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FOREWORD

The Clima 2007 congress is an official congress of Federation of European Heating and Air-conditioning societies (REHVA). The Clima Congress series was established to transfer research results into practice in the area energy efficient building services and to increase the exchange of information between countries with different experience in building services engineering. The first Clima congress was held in Milan (1977), then in Copenhagen (1981), Sarajevo (1985), London (1989), Budapest (1993), Brussels (1997), Naples (2001), and Lausanne (2005).

The 9th Congress in Helsinki held in Finland June 2007 focused particularly on how to design, build and maintain healthy and productive indoor environment with low use of energy. The title of the congress “WellBeing indoors” reflects this goal. The Clima 2007 Congress was organised in co-operation of four organisations: Finnish Association of HVAC Societies (FINVAC), the official Finnish member of REHVA, Helsinki University of Technology (TKK), Finnish Association of Mechanical Building Services Industries (FAMBSI), and Finnish Society of Indoor Air and Quality (FiSIAQ).

The invited key-note lectures and the one-page summaries of the over 460 papers presented in the Clima 2007 congress are published in the Abstract book. The full papers are published in the Proceedings CD, organized in 30 sessions under the following themes:

Theme A: Healthy and productive indoor environment
Theme B: Sustainable energy use of buildings
Theme C: Intelligent design and management of buildings
Theme D: Energy efficient heating and cooling systems

The main objective of the congress was to share the knowledge between scientists, practitioners and manufactures. A significant outcome of the congress are the reports from the 20 workshops organised during the congress. The results of the workshops will be available at REHVA website (*www.rehva.eu*). The summaries of the workshops will also be published as a booklet. After one year of the Congress the papers will be also available in a AIRBASE bibliography database at *www.aivc.org*.

Many people have worked hard to make the Clima 2007 congress successful. We wish to thank all of you for the effort. We are also grateful for the congress partners, sponsors and supporters of the congress whose support made the congress possible. We wish to present our thanks also to the scientific committee that reviewed the papers and helped in organising the sessions and proceedings. For technical assistance in compiling the proceedings we thank Mr Jarkko Narvanne.

Espoo, May 2007

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EUROHEAT & POWER	International Association for District Heating, District Cooling and Combined Heat and Power
EUROVENT	European Committee of Air Handling and Refrigeration Equipment Manufacturers
ISIAQ	International Society of Indoor Air Quality and Climate
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Providing Better Indoor Environmental Quality Brings Economic Benefits

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SUMMARY

This paper summarizes the current scientific evidence that improved indoor environmental quality can improve work performance and health. The review indicates that work and school work performance is affected by indoor temperature and ventilation rate. Pollutant source removal can sometimes improve work performance. Based on formal statistical analyses of existing research results, quantitative relationships are provided for the linkages of work performance with indoor temperature and outdoor air ventilation rate. The review also indicates that improved health and related financial savings are obtainable from reduced indoor tobacco smoking, prevention and remediation of building dampness, and increased ventilation. Example cost-benefit analyses indicate that many measures to improve indoor temperature control and increase ventilation rates will be highly cost effective, with benefit-cost ratios as high as 80 and annual economic benefits as high as $700 per person.

INTRODUCTION

Recently completed research, reviewed in this paper, indicates that there is a large untapped opportunity for economic benefits resulting from improvements in indoor environmental quality (IEQ) in non-industrial work places and homes. The most clearly established sources of economic benefits include improved work performance, e.g., work speed or quality, reduced absence, and reduced health care costs. There is also evidence that providing better IEQ can improve student learning which, in turn, should lead to more effective future workforces. At the societal level, economic value can also be assigned to the reduced suffering of ill health and to extended average lifetimes expected when IEQ is improved.

This paper will summarize the current scientific evidence that improved IEQ can improve work performance and health, building upon the conceptual framework of Seppanen et al. [1]. The paper will address only the thermal and air-quality-related aspects of IEQ. When possible, the relationship between IEQ parameters and health or performance outcomes will be expressed in quantitative terms that provide a basis for cost-benefit analyses applicable to building design and operation [2]. Calculations for hypothetical case studies will contrast the expected economic benefits with the required investment costs. The paper will also argue for using the most energy efficient options possible to bring about the improvements in IEQ.

METHODS

This paper is based on a review and analysis of the published scientific literature addressing the linkages of IEQ with health and work performance. The original scientific research employed a variety of study designs. Some of the research employed cross-sectional multi-building surveys of IEQ conditions and health, absence, or work performance outcomes.

These studies used statistical models to analyze the resulting data and quantify the effect of specific factors, such as ventilation rates, on outcomes (e.g., absence rates), controlled for other factors that also influence the outcome (e.g., student age and socioeconomic status). Other research has experimentally modified IEQ factors (e.g., temperature or ventilation rate) in real buildings and measured the resulting health or performance changes. Several of these studies were performed in call centers where workers interact with the public via telephone and use computers to obtain, input, and process information. Speed of work, such as average time to complete a call and associated information processing, was used as an indicator of work performance. Numerous experimental studies have also been performed in laboratories. With few exceptions, the laboratory spaces have been very similar to real workspaces, but the laboratory setting enabled more precise control of experimental variables. Normally the subjects performed simulated work tasks, such as addition, typing, and proof reading, with the speed and accuracy of task performance measured under different IEQ conditions.

In other papers reviewed, various approaches have been used estimate economic costs of health effects, so that the economic value of improved health from better IEQ can be estimated. In one approach, the total national annual cost of a health outcome, e.g., total annual cost of asthma in a country, was multiplied by the estimated fraction of ill health attributable to an IEQ risk factor, e.g., to moldy buildings. In another approach, each health outcome was assigned a unit cost per incident (e.g., cost per hospitalization for asthma) and this unit cost was multiplied by the estimated percentage decrease in the health incidents resulting from improved IEQ. One of two different types of unit costs are utilized. “Cost of illness” (COI) unit costs reflect the cost of health care and often also include the cost of lost work. “Willingness to pay”(WTP) unit costs are estimates of how much individuals are willing to pay to avoid a health effect, often based on surveys and analyses of consumer spending for safer products. WTP values, thus, assign an economic value for avoidance of pain or suffering or premature death from ill health and are often much larger than COI unit costs. COI and WTP values are available from various sources including the U. S. Environmental Protection Agency [3].

IMPACTS OF IEQ ON WORK AND SCHOOL PERFORMANCE

Temperature and office work performance

The influence of indoor air temperatures on objective (measured) work performance relevant to offices has been assessed experimentally, primarily in call centers and laboratory settings representative of real offices. A few additional studies used vigilance tests to measure ability to concentrate. Most of the studies experimentally manipulated temperatures while holding other factors constant to investigate the influence of temperature on performance, although one call center studied relied on natural changes in air temperature. Recently, a formal statistical meta-analysis of 24 of these studies was completed [4] to assess the average relationship between temperature and performance of work. The authors analyzed primarily office studies and laboratory studies that simulated office work, although three of 24 studies were performed in classrooms. Their analyses are the source of Figure 1 illustrating a best estimate of how office work performance varies with temperature. The graph in Figure 1 shows that performance increases with temperature up to 21 to 22 ^{o}C, and that performance decreases with temperature above 22 or 23 ^{o}C. The maximum performance, i.e., relative performance equal to unity, occurs at temperature of 21.8 ^{o}C. The equation for the curve in Figure 1 is

$$RP = 0.1647524 \cdot T_C - 0.0058274 \cdot T_C^{\ 2} + 0.0000623 \cdot T_C^{\ 3} - 0.4685328 \qquad (1)$$

where RP is relative performance, i.e., performance relative to maximum value, and T_C is room temperature, ^oC.

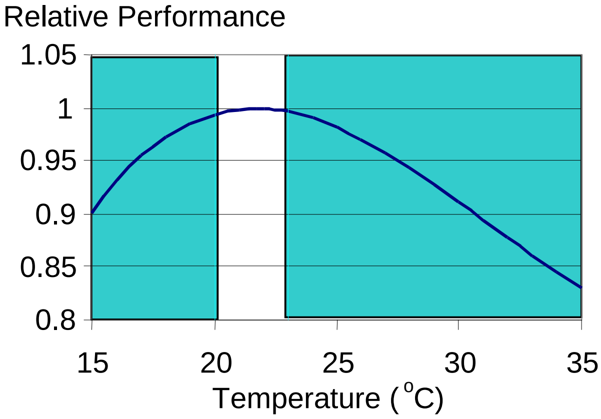

Figure 1. Relationship between office work performance and indoor temperature based on a statistical analysis of reported data. The shaded areas in the figure represents the regions where a statistical analyses indicates that decrements in performance have less than a 10% probability of being the result of chance.

Temperature and school work performance

Several studies conducted in the 1950s and 1960s found that students performed better in thermally conditioned classrooms than in classrooms without heating or cooling [5]. However, there have been few studies of the influence of temperature in thermally-conditioned classrooms on school work performance or learning. In the late 1960s, six groups, each with six students, were brought to a climate-controlled chamber at a U.S. university [5]. Each group of students performed simulated school work with chamber temperatures ranging from 17 to 33 ^oC. Error rates and speed of work were used as performance indicators. Two out of four performance measures, error rates and time required to complete assignments, were affected by temperature. The error rate was highest at 17 ^oC and lowest, about 20% lower, at 27 ^oC; however, students worked most slowly at 27 ^oC and fastest, about 10% faster, at 17 ^oC. Several similar studies were also performed in the 1960s [6]. Some of these studies performed in climate chambers and other studies in actual classrooms found reading speed, reading comprehension, and multiplication performance of school children to be poorer with temperatures of 27 to 30 ^oC, relative to 20 ^oC. In one case, the performance decrement was as large as 30%.

The influence of more moderately elevated temperatures on student performance was investigated recently via field studies conducted in classrooms [7, 8]. Classroom temperatures were manipulated by turning cooling systems on and off, while keeping all other factors constant to the degree possible, although, teachers opened windows "slightly more often when it was warm in the classroom". Performance tasks representing eight aspects of schoolwork, from reading to mathematics, were embedded into the normal school work. The speed and accuracy of task performance was assessed. The average speed of eight simulated school work tasks was increased by approximately 2% per 1 ^oC as temperatures decreased from 25 ^oC to 20 ^oC, although from visual inspection of the data there appeared to be no significant change in performance as temperature decreased from 23 to 20 ^oC. The number of errors in school work was not significantly affected by temperature changes in this temperature range. Figure 2 provides more detailed results from this study.

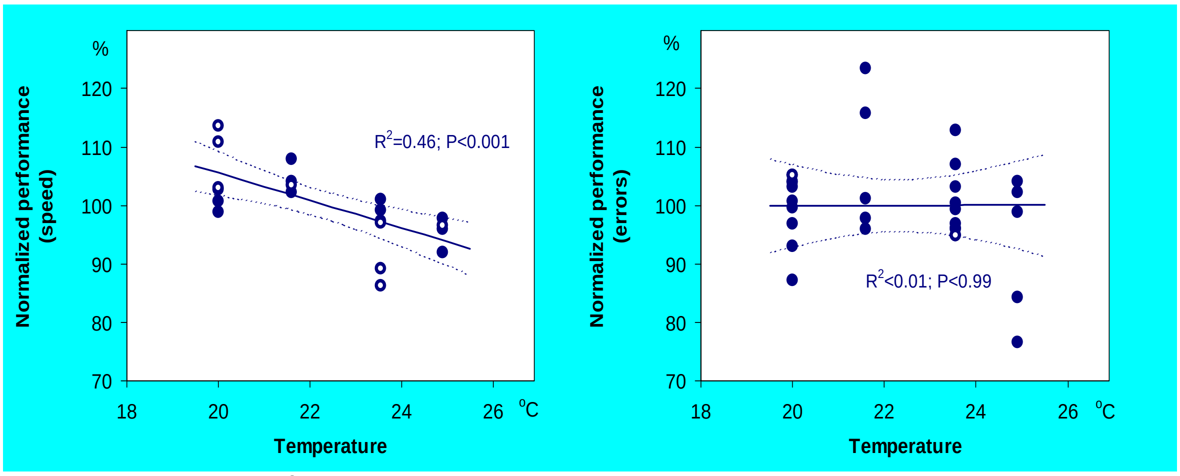

Figure 2. Student performance versus temperature based on study in Denmark [8]. Performance was based on the speed (left figure) and accuracy (right figure) of completing various school work tasks. Open dots represent tasks for which the performance increases with ventilation rate were statistically significant. [Figure 2 reproduced with permission.]

Ventilation rates and office work performance

The influence of ventilation rates (i.e., rates of outdoor air supply) on office work performance has been assessed experimentally in call centers and laboratory settings using the same methods described previously to assess how temperature affects work performance. Seppanen et al.[9] performed a statistical meta-analysis of eight studies of how ventilation rate affects office work, plus one study in schools of how ventilation rate affected concentration and vigilance, to assess the average relationship between ventilation rate and performance of work. Their analyses are the source of Figure 3 illustrating a best estimate of how office work performance varies with ventilation rate.

The curve in figure 4 is well fit by the equation

$$RP = (5.56 \text{ x } 10\text{-}8)\ V^3 - (1.48 \text{ x } 10\text{-}5)\ V^2 + (1.49 \text{ x } 10^{-3})\ V + 0.983 \qquad (2)$$

where *RP* is the relative performance and *V* is the ventilation rate in L/s per person.

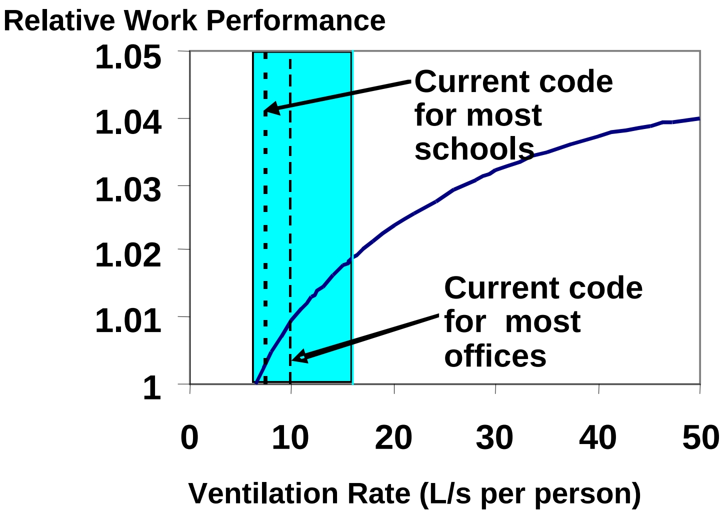

Figure 3. Performance of office work at various ventilation rates relative to performance with a ventilation rate of 6.5 L/s per person based on the meta analysis by Seppanen et al. [9]. In the shaded region, the trend of increased performance with ventilation rate has a 10% or smaller probability of being the result of chance.

Ventilation rates and school work performance

Four studies in schools have investigated the linkage of ventilation rates to objectively measured, as opposed to self-reported, school work performance. A Norwegian study [10] performed in 35 classrooms used reaction times to measure student concentration and vigilance. Reaction times were 5.4% less (i.e., faster) with a ventilation rate of 8.1 ach (12 L/s per person) compared to 2.6 ach (4 L/s per person). A U.S. study [11] in 5th grade classrooms from 54 schools, used student performance in standard academic tests as the measure of performance. Performance in both math and reading tests increased with ventilation rate. Test scores increased about 13% from classrooms with the lowest ventilation rates (less than 2.2 L/s per student) to classrooms with the highest ventilation rates (greater than 4.5 L/s per student); however, statistical analyses indicated greater than a 10% probability that the increases in performance were due to chance. In a Danish study in six classrooms [8], Wargocki and Wyon used performance tasks representing various aspects of schoolwork, from reading to mathematics that were embedded into the normal school work. The speed and accuracy of task performance were assessed. This study reported an 8% increase in speed of school-work tasks with a doubling of ventilation rate. There was no statistically significant influence of ventilation rate on the number of errors made by students. Figure 4 provides more detailed results from this study.

In Japan, college student performance in classroom settings and in a laboratory setting on standardized tests was evaluated at different ventilation rates [12, 13]. In three tests implemented in the field study, one on theory and two involving memorization, performance improved 5.4%, 8.7 %, and 5.8%, respectively, with increases in ventilation rate from 0.4 to 3.5 ach. The laboratory study included only tests of memorization performance, and had results similar to those from the field study. However, in these studies, the ventilation rates

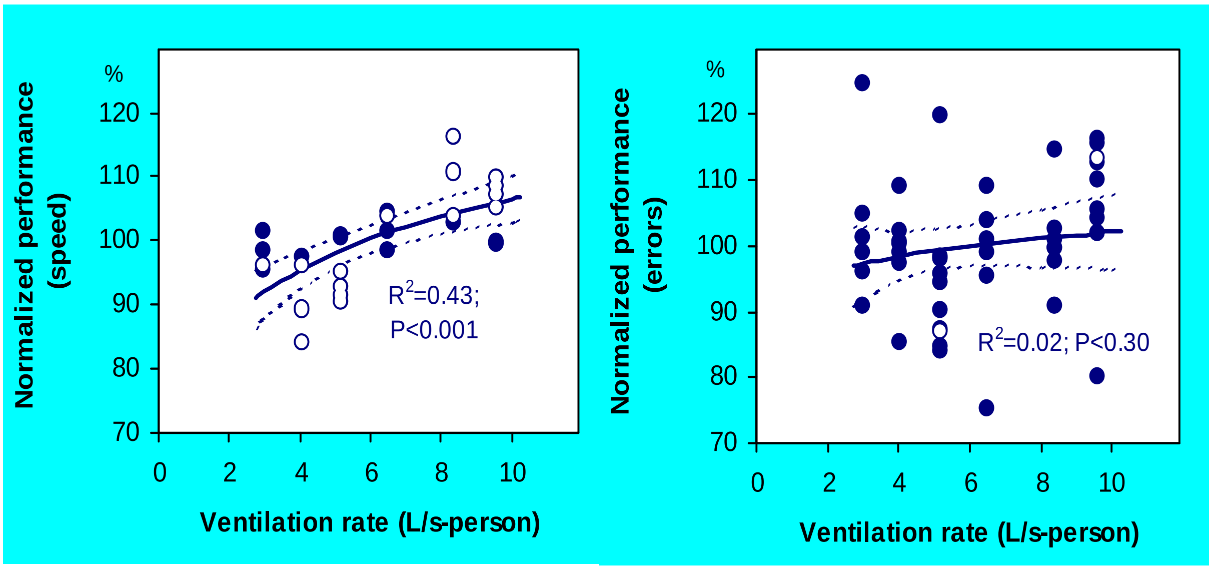

Figure 4. Student performance versus ventilation rate based on study in Denmark [8]. Open dots represent tasks for which the performance increases with ventilation rate were statistically significant. [Figure 4 reproduced with permission.]

per person at the lower air exchange rates were very low even for classrooms, e.g., less than 1 L/s per person. In addition, this study intentionally did not disentangle the effects of ventilation and temperature. Temperatures were higher by approximately 2 °C in the low-ventilation conditions, as they would be in building cooled by the outdoor air supply. Higher temperatures, in the temperature ranges encountered in this study, have been shown to reduce work performance [14]. Thus, these two studies do not provide information about how ventilation rates affect student performance when temperature is maintained constant.

Higher classroom ventilation rates have also been linked to a reduction in student absence [15], which, in turn, may improve student learning.

In summary, while the applicable research is limited, the available scientific literature indicates the potential for 5% to 10% increases in aspects of student performance with increased classroom ventilation rates. Ventilation rates in roughly half of U.S. public elementary school classrooms appear to be less than specified in codes [11, 15, 16]; thus, the opportunities for increasing student performance by increasing ventilation rates, at least in U.S. schools, may be substantial.

Indoor pollutant sources and performance

The improvements in work and school work performance with higher ventilation rates is presumed to be a consequence of the reduction in indoor concentrations of indoor-generated air pollutants with more ventilation. If this explanation is correct, we would expect performance to also increase if indoor pollutant concentrations were reduced by reducing the emissions from indoor pollutant sources, e.g., by removing indoor pollutant sources. We have limited research that directly assesses the effects of source reduction measures on performance, but the results of this research are at least qualitatively consistent with the finding that increased ventilation rates improve work performance. Most of the relevant studies were performed in a laboratory setting representative of real offices [17-19]. In these studies, subjects performed tasks representative of office work, such as proof reading of text, text typing, and simple arithmetic operations and speed and accuracy in these simulated work tasks was measured and used to indicate work performance. In three studies [17, 18], a section of carpet removed from a complaint building was placed in the laboratory, but hidden from the study subjects. Performance, based on typing, addition, and proof reading tests, was improved by approximately 4% by removing the carpet, as depicted in Figure 5.

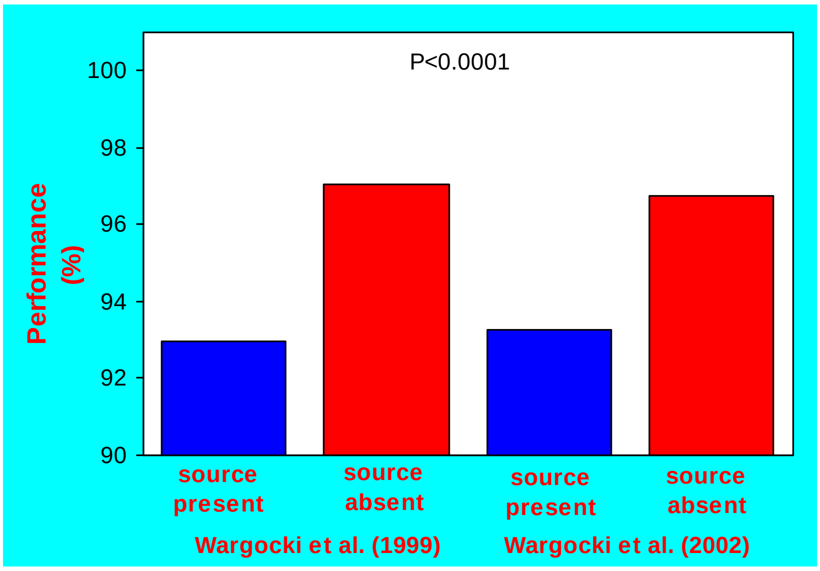

Figure 5. Controlled laboratory studies performed in Denmark show that performance, based on typing, addition, and proof reading tests, improved when a carpet taken from a complaint building was removed. [Figure 5 reproduced with permission.]

Bako-Biro [19] performed similar laboratory studies, but with three-month old personal computers with CRT monitors as the pollutant source. When the computers were removed, performance in proof reading and arithmetic calculations were unaffected; however, text typing errors diminished by 16% and typing speed improved slightly. The improvements in typing performance were statistically significant. Additional research by Bako-Biro [20] employed three-year-old linoleum flooring, shelves with books and paper, and three-month-old caulk sealant as the pollutant sources. Speed and accuracy in arithmetic calculations were used to assess work performance. In this study, few and only small statistically significant impacts of the pollutant sources on performance were identified, although there is some evidence of improved text typing when the pollutant sources were removed.

One additional study of the effects of pollutant source removal on work performance was performed in a call center with the time required to talk with customers used as an measure of work performance [21]. The pollutant source was a 6-month old particle filter in the building's ventilation system. When the used filter was replaced with a new filter, at times unknown to the call center workers, average talk time decreased by approximately 10%.

Perceived Indoor Air Quality and Work Performance

Peoples' perceptions of indoor air quality, as reported on questionnaires, have often been used as subjective indicators of the quality of indoor air. The questionnaires normally ask subjects to rate air quality on a scale ranging from clearly acceptable to clearly unacceptable. Research, performed primarily in the laboratory [18-20, 22-24] , has found that improved perceptions of, or satisfaction with, indoor air quality are associated with improvements in some aspects in work performance. In this research, performance of some tasks has increased by approximately 1% for each 10% reduction in the percentage of the occupants dissatisfied with indoor air quality. However, it is not known if an increased satisfaction with indoor air actually causes people to work better. It could be that the indoor environmental exposures, e.g., pollutants, directly cause both poorer work performance and dissatisfaction with indoor air, leading to a correlation of performance with perceived air quality.

HEALTH-RELATED ECONOMIC BENEFITS OF IMPROVED IEQ

Eliminating Environmental Tobacco Smoke

Environmental tobacco smoke (ETS) is the tobacco smoke to which non-smokers are exposed. Most ETS exposures occur indoors. Based on reviews of a very large body of research, various organizations or panels of experts have concluded that ETS is the cause of an increased number of several health outcomes including premature death from lung cancer and cardiovascular disease, and cases of asthma exacerbation. A critical review by a California health organization provides estimates [25] of annual ETS-caused health effects in the U.S.. Fisk [26] used these data and unit costs for health outcomes to estimate the preventable cost of ETS exposures, assuming that ETS exposures are 100% preventable. The results are shown in Table 1.

Table 1. Estimated annual health effects and preventable costs ($U.S.) in the U.S. of indoor ETS exposures in 2000.

Heath Effect	**Annual U.S. Cases (1000s)**	**Unit Cost ($ U.S.) [cost type^]**	**Total Cost in U.S. (billions of $U.S.)**	**Cost per U.S. smoker# ($U.S.)**
Death	40.5 – 68.8	6300K [WTP] (2100K–10600K)	$260 - $430* $85 - $730+	$4000 - $6600* $1300 - $11000+
Asthma induction	8.1 - 26	$33K [WTP]	$0.27 - $0.86*	$4 - $13
Day of asthma exacerbation	400 – 1000	$42 [WTP]	$0.017 - $0.042*	$ 0.26 - $0.65
Bronchitis or pneumonia hospitalizations	7.6 – 15.2	$11K [COI]	$0.084 - $0.170*	$1.30 -$2.63
Acute Bronchitis or Pneumonia	150 – 300	$59 [WTP]	$0.009 - $0.018*	$0.13 – $0.28
All morbidity			$0.38 – $1.09	$5.87 - $16.80

#based on 64.7 million smokers in the U.S. in 2000 [27] *range in costs reflects only the range in number of health effects +range in costs reflects the range in number of health effects and the range in the estimated unit costs per health effect, when available ^ WTP = willingness to pay, COI = cost of illness

From the numbers in this table, it is evident that the health-related costs of ETS are still very large in the U.S., despite considerable progress in reducing smoking rates. On average, each smoker is responsible for 3100 to 8500 € ($4000 to $11000) of cost per year from premature death in non-smokers and for 5 to 12 € ($6 to $16) in annual health costs for morbidity. The estimated health cost imposed per smoker may be roughly similar in other regions of the world with similar living standards.

Dampness and Mold

Many studies have found that dampness or mold is common in homes and is associated with increases in asthma exacerbation and various respiratory symptoms [28]. To estimate the cost of asthma from indoor dampness and mold (C_{DM}), Fisk [26] used the following equation

$$\boldsymbol{C_{DM}} = \text{(attributable fraction of asthma) X (total cost of asthma)} \qquad (2)$$

The same basic approach was used by Nguyen et al. [29] for the asthma attributable to dampness or mold in Finland. The fraction of asthma symptoms attributable to dampness and mold was estimated using a standard equation for calculating the attributable fraction (AF)

$$AF = [P(RR-1)]/[P(RR-1)+1] \qquad (3)$$

where P is the prevalence of the risk factor (e.g., household mold contamination) and RR is the relative risk, which indicates the increased prevalence of the health effect in the population with the risk factor. Using the results of four large studies, three from North America and one from Europe, approximately 13% of asthma symptoms were attributable to asthma and mold in housing [26]. The total annual cost of asthma in the U.S. in year 2000, was estimated to be between 5.8 to 10.7 billion € ($7.5 to $13.9 billion U.S.) [26]. Thus, the annual cost of asthma in the year 2000 attributable to dampness and mold in housing is 13% of this range or approximately 0.8 to 1.4 billion € ($1 to $1.8 billion U.S.). This equals 19 to 35 € ($25 to $45 U.S.) of annual asthma-related costs for every home with dampness and mold, given that the U.S. had 121 million housing units [27] in 2000 and roughly one third of homes have visible dampness or mold [26]. This 19 to 35 € ($25 to $45 U.S.) estimate may be a reasonable initial estimate of the annual asthma-related costs for damp homes in other countries with similar living standards. Based on the prior analyses of the asthma health cost of dampness and mold in Finland in 1996 [29], one can estimate that the cost was 18 € ($29) per home or apartment in need of repair due to moisture. The costs of other health effects of dampness in mold have not been estimated but may be substantial. In addition, dampness and mold in workplaces and schools is common and is expected to impose additional health costs.

At present, we do not have a defensible estimate the portion of dampness and resultant indoor mold that is preventable. Many dampness problems, possibly a majority, result from water leaks that could be prevented through better building maintenance and improved design and construction. Better ventilation in winter and use of dehumidifiers in summer could reduce dampness problems that result from high indoor humidity. Thus, it is technically feasible to eliminate a large majority of the dampness problems. However, poverty, insufficient training of homebuilders, lack of public awareness, and resistance to behavioral changes remain substantial obstacles.

Communicable Respiratory Illnesses and Sick Leave

The limited data available suggest that occupants of buildings or spaces with higher ventilation rates on average have fewer communicable respiratory illnesses, e.g., common colds and influenza, and less absence from work or school. These data are from buildings in which multiple occupants share the same airspace, i.e., air is transported from person to person. Much of the research has been performed in buildings with a high occupant density such as jails or military barracks. Three studies have investigated linkages between ventilation rates and absentee or sick-leave rates in more typical buildings; two were performed in offices [30, 31] and one in elementary grade classrooms [15]. One office study [31] found no association of sick leave with building carbon dioxide (CO_2) concentrations, as indicators of ventilation rates; however, this study included only two buildings and had experimental periods that did not integrate over the yearly cycle of respiratory disease. The two much larger studies, one of offices [30] and the second of classrooms [15], assessed absence over full-year periods. In the office study, short term sick leave was 25% less in buildings with 24 L/s-person compared to 12 L/s-person of outdoor air. In the classroom study, a CO_2 concentration decrease corresponding to a 10 L/s per person increase in ventilation rate was associated with a 9% to 17% decrease in total student absence.

From the office building study, one can estimate the economic benefit of the 12 L/s-person increase in ventilation rate. In the office workers, the total average sick leave rate was 1.71% or 4.3 days assuming 250 work days per year. The 25% lower sick leave in the buildings with 12 L/s-person more ventilation corresponds to an average annual reduction in absence of 1.1 days per person. If the annual salary plus benefits is 77,000 € ($100,000 U.S.), the reduction in sick leave is worth $338 € ($440 U.S) per worker.

COST EFFECTIVENESS OF IMPROVEMENTS IN IEQ

To inform decisions about investments in building design, retrofit, or operation, it is common to employ cost-benefit analyses that account for initial equipment costs, energy costs, maintenance costs, and taxes. The economic value of changes in work performance, absence, and health are normally neglected. However, using the information provided above, it is now possible to account for some of these changes in human outcomes. Although uncertainty about the magnitudes of changes in performance, absence, and health remain high, application of best available estimates should lead to better decisions than the current practice of neglecting these factors. This section provides some examples.

Night Time Ventilative Cooling

In buildings without mechanical cooling, work-time air temperatures can be diminished by using fans (or by opening windows) to ventilate the building during the cooler night-time periods. The night-time ventilation reduces the temperature of thermal mass in the building and, by absorbing heat during the subsequent workday, this cooled thermal mass reduces workday temperatures. Kolokotroni et al [32] report temperatures in a Finnish office building with and without ventilative cooling. We previously used these temperatures with a very simple temperature-performance relationship [33], to compare the value of predicted work performance increases with the costs of running the fans to ventilate the building at night. We assumed that 2.5 kW of fan power was required per each 1 m^3/s of ventilation, a 4 air change per hour ventilation flow rate, 83 m^3 of indoor air volume per occupant, 8 hours of fan operation, and that salary, benefits, and overhead cost the employer 25 € per hour per worker. We now provide an updated analysis using the temperature-performance relation in equation 1. For each hour of the workday, the percent decrease in work performance and equivalent lost working time were calculated using the indoor temperatures with and without night-time ventilative cooling. Table 2 shows the calculation of lost working time. It was estimated that ventilative cooling reduces the lost work time from 26.1 to 8.8 minutes per day, which has a value of 7.2 € per worker per day. If the electricity to operate fans cost 0.10 € per kWh, the daily energy cost is 0.18 € per person. The benefit-cost ratio is then 7.2/0.18 or 40. With electricity costs of 0.05 and 0.15 € per kWh, the benefit-cost ratios are 80 and 27, respectively.

Table. 2. Example calculation of work performance gains from night time ventilative cooling in an office building without air conditioning.

Hour	No Ventilative Cooling			With Ventilative Cooling		
	T ^oC*	RP	Lost Min	T ^oC*	RP	Lost Min
8 - 9	27.0	0.954	2.73	23.4	0.991	0.55
9 - 10	27.2	0.952	2.90	23.5	0.990	0.58
10 - 11	27.6	0.946	3.24	23.9	0.988	0.74
11 - 12	27.8	0.943	3.42	24.2	0.985	0.88
13 - 14	27.8	0.943	3.42	25.2	0.976	1.42
14 - 15	27.8	0.943	3.42	25.4	0.974	1.55
15 - 16	27.9	0.941	3.51	25.4	0.974	1.55
16 - 17	27.8	0.943	3.42	25.4	0.974	1.55
Total			26.08			8.83

*Estimated operative temperature as average of air and slab temperatures
Lost time each hour = (1 – RP from equation 1 at T*) x 60 minutes

Comparison of Building Cooling Strategies

In another set of example cost-benefit analyses [34], the following options for cooling an office building located in Helsinki were compared: a) base case with no mechanical cooling and a 2 L/s-m^2 supply air flow 10 h per day; b) mechanical cooling with capacity of 20 W/m^2, c) base case but with the ventilation system operation increased from 10 to 24 h per day; d) base case but with the ventilation system operation increased from 10 to 24 h per day and supply flow rate increased to 4 L/s per m^2; e) base case plus all measures in b, c, and d. The assumed value of an hour of work was 32.3 € and the cost of heat and electricity were 0.04 €/kWh and 0.1 €/kWh, respectively. Other assumptions are provided in reference [34]. The key results of this analysis are provided in Table 3.

Table 3. Comparison of life cycles costs of cooling an office building in Helsinki [34]

Factor	Base Case	Mechanical Cooling	Increased Operation Time (No Mech. Cooling)	Increased Outdoor Air Flow (No Mech. Cooling)	All Meas -ures
Increased annual energy cost per person, €	--	1.6	6.3	40.0	39.9
Increased first cost, € per person	--	9.5	0	10.0	19.5
Effective lost work hours per person-year, h	21.2	15.5	12.6	6.5	4.4
Value of lost work hours per person-year, €	686	501	408	211	141
Value of improved work, €	--	184	278	475	545
Total annual savings per person, €	--	131	272	380	398

This example analysis projects annual savings of 131 € to 398 € per worker from implementing the various measures for space cooling. In the Helsinki climate, with relatively few days where cooling is needed, the mechanical cooling option consumes less energy than increasing ventilation time or flow rate.

Addition of an Outdoor Air Economizer

In an air-conditioned building, an economizer system increases the outdoor air supply above the minimum amount required by codes whenever doing so eliminates the need for energy-intensive mechanical cooling. Normally, economizers are used in buildings with air recirculation by reducing the amount of recirculation and increasing the outdoor air supply. Economizer systems save energy. In many climates, economizers also dramatically increase time-average ventilation rates. Economizer systems are common in large U.S. HVAC systems with air conditioning, but are often considered too costly for small HVAC systems.

A prior paper [35] evaluated the energy savings and projected financial benefits from reduced sick leave when an economizer system was used in an office building located in Washington, DC. A building energy simulation model predicted hourly ventilation rates and energy consumption with and without an economizer system. A model of airborne disease transmission calibrated with empirical data on the relationship of ventilation rates with respiratory disease and absence was employed to estimate sick leave rates with and without the economizer. The doubling of the work-time average ventilation rate, from 10 to 20 L/s per person, when an economizer was used saved about 23 € ($30) in energy costs per person per year and reduced annual sick leave by an estimated 0.9 to 1.2 days per person. With each day of sick leave valued at 154 € ($200 U.S), corresponding to an annual cost of salary plus benefits of 38,500 € ($50,000 U.S.), the annual value of the reduced sick leave is approximately 160 € ($210 U.S.) per person.

One can instead easily use the ventilation rates in this paper and estimate the work performance benefits from increased ventilation via Figure 4 or equation 2. The annual value of the projected 1% increase in work performance is $380 € ($500 U.S.)

Table 4 summarizes the results of these two analyses. Decisions about whether or not to use an economizer would normally be based on a comparison of the economizer system costs with the 23 € ($30) per person projected annual energy savings. However, when best estimates of the impacts of ventilation rate on absence and work performance are considered, the economizer saves roughly 570 € ($740 U.S.) per person per year.

Table 4. Example analyses of financial benefits of an economizer system in an office building located in Washington, DC.

Increase in annual average ventilation rate	10 L/s-person
Normally considered benefit Annual Energy cost savings	 $30 U.S. (23 €) per person
Normally neglected benefits Annual value of reduced absence Annual value of productivity increase	 $210 U.S. (160 €) per person $500 U.S. (380 €)

Increased Ventilation Rates in an Office Building

Another example analysis [34] compared the energy and equipment costs of providing a higher ventilation rate in an office building with the value of work performance increases estimated from Figure 4 or Equation 2. The office building had a typical design for a medium-size northern-European, mechanically-cooled building with supply and exhaust air flows, no recirculation, and air-to-air energy recovery with a 75% temperature efficiency. Energy consumption and energy costs were estimated with a simulation program, Helsinki climate data, and energy costs of 0.04 € per kWh for heat and 0.1 € per kWh for electricity. The annual value of work per employee was 30,000 C. Table 5 summarizes the main results from this analysis. Increasing the ventilation rate results in a 4.4 to 22 € increase in annual energy cost per employee, a similar magnitude increase in maintenance costs, a 22 to 63 € annualized increase in first costs, performance increases of 1% to 2.4 % valued at 300 to 690 € per person per year. The estimated benefit-cost ratio ranges from 6.2 to 9.4. The largest benefit-cost ratio was predicted for increasing ventilation rates from 6.5 to 10 L/s per person, but further economic benefits are predicted from increasing the ventilation rate from 10 to 20 L/s per person. This example analysis neglected the value of reduced health effects and absence from increased ventilation.

Table 5. Predicted costs and benefits from increasing ventilation in an office building [34].

Change in Ventilation Rate (L/s-person)	Annual Cost Increase (€ per person)			Increase in Performance		Benefit-Cost Ratio
	Energy Consumption	Maintenance	Equipment (annualized)	%	€ per person	
6.5 to 10	4.4	4.7	22.5	1.0	300	9.4
6.5 to 20	22.5	13.3	63	2.3	690	7.0
10 to 20	18.1	8.5	41	1.4	420	6.2

DISCUSSION

Main findings

Based on the information presented above, we have fairly compelling evidence that indoor environmental factors and related building operating conditions affect health and work performance. Higher ventilation rates are associated with improved health and better work and school performance. Removal of pollutant sources is associated with improved work performance, at least in some cases. Indoor temperatures in the 21 to 22 ^{o}C range are associated with maximum overall work and school performance. We now have estimates of quantitative relationships between some of these indoor environmental factors and health or performance outcomes. These quantitative relationships enable an accounting of some health and productivity benefits in cost-benefit calculations pertaining to building design and operation. Example cost-benefit analyses indicate that many measures to improve indoor temperature control and increase ventilation rates will be highly cost effective, with benefit-cost ratios as high as 80 and annual economic benefits as high as 500€ ($700) per person.

Implications

The research findings and cost-benefit analyses summarized in this paper suggest that more effort and investment be devoted to improvement of IEQ, which will often lead to more productive and healthy workers and financial benefits that greatly exceed costs. Designers and those who construct buildings must be educated about the technologies and practices needed to enable good IEQ. To assure that good IEQ is maintained over the life of buildings, periodic or continuous commissioning and building and HVAC system maintenance will be necessary. It would be helpful to develop and utilize lease and maintenance contracts that incentivize building owners and maintenance providers to maintain good IEQ. In addition, building operators and maintenance staff should be educated about means of maintaining good IEQ.

Limitations

There remain substantial uncertainties in the quantitative relationships between indoor environmental factors and health and work performance outcomes. The limited body of related research is one source of uncertainty. In addition, much of the research completed to date uses the speed and accuracy of simulated work tasks as the work performance outcome. While such outcomes are clearly relevant to real work, the quantitative relationship between average performance of these simulated work tasks and average performance of overall real-world work remains uncertain. In particular, speed and accuracy of simulated work tasks, such as addition, typing, and proof reading, might not be a good predictor of higher-level cognitive functioning such as the decision making of many managers and executives. Another limitation of the currently available quantitative relationships is that they are not tailored for specific building conditions, types of work, or types of workers. Rather, the existing quantitative relations presented in this paper are estimates of the average relationships for the average building, type of work, and worker. In specific situations, larger or smaller improvements in health and performance may be expected. For example, an increase in ventilation rate would seem more likely to substantially improve health or work performance in a building with initially poor IEQ due to the presence of strong indoor pollutant sources. Despite the substantial uncertainties, application of the best information available today should lead to better decisions about building design and operational practices than the current practice of totally neglecting impacts on health and work performance.

Who Benefits and Who Pays?

Users of quantitative relationships between IEQ factors and health and productivity must recognize that costs and benefits may affect different parties. For example, in a leased building, the building owner usually pays for building or HVAC improvements while the employer benefits most directly from improved work performance and reduced absence. However, the owner of a leased building may indirectly benefit the improved worker health and work performance through increased rent, higher property values, and longer tenant retention when IEQ is improved and the building has a more positive reputation. The employee and health care insurer may directly benefit from reduced health care costs, but often the associated financial benefits will not be passed on to the employer. In an owner-occupied building, the incentive for investments to improve IEQ may be greatest because the owner-employer benefits directly from both improved work performance and reduced absence.

Energy use concerns

Because increased ventilation rate is associated with improved work performance and better health, and often increases energy use, there is a conflict between energy efficiency and health/productivity goals. Given the broad concerns about global climate change resulting from use of fossil fuels for energy, there is a strong need for technologies and practices that bring about the productivity benefits without increasing energy use, and ideally with simultaneous energy savings. The economizer system, described above, is an example of such a technology. In the longer term, we should be able to identify and utilize pollutant source control measures and air cleaning technologies to bring about the same health and productivity benefits as increased ventilation, but with no or minimal increases in energy use.

Priority research needs

While there are many research needs related to the subjects discussed in this paper, the following are suggested as high priority research needs:

- the influence of IEQ factors on high-level cognitive performance such as decision making;
- the influence of IEQ factors on quality and speed of real work, as opposed to simulated work tasks;
- the relationship between performance of school work tasks and student learning;
- the relationship between worker health, which has already been shown to be associated with many IEQ factors, and work performance;
- means of obtaining the health and productivity benefits of increased ventilation with minimal increase in energy use, or ideally with energy savings.

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Effect of outdoor generated pollutants on indoor air quality and health

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SUMMARY:
The significant public health effects of ambient air pollution are mostly caused by exposures in indoor environments, where we spend over 90 % of our time. Indoor concentrations of air pollutants are in general higher than and often also independent of the outdoor air concentrations. In average, urban outdoor air contributes significantly to indoor concentrations of carbon monoxide (CO), nitrogen dioxide (NO_2), benzene and some other aromatic VOC:s, ozone (O_3), and fine particulate matter ($PM_{2.5}$), but not to, e.g., most VOCs and carbonyls or to the highest concentrations of CO and PM. Indoor concentrations of pollutants of outdoor or ambient origin, however are at most equal to, (e.g. CO) and for reactive (e.g. O_3 and ultrafine particles) significantly lower than the outdoor air levels. $PM_{2.5}$ is the urban air pollution which causes the biggest mortality, over 300 000 excess deaths annually in Europe. To the vulnerable population groups the current building envelope, ventilation and air conditioning technologies can provide effective and targeted protection against the risks of outdoor air fine PM and O_3.

INTRODUCTION

Air pollution epidemiologists have - with remarkable success - studied the short term associations of daily mortality and morbidity of urban populations with respective daily ambient air (outdoor air) pollution monitoring data [1,2] and the long term mortality of urban cohorts with respective long term urban ambient air pollution monitoring data [3,4]. Epidemiologists talk about the health effects of outdoor air pollution, which has sometimes given rise to a false interpretation, namely that the observed health effects are due to the exposure that the population acquires while outdoors. To avoid confusion, some USEPA documents and articles use the terms 'air pollution of ambient origin' and 'air pollution of indoor origin' [5,6,7]. The current presentation focuses on the former, air pollution of ambient/outdoor origin, acknowledging that – due to our time-microenvironment-activity patterns - most of the exposure to such pollution occurs in indoor environments.

The significance of indoor exposures – even to pollution of ambient origin - arises from the fact that most people in the industrialised societies spend a very high portion of their total time indoors. One of the objectives of the *EXPOLIS* study [8,9], was to determine the time spent - average hours per workday - by active adult European urban populations in different microenvironments and activities. Grand average time spent indoors was 22.3 h/d or 93%, which does not include time spent in transit - also mostly indoors. [10]

Outdoor generated air pollutant is relevant for indoor air quality and health, if it meets three criteria; (i) penetrates effectively indoors, (ii) contributes a significant portion of the level of that pollutant in indoor air and (iii) is harmful for the health of at least some fraction of the population at the observed indoor levels.

Table 1. Time allocations for indoor Environments in seven European cities; Athens, Basle, Grenoble, Helsinki, Oxford and Prague. (h/d)

Indoor	Min (city)	Max (city)
Home	13.5 (Milan)	15.8 (Oxford)
Work	6.1 (Athens)	7.5 (Milan, Prague)
other	1,3 (Oxford)	2.2 (Grenoble)

CONTRIBUTION OF OUTDOOR AIR POLLUTION TO INDOOR AIR QUALITY AND HEALTH EFFECTS

The contribution of outdoor air pollution to indoor air quality and health vary greatly in relation to the pollutant, the climate zone and building stock, and also between the individual buildings. The current presentation, by necessity, focuses on the typical and average conditions and situations in urban areas across Europe, but it is important to realise that individual buildings may behave quite independently of these averages. Still, the variation of outdoor air contribution to indoor air quality is much smaller than the variation induced by the indoor sources.

Carbon monoxide (CO), and nitrogen dioxide (NO_2)

The health impacts of CO are caused by its strong affinity to blood haemoglobin, where it blocks the oxygen carrying capacity of the blood. Acutely dangerous CO concentrations in indoor air are responsible for thousands of lethal and many more non-lethal poisonings annually in Europe. These very high levels, up to hundreds of ppm, however, originate almost exclusively from indoor sources. Also ambient air CO concentrations are associated with daily mortality [11,12], although it is unclear how much of this is due to its strong correlation with other combustion generated pollutants - ultrafine and fine particles, NOx, CO etc. - and how much to the independent effects of CO. CO is a universal marker of incomplete combustion products, which also include gas, vapour and particle phase organics, soot particles and numerous carcinogenic substances.

The health effects of NO_2 at the environmentally relevant levels are not nearly as well understood as those of CO, although a body of epidemiological evidence associates bronchitic symptoms of asthmatic children with NO_2 concentration, and reduced lung function growth in children is linked to elevated NO_2 concentrations within communities already at current North American and European urban ambient air levels. Yet, in the recent risk assessments these effects have been increasingly assigned to other combustion and atmospheric chemistry generated co-pollutants, which are highly correlated with NO_2 [13]

The sources of CO and NO_2 are the same, stationary and mobile combustion devices. CO is generated in incomplete combustion conditions with low quality fuel, poor mixing or insufficient quantity of combustion air and/or low combustion temperature. The opposite conditions, well mixed excess air and high combustion temperature, produce NO, which is in the presence of ozone (O_3) in ambient air rapidly oxidised to NO_2.

Although urban ambient air CO levels have been reduced by an order of magnitude in the past 30 years, CO remains the most abundant urban air pollutant, and its concentrations range from a few hundred up to a few thousand $\mu g/m^3$. In indoor and urban air CO is an inert gas with an atmospheric half life of weeks, and therefore, in the absence of indoor sources, the indoor air CO concentration smoothly follow the 1..3 hour average ambient air concentration.

Unlike CO, NO_2 rapidly reacts with co-pollutants in the air or is absorbed by many indoor materials. When integrated over a full day or longer, indoor concentrations of NO_2 are usually, even in the presence of some indoor sources, like gas stoves or smoking, lower than respective ambient air concentrations, as can be seen from Table 2. In the PRIMEQUAL study on I/O relationships of air pollutants in 8 French Schools, the I/O ratios of NO_2 ranged from 0.87 to 1.17. The variation of I/O ratios for NO was much wider, 0.5 to 2.59 [14].

Table 2. 48 h. average outdoor and indoor NO_2 concentrations ($\mu g/m^3$) in three EXPOLIS cities [15]

City	Outdoor	Indoor	Workplace
Helsinki	24 ± 12	18 ± 11	27 ± 15
Basel	36 ± 13	27 ± 13	36 ± 24
Prague	61 ± 20	43 ± 23	39 ± 18

Short term concentrations of NO_2 in kitchens with gas stoves and (bath)rooms with gas fired unflued hot water heaters, however, may exceed 1000 $\mu g/m^3$, far above the levels ever observed in outdoor air [16].

Volatile organic compounds (VOC)

After CO, VOCs are found in highest indoor air concentrations and are consequently also responsible for the highest long-term population exposures, but high indoor VOC concentrations originate only rarely from outdoor air.

The health effects of concern for VOC:s are as variable as the VOC:s themselves. Benzene is an IARC classified category 1 carcinogen, causing leukaemia. A number of other VOCs and carbonyls (e.g. 1,3 butadiene, formaldehyde, etc) are suspected or known carcinogens, some are neurotoxic, many irritating and odorous. Their combined “cocktail effects” have risen great interest over the years, but are very complicated to study in practice. As for their population level health risks, on the one end the contribution of residential indoor VOC:s to human carcinogenesis is likely to be real but small, on the other end the sensory effects of VOC cocktails are mild but affect millions of individuals.

The simplest and often useful approach for assessing the relative contributions of outdoor and indoor sources to indoor concentrations, is to look at the indoor/outdoor (I/O) concentration ratios, which have been extensively reported since the late '70's. Ratios close to 1.0 indicate outdoor air as the main contributor to indoor exposure. High ratios up to 10 and above indicate that indoor sources dominate. This approach, however, is not reliable for compounds with high reactivity or absorbance to indoor surfaces.

Outdoor sources contribute the most (½ or more) to indoor air concentrations of, e.g., benzene (and often other aromatics), butanal and carbon tetrachloride [17,18,19,20]. Of these benzene warrants a closer look. Combining the result of the *EXPOLIS* [17] and Macbeth [21] studies provides a cross section of the residential indoor, outdoor, workplace (*EXPOLIS* only) and personal exposure levels of benzene in 10 European cities, see Figure 1.

Figure 1 highlights the large regional variations in the average contributions of urban outdoor air benzene to indoor concentrations and personal exposures. Outdoor air contribution to indoor air levels was highest, about 100% in Padua, Athens and Murcia, lowest about ½ in Basle, Rouen and Antwerp. Respectively the outdoor air contribution to personal exposure was highest in Oxford, and lowest $^{1}/_{3}$ – ¼ in Rouen, Antwerp and Basle. For most other VOCs and carbonyls the indoor source contributions to the indoor air concentrations are higher, close to or above 90 % in average for, e.g., acetone, formaldehyde, propionaldehyde, other aldehydes, limonene, isopropanol, α-pinene, camphene, undecane, dodecane and decane [18,19,22,23,24]. For the indoor air concentrations and possible health effects of these compounds, outdoor air contributes, in average, very little.

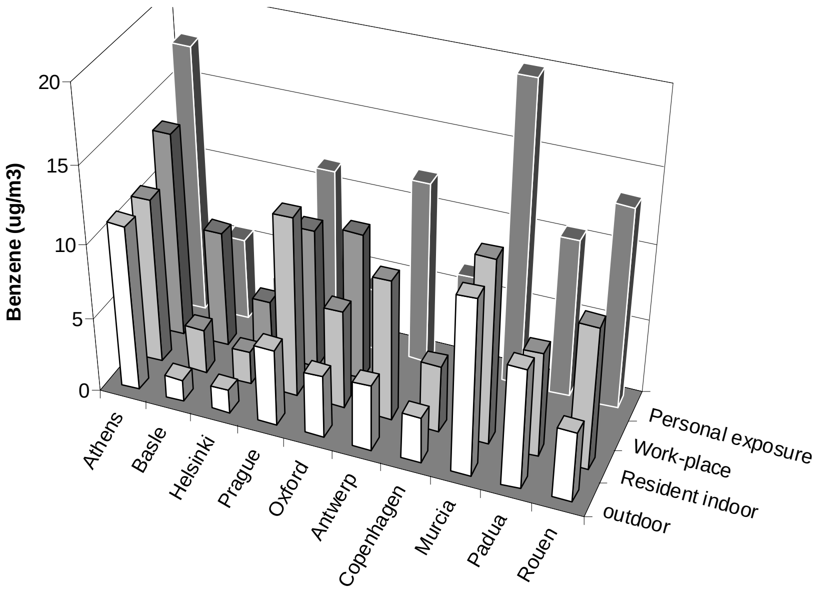

Figure 1. Averages of outdoor, residential and workplace indoor concentrations and personal exposures to benzene in 10 European cities.

Urban - rural: In a German study outdoor and indoor concentration differences for benzene, toluene, ethylbenzene and xylenes (BTEX) were assessed between urban (High traffic area in the city of Hannover) and rural (in Wedemark, no major traffic nearby) areas [25, 26]. Although the average outdoor air concentration were about 10 times higher in the urban vs. rural area, the respective indoor air concentrations of benzene were only about 50% higher in the city. For the other aromatics the average indoor concentrations in the urban and rural areas were even closer. I.e. the indoor sources for BTEX all but eliminated the urban-rural outdoor concentration differences. Because the outdoor source contributions are generally higher for BTEX than for the other VOCs, urban-rural indoor exposure differences for the other VOC:s - terpenes in particular - are likely to be even smaller and only weakly reflect the respective outdoor air concentration differences.

Ozone (O_3)

Ozone levels in ambient air are almost inversely proportional to most other air contaminants. The highest ozone levels are found in rural and seaside areas away from the pollution sources, and in urban areas in sunny afternoons, when the vertical mixing is at its highest. Conversely the levels are very low in the street canyons under high traffic conditions, where traffic emitted NO consumes all O_3 in its oxidation reaction to NO_2. Because of the very different concentration pattern compared to other air pollutants, the health effects of ozone are relatively easy to differentiate from those of the other pollutants.

According to the European CAFÉ report [27] O_3 is the other pollutant, after fine PM, which causes significant excess morbidity in Europe. Ozone is not clearly associated with increased mortality, but it reduces the lung function in children, and induces and aggravates asthma and other inflammatory respiratory conditions [13]. The difficulties in long term ozone epidemiology studies arise from the facts that the highest ozone levels are found in sparsely populated rural areas, and that the indoor levels are generally much lower, and highly variable in relation to the outdoor concentrations. In other words, ambient air ozone monitoring reflects poorly the ozone exposures of the population. On the other hand, indoor ozone sources (large or old photocopiers, ozonizers) are rare and therefore do not disturb the I/O ratios in most indoor environments.

Ozone is clearly more reactive than the other regulated air pollutants, and therefore indoor air ozone levels remain below the ambient air concentrations even under open window ventilation. Ozone levels in sealed and centrally air conditioned buildings are only in the order of 1/10 of the ambient air levels. In the PRIMEQUAL study the indoor air ozone levels in French schools ranged from 0 to 45% of the ambient air levels [14].

Particulate matter (PM)

Particulate air pollution is (in sampling, regulation and research) usually divided according to particle size into coarse (2.5 - 10 µm aerodynamic diameter), fine ($PM_{2.5}$ < 2.5 µm) and ultrafine (< 0.1 µm) fractions. The two first fractions are measured as mass concentrations, the third by counting the particles in a unit air volume.

After years of debate and research, the general understanding has emerged that urban fine PM represents the air pollution that has the biggest health effects [1,2,3,4]. CAFÉ report [27] estimates that in 2000, fine PM with a smaller contribution from O_3 caused 370 000 excess deaths and 3.62 million lost life years in Europe. These numbers exceed by far the mortality or lost life years estimates for any other environmental contaminants in Europe. Most of this excess mortality is due to acute cardiac death, but also respiratory disease and lung cancer contribute significantly.

In the Western urban context, the most important source of indoor air particles - in the absence of smoking - is usually outdoor air. PM2.5 is the particle size fraction that is most evenly distributed over large urban areas. Ultrafine particle numbers (UFP#) are, instead, quite unevenly distributed, and data at two urban monitoring sites correlate poorly with each other [28], let alone with personal and indoor air concentrations. Penetration of particles from outdoor air into indoor spaces is therefore a matter of great interest for indoor air quality and public health.

Of the commonly monitored particle size fractions, $PM_{2.5}$ infiltrates best (30 - 90%) from outdoor to indoor air. Indoor infiltrations of ambient coarse ($PM_{2.5\text{-}10}$) and ultrafine particles are lower. In non-smoking homes in Boston, MA, minimum infiltrations were observed for ultrafine and coarse, maximum for 0.1 - 0.5 µm particles. Infiltration efficiency depends also on the season and home characteristics - high in the summer with no air conditioning (AC) and high air exchange rate (AER), but low with AC and low AER) [29]. In two studies made in very different climate zones and building types, namely residences in Brisbane, Australia

[30], and offices in Helsinki, Finland [31], indoor concentrations followed the outdoor concentration changes in a smoothed and delayed pattern. In the Australian homes, however, the I/O ratios were 0.8 - 1.0 for both particle number and $PM_{2.5}$ mass concentrations, while in the Finnish offices the I/O ratios fell within a much broader range, 0.05 - 0.75. In another Finnish office building with a mechanical ventilation system equipped with EU7 class filters, infiltration of outdoor air particles was highest (0.25 - 0.3) for 0.2 - 0.5 um particles, lower (0.15 - 0.2) for 0.1 um, and lowest (0.02 - 0.06) for 0.01 um particles [32]. In the PRIMEQUAL study on French schools, the I/O ratio increased with increasing particle size (0.3 to 20 µm), and averaged 0.82 (0.41 – 1.18) for $PM_{0.3\text{-}0.4}$, but was 3.7 (0.15 – 9.0) for $PM_{5\text{-}7.5}$. [14]

In an American study of two retirement homes (in Baltimore, DE, and in Fresno, CA), indoor $PM_{2.5}$ concentrations in air conditioned apartments were, in average, only 45% of outdoor concentrations, but reached 80 % when AC was turned off and homes were ventilated through open windows (April - May in Fresno, CA). Personal exposures of the residents followed closely the indoor air concentrations with a mean 3 $\mu g/m^3$ increase from personal cloud [33].

I/O ratios of fine PM and NO_X depend also on outdoor meteorological parameters, which influence both the ventilation methods and air exchange rates. In Hong Kong the impacts were quite similar for both pollutants, namely strongly increased I/O for increased ambient temperature and humidity, weak increase for solar irradiation and no effect for wind speed and atmospheric pressure. [34]

RISK MANAGEMENT OPTIONS FOR AMBIENT AIR POLLUTION INDOORS

Most of the population exposure to and therefore - logically - also mortality and morbidity from ambient air $PM_{2.5}$ is caused by indoor exposure to $PM_{2.5}$ of ambient origin. An interesting epidemiological study [35] pointed out that, indeed, increased fraction of buildings with central air conditioning within the community significantly reduces the community wide association between ambient air PM and serious health effects (CVD, COPD, Pneumonia). This predictable fact opens a significant reduction potential for urban $PM_{2.5}$ mortality risk in buildings.

In more recently built buildings with advanced two way mechanical ventilation with intake and circulation air filters and air conditioning systems, the infiltration of outdoor air $PM_{2.5}$ is significantly lower than in older buildings with natural ventilation via open windows and vents. A probabilistic exposure modelling exercise demonstrated that reducing the $PM_{2.5}$ infiltration into all buildings in the city of Helsinki to the level of the office buildings built after 1990, would reduce the population exposure to $PM_{2.5}$ from ambient origin as well as its adverse health effects by 27%, in fact almost as much as total elimination of all traffic sources from within the metropolitan area limits. [36]

O_3 is another air pollutant with significant public health risks, against which buildings and ventilation & AC systems can provide a significant level of protection. Also *nitrogen dioxide* is sufficiently reactive to be significantly reduced indoors vs. outdoors.

Urban air pollution risk management applying building technologies should not be seen as an alternative, but rather as a complementary strategy for urban air pollution risk reduction. This is highlighted by interesting differences between the two strategies. Reduction of local air pollution sources has no impact on exposure to the air pollution load from extra-urban sources, which are often responsible for 50...66 % of the exposure. Building technologies reduce pollution exposure from intra- and extra-urban sources equally. Reduction of local air pollution sources has only limited and slowly materialising potential to protect vulnerable population groups or individuals, while building, ventilation and filtration technologies can be targeted for efficient and rapid individual or building level protection.

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Integrated management and control of building services

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SUMMARY

Professional building owners and facility managers require system solutions featuring integrated building management comprising building automation and control along with fire safety and/or security for the operator and at technical levels. Those solutions require coordinated procedures during the design phase as well as during the execution phase. The integration of different systems is carried out by using standardized communication protocols. One of these protocols to combine different building systems is the globally accepted protocol BACnet according to EN ISO 16484-5. The same standard, Part 3 defines the “BACS functions”. The definitions are valid for the entire branch of the building automation and control (BAC) industry to serve as a common language for all those involved in the design, implementation and operation of such systems.

A functional networking of systems from different disciplines calls for involving complex efforts in terms of design engineering and implementation. The uniform object model of BACnet provides the brand and system neutral application and WEB services for interoperable integration with enterprise systems and other BACS using XML and the internet technology for networking, the transport of data.

The standards for communication protocols in the various fields of building applications are BACnet, LonWorks with LonMark and KNX/EIB as well as more or less “open” methods as OPC, and MODbus. LonWorks is a national American and national European standard under the property rights from Echelon Inc. used mainly for data transportation and complex functionality in the field of room automation and smaller BACS. LonMark is a private “standard” to achieve interoperability using LonWorks products. KNX/EIB is also an American and European standard, but for 100% interoperability of products used in general electrical installation that also includes integrated room automation.

The field of building automation and control and the integration of related building services needed a more comprehensive functionality. So BACnet became the international communication standard for BACS. Both, LonTalk (the transport protocol of LonWorks) and KNX/EIB have become a normative part of BACnet and therefore they are already part of the global EN and ISO standard for building automation and control systems. To use LonMark products within a BACnet environment, a Gateway to match the different semantics is necessary.

BACnet already has defined provisions for the functional integration of integer fire safety and access control systems supporting the BACnet standard; the “life safety” communication objects and services.

The question of responsibility in projects with combination of autonomous systems for safety/security alarming and building automation and control is a serious matter of the various construction contracts – and is not covered by standards yet.

INTRODUCTION

Integrated technical building management

Technical building management probably represents the most important task within the scope of facility management while a building is being used. This is where all technical tasks to manage a building and its equipment come together.

The control of plants for technical building services as well as with building operation monitoring for life safety and security management, energy management and supervisory control for of power distribution installations are the tasks for technical building management.

The operating of modern industrial and commercial buildings requires clearly delineated building management tasks:

- Commercial building management focuses on managing and controlling processes.
- Infrastructural building management for general services rendered during building use.
- Technical Building Management with tools for operation and supervision.

Communications for human interaction and for alarm notifications is using facilities such as telephone, pager, intercom, radio, and staff paging systems. However communication within and between technical plants is carried out via networks using communication protocols.

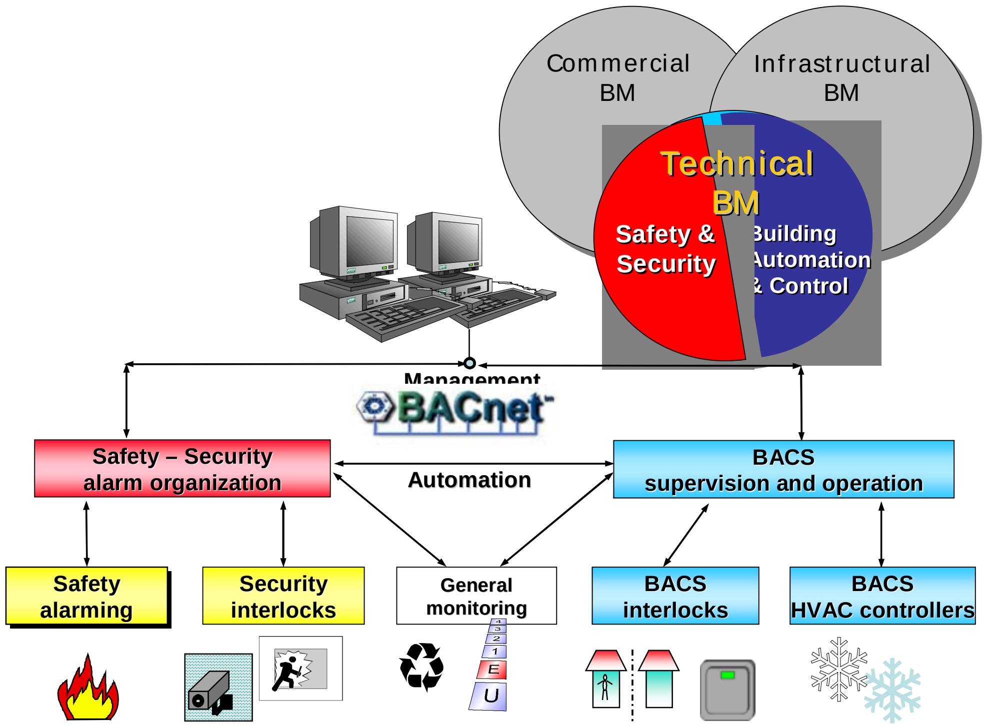

Figure 1. Technical building management and different systems in interaction.

The discussion of system integration has become an increasingly important topic on the agenda of engineers and other experts involved building automation and control systems technology. This discussion is influenced by contributions from the different bus systems or networks available for building applications. The heart of the matter is to solve the problem of exchanging unambiguous “functional” information between different technical systems. Comprehensive, project and estate-specific data management with a building management system will be cheaper from the lifetime point of view because the owner only has one database that needs to be managed and updated. It also eliminates data inconsistencies resulting from a multitude of different local or department-specific databases. The user can then transform the financial advantages into increased competitiveness.

RESULTS

Combined technical building systems

To understand the needs of systems integration, first a definition the different systems installed in a building is essential. The possibilities of an interoperable connection between life safety/security alarm systems and systems for building automation have to be considered.

Figure 2. The different disciplines to be interoperable combined.

The operating of modern industrial and commercial buildings requires clearly delineated building management tasks. The European standardization bodies have subdivided the management tasks as follows:

Commercial building management

Commercial building management focuses on managing and controlling processes. Examples for commercial building management are:

- Contract management,

- Tennant billing,
- Cost planning and controlling,
- Object accounting,
- Procurement,
- Project management.

Infrastructural building management

The infrastructural building management comprises general services rendered during building use such as:
- Security services,
- Guard services,
- Telephone services,
- Cleaning and waste disposal,
- Moving services,
- Campus services,
- Winter/snow removal services,
- Courier services/internal mail services.

Life safety and security systems

We can subdivide these systems into:
- Danger Alarm Systems:
- Fire alarm systems (fire safety),
- Burglar alarm systems (security),
- Raid alarm systems,
- Transmission systems for such alarms.

Often the following installations are combined with the detection and alarming systems:
- fire extinguishing devices,
- fire alarm devices,
- CCTV systems for room monitoring,
- access control systems,
- watchman's reporting systems.

Criteria for life safety and security systems

From the security engineer’s point of view, the terms "integrity" and "integration", on closer consideration, become antonyms. The term "integrity" is an important factor within their security concept. A look in the dictionary might be helpful in casting light into the depths of terminological confusion.
- integer (Latin adj.): untouched, entire, soundness;
- integration: the act of combining or adding parts to make a unified whole.

Asking the users of security systems about their expectations with regard to the performance and usefulness of these systems, their main criteria are as follows:
- fail-safety/availability,
- low incidence of false alarms,
- fastness and reliability of repair service,
- state-of-the-art technology,

- user-friendliness,
- service quality,
- delivery safety,
- confidentiality.

(It is interesting to note that the above criteria frequently even range before the price!)

Building automation and control systems (BACS)

The concepts of "supervisory control systems", "centralized control systems" or "building services management systems", have been supplanted by the term "building automation and control systems" (BACS), as a complete system for digital measuring, control, supervision and management.

Almost invisibly, building automation and control systems and their operators ensure ecological, economical and safe operation of a building.

The past experience made sure that it never would work sufficient to interconnect functions without standardized semantics.

Functions within the building automation and control systems

The BACS industry has learned that automatic functions need to become clearly defined and to be standardized for reducing the risk during tendering and construction (for both sides). This includes the I/O-functions as well as the complex processing and optimization functions such as "start-up control", "sequence control", "frost protection", "night cooling mode", "peak load limitation", "long term history", "dynamic display" etc.

Therefore the industry hardly is promoting standard descriptions of building automation functions and of a communication standard describing semantic communication objects.
The common definition of "BACS functions" is given in the global standard EN ISO 16484-3 to serve as a common language for all those involved in BACS projects. Fifty standardized functions include the I/O-functions as well as the complex processing, optimization and operator functions.

Those fifty standardized functions cover:

1. Hardware-related and communicative (shared) input and output functions for event detection, measuring, counting, switching and actuating – the "basic I/O functions",
2. Functions for monitoring and display of the different processes, as well as alarm functions in case of faults or of limit violations,
3. Processing functions for safe local automation - closed-loop and open-loop and PID-control - of building services and user-specific installations,
4. Processing functions for energy-saving operation of the different processes (local optimization),
5. "Centralized" processing functions for time scheduling and comprehensive optimization strategies covering all areas and components of a building (general optimization and coordination of plant operation),
6. Process control functions for "human system interface", including service operation on the "field level" with local direct process control and indication devices,
7. Management functions for analysis and statistical display of current or historical process values to facilitate decisions,

8. Tools for calculation or estimation of optimized operation parameters, and for combination and programming of new modes of operation,
9. Functions for data exchange and interoperability with other systems (system integration),
10. Remote control or telecontrol functions via the web or public networks,
11. Tools for modifications, expansions, and testing,
12. Data archiving functions for backup and restore.

The complexity of combining systems

Fire detection systems have been integrated with door locks and with HVAC fan and damper controls for smoke management for several years, but these systems have relied on relays controlled by the fire alarm system to override the normal controls. This kind of integration has primarily involved constant-volume HVAC systems and required only on/off control of fans and dampers to be moved to fully open or fully closed positions. Many modern HVAC systems are far more complex. Variable air volume and individual room control systems are used to reduce energy consumption. These systems require sophisticated control algorithms to operate either a continuously variable-speed fan or inlet guide vanes to control the static pressure in the supply air duct. Variable air volume boxes control the airflow from the supply duct into individual rooms by modulating dampers. What is needed is a way for the fire alarm system to command the HVAC control system to enter zone wise a smoke control mode and let the HVAC controllers manage the equipment.

Purchasing of system integration

To purchase (and to offer) system solutions featuring integrated building management
- fire safety and/or security along with building automation and control -
coordinated procedures during the tendering phase, the construction phase as well as during service are required.

Another important aspect is the intentional interdependence of different functions. This means that, for instance, signals from safety and security installations (fire/burglar alarm) should trigger suitable reactions in the lighting, elevator and supply systems. Mutual exchange of electronic information between the different systems, without restricting their individual autonomy, would thus be useful for all systems involved.

Damage to staff and/or buildings can be limited through automatic "secondary" actions in the technical installations initiated by alarms from the safety and security installations, which, in turn, initiate "primary" actions (emergency call, fire extinction etc.). In many cases, these functional chains are stipulated by the laws on the erection of buildings.

Integration is not just a question of supplying equipment, but it is a task in its own right. For this reason, the networking of functions requires that the objects, properties and services used for the interaction of these functions are standardized as well as there is a need to restrict the methods for the data transport (the lower communication protocols in the ISO/OSI reference model), i.e. the interface definitions must describe the data transfer medium, the transport protocol, but above all the "application objects".

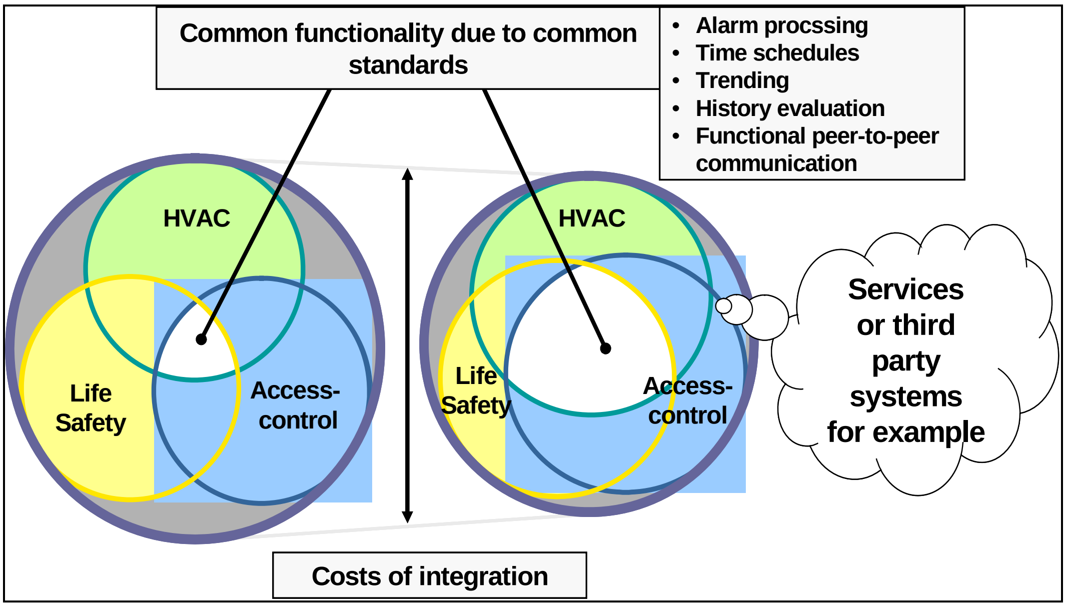

Figure 3. The van diagram of system integration and standardization

Characterization of combined and integrated systems

“Combined systems” fulfill their functions with jointly used system components within the valid standards, making sure that there is no impact of faults on the defined main functions. If faults occur in a specific installation these have no impact on any other system.
“Integrated systems” are characterized by a joint utilization of system components with possible interferences between system components. In this context, new and more detailed regulations and standards are necessary which take into account the need to preserve the integrity of the individual system components.

The discussion of system integration has become an increasingly important topic on the agenda of engineers and other experts involved in alarm systems technology. This discussion is influenced by contributions from the different bus systems or networks available for building applications. The heart of the matter is to solve the problem of exchanging unambiguous “functional” information between different technical systems.
All together is what we call “systems integration”.

Methods for combining different systems

Coupling of different systems can be implemented through direct "parallel" transfer of each information via coupling relays, or by means of "serial communication" in the case of larger building complexes. The different systems must be able to "understand", but not interfere, with each other. It is even more important that the aforementioned actions/measures do not reduce the functionality of the involved systems or installations. Of course, it also is necessary that the performance of different integrated systems can be accounted for individually (acceptance test/warranty). Furthermore, we have to be aware that systems integration or simple coupling of systems is subject to certain restrictions caused by safety and insurance regulations.

We have seen that the principal expectations of the two user groups for automation and safety differ considerably. Nevertheless, it is important to look for common ground because there is undoubtedly a large demand for combined/integrated systems on the market. However, it is important to judge these systems on the basis of safety aspects.

Great diversity and complexity also mean an increased interdependence between the different systems. The different processes required for project planning and design, erection, commissioning, acceptance test, operation and service involves a continually increasing number of people who have to cooperate and communicate by means of various different interfaces.

Is OPC a solution?

Control devices and operator interfaces often are interconnected via an OPC interface. OPC is a general industry standard depending on and restricted to the procedures of the former Microsoft Windows operating system, it does not work under MS Vista. It also is not a standard in the ISO/EN sense. It is used for the connection of PLC’s (controllers) and monitoring stations under Windows.

For professional BACS, OPC has too many functional restrictions and limited reliability with respect to the data transmission within Windows-PC. OPC does not use semantics, data points and application objects. This means that the relationships between the functions of an object, e.g. between a counter input and the associated limit values and other parameters get lost. It requires double and more engineering efforts because all variables of a control data point need to be created within the controllers and to be re-created for the monitoring system.

Uniform Object Model

One can easily imagine that a well-planned data model for building automation and control is a lucrative asset both for owners, for prime contractors and for facility managers particularly if this data model is based on a globally accepted International Standard. A uniform data model, however, is only possible through the creation of a uniform object model which enables access to the internal data of different systems.

Common ground

Fortunately, with “BACnet” we have the global standard that provides such an object model, the required procedures and data access services. And… it supports the appropriate data transport methods by referencing the most common networks used in buildings. In the following we will have a closer look to BACnet.

What is BACnet?

BACnet is originally the ASHRAE data communication protocol for building automation and control networks. The field of application covers the communication needs systems for applications such as heating, ventilating, and air-conditioning control, lighting control, access control, and fire detection systems, and some more are in development [1].
BACnet was not intended for use in the application fields for home automation, medical applications, white and brown goods, and consumer electronics.

After a parallel voting within ISO and CEN, BACnet became an ISO and an European Standard in 2003. It now is: **EN ISO 16484 Part 5 „BACS protocol“** [2] [3]. “EN” shows that this also is a National Standard in 30 European (CEN) countries.

The protocol is describing:
1. Object types for application messages,
2. Services for data-/information access, with messages and their format and syntax,
3. Data transport to convey data by referencing standards. with networking options (LANs, WANs, Dial-up),
4. Joining networks together to form "internetworks".

BACnet is designed specifically for building operation - some functional examples are: Alarm and event processing, scheduling, trending, command prioritization, control logic, life safety and security/access objects. But: The protocol does not specify the system itself, only provisions for interoperability of different systems or products.

BACnet Object Types

The BACnet “objects” represent the behavior of physical inputs, outputs and software processes. Each object is characterized by a set of “properties” that govern its operation. The BACnet Standard in the moment defines a collection of 28 standard object types. For example, an analog input is represented by a BACnet Analog Input object that has a set of properties that include its present value, sensor type, location, alarm limits, and others. Some properties are required and others are optional, depending on the functional use. A device is represented by an appropriate collection of network-visible objects. Once the information and functionality of a device are represented on the network in terms of standard objects and properties, messages can be defined to access and manipulate this information in a standard way.

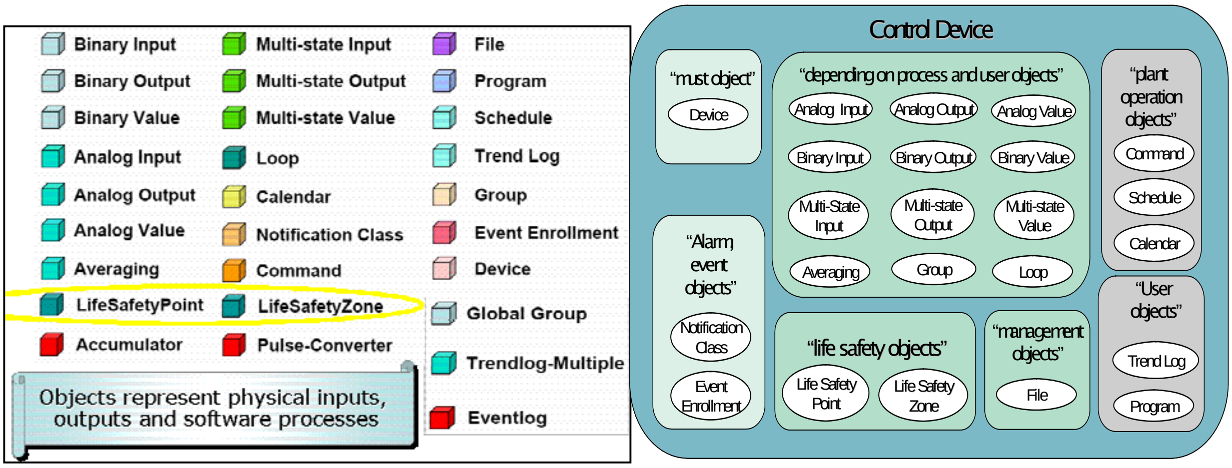

Figure 4. a) The recent BACnet object types, and b) objects in a BACnet device

The object-oriented structure also provides a way to add new application functionality to BACnet by defining new objects and/or new application services. The safety/security additions were developed with assistance from the fire alarm industry and have been approved by the BACnet committee. Other new object types have been developed for video surveillance (CCTV), access control, other security applications, utility and lighting control integration. Hey are being integrated into the global standard soon.

Application services

A communication with BACnet is described by reading and writing of information as contents of the objects' properties. The definitions are unambiguous and complete.

1. Alarm and Event Services,
2. Data reading and writing Services,
3. File Access Services,
4. Object Access Services,
5. Remote Device and Network Management Services,
6. Virtual Terminal Services (for engineering),
7. WEB services for integration with enterprise systems using XML.

XML is a mechanism to describe information in a manner that is flexible and can easily be communicated across IP networks. Web Services can best be described as a mechanism for applications (both system and device-centric) to communicate with each other across the Internet and intranets.

These services include provisions for Scheduling, calendar and time depending actions and for Trending with Logs of analog values and events.

BACnet's data exchange philosophy follows the well known client and server concept, but dedicated to functionality not to certain devices. Additional BACnet is providing procedures for applications and other services. Examples are the prioritizing of event notifications and commands. Other procedures exist for system start up, power recovery, and failures with additional error, reject, and abort codes.

BACnet Networking Options

BACnet provides a wide variety of options for the transport of data. These options comprise:

1. Ethernet, IEC 8802-3,
2. ARCNET, ATA/ANSI 878.1
3. Master-Slave/Token-Passing (MS/TP), ISO 16484-5, EIA RS 485,
4. Point-to-Point (PTP), EIA RS 232-C
5. Echelon's LonTalk, EIA/CEA 709.1-B
6. BACnet/IP, (Internet protocol),
7. BACnet/WS (Web services), ISO 16484-5 Am. 2004-c
8. In recent discussion are wireless networks such as ZigBee or Bluetooth, WLAN already is provided by Ethernet/IP.

The first option is ISO 8802-3, better known as "Ethernet", also together with the Internet Protocol (IP). It is the fastest option and is typically used to connect workstations, controllers and high-end field devices. Further on, in combination with the internet protocol Ethernet develops as the most future proof method. The second option is ARCNET, which comes in slower, lower-cost versions (mainly in US). BACnet defines the MS/TP (master-slave/token-passing) network designed to run over twisted-pair wiring. Echelon's LonTalk network can also be used. The Ethernet/IP, ARCNET, and LonTalk options all support a variety of physical media. BACnet also defines a dial-up or "point-to-point" protocol called PTP for use over phone lines or hardwired EIA-232 connections.

A key point is that BACnet messages are the same no matter which LAN is used. This makes it possible to easily combine LAN technologies into a single system.

BACnet/WS Web Services facilitate integration of BACS and "enterprise" systems for scheduling, energy management, preventive maintenance, and other global applications, using XML, SOAP and other standard Internet Protocol capabilities.

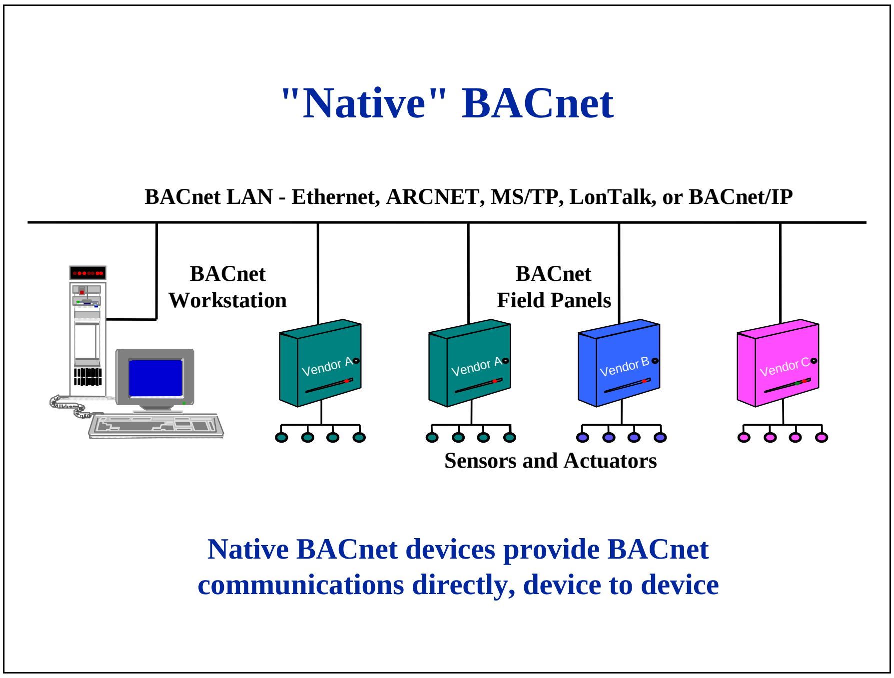

Figure 5. A flat architecture

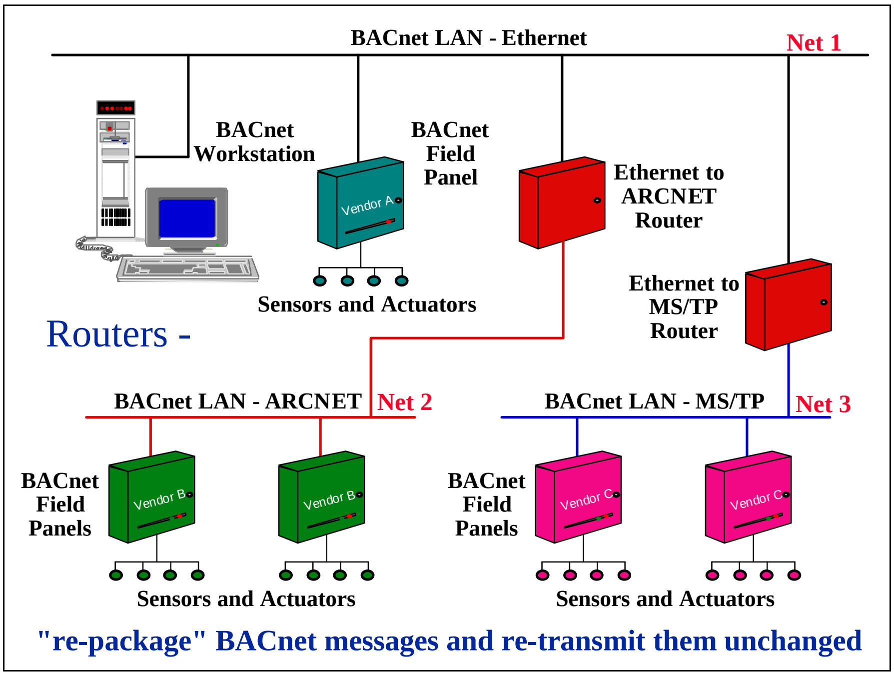

Figure 6. A hierarchical architecture – BACnet allows both

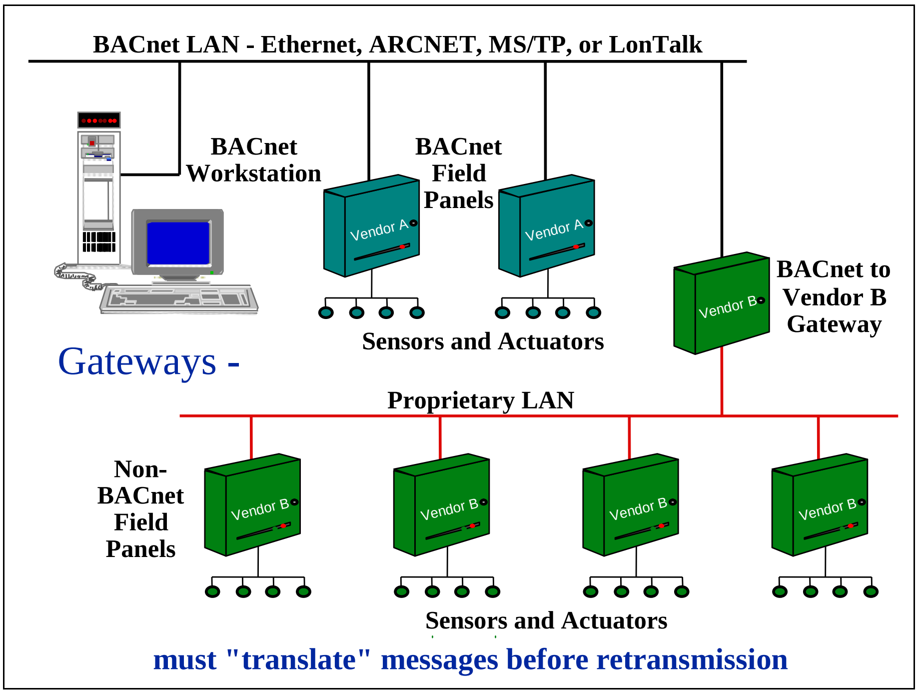

Figure 7. Integration of existing systems by gateway

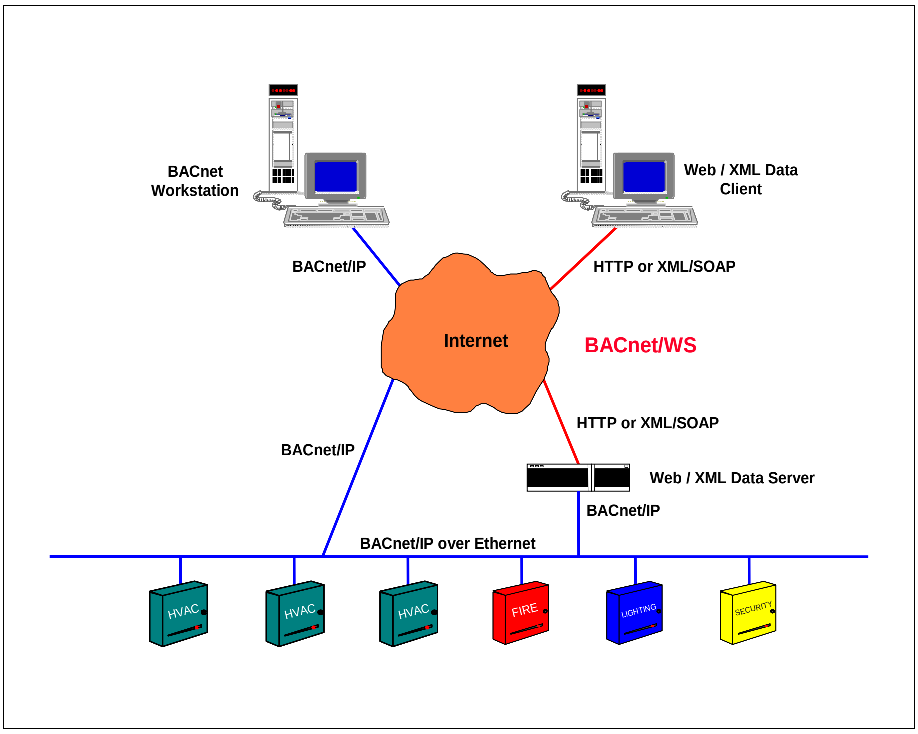

Figure 8. BACnet via internet

BACnet maintenance

BACnet is maintained under ASHRAE “continuous maintenance” procedures by a committee (SSPC 135) representing all sectors of the industry, including liaisons from Europe and Asia. The current ASHRAE SSPC 135 Working Groups:

1. Applications (AP-WG)
2. Internet Protocol Issues (IP-WG)
3. Lighting Applications (LA-WG)
4. Life Safety and Security (LSS-WG)
5. MS/TP LAN Issues (MS/TP-WG)
6. Network Security (NS-WG)
7. Objects and Services (OS-WG)
8. Utility Integration (UI-WG)
9. XML (XML-WG)

Changes can be proposed at any time by the public or a committee member. All changes are subject to public review and comment. There are formal liaisons with:

- ABOK (Russian HVAC society),
- CEN/TC247 (Building Automation and Building Management),
- IEIEJ (Institute of Electrical Installation Engineers of Japan),
- NEMA (National Electrical Manufacturers Association),
- SIA (Security Industries Association),
- SWEDVAC (Swedish HVAC society).

BACnet and Conformity

Open communication is designed to enable the erection of dissimilar systems with common functionality, including systems with products and/or engineering by different suppliers and manufacturers.

During the development of interfaces and other software, faults with the implementation of communication protocols cannot always be avoided. It therefore is necessary to test the devices for conformity with the given specification or standard. The global standard BACnet already has such a test specification as EN ISO 16484-6. The association "BACnet International" has created the "BACnet Testing Laboratories" (see BTL-sign) in USA, Europe and India.

This testing is a type test of whether or not the reaction of the product's software being tested conforms to significant test parameters of the protocol. These tests are carried out by authorized institutes, in accordance with standard testing conditions, so as to ensure that all products are tested under identical conditions. These tests do not provide insights into the dynamic behavior of the test object within an integrated system as a whole.

Only an interoperability test under typical working conditions which includes all the different products involved in one project would bring up the system behavior under different fault conditions. But this can be very expensive indeed – so it is better to leave this task to a trustworthy company.

Figure 7. The BACnet test logo

BACnet testing

The Companion standard ANSI/ASHRAE 135.1, Method of Testing Conformance to BACnet was published in 2003 [4]. The BACnet functionality of listed products is published on a web site, accessible to everyone: www.bacnetassociation.org.

Testing requirements and procedures are made public. Most testing tools are easy to obtain from http://sourceforge.net/projects/vts. Another automated test software was developed in Europe. Information is given by the BIG-EU: - www.big-eu.org [5].

Manufacturers can run tests in their own labs before submitting products. Organizations in North America and Europe worked together to create a unified testing and listing program. For the tests, the functional scope of the BACnet devices is defined by the manufacturers and – if successful tested - listed by BTL.

The BACnet standard defines the PICS (Protocol Implementation Conformance Statement) and the BIBBs (BACnet Interoperability Building Blocks) to allow an accurate assessment of interoperability. A classification of BACnet devices is described with device profiles. Each device profile (operator workstation, building controller, advanced application controller, application-specific controller, smart actuator, and smart sensor) has a minimum of BIBBs expected of it.

BIBBs are definitions of functions which are important to operate and monitor the special objects subdivided into data sharing, alarm and event management,
scheduling, trending, device and network management.
PICS are provided by manufacturers and give information about devices:
1. The implemented object types, (input, output, schedule ….) for the Application,
2. Application services and their role (initiate or execute),
3. Physical media and lower protocol layers,
4. Character sets supported.

BACnet Vendors and Users

There is a steady growing community of BACnet vendors, as of May 2, 2007, 255 Vendor IDs have been issued and are distributed internationally from 29 countries [6].
There are ten-thousands of installed systems ranging in complexity from a single gateway to very large office buildings with top-to-bottom native BACnet systems, to campus or city-wide systems linking multiple buildings in over 82 countries and on all continents of the earth.

The benefits of BACnet

This protocol has been designed specifically for building operation - some functional examples are:

* Alarm and event processing,
* Scheduling,
* Trending,
* Command prioritization,
* Control logic,
* Life safety and security objects.

Remember: BACnet does not specify the system itself, only provisions for interoperability of different systems or products.

1. BACnet doesn't depend on current technology,
   * BACnet is implemented through software, not hardware,
   * BACnet's objects and services are independent of the underlying network technology,
   * BACnet/WS allow communication between BACS and enterprise applications.
2. BACnet has no fixed architecture,
   * BACnet devices can be arranged in flat, bus-like topologies,
   * BACnet devices can be arranged hierarchically.
3. BACnet can be implemented in devices of any size:
   * General purpose, programmable controllers,
   * Configurable, fixed program controllers,
   * Application specific controllers,
   * Workstations and Web servers,
   * Tool devices and protocol analyzers.
4. BACnet can be easily enhanced and improved by extending the object model.
5. There is no charge for the use of BACnet - no fees, licenses, royalties, etc.!

DISCUSSION

A question of responsibility

One important motive for the client to contract only one supplier is to be able to assign responsibility in case of faults. It is therefore obvious that all attempts at integration of different functions or systems have been limited to individual manufacturers or suppliers of systems.

Faults occurring during the integration and operation of systems, including signaling failure, malfunctions, or other faults, inevitably lead to the question as to who is to account for the resulting damage. Another problem is to allocate the proper share of responsibility for faults or damage to every single supplier involved. “External" systems, for instance, can be infinite sources of faults. For this reason, the question of insurance is particularly important, especially where the protection of life and limb is concerned [7].

Combination or integration of security and signaling systems with other systems is only acceptable if the possibility to detect faults, and the possibility to document any external influence on the safety alarm system can be fulfilled. Combination of autonomous systems for safety/security alarm and building automation and control becomes feasible if responsible-minded companies create interfaces ensuring that all external influences on the system can be traced back to their origins.

Anyway - whatever the future may have in store with regard to system integration, we should not let ourselves be guided by wishful thinking and idealized notions about technological feasibility. Instead, we should always be aware of the human weaknesses and their impact on the functioning of combined or integrated systems.

With regard to project integration, we can normally assume that a trustworthy supplier of "combined systems" assumes the role of the general manager and therefore also ensures the necessary degree of integrity.

ACKNOWLEDGEMENT

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Heating and cooling systems for better energy efficiency

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SUMMARY

Heating, ventilation and cooling of buildings is responsible for 30-40 % of the energy consumption in buildings and a corresponding significant amount of CO_2 emission. Since 2006 the European Energy Performance of Buildings Directive (EPBD) is being implemented in building codes on a national level. For new and existing buildings this requires a calculation of the energy performance of the building including heating, ventilation, cooling and lighting systems, based on primary energy. Each building must have an energy certificate and regular inspections of heating, cooling and ventilation systems must be performed. This has increased the focus on energy efficiency of systems and there components. The concept of the European standards developed to support the EPBD is presented in this paper. These standards may include alternative calculation methods and many European countries are using national methods. This means the heating system may be evaluated differently in different countries.

In Northern Europe a heating system is still needed even if the insulation of buildings will increase significantly. However, in middle Europe the increased insulation may result in a very low heat demand and a full heating system may not be needed. Also global warming will in the future impact the need for a heating system in many geographical areas. This may lead to alternative solutions for heating of buildings.

On the other hand peoples wish for more comfort, future higher outside temperatures, heat islands in big cities will increase the use of cooling systems. Further more we must increase the use of renewable energy sources. This requires systems, which can be used for heating at relative low temperature (air-water) and used for cooling at relative high system temperatures. The paper will present studies on the use of water based radiant heating and cooling systems with water temperatures close to room temperatures. The paper also introduces the concept of exergy for evaluation of sustainable heating and cooling systems.

INTRODUCTION

In 2003 the European Commission (EC) issued a directive, 2002/91/EC [1]. The objective of this directive is to promote the improvement of the energy performance of buildings within the community, taking into account outdoor climatic and local conditions, as well as indoor climate requirements and cost-effectiveness. The directive refers to the energy use and dos not take into account the life cycle energy demand (energy used to produce the products used for the building) For new and existing buildings this requires a calculation of the energy performance of the building including heating, ventilation, cooling and lighting systems, based on primary energy. Each building must have an energy certificate and regular inspections of heating, cooling and ventilation systems must be performed. This directive required all member countries by January

2006 to implement the directive in the building codes on a national level. Until now this has only been implemented fully in a couple of countries.

CONCEPT OF THE EPBD-STANDARDS

A mandate to the European Organization for standardization (CEN) from the Commission, M343-EN-2004 [2] was issued. This mandate asks CEN to elaborate and adopt standards for a methodology, calculating the integrated energy performance of buildings and estimating the environmental impact, in accordance with the directive. To coordinate the standardization related to the EPBD, CEN established an EPBD-Project Group including the following Technical Committees (TC's):

TC 89 Thermal performance of buildings and building components;
TC156 Ventilation for buildings;
TC169 Light and Lighting;
TC228 Heating systems in buildings;
TC247 Building automation, controls and building management

The standards under the mandate shall constitute an integrated and interacting methodology for the calculation of the energy uses and losses for heating, cooling, ventilation, domestic hot water and lighting systems, taking into account natural lighting, passive solar systems, passive cooling, position and orientation, automation and controls, and auxiliary installations necessary for maintaining a comfortable indoor environment. The methodology shall integrate, where relevant, the positive influences of active solar systems and heat and electricity from renewable energy sources, as well as quality co-generation heating plants (CHP, including micro-CHP) and district heating and cooling systems. It should also facilitate an estimation of the environmental impact from this energy use and provide data requirements for carrying out standard economic evaluations for the use of different systems.

This series of standards (about 40) have or are being voted for as final standards. The objective is to establish common calculation methods in Europe for energy performance of buildings and HVAC systems. Unfortunately this did not happened because the standardization work started too late and several countries have adopted national calculation methods. Also some of the standards do include alternative methods, which mean the energy performance of the same system may be evaluated differently in different countries.

Energy performance of heating systems

A basic standard for the calculation of the building energy demand (EN ISO 13790 [3]) will form the central point of the calculation procedure. To perform this calculation, input data for indoor climate requirements, internal loads, building properties and climatic conditions are needed. Standards and methods for these input data exist already to a great extent. The calculation of the building energy demand does not take into account the heating-cooling-ventilation system. The calculated building energy demand serves then as an input to the calculation of the system energy requirement.

The boundary between building and system is shown in Figure 1 for a heating system. The additional losses are calculated for heat emission, distribution, storage and generation. The auxiliary electrical energy needed for fans, pumps etc. will also be calculated. The effect of the control system is included in the building energy demand as well as the additional losses from the system due to sub-optimal control. The additional energy savings obtained with a whole building automation system (heating, cooling, ventilation, electrical appliances, light etc.) will be taken into account in a separate standard (EN 15232[4]).

Output from the calculation (Figure 1) will be the net energy (building energy demand) together with the required heating/cooling ventilation energy for the HVAC systems, including the auxiliary energy. Finally, the total delivered energy for the building/system can be calculated by adding the required energy for all the systems, including lighting. This will be converted to Primary Energy, taking into account renewable energy sources and national conversion factors. The calculation process comprises three basic points, which are, calculation of net energy (building energy demand), calculation of delivered energy (system energy demand), and conversion to primary energy. Delivered energy takes into account the losses coming from the heat emission, heat distribution and heat generation system.

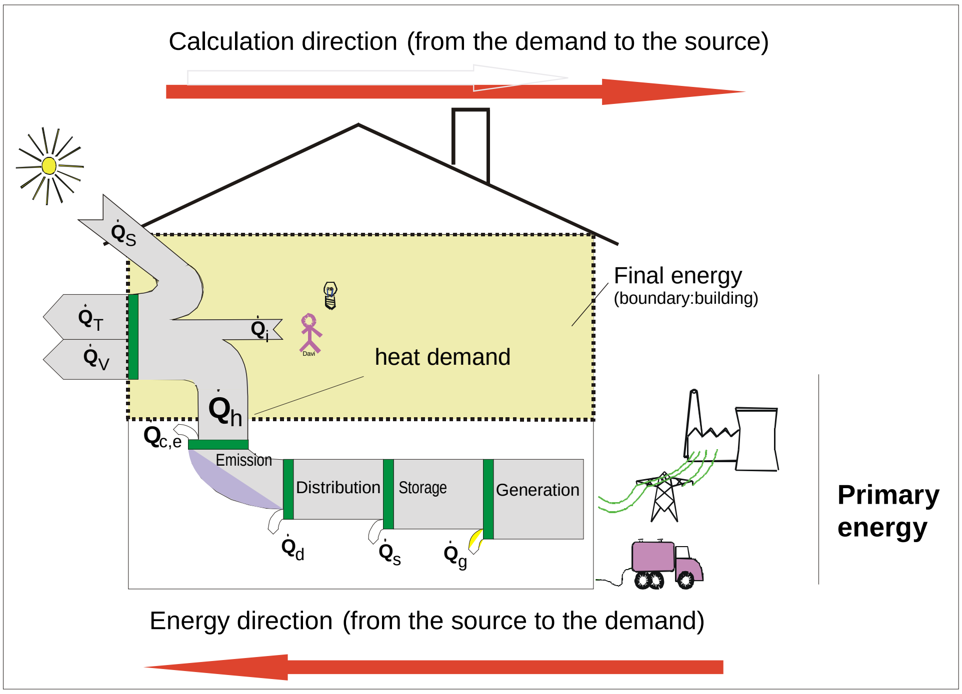

Figure 1. Calculation concept and building-system boundaries for heating (EN15316-1 [5])

Heat losses for the heat emission system (EN15316-2.1)

Emission losses are due to three factors, namely, non-uniform temperature distribution, losses to the outside from embedded heating devices in the structure, and losses due to non-perfect control of the indoor temperature. The heat energy losses of heat emission are calculated as:

$$Q_{em,ls} = Q_{em,str} + Q_{em,emb} + Q_{em,ctr} \qquad [J] \qquad (1)$$

where:

$Q_{em,str}$ heat loss due to non-uniform temperature distribution in Joule (J);
$Q_{em,emb}$ heat loss due to emitter position (e.g. embedded) in Joule (J);
$Q_{em,ctr}$ heat loss due to control of indoor temperature in Joule (J).

Two methods are recommended in the standard. The two methods do not give exactly the same results, but the same trend. The two methods shall not be mixed.

Method using efficiencies of the emission system

The evaluation of $Q_{\mathrm{em,ls}}$ takes place monthly or by another time period in accordance with equation (2).

$$Q_{\mathrm{em,ls}} = \left(\frac{f_{\mathrm{Radiant}} f_{\mathrm{int}} f_{\mathrm{hydr}}}{\eta_{\mathrm{em,ls}}} - 1 \right) Q_{\mathrm{H}} \qquad (2)$$

where

$Q_{\mathrm{em,ls}}$ is the additional loss of the heat emission (time period), in kWh;
Q_{H} is the net heating energy (time period) (EN ISO 13790), in kWh;
f_{hydr} is the factor for the hydraulic equilibrium.
f_{im} is the factor for intermittent operation (as intermittent operation is to be understood the time-dependent option for temperature reduction for each individual room space);
f_{rad} is the factor for the radiation effect (only relevant for radiant heating systems);
η_{em} is the total efficiency level for the heat emission in the room space.

The total efficiency level η_{em} is fundamentally evaluated as

$$\eta_{em} = \frac{1}{(4 - (\eta_{str} + \eta_{ctr} + \eta_{emb}))} \qquad (3)$$

where

η_{str} is the part efficiency level for a vertical air temperature profile;
η_{ctr} is the part efficiency level for room temperature control regulation;
η_{emb} is the part efficiency level for specific losses of the external components (embedded systems).

In individual application cases this breakdown is not required. The annual expenditure for the heat emission in the room space is calculated as

$$Q_{\mathrm{em,ls,a}} = \sum Q_{\mathrm{em,ls}} \qquad (4)$$

where

$Q_{\mathrm{em,ls,a}}$ is the annual loss of the heat emission, in kWh;
$Q_{\mathrm{em,ls}}$ is the loss of the heat emission (in the time period) in accordance with equation (2), in kWh.

Default values fro the different efficiencies and factors can be found in an informative annex to the standard. Some of these values are based on real data from experiments and/or computer simulations, while others are made by agreement. Examples of the values included in the annexes are given in table 1 to 3.

Method using equivalent increase in internal temperature

The internal temperature is increased by:

- The spatial variation due to the stratification, depending on the emitter;
- The control variation depending on the capacity of the control device to assure a homogeneous and constant temperature.

The equivalent internal temperature, $\theta_{\mathrm{int,inc}}$ taking into account the emitter, is calculated by:

$$\theta_{\mathrm{int},inc} = \theta_{\mathrm{int},ini} + \Delta\theta_{str} + \Delta\theta_{ctr} \qquad (°\mathrm{C}) \qquad (5)$$

where:

$\theta_{\mathrm{int,ini}}$ initial internal temperature (°C);

$\Delta\theta_{str}$ spatial variation of temperature;
$\Delta\ \theta_{ctr}$ control variation.

The influence of an equivalent increase in internal temperature. of the heat emission system may be calculated in two different ways:

- by multiplying the calculated building heat demand, Q_H, with a factor based on the ratio between the equivalent increase in internal temperature, $\Delta\theta_{int,inc}$, and the average temperature difference for the heating season between the indoor and outdoor temperature for the space:
  $Q_{em,ls} = Q_H \cdot (1 + \Delta\theta_{int,inc} / (\theta_{int,inc} - \theta_{e,avg}))$ [J] (6)

- by recalculation of the building heat energy requirements, according to EN ISO 13790, using the equivalent increased internal temperature. as the set point temperature of the conditioned zone. This second approach leads to a better accuracy.

For η_{str} an average value is to be formed from the data for the main influence parameters "over-temperature" and "specific heat losses via external components".

$\eta_{str} = (\eta_{str1} + \eta_{str2})/2$

Table 1. Efficiencies for free heating surfaces (radiators); room heights ≤4 m.

Influence parameters		Efficiencies			
		η_{str}		η_{ctr}	η_{emb}
Room space temperature regulation	unregulated, with central supply temperature regulation			0.80	
	Master room space			0.88	
	P-controller (2 K)			0.93	
	P-controller (1 K)			0.95	
	PI-controller			0.97	
	PI-controller (with optimisation function, e.g. presence management, adaptive controller)			0.99	
		η_{str1}	η_{str2}		
Over-temperature (reference Θ_i = 20 °C)	60 K (e.g. 90/70)	0.88			
	42.5 K (e.g. 70/55)	0.93			
	30 K (e.g. 55/45)	0.95			
specific heat losses via external components (GF = glass surface area)	radiator location internal wall		0.87		1
	radiator location external wall				
	- GF without radiation protection		0.83		1
	- GF with radiation protection[a]		0.88		1
	- normal external wall		0.95		1

[a] The radiation protection must prevent 80% of the radiation losses from the heating body to the glass surface area by means of insulation and/or reflection.

EXAMPLE: radiator external wall; over-temperature 42.5 K; P-controller (2 K)

$\eta_{str} = (\eta_{str1} + \eta_{str2})/2 = (0.93 + 0.95)/2 = 0.94$; $\eta_{ctr} = 0.93$; $\eta_{emb} = 1$

$\eta_{em} = 1/(4 - (0.94 + 0.93 + 1)) = 0.88$

Factor for intermittent operation f_{im} = 0.97
Factor for radiation effect: f_{rad} = 1.0

Table 2. Factor for hydraulic balancing: $f_{hydr.}$

	Influencing parameters	Factor for hydraulic balancing, f_{hydr}
Hydraulic balance	non balanced systems	1.03
	Signed balancing report and in compliance with EN 14336 • more than 8 emitters per automatic differential pressure control or only static balanced systems	1.02
	Signed balancing report and in compliance with EN 14336, • Max 8 emitters per automatic differential pressure control	1.00

For η_{emb} an average value is to be formed from the data for the main influence parameters "system" and "specific heat losses via laying surfaces".

$$\eta_{emb} = (\eta_{emb1} + \eta_{emb2})/2 \qquad \text{(A4)}$$

Table 3. Efficiencies for component integrated heating surfaces (panel heaters); room heights ≤4m.

influence parameters		**Part efficiencies**			
		η_{str}	η_{ctr}	η_{emb}	
Room space temperature regulation	Heat carrier medium water				
	- unregulated		0.75		
	- unregulated, with central supply temperature regulation		0.78		
	- unregulated with average value formation ($\vartheta_V - \vartheta_R$)		0.83		
	- Master room space		0.88		
	- two-step controller/P-controller		0.93		
	- PI-controller		0.95		
	Electrical heating				
	-two-step controller		0.91		
	- PI-controller		0.93		
System	Floor heating			η_{emb1}	η_{emb2}
	- wet system	1		0.93	
	- dry system	1		0.96	
	- dry system with low cover	1		0.98	
	Wall heating	0.96		0.93	
	Ceiling heating	0.93		0.93	
Specific heat losses via laying surfaces	Panel heating without minimum insulation in accordance with DIN EN 1264				0.86
	Panel heating with minimum insulation in accordance with DIN EN 1264				0.95
	Panel heating with 100% better insulation than required by DIN EN 1264				0.99

EXAMPLE: Floor heating - wet system (water); two-step controller; floor heating with high level of heat protection

$\eta_{str} = 1.0$; $\eta_{ctr} = 0.93$; $\eta_{emb} = (\eta_{emb1} + \eta_{emb2})/2 = (0.93 + 0.95)/2 = 0.94$

$\eta_{em} = 1/(4 - (1.0 + 0.93 + 0.94)) = 0.88$

Factor for intermittent operation:	f_{im}	= 0.98
Factor for radiation effect:	f_{rad}	= 1.0
Factor for hydraulic balancing:	f_{hydr}	same as for radiators

Heat losses for the heat distribution and generation system

The heat losses of a distribution system depend on the average temperature of the heating medium, the temperature of the surrounding envelope, length and insulation of the pipes. For the heat losses in a time step the following formula applies:

$$Q_D = \sum_i U' \cdot (\vartheta_m - \vartheta_a) \cdot L \cdot t_H \qquad \text{[J]} \qquad (10)$$

where:

U' U-value per length, [W/mK]
ϑ_m average medium temperature, [°C]
ϑ_a surrounding temperature, [°C]
L length of the pipe [m]
t_H heating hours in the time step, [h]

The standard EN15316-2.3 gives both approximated and detailed methods. The most important factors are the temperature level of the heating medium. Low temperature heating will result in decreased losses.

The calculation of losses from heat generation systems like boilers, heat pumps, CHP, district heating, solar heating and biomass combustion systems are considered in other standards. Also here a lower water temperature will result in increased energy efficiency.

Figure 2 show an example of the energy performance calculation for a one-family house for a radiator and a floor heating system. It can be seen that the additional losses from the heating system contributes with about 20% to the total primary energy.

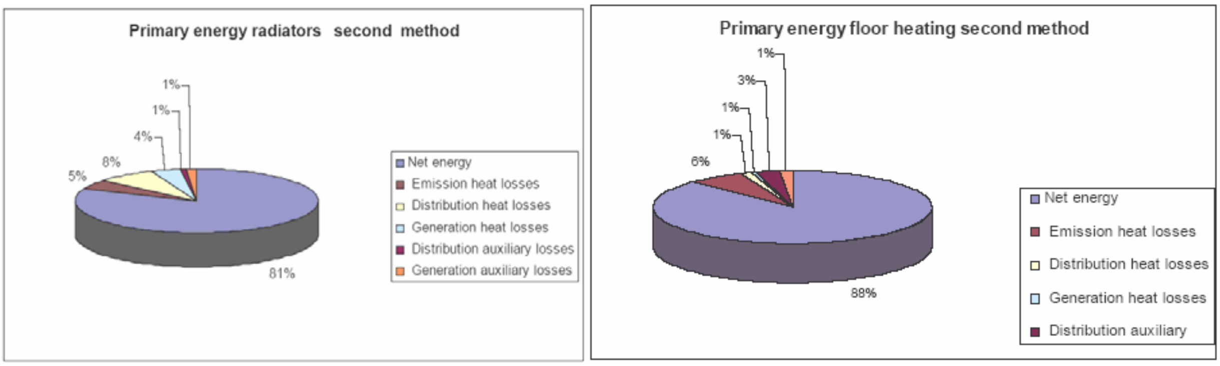

Figure 2. Losses in primary energy for a radiator and a floor heating system.

EMBEDDED WATER BASED SYSTEMS FOR LOW TEMPERATURE HEATING AND HIGH TEMPERATURE COOLING

In Europe it is mainly water-based heating systems that are used. These systems use radiators or floor heating as heat emitters. One advantage compared with air systems is the more efficient means of transporting energy. The demand for comfort, better insulation of buildings, and greater internal loads from people and equipment have increased interest in installing also a cooling system to keep indoor temperatures within the comfort range. This resulted first of all in the introduction of suspended ceiling panels for cooling and in recent years also in the use of floor systems for cooling (Simmonds et al. [8]; Olesen [9]). Typical positioning of pipes for wall, floor and ceiling systems is shown in Figure 3.

A new trend, which started in the early nineties in Switzerland (Meierhans [10]), is to use the thermal storage capacity of the concrete slabs between each storey in multi-storey buildings. Pipes carrying water for heating and cooling are embedded in the centre of the concrete slab (Figure 3).

By activating the building mass, you will not only get a direct heating-cooling effect, but you will also, due to the thermal mass, reduce the peak load and transfer some of the load outside the period of occupancy. Because these systems for cooling operate at a water temperature close to room temperature, they increase the efficiency of heat pumps, ground heat exchangers and other systems using renewable energy sources.

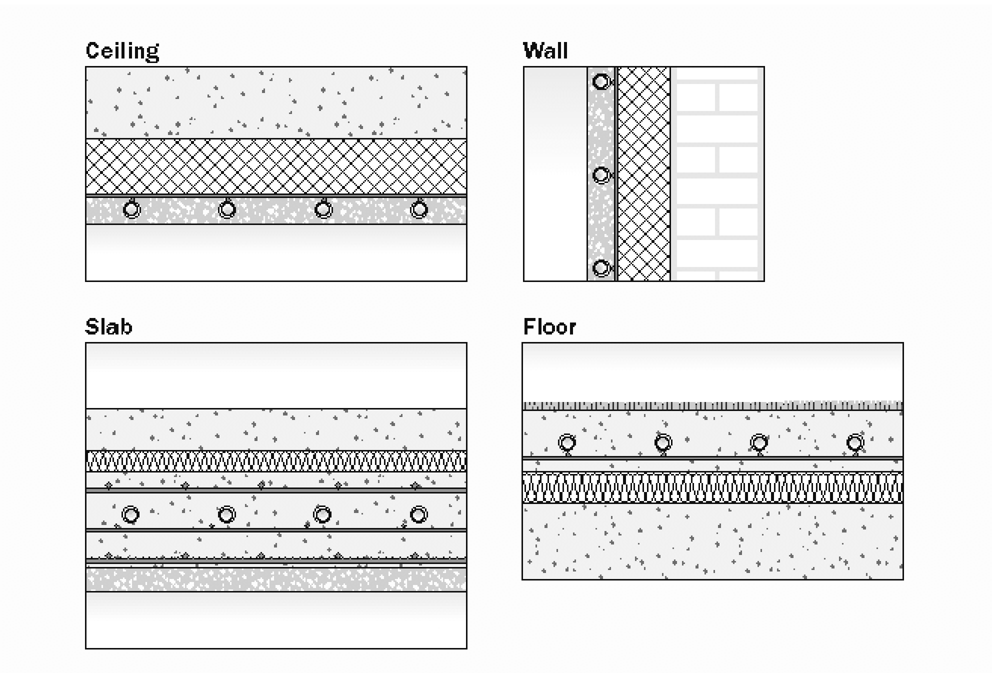

Figure 3. Examples of the positioning of pipes in floor, wall, ceiling and slab.

Even if surface heating and cooling systems often have a higher thermal mass than other heating/cooling systems, they have a high control performance. This is partly due to the small temperature difference between the room and the system (water, surface) and the resulting high degree of self-control. Studies on controllability of floor heating/cooling (Olesen [11]) show that floor heating control the room temperature as good as radiators. To avoid condensation on a cooled surface, there is a need to include a limitation on water temperature, based on the space dew-point temperature.

Design and dimensioning of these systems including calculation of heating and cooling capacity can be done according to the new standards EN15377-1 and 2. [12, 13]

Thermo Active Building Systems (TABS)

A Thermo-Active-Building-System (TABS) is a water based heating and cooling system, where the pipes are embedded in the central concrete core of a building construction. The heat transfer takes place between the water (pipes) and the concrete, between the concrete core and the surfaces to the room (ceiling, floor) and between the surfaces and the room.

The peak-shaving is the possibility to heat and cool the structures of the building during a period in which the occupants may be absent (during night time), reducing also the peak in the required power (Figure 4). In this way energy consumption may be reduced and lower night time electricity rate can be used. At the same time a reduction of the size of cooling system including chillier is possible.

The performance and dimensioning of TABS can be done by full dynamic building simulations with commercial programs including calculation models for embedded pipes. (Olesen and Dossi, [14]). A study was performed with the aid of the dynamic simulation program. The system considered is shown in Figures 5 and 6. The meteorological ambient boundary conditions correspond to those of Würzburg/Germany and Venice/Italy. The external temperature data for winter and summer design days are shown in Table 3. Summer was the period from 1 May to 30 September, and winter was the period from 1 October to 30 April

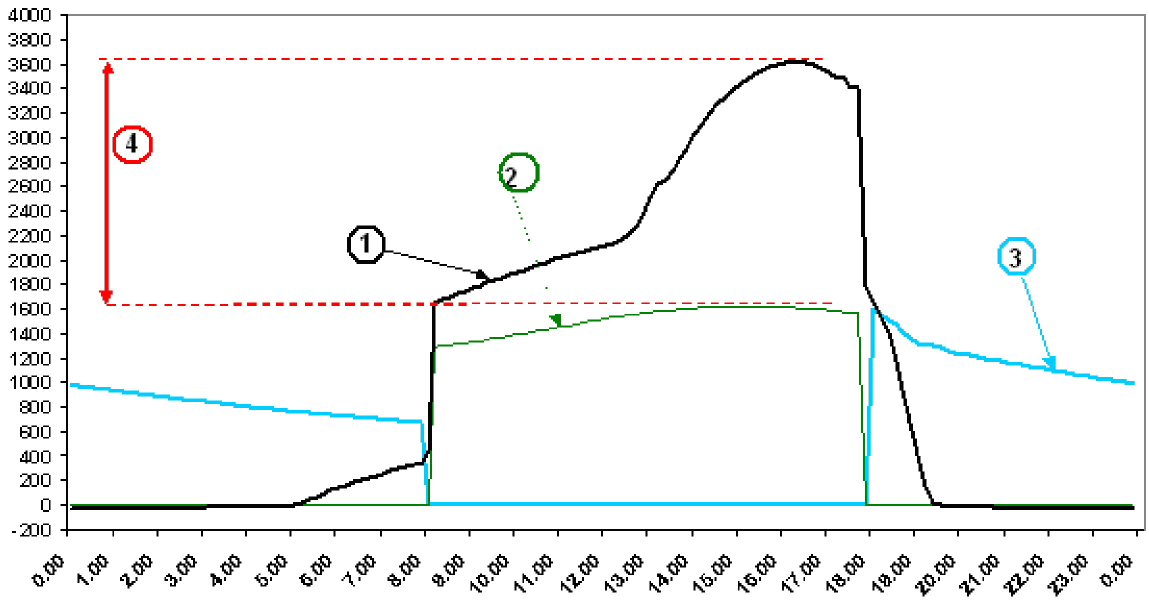

Figure 4. Example of peak-shaving effect (X-axes: time; y-axes: cooling power W)
1) heat gain, 2) power needed for conditioning the ventilation air, 3) power needed on the water side, 4) peak of the required power reduction

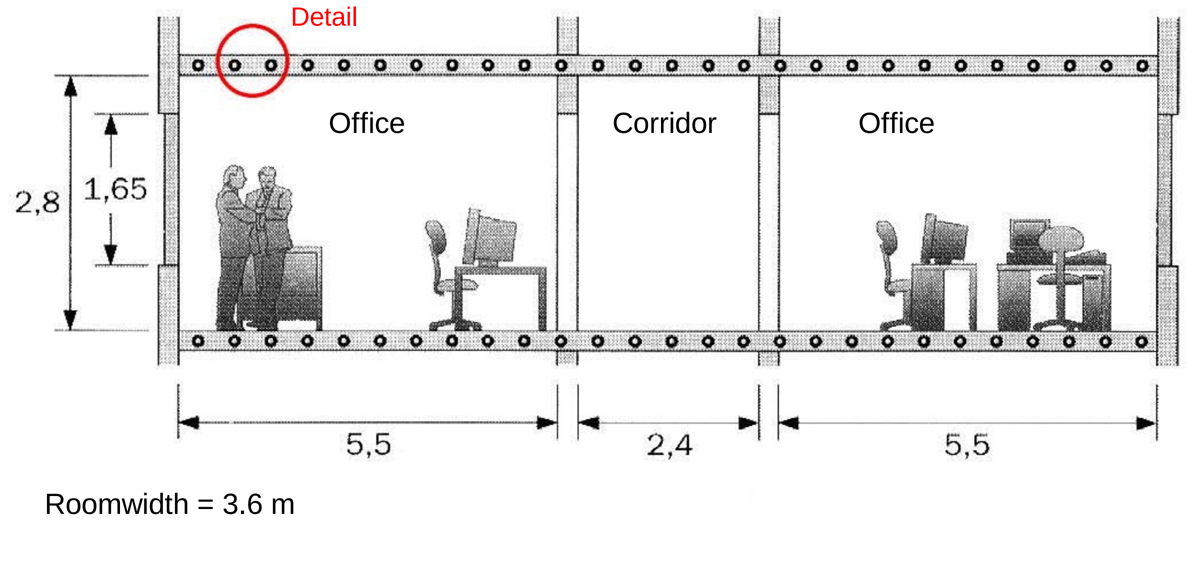

Figure 5. Central room module used for the computer simulation of a building with concrete slab cooling. All dimensions are in metres.

The time of occupancy was Monday to Friday from 8.00 to 17.00, with a lunch break from 12.00 to 13.00. The system was in operation only outside the period of occupancy, from 18:00 to 06:00. *Internal heat sources: d*uring occupied periods corresponding to 27.8 W/m^2. *Ventilation (ach*): outside time of occupation 0.3 h^{-1} (infiltration); during occupation 1.5 h^{-1} (~ 11 l/s per person). *Sun protection*: during occupation, by direct exposure of sunlight and operative temperature above 23°C, reduction factor z = 0.5.

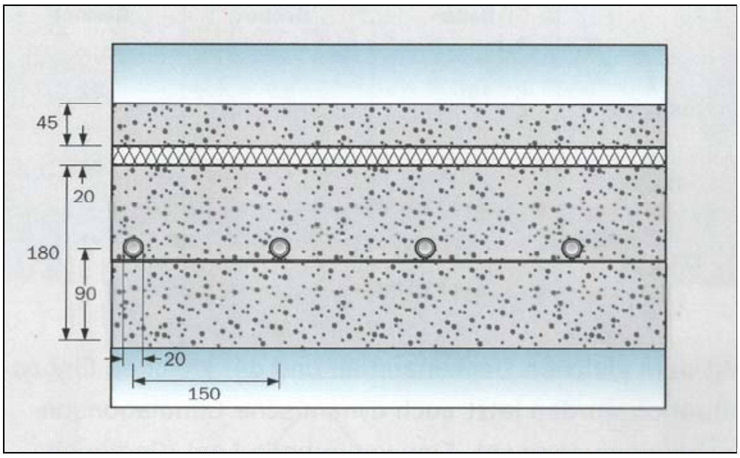

Figure 6. Position of the plastic pipes in the concrete slab between two storeys.

Table 4: Design day outdoor temperatures for Würzburg, Germany and Venice, Italy.

City	Lat. [°]	Long. [°]	Elev. [m]	Heating Dry Bulb [°C]		Cooling Dry Bulb [°C]	
				99,6%	99%	0,4%	2%
Venice	45.30 N	12.20 E	6	-4,9	-3,1	30,8	28,2
Würzburg-Frankfurt	50.05 N	8.60 E	113	-11	-8,2	30,3	26,7

The goal for the system used in the study was to operate water temperatures as close to the room temperature as possible. If very high or very low water temperatures are introduced into the system it may result in over-heating or under-cooling.

In the present study, the supply water temperature was controlled so that it was not lower than the dew point in the space. For this purpose, a humidity balance (latent loads from people, outside humidity gain from ventilation) was also included in the simulation. It was then possible to calculate the dew point in the room for each time step in the simulation.

Instead of controlling the supply water temperature it may be better to control the average water temperature. The return water temperatures are influenced by the room conditions. By maintaining a constant supply water temperature, an increase in internal loads from sun or internal heat sources will increase the return temperature. The average water temperature will then increase and the cooling potential will decrease. If, instead, the average water temperature ($\frac{1}{2}(t_{return} - t_{supply})$) is controlled, an increase in return temperature will automatically be compensated for by a decrease in supply water temperature.

In well designed buildings with low heating and cooling loads it may be possible to operate the system at a constant water temperature. The following concepts for water temperature control were studied:

Supply water temperature is a function of outside temperature according to the equation:

$$t_{supply} = 0{,}52 * (20 - t_{external}) + 20 - 1{,}6 * (t_{operative} - 22) \quad °C \qquad \text{(case 801)}$$

Average water temperature is a function of outside temperature according to:

$$t_{average} = 0{,}52 * (20 - t_{external}) + 20 - 1{,}6 * (t_{operative} - 22) \quad °C \qquad \text{(case 901)}$$

Average water temperature equal to: 22°C in summer and 25°C in winter. (case 1201)

Supply water temperature is a function of outside temperature according to the equation:

$$t_{supply} = 0{,}35 * (18 - t_{external}) + 18 \quad °C \quad \text{summer} \qquad \text{(case 1401)}$$

$$t_{supply} = 0{,}45 * (18 - t_{external}) + 18 \quad °C \quad \text{winter} \qquad \text{(case 1401)}$$

The results of the simulation are shown in Table 5 for summer conditions and in Table *6* for winter conditions. The operative temperature of the cases 0801, 0901 and 1401 (Table 5) is for most of the time (>85%) in a comfort range (22-26°C). In Würzburg, 27°C is never exceeded and 26°C is exceeded less than 5% of the time. In Venice, only 5% of the temperatures are above 27°C. The difference between controlling the supply water temperature (case 0801) or the average water temperature (case 0901) is very small. In the case of 1401, the control does not take into account the internal operative temperature, but the results are almost identical to cases 0801 and 0901. With a constant average water temperature (22°C), the cooling effect is too low and the operative temperature is often too high (60% of the time above 27°C in Venice and 27% in Würzburg). The energy use is the same for the cases 0801, 0901 and 1401 in Venice. For Würzburg, case 1401 is the energy use, but it is about 10% lower than case 801 and 901. Energy use in case 1201 with a constant water temperature is relatively high. The pump running time for case 1401 is equal to or lower than for the other cases. In the summer, case 1401 is overall better than the others. Due to the warmer climate in Venice (Table 5) the room temperatures are higher, and energy use and pump running time are also higher compared to Würzburg.

Table 5: Operative temperatures, temperature drift, pump running time and energy transfer for different water temperature control strategies. Summer conditions. Dead-band 22–23°C. Ventilation rate: 0.3 ach from 17:00 to 8:00, 1.5 ach from 8:00 to 17:00.

	May to September Time of operation 18:00-06:00								
		Venice				Würzburg			
Water temperature control		Supply = F (outside) 0801	Average = F (outside) 0901	Average = 22°C 1201	Average =F (outside) 1401	Supply= F (outside) 0801	Average =F (outside) 0901	Average = 22°C 1201	Average =F (outside) 1401
Operative temperature interval	°C	%	%	%	%	%	%	%	%
	<20	0	0	0	0	0	0	0	0
	20-22	0	0	0	0	3	3	1	5
	22-25	56	58	8	56	75	78	30	77
	25-26	26	25	13	25	18	16	21	14
	26-27	13	12	19	14	5	4	22	4
	>27	5	5	60	5	0	0	27	0
Pump running	hours	1254	1190	1417	1214	1091	971	1327	953
	% of time	34	32	39	33	30	26	36	26
Energy KWh	Cooling	1104	1109	1297	1106	763	785	978	749
	Heating	1	2	0	0	29	41	2	2

Standard EN15377-3 [15] includes more simplified methods to evaluate the dynamic performance of TABS. Besides the standard includes diagrams like the one shown in Figure 7 [16]. This simplified diagram give the relation between internal heat gains, water supply temperature, heat transfer on the room side, hours of operation and heat transfer on the water side. The diagrams correspond to a concrete slab with raised floor (R=0.45 m^2K/W) and a permissible room temperature range of 21 °C to 26 °C. The upper diagram shows on the y-axis the maximum permissible total heat gain in space (internal gains plus solar gains) W/m^2, and on the x-axis the required water supply temperature. The lines in the diagram correspond to different hours of operation (8h, 12h, 16h, and 24h) and different maximum amount of energy supplied per day Wh/m^2 d. The lower diagram shows the cooling power W/m^2 required on the water side (for dimensioning of chiller) for thermally activated slabs as a function of supply water temperature and operation time. Further, the amount of energy rejected per day is indicated $Wh/(m^2$ d). The example shows, that by a maximum internal heat gain of 38 W/m^2 and 8 hour operation, a supply water temperature of 18,2 °C is required. If, instead, the system is in operation for 12 hours, a supply water temperature of 19,3 °C is required. In total, the amount of energy rejected from the room is app. 335 Wh/m^2 per day. The required cooling power on the water side is by 8 hours operation 37 W/m^2 and by 12 hours operation only 25 W/m^2. Thus, by 12 hours operation, the chiller can be much smaller. The total heat rejection on the water side is app. 300 Wh/m^2 per day.

Table 6: Operative temperatures, temperature drift, pump running time and energy transfer for different water temperature control strategies. Winter conditions. Dead-band 22–23°C. Ventilation rate: 0.3 ach from 17:00 to 8:00, 1.5 ach from 8:00 to 17:00.

	October to April Time of operation 18:00-06:00								
		Venice				Würzburg			
Water temperature control		Supply= F (outside) 0801	Average= F (outside) 0901	Average= 25°C 1201	Average= (outside) 1401	Supply= F (outside) 0801	Average= F (outside) 0901	Average= 25°C 1201	Average= F (outside) 1401
Operative temperature interval	°C	%	%	%	%	%	%	%	%
	<20	0	0	0	1	0	0	4	4
	20-21	1	1	6	14	9	7	19	24
	21-23	72	75	50	63	77	80	50	63
	23-24	14	15	5	14	8	7	7	7
	24-26	12	10	23	8	6	5	15	2
	>26	0	0	16	0	0	0	5	0
Pump running	Hours	837	642	1487	1166	813	664	1533	1322
	% of time	16	13	29	23	16	13	30	26
Energy KWh	Cooling	144	144	143	143	57	64	63	45
	Heating	551	554	407	421	816	834	684	717

EXERGY ANALYSIS

"Energy saving" and emission reduction are both affected by the energy efficiency of the built environment and the quality of the energy carrier in relation to the required quality of the energy. Taking into account qualitative aspects of energy leads to introduction of the exergy concept in comparison of systems, which is the key idea of IEA-Annex 37. Energy, which is entirely convertible into other types of energy, is exergy (high valued energy such as electricity and mechanical workload). Energy, which has a very limited convertibility potential, such as heat close to room air temperature, is low valued energy. Low exergy heating and cooling systems use low valued energy, which is delivered by sustainable energy sources (e.g. by using heat pumps, solar collectors, either separate or linked to waste heat, energy storage etc.). Common energy carriers like fossil fuels deliver high valued energy. The reason for "energy saving" being in quotation marks in the first sentence, is that we actually are talking about saving exergy, not energy!

Future buildings should be planned to use or to be suited to use sustainable energy sources for heating and cooling. One characteristic of these energy sources is that only a relatively moderate temperature level can be reached, if reasonably efficient systems are desired. The development of low temperature heating and high temperature cooling systems is a necessary prerequisite for the usage of alternative energy sources. The basis for the needed energy supply is to provide occupants with a comfortable, clean and healthy environment.

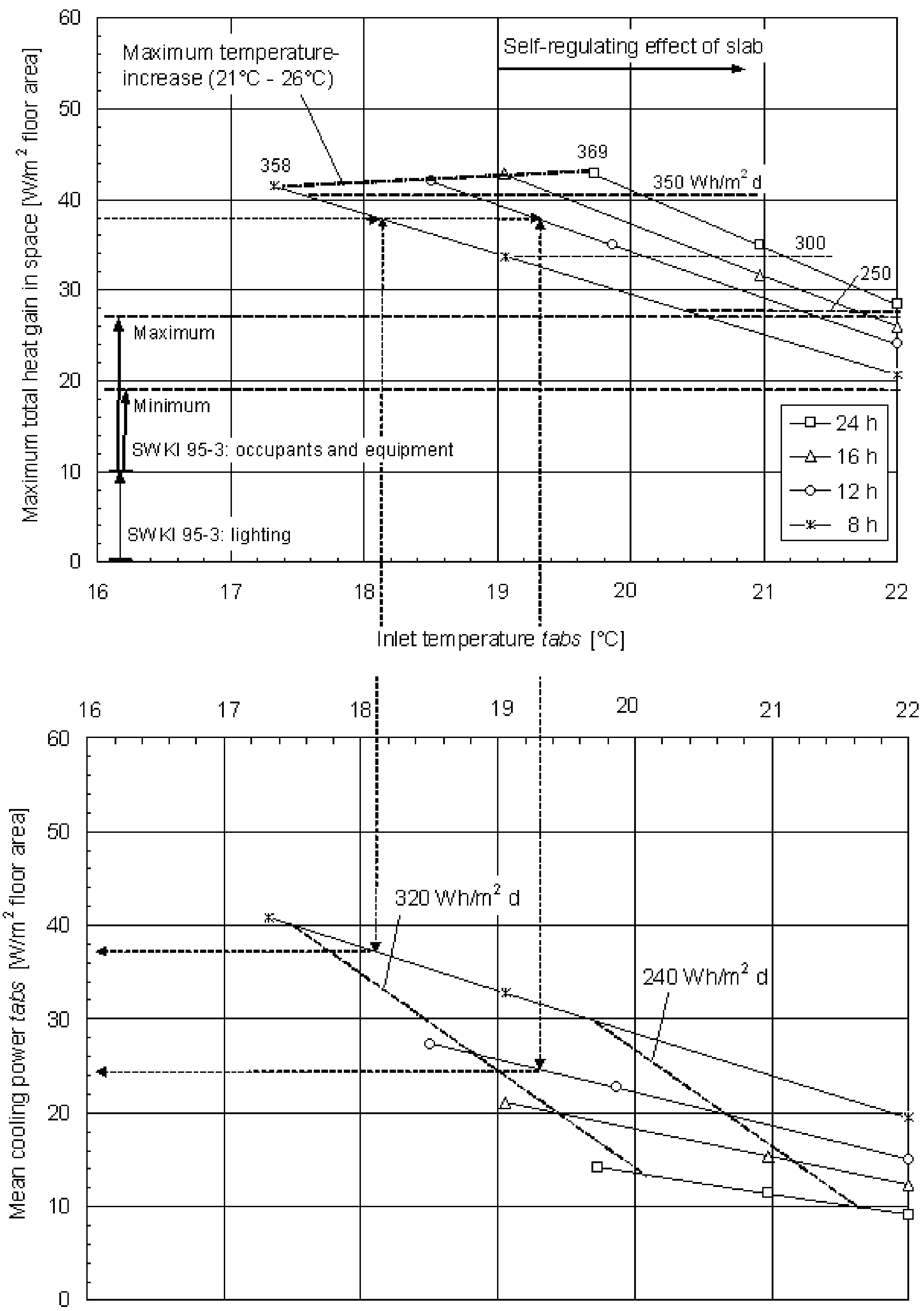

Figure 6 – Working principle of TABS (Koschenz and Lehmann [16].

Figure 7 is showing an example of a calculated energy flow and exergy flow from primary energy, heat generator, emission in room and loss to the outside. The figures show that a low temperature heating system will result in less exergy consumption.

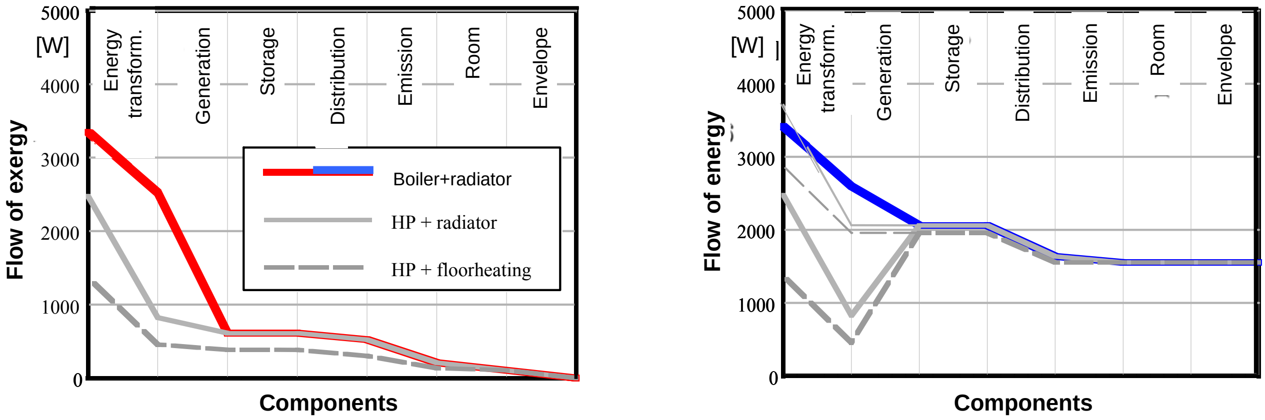

Figure 7. Examples of exergy and energy flow for three type of heating systems. (IEA-Annex 37).

DISCUSSION AND CONCLUSIONS

New demands for lowering the energy consumption of buildings require increased insulation of houses, tight constructions with mechanical ventilation and heat recovery. A trend for residential building is the concept of passive houses, where a “normal” heating system may not be needed. This trend will result in much smaller heating systems and in many cases systems using air as the energy transport medium. Also the global warming will result in less need for heating.
Because of the EPBD all European countries will in a couple of years have implemented a requirement for calculation or estimation (measurements/calculation) of the energy performance of new and existing buildings. With the development of the new CEN standards related to the EPBD it will in the future be possible to use common method and avoid that the same building system will be evaluated differently. While it is relative well established how to calculate the energy performance of the building alone, there is a lack of data on energy performance of HVAC systems. Especially for cooling systems the calculation methods are not well established. Many manufacturers will try to show that their system has a high energy performance and will also aim at developing more efficient systems and controls. But if the energy performance evaluation methods used are not detailed enough, new development may not be reflected in the calculations. On the other hand detailed calculations may make the methods more complicated. Another trend in Europe is the increasing need for cooling. Peoples increased comfort requirements (air conditioning in cars), increasing thermal loads in many office buildings, global warming, heat islands and the aim for optimal productivity of people, are all issues that will result in a higher demand for cooling.

In middle Europe a trend to use water based cooling systems at high water temperature is a way of making the cooling more sustainable with less energy consumption compared to full Air Conditioning. Some of theses systems also use the building mass (TABS) to reduce the peak cooling load and transfer some of the cooling from daytime to night. This trend has resulted in new CEN standards for design and dimensioning of systems and evaluation of the dynamic effect. Also new research studies are looking at the performance and optimal control of such systems.

Procedures for calculating the steady-state heating/cooling capacity are available. By a proper control the risk for condensation on the cooled surfaces can be limited. The results of a dynamic computer simulation of different control concepts for a water-based radiant cooling and heating system with pipes embedded in the concrete slabs have been presented. The system was studied for both the summer period May to September and the winter period October to April in two geographical locations, Venice, Italy and Würzburg, Germany. The best performance regarding comfort and energy is obtained by controlling the water temperature (supply or average) as a function of outdoor temperature. There is no need to take into account the room temperature. The system was able to keep the room temperatures within a comfortable range, in both summer (cooling) and winter (heating), and in both climatic zones.

Due to the use of water temperatures close to room temperatures water based surface heating and cooling systems will increase the possibility to use renewable energy sources like ground source heat pumps, ground heat exchangers, geothermal energy, solar energy, evaporative cooling etc. The level of water temperatures used also increase the efficiency of boilers, chillers and heat pumps.

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Contribution of the Eurovent Performance Certification programs into providing incentive to manufacturers to improve the energy efficiency of HVACR products

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SUMMARY

The Eurovent Certification company was established in 1994 and administers industry driven certification programs consisting of verification testing in a network of independent test laboratories of efficiency and capacity ratings of HVACR products against published data in manufacturer's sales catalogues.

A Certified Products Directory is available to consultants, architects and equipment specifiers for evaluation and selection of equipment, providing assurance that the products will perform as indicated in the directory.

Nowadays, nine groups of air-conditioning and refrigeration related products, are verification tested and the latest list of participating manufacturers, has grown impressively to more than 180 manufacturers.

As energy efficiency became one of the most important issues for the HVACR industry, the future trend for the Eurovent Certification Programs is to evolve from the original set-up of providing a fair competition level ground for manufacturers and increasing customer's confidence towards an additional instrument of providing incentive for manufacturers to improve the energy efficiency ratio of HVACR products.

This paper will shed a light on the certification programs and how the future trends in Eurovent Products Certification ,with regard to Energy Efficiency, are dealt with.

INTRODUCTION

Eurovent is the European Committee of Air Handling and Refrigeration Equipment Industries. The Committee was established in 1959 and has amongst its members 15 national associations active in the field of HVACR industry.

List of member associations in Eurovent:

Country	National Association	National Association
Belgium	AGORIA	
Denmark	DANSK Ventilation	
France	UNICLIMA	
Germany	FV Alt im VDMA	
Italy	ANIMA	CO.AER
Finland	FAMBSI	
Netherlands	FKL	
Norway	NVEF	
Slovenia	HVAC Cluster	
Spain	AFEC	ANEFRYC
Sweden	KTG	SWEDVENT
Turkey	ISKID	

The Committee represents more than 800 companies , mainly in the manufacturing industry and producing various products to be applied in Air Conditioning, Refrigeration, Air Handling and Ventilation as well as auxiliary refrigeration and air distribution component manufacturers. The estimated turnover of the companies is in excess of 20 billion €.

The essential activity is organised through working groups where manufacturers of a particular type of products meet together, examine relevant issues and decide actions to undertake. Working groups for fans, liquid chilling packages, air conditioners, fan coil units, air handling units, air filters, refrigerated display cabinets are particularly active having usually several meetings a year. A number of application documents, guides, recommendations and other agreements have been finalised and published.

A very important feature of Eurovent organisation is the existence of the certification programmes. Eurovent Certification Company was established in 1994 on a voluntary basis by the manufacturers of HVACR equipment. The performance characteristics of HVACR products are checked by at random sampling testing of products and equipment in independent laboratories. Eurovent certification covers a large part of relevant products and more than 180 manufacturers participate in one or more programmes. Checking data claimed in catalogues is extremely important as it was shown by testing in independent laboratories that there are always actors on the market not respecting clarity and honesty in data presentation. For instance a decision on minimum efficiency for particular equipment if not supported by certification will not be applied by all manufacturers and distributors. Only regular checking by an organised system will guarantee the performance of the application.

As all products in the scope of Eurovent consume energy – mainly in the form of electricity, the energy efficiency – intrinsic to the products but also related to use and installation – has become one of the most important issues for our industry. The implementation of the Kyoto protocol has now a high priority for the European Union and strong measures shall be applied to achieve its targets fixed for Europe as a reduction of 8 % of equivalent CO_2 emission in period 2008 – 2012.

In almost all Eurovent working groups the issue of energy efficiency has been discussed during the last several years. There is a general agreement that industry must be proactive and propose relevant actions in advance, before some mandatory measures are being decided by the European Commission or National Authorities. That would be the best way to help the industry to meet this important challenge.

EUROVENT CERTIFICATION – INCREASING INTEGRITY [1]

Eurovent Certification administers and promotes the Product performance and energy efficiency certification by means of Certification Programmes.

Existing Certification Programmes:

Air Conditioners up to 12kW cooling capacity
Air Conditioners from 12kW to 45kW cooling capacity
Air Conditioners from 45kW to 100kW cooling capacity
Close control Air Conditioners for computer rooms
Fan Coil units
Liquid Chilling Packages up to 750kW
Air Coolers for Refrigeration
Air Cooled refrigerant Condensers
Dry Coolers
Cooling Towers
Cooling and Heating Coils
Air Handling Units
Refrigerated Display Cabinets
Air to Air Plate and Rotary Heat Exchangers
Filters
Chilled Beams

Eurovent Certification maintains and publishes an updated list of certified Products in the Certified Products Directory and on the Eurovent Certification Website (www.eurovent-certification.com).

The system of Product Certification started in 1994 on a voluntary basis . The performance characteristics of HVACR products is checked by at random sampling testing of products and equipment in independent laboratories.
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The total number of participating companies today amounts to 185 companies.

Major issues besides the data checks of actual products versus the published data in sales and promotion catalogues are the Energy labelling implementation beyond what is provided for in the existing European directive (for air conditioners with a cooling capacity up to 12kw) and the determination of minimum energy efficiency levels below which certain products are not eligible to carry the logo "Eurovent Certified Performance".

A Certification Manual defines the organisation and general rules of Eurovent Certification and gives principles to follow in each programme. It must be emphasised that the Manual is regularly updated in order to keep the integrity at the highest possible level.

EUROVENT CERTIFICATION

General Principles

The following general principles are used in all certification programmes. The staff of the Certification Company must check their application and inform the Compliance Programme and Policy Committee (CPPC)of any deviation.

- Eurovent Certification Programmes are open to all manufacturers of relevant products – European and non-European - without any discrimination.
- Eurovent Certification is intended for European market and only products that could be sold in Europe* may be certified.
- Products produced by European or non-European manufacturers for other markets and which could not be sold in Europe (for instance 60 Hz units, non-authorised refrigerants etc.) could not be certified.
- Products which could be sold in Europe but are intended for other markets must comply with specific rules applied in Europe (for instance refrigerants, energy labelling for air conditioners and minimum efficiency).
- Conformity with safety directives (LVD, EMC, MD, PED) shall not be part of certification but will be required if applicable. However independent laboratories shall be asked to inform participants and Eurovent on possible irregularity .If non-conformity is confirmed the certification shall not be granted.

  *Europe represents members of European Union, and associated countries.

EUROVENT CERTIFICATION

Certify-All

The scope of each certification programme is clearly defined. Only products that may be tested by an independent laboratory either inside the laboratory or at participants' facilities shall be included. If for some large equipment there is no available laboratory above some defined size, the scope shall be limited to smaller units.
For some products like Air Handling Units the selection software is certified and the smallest size of the range must be such that the testing in an independent laboratory is possible.

All products of the relevant certification programme manufactured or sold by a participant inside the defined scope must be certified. When applicable, "certify-all" procedure means at least "all products inside the defined scope presented on the European market", but each Compliance Committee may implement larger applications.

"Certify-all" brings clarity and transparency and therefore increases the value of the whole system. If for some products due to their nature it is difficult to define a scope satisfying all participants the approval of the CPPC must be asked.

EUROVENT CERTIFICATION

Independent Laboratories

For each certification programme only one independent laboratory shall be preferably selected. The laboratory must be independent with no connection to any participant in the programme. Accreditation in accordance with the EN 45001 shall be requested. If for some certified characteristics this accreditation is missing, Eurovent staff shall organise the verification of technical competence of the concerned laboratory by a competent expert. An audit following the procedures given in ISO 17025 shall be performed.

The independent laboratory shall be regularly inspected by the technical staff of the Certification Company, following a schedule decided by the Compliance Committee.

If more than one laboratory is selected for a same type of products, the regular comparison testing shall be organised in order to assure that all laboratories provide similar results. The Compliance Committee may modify selection of laboratories if necessary.

List of Independent Laboratories involved in the performance testing:

TEST LABORATORY	CITY	COUNTRY
CEIS	Madrid	SPAIN
CETIAT	Lyon	FRANCE
DMT	Essen	GERMANY
HTA	Luzern	SWITZERLAND
IMQ	Amaro/Milano	ITALY
SP	Boras	SWEDEN
TUV	Essen/Munich	GERMANY
VTT	Espoo	FINLAND
WSPLab	Stuttgart	GERMANY

EUROVENT WORKING GROUPS ACTIVITIES ON ENERGY EFFICIENCY

WG 6B "Air Conditioners" [2]

The WG 6B "Air Conditioners" has been directly involved in the SAVE project EERAC concerning energy efficiency of room air conditioners. This project was finalised at the end of 1999. Using the data provided by manufacturers and available from Eurovent Directory of Certified Products, it has been observed that energy efficiency of air conditioners presented on the market varies widely: the best units have sometimes two times higher efficiency than the worst.

Two possible policies have been analysed in detail: energy labelling and minimum efficiency.

Energy Labelling as applied with success to many home appliances like domestic refrigerators was an obvious measure. However the labelling itself means only a clear information to buyers – it is expected that buyers will prefer better equipment and that in this way the global, average efficiency of the products sold on the market will increase. Labelling Directive for small Air Conditioners has been prepared by the European Commission and was published in April 2002. The WG 6B took a very active part examining drafts, proposing modifications and clarifications to the Labelling Directive.

Mandatory minimum efficiency has been introduced in many countries outside of Europe. This is the simplest way to eliminate low efficiency products from the market but it has a very important effect on the industry and must be applied carefully. In order to avoid such regulation the WG 6B has prepared a proposal on voluntary minimum efficiency to be supported by Eurovent certification. After many discussions the proposal received overwhelming approval from individual manufacturers and national associations.

The first practical action was applied on 1 January 2004 when air conditioners under 12 kW cooling capacity having the lowest classification (class G) were eliminated from the certification programme and, as the certify-all procedure must be applied, from the market. The effect of these measures can be seen on following table .Units with lowest efficiency (Class G) were eliminated and the minimum value is now 2.21. Manufacturers are presenting units with higher efficiency and the average energy efficiency of all air conditioners is increasing.

	EER (min)	EER (max)	EER (mean)	N0. of Units
2001	1.64	3.63	2.55	2597
2002	1.76	3.97	2.55	3251
2003	1.75	3.85	2.58	3078
2004	2.20	4.55	2.68	2081
2005	2.21	4.64	2.87	3502
2006	2.21	5.51	2.88	2396

The next step to eliminate classes F and E has been scheduled to be implemented in 2008. A proposal in connection with the Energy using Product Directive has been prepared by the WG6B in June 2005.

WG 6A "Chillers" [3]

The WG 6A "Chillers" has been examining the energy efficiency issues for liquid chilling packages – involved products cover capacity range from few kW to several MW. Several possible actions have been examined. In the certification programme, it was already decided that the energy efficiency (EER and COP) replace the power input as the certified characteristics. A voluntary classification system has been implemented although for such large equipment no physical labels will be fixed. However product classes will increase awareness of users that the energy efficiency is an important issue but also the manufacturers when they design new equipment. With a classification it may be easier to eliminate, like for air conditioners, the low efficiency products. The effect on the market has already been identified as some users simplify their request asking Class A products.

The use of chillers and the real annual energy consumption has been largely discussed. As the part load efficiency has a very strong impact on consumption, in the SAVE study EECCAC – (Energy Efficiency and Certification of Central Air Conditioners) - this issue has been treated exhaustively. This study was initiated by Eurovent/Cecomaf and manufacturers are actively participating. An integrated energy efficiency index called ESEER (European Seasonal Energy Efficiency Ratio) similar to IPLV (Integrated Part Load Value) developed in the USA and used by ARI, was established for the European conditions. The study was finalised in 2004 and experimental application started in 2005. Part load certification was implemented in the Eurovent certification in 2006.

From the certified part load table, Eurovent will compute ESEER allowing the comparison of chillers performances in the cooling mode. This global single figure shall be published in Eurovent directory together with cooling capacity and power input for standard conditions at full load. At present, the Energy Labelling Directive is restricted to household appliances. Indeed, the label is mandatory only for Room Air Conditioners with capacity equal to or lower than 12 kW. However, Eurovent WG6A established classification for full load Energy Efficiency Ratio of each type of chillers. The classification follows the A to G approach used in the European Energy Label for household appliances but the limits between classes have been defined for the existing chillers as listed in Eurovent directory, see Table1 for cooling mode.

EER Class A	Air Cooled EER≥3.1	Water cooled EER≥5.05	Remote condenser >=3,55
B	2,9≤ EER< 3,1	4,65≤ EER<5,05	3,4≤ EER<3,55
C	2,7≤ EER< 2,9	4,25≤ EER<4,65	3,25≤ EER<3,4
D	2,5≤ EER< 2,7	3,85≤ EER<4,25	3,1≤ EER<3,25
E	2,3≤ EER< 2,5	3,45≤ EER<3,85	2,95≤ EER<3,1
F	2,1≤ EER< 2,3	3,05≤ EER<3,45	2,8≤ EER<2,95
G	<2,1	<3,05	<2,8

Next steps will be the elimination of Class G chillers (poorly efficient) from Certification and reviewing the classification of chillers according to ESEER instead of EER.

WG 6C "Air Handling Units" [4]

The WG 6C "Air Handling Units" after finalising the Life Cycle Cost project has realised that the most important part of this cost is the energy consumption. Although other factors, like maintenance, are also very important, the best means to decrease the life cycle cost of an air handling units is to design and operate in the optimum way with respect to energy consumption.

A Eurovent Document 6/8 called "Recommendations for calculations of energy consumption for Air Handling Units" has been prepared. This document provides the detailed analysis of all components: fans, cooling and heating coils, heat recovery units. The document 6/8 was published in May 2005.

WG 4B "Air Filters"

The Eurovent WG 4 B "Air Filters " has found, during the work on Life Cycle Cost project, that energy consumption represents the largest part of the cost. Although the filters are passive elements and do not consume energy directly, the pressure drop they introduce in the flow is related to fan consumption. Selection of filters and their replacement should be made taking into account not only their primary purpose (cleaning the air) but also the possible energy saving. The relevant Eurovent Recommendation concerning calculation of Life cycle Cost of Air Filters was revised in 2004.

WG14 "Refrigerated Display Cabinets"

The WG 14 "Refrigerated Display Cabinets" has been working on annual energy consumption. The Group has been very much involved in the action undertaken by public authorities in the United Kingdom where Market Transformation Programme is aimed to increase the use of more efficient products. In order to implement the Eurovent Certification as the single data base for all Europe it will be necessary to highlight more efficient display cabinets. In addition, the users need to estimate the cost of annual energy. Conventional formulas have been presented, but more study will be necessary to achieve a valid method.

The energy efficiency appears now everywhere, for almost all products and under different aspects. In standardisation, the valid methods of testing have to be established. The certification will be more and more used not only to check the validity of claimed data but also to remove low efficiency products from the market.

The studies of Life Cycle Cost almost always shows that the energy consumption represent the highest part of this cost; even for passive elements like filters, a small decrease of pressure drop represents a huge economy of energy on the life cycle basis.

Eurovent/Cecomaf continues to show its commitment to energy efficiency improvement and tries to propose practical and easily verifiable measures.

Conclusion

Undeniably Eurovent Certification has greatly contributed in providing ground for fair competition and in delivering a framework for assuring to the users of HVACR products the cooling / heating performances to be expected from the products applied.
Equally the energy input at specified conditions could be certified.
Gradually in the last few years, when energy efficiency became a more hot topic in the general media in relation to global warming, the working groups within Eurovent took and still take important steps to distinguish further the more energy efficient products and determining minimum levels of energy efficiency, below which, products cannot be certified and thus cannot be listed in the Directory of Certified performance or the website of Eurovent Certification. Such orientation can only help the markets in applying more energy efficient products to the benefit of society as a whole and it's well being indoors.

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11 June 2007 at 10:30 - 12:30

A01
Indoor environment, health and productivity

Productivity, Energy, and Economics in Modern Offices

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SUMMARY

To promote the effort for energy conservation, it is also important to estimate the indoor environmental quality (IEQ) from the aspect of office worker's productivity. A conceptual diagram for the evaluation of the effect of IEQ on productivity was proposed. The results from subjective experiments of short term showed that human responses, such as fatigue, cerebral blood flow, sensation to the environment and SBS symptoms, were important for the evaluation of productivity. The results from a subjective experiment of long term exposure revealed the relationship that performance decreased with the increase in the level of fatigue. The results from another subjective experiment showed that the subjects performed well when they were satisfied with the IEQ. A survey in call-center showed the decline in performance in warmer environment. Economical and global environmental impacts from the modification of cooling temperature settings in office were also discussed.

INTRODUCTION

Productivity in general is defined as the quotient of Output over Input. The economic index for expressing the cost effectiveness of the improvement of IEQ can be proposed by dividing the benefits (Output) obtained through the improvement of IEQ by the cost factors (Input) for the improvement. Input-side cost factors, e.g. expenditure of the facility investment, are rather simple to estimate. Output-side benefits, on the other hand, were calculated by converting the quantified performance changes in terms of salaries [1], [2]. However, the output-side factors, e.g. the performance of the workers, are difficult to quantify in most cases, because the factors include intellectual activities, which can be influenced by psychological factors such as arousal level and motivation, as pointed out by McIntyre [3]. From the results of our laboratory experiments [4], [5], the effects of indoor environment were observed in human responses, such as fatigue, cerebral blood flow, sensation to the environment and SBS symptoms, even when there was no significant change in the performance. The performance would eventually decrease, in hypothesis, when workers were tired or dissatisfied with the environment.

A conceptual diagram for the evaluation of the effect of IEQ on productivity is proposed as shown in Figure 1. As indicated above, human responses are considered to be the intermediate factors affecting the performance of the workers by the indoor environment, and hence included in the conceptual diagram. In this paper, several evidences from subjective experiments and a field survey to explain the diagram. Also, an example of the evaluation of productivity was shown on the modification of cooling temperature settings in a typical office in Japan.

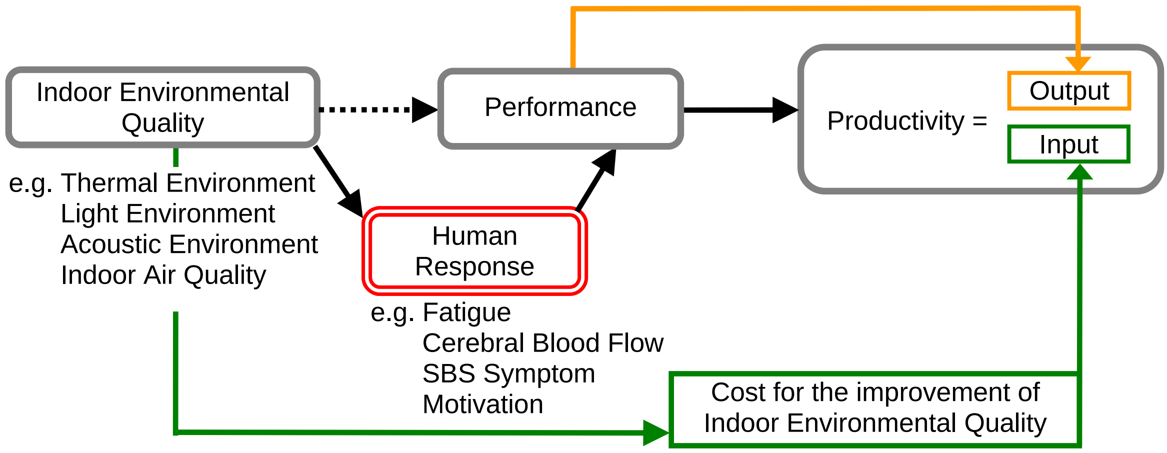


Figure 1. A conceptual diagram for the evaluation of the effect of indoor environmental quality on productivity

EFFECT OF HIGH TEMPERATURE ON TASK PERFORMANCE AND FATIGUE (SHORT EXPOSURE)

Method

To study the effect of moderately high temperature on task performance and fatigue, a subjective experiment was conducted in a climate chamber [6]. College-age subjects, 20 males and 20 females, participated in the experiment. The chamber was conditioned at operative temperatures of 25.5°C, 28.0°C, and 33.0°C with still air. Previous to these three conditions, a practice session at an operative temperature of 25.5°C was conducted. Relative humidity was controlled around 50%. Subjects wore a uniform with an insulation value of 0.76 clo. Task performance tests on computer were conducted for 1.5 hours.

The experimental procedure is shown in Figure 2. After entering the climate chamber, subjects waited in a sedentary position for 30 minutes, and then reported their first thermal sensation in the chamber and their feeling of fatigue. Four tests were carried out: the addition test for 10 minutes, the positioning test for 5 minutes, the text typing test for 5 minutes, and the Walter Reed Performance Assessment Battery test (PAB) [7] for about 15 minutes. After each test, an intermission of 5 minutes was taken and the subjects reported their thermal sensation, their feeling of fatigue, and their evaluation of the task load. Subjects filled in the sheets for evaluation of subjective symptoms of fatigue 30 minutes after entering the climate chamber as "before task" and after all computer tasks were finished, as "after task".

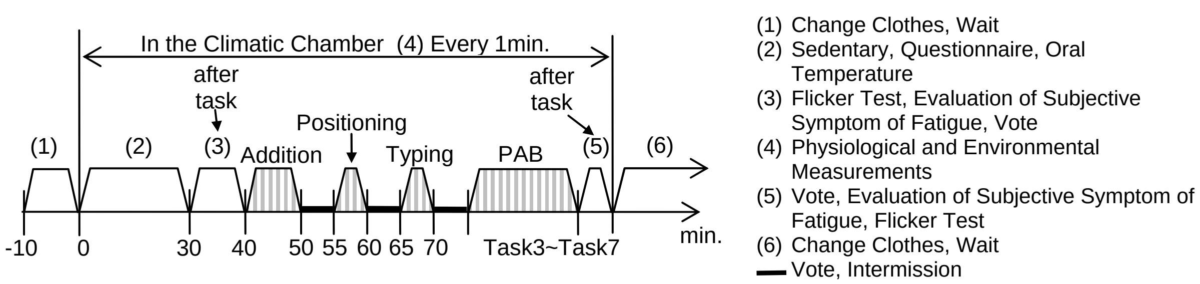


Figure 2. Experimental procedure

Task Performance

For female subjects, there was no significant difference in the performance of all computer tasks under the environmental conditions. For male subjects, there was no significant difference in the performance of the addition test and the positioning test under the environmental conditions. The performance of addition task is shown in Figure 3. In this study, the effects of thermal environment on task performance were contradictory among the task types as in previous findings [8], [9]. It is difficult to evaluate the effect of thermal environment on productivity by measuring only task performance.

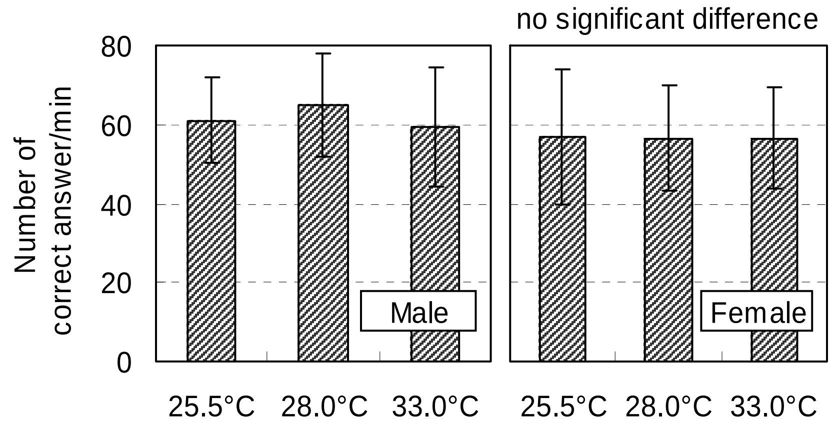


Figure 3. Performance of addition task

Evaluation of Subjective Symptoms of Fatigue and Cerebral Blood Flow

To evaluate the feeling of fatigue, subjects filled in the sheets of "Evaluation of Subjective Symptoms of Fatigue" [10]. It consists of three categories; I-group consists of 10 terms about "drowsiness and dullness", II-group consists of 10 terms about "difficulty in concentration", and III-group consists of 10 terms about "projection of physical disintegration". The subjects complained of a feeling of mental fatigue and complained the most just being in the room with operative temperature of 33.0°C.

Previous studies reported that increase of ΔO_2Hb and Δtotal Hb and decrease of ΔHHb were the typical findings by NIRS during the brain activation and mental work [11], [12]. In another experiment, it was showed that there was no significant difference in task performances between 26.0°C and 33.5°C conditions, but the increase of ΔtotalHb was higher at 33.5°C than that at 26.0°C [13]. Hot environments required more cerebral blood flow to maintain the same level of task performance.

EFFECT OF HIGH TEMPERATURE ON TASK PERFORMANCE AND FATIGUE (LONG EXPOSURE)

Method

To study the relationship between the task performance decrement and the work time, six hours of subjective experiment was conducted in a climatic chamber [14]. Fifteen college-age male subjects participated in this experiment. The conditions were set by the combination of operative temperature in the chamber and the amount of clothing insulations as: 25.0°C with 1.0 clo; 28.0°C with 1.0 clo; 28.0°C with 0.7 clo. Previous to these experimental conditions, a practice at 25.0°C with 1.0 clo condition was conducted.

The experimental procedure is shown in Figure 4. After entering the chamber, the subjects waited for 30 minutes in sedentary position, and then reported their first thermal sensation in the chamber and their feeling of fatigue. Subjects were then assigned nine sessions of addition tasks. In each session, the subjects worked for 30 minutes on the task and they reported thermal sensation, their feeling of fatigue, their evaluation of the task load in 5 minutes.

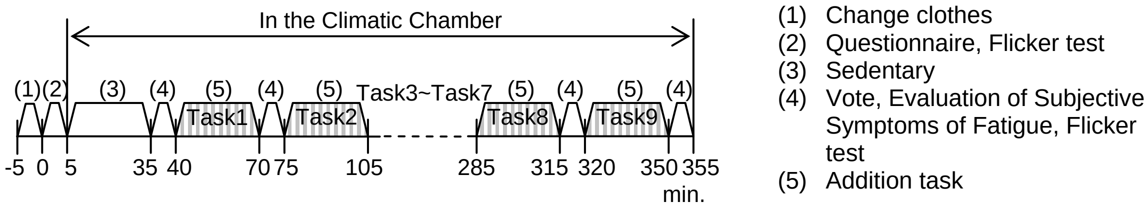


Figure 4. Experimental procedure

Task Performance

Figure 5 shows the result of normalized performance. The normalized performance did not changed significantly at 25.0°C with 1.0 clo condition over the time. However, the performance was significantly lower after sixth session (each $p<0.05$) at 28.0°C with 1.0 clo condition. Also, the performance was significantly lower after the third session (each $p<0.05$) at 28.0°C with 0.7 clo.

Fatigue and Performance

To determine the relationship between the level of fatigue and performance, the results of "Evaluation of Subjective Symptoms of Fatigue" were analyzed. Personal rates of complaints of fatigue, which were the general rates of complaints of fatigue for individual in each environmental condition excluding the results of practice session, were calculated. Figure 6 shows the correlation of personal rate of complaints of fatigue and normalized performance. It showed strong relationship that performance decreased with the increase in the level of fatigue. The correlation coefficient was -0.76. From the regression equation obtained from the results, increase in 10% of fatigue corresponds to the decrement in performance by 1.7%. The result showed the link from the fatigue of the occupants to the performance of them. It implies that evaluating fatigue is useful for estimating the effect of indoor environment on performance in offices and in experiments.

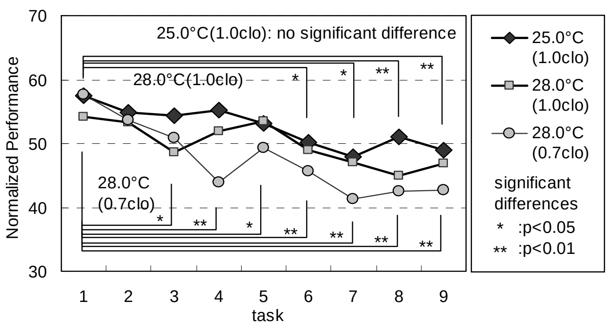


Figure 5. Result of normalized performance

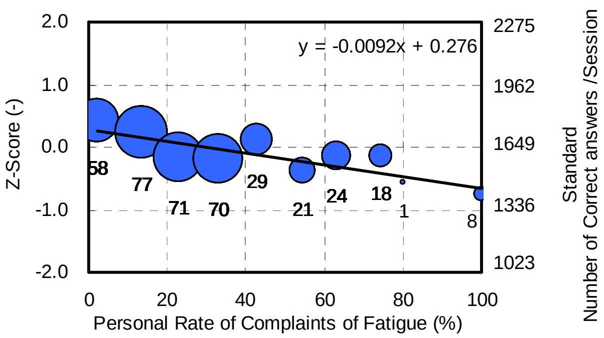


Figure 6. Correlation of the personal rate of complaints of fatigue and normalized performance

EFFECTS OF SATISFACTION WITH IEQ ON PERFORMANCE

Method

A subjective experiment was conducted in a climatic chamber to evaluate the effects of improvement of IEQ on occupant's performance [15]. The chamber was conditioned with the combination of: operative temperatures of 25°C or 28°C, illuminance of 750lx or 400lx and with or without traffic noise. Eight conditions were named as "Control(28°C, 400lx, with traffic noise)", "T(25°C, 400lx, with traffic noise)", "N(28°C, 400lx, without traffic noise)", "L(28°C, 750lx, with traffic noise)", "TN(25°C, 400lx, without traffic noise)", "TL(25°C, 750lx, with traffic noise)", "NL(28°C, 750lx, without traffic noise)" and "TNL(25°C, 750lx, without traffic noise)".

The experimental procedure is shown in Figure 7. Duration of the experiment lasted for 210 minutes. First, subjects entered the climatic chamber, and they voted satisfaction with indoor environment and fatigue on a PC screen. They rested in a sedentary position for 45 minutes to adopt themselves to the environment. Then, they performed thirty-minutes of multiplication task four times, and voted satisfaction with indoor environment and fatigue before and after the tasks. They were asked to answer predicted-performance-vote at the end of the day.

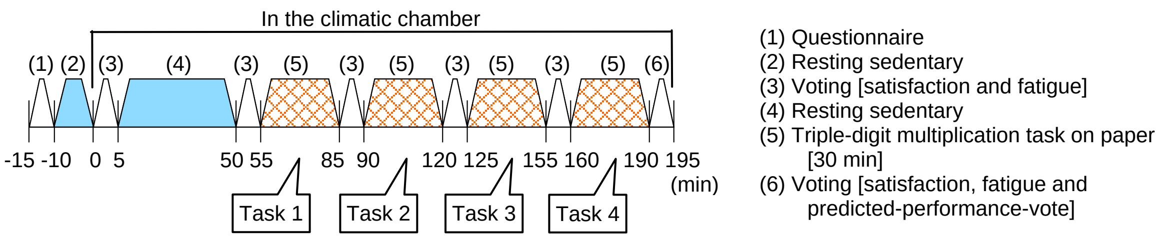


Figure 7. Experimental procedure

Relationship between Satisfaction with Indoor Environment and Performance

Subjects voted their satisfaction with indoor environment six times a day, before and after each session of resting and task. The results of the votes of satisfaction with indoor environment are shown in Figure 8. They were satisfied with environments of TN and TNL, and dissatisfied with those of all other conditions.

To evaluate how much satisfaction with indoor environment affects task performance, the relationship between satisfaction with indoor environment and the Z-score of multiplication task were examined. The data set of satisfaction with indoor environment by the interval of 0.2 was averaged and corresponding Z-score was also averaged. The relationships between satisfaction with indoor environment and the Z-score are shown in Figure 9. The Z-score was related to subjects' satisfaction with indoor environment (r=0.43). The results from the experiment showed that the subjects performed well when they were satisfied with the IEQ. To promote increase of productivity in the office, it is effective to build up the environment providing high occupants' satisfaction.

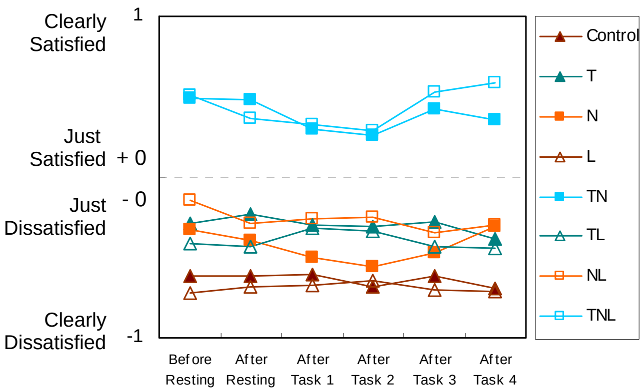


Figure 8. Satisfaction with Indoor Environment

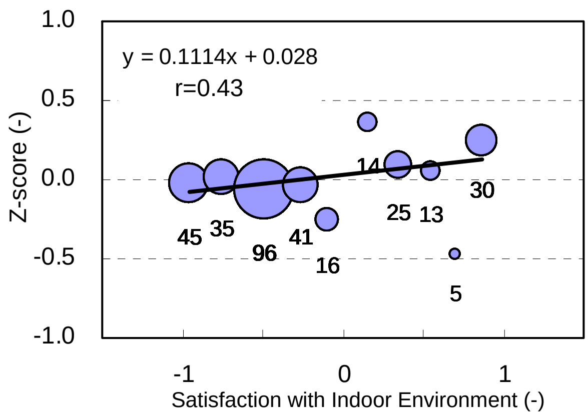


Figure 9. Relationship between Satisfaction with Indoor Environment and Performance

FIELD SURVEY IN CALL CENTER

Method

Data in a call center was collected for one year [16]. Indoor air temperature, humidity, lighting, and CO_2 concentration were monitored. Measurement was conducted from February 18, 2004 to February 1, 2005. Figure 10 shows the interior of call center. Call data of 13,169 were collected in total. This call center is performing guidance and consultation for customers, and technical support. Although the number of communicators changes with a day and time, total number of workers is about 70-120 persons/day in general. Business-hours of a call center are 8:45-19:00 from Monday to Saturday. Air-conditioning is operated from 8:00 to 20:00. Detailed research result is published by Kobayashi et al., see reference [16].

Call Response Rate

The relationship between indoor air temperature and the average call-response rate is shown in Figure 11. A linear regression model weighted by the number of relevant operators was obtained with the correlation coefficient of -0.69. Based on the regression model, the increase in air temperature by 1.0°C is relevant to the decrease in call response rate by 0.15calls/h. In particular, increasing indoor air temperature by 1.0°C from 25.0 to 26.0°C would result the decrement in call response rate from 7.79 to 7.64calls/h, which is 1.9% loss in performance. Seppänen et al. reported that 1.0°C of room air temperature rise is equivalent to decline in 2% of working performance [17]. The result of this research agreed quite well with the model of Seppänen. The average arrival telephone calls time per response number of cases, average receivable time, and the average desk work time were performed stable for whole period. Call-response rate was adopted as an index to evaluate the worker performance. For long period field study indoor air temperature has effects on workers' productivity.

Figure 10. Interior view of a call center

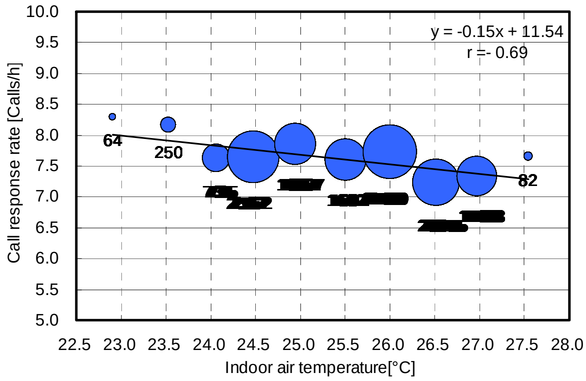
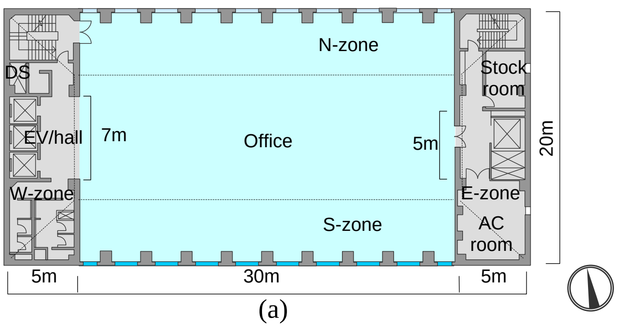


Figure 11. Relationship between indoor air temperature and the average response number of cases

ECONOMICAL AND GLOBAL ENVIRONMENTAL IMPACTS FROM THE MODIFICATION OF COOLING TEMPERATURE SETTINGS IN OFFICE

The effects of the modification of the cooling temperature settings were evaluated and analyzed. The primary energy consumption of the typical office building in Japan, Building K, was calculated by the Building Energy Consumption Simulator (BECS), which is commonly used in Japan for the Energy-Saving Law. The conditions for the simulation are shown in Figure 12. The temperature settings of the air-conditioning system changed by the seasons as: spring and autumn at 24°C (April, May, October, and November); summer at 26°C (June to September); and winter at 22°C (December to March).



(a)

Building K	
Building use	Office
Place	Tokyo, Japan
Floors	BF1 to 11F
Total floor area	10,001.76m^2
Floor area for AC	6082.90m^2
Heat source	2 absorption chillers
Air-conditioning System	Office/Dining hall
	AC for each floor (CAV) and FCU
Office hour	9:00-18:00

(b)

Figure 12. Conditions for the energy consumption simulation. (a) typical floor plan for the Building K, and (b) description of the Building K

The cases studied were: the standard operation of the office buildings as Case-26°C; and the modification of cooling temperature settings during summer for two Centigrade, at 28°C, as Case-28°C. Same air-conditioning systems were assumed to be operated in both cases, and the simulations were conducted over a year. From the results, the primary energy consumption was 1180.5MJ/m^2/year for Case-26°C and was 1161.2MJ/m^2/year for Case-28°C. The difference between the two cases was only 19.3MJ/m^2/year (-1.64%), because chillers were not always operated at the rated point.

The effects of the modification of the cooling temperature settings were evaluated in terms of: (I) the benefit by the reduction in the running cost (yen/m^2/year), (II) the cost by the performance changes (yen/m^2/year), and (III) the reduction in the amount of emission of carbon dioxide (kg-CO_2/m^2/year). They were calculated with the following three equations

$$\text{(I)} = \Delta E_{AC} \times M_E, \quad (1)$$
$$\text{(II)} = k_{summer} \times M_p \div S_p \times \Delta P, \quad (2)$$
$$\text{(III)} = k_{CO2} \times \Delta E_{AC}, \quad (3)$$

where, ΔE_{AC} is the reduction in energy consumption (kWh/m^2/year), M_E is the price of energy (yen/kWh), k_{summer} is the number of months in summer to year (=4/12), M_p is the average salary ($\approx$4 million yen/person/year [18]), S_p is the space per person (=10m^2/person), ΔP is the change in performance (%), and k_{CO2} is the CO_2 emission rate for a thermal power plant (=0.378kg-CO_2/kWh). The building facility was assumed to be operated by electricity from a thermal power plant, whose conversion factors are 9MJ/kWh [19] and 11.08yen/kWh in Tokyo. The change in performance was cited from the result of the survey, 1.9%/°C.

The results of the cost analysis and the reduction in carbon dioxide emission when the cooling temperature settings were modified by 2°C in the Building K are shown in Table 1. The net profit by the modification was -5043 yen/m^2/year, while the reduction in CO_2 emission was 0.81kg-CO_2/m^2/year. It was suggested from the results that (I) the benefit by the reduction in the running cost and (III) the reduction of the CO_2 emission would be minimal compared to the possible cost generated by the performance decrement. The reduction in the emission of greenhouse gases is essential for the world wide concern of the global warming issue; however, it is also important to give considerations on productivity for the measures in offices.

Table 1. The results of the cost analysis and reduction in carbon dioxide emission by modifying the temperature settings by 2°C

(I) Reduction in running cost	(II) Cost by performance change	(III) Reduction in CO_2 emission	(I)-(II) net profit
23.8 yen/m^2/year	5067 yen/m^2/year	0.81kg-CO_2/m^2/year	-5043 yen/m^2/year

CONCLUSION

1) A conceptual diagram for the evaluation of the effect of indoor environmental quality on productivity was proposed.
2) The results from subjective experiments of short term showed that human responses, such as fatigue, cerebral blood flow, sensation to the environment and SBS symptoms, were important for the evaluation of productivity.
3) The results from a subjective experiment of long term exposure revealed the relationship that performance decreased with the increase in the level of fatigue.
4) The results from another subjective experiment showed that the subjects performed well when they were satisfied with the IEQ.
5) A survey in call-center showed the decline in performance in warmer environment.

6) From the simulation on the economical and global environmental impacts for the modification of cooling temperature settings in an office, it was suggested from the results that the benefit by the reduction in the running cost and the reduction of the CO_2 emission would be minimal compared to the possible cost generated by the performance decrement.

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The influence of exposure to multiple indoor environmental parameters on human perception, performance and motivation

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SUMMARY

The impact of simultaneous short-term exposure to three environmental parameters (temperature, noise and air quality) on human perception and performance was studied in two identical climate chambers. Eight conditions were created exposing the subjects for 20 minutes to combinations of two levels of operative temperature (23,5 ºC and 28,0 ºC), two noise levels (52 dB(A) and 60 dB(A)) and two pollution loads (pollution load absent, pollution load present). 56 participating subjects performed simulated office work (addition) and completed questionnaires concerning their perception of the environment and adverse symptoms in each combination of conditions. The subjects were informed of the conditions so they were able to make a conscious or unconscious choice to work more or less according to their perception of the conditions and their attitude to them. Despite the short time allocated for performance of the addition task a significant decrease in performance in warm noisy, warm polluted and warm noisy polluted conditions could be demonstrated. Subjects reported a highly significant reduction in the acceptability of the indoor environment, in self-reported performance and in their ability to concentrate in all deteriorated conditions, i.e. they tended to overestimate the negative effects of each factor. This was especially the case for noise.

INTRODUCTION

The impact of indoor climate parameters on human comfort, health and performance has been studied for decades. Most of this research has evaluated the influence of one environmental parameter at a time. Very little research has been performed on the effects of exposure to multiple indoor environmental parameters as they occur in real life. In real buildings, interventions to improve one factor may fail to achieve a positive effect when several other factors are causing discomfort or they may not achieve as large a positive effect as if more factors had been altered simultaneously. An intervention that reduces one negative factor may even increase the discomfort caused by other factors. A familiar example of this is when opening windows to refresh the air in the room causes increased discomfort due to the traffic noise that can be heard through the open windows. It is therefore essential to study not only the influence that each individual parameter has but also what interactions exists between them in terms of their effects on occupants, and what influence they have on each other.

Toftum [1] concluded in a literature review that there is only limited evidence for the existence of significant interactions between different aspects of the indoor environment. Only for the effect of air temperature and air humidity on perceived air quality are well-established relationships available. Fang et al. [2,3] showed that the acceptability of inhaled air decreased with both increasing air humidity and air temperature, whereas the odour intensity of the air was independent of its psychrometric properties. Furthermore, Fang et al. [4] showed that a decrease in outdoor air supply rate could be compensated for by decreasing indoor air enthalpy so as to avoid reducing perceived air quality. The effects of other combined factors were described by Clausen et al. [5], who determined the relative importance of sensory air pollution, thermal load and noise. Their study showed that a 1 ºC change in operative temperature had the same effect on comfort as a change of 2,4 decipol in the perceived air quality or a change of 3,9 dB in the noise level. Witterseh [6] investigated the impact of combinations of the same three parameters on environmental perception, SBS symptoms and the performance of office work. The combination of ventilation noise and emissions from carpet was found to cause significantly more self-reported fatigue and there was a tendency for a negative effect on performance. The interaction between noise and temperature was found to have a significant effect on the performance of office tasks and on self-estimated performance, ability to think clearly, ability to concentrate and fatigue. Banhidi at al. [7] studied the effect of combinations of two levels of temperature (20 and 30ºC), noise (60 and 70dB) and lighting (280 and 920lux) on performance and physiological measures. A significant effect was found for elevated noise level negatively affecting the performance of a game (Tetris, a falling blocks puzzle video game requiring high concentration and logical thinking) and interaction between lighting and temperature affecting the number of characters written. The influence of a combination of irrelevant speech and indoor lighting on performance was examined by Knez and Hygge [8]. No interactions between noise and light were shown but there were negative effects of presence of noise and cool-white light on long-term memory recall. Furthermore, unpleasantness increased over time in the silence condition and decreased when the subjects were exposed to irrelevant speech. Clausen and Wyon [9] further investigated the influence of 6 combined factors on the acceptability of the environment and on the performance of office work. One group of subjects performed simulated office work in a set of poor environmental conditions with overhead fluorescent lighting, recorded traffic noise, 27 ºC operative temperature, supply air polluted by emissions from linoleum, recorded open office noise, and almost no daylight. The realistic annual cost of improving each of the six conditions was estimated and expressed as a percentage of the total sum. A second group of subjects briefly experienced all poor and improved conditions and individually selected the improvements they preferred, up to a 50%-budget, while the members of a third group of subjects were randomly paired with each of these subjects. The fourth group was exposed to 100%-budget conditions. Significant improvements in subjective assessments of the environment occurred at high budget levels and when individual choice was provided, and the self-reported performance of office tasks improved, but no effects on performance could be shown.

The studies reviewed above showed that the acceptability of the indoor environment and human performance decreases with deteriorating conditions, although in some of them the decrement in performance was not always significant or as large as could have been expected. This might be due to the motivation of people to counteract the uncomfortable conditions and perform well in the experimental setting. However, in reality, if people are aware that their working conditions vary from day to day or during a day, it is conceivable that they will avoid working hard in poor conditions and choose to work better in more comfortable conditions.

The purpose of the present experiment was to study the impact of exposure to three environmental parameters (temperature, noise and air quality), occurring individually or at the same time, on the overall acceptability of the indoor environment and on office work performance, taking into account subjects' motivation to work. It is presumed that if the subjects are aware of the conditions they are about to encounter, and are aware that conditions will later improve or deteriorate, they will make a choice (consciously or unconsciously) to exert more or less effort in their work according to the conditions prevailing at the time.

METHODS

The experiment was carried out in two identical climatic chambers at the International Centre for Indoor Environment and Energy, DTU in June 2006. Both chambers were equipped with four workstations, each comprising a chair, desk and computer.

56 subjects participated in the experiment. The participants were recruited among students attending DTU. The average age of the group was 24 years and it consisted of 26 females and 30 males. The participants were paid for their participation.

Eight conditions were created and the subjects were exposed in groups of four to all combinations of two levels of operative temperature, two noise levels and two states of indoor air quality (Table 1.).

Table 1. Environmental parameters

Environmental parameter		Reference condition	Deteriorated condition
1	Operative temperature	23,5 ºC (PPD 5 %)	28,0 ºC (PPD 38 %)
2	Traffic noise	52 dB(A)	60 dB(A)
3	Air quality	Pollution load absent	Pollution load present 23 m^2 of old carpet

The eight experimental conditions are shown in the Table 2. The groups of subjects were exposed to these conditions in randomized balanced order.

Table 2. Combinations of exposures

Symbol	Pollution load	Operative temperature	Noise level
C1	Absent	23,5 ºC	52 dB(A)
C2	Absent	23,5 ºC	60 dB(A)
C3	Absent	28,0 ºC	52 dB(A)
C4	Absent	28,0 ºC	60 dB(A)
C5	Present	23,5 ºC	52 dB(A)
C6	Present	23,5 ºC	60 dB(A)
C7	Present	28,0 ºC	52 dB(A)
C8	Present	28,0 ºC	60 dB(A)

A recording of traffic noise was played inside the chambers using a CD player and one speaker in each chamber. In the reference condition of 52 dB(A) the traffic noise was barely audible as through a closed window; in the deteriorated condition of 60 dB(A) the traffic noise was clearly audible as if the window was open. An old carpet taken from an office was used as a source of pollution; 23 m^2 of carpet were used corresponding to the floor area of the chamber. The carpet was placed in a "pollution chamber", through which the supply air to one chamber was passed. The outdoor air supply rate was kept constant at a high level of 26 l/s per person to eliminate the effect of bioeffluents and other pollution sources than the old carpet. Prior to the experiments the subjects participated in a one-hour preliminary session. During the preliminary session 10 minutes were allowed for experiencing the environmental conditions and the subjects performed an olfactory ranking test [10] to ensure they had a normal sense of smell.

During the experiment the subjects were exposed to sequentially changing combinations of conditions while working on a computer. Only minor breaks disturbed the continuity of the subjects' work when they exited one combination of conditions and entered another. The subjects performed simulated office work (addition) for 15 minutes in each combination of conditions. In the last 5 minutes of each exposure they then completed a questionnaire concerning their perception of the environment, thermal comfort and adverse symptoms. The task and questionnaires were administered using the DTU software tool "Remote Performance Measurement" (RPM) [11]. At the end of the exposure period the experimenter signalled the subjects to exit the chamber. In this very short break the subjects were asked to state which of the environmental parameters they considered it would be the most important to improve if they were working in such an environment and which would be the second most important to improve. They then received a notice describing in informal terms the next conditions they were to work in and returned to work in one of the chambers where the respective conditions were set. This break served also as an exercise to keep the activity of subjects at approximately the same level as in a real office. The whole experimental session lasted for 3 hours.

The data obtained in the experiment and their residuals were analysed for normality using the Kolmogorov-Smirnov test. Paired t-tests and the Wilcoxon matched-pairs signed-rank test were then used for analyzing normally and not normally distributed data respectively. The results of a comparison analysis between the reference condition C1 (23,5 ºC; 52 dB(A); pollution load absent) and the seven deteriorated conditions and between the deteriorated conditions themselves are presented.

RESULTS

The results of an addition test, i.e. the number of completed additions per exposure, are presented in Figure 1. The difference from the result obtained in C1 was significant in conditions C4 and C7 at the level of significance $p<0{,}05$ and in the condition C8 at the level of significance $p<0{,}01$. No significant difference was found between conditions in terms of the percentage errors committed. Self-estimated performance showed a significant drop in all deteriorated conditions compared to C1 ($p<0{,}001$) (Figure 2).

Figure 1. Number of completed additions per exposure ± standard deviation

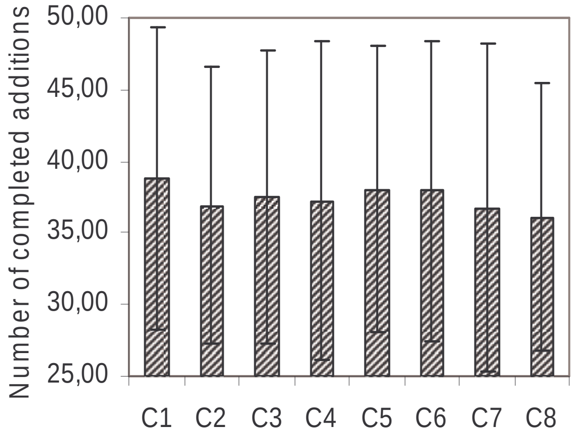
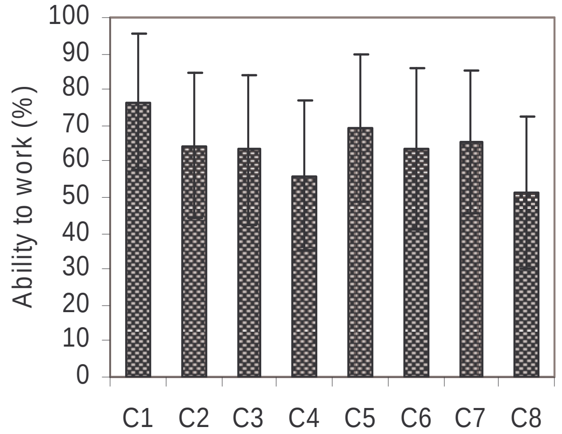


Figure 2. Self-estimated performance (%) ± standard deviation

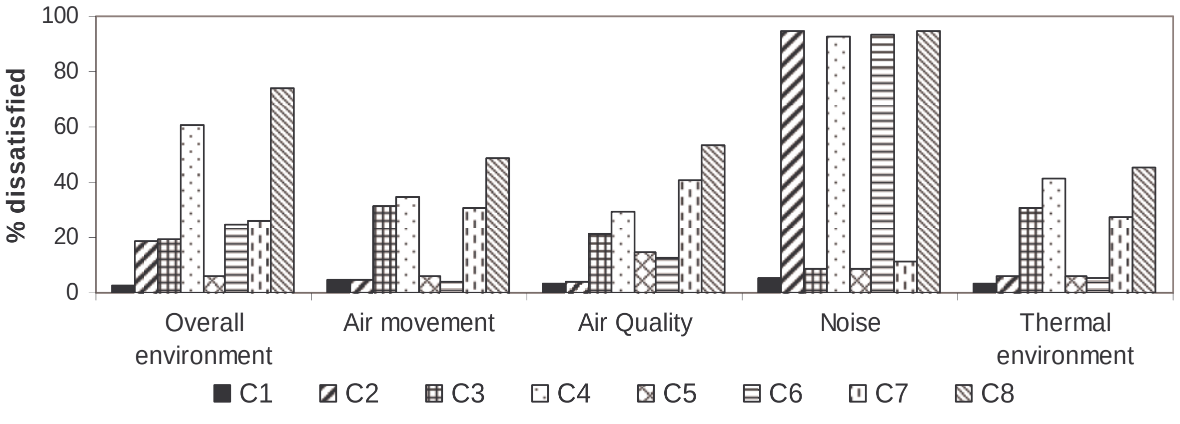


The mean acceptability votes are shown in Table 3 and the percentages dissatisfied with the overall environment, air movement, air quality, noise and thermal environment are presented in Figure 3.

Table 3. Acceptability of the environment and its factors

	Mean acceptability ± standard deviation							
	C1	**C2**	**C3**	**C4**	**C5**	**C6**	**C7**	**C8**
Overall indoor environment	0,64 ±0,30	0,25 ±0,36	0,24 ±0,40	-0,11 ±0,47	0,48 ±0,35	0,18 ±0,45	0,16 ±0,42	-0,24 ±0,50
Air movement	0,55 ±0,32	0,55 ±0,33	0,12 ±0,46	0,09 ±0,47	0,49 ±0,33	0,55 ±0,28	0,12 ±0,43	-0,02 ±0,44
Indoor air quality	0,61 ±0,31	0,56 ±0,33	0,21 ±0,41	0,13 ±0,45	0,30 ±0,43	0,33 ±0,45	0,04 ±0,46	-0,06 ±0,43
Noise	0,51 ±0,38	-0,57 ±0,38	0,41 ±0,42	-0,52 ±0,49	0,41 ±0,33	-0,54 ±0,40	0,36 ±0,43	-0,58 ±0,39
Thermal environment	0,61 ±0,30	0,49 ±0,39	0,12 ±0,42	0,03 ±0,41	0,48 ±0,39	0,51 ±0,34	0,15 ±0,45	0,00 ±0,45

Figure 3. Percentage of dissatisfied with the environment and its factors



The acceptability of the overall environment was significantly lower in all deteriorated conditions compared to the reference condition C1 ($p<0,001$; except C5 where $p<0,01$). However, when the deteriorated conditions that differ only in pollution load were compared between each other (i.e. C2 against C6, C3 against C7, C4 against C8), the analysis did not show a significant difference in acceptability. A major drop in acceptability can be observed in the two conditions exposing subjects to both elevated operative temperature and elevated noise level (i.e. C4 and C8).

The air movement was significantly less acceptable in the conditions with elevated temperature C3, C4, C7 and C8 ($p<0,001$) than in the reference condition C1.

The air quality was significantly less acceptable in all deteriorated conditions compared to C1 ($p<0,001$), except C2. The acceptability of air quality decreased significantly when the pollution load was present in conditions C5, C6, C7 and C8 but also in the conditions where the pollution load was absent but the temperature was elevated (C3 and C4). In the condition with pollution load present and low temperature (C5), 15% occupants were dissatisfied with the air quality. Elevating the temperature only (C3) had an even greater impact on the acceptability of the air quality, with 21% dissatisfied; however the difference between the acceptability of the air quality in C3 and in C5 is not significant. As many as 41% were dissatisfied with the air quality in the combination of elevated temperature and presence of pollution load (C7). The effect of the combination was significant comparing to both the effect of elevated temperature only in C3 ($p<0,05$) and to the effect of pollution load only in C5 ($p<0,01$). The addition of an elevated noise level had a significant effect on the acceptability of air quality only when noise was added to both elevated temperature and pollution load ($p=0,04$) (C7 compared against C8).

The conditions with elevated noise level caused the acceptability of noise in the environment to decrease ($p<0,001$). Similarly, the acceptability of thermal environment was significantly lower in the conditions with elevated temperature ($p<0,001$) than in the reference condition C1.

The ability to concentrate was negatively affected in all deteriorated conditions ($p<0,001$; except C5 where $p=0,01$). The subjects felt worse (subjective feeling marked on a continuous scale from feeling bad to feeling good) in all deteriorated conditions ($p<0,001$; except C5 where $p=0,02$).

The subjective importance of the three deteriorated conditions is summarised in Table 4. The results shown were obtained after the last exposure when the subjects had already experienced all possible combinations of conditions and were making the final decision on the relative importance of the three factors.

Table 4. Ranking of the deteriorated conditions by subjects after the last exposure

Condition	**Chosen by subjects (%) to be**	
	The most important to improve	**The second most important to improve**
Elevated noise level	42	40
High temperature	36	35
Deteriorated air quality	22	25

DISCUSSION

Despite the very short time allocated for performance of the addition test in each condition, it was possible to demonstrate a deterioration in the subjects' performance in conditions C4, C7 and C8 compared to the reference condition C1. It seems likely that in longer exposures the effect on performance would be larger.

The highly significant differences in the acceptability of the indoor environment and its component factors may have been due to the experiment not being blinded. This was the expected bias. The subjective overall ranking of the three deteriorated conditions are supported by the acceptability ratings: the air quality seemed the least important to subjects when they were asked directly and the addition of air pollution did not cause a significant drop in the acceptability of the overall environment. The assignment of lower importance to air quality may also have been due to the subjects completing the questionnaires at the end of each exposure when they were fully adapted to the air quality in the climate chamber.

The air movement was considered significantly less acceptable in the conditions with elevated temperature, compared to the reference condition. The subjects were not asked whether they would prefer more or less air movement but it is likely that they would have preferred more air movement to provide some cooling in the warm conditions.

The results on acceptability of air quality show a decrease in acceptability when the pollution load was present or the temperature was elevated. These results are in agreement with the findings of Fang [2,3] on how perceived air quality is affected by raised temperature. The results indicate that moderately raised air temperature may have even greater influence on acceptability of the air quality than the pollution caused by an old carpet.

The subjects' choice of the noise level being the most important to improve and the air quality the least important accords with the results of the experiment conducted by Clausen and Wyon [9]. However, subjects still differed as to which of the three conditions it was the most essential to improve.

The most important implications of these results for practice are that: 1) Although self-estimated performance predicted environmental effects on actual performance quite well, it exaggerated the magnitude of the effect (33% in the most negative condition instead of 7%); 2) Warm air had almost the same negative effects as polluted air on SBS and on environmental perceptions, providing support for the widespread practice of lowering the temperature in offices to improve perceived air quality; and 3) Although noise had a large and consistent negative effect on the perceived ability to concentrate, clearly more than that of any other factor, it had virtually no other effect of any kind, i.e. subjects tended to over-estimate the importance of noise, so more selected it as the most important to change, even though it was not the most important.

CONCLUSIONS

- The addition test was performed significantly less well in the warm noisy condition, the warm polluted condition and the warm noisy polluted condition.

- Self-estimated performance and the overall acceptability of the environment were both significantly worse than in the reference condition when any negative factor was present.
- Poor air quality had less of an effect than elevated temperature or noise on the overall acceptability of the environment.

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Effect of Work Motivation on the Task Performance under the Different Thermal Conditions

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SUMMARY

The impact of the work motivation on the occupants' work performance of serial one-figure addition task and proofreading task was studied in two types of experiments, i.e., type A of cool condition (22 deg C) and type B of neutral condition (25 deg C). During the occupancy of 120 minutes under each condition, the subjects performed the above tasks with financially motivated situation and without motivated situation. The subjects also answered the questionnaire, which asked the indoor environmental quality and sick building syndrome symptoms during the conditions. It was found that there was significant difference in the performance of theses two tasks between the motivated situation and no-motivated situation. The shape of the performance curve for type A during no-motivated situation was similar to that during motivated situation. The performance of addition task was well correlated with the thermal acceptability vote during motivated situation. On the other hand, the good correlation was not found between the performance and thermal acceptability during no-motivated situation.

INTRODUCTION

Many researches in various research fields, such as occupational hygiene, ergonomics, psychology, architecture and so forth, have investigated the effects of indoor environmental factors on productivity since Vernon's research. He investigated the monthly output of five plating factories with monthly outside temperature and concluded that an output fallen along with the rise of temperature [1]. In the early stage of the psychological research on workplace, the effect of an environmental stimulus was considered to be uniform to everybody. Sundstrom *et al* mentioned that the relation between people and environment was mechanistic and deterministic [2]. They explain that, in theory, a person's response to the environment may involve several processes, which in turn will affect his/her performance. The main processes are, arousal, stress, distraction, overload, fatigue and adaptation. Sundstrom reports that mental tasks have generally been unaffected by heat whereas motor tasks have suffered in heat. As a result of the experiments the effect of various illuminance levels on the productivity of assembly workers was studied in Hawthorne Works, it was found that even decreasing the illuminance level increased worker productivity. It was thought that increased motivation could lead to increased productivity; this is commonly referred to as the Hawthorne effect [3].

In the work performance studies in Japan, the serial one-figure addition task has been used as an index of the productivity from prewar days. Since this task was developed as "Uchida-Kräpelin mental work test" in psychology, the performance curve of the one-figure addition task could be used to investigate a person's aspect of mental process [4]. Many researches on productivity studies in built environment use total amount of calculation of the addition task

within given time as an index of performance. However we use the performance curve as an index of his/her mental aspect [5].
The purpose of this study was to investigate the impact of the work motivation on the occupants' work performance of serial one-figure addition task and proofreading task in two conditions, i.e., type A under cool condition (22 deg C) and type B under neutral condition (25 deg C). During the occupancy of 120 minutes under each condition, the subjects performed the above tasks with financially motivated situation and without motivated situation. The difference between the performance conducted during motivated situation and that during no-motivated situation could be discussed.

METHODS

Experimental condition

The experiments were conducted in the test room shown in Figure 1. The room with two windows facing south had a floor area of 32m^2 and a volume of 96 m^3 (L×W×H=8×4×3). Two different thermal conditions were set up in the room as listed in Table 1: 1) type A, room temperature 22 deg C, 2) type B, 25 deg C. The air temperature was controlled by the air conditioner equipped in this room.

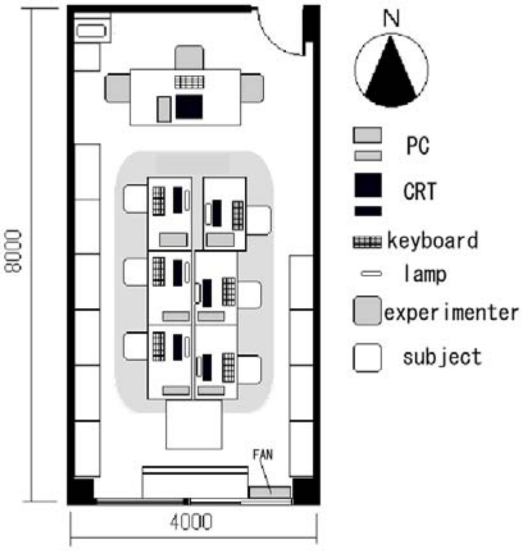

Figure 1 Experimental Room

Table 1 Experimental type

type	Room temperature	Relative humidity
A	22 deg C	40～
B	25 deg C	

Subjects sat at six workstations, each consisting of a table, a chair, a desk lamp and a personal computer (PC). Thirty female subjects participating in the present experiments were all students. The subjects were paid a salary for participating in the experiment at a fixed rate. To increase their motivation during the motivated situation, they were also paid a bonus of up to 15% of the total salary, depending on their performance.

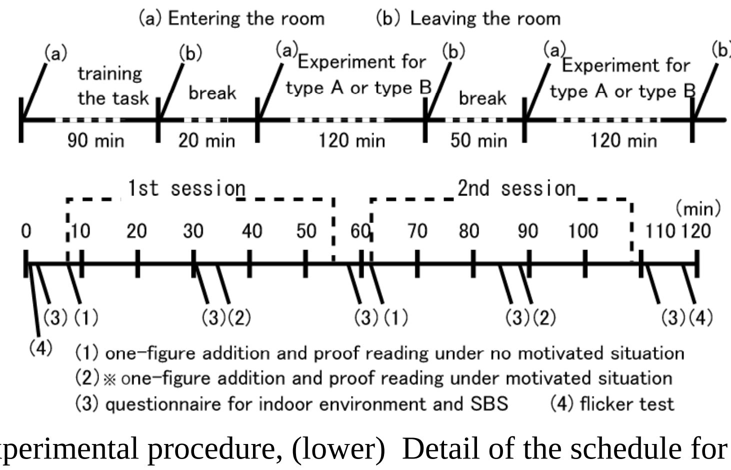

Figure 2(upper) Experimental procedure, (lower) Detail of the schedule for an experiment

Table 2 Questionnaire for indoor air condition

Assess Odour Intensity	Assess Thermal Comfort	Assess Indoor Quality	Assess Irritation in (Eyes Nose Throat)
	3 Hot		
0 No odour			0 No irritation
	2 Warm	1 Clearly acceptable	
1 Slight odour			1 Slight irritation
	1 Slight warm	0.05 Just acceptable	
2 Moderate odour			2 Moderate irritation
	0 Neutral	−0.05 just not acceptable	
3 Strong odour			3 Strong irritation
	−1 Slight cool	−1 Cleary not acceptable	
4 Very strong odour			4 Very strong irritation
	−2 Cool		
5 Overpowering odour			5 Overpowering irritation
	−3 Cold		

Table 3 Questionnaire for self-condition and SBS symptoms

Right now my environment can be described as follows:

	Extremely	Very	Slight	Neutral	Slight	Very	Extremely	
Too humid	☐	☐	☐	☐	☐	☐	☐	Too dry
Air stuffy	☐	☐	☐	☐	☐	☐	☐	Air fresh
Too dark	☐	☐	☐	☐	☐	☐	☐	Too bright
Too quiet	☐	☐	☐	☐	☐	☐	☐	Too noisy
	−3	−2	−1	0	1	2	3	

Right now I feel as follows:

	Extremely	Very	Slight	Neutral	Slight	Very	Extremely	
Nose blocked	☐	☐	☐	☐	☐	☐	☐	Nose clear
Nose dry	☐	☐	☐	☐	☐	☐	☐	Nose running
Throat dry	☐	☐	☐	☐	☐	☐	☐	Throat not dry
Mouth dry	☐	☐	☐	☐	☐	☐	☐	Mouth not dry
Lips dry	☐	☐	☐	☐	☐	☐	☐	Lips not dry
Eye dry	☐	☐	☐	☐	☐	☐	☐	Eyes not dry
Eye smarting	☐	☐	☐	☐	☐	☐	☐	Eyes not smarting
Eye feel itchy	☐	☐	☐	☐	☐	☐	☐	Eyes not itchy
Severe headache	☐	☐	☐	☐	☐	☐	☐	No headache
Feeling bad	☐	☐	☐	☐	☐	☐	☐	Feeling good
Tired	☐	☐	☐	☐	☐	☐	☐	Rested
Difficult to concentrate	☐	☐	☐	☐	☐	☐	☐	Easy to concentrate
Depresed	☐	☐	☐	☐	☐	☐	☐	Positive
Alert	☐	☐	☐	☐	☐	☐	☐	Sleepy
	−3	−2	−1	0	1	2	3	

Completion of tasks requires:

☐ 100% ☐ 90% ☐ 80% ☐ 70% ☐ 60% ☐ ( )% ☐ 0%

Experimental procedure

The experimental procedure is presented in Figure 2 (upper). The experiment was carried out for 7.5h from 9:00 to 16:30. Figure 2 (lower) shows in detail the schedule. From 9:30 to 11:00 subjects took the training for the tasks, i.e., serial one-figure addition and proof reading, used in the following experiments. After 30 minutes break, subjects participated the experiment of type A or type B (random order) for 120 minutes (11:30 – 13:30). Then subjects participated the experiments of type A or type B for 120minutes (14:20 – 16:20). For each experiment, subjects entered the room and approached their workstations. Once seated, they assessed perceived air quality, general perceptions of the environment, SBS symptoms and thermal comfort. These evaluations were made several more times during exposure in the experiment.
The questionnaire for the indoor air condition is listed in Table 2 and for self-condition and SBS symptoms in Table 3. While exposed to the conditions in the room, subjects performed tasks. In 1st session, they performed serial one-figure addition task for 10 min. and proofreading task for 10 min. under no-motivated situation. After the no-motivated situation, they performed serial one-figure addition task for 10 min. and proofreading task for 10 min. under motivated situation. In 2nd session, the above tasks were repeated.

RESULTS

The measured levels of the general parameters describing the indoor climate inside the room are shown in Table 4. The average air-change rate of the room measured by tracer gas decay method was 0.8 h^{-1}.

Table 4 Measured room temperature and relative humidity

	Type A	Type B	Outdoor
Room temperature (deg C)	22.0±0.9	25.3±0.8	13.8±1.6
Relative humidity (%)	46±5	39±4	58±15

(average±SD)

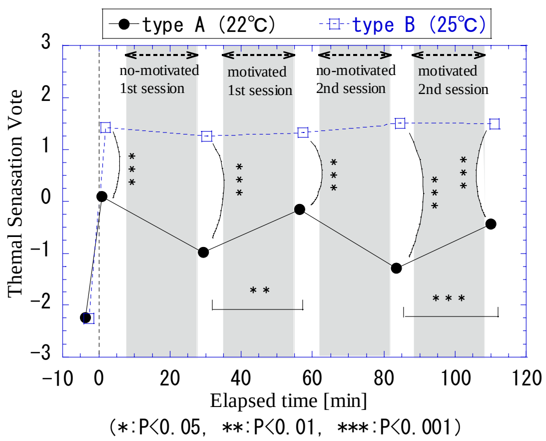

Figure 3 Thermal Sensation Vote as a function of time

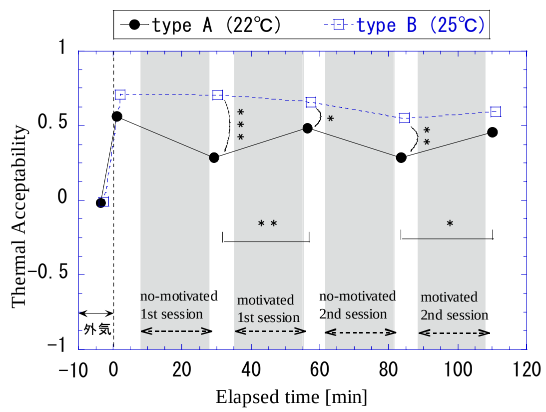

Figure 4 Thermal acceptability as a function of time

The average Thermal Sensation Vote (TSV) voted by subjects during no-motivated situation and motivated situation for type A was –0.7 and that for type B was 1.4. The average TSV value as a function time was presented in Figure 3. For type A, the average TSV values voted just after motivated situation was significantly higher (warmer) than those just after no-motivated situation. However there was no impact of the motivation on the TSV for type B.

Figure 4 shows the average thermal acceptability vote as a function of time. Since the experiments were conducted in autumn, subjects preferred the warmer condition of type B.
The average amount of calculation in one-figure addition task as a function of time is shown in Figure 5. There was no significant difference in the amount of calculation between type A and type B. On the other hand, the amount of calculation during motivated situation was significantly more than that during no-motivated situation both in type A and type B (P<0.01).

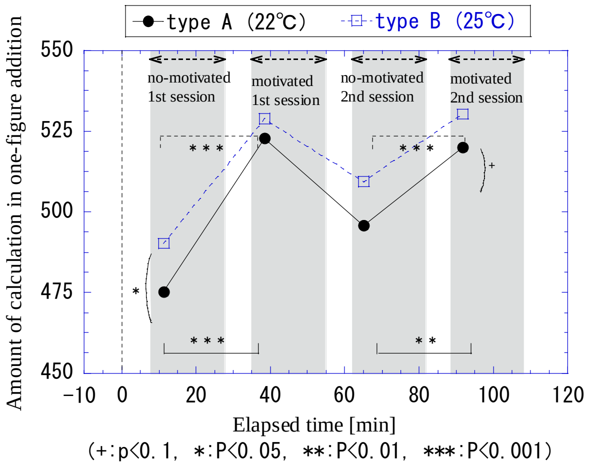

Figure 5 Average amount of calculation in one-figure addition task as a function of time

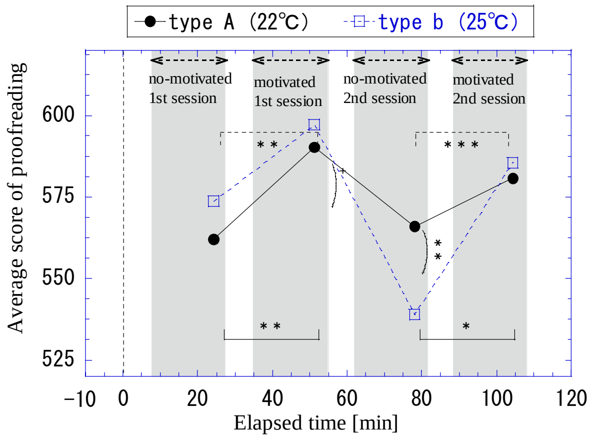

Figure 6 Average score of proofreading task as a function of time

The average score of proofreading task as a function of time is presented in Figure 6. Comparing the score of proofreading in type A with that in type B, the score in type A was significantly less than that in type B only during no-motivated situation in 2nd session. In terms of the motivation, it was found that the score during motivated situation was significantly more than that during no-motivated situation both in type A and type B as same as the result of one-figure addition task.

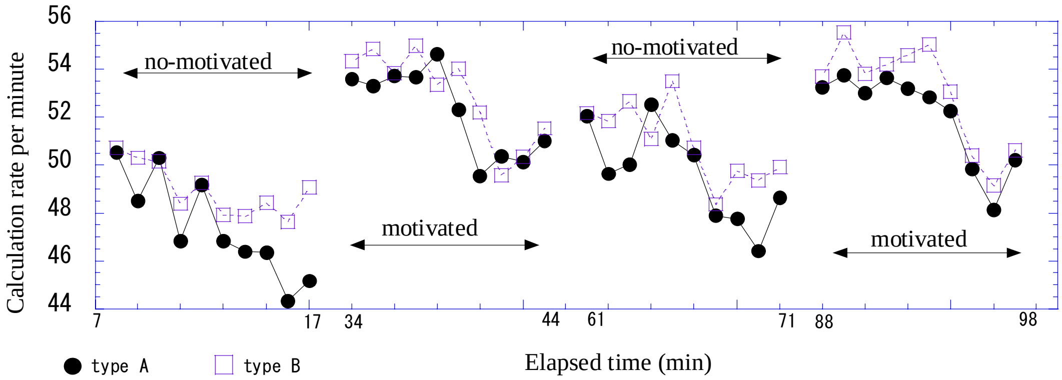

Figure 7 Average calculation rate per one minute as a function of time of one-figure addition task (ten minutes each)

In “Uchida-Kräpelin mental work test”, the performance curve of the one-figure addition task has been used to investigate a person’s aspect of mental process in psychology. Also for productivity study in the built environment, this performance curve was used to guess the mental process of subjects who were conducting tasks [5]. The average calculation rate per one minute as a function of time of one-figure addition task (ten minutes each) is shown in Fig.7. This figure shows that the calculation rate change curve (performance curve) had time decay trend. For each task, the calculation rate in the latter 4 minutes was much lower than

that in the first 6 minutes during no-motivated situation for type A. During motivated situation in 2nd session, the calculation rate in the first 6 minutes for type B was higher than that in the first 6 minutes for type A.
Therefore the calculation rates in the first 6 minutes and that in the latter 4 minutes for each experimental type were determined from the performance curve in Figure 7. The calculation rates during no-motivated situation were drawn in Figure 8 and those during motivated situation in Figure 9. The calculation rate in the latter 4 minutes for type B was significantly more than that for type A during no-motivated situation both in 1st session and 2nd session (Figure 8). On the other hand, the calculation rate in the first 6 minutes for type B was significantly more than that for type A during motivated situation in 2nd session (Figure 9).

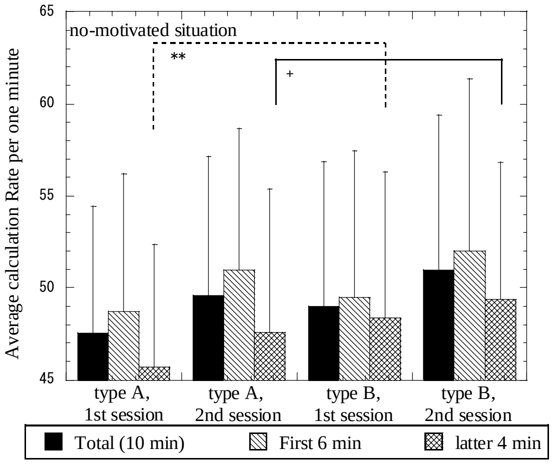

Figure 8 Average calculation rate for each session during no-motivated situation

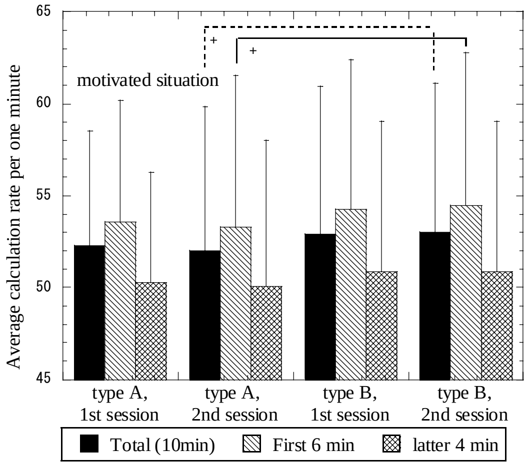

Figure 9 Average calculation rate for each session during motivated situation

DISCUSSION

While there was no large difference in performance curve between type A and type B during motivated situation, large difference was seen in calculation rate in the latter 4 minutes between type A and type B during no-motivated situation (Figure 7). The time decay trend was found in the performance curve for type A more remarkably than type B during no-motivated situation. The above time decay trend might be used for the index of arousal since this trend was seen during motivated situation. Figure 10 shows the relation between the average thermal acceptability and calculation rate in the first 6 minutes and that between the average thermal acceptability and calculation rate in the latter 4 minutes.
It was found that there was higher correlation between the calculation rate and the thermal acceptability during motivated situation than that during no-motivated situation. During motivated situation, arousal level of the subject could have been high because of the chance of bonus. Therefore lower arousal stimulus by indoor environment might have been needed during motivated situation. During no-motivated situation, arousal level of the subjects could have been low because of no chance of bonus. In this case, the subject performed more task in the cooler environment of the higher arousal stimulus even the thermal acceptability was low. In this study the thermal environment had no significant effect on the work performance during motivated situation. However there was effect of thermal environment by arousal on the work performance during no-motivated situation.

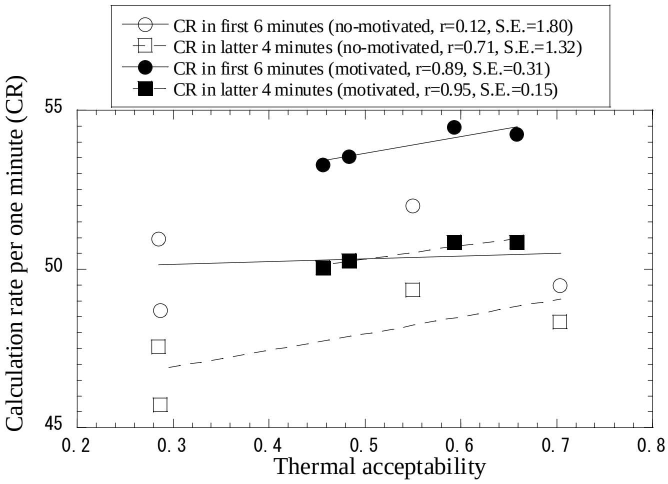

Figure 10 Relation between the average thermal acceptability and calculation rate in the first 6 minutes and that between the average thermal acceptability and calculation rate in the latter 4 minutes.

CONCLUSIONS

1) Thermal acceptability voted by subjects for type B (25 deg C) was significantly higher than that for type A (22 deg C) during no-motivated situation.
2) There was no significant difference in work performance (i.e., amount of calculation in one-figure addition task and average score of proofreading task) between type A and type B. For each experimental type, work performance during motivated situation was significantly better than that during no-motivated situation.
3) Performance curve, which presents the time change of calculation rate in one-figure addition task, showed time decay trend. During motivated situation, the calculation rate in the latter 4 minutes was much lower than that in the first 6 minutes. This time decay might be an index of the arousal. During no-motivated situation, the time decay trend was remarkable for type A.
4) It was found that there was higher correlation between calculation rate and the thermal acceptability during motivated situation than that during no-motivated situation. During motivated situation, arousal level of the subject could have been high because of the chance of bonus. Therefore lower arousal stimulus by indoor environment might have been needed during motivated situation. During no-motivated situation, arousal level of the subjects could have been low because of no chance of bonus. In this case, the subject performed more task in the cooler environment of the higher arousal stimulus even the thermal acceptability was low.
5) In this study the thermal environment had no significant effect on the work performance during motivated situation. However there was effect of thermal environment by arousal on that during no-motivated situation.

ACKNOWLEDGEMENT

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Distribution of Thermal Sensation Votes in Offices used to Model Annual Mental Performance Decrements

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SUMMARY

This paper presents the development of distribution models of office employee's thermal sensation and an introduction to a methodology how to integrate the fact that humans perceive temperature differently with the effects of temperature on mental performance. Two different distribution models have been developed; one for occupants in mechanically ventilated buildings and one for occupants in naturally ventilated buildings. The results show that there is a significantly difference ($P<0.05$) in how people assess the indoor thermal environment depending on ventilation principle. Thus when calculating the effects of temperature on occupants' mental performance it is important not only to take the individual differences into consideration between humans, but also building ventilation principle.

INTRODUCTION

Effects on employee performance of different indoor climate parameters have received considerable attention during recent years [1, 2, 3, 4]. Several studies have shown an increase in mental performance from 5-10% (in some cases up to 20%) when improving the indoor climate e.g. by increasing the air change rate, provide better temperature control or use better filtration [1, 5, 6, 7]. Based on substantial numbers of field and laboratory studies, Seppänen et al. [2, 3, 8] have suggested models to describe relationships between performance and different indoor climate parameters. Design tools for evaluating the economical implications of investments in indoor environmental quality, which take into account employee productivity, may provide strong arguments to companies to invest in improved indoor environment. Such tools need appropriate relation between indoor environment factors and performance to be functional and convincing.

Development of relationships between indoor climate parameters and mental performance (e.g. an indoor air temperature of 30 °C reduce performance of office workers with 10%) is the first element of a model that can be used for estimating the potential economical gains of an investment in better indoor environment. Two other elements should be included in order to make the model stronger and more precise:

1. Hourly data of the indoor environment
2. Distribution models of the employees in a given environment

Since the indoor climate is normally dynamic (variation over day and season) calculation on an hourly basis during a year may be used as a more representative input to estimate the effects on mental performance of the indoor climate. These hour by hour values of the indoor climate can typically be obtained using computer simulation. Many different building simulation tools to evaluate the indoor climate exit on the market. Using the output from such tools in a performance calculation program, enables easy comparison of different building designs and their effects on performance.

Even when exposed to a uniform indoor environment (e.g. in well controlled climate chambers) humans perceive the indoor environment differently e.g. [9]. It is thus likely that occupant performance will be affected differently in different individuals, e.g. as a consequence of individuals having a different thermal sensation even though their exposed to only one (uniform) temperature.

Using distribution models of the employees' perception of the indoor environment allows differences between individuals to be taken into account. People do not sense the indoor climate the same way, some find it draughty, too cold, and too stuffy while others find the same conditions comfortable. Knowledge of the distribution of sensation of different indoor climate parameters may be a step to better assess indoor climate effects on performance.

Temperature is one of the most investigated parameters and it is one of the indoor climate parameters that cause most complaints [10]. Figure 1 shows schematically, the different steps of the methodology on how to evaluate effects on mental performance of temperature. Step 1, first the air temperature needs to be calculated on an hourly basis using building simulation tools. Step 2, air temperature is used with other factors (clothing insulation, relative humidity, air velocity, ventilation system etc.) to evaluate mean thermal sensation distribution of the employees. Step 3, the corresponding perceived temperature for each level of thermal sensation in the distribution is calculated. Step 4, mental performance decrement can be calculated on an hourly basis during a year.

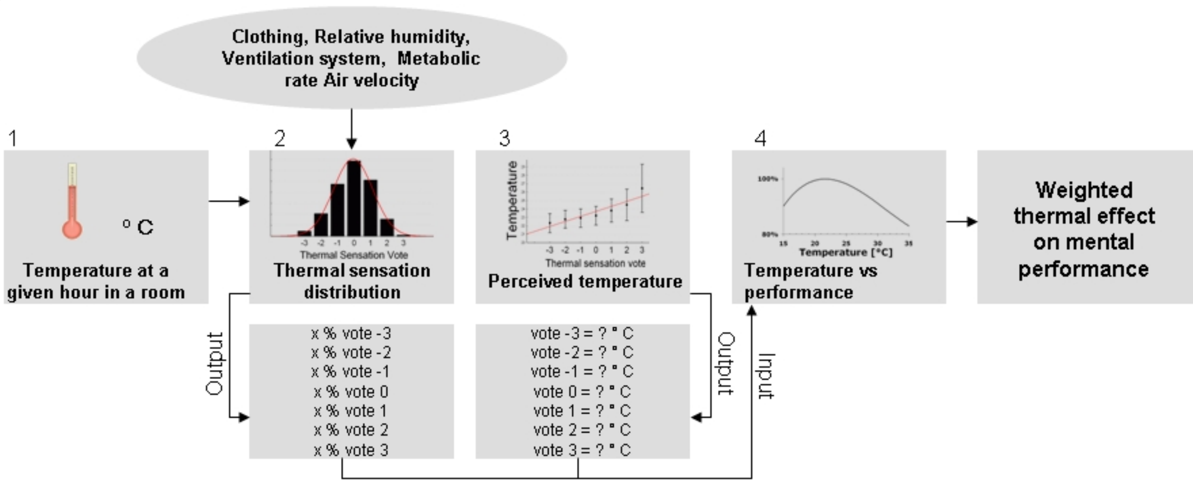

Figure 1. Methodology of calculating mental performance loss due to the effects of air temperature

In this paper distributions of thermal sensation votes will be used to assess the resulting distribution of perceive temperature and the corresponding weighting factors. The distribution will be based on data from ASHRAE RP-884 Adaptive Model Project [11]. The database contains corresponding set of indoor climate measurements and thermal sensation votes from thousands of office employees around the world, occupying hundreds of mechanically and naturally ventilated buildings. The distribution will be based on responses from building occupants, rather than subjects in laboratories, to reflect better the non-uniform condition that occur in buildings in practice. This distribution will be used as input to model effects mental performance effects of the thermal environment during a full year.

METHODS

Taking into consideration that individuals respond differently, even when exposed to the same environment, it is likely that at 22 °C some occupants will feel warm or neutral while others may feel cool or cold. Fanger [9] observed a distribution of thermal responses from sensation votes under different air temperatures. This distribution was based on votes cast by human subjects during exposure to uniform conditions in laboratory environments. During the exposure, subjects were asked to vote on the 7-point ASHRAE scale ranging from Cold to Hot (see Figure 2).

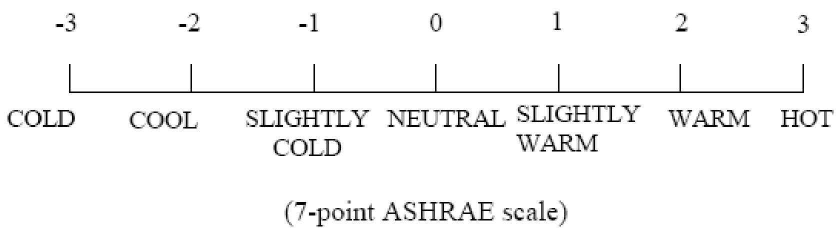

Figure 2. ASHRAE 7-point Thermal Comfort scale

Instead of using climate chamber data we choose to investigate data from field studies. These studies used the scale in Figure 2, but in real office environments. Table 1 shows an overview of the data used in this paper. Data has been divided into two main categories: Mechanically ventilated and naturally ventilated (NV) buildings.

Table 1. Overview of investigated data

Ventilation system	**Number of subjects**	**Buildings**
Mechanical	8119	100
Natural	4679	24
Total	12798	124

The data consists of information from over 12700 occupants in 124 different buildings. Subjective assessments of the thermal conditions and physical indoor climate parameters (e.g. air temperature, air velocity, relative humidity) have been monitored. The buildings were primarily office buildings. Only data with occupant activity levels below 1.3 Met and clothing insulation values below 1.1 clo was included. The data from ASHRAE RP-884 Adaptive Model Project was classified in three broad classes: Class I, Class II and Class III. Class III field studies were based on simple measurements of temperature and simple questionnaires, and these data where excluded from the present investigation. Only data from investigations which were classified as Class I or Class II was included in the present analysis. Class II data includes data where all indoor physical environmental variables necessary for the calculation of PMV/PDD indices were made at the same time and place as the thermal questionnaires were administered. Class I data consists of data of same type as Class II data, but the measurements and procedures were in 100% compliance with ASHRAE Standard 55 (1992) and ISO 7730 (1984) (e.g. measurements in three heights, measurement of turbulence intensity etc.) The statistical program Statistica 7.1 was used for data management and statistical calculations.

For each one-degree interval of indoor air temperature from 20-30°C measured in the mechanically ventilated buildings and from 20-33°C in naturally ventilated buildings, the count of corresponding thermal sensation votes in one scale-unit intervals from -3 to +3 was recorded and used to describe the distribution.

RESULTS

Table 2 and 3 show the distribution of thermal sensation votes for occupants in mechanically and naturally ventilated buildings. The tables show the number of occupants who were exposed to a certain air temperature and the corresponding percent of occupants who voted a certain category on the 7-point ASHRAE thermal comfort scale.

Table 2. Distribution of Thermal Sensation Votes (%) in mechanically ventilated buildings

°C	N	-3	-2	-1	0	1	2	3
20	109	9%	27%	34%	24%	6%	1%	-
21	374	8%	14%	30%	32%	11%	4%	1%
22	1590	7%	16%	32%	30%	12%	3%	-
23	2712	3%	11%	28%	35%	19%	4%	-
24	2093	2%	10%	23%	33%	24%	7%	-
25	752	-	4%	16%	30%	35%	14%	2%
26	234	-	-	5%	24%	44%	22%	5%
27	113	-	2%	5%	8%	39%	31%	15%
28	32	-	-	-	3%	22%	41%	34%
29	15	-	-	-	-	53%	27%	20%
30	43	-	-	-	-	16%	44%	40%

Table 3. Distribution of Thermal Sensation Votes (%) in naturally ventilated buildings

°C	N	-3	-2	-1	0	1	2	3
20	37	3%	27%	32%	11%	19%	8%	-
21	129	7%	16%	23%	20%	17%	15%	2%
22	212	2%	10%	24%	28%	20%	14%	2%
23	308	2%	7%	19%	32%	24%	13%	2%
24	243	2%	3%	15%	36%	26%	14%	4%
25	301	3%	13%	24%	23%	24%	11%	3%
26	531	2%	13%	26%	25%	25%	9%	1%
27	626	1%	7%	24%	30%	25%	12%	1%
28	390	-	3%	16%	28%	30%	19%	4%
29	434	-	-	6%	35%	40%	15%	2%
30	544	-	1%	3%	29%	42%	21%	4%
31	511	-	1%	3%	27%	42%	23%	4%
32	245	-	-	3%	22%	46%	25%	4%
33	88	-	-	1%	17%	61%	18%	2%

Table 2 and 3 show that at e.g. 21 °C, 24 °C and 27 °C there is a difference in the distribution of thermal sensation votes between occupants in naturally and mechanically ventilated buildings. This difference is significantly ($p<0.05$) for all temperatures between 20-30 °C, a finding also described in [11]. It is evident that when temperature increases, the thermal sensation of the occupants also increases in both types of buildings, but at the indoor temperature 21 °C in NV buildings, 40% of the occupants was dissatisfied (voting -3,-2, +2 or +3 on the 7-point scale) compared to only 27% dissatisfied occupants in mechanically ventilated buildings. At 24 °C the number of dissatisfied is similar comparing the two different ventilation principles, 23% and 20% for NV and mechanically ventilated buildings, respectively. In warm indoor environments at 27 °C in mechanically ventilated buildings occupants are more dissatisfied than occupants in NV buildings, 48% dissatisfied compared to 21% dissatisfied, which corresponds with the rationale behind the adaptive model of thermal comfort [11].

Perceived air temperature

This data set allows establishment of a relationship between thermal sensation votes and temperature. This can be used to estimate occupants' perceived air temperature corresponding with a given vote on the thermal sensation scale. Figure 3 shows this relationship in mechanically and naturally ventilated buildings, respectively.

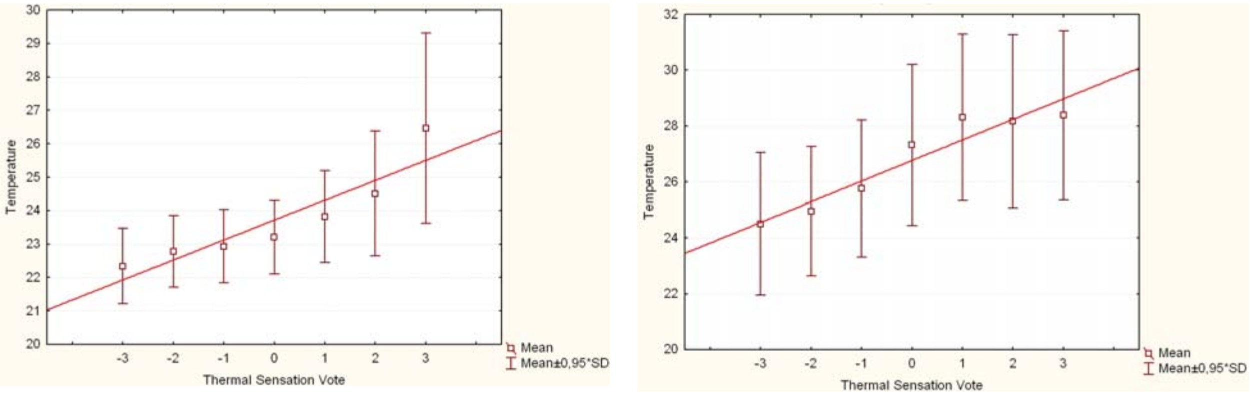

Figure 3. Relationship between office employees' thermal sensation votes and measured temperature in mechanically ventilated building (left figure) and NV buildings (right figure)

Figure 3 shows the mean temperature measured for each thermal sensation vote and the standard deviation. The mean temperature perceived as neutral (the average temperature measured for occupants who voted 0 on the thermal sensation scale) in mechanically ventilated buildings was 23.2 °C and 27.3 °C in naturally ventilated buildings. A linear regression line is used to describe the relationship between thermal sensation and temperature. E.g. for occupants in mechanically ventilated buildings voting 2 (warm) on the sensation scale the corresponding perceived air temperature will be 24.9 °C.

Table 3 shows an example of the combination of the thermal distribution at 24 °C for occupants in mechanically ventilated buildings and the corresponding perceived air temperature (the point on the regression line from Figure 3).

Table 3. Perceived air temperature for occupants exposed to 24 °C indoor environment

Thermal sensation vote	-3	-2	-1	0	1	2	3
Thermal distribution	2%	10%	23%	33%	24%	7%	-
Perceived air temperature	21,9 °C	22,5 °C	23,1 °C	23,7 °C	24,3 °C	24,9 °C	25,5 °C

From table 3 it is seen that 33% of the occupants in mechanical ventilated buildings who voted 0, may perceive the air temperature as being 23.7 °C, but 7% perceive the temperature as 24.9 °C. When calculating the effects of temperature on mental performance, occupants are affected differently even when exposed to the same temperature, due to the differences in thermal sensation between individuals. When the temperature varies the distribution of thermal sensation changes, thus by calculating this weighted effect of temperature on performance hourly over a year, an expression of the total yearly performance can be estimated.

DISCUSSION

In order to estimate the effects of temperature or other indoor climate parameters on mental performance of office work, the distribution of individual perceptions may be used as input to the calculations. With thermal sensation it is well established that not all people feel neutral at 22 °C, but as seen from this study e.g. 14% of occupants in NV buildings sense 22 °C as “warm”. This could be due to behavioral or psychological differences, a variation that normally occurs between occupants in offices around the world. Over an entire population occupants voting “warm” on the thermal sensation scale, would perceived the indoor air temperature to be 28.2 °C. Thus this is the temperature that will affect the mental performance.

By using a relationship between mental performance and air temperature established in e.g. [2], the distribution of thermal sensation votes and their corresponding perceived air temperature, a weighted effect of temperature on mental performance of office workers can be calculated. Since some occupants will perceive temperatures that differ from the optimal performance point of view, it will not be possible to reach 100% performance for a group of occupants even under thermal conditions that are acceptable to a majority of the occupants.

Future challenges are to integrate hourly temperature output from building simulation programs and a relationship between mental performance and temperature with the developed distribution and perceived temperature in a design decision tool. This tool could make it possible to compare different building designs in relation to investments in improvement of the thermal indoor climate and the corresponding effects on occupants’ mental performance.

CONCLUSION

The distributions developed in this paper can be used for future estimation of the effects of temperature on mental performance. Occupants do not perceive the temperature or other indoor parameters uniformly. The approach suggested in this paper allows individual differences to be accounted for when assessing effects on performance of indoor climate, as it varies during the year.

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We would like to thank Richard De Dear for providing the data used in this paper. It is now possible to download the datasets from the internet [11]. We would also like to acknowledge Professor P.O Fanger for his pioneering work in the field of thermal comfort.

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Effect of Overcooling on Productivity Evaluated by the Long Term Field Study

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SUMMARY

A long-term field study was conducted in an office for seven months to clarify the effect of thermal environment on productivity. Seven male software programmers participated as subjects. They answered the questionnaires on their personal computer including thermal sensation votes, complaints about their working environment, fatigue, vitality, mental workload and self estimated productivity every workday. The numbers of keystrokes were automatically recorded during the operation of their computers. Advanced Trail Marking Test (ATMT) was also conducted as the estimation of performance. Under lower temperature, the percentage of dissatisfied about “Low temperature” and “Draft” were higher and the vitality level was lower. The decrement in the total number of keystrokes of one day was 7.8% and that in the numbers of reaction per second of ATMT was 2.6% as temperatures dropped by 1.0°C below thermally neutral temperature. The result showed that overcooling brought the decrement of performance in the office.

INTRODUCTION

The objective of this study was to clarify the effect of thermal environment on productivity in the actual office. Especially, the effect of overcooling on office workers’ productivity was discussed in this study. A field study was conducted in a programmers’ office for seven months. The air temperatures were measured near the subjects’ desks. The relationships of air temperature, subjective vote and their performance were analyzed.

It is not simple to quantify the effects of thermal environment on performance of actual office works objectively. In this study, the numbers of keystrokes of the programmers were automatically recorded during their work time as their performance. Advanced Trail Marking Test (ATMT) [1] which was the standardized performance test that requires searching and clicking on the target character from twenty-five randomly appeared alphabets on their computers’ screen, was also asked to subjects.

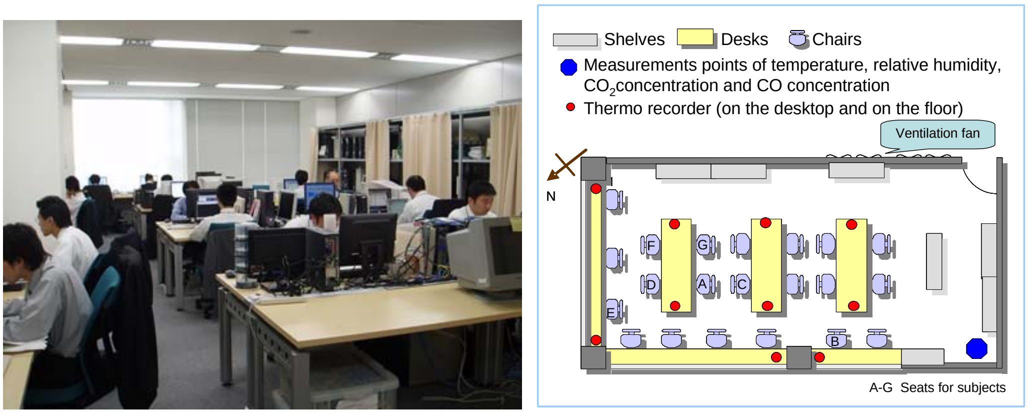
METHODS

The field study was conducted from July 2005 to January 2006 in the programmers’ room. During the field study, the office was cooled by air-conditioner. The setting temperature of air-conditioning unit during the survey is shown in Table 1. The scene of the programmer’s room and the floor plan of the programmers’ room are shown in Figure 1.

Air temperature, relative humidity, the concentrations of CO_2 and CO were measured 10-minute intervals by IAQ -Calc Model 8762 (TSI) during the survey. The concentration of CO_2 was 577±135ppm and that of CO was 2.6±0.4ppm. Air temperature and relative humidity near the subjects' seats were measured. At each with measurement point, the environmental factors at the height of desktop and floor were measured 10-minute intervals by thermo recorder TR-72U（T and D）. The measurement points are shown in Figure 1.

Table 1. The setting temperature of air-conditioning unit

Setting temp. (°C)	4-8 Jul.	11-15 Jul.	19-22 Jul.	25-29 Jul.	1-5 Aug.	8-12 Aug.	22-26 Aug.	29 Aug. -2 Sep.	5-6 Sep.	7-9 Sep.	12-30 Sep.	3 Oct. -18 Jan.
Working hour 10:00-17:00	21	23	24	22	24	24	26	26	25 -28	25	Un-kno wn	24
Others	19	21	22	19	22	21	24	25		24		23



(a) (b)

Figure 1. (a) Scene and (b) the floor plan of the programmers' room

In the room, ten to twenty programmers were working depending on the type and scale of their projects. In this study, seven male software programmers, with their age of 28.4±2.9, height of 168.1±3.5cm, and weight of 57.6±6.6kg, participated as the subjects. They were in their suits as usual and they were allowed to adjust their clothing.

They answered the questionnaires on their computer including thermal sensation votes, complaints about office environmental quality, fatigue and vitality at the beginning and the end of their office hours. They also answered mental workload and self estimated productivity at the end of their office hours every workday.

The voting items are shown in Table 3. Thermal sensation was voted on scales. Complaints on other indoor environmental elements were reported by checking the applicable items listed in "Complaints about indoor environment".

As the scale for subjective evaluation on mental workload, National Aeronautics and Space Administration Task Load Index (NASA-TLX) was proposed by Hart and Staveland [2]. The Japanese version of NASA-TLX [3] was used. NASA-TLX consists of six components: 'mental demand', 'physical demand', 'temporal demand', 'performance', 'effort' and 'frustration level'. The subjects indicated these components by a mark on each scale. The leftmost end was 'good (0)', and the rightmost end was 'poor (100)' for 'performance' scale. The leftmost ends were 'low (0)', and the rightmost ends were 'high (100)' for the other scales. For a comprehensive evaluation of the mental workload, Raw TLX (RTLX) proposed by Miyake was used. RTLX were calculated by averaging ratings of six components.

To evaluate the feeling of fatigue, 'Evaluation of Subjective Symptoms of Fatigue' suggested by the working group for occupational fatigue of the Japan Society for Occupational Health [4] was used. 'Evaluation of Subjective Symptoms of Fatigue' has been used in the field of labor science and ergonomics in Japan. It consists of 30 terms of fatigue symptoms. 'General rate of complaints' was defined as the rate of complaints about all 30 symptoms and calculated by equation (1) for each subject.

Rate of complaints (%)

$$= \frac{\text{Total number of a corresponding fatigue symptom of each subject}}{\text{Total number of symptoms on the evaluation sheet}} \times 100 \qquad (1)$$

To evaluate the vitality level, each subject listed up 20 items that they want to do on their free time before the survey, and they filled in whether they want to do them at the time of voting. Along with 'Evaluation of Subjective Symptoms of Fatigue', the vitality level was defined as the rate of applicable items about respective 20 items.

Subjects also reported the self estimated performance by marking a scale. The scale was the estimation on how subjects thought their productivity at work would be enhanced or interfered by the environment near their seat (+100: enhance their productivity, 0: interfere their productivity).

Table 3. Voting items

<table>
<tr><th>Subjective votes about indoor environment</th><th>Mental workload</th></tr>
<tr><td rowspan="7">●Thermal sensation

-3 Cold | -2 Cool | -1 Slightly cool | 0 Neutral | 1 Slightly warm | 2 Warm | 3 Hot

●Complaints about indoor environment
(check the applicable items)
□High temp. □Stuffy air
□Low temp. □Humid air
□Adjustability of temp. □Dry air
□ Nonuniform temperature □Not enough air velocity
□Not enough lighting □Unpleasant odor
□Glaring □Distracted by other people
□Too much lighting □Draft
□Noise □Dust and dirt</td><td>NASA-TLX Japanese Version</td></tr>
<tr><td>The feeling of fatigue</td></tr>
<tr><td>Evaluation of subjective symptoms of fatigue</td></tr>
<tr><td>The feeling of vitality level</td></tr>
<tr><td>Beforehand subjects list up 20 items what they want to do on their free time. And they fill in whether they want to do them at the time.</td></tr>
<tr><td>Self estimated productivity</td></tr>
<tr><td>The estimation on how subjects thought their productivity at work would be enhanced or interfered by the environment near their seat (+100: enhance their productivity, 0: interfere their productivity).</td></tr>
</table>

For the performance of work, their numbers of keystroke were automatically recorded during the operation of their computers, since much of their works involve in typing. For the analysis, the data for ten minutes at the beginning and the end of the operation time of their computers were eliminated to exclude the possible keystrokes for responding questionnaire.

ATMT was also conducted at the beginning and the end of office hours as the estimation of performance. In this test, subjects were asked to search and click on the target character "R" in twenty-five alphabets on their computers' screen. The distributions of the alphabets are presented randomly and repeatedly 70 times on their screen. For the analysis, the numbers of reaction per second at the end of their office hours were calculated.

The air temperature at the height of desktop near each subjects were averaged from ten o'clock to eighteen o'clock. The data sets of air temperature by the interval of 0.5°C were averaged, and corresponding subjective vote and their performance were averaged. The size of the plots shown in figures describes the number of corresponding subjects. For analysis of correlation between the air temperature and corresponding subjective vote and their performance, Pearson's product moment correlation coefficients weighted by the number of subjects were used.

RESULTS

Thermal sensation vote

The relationship of air temperature and thermal sensation vote with the linear regression model obtained from the plots is shown in Figure 2. Thermally neutral temperature was derived from the regression model as the temperature at which the thermal sensations vote becomes zero. The temperature was 28.6°C in this study. The reasons that thermally neutral temperature was relatively high were: air velocity in the room was 0.2-0.4m/s and was relatively high for offices, and subjects were able to adjust their clothes.

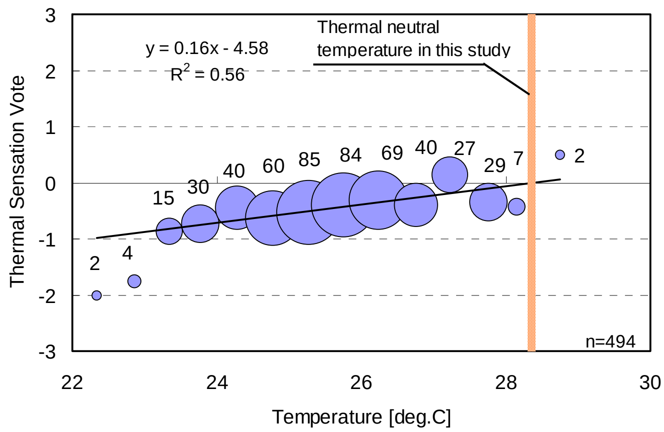


Figure 2. The relationship of air temperature and thermal sensation vote

Complaints about indoor environment

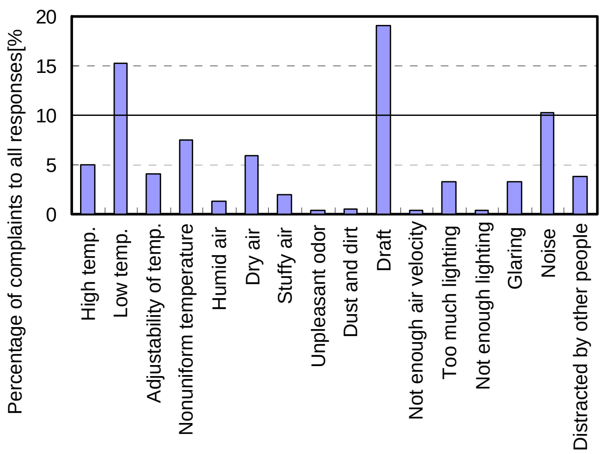
The percentages of complaints to all responses are shown in Figure 3. The percentage exceeded 10% were "Low temperature", "Draft" and "Noise". The equivalent sound level of the office was 60dBA, which it might have lead to the high rate of complaints on "Noise". Figure 4 shows the breakdown of these data for each air temperature range. Complaints about "Low temperature" and "Draft" were high under 23.0°C of air temperature in this room.

Mental workload

The relationship of air temperature and RTLX is shown in Figure 5. There was no significant correlation between air temperature and RTLX.

Fatigue

The relationship of air temperature and general rate of fatigue is shown in Figure 6. There was no significant correlation between air temperature and general rate of fatigue.



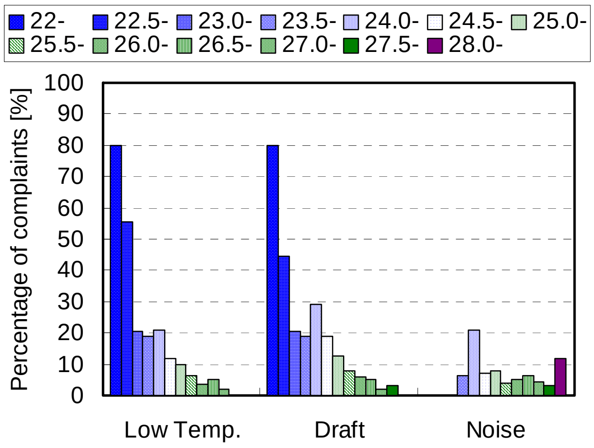
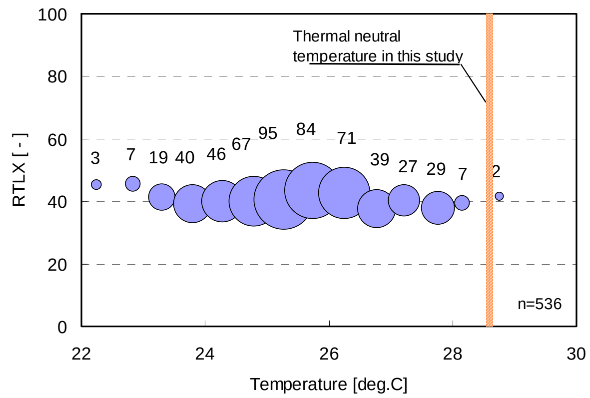


Figure 3. Percentage of complaint to all responses

Figure 4. Percentage of complaints for each air temperature range



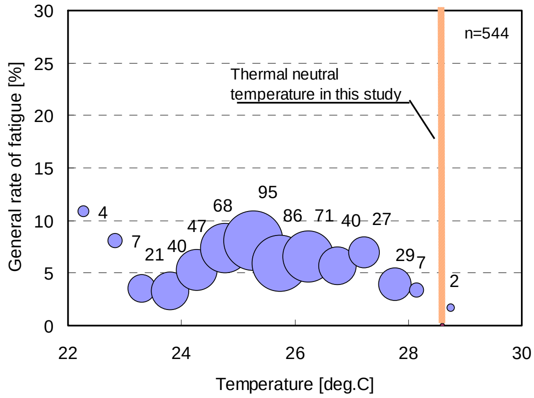
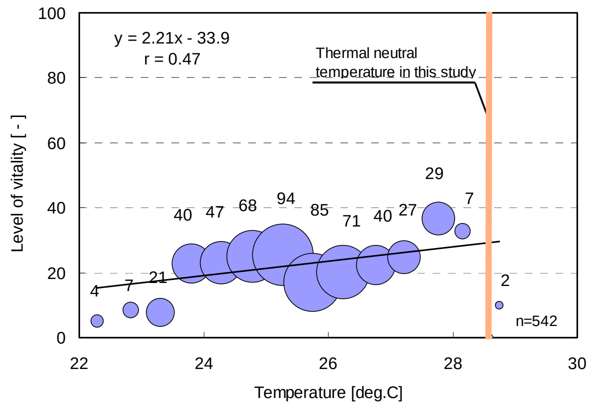


Figure 5. Air temperature and RTLX

Figure 6. Air temperature and General rate of fatigue

Vitality level

The relationship of air temperature and the level of vitality is shown in Figure 7. The correlation coefficient of air temperature and the level of vitality was 0.47. From the result, the vitality level decreased when the air temperature decreased from the thermally neutral temperature.



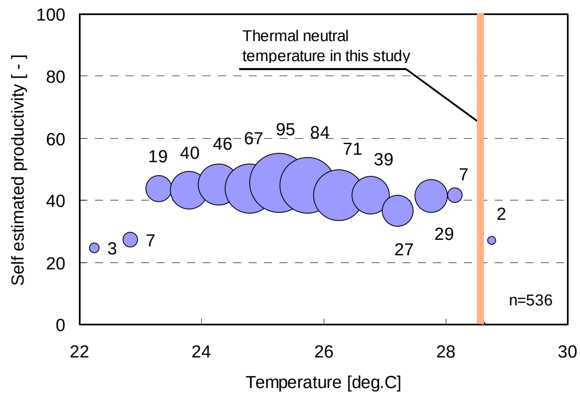


Figure 7. Air temperature and vitality level

Figure 8. Air temperature and self estimated productivity

Self estimated productivity
The relationship of air temperature and self estimated productivity is shown in Figure 8. There was no significant correlation between air temperature and self estimated productivity.

The numbers of their keystrokes per day
The relationship of air temperature and the number of keystrokes per day is shown in Figure 9. The correlation coefficient of this relationship was 0.70. From the linear regression model obtained from the results, decrease in 1.0°C of air temperature from thermally neutral temperature of 28.6°C corresponds to the decrement in the number of keystrokes per day by 7.8%.

The numbers of reaction per second of ATMT
The relationship of air temperature and the number of reactions per second is shown in Figure 10. The correlation coefficient of this relationship was 0.85. From the linear regression model obtained from the results, decrease in 1.0°C of air temperature from thermally neutral temperature of 28.6°C corresponds to the decrement in the number of reactions per second of ATMT by 2.6%.

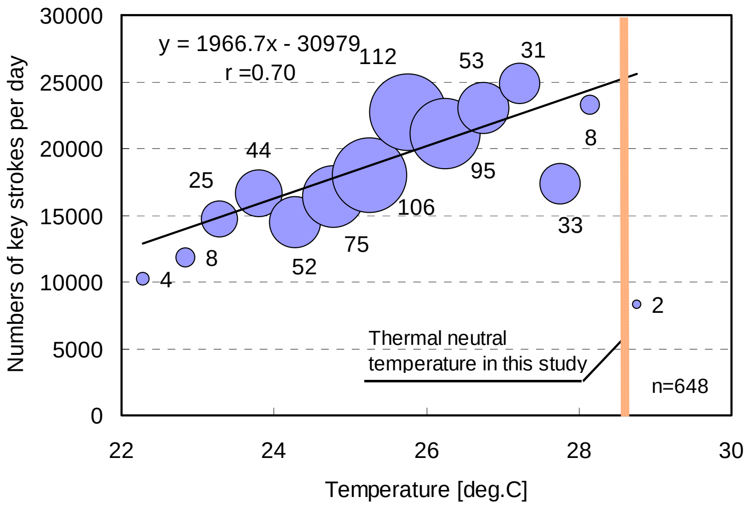


Figure 9. Air temperature and the number of keystrokes per day

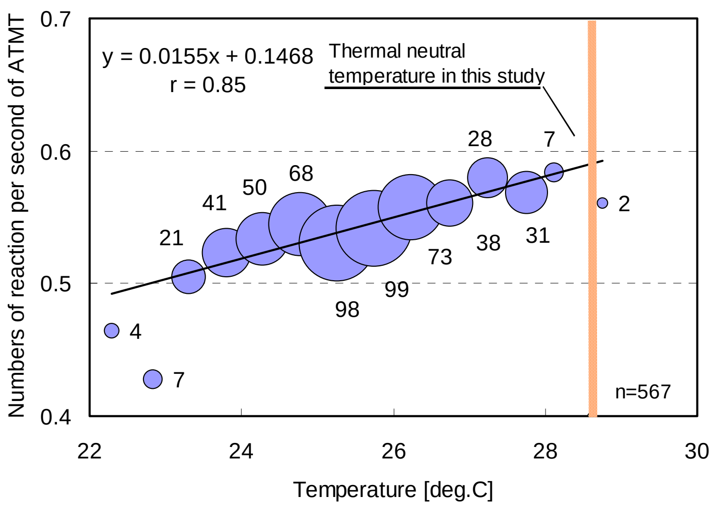


Figure 10. Air temperature and the number of reactions per second of ATMT

DISCUSSION

The results showed that the decrement in the total number of typing of one day was 7.8% and that in the numbers of reaction per second of ATMT was 2.6% as temperatures dropped by 1.0°C below thermally neutral temperature of 28.6°C. This result showed that overcooling brought the decrement of performance in the office. In our previous study of a year-long field survey of a call-center, the regression model of indoor air temperature and call response rate had shown that the rising of indoor air temperature by 1.0°C from 25.0 to 26.0°C would lead to the decrement in performance by 1.9% [5]. These studies showed the tendency that the deviation from the optimal temperature for office work lead to performance decrement. The relation of temperature and performance that had been developed by Seppänen et al [6] also showed this tendency.

It is important to specify the optimal temperature for work for creating the productive work place. However, it is not easy and the optimal temperature differed from one study to another. For example, Seppänen et al. mentioned that performance increases with temperature up to 21-22°C, and decreases at temperature above 23-24°C. On the other hand, in this study, thermally neutral temperature was 28.6°C and relatively high because of high air velocity in the room (0.2-0.4m/s) and the adjustability of clothes. Thermal environmental factors (not only air temperature, but also radiant temperature, air velocity, humidity, clothes and activity level) are needed for consideration. Additionally, the difference of task types may affect the optimal thermal environment for work. In this study, programmers' performances were estimated by the tasks, most of which requires the use of fingers such as keystrokes, and clicking response and they were relatively susceptible to cooler environment.

The results of this study and our previous call center study [5] showed the significant relationship between air temperature and performance at work. Tham et al. [7] and Niemelä et al. [8] also showed the significant relationship between air temperature and performance by the field studies in call centers. On the other hand, our previous laboratory experiment showed that it was difficult to evaluate the effect of thermal environment on productivity by measuring only task performance in the short-term exposure studies (about 2 hours) [9]. It was important to measure the human responses, such as fatigue level, physiological responses, and psychological responses, as the factors to influence workers' performance. For longer exposure, it became clearer that thermal environment affect physiological and psychological process such as fatigue, mental effort and so on, which would consequently affect workers' performance. The result from long-term exposure study (about 6 hours) had revealed that increase of fatigue level causes decrement in performance [10]. In the field study, relatively strong relationship of indoor air temperature and their performance could be obtained compared to laboratory experiments. The reason why the stronger relationship of indoor air temperature and their performance at the actual office work than that at laboratory experiments may be that the motivation level is higher in the short exposure studies and the human responses makes it hard to verify the direct relation between air temperature and performance.

CONCLUSIONS

A long-term field study was conducted in an office for seven months to clarify the effect of thermal environment on productivity. Seven male software programmers participated. The following results were obtained.

1) Complaints about "Low temperature" and "Draft" were high under 23.0°C of air temperature in this programmers' room.
2) The vitality level decrease when the air temperature decreased from thermally neutral temperature of 28.6°C.
3) The decrement in the total number of typing of one day was 7.8% and that in the numbers of reaction per second of ATMT was 2.6% as temperatures dropped by 1°C below thermally neutral temperature of 28.6°C.
4) The result showed that overcooling brought the decrement of performance in the office.

ACKNOWLEDGEMENT

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Development of Survey Tools for Indoor Environmental Quality and Productivity

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SUMMARY

A computer program for both field survey and laboratory experiment was developed to evaluate performance of office works and to link the findings from experiments to the actual offices. The program has three parts: namely, Voting Tool, Task Tool, and Performance Evaluation Tool. In this study, the Performance Evaluation Tool, which is consisted of twelve standard tests, was examined. The abilities required for the office works in two companies and one branch of the government office were collected to verify the applicability of the Performance Evaluation Tool. Two subjective experiments were conducted to determine the abilities required to work on each of twelve tests and to examine the reliability as a measurement tool for performance. From the results, there were some abilities required in offices but not for the tests, and the need for further modification was addressed. The performances of the tests were positively correlated with that of multiplication task (r=0.86).

INTRODUCTION

Performance is essential for the evaluation of productivity. Unfortunately, the performance of office works is mostly hard to quantify. To evaluate the effect of indoor environmental quality on performance quantitatively, mental tasks, such as three-digit multiplication, text-typing, and memorization of letters, have been assigned in the subjective experiments [1][2][3][4]. However, there is no guarantee that those simulated tasks are good representations of actual office works. Also, the importance of evaluation of human responses, such as fatigue, cerebral blood flow, sensation to the environment, was reported from the results of short-term experiments [5][6]. The survey tools developed in this study were to equip the quantitative measurement method of performance with the appropriate relationship to the performance of office works and the measurement method of human responses as used in the previous studies. The main purposes of this paper were to introduce this survey tools and to examine the characteristics and practicality of the Performance Evaluation Tool, which was developed as a part of the survey tools to measure performance quantitatively in both field surveys and laboratory experiments.

METHODS

First, a summary of the survey tools was briefly described and the tests of the Performance Evaluation Tool were introduced. Then, the questionnaire form was described with the categories of cognitive abilities, which were used to relate the Performance Evaluation Tool to tasks and office works. Also, the methods of this study, which includes questionnaire surveys on two companies and one branch of the government office and two subjective experiments, were explained.

Survey Tools for Indoor Environmental Quality and Productivity

PC-based survey tools were developed to evaluate the effect of indoor environmental quality on productivity. The tools are programmed as computer software with a mouse and a numeric keypad to respond. The tools include three types of evaluation tools, which include Voting Tool (V-Tool), Task Tool (T-Tool) and Performance Evaluation Tool (P-Tool). V-Tool is consisted of questionnaire forms including: votes on environment, Evaluation of Subjective Symptoms of Fatigue [7], and NASA-TLX [8], to evaluate human responses and fatigue. T-Tool is a set of experimental assignments such as multiplication tasks. P-Tool contains twelve standard tests as described in the next section.

Performance Evaluation Tool

P-Tool was developed to evaluate objectively the performance level of the persons in both field survey and laboratory experiment. The P-Tool was also rebuilt as web-based programs for the field surveys, because it has recently been difficult to directly install personally-built programs into the computers in offices for the security reasons. The P-Tool was installed into the PCs of the laboratory and was run without internet access in this study. It is consisted of twelve standard tests programmed with references from the Walter Reed Battery Tests [9] and examples given in the literature [10].The twelve tests are:

(I) Manikin – a manikin surrounded by circle or square carries circle and square on his hand appears on the screen. Subjects are to answer the side of hand that carries same or different shape compared to the shape surrounds manikin;

(II) Coordinate – based on the table of numbers with corresponding alphabets, subjects are to choose the coordinate point that are implied by the randomly selected combination of two alphabets;

(III) Coding - based on the table of numbers with corresponding alphabets, subjects are to input a number that is implied by a randomly selected alphabet letter;

(IV) Four choice – a panel with 1,2,4,5, corresponding to a portion of numeric keypad, is appeared with background color of one number is highlighted. Subjects are to input the highlighted number;

(V) Nine choice – similar to the Four choice but the panel with numbers from 1 to 9;

(VI) Time-lag input – subjects are to input the number that is appeared previously;

(VII) Arithmetic symbol – subjects are to answer the correct arithmetic symbol to complete the equation; appeared on the screen

(VIII) Positioning – subjects are to select the number as instructed from the randomly positioned 25 numbers appear on the screen;

(IX) Pattern memory – subjects are to answer whether randomly positioned dots appear twice are the same or different;

(X) Letter search – from the thirteen alphabet letters, subjects are to answer whether the target two letters are included both, only one, or neither;

(XI) Classification rule – subjects are to answer the rule that classifies the boxes with □■△▲○● as appeared; and

(XII) Pattern search – target pattern of sixteen mosaics with alternatives were shown. Subjects are to select the same pattern from the alternatives.

Cognitive abilities

In this study, the abilities required for the office works and P-Tool were in focus. The ability classification questionnaire was used in the questionnaire surveys and in the Subjective Experiment 1. The abilities and its definitions were cited from the Cognitive Abilities proposed by Fleishman [10]. The abilities include: (1) Oral comprehension; (2) Written comprehension; (3) Oral expression; (4) Written expression; (5) Fluency of ideas; (6) Originality; (7) Memorization; (8) Problem sensitivity; (9) Mathematical reasoning; (10)

Number facility; (11) Deductive reasoning; (12) Inductive reasoning; (13) Information ordering; (14) Category flexibility; (15) Speed of closure; (16) Flexibility of closure; (17) Spatial orientation; (18) Visualization; (19) Perceptual speed; (20) Selective attention; and (21) Time sharing. The abilities required to the subjects or workers for their works were marked with the ability classification questionnaire, which is shown in Figure 1.

Q1. Briefly explain the type of your work ________________________________.

Q2. Mark in the left box of the following abilities that are required for your work.

	Ability to understand spoken (English) words and sentences
	Ability to understand written sentences and paragraphs
	Ability to use (English) words or sentences in speaking so others can understand
	Ability to use (English) words or sentences in writing so others can understand
	Ability to produce a number of ideas about a given topic
	Ability to produce unusual or clever ideas about a given topic or situation
	Ability to remember information, such as words, numbers, pictures, and procedures
	Ability to know when something is wrong or is likely to go wrong
	Ability understand and organize a problem and then to select a mathematical method or formula to solve the problem
	Ability to add, subtract, multiply, divide, and manipulate numbers quickly and accurately
	Ability to apply general rules to specific problems and to come up with logical answers
	Ability to combine separate pieces of information, or specific answers to non-mathematical problems, or to form general rules or conclusions
	Ability to correctly follow a rule or set of rules specifying how to arrange things or actions in a certain order
	Ability to produce many rules in such a way that each rule tells how to group a set of things in a different way
	Ability to quickly make sense of information which initially seems to be without meaning or organization
	Ability to identify or detect a known pattern (e.g., a figure, word, or object) that is hidden in other material
	Ability to know one's location in relation to the environment one is in or to know where an object is in relation to oneself
	Ability to imagine how something will look when it is moved around or when its parts are moved or rearranged
	Ability to compare letters, numbers, objects, pictures, or patterns, quickly and accurately
	Ability to concentrate on a task over a period of time
	Ability to shift back and forth efficiently between two or more activities or sources of information

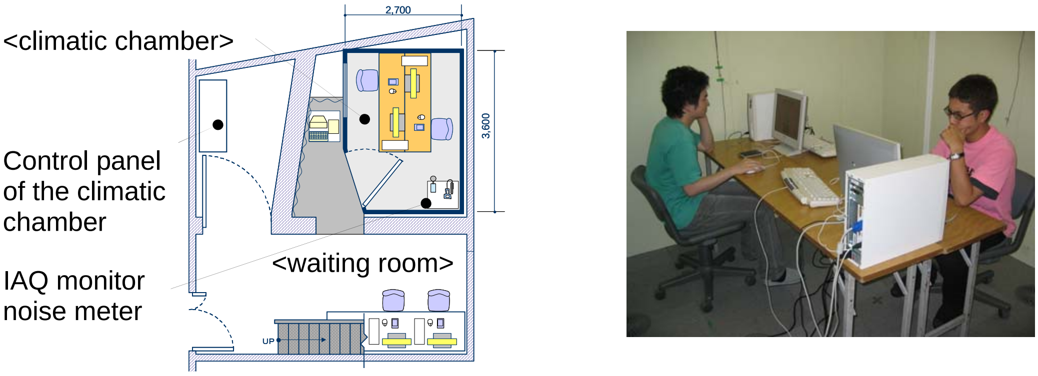
Figure 1. The ability classification questionnaire

Questionnaire surveys

To obtain the balance of abilities of the office works, the ability classification questionnaire had been collected in Company M, Company T and one branch of the government office. In Company M, 75 workers were surveyed; of 69 took over clerical job, of 3 took over technical job and of 3 had no response. In Company T, 50 workers were surveyed; of 46 took over technical job and of 4 took over clerical job. In the government office, 76 workers were surveyed; of 66 took over clerical job and of 10 took over technical job.

Subjective Experiment 1

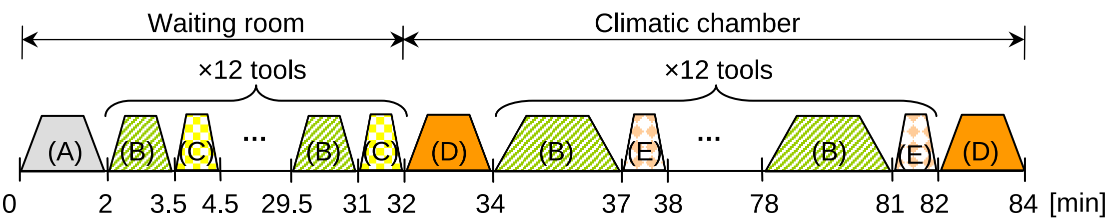
The purposes of the experiment were to gather information to modify the operationality of P-Tool and to understand the characteristics of P-Tool by obtaining the abilities required for each test. Thirty-one college-aged subjects, eight female and twenty three male, participated in the experiment. Subjects wore their own clothes and were allowed to adjust their clothes during the experiment. The experiment was conducted on July 26-28, 2006 in a climatic chamber and the waiting room. The layout of the chamber with the waiting room and a picture during the experiment were shown in Figure 2. Two subjects were paired and participated only once in this experiment. The physical environment for the climate chamber was measured with IAQ monitor for air temperature, relative humidity, the concentration of CO_2, and noise meter for sound level. The conditions of the climatic chamber were as follows: average air temperature was 25.4°C with standard deviation of 0.1°C; average relative humidity was 50% with standard deviation of 0%; the concentration of CO_2 was 576ppm with standard deviation of 55pppm; and equivalent sound level was 56dBA.



(a) (b)

Figure 2. (a) The layout of the climatic chamber and the waiting room; and (b) a picture during the Subjective Experiment 1 in the climatic chamber.

Procedure of the experiment is shown in Figure 3. In the waiting room, instructions and procedures of the experiment were given to the subjects first, and the subjects worked on each test of P-Tool for 1.5 min and then reported on the evaluation sheets on the test. After all twelve tools were finished; subjects moved into the climatic chamber and reported the Evaluation of Subjective Symptoms of Fatigue. They again worked on each test of P-Tool for 3 min and then reported the ability classification questionnaire on the test. After all twelve tests were finished, subjects reported the Evaluation of Subjective Symptoms of Fatigue.



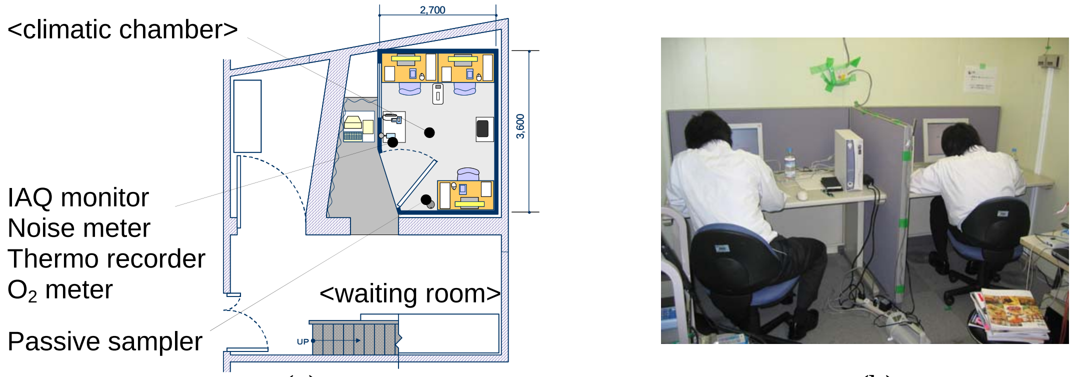
(A) instruction
(B) test of P-Tool
(C) evaluation sheets on the test
(D) Evaluation of Subjective Symptoms of Fatigue
(E) NASA-TLX and the ability classification questionnaire

Figure 3. Procedure of Subjective Experiment 1

Subjective Experiment 2

Another subjective experiment was conducted to evaluate the compositive effect of thermal environment and indoor air quality. From the perspective of evaluating the P-Tool, this experiment was conducted to examine the relationship between the performance of the P-Tool and multiplication task. The experiment was conducted in the climatic chamber as shown in Figure 4 with twelve male subjects from September 18 to October 19, 2006. Three subjects were participated together in this experiment. Subjects wore shirt with long sleeves and pants (0.66clo) [11] during the experiment. A practice and four experimental conditions were set: A) Practice with operative temperature of 28.5°C and Low ventilation rate; B) 28.5°C-Low ventilation rate; C) 28.5°C-High; D) 25.5°C-Low; E) 25.5°C-High. Subjects experienced

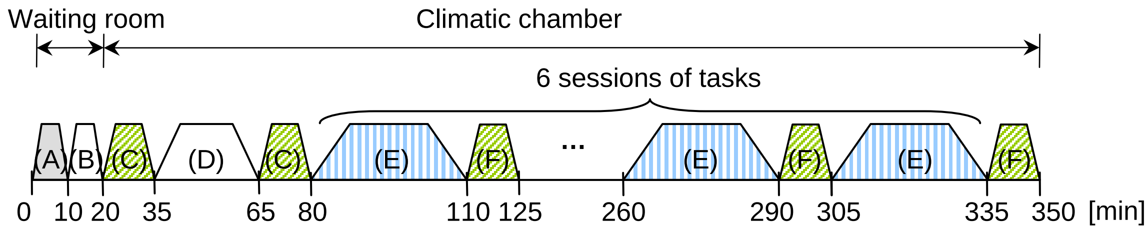
Practice first and then these four conditions in balanced order. They were exposed on the same time of the day in the successive weeks to avoid the influence of time. The tests of P-Tool used in this experiment were selected based on the results of the Subjective Experiment 1. Because one subject had fever in one of the conditions, his data was excluded for the analysis, i.e. data from eleven subjects were used for analysis in this experiment.



(a) (b)

Figure 4. (a) The layout of the chamber; and (b) a picture during the Subjective Experiment 2

The procedure of the experiment is shown in Figure 5. In the waiting room, subjects changed their clothes, instructions were given, and they rested in sedentary position for ten minutes. Then, the subjects moved into the chamber, reported on the environment and the Evaluation of Subjective Symptoms of Fatigue, and worked on P-Tool. After resting in sedentary position for thirty minutes, subjects voted on the environment, reported the Evaluation of Subjective Symptoms of Fatigue, and worked on P-Tool. Then six sessions of multiplication task with votes and P-Tool between the tasks were assigned to the subjects. After the sessions, they reported the predicted-performance-vote [12] at the end of the experiment.



(A) change clothes and instruction
(B) rest in waiting room
(C) votes on environment, Evaluation of Subjective Symptoms of Fatigue, P-Tool
(D) rest in climatic chamber
(E) multiplication task
(F) NASA-TLX + (C)
(G) NASA-TLX + (C) + predicted-performance-vote

Figure 5. Procedure of the Subjective Experiment 2.

RESULTS

Questionnaire surveys

The results of ability classification questionnaires were shown in Figure 6. Abilities were evaluated by the fraction of the number of workers who responded the ability to be required to the number of total workers. In these offices, (1) Oral comprehension, (2) Written comprehension, (3) Oral expression, and (4) Written expression were the abilities that were evaluated as "required" by over 85% of workers. The abilities such as (6) Originality, (11) Deductive reasoning, (12) Inductive reasoning, (14) Category flexibility, (16) Flexibility of closure, were required in Company T, in which most of workers took over technical jobs unlike the other two. The ability classification questionnaire can be a useful method for understanding the characteristics of the office.

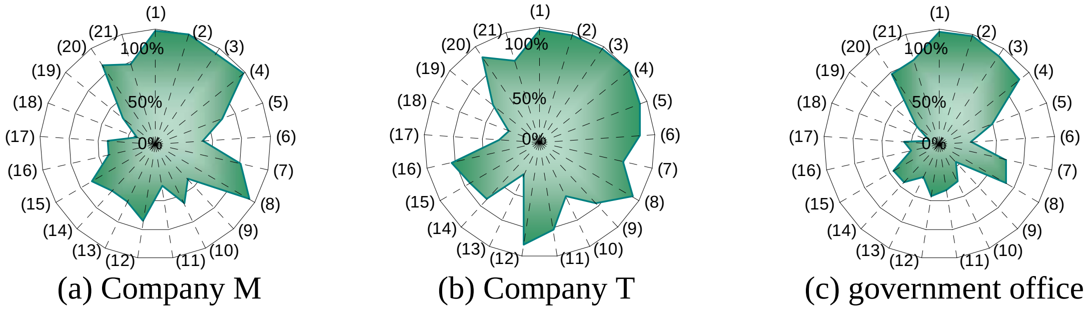


Figure 6. Results of ability classification questionnaire at (a) Company M, n=75; (b) Company T, n=50; and (c) government office, n=76. The numbers in parenthesis in the figures correspond to the abilities described in the section “Cognitive abilities”.

Subjective Experiment 1

The balance of the abilities required to work on each test are shown in Figure 7. (19) Perceptual speed and (20) Selective attention were the abilities that were highly required in all of the tests. Other than the two, the balance of the abilities required differed from test to test. There were some tests, however, that were similar in terms of required abilities. For the practical use in future researches, six tests were selected keeping the new set of tests to have similar balance of required abilities as a total. The selected six tests, together called P-Tool$_6$, were: (I) Manikin; (II) Coordinate; (IV) Four Choice; (VII) Arithmetic symbol; (XI) Classification rule; and (XII) Pattern search.

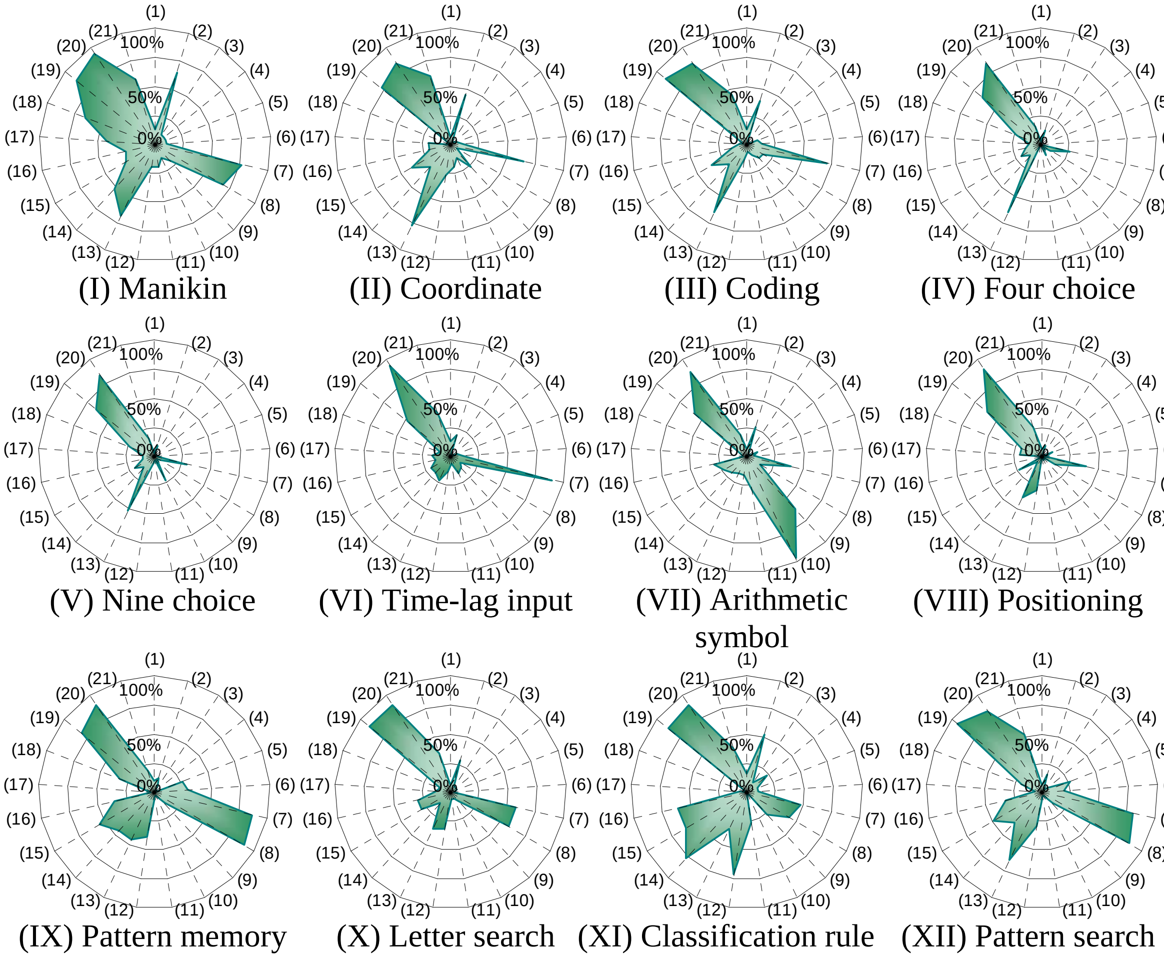


Figure 7. Results of abilities classification questionnaire of each test of the Performance Evaluation Tool. The numbers in parenthesis in the figures correspond to the abilities described in the section “Cognitive abilities”.

Subjective Experiment 2

To examine the reliability of the P-Tool, it was compared with the multiplication task. The results of ability questionnaire survey for the P-Tool$_6$ from the Subjective Experiment 1 and that for the multiplication task are shown in Figure 8. The balance of the abilities required for the P-Tool$_6$ covered that for multiplication task.

(a) P-Tool$_6$ (b) Multiplication task

Figure 8. Results of ability classification questionnaire for (a) P-Tool$_6$ and (b) multiplication task. The numbers in parenthesis in the figures correspond to the abilities described in the section “Cognitive abilities”.

Performances of both the P-Tool$_6$ and multiplication task were normalized to exclude individual variability by obtaining z-scores for each subject. The data set of performance of the P-Tool$_6$ with interval of 0.5 in z-score was averaged and corresponding performance of the multiplication task was averaged. The relationship between the two performances is shown in Figure 9 with a linear regression model weighted by the number of subjects. It was positively correlated with the correlation coefficient of 0.86. The result gave some proof on the adequacy of the P-Tool as a measurement method of the workers’ performance indirectly.

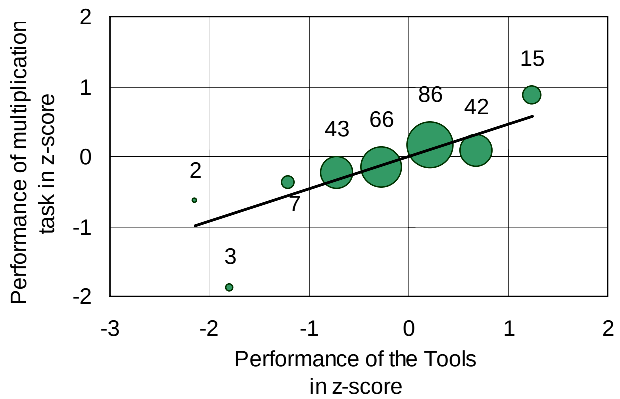


Figure 9. The relationship between normalized performance of the multiplication task and of the P-Tool$_6$, and a linear regression model weighted by the number of subjects.

DISCUSSION

Cognitive abilities required in offices and in the Performance Evaluation Tool

From the results of ability classification questionnaire in offices and Subjective Experiment 1, there are some differences in the abilities required between office works and the P-Tool. For office works, major abilities required were of (1) Oral comprehension, (2) Written comprehension, (3) Oral expression, and (4) Written expression, which were not evaluated as much for the P-Tool. Some of the methods to test these abilities have been introduced in the previous works [10]. However, they are not suitable for repetative use, for scoring objectively and automatically, to avoid uniformality of the difficulty of each question, etc., that are not negligible when the use in offices is assumed. Further modification of the P-Tool for these abilities may be needed.

Performance Evaluation Tool as measurement of performance
From the results of Subjective Experiment 2, the performance of the P-Tool was correlated well with the performance of multiplication task. There would be still some more studies needed on the tasks other than multiplication for conclusive evidences to convince that the P-Tool can be applied in the situation where direct performance is difficult to quantify. Also, the web-based version of the P-Tool needed to be practiced in the field.

CONCLUSION

In this paper, the survey tools for indoor environmental quality and productivity were introduced, and the characteristics and practicality of the Performance Evaluation Tool were examined.

1) The results from survey and experiment, there were some abilities required in offices but not for the tests, and the need for further modification was addressed.
2) The balance of the abilities required for the Performance Evaluation Tool covered that for multiplication task. The performance of the Performance Evaluation Tool was positively correlated with the performance of multiplication task.

ACKNOWLEDGEMENT

The authors express gratitude to the subjects of the experiments and the workers participated for the surveys. The authors also appreciate Miss M. Isemura, Miss K. Nakajima and Mr. S. Nakamura for their contributions to this study. This study was partially funded by the Global Environment Research Fund (H-061) by the Ministry of the Environment, Japan and by the Project Research of Advanced Research Institute for Science and Engineering, Waseda University.

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The cost of indoor climate systems in dwellings taking into account airflow rate, health and productivity

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SUMMARY

The life cycle cost (LCC) of the heating and ventilation system was simulated for a typical multi family dwelling and a typical detached house. Different ventilations systems were simulated resulting in different heating system designs. A health related cost dependent on the airflow rate which was based on recent studies was added to the life cycle cost. The health related costs are not analyzed but varied in a parametric study. An optimal airflow rate can be found depending on the assumed size of the health related cost. It is not well known how the airflow rate influences the health and comfort. Furthermore, it is not well known or defined how to valuate that possible influence. Despite this, this study is a demonstration of a method which can be used to optimize airflow rates in buildings if the influence and costs on human health related to the airflow rate is known.

INTRODUCTION

People spend up to 90% of their time indoors [1]. To ensure people's health and comfort when they are indoors, the indoor air quality and thermal comfort must be appropriate. An indoor climate system, consisting of heating, cooling and ventilation, serves this purpose [2]. The aim of the indoor climate system is to provide the building with a good thermal comfort and indoor air quality. An increasing number of studies have indicated a significant relationship between the ventilation airflow rate and health and productivity, in regards to offices, schools and dwellings.

In this study, the life cycle cost (LCC) of the heating and ventilation system was simulated for a typical multi family dwelling and a typical detached house. Different ventilations systems were simulated resulting in different heating system designs with the computer program ProLive [3]. A health related cost dependent on the airflow rate which was based on recent studies was added to the life cycle cost. A higher airflow rate results in a more expensive indoor climate system both regarding the life cycle cost and the initial cost.

No comprehensive literature review was done regarding the cost and health impacts from the ventilation airflow rate. Only theoretical buildings have been simulated, one typical multi family dwelling and one typical detached house. Field measurements were not an alternative, since it would be difficult to measure the life cycle cost, particularly for different airflow rates. Only mechanical ventilation systems were included.

METHODS

One problem with the design of an indoor climate system is that there has been a predominant focus on initial costs. A life cycle approach could improve both the energy and economic performance of the indoor climate system. The life cycle cost of a product is the sum of all costs related to that product over its entire life span. Future costs are discounted to the value of today, the net present value, by the use of a discount rate of interest.

The computer program ProLive was developed to calculate the life cycle costs for indoor climate systems in offices, schools and dwellings [3]. It handles thousands of combinations of heating, cooling and ventilation systems typically used in Sweden, and takes into account the initial costs for buying and mounting components through a power demand calculation, energy costs, maintenance costs, repair costs and costs for space loss due to system components. Costs are based on Swedish prices from Sektionsfakta [4], which is a known cost database for the building sector. Outdoor climate data is obtained from the computer program Meteonorm [5], which simulates outdoor climate data for the entire world. ProLive has features to model productivity and health costs based on indoor temperature and airflow rate even if the user has to provide the correlation data between the parameters and the costs. ProLive uses Swedish costs in SEK excluding VAT (25% value added tax). 1 SEK ≈ 0.14 US$ ≈ 0.11 € as of 2007-02-15.

In this study, the price of heat was set to 0.6 SEK/kWh and the price of electricity was set to 0.8 SEK/kWh excluding VAT. The discount interest rate was assumed to be 1% for electricity, 2% for heat and 3% for other costs representing a real price increase for heat and even more for electricity. An annual value of 800 SEK/m² was assumed for space loss. It was assumed that there is a 20% deduction from the initial costs for both air and hydronic components compared to Wikells [4]. A 40 year calculation period was used. The scrap value was assumed to be zero.

Buildings

None of the buildings were set up with cooling. Hydronic radiators were used for heating. Assumed data for the buildings are given in Table 1. The outdoor climate data are from Stockholm, Sweden. Table 2 gives the apartment areas, the window areas, the heat transmission areas and the number of people per apartment. The rooms with supply devices are the rooms that are not kitchens or bathrooms. Exhaust devices are located in the kitchen and in the bathrooms.

The multifamily dwelling was assumed to have four storeys with two one-room, two two-room, two three-room and two four-room apartments on each storey. It was assumed that there were two exhaust devices for each apartment, except for the four-room apartments where it was assumed that there were three exhaust devices. The building was assumed to be a medium weight construction.

The detached house was set up with two storeys. There were three exhaust devices on the bottom floor and two on the second floor while there were two supply diffusers or air inlets on the bottom floor and three on the second floor. It was assumed that four persons were living in the house.

Table 1. Data for the simulated buildings. Internal load means internal heat gain not from people. Heat transmittance is the average value for the building.

	Detached	Multi family		Detached	Multi family
Storey height	3 m	3 m	Chiller COP	2	2
Building length	12 m	43.3 m	Room temperature	22°C	22°C
Building width	8 m	12 m	Internal load presence	3 W/m²	3 W/m²
Total floor area	192 m²	2080 m²	Internal load abscence	1 W/m²	2 W/m²
Heat transmission area	288 m²	Table 2	Heat recovery eff.	0.8	0.8
Window area	15 m²	Table 2	Heat plant eff.	90%	0.9
Heat transmittance	0.25 W/(m²·K)	0.365 W/(m²·K)	Solar rad. trans.	0.5	0.4
Supply air temperature	18°C	18°C	Leakage at 50 Pa	0.8 l/(s·m²)	0.8 l/(s·m²)

Table 2. Data for the multi family apartment. The building was set up to consist of two of each listed apartments per storey and four storeys.

Nbr of rooms/apartment	1 room	2 rooms	3 rooms	4 rooms
Nbr of persons/apartment	1	2	3	4
Window area	3 m²	6 m²	8 m²	9 m²
Apartment area	30 m²	50 m²	80 m²	100 m²
Heat transmission area	34.2 m²	57.0 m²	91.2 m²	114 m²

Ventilation systems

Specifications of the indoor climate regarding thermal comfort and air quality are determined by requirements, recommendations, national regulations or by the building's user. This helps to simplify the design process of an indoor climate system. Usually, the minimum and maximum temperatures and the supply airflow rate are set depending on the activity in the building. In Sweden, there is a requirement of 0.35 l/(s·m²) [7].

The included systems in this study were exhaust ventilation system (E) with air inlets at the windows, exhaust ventilation system with a heat pump (EHP) recovering heat from the exhaust air to tap water and heating, and supply and exhaust ventilations system with heat recovery (SEH) with ceiling diffusers for supply air. A constant airflow rate was assumed. For the exhaust system with heat pump, it was assumed that half of the energy use for hot tap water, 30 kWh/(m²·year) [8], was covered, that is 15 kWh/(m²·year), plus half of the heating energy. For the supply and exhaust system, the heat recovery temperature efficiency was set to 70%.

Health costs

It is proposed that a health related cost can be expressed by Equation 1. The benefit from a raised airflow rate goes towards zero at high airflow rates. At zero airflow rate the equation is not valid.

The constants k_1 and k_2 must be determined in relation to the units of the health related cost, C_{health}, and the airflow rate, q. Two examples are given in this paper based on recent studies.

$$C_{health} = k_1 \cdot e^{-k_2 \cdot q}, \qquad (1)$$

Sick leave could reasonably be blamed on poor ventilation at home and not only at work. If it is assumed that the influence is the same, Johansson [3] gives an equation based on a compilation of other studies over the world [9]. If it is assumed that the labour cost is 200

SEK/h, the number of annual working hours is 1700, the calculation period is 40 years, the discount interest rate is 3%, the detached house is taken into account with four people in 192 m², k_1 becomes 774 and k_2 becomes 3.28 with [q] = l/(s·m²) referring to the floor area and [C_{health}] = SEK over the calculation period.

In the other example, Bornehag et al. [10] performed a case-control study on children in 390 Swedish dwellings and found that the prevalence of allergic symptoms decreased significantly with an increased ventilation airflow rate. The dwellings were sorted into four quartiles depending on the airflow rate of 0.44 to 1.43 air changes per hour at a median of 0.62 as reference case. The odds ratio increased constantly to 1.95 for the group with 0.05 to 0.24 air changes per hour with a median at 0.17. The mean air change rate from the study was within the air change rate quartile with an odds ratio of 1.14 compared to the reference case. This quartile had a mean air change rate of 0.38. The reported prevalence of allergic symptoms was 9.7% based on a former cross-sectional study. Allergy and asthma was reported in around half of the cases of allergy [11]. Jansson [12] determined a cost for asthma of 15919 SEK/(person·year) based on the OLIN-studies in Sweden. If it is assumed that the reported prevalence of allergic symptoms and asthma corresponds to the average mean air change rate of 0.38, the prevalence of asthma can be approximated to 5% at 0.38 air changes per hour. If the risk ratio is approximated with the odds ratio, the prevalence becomes 4.4% for an air change rate of 0.17 per hour and 8.6% for an air change rate of 0.62 per hour. If the house has a height of 2.4 m, k_1 becomes 10.9 and k_2 becomes 2.23 if C_{health} represents the expected prevalence of asthma in percent at a certain airflow rate, q, in l/(s·m²) referring to the floor area. 10.9 must be multiplied with 159.19 which is the cost per percent prevalence and 23.115 which sums 40 years with 3% discount interest and 4 due to 4 people per house and must be divided by the floor area of 192 m² to get C_{health} to represent health cost over 40 years per floor area. k_1 becomes 838.

RESULTS

Figure 1 gives the life cycle cost for the multi family dwelling and the detached house respectively split into components and parts as a function of the airflow rate, q, for a simulated exhaust ventilation system. The heating pipes do not change relative to the airflow rate, while the radiators become larger to be able to heat the supply air. Figure 2 gives the same for the supply and exhaust ventilation system with heat recovery. For the detached house, there is a new brand of air handling unit installed at 1.2 l/(s·m²). Figure 3 shows the resulting life cycle cost for the exhaust ventilation system with heat pump. Space loss is the space that is taken up by ducts and air handling unit. At some points the cost increases in large steps due to the available component sizes on the market.

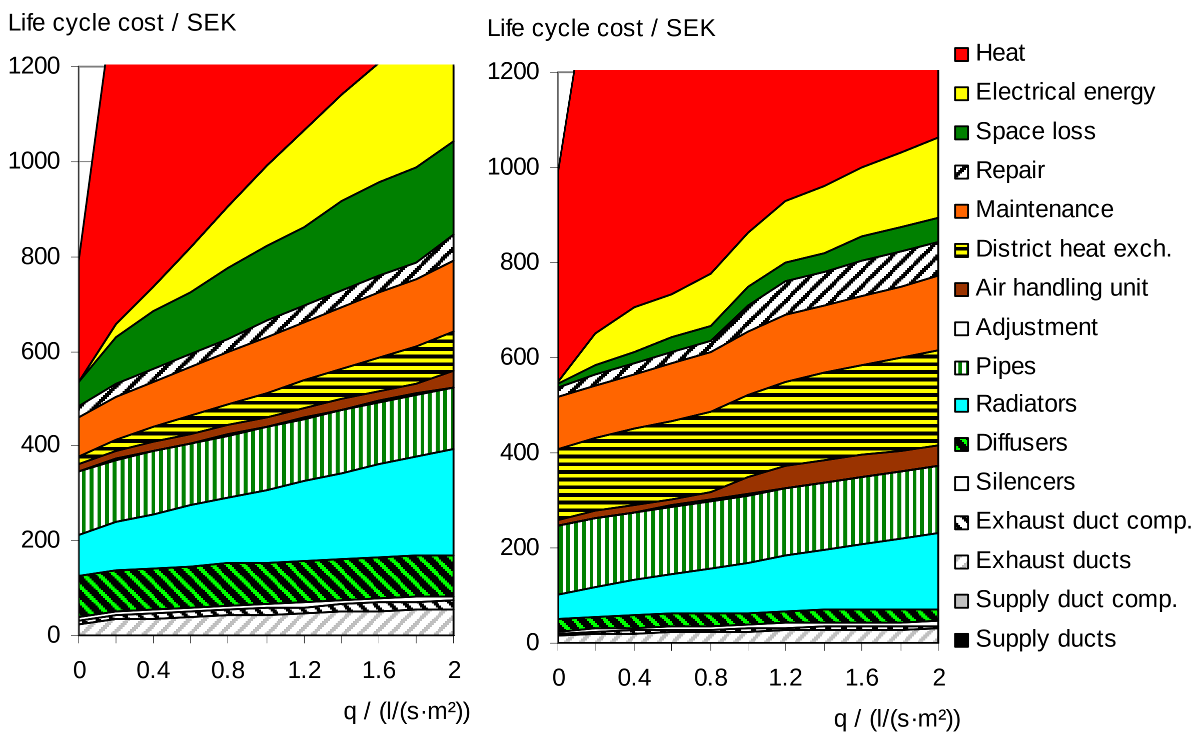

Figure 1. The life cycle cost as a function of the airflow rate for the multi family dwelling, to the left, and for the detached house, to the right, with exhaust ventilation system (E). There are no supply duct components or supply ducts within this system.

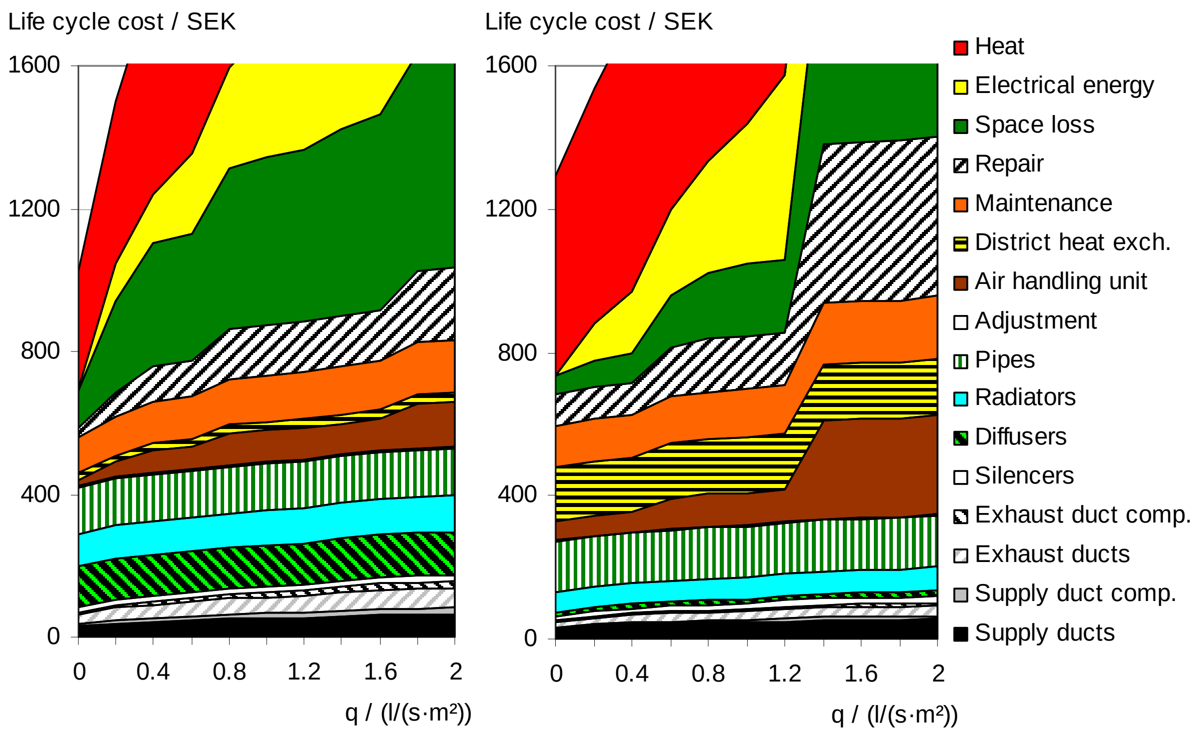

Figure 2. The life cycle cost as a function of the airflow rate for the multi family dwelling, to the left, and for the detached house, to the right, with exhaust ventilation system (E).

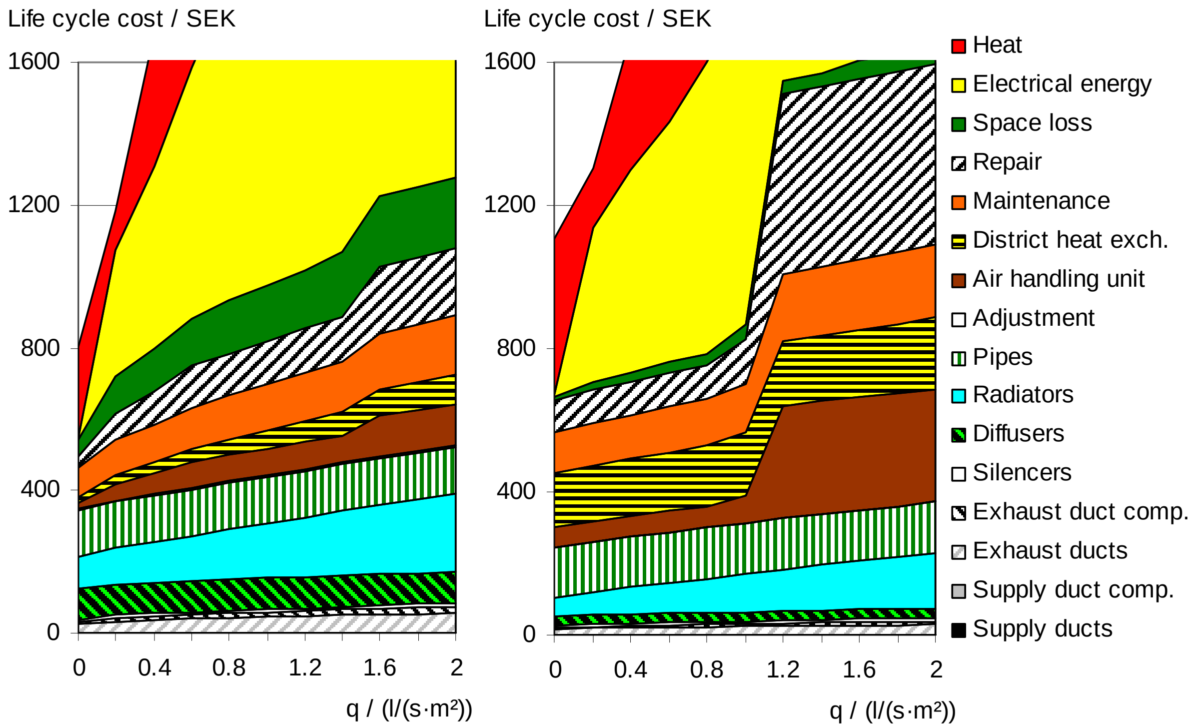

Figure 3. The life cycle cost as a function of the airflow rate for the multi family dwelling, to the left, and for the detached house, to the right, with exhaust ventilation system (E). There are no supply duct components or supply ducts within this system.

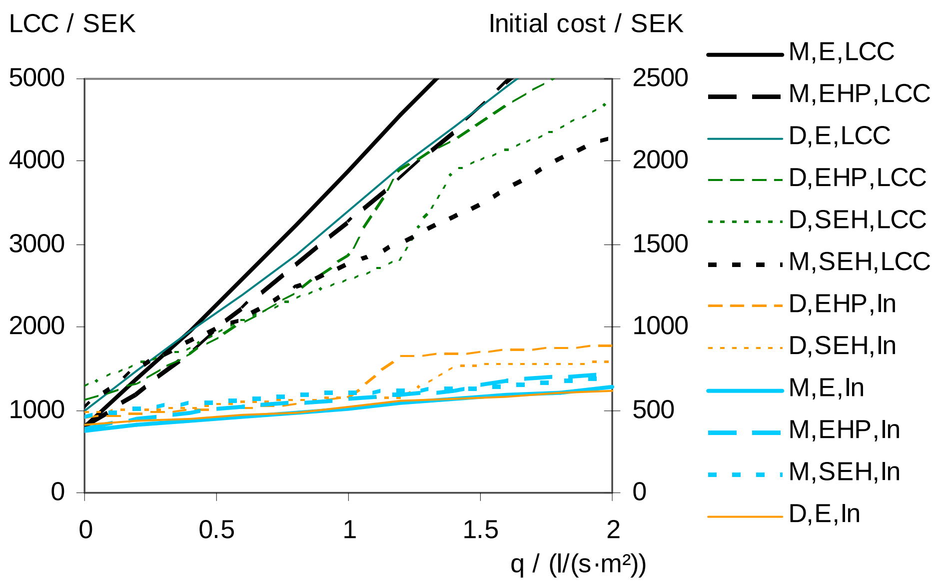

Figure 4. Life cycle cost (LCC) and initial cost (In) per floor area for the simulated buildings. E means exhaust ventilation, SEH means supply and exhaust ventilation, EHP means exhaust ventilation with heat pump.

Figure 4 shows the resulting life cycle cost for both buildings and all three systems together with the initial cost. In Figure 5, the health related cost according to the given examples is added for the detached house with exhaust ventilation and supply and exhaust ventilation. In Figure 6, k_1 is varied with k_2 = 2.23 and the optimal airflow rate is shown for the detached house.

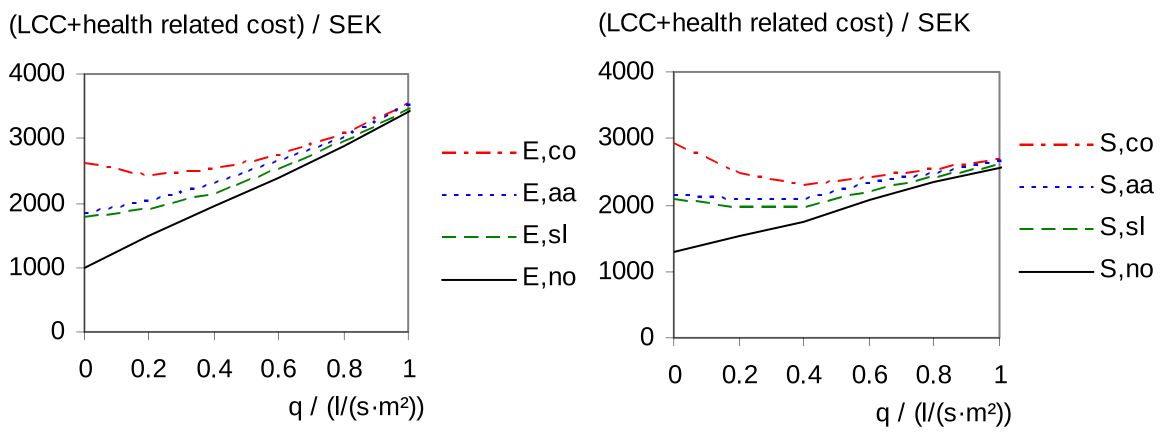

Figure 5. Life cycle cost (LCC) including health related costs for the simulated detached house. E means exhaust ventilation, S means supply and exhaust ventilation. The sick-leave example is denoted sl, the asthma example aa and the combination cost of the sick leave and asthma example co.

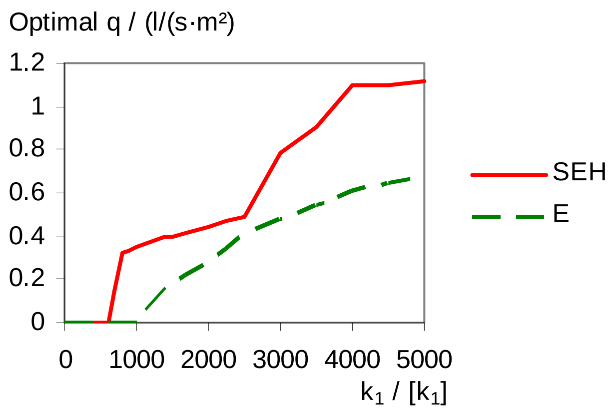

Figure 6. Optimal airflow rate, q, resulting in lowest sum of system life cycle cost and health related cost with k_2 = 2.23 for exhaust ventilation (E) and supply and exhaust ventilation (SHE) for the detached house.

DISCUSSION

The presented life cycle costs for heating and ventilation systems of multi family dwellings and detached houses show that heat is the largest part of the life cycle cost except for the system with heat pump where heat comes from electricity. The initial cost of the systems in dwellings is not negligible. It is also shown that supply and exhaust ventilation has a lower life cycle cost than exhaust ventilation at required airflow rates. The exhaust ventilation with heat pump is even lower even if exhaust systems have problems with controlling the air movements and draughts. There is also a problem with knowing the ratio between electricity and heat price in the future, which can make the heat pump unprofitable.

If the assumed health related costs is added to the costs of the heating and ventilation systems, an optimum is found in some cases. A combination of the two examples of health related costs is realistic and results in a lowest life cycle cost for supply and exhaust ventilation with an airflow rate slightly above the required 0.35 l/(s·m²). In fact, Bornehag [10] indicates that the airflow rate is lower than required, which is no surprise. If so, it means that the airflow rate should be increased in Swedish dwellings. The examples of health related costs does not take into account possible productivity increases and only takes into account the illness asthma. Even if asthma has high prevalence, there could be a number of other illnesses which costs money and are influenced.

With higher airflow rates, the relative humidity indoors would decrease which would decrease the risk of moisture problems, which in turn can be suspected to decrease costs for repair and for illnesses influenced by moisture problems. On the other hand, with higher airflow rate there is a risk of having air that is too dry during cold winter days.

A feature of the ventilation system would be to use a variable airflow rate which is controlled by the occupancy level. That means that the assumed benefit would be the same, the initial cost would be the same since the designed airflow rate is the same, but the energy use would decrease which in term would increase the optimal airflow rate.

Parametric studies of parameters, such as discount interest rates, building data or system data, would give different results, but most likely, the results would be rather close and representative for this kind of building. The health related costs are not analyzed in this study but varied in a parametric study. It must be noted that it is not well known how the airflow rate influences the health and comfort. Further, it is not well known and defined how to valuate that possible influence. Despite this, this study is a demonstration of a method which can be used to optimize airflow rates in buildings if the influence and costs on human health related to the airflow rate is known.

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Ventilation and building related symptoms

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SUMMARY

Due to criteria for building energy efficiency today's buildings are better insulated and the envelope is more air tight. These improvements have led to a more comfortable buildings and lower running costs. However, the new indoor environments are more dependent on controlled ventilation system thus, the role of ventilation is emphasised.

Impurities in indoor air can originate from different sources: e.g. materials, structures, adjacent zones such as crawl spaces or attics or ventilation systems. Ventilation and pressure differences as well as leakage routes are key factors in understanding the concentrations of impurities in indoor air and their penetration to indoor air. This paper reviews the current understanding of the relationship between ventilation and its effect on sick building syndrome.

INTRODUCTION

In epidemiological population studies, moisture damage and microbial growth in buildings have been associated with a number of health effects, including respiratory symptoms and diseases and other symptoms [1-4]. The health effects associated with moisture damage and microbial growth seem to be consistent across different climates and geographical regions [5-8].

Microbiologically clean buildings probably do not exist, as some contamination in the structures begins to build up even during the construction phase. In the case of moisture damage, even after repairs, a significant amount of contaminated materials can remain in structures [9]. Therefore, the potential transport of pollutants from and through structures has great importance in respect of the design and construction of buildings. It is important to know which factors affect the release and penetration of pollutants and possible measures, such as sealing and pressurising, that can be taken to avoid this.

Ventilation does not directly affect on occupant health or perception outcomes, but the rate of ventilation affects indoor environmental conditions including air pollutant concentrations that, in turn, may modify the occupants' health or perceptions. The air pollutant concentrations in a given space depend on several factors other than air flow rate.

In steady state conditions where the ventilation air is well mixed to indoor air the concentration of indoor air can be calculated from simple balance equations. In practice, however the situation is not that simple. Typically pollution source is not uniform and mixing not complete thus, the concentration of pollutants in breathing zone may vary significantly depending on the air distributions pattern and the locations of the pollutant sources. Also the indoor pollutant generation rate (source strength) is usually not constant and has very high variations. Pollutants may be adsorbed by room surfaces during high concentration periods. Therefore non emitting surfaces may be emitting sources. Indoor pollutant source strengths are highly variable among buildings, and considered the biggest cause of the variation in pollutant concentrations among buildings [10].

In many buildings, ventilation rates are not constant. Often old residential buildings are running with a natural ventilation thus, the climatic conditions are strongly affecting on ventilation rates. Typically in office buildings ventilation systems may not operate at night or are operated at reduced level, and the rates of ventilation during operation may change with internal heat loads or with outdoor air temperature. Since the reduced night ventilation is usually created only by using exhaust fans, the pressure difference over the building is also dramatically changing between day and night. Pollutant concentrations may not reach equilibrium or they reach that until several hours after ventilation rates stabilize. Thus, the indoor air quality is also dependant on the operating schedule of the ventilation system.

In addition to above mentioned factors, the concentration of pollutants in indoor air is affected by following other major forces: the level and type of pollutants outdoors, possible recirculation of return air, the location of the outdoor air intake relative to outdoor air pollution sources including exhaust air outlets, pollution sources in the air handling unit, and pollutant removal from supply air filters, sorbents or deposition on duct surfaces. Therefore with a same ventilation rate indoor air quality may vary significantly due to variations in the quality of the supply air [11,12].

VENTILATION, PRESSURE DIFFERENCE AND PENETRATION OF IMPURITIES

Residential buildings often have mechanical exhaust ventilation, where intake air comes through inlets and cracks. In cold climates, inlets will cause a draught in winter and they are often closed, resulting in a high under-pressure indoors and forced airflow through cracks. Typically, mechanical exhaust ventilation creates an under-pressure of 5-10 Pa in the apartment [13] if the inlets are open. A field study [14] showed high air flow rates through leaks in the base floor to the apartment at a pressure difference of around 6 or 15 Pa, depending on the speed of the exhaust fan. Field measurements [15,16] have shown evidence of fungal spores transported indoors from a crawl space. According to a literature review by [4], concentrations of viable fungi in the indoor air of residential buildings vary between 10–100,000 cfu/m^3. Although the range is wide, average levels of 100–10,000 cfu/m^3 are typical. Lower levels have been reported in winter not only in a cold climate [17] but also in a subtropical climate [7].

Concentration of fungal spores and ventilation system

Many studies have proven that the concentration of microbes outdoors is clearly higher in summer than in winter. However, there is only a very limited number of reports on concentrations in the crawl spaces or attics, even though impurities might be transported from there as well. Especially crawl spaces could have favorable conditions for fungal growth especially in the summer, when the outdoor air is usually warmer and has a higher moisture content, which is transported via ventilation to the crawl space. The extra moisture condensates to the surfaces in the crawl space giving better conditions for microbes in the surfaces.

The concentrations of viable fungal spores in the crawl spaces were a few thousand cfu/m^3 [16]. There are no guidelines or sufficient reference data in Finland for the concentration of fungal spores in crawl spaces. The high concentrations of fungal spores in the air under the floor may exposure occupants to fungal spores or fungal metabolites if there are cracks or holes between the spaces.

In subartic climate, in summer the concentration of fungal spores indoors is usually smaller than the concentration outdoors, whereas in winter the indoor/outdoor –ratio often is over 1 in the apartments as well as in the offices [18]. Generally outdoor air is the main source of fungal spores and the seasonal variation is wide; three to four order of magnitude in subartic climate (Finland), [e.g. 19,16] .

Indoor/outdoor –ratio had been higher in buildings with mechanical exhaust ventilation than in buildings with mechanical supply and exhaust ventilation [e.g. 19]. In a study [16] such a relationship couldn't be found. A reason for this might be that in this study all the measured buildings with mechanical supply and exhaust ventilation were primary schools or day-care centres, and the measurements were done during the working day and the influence of outdoor air to indoor air concentrations could not be avoided. Also the clothes of the children might be a potential carrier of spores [20].

Transport of fungal spores indoors

It has been found out in many studies that in indoor air particle concentration is highly influenced by outdoor particle concentration. Also the type of the particles is similar. According to [16] the total concentration of fungal spores did not correlate statistically between indoor and crawl space but higher concentrations in crawl spaces could be seen elevated concentrations indoors as well. Crawl space concentrations of fungal spores were tens or thousands of times higher than indoors.

However, when specific species of fungal spores were examined a relationship was found in Acremonium which is not typical microbe indoors, indicating presence of high counts in crawl spaces reflecting as high counts in indoor air.

The size of a particle has an important role in respect of penetration of a crack in a building envelope, thus, it is important to study the size distribution of fungal spores to estimate their penetration indoors. The size distribution is varying depending on microbial species. According to [16] fungal spores analysed from indoors, outdoors and crawl space were similar in shape when size distributions were compared. The highest counts were impacted (Andersen) in stages whose average aerodynamic diameter was between 1.4-2.6 μm. This size range is interesting as the alveolar deposition of particles above 0.5 μm has a maximum at about 3 μm. Therefore, even small changes in particle size around this maximum value effect on the deposition pattern of particles [21].

There are many factors affecting the size of fungal spores such as age, dehydration, agglomeration and relative humidity of surrounding air. In a study [21] it was found out that the most common fungal spores, such as *Penicillium, Cladosporium, Aspergillus* and yeasts, have their maximum concentrations in the size range 2.1 – 3.3 μm. In this study, the maximum counts were observed in smaller fungal spores, between 1.4 – 2.6 μm, indicating that the fungal spores are mainly detected as single spores and not as aggregates.

Air change rate and air flow patterns

In [22] it was found out that the levels of airborne microbes were higher in naturally ventilated office buildings than in offices with mechanical ventilation. In a wide review by [23] it was found out in that there is a strong evidence between ventilation and the control of air flow directions in buildings and the transmission and spread of infectious diseases such as measles, TB, chickenpox, anthrax, influenza, smallpox, and SARS. However, they didn't found data to support the specification and quantification of the minimum ventilation requirements in schools, offices and other environments in relation to the spread of airborne infectious diseases.

Particles and their penetration indoors

It is not only the moisture and microbes which can cause problems, but also exposure to fine particles. According to recent studies particles smaller than 2.5 μm in diameter can reach the lower respiratory system and cause adverse health effects [24,25]. However, usually the limit value for respirable particles is considered between 2.5-5 μm. Comite European de Normalisation (CEN 1992) and International Organization for Standardization (ISO 1992) both define the limit value for respirable particles 4 μm. It is still unclear what causes the health effects of such particles, but it seems that the type of particle has some role. According to studies [26,27] sand dust is not so harmful as particles of the same size originating from the combustion of fuels. About 70-90% of the viable fungi in indoor air have been estimated to be in the respirable size fraction [7]. The median aerodynamic diameter for fungi in indoor air is typically 2-3 μm [28]; the highest concentrations of airborne viable fungi are also usually in the same size range.

A number of studies have focused on properties of ambient particulate matter in recent years. Elevated concentrations of aerosol particles have been associated with increases in mortality as well as adverse health effects [29] Even though the particulate matter outdoors and its penetration indoors is very important topic, it is typical for all kind of buildings and not specific to buildings with SBS symptoms.

Many studies have been carried out to estimate the penetration of particles through cracks. [30] report penetration factors of 0.5-0.8 for particles in a size range of 0.5-2.5 μm. [31] found that at a pressure of 5 Pa 40% of 2-μm particles and <1% of 5-μm particles penetrate through horizontal slits of a height of 0.5 mm. In a study by [32] particles of 0.1-1.0 μm are predicted to have the highest penetration efficiency, nearly unity for crack heights of 0.25 mm or larger at a pressure difference of ≥4 Pa. These results are important, since the peak of the size distribution of fungal spores is often between 2-3 μm and is very suitable for penetration. [33] reported that Sulfur-containing particles of sizes below 1.5 μm penetrated a office building's filter unit in ventilation channel easily, elemental removal efficiency of the filter being only 7%. Only a limited number of studies have been carried out on those structures commonly used in buildings. [32] predict, by a simulation model, that penetration through mineral wool insulation is negligible. In reality structures mineral wool insulation is seldom perfectly installed, and timber frame structures are, in respect of particle transport, a combination of cracks, surface contacts, and mineral wool.

Ventilation rate of a building affects indoor particle concentrations, and as the fluxes from indoor to outdoor and outdoor to indoor are typically different, indoor air might become contaminated from other sources than outdoors, such as sources from crawl space or from contaminated building materials. Indoor concentrations are reduced by deposition and increased by re-emission of deposited material. The amount of deposited material depends e.g. on deposition rate, re-emission rate and the cleaning frequency.

Recent studies on particle deposition to indoor surfaces make it clear that the deposition rate varies broadly across conditions. Particle size is undoubtedly important. However, other factors can also influence the deposition rate significantly including interior furnishing (quantity and type of the surface) and the indoor air movement. According to [34] the air movement from zero to 0.142 m/s (which is a typical air velocity indoors) increased the deposition rate by 15% for 1 μm particles and by 24% for 1.9 μm particles. Furnishing had a great impact deposition rate; deposition rate increased over 100% for 1 μm particles at air velocity 0.054 m/s and 78% for 2.5 μm if the empty room was furnished [34]. The highest deposition rates had been reported mainly from field settings [e.g. 30], where the control of experimental settings is limited. Even in the best conditions it is difficult to isolate deposition from the many competing factors that can influence airborne particle concentrations.

In studies [30-32], penetration factors of particles within range of 0.5-2.5 μm have been between 0.5 and 1 depending on the dimensions of studied cracks. These results cover the size distribution of fungal spores; and the peak of spores of 1-2 μm seems to be very capable for penetration if the crack height is higher than 0.1 mm [32]. [32] have estimated that the penetration through mineral wool insulation is negligible. In [16] study mineral wool was inside a real base floor structure, and still the results showed penetration occurred, due to the installation of mineral wool which is seldom perfect allowing some routes for particles. The penetration factors determined by [16] were significantly different than in former studies indicating that surface contacts of mineral wool and other building elements are likely to have important effect on the penetration. The importance of surface contacts seems to be confirmed by tests; e.g. for 1.3 μm particles the penetration factor at 20 Pa was 0.19 when the boards didn't have any penetration and 0.15 when the four 10 mm holes on the surfaces were open, when the pressure was decreased to 6 Pa the penetration factors were 0.08 and 0.03 respectively. Thus, the holes on the surface boards did not have any effect on the measured penetration of particles but probably the small leakage routes between surface contacts of mineral wool and other building elements.

[35] measured particle and fungal spore penetration through a wooden structure in laboratory. They found out that the penetration was roughly the same within particle size range of 0.6-2.5 μm. The penetration was highly dependant on the pressure difference on the and not that much on the holes in surface boar of the structure (the structure included mineral wool layer under the structure acting as a filter).

Particle emissions from HVAC components

Fibre emission from HVAC components such as round silencers can be remarkable pollutants in the indoor air. In a field study [36] particle emission from HVAC components and occupational exposure were studied in 10 office, school and day care centres. The surface fibre dust density varied in the range of < 0.1 – 4.9 fibres/cm^2 and the accumulation during two weeks period in the range of < 0.1 – 2.6 fibres/cm^2. Fibre concentration in the supply air varied between 0.01-85 fibres/cm^3(fibre length >20 μm). In the study the silencers were also studied in laboratory but no relationship between silencers surface area to the particle emissions could not be generally be drawn but it was feature of different silencer. Detailed results from that study are presented by Kovanen in this Clima2007 congress.

Ventilation rate and VOCs

There are many studies in which the VOC concentrations are evidently lower in buildings with higher ventilation rates. In United States new buildings were measured, built with low-VOC and conventional building products [37]. In that study the air change rate varied between 0.14 to 0.78 ach, TVOC concentrations varying between 1.5 to 2.7 mg/m^3. In a Finnish study by [38] “Allergy House” building with 1.7 ach had the TVOC concentration 10-fold lower than in reference building with 0.8 ach. However, also the materials in allergy building were low emitting, thus, the low concentrations are not only because of higher ventilation rate.

In a recent study by [39] there was not a clear trend showing the concentrations of VOCs being lower in buildings with higher ventilation rate. In the newly finished buildings, the TVOC concentration was significantly lower ($p<0.01$) in the buildings with mechanically supply and exhaust air than in the buildings with mechanical exhaust air. The TVOC and formaldehyde concentrations were also lower ($p<0.01$) in the 6- and 12-month-old buildings with combined mechanical ventilation.

DISCUSSION

Strikingly, there are only a few studies in which the ventilation system of the measured building was mentioned and even less reports of the ventilation rate. However, the ventilation rate and pressure differences as well as leakage routes of the studied building/zone are key factors in understanding the concentrations of impurities in indoor air and their penetration to indoor air. Impurities can arrive indoor air from materials and their surfaces or adjacent zones such as crawl spaces or attics. Also the filters and their contamination and its influence indoors are not well known.

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Effect of household specificity on exposure time to CO_2 when balanced ventilation systems are used

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SUMMARY

The purpose of this study is to evaluate the relevance of household characteristics on exposure time to CO_2 in houses with balanced ventilation. Statistical tests were conducted to measure the significance of household differences on time indoors. Household scenarios and ventilation scenarios were constructed to determine the level of CO_2 indoors. The production of CO_2 was calculated with the metabolic rate per age of the members of the household. Households are exposed to levels of CO_2 higher than 1000 ppm in houses with low infiltration rate. In practice, the air flow does not comply with the requirements because of the use of the system and for lack of maintenance. High levels of CO_2 (over 5000 ppm) were not shown to be a risk in this study, but exposure to levels of CO_2 between 1000 and 2000 is used as an indicator of other pollutants indoors.

INTRODUCTION

CO_2 emissions in the indoor environment may have negative effects on the occupants. The effects produced by exposure to CO_2 have been studied extensively in the past [1,2]. Under concentrations less than 5000 ppm, CO_2 is not particularly toxic, but is accepted as an indicator for the presence of other contaminants and therefore for indoor air quality. Concentrations above 1000 ppm are related to complaints like fatigue or headache. Some studies tend to show that certain subjects already show these complaints at concentrations as low as 600 ppm [1,2] In the indoor environment of dwellings, CO_2 emissions, as an indicator of air quality, may affect more vulnerable occupants, such as children, the elderly, and people with poor health condition, because they tend to spend more time in the dwelling [3]. In reducing such effects, ventilation has a prominent role. The volume flow rate of ventilation needed to keep the carbon dioxide concentration at the desired level depends on the production rate of CO_2 in the indoor environment. CO_2 levels depend on several factors closely related to household composition, size of household, age and activities of the dwellings occupants. Building regulations set minimal ventilation flow rates per squared meter but recent studies have shown that these minimal ventilation rates may not be sufficient for bedrooms [2]. This research aims to study if differences in households can lead to differences in exposure to CO_2, and therefore, to different requirements of ventilation flow rates.

In order to calculate the indoor exposure to CO_2 in Dutch dwellings, it is necessary to determine the time of exposure, the type of activities realized indoors and the sources of CO_2 found indoors. For this, the Dutch Time Budget Survey (TBO) was used. The exposure time was analyzed by statistical means in order to test the influence of occupants' and households' characteristics on the exposure to CO_2. Scenarios of exposure per household were constructed to test the relevance of differences on exposure time. CO2 production and needed ventilation flow rates for the different household scenarios are then modelled, calculated and compared.

METHODS

Household specificity

The data of the TBO were collected by means of a self-administered diary. Respondents were asked to record their daily activities and place of activities during a period of one regular week. Only groups of subjects with a sample large enough for the statistical analysis were used. Mean times were calculated for the total time indoors, the time spent in spaces of the house and the type of activity realized (table 1).

Table 1. Categories of activities

Category (Place and activity level)	Sub-category (Activities)
Kitchen (medium level)	Cooking
Living room (medium level)	Sporting, household work, caring for pets and plants, playing indoors, childcare
Living room (low level)	Eating, studying and writing, group activities, working at home, hobbies, telephoning, reading, watching TV
Bedroom (very low level)	Sleeping
Bathroom (low level)	Personal care

The survey collected information of 1813 respondents from 12 years and older, therefore, the exposure times of children below 11 years old is not taken into account. It was realized during regular days, so variation in exposure time in other days such as holidays is neglected. In the case of non-response, it can be attributed to lack of time due to a special activity (e.g. birthday), therefore it does not cause bias in the data collected [4]

Statistical Analysis

In the Kolmogorov-Smirnov and Shapiro-Wilk tests of normality, the distributions per place and level of activity were shown to be non-normal with a large percentage of the Sig. values falling below 0.05; though the histograms of some activities showed resemblance with a normal distribution, and the normal probability plots showed a reasonable straight line. Therefore, a non-parametric test was conducted to evaluate the significance of the differences in exposure time within the groups. The Kruskal-Wallis test showed very significant results, therefore adding the fact that the sample is large, parametric tests were conducted to identify the location of the differences in exposure time within the groups, as well as the magnitude of the differences. An independent-samples t-test was conducted for groups of gender, and ANOVAs for groups of age and household size. In order to determine the magnitude of the differences, the eta-squared per variable was obtained, considering an eta-squared of 0.02 as a small effect, 0.06 as moderate effects and 0.12 as large effect [5].

Differences in exposure time attributable to gender

An independent samples t-test (ANOVA) was conducted to analyse the relevance of the differences in exposure time in different places indoors that are attributable to gender. The test showed that there is no significant difference for time spent in the bedroom (p-value=0.379) and the bathroom (p-value=0.745). The differences are significant for the time spent in the living room (p-value=0.013) and the kitchen (p-value=0.00). The differences between time spent in the different spaces were shown to be small for the living room (eta-squared=0.01), bedroom (eta-squared=0.029), and bathroom (eta-squared=0.013). The differences for the time spent in the kitchen do show a large effect (eta-squared=0.16). The results of the t-test showed that there is a significant difference for low level activities (p-value=0.00) and high level activities (p-value=0.00). The differences in the mean of low level activities show a small effect (eta-squared=0.001), while high level activities showed to have large effect (eta-squared=0.18). In other words, gender affects principally the time spend doing high level activities in the kitchen.

Differences in exposure time attributable to household size

A one-way analysis of variance test was conducted for analysing the relevance of the differences in mean times within groups of household size. Subjects were grouped according to the size of the household they belong. The households of 3 persons were not further considered because of the lack of representation within the sample. The results showed that there is a significant difference for the mean scores of the time spent in the living room (p-value=0.00), bedroom (p-value=0.012), and kitchen (p-value=0.005). There is no significant difference for the time spent in the bathroom (p-value=0.72). An eta-squared calculation showed that the differences for time spent in bedroom (eta-squared=0.007), and kitchen (eta-squared=0.008) have a small effect; while the differences for the time spent in the living room (eta-squared=0.059) shows a moderate effect. Post Hoc comparisons showed that 2 person's households spend more time in the living room, followed by 4 person's households. Singles tend to spend less time in the living room. 4 person's households spend more time in the bedroom, while singles and couples spend less time. The time spent in the kitchen is lower for couples than is for singles and 4 person's households. Low and high level activities showed to be significantly different among the groups (p-value=0.00). Nevertheless, the eta-squared calculation showed only a small effect for low level activities (eta-squared= 0.058), and moderate effect for high level activities (eta-squared=0.035). Post Hoc tests showed that singles spend less time in high level activities than 4 person's households and couples. Couples are the households that spend more time involved in high level activities. Households of 4 persons and singles spend less time in low level activities than couples.

Differences in exposure time attributable to age

A one-way analysis of variance test was conducted to estimate the effect of age in variations of mean times in the indoor environment. Six groups were formed according to the age range of the subjects (Group 1: 12-24, Group 2: 25-34, Group 3: 35-44, Group 4: 45-54, Group 5: 55-64, Group 6: 65 and older). The results of the ANOVA for the time spent in different places showed a significant difference for the time spent in the living room (p-value=0.00), kitchen (p-value=0.00), and bedroom (p-value=0.00). The mean differences for the time spent in the bathroom did not show significant difference (p-value=0.095). An eta-squared calculation showed large effects in the differences on the means for time spent in the living room (eta-squared=0.20), and in the kitchen (eta-squared=0.15). The effect is moderate for the time spent in the bedroom (eta-squared=0.06). Post Hoc comparisons showed that the time spent in the living room and in the kitchen increases with age. The time spent in the bedroom also increases with age, except for group 1 (12-24 years), which tend to spend more time in the bedroom. Significant differences were found to have large effects for low level activities (p-value=0.00, eta-squared=0.26), and high level activities (p-value=0.00, eta-squared=0.13). The time spent in high activities increases with age. The time spent in low level activities increases with age, except for group 1 (12-24 years) which tend to spend more time than groups 2, 3 and 4; but not as much as groups 5 and 6.

The trends on time spent indoors are shown in table (2). The exposure time in all cases is higher for women and elderly people. Though there is a relation between household size and mean times, the variations do not show a clear trend. The time spent in the bathroom is considered not to be different for the different groups, and the time spent in the bedroom shows little variation.

Table 2: Trends in time spend in different rooms per occupants' characteristics

Place / activity level	Gender		Household size		Age range	
	Effect	Trend	Effect	Trend	Effect	Trend
Bathroom	-	-	-	-	-	-
Bedroom	-	-	Small	4>2,1	Moderate	Increases with age
Living room	Small	Females > males	Moderate	4,2>1	Large	
Kitchen	Large	Females > males	Small	4,1>2		
Low level activities	Small	Females > time	Small	2>4,1		
High level activities	Large	Females > time	moderate	2>4>1		

Based on these data, four different scenarios are identified that could cover the bandwidth of Dutch households. The household's scenarios are: 1a) single male adult 20-24 years old, 1b) single female elderly, 2a) 4 person's household composed by two adults and two children (12 years), and 2b) 4 person's household composed by two adults and two children (20 years)

Method for the calculation of exposure to CO_2

For this research we assumed that CO_2 is produced only by human beings. Other sources of CO_2 production like cooking hobs, boilers and plants are neglected. The CO_2 volume flow rate in a room qco2(m^3/s) can be described in steady state as:

qco2 = V.(Ci-Co)

Where Co is the CO_2 concentration of the fresh air supplied to the room (usually about 350 ppm) and Ci is the concentration of CO_2 in the air leaving the room. V is the volume flow rate of air. It is assumed that the quantity of air leaving the room is equal to the quantity of air supplied to the room. For Dutch living rooms and bedrooms of average dimensions steady state conditions are assumed to be achieved after 2 hours constant occupancy when the air change rate (ACH) is above 0,8 h-1 [2]. Once qco2 and V are known, it is possible to calculate the concentration of CO_2 in the room (Ci).

Calculation of the CO_2 production rate qco2

In general, energy production in living organisms is obtained from the oxidation of glucose, see equation 1.

$C_6\ H_{12}\ O6 + 6O_2 \rightarrow 6CO_2 + 6H_2O$ - 2843 KJ mol-1 glucose, (1)

Under aerobic conditions the energy consumption by this reaction is 2843 kJ per mol glucose. At the same time, 6 mol of CO_2 are produced. This means that the consumption of 474 kJ by the human body produces 1 mol of CO_2. With the specific volume of CO_2 being 22,4 m3/kmol, an energy consumption rate of 1.0 J/s will result in 4.7×10^{-8} m^3/s of CO_2.

The carbon dioxide production rate follows therefore from the energy consumption rate of the human body. This energy consumption rate is called metabolic rate and is dependent on the activity performed and the surface area of the body. The calculation of the metabolic rate is conducted according to [6]. It is assumed that the metabolic rates at a same level of activity and age are identical for men and women. The surface area of the body Adu (m^2) is calculated using the formula of DuBois [6].

Adu= 0.202.W0,245 .H0.725

where W is the body weight in kg and H the body height in m.

In table 3, calculated metabolic rates and CO_2 production rates are given as a function of age and activity.

Table 3: Metabolic rates and CO_2 production per age

Age	Weight (m)	Height (kg)	CO_2 production [m^3/s] per level of activity			
			Sleeping	Low	Middle	High
12	28	1.54	2.4 $x10^{-6}$	5.4 $x10^{-6}$	8.9 $x10^{-6}$	$12x10^{-6}$
20	74	1.84	4.1 $x10^{-6}$	9.3 $x10^{-6}$	15 $x10^{-6}$	21 $x10^{-6}$
35	70	1.7	3.8 $x10^{-6}$	8.5 $x10^{-6}$	14 $x10^{-6}$	20 $x10^{-6}$

Calculation of the concentration of CO_2 at room and dwelling level

The concentrations are calculated for a typical Dutch terraced house with three bedrooms. The Dutch Building Decree prescribes the following minimum flow rates given in table 4 [11]. The floor areas and the minimum ventilation flow rates for this dwelling calculated according to the Dutch building decree are summarized in table 5.

Table 4: Minimum flow rates

	Living area (bedrooms and living room)	Kitchen	toilet	Bathroom
Flow rate (m^3s^{-1})	$0,9x10^{-3}$ per square meter with a minimum of 7 $x10^{-3}$ total	$21x10^{-3}$ total	7 $x10^{-3}$ total	14 $x10^{-3}$ total

Table 5: Areas and minimum flow rates for typical Dutch house

	Living room + kitchen	Bedroom 1	Bedroom 2	Bedroom 3	Bathroom
Area (m^2)	37,1	9,9	9,7	6,0	6,6
Flow rate (m^3s^{-1})	33,4 $x10^{-3}$	8,9 $x10^{-3}$	8,7 $x10^{-3}$	7 $x10^{-3}$	14 $x10^{-3}$

To limit costs, balanced ventilation systems are mostly designed to meet the requirements on the minimum ventilation rates, but not more. Furthermore, the legislation prescribes that these minimum ventilation flow rates must be achieved by the system when it runs at its highest level (level 3). Balanced ventilation systems in dwellings are mostly equipped with switches that can be put in levels 1, 2 or 3. Studies by Hasselaar [7] and Soldaat [8] show that the systems are running most of the time in level 2 in which only 50 % of the capacity is available. At night time, the system mostly runs in level 1 due to noise pollution from the ventilator, with only 30 % of the capacity. Furthermore, studies have shown [9] a reduction of the capacity of 10% a year, due to a too low level of maintenance (cleaning). After 5 years, the maximum capacity is reduced to 65 % of the original capacity. The total amount of fresh air supplied to the dwelling is also dependent on the air-tightness of the construction. The air-tightness of Dutch new dwellings should not exceed 200 dm^3/s per 500 m^3 building volume. For the reference dwelling studied, the air flow rate should be lower than 200 dm^3/s, which is about 1.5 dm3/sm2. In the calculations shown in this paper, we used this value as the upper limit of the infiltration flow rate, referred as “high infiltration rate” in the graphics. In the Dutch mandatory energy performance norm [10], it is assumed that an air infiltration flow rate lower than 0.4 dm3/sm2 cannot be achieved. We used this value as the lower limit for the infiltration rate (referred as “low infiltration rate” in the graphics).

Four household scenarios and six ventilation scenarios are studied for the reference dwelling. The household scenarios were described earlier in this paper. The six ventilation scenarios are:

1. Non optimal use of a 5 year- old system: 50 % maximum capacity in living room, kitchen and bathroom, 30 % maximum capacity in bedrooms (night). The maximum capacity is 65 % of the mandatory capacity as given in table 4. With 1) low infiltration rate and 2) high infiltration rate

2. Non optimal use of a new system: 50 % mandatory capacity in living room, kitchen and bathroom, 30 % mandatory capacity in bedrooms (night). With 1) low infiltration rate and 2) high infiltration rate
3. Optimal use of a new system: 100% mandatory capacity in living room, kitchen, bathroom and bedrooms. With 1) low infiltration rate and 2) high infiltration rate

Calculation of exposure time

The time spend doing an activity (sleeping, low level or high level) is calculated from the statistical data for each room (living room, three bedrooms, kitchen en bathroom) and each person of each household type (1a, 1b,2a, 2b). The CO2 concentration per room per activity is calculated according to the method described here above, taking into account the number of persons present in the room. The exposure time per person per household is then obtained by adding the times of exposure to a certain level in the different rooms of the house. Four types of exposure levels are distinguished, according to the findings reported in the introduction. These exposure levels are above 2000 ppm, between 1000 and 2000 ppm, between 600 and 1000 ppm and lower than 600 ppm.

RESULTS

For single person households, scenarios 2 and 3 (fig 2 and 3) have a good performance; there is only a slight variation in exposure to CO2 levels between the optimal and non-optimal use of the new system. The exposure to higher levels of CO2 increases in dwellings with low infiltration rate and non-optimal use of 5 years old system (scenario 1.1). Scenario 1.2 (high infiltration rate and non-optimal use of old system) shows also a good performance for single person households (fig 1). For four person households, there is little difference between old and new ventilation systems when these are not optimally used. In dwellings with low infiltration rate, the exposure to levels of 1000 to 2000 ppm is very high. The exposure decreases in dwellings with high infiltration rate, in which the exposure is mainly to levels of 600 to 1000 ppm. The exposure decreases drastically for scenario 3 (fig 3), but the exposure is still high for households of 2 adults and 2 children of 20.

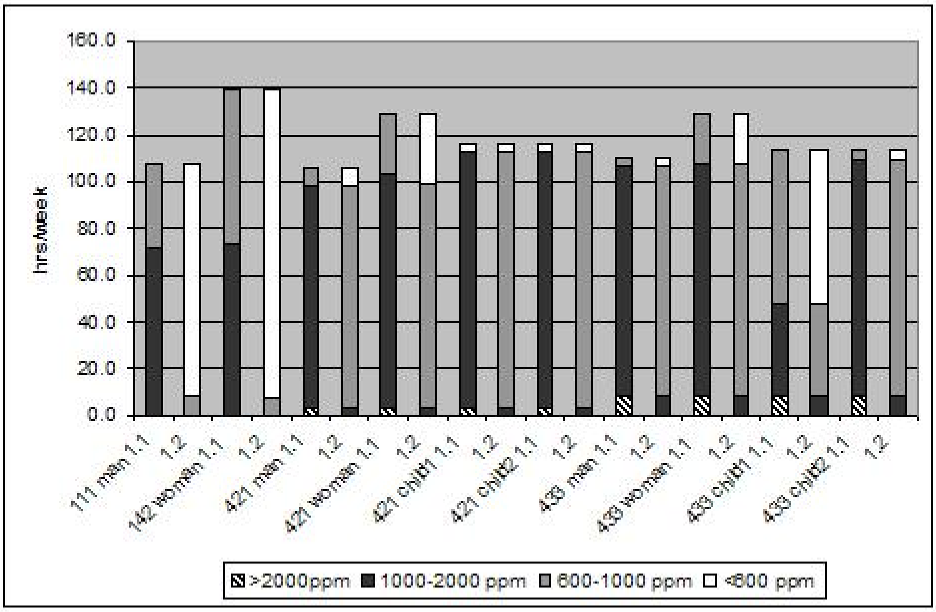

Figure 1: Time of exposure to levels of CO2 per occupant in ventilation scenario 1.1) airtight dwelling, and 1.2) no airtight dwelling, child1= bedroom 2, child2= bedroom 3

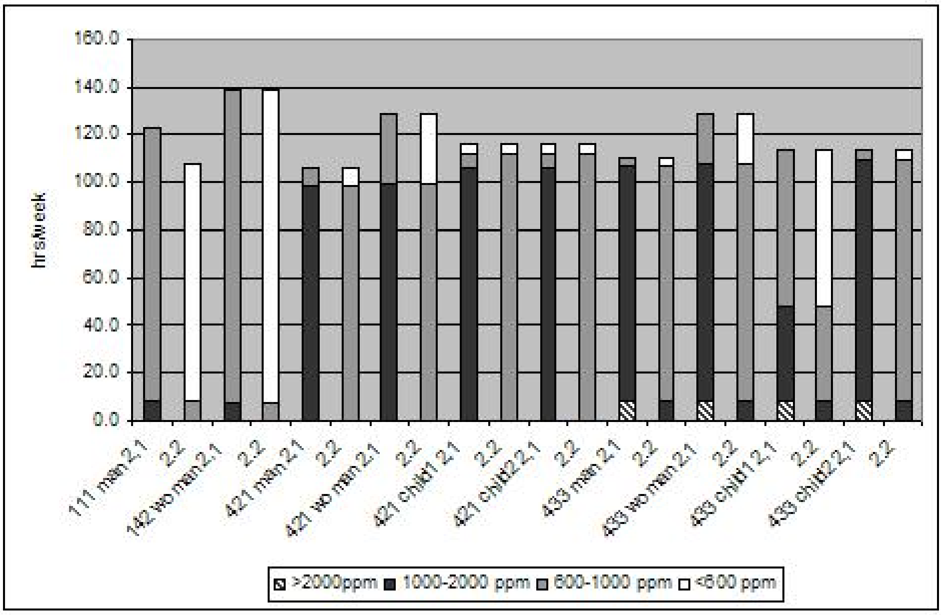

Figure 2: Time of exposure to levels of CO2 per occupant in ventilation scenario 2.1) airtight dwelling, and 2.2) no airtight dwelling, child1= bedroom 2, child2= bedroom 3

In dwellings with low infiltration rates with a ventilation system not used at its optimal capacity, all occupants show a high exposure time to CO2 levels of 1000 to 2000 ppm. In addition, large households may be at risk of exposure to levels of CO2 of more than 2000 ppm for few hours per week. For these dwellings, if the ventilation system is optimally used, the time exposed to high levels of CO2 decreases; though the occupants may be still exposed to CO2 levels of 600 to 1000 ppm. In dwellings with a high infiltration rate, regardless the age or use of the ventilation system, large households are exposed to levels of CO2 of 600 to 1000 ppm, which is under the level recommended by the Ashrae standards; and singles are not exposed to harmful levels. Nevertheless, the exposure of children to possible harmful levels increases when the ventilation system is older than 5 years.

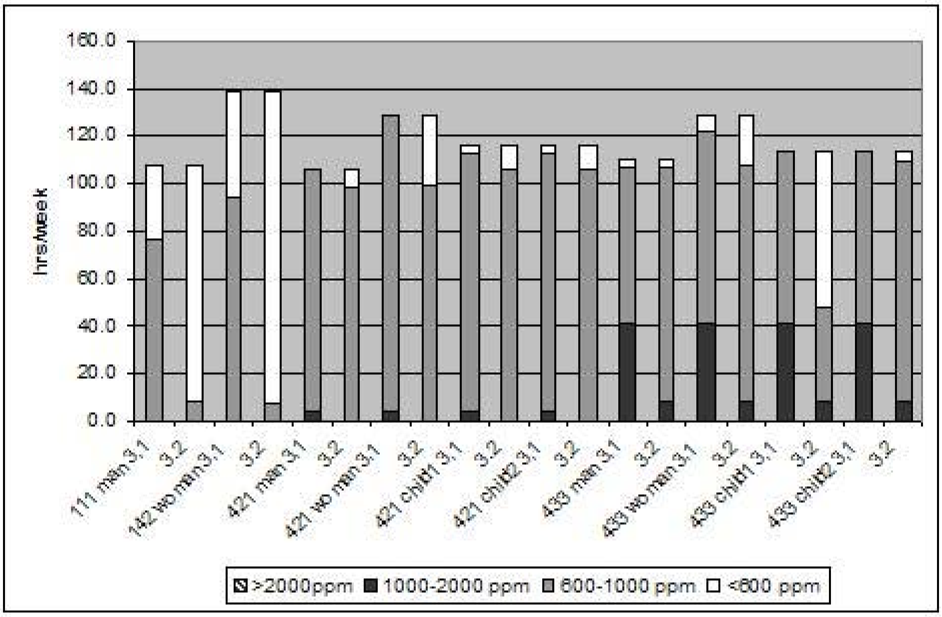

Figure 3: Time of exposure to levels of CO2 per occupant in ventilation scenario 3.1) airtight dwelling, and 3.2) no airtight dwelling, child1= bedroom 2, child2= bedroom 3

CONCLUSION AND DISCUSSION

From this study, we can conclude that the regulation on air flow rates does not comply with the needed ventilation for large households composed by 4 adults. Therefore, large household are at risk, which confirms some results of Hasselaar [7].

Because of the use of the system (users regulate the levels), the ventilation is in practice not enough. This could be solved by having the minimal rates already obtained in the lower levels. In addition, lack of maintenance of the ventilation systems decreases their capacity of satisfying the needed flow.

The statistical analysis of time spent indoors, showed significant differences within different age ranges and gender, nevertheless, differences in exposure between one person household may be neglected under the assumptions we made in this paper.

This research showed that differences in exposure time can be relevant for determining the ventilation required for a good indoor air quality. Though carbon dioxide does not cause health effects at the concentration levels considered for the analysis, it can be used as an indicator of the presence of other harmful substances in the indoor environment. Some groups of people are more exposed to the effects of the indoor environment. The statistical analysis of the time spent indoors showed that the elderly and women tend to spend more time in the indoor environment (table 2). The calculation of exposure time to CO2 showed that large households are more exposed to higher levels. For the statistical analysis, only Dutch people older than 12 years were considered. The exposure time of young children, and households with non-Dutch backgrounds should be considered for further research. Exposure to levels above 2000 ppm exists only for short time. In our calculations, this time is probably overestimated, because we worked with steady state concentrations. In case of short staying times, the steady state conditions would probably not be achieved. In further work, this should be taken into account.

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Office noise and work performance

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SUMMARY

Field studies have shown that speech is the most distracting sound in open-plan offices. On the other hand, laboratory studies have shown that speech impairs the performance of cognitively demanding work tasks, e.g. reading and memory-recall tasks. The sound level of speech does not determine the degree of distraction caused by speech. Instead, Speech Transmission Index, *STI*, which correlates with speech intelligibility can explain the distracting power of speech much better. *STI* can be determined also between workstations in open offices. This study presents a new general model that gives the decrease in work performance as a function of *STI*. Work performance decreases with increasing *STI*. The average performance of cognitively demanding tasks decreases at least by 7 percent when speech is highly intelligible. In practical design, the ability to calculate the payback time of investments leading to proper acoustical environment is important. The new model represents the missing link in such economic estimations. This paper is based on Reference [1].

INTRODUCTION

According to field surveys, noise is often the most detrimental factor of the indoor environment in open-plan offices [2-4]. Speech is usually the most distracting sound source in offices. Sounds of phones ringing and walking on corridors are on the second place. Ventilation noise and traffic noise are very seldom causing distraction.

The effects of continuous steady noise, such as ventilation noise or pink noise, on performance have been studied several decades. Steady noises have not seen to affect work performance directly. Colle and Welsh (1976) were among the first researchers to study the effect of speech on the performance of memory-recall tasks. The effect of speech on task performance was strong [5]. Their study has been repeated by many researchers using different tasks and sound environments. It is a well-known fact that the speech interferes with the performance of cognitively demanding work tasks. However, the distraction mechanism of speech on working memory is still under research and this is beyond the scope of this study.

Colle (1980) found that it is not the level of speech that causes the distraction but speech intelligibility [6]. The decrease in performance was almost independent of speech level within 40 to 80 dBA (Colle 1980). At 20 dBA, performances in silence and speech did not differ from each other. This was obvious because the level of consonants above 500 Hz fell below the auditory threshold level.

The aim of this study is to emphasize the important finding of Colle and to convert the finding into such form that it can be applied in room acoustical design.

Currently, there are no methods for showing that room acoustical improvements enhance work performance. Therefore, the investments on room acoustical conditions are difficult to justify as their positive effects on workers are unknown.

The primary aim of this study is to present a new general model that predicts the decrement in work performance as a function of speech intelligibility.

METHODS

Speech intelligibility is a subjective measure that describes the percent of correctly heard items. Items can be syllables, words or sentences. Speech intelligibility can be determined by using listening tests.

A good estimate of subjective speech intelligibility can be made by measuring the Speech Transmission Index, *STI*, between the speaker and listener. The correlation of *STI* and subjective speech intelligibility is presented in Figure 1.

STI has been used for 20 years to characterize the speech intelligibility in auditoria, class rooms and telephone communication. It has been started to apply in open offices recently [7-9]. The *STI* of an open-plan office is easy to measure and most acoustic consultants own the apparatus to measure *STI*. Therefore, the selection of *STI* as the explaining parameter of the model is meaningful and topical. The measurement arrangement of *STI* between office workstations is presented in Figure 2.

The subjective meaning of *STI* is presented in Table 1. *STI*=1.00 and *STI*=0.00 mean perfect and zero speech intelligibility, respectively. It should be noted that the aim of acoustical design of offices is good speech privacy, i.e. bad speech intelligibility.

Table 1. The subjective meaning of Speech Transmission Index, *STI*. Good speech intelligibility is desired in auditoria. In open offices, the opposite situation is appropriate.

STI	Speech intelligibility	Speech privacy	Examples in offices
0.00 … 0.05	very bad	confidential	Between two single-person office rooms, high sound insulation
0.05 … 0.20	bad	good	Between two single-person office rooms, normal sound insulation
0.20 … 0.40	poor	reasonable	Between workstations in a high-level open-plan office
			Between two single-person office rooms, doors open
0.40 … 0.60	fair	poor	Between desks in a well designed open-plan office
0.60 … 0.75	good	very poor	Between desks in an open-plan office, reasonable acoustical design
0.75 … 0.99	excellent	no	Face-to-face discussion, good meeting rooms
			Between desks in an open-plan office, no acoustical design

More than 30 psychological laboratory studies were reviewed to evaluate the performance decrements caused by speech or office noise. Only statistically significant differences were considered in this review. Most of the studies had been carried out by brain researchers that study the operation of working memory using speech stimulus. Therefore, the experiments had been carried out in two situations, with and without speech. They did not pay attention to the effect of speech intelligibility which would be of main interest if the results were to be applied in acoustic design. However, these studies are valuable since they prove that unwanted speech undoubtedly impairs task performance.

The decrease in performance, *DP*, was defined from the previous studies as the arithmetic difference in error percentage during speech and silence, that is,

$$DP = P_{speech} - P_{silence} \qquad (1)$$

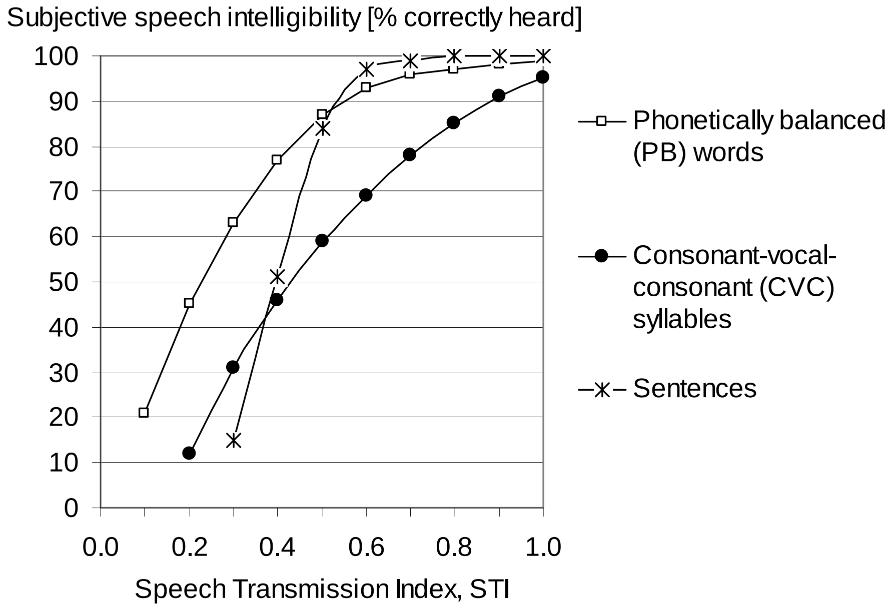

Figure 1. Experimental relations between subjective speech intelligibility and *STI* [10].

Figure 2. Measurement of STI between two neighboring workstations. Speaker is placed on one workstation and the measurement is carried out in the other. Workers are absent during the measurement.

RESULTS

Office noise containing speech deteriorates work performance significantly. The summary of literature review is presented in Table 2. Performance decrements due to speech have been between 4 and 41 % compared to performance in silence. Large variation results from the differences in types of tasks and sound environments.

Table 2. Summary of literature review showing the decrease in performance depending on task type and sound environment. *N* is the number of studies.

		Decrease in performance [%]		
Task type, sound environment	***N***	**AVERAGE**	**Minimum**	**Maximum**
Memory for letters presented visually, speech	4	19	15	29
Memory for 9 digits presented visually, speech	5	11	9	13
Reading comprehension, speech	1	10	10	10
Proof-reading, speech	3	7	4	10
Other tasks, speech	7	16	7	29
Varying tasks, office noise with or w/o speech	5	26	13	41
Short-term memory of digits, music with or without voice	1	10	4	14
	26	14	4	41

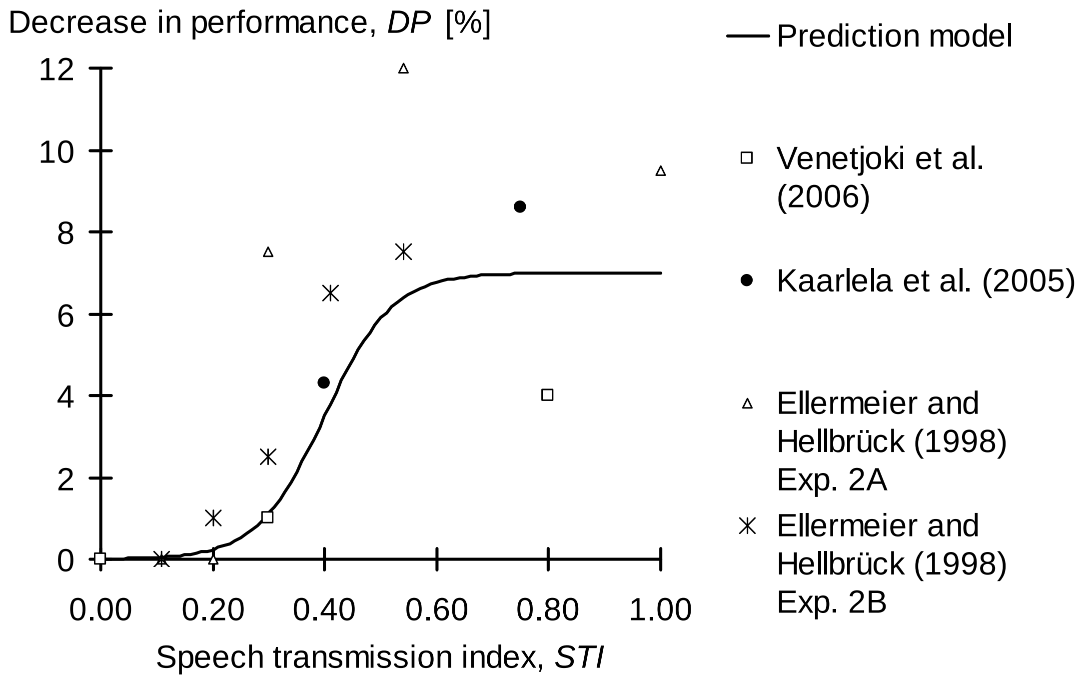

Figure 3. Work performance decreases with increasing speech intelligibility in offices.

There is clear evidence that distraction of speech increases with increasing speech intelligibility. Therefore, it was justified that the *STI*-dependence of work performance can be based on the relation between *STI* and subjective speech intelligibility of sentences, as shown in Figure 1. The mathematical form of the prediction model is

$$DP = \frac{-A}{1+\exp[(STI-0.4)/0.06]} + A \qquad (2)$$

where *DP* is decrease in performance [%] and *STI* is the average Speech Transmission Index at the office workstation. The mathematical model is outlined in Figure 3. [1]

The value of constant A depends on task type and sound environment. For this study, a value A=7 % was used which means that the highest decrease of productivity is assumed to be at least 7 %. It should be noted that this is not a universal value. The choice represents merely a bottom line of previous research. Most studies resulted in larger *DP* values as Table 2 indicated. The maximum decrease in performance depends on the cognitive demands of the office work. Personality factors explain the differences between individuals.

DISCUSSION

The main criticism should concern the existence and the shape of the prediction model of Figure 3 and Equation (2). The model was based on three corner stones:

1. highest performance is obtained when speech is absent, *STI*=0.00,
2. the largest decrease in performance is 7 % and this is reached when speech is perfectly heard, i.e. *STI*=1,00, and
3. the range *STI*=0.00 - 1.00 is based on the *STI* vs. sentence intelligibility curve of Figure 1.

The model should be validated with experimental data where *STI* of speech is varied. This work is mainly the topic of our future work. So far, there are three experimental studies to support the shape of the model. They are indicated as individual points in Figure 3 and discussed below.

Venetjoki et al. (2006) studied the effect of speech of varying speech intelligibility on proof-reading performance [11]. Three *STI* values were used, 0.00, 0.30, and 0.80. Their study clearly showed that performance is not affected when *STI* remains within 0.00 and 0.30. This study will be continued in FIOH. The aim is to find more data to the *STI* range 0.20 - 0.60 where the performance is expected to be decreased very rapidly. This range is of main interest for the motivation of acoustic improvements in offices.

Ellermeier and Hellbrück [12] studied the recall of digits in different signal-to-noise ratios of speech. They carried out two laboratory experiments with speech-to-noise ratios +4, -4, -12 dB in experiment 2A, and speech-to-noise ratios of +4, 0, -4, -8, and -12 dB in experiment 2B. These signal-to-noise ratios were converted by the author into *STI* values using the model of Hongisto et al. (2004) [7]. The results of both experiments followed the formulated prediction model reasonably well in a wider *STI* range.

Kaarlela-Tuomaala et al. (2004) studied the effects of office noise on office workers using questionnaires [13]. The study was carried out in a company which moved from a single-room office to an open office. The self-estimated daily waste of working time due to noise was 4.3 % in a single-room office (*STI*=0.40) and 8.6 % in an open-plan office (*STI*=0.70). The field studies of Helenius et al. confirm this finding. [2]

CONCLUSIONS

Work performance declines with increasing speech intelligibility. The performance decline due to office noise can be predicted when the average Speech Transmission index, *STI*, between office workstations is known. Present acoustical design guidelines already aim at the reduction of *STI* between office workstations [14]. The new model enables the estimation of the payback time of noise control investments that lead to better speech privacy. The model can be utilized in promoting actions that aim at better acoustic environment.

ACKNOWLEDGEMENTS

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Impact of Indoor Humidity, Local Air Velocity and Illuminance on Subjective Comfort, Performance and Fatigue

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SUMMARY

A subjective experiment was conducted using 15 college-aged subjects of both genders in order to evaluate their physiological and psychological reactions, performance and fatigue under the different combinations of indoor humidity, local air velocity and illuminance.
The five-hour exposure periods were divided into three sections of 1.5 hours by 10-minute breaks. During each section, subjects performed 3 times of 20-minute task. During the exposure time subjects rated their sensations, visual fatigue and general fatigue, and measured their break up time (BUT) and skin moisture after each task.
Higher rate of complaints related to visual fatigue was reported under the condition with local air velocity than without it at the same humidity. High humidity caused subjective low visual fatigue. Subjective BUT was shorter in the environment with local air velocity or at low humidity, where people rated greater eye dryness sensation. High illuminance caused no difference in occupants' physiological and psychological reactions although correct answer rate was higher in environment with high illuminance.

INTRODUCTION

Tsutsumi et al. [1] [2] [3] conducted subjective experiments on the effects of low humidity on occupants' comfort and productivity. As the result, occupants' discomfort caused by dryness was not found in the thermally neutral condition at below 40%RH. On the other hand, visual data acquisition could be interfered at very low humidity. In the daily life, local air velocity and illuminance could cause occupants' general dryness sensation and eye dryness, blink, and visual fatigue as well as indoor humidity. Wyon et al. [4] reported other environmental factor like air velocity and illuminance also cause eye and skin dryness as well as indoor humidity. Present paper reports the subjective experiments carried out to evaluate their physiological and psychological reactions, performance and fatigue under the different combinations of indoor humidity, local air velocity and illuminance.

METHODS

Experimental design

A subjective experiment was conducted in a climate chamber using 15 college aged subjects, 7 males and 8 females, in order to evaluate their physiological and psychological reactions, performance and fatigue under the different combinations of indoor humidity, local air velocity and illuminance.

Experimental conditions:
The experimental conditions are listed in Table 1.

For all conditions, air temperature was kept at 25.0 °C. Two levels of humidity conditions, 30%RH and 70%RH, were set. For each humidity condition, two conditions of local air velocity to subject's face with a small fan on the desk, with and without velocity of 1.0 m/s, were examined. A condition with high illuminance at 30%RH without local air velocity was also studied. Under the condition with high illuminance at 30%RH, vertical illuminance in front of PC screen on the desk of subject was about 1200 lx while under other four conditions it was 400-450 lx. Each subject wore the clothing ensembles that consisted of a long-sleeve shirt, short-sleeve T-shirt, trousers, socks and shoes. All subjects wore their own underwear. Clo value was measured to be 0.85clo by using a thermal manikin.

Table 1. Experimental conditions.

Conditions	Air Temperature (=MRT)	Relative Humidity	Local Air Velocity	Vertical Illuminance
30%RH	25.0°C	30%RH	Without	400-450lx
30%RH +Local Air Velocity (LV)			With	
30%RH +High Illuminance (HI)			Without	1200lx
70%RH		70%RH	Without	400-450lx
70%RH+LV			With	

Experimental procedure:

Figure 1 shows the experimental procedure. People were exposed in a climate chamber for 5 hours. The five-hour exposure periods were divided into three sections of 1.5 hours by 10-minute breaks. During each section, subjects performed 20-minute task three times: 1) Short-term memory task [5], 2) Vigilance task [5], 3) Visual recognition and reaction task. Subjects could only drink a bottle of water provided by the experimenters during the exposure time. During 10-minute break between sections, people were allowed to go out the climate chamber. Air temperature, relative humidity and globe temperature in the chamber were logged every minute. Air velocity before and after the exposure time was also recorded.

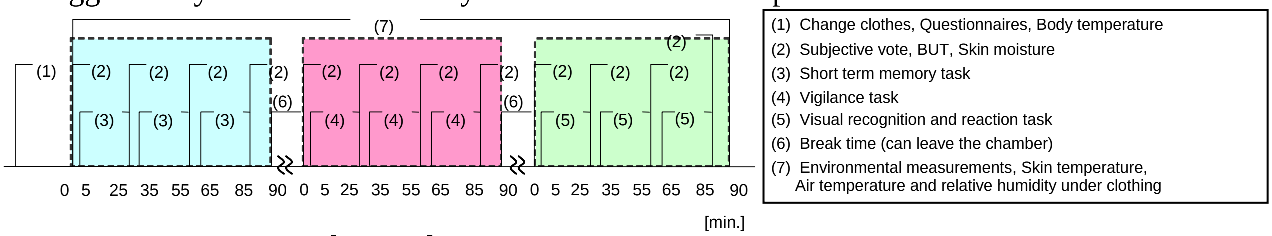
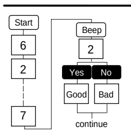
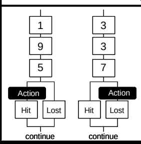
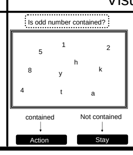


Figure 1. Experimental procedure.

Tasks:

During the exposure, subjects performed 3 types of task: 1) Short-term memory task [5], 2) Vigilance task [5], 3) Visual recognition and reaction task, as presented in Table 2. High visual data acquisition, quick judgments and response are needed for all tasks. Therefore, short Break up time (BUT) or visual fatigue due to the indoor environment could affect their performance. Subjective discomfort to the environment would also cause general fatigue that reduces their performance.

Table 2. Detail of 3 tasks.

Short-term memory task		Vigilance task		Visual recognition and reaction task	
	1. Total of six numbers were displayed every second. 2. One number to be judged were shown. 3. If the number was included in 6 numbers, subjects reacted as soon as possible. If not, they didn't. Time limit was 3 seconds.		1. Total of 3 numbers were displayed every second. 2. When three different odd numbers were shown, subjects reacted as soon as possible. If not, they didn't. Time limit was 3 seconds.		1. For three seconds, ten characters that consist of numbers (1-9) and alphabets (a-z) were shown at random. 2. When an odd number appears, subject reacted as fast as possible. If not, they didn't.

Statistical analysis

Data obtained in the experiments were analyzed with Non-parametric statistical analysis method [6]. The Wilcoxon Matched-Pairs Signed Ranks test was administered between each condition. Indoor humidity effect was discussed in the pair-wise comparison between 30%RH and 70%RH, and between 30%RH+LV and 70%RH+LV. According to the comparison between 30%RH and 30%RH+LV, and between 70%RH and 70%RH+LV, local air velocity effect was reported. Pair wise test between 30%RH and 30%RH+HI reported the effect of indoor illuminance. P-values presented in the figures indicate the level of significance.

RESULTS AND DISCUSSION

Physiological responses

Skin moisture

Skin moisture was measured on a subject's left forearm with "Skicon-200" (IBS Corp.). Skicon-200 adopts the high frequency impedance method [7]. As shown in the Figure 2, skin moisture after all tasks at 30%RH was lower than at 70%RH and at 30%RH+LV than 70%RH+LV ($p<0.01$). Low humidity caused their low skin moisture. No significant difference was observed between 30%RH and 30%RH+LV and between 70%RH and 70%RH+LV. Local air velocity did not have great impact on subjective skin moisture. No significant difference was observed between 30%RH and 30%RH+HI. The effect of illuminance was moderate in this experiment.

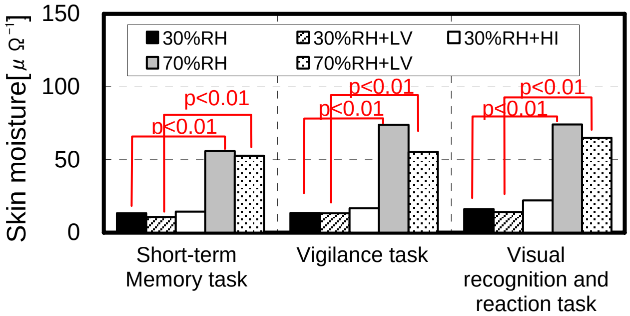


Figure 2. Skin moisture.

Break up time (BUT)

Break up Time (BUT) of precorneal film is one of the physiological reactions that might affect the subjective eye comfort [8].During the exposure time, the subjects measured their interval time between each blink by themselves using a stopwatch as "BUT" after each task for this research.

BUT recorded at the end of exposure was shown in Figure 3. BUT after all task under 30%RH+LV was significantly shorter than under 30%RH ($p<0.02$). Shorter BUT was observed under 70%RH+LV than 70%RH although no statistically significant difference was observed. It is found that BUT became shorter in the environment with local air velocity. There was significant difference in BUT after Visual recognition and reaction task between 30%RH+LV and 70%RH+LV ($p<0.05$). BUT measured after Short-term memory task and Vigilance task at 30%RH and at 30%RH+LV were shorter than 70%RH and 70%RH+LV respectively. Low humidity had the impact that makes subjective BUT gotten shorter.
No statistically significant difference was observed between 30%RH and 30%RH+HI.

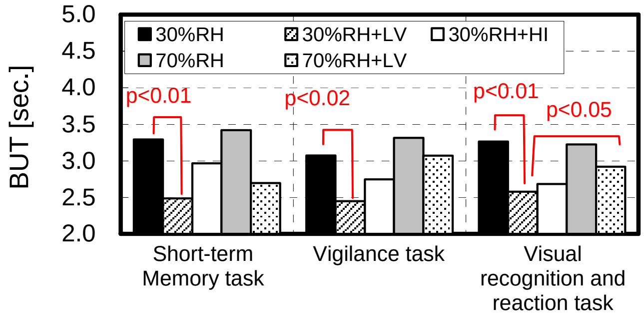


Figure 3. BUT measured at the end of each task.

Psychological reactions

General humidity sensation

Figure 4 presents the general humidity sensation rated by subjects after each task. Subjects tended to feel dryer after Short-term memory task and vigilance task under 30%RH+LV than that under 30%RH ($p<0.1$). While no significant difference was found, general dryness sensation under the condition with local air velocity was higher than under the condition without local air velocity at the same humidity.

General dryness sensation tended to be higher at 30%RH+LV than 70%RH+LV after Vigilance task and the Visual recognition and reaction task ($p<0.1$). Subjects reported significantly higher dryness sensation at 30%RH than at 70%RH after the Visual recognition and reaction task ($p<0.03$). It is found that people perceived air to be dryer under low humidity condition.

No significant difference was observed between 30%RH and 30%RH+HI. Difference of illuminance examined in this experiment did not have great impact on subjective general humidity sensation.

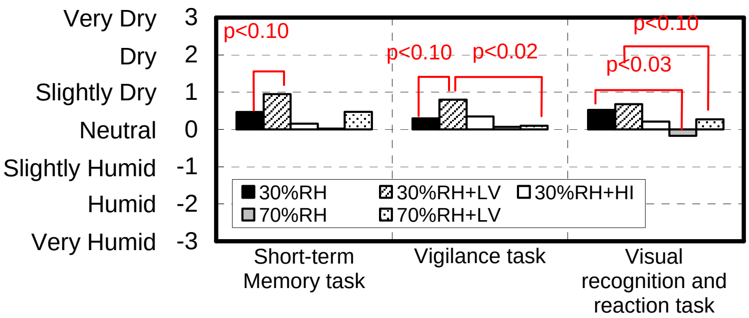


Figure 4. General humidity sensation.

Sensation of eye dryness

Eye dryness sensation vote after each task was shown in Figure 5.

Eye dryness at 30%RH was significantly higher than at 70%RH after all tasks in pair-wise comparison ($p<0.01$). Subjects reported greater eye dryness at 30%RH+LV than 70%RH+LV after all tasks although no significant differences were gotten. It proved that indoor humidity has an impact on subjective eye dryness sensation.

Statistically significant differences occurred between 70%RH and 70%RH+LV after all tasks, and between 30%RH and 30%RH+LV after the Short-term memory task and Vigilance task ($p<0.04$). People reported their eyes were dryer in the environment with local velocity than at the same humidity without velocity.

Since no significant difference was found between 30%RH and 30%RH+HI, it is concluded that the effect of illuminance on subjective sensation of eyes were not found in this experiment

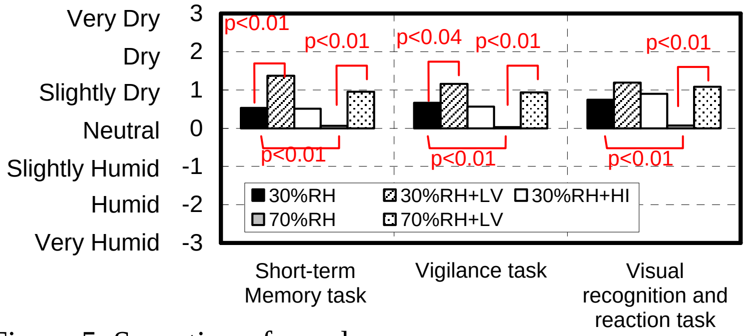


Figure 5. Sensation of eye dryness.

Sensation of eye comfort

As shown in Figure 6, there was no significant difference between 2 humidity levels for both of with and without local velocity. Subjective discomfort was significantly greater at 30%RH+LV than 30%RH after Short-term memory task and Vigilance task and at 70%RH+LV than 70%RH after Short-term memory task and Visual recognition and reaction task ($p<0.04$). People reported greater eye discomfort at 30%RH+LV than at 30%RH after Visual recognition and reaction task, and at 70%RH+LV than 70%RH after Vigilance task but these findings were not significant. This result demonstrated that subjects felt their eyes were more uncomfortable in the air with velocity.
According to no significant difference between 30%RH and 30%RH+HI, illuminance did not affect subjective sensation of eye comfort.

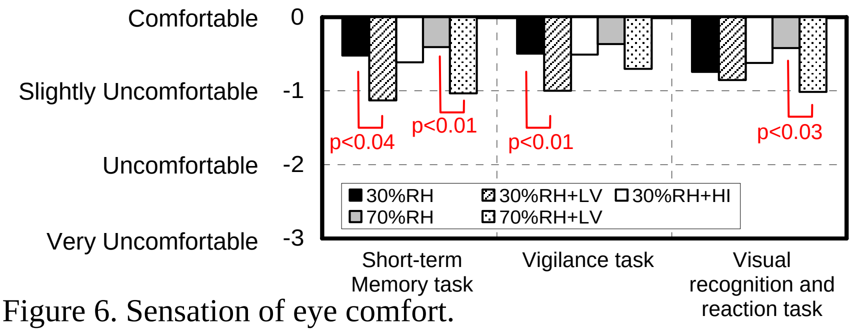


Figure 6. Sensation of eye comfort.

Performance and fatigue

Correct answer rate

Figure 7 shows the correct answer rate of last task for each section. The correct answer rate indicates the ratio of correct answers to all questions during a 20-minute task. Correct answer rate of Vigilance task was above 90% for all conditions, while that of Short-term memory task was between 60 and 85%. It was found that Short-term memory task was the most difficult task and Vigilance the easiest.
Same tendency of the correct answer rate under different conditions were obtained for all tasks. Correct answer rate at 30%RH+HI tended to be higher than at 30%RH for all tasks.($p<0.07$)
No significant difference between 70%RH and 70%RH+LV were found. Correct answer rate of Short-term memory task at 30%RH+LV was significantly higher than at 30%RH ($p<0.05$) and that of Visual recognition and reaction task tended to be higher at 30%RH+LV than at

30%RH ($p<0.06$). This might be caused by the comfortable thermal environment at 30%RH with air velocity, where PMV (Predicted Mean Vote) was +0.72, between "0: neutral" and "+1: slightly warm", at 30%RH and –0.1, near "0: neutral", at 30%RH+LV.
As for the effect of indoor humidity, correct answer rate of Short-term memory task at 30%RH tended to be lower than 70%RH ($p<0.08$). Higher rate of correct answer at 70%RH was found compared with that at 30%RH, nevertheless there was no statistically significant difference.

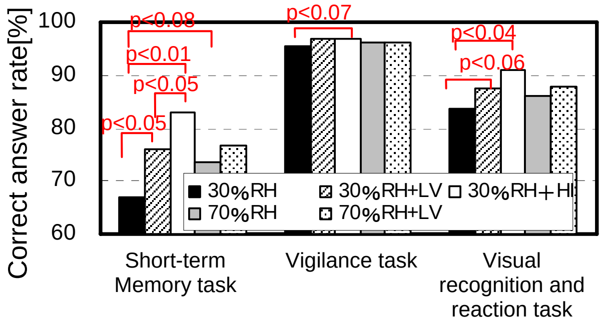


Figure 7. Correct answer rate of each task.

Figure 8 presents the relationship between subjective BUT and correct answer rate of the Short-term memory task and the Visual recognition and reaction task. Both figures demonstrated that longer BUT could cause higher correct answer rate. It is concluded that in case that the subjective BUT gets shorter due to indoor environment, subjective performance would become lower.

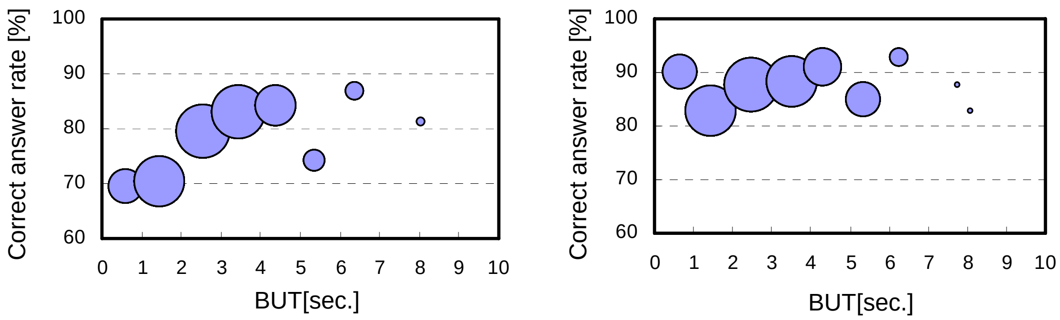


Figure 8. Relationship between BUT and correct answer rate.
(Left: Short-term memory task, Right: Visual recognition and reaction task)

General fatigue

Subjects were asked to assess their general fatigue [9]. The questionnaire is composed of 3 groups. Each category has 10 symptoms related to subjective fatigue. Subjects marked "O" if they had the given symptoms, and marked "X" if they did not. Ratio of complaints was calculated for each category using the equation below:

$$\text{Rate of complaints [\%]} = \frac{\text{Total number of complaints}}{\text{The number of symptoms} \times \text{ The number of subjects who used a questionnaire}}$$

Three patterns were suggested by comparing the rate of complaints for each category: I>III>II: I-dominant, I>II>III: II-dominant, and III>I>II: III-dominant. "I-dominant" indicates a general pattern of fatigue, "II-dominant" a typical pattern of fatigue for mental work or night work and "III-dominant" a typical pattern of fatigue for physical work.

Table 3 listed the rate of complaints. Total rate of complaints was about 10% for all conditions examined in this experiment. On the other hand, type of fatigue reported after Short-term memory task under 30%RH+LV, 30%RH+HI and 70%RH was I>II>III; II-dominant, while I>III>II was rated under any other condition. It is concluded that mental work load was greater during performing the Short-term memory task than any other task.

Table 3. Rate of complaints related to fatigue.

Task	Condition	Category I[%]	Category II[%]	Category III[%]	Type of fatigue		Total rate of Complaints[%]
Short term memory task	30%RH	27.3	0.7	6.0	I>III>II	I-dominant	11.3
	30%RH+LV	14.7	7.3	4.0	I>II>III	II-dominant	8.7
	30%RH+HI	22.8	8.7	6.0	I>II>III	II-dominant	12.5
	70%RH	21.3	7.3	6.0	I>II>III	II-dominant	11.6
	70%RH+LV	15.3	4.7	7.3	I>III>II	I-dominant	9.1
Vigilance task	30%RH	13.3	0.7	8.0	I>III>II	I-dominant	7.3
	30%RH+LV	17.3	3.3	4.0	I>III>II	I-dominant	8.2
	30%RH+HI	25.3	7.3	5.3	I>II>III	II-dominant	12.7
	70%RH	9.3	2.0	4.7	I>III>II	I-dominant	5.3
	70%RH+LV	18.0	6.0	8.0	I>III>II	I-dominant	10.7
Visual recognition and reaction task	30%RH	14.0	3.3	7.3	I>III>II	I-dominant	8.2
	30%RH+LV	16.7	3.3	5.3	I>III>II	I-dominant	8.4
	30%RH+HI	15.3	2.7	5.3	I>III>II	I-dominant	7.8
	70%RH	8.7	1.3	4.7	I>III>II	I-dominant	4.9
	70%RH+LV	23.3	5.3	7.3	I>III>II	I-dominant	12.0

Visual fatigue

Subjects reported their visual fatigue on the questionnaire referred to Takahashi [10]. Twenty symptoms related to visual fatigue were shown on the questionnaire. Subjects marked “O” if they had the given symptoms, and marked “X” if they did not. Total rate of complaint related to visual fatigue was calculated by using the same method as equation mentioned in the session of general fatigue.

Figure 9 presents the visual fatigue rated at the end of each task. People complained more after all tasks in the environment with local air velocity compared with the condition without air velocity at same humidity. More complaints were reported at 30%RH than 70%RH after all tasks.

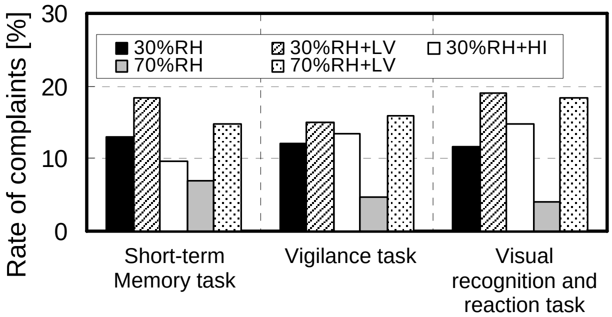


Figure 9. Visual fatigue rated at the end of each task.

CONCLUSION

A subjective experiment was conducted using 15 college-aged subjects of both genders in order to evaluate their physiological and psychological reactions, performance and fatigue under the different combinations of indoor humidity, local air velocity and illuminance.

The five-hour exposure periods were divided into three sections of 1.5 hours by 10-minute breaks. During each section, subjects performed 3 times of 20-minute task.

Low humidity caused subjects’ low skin moisture and general dryness sensation. On the other hand, local air velocity did not have great impact on subjective skin moisture while general

dryness sensation under the condition with local velocity was higher than under the condition without local velocity at the same humidity.
It is found that BUT became shorter and visual fatigue became greater in the environment with local air velocity and at low humidity. It proved that of indoor humidity and local velocity has negative impact on subjective eye dryness sensation and visual fatigue, while only local velocity affects their sensation of eye comfort.
Subjective mental work load was found to be greater during performing short term memory task than other tasks.
According to the results in their performance, it is concluded that if the subjective BUT gets shorter due to indoor environment, subjective performance would become lower.
High illuminance caused no difference in occupants' physiological and psychological reactions although correct answer rate was higher in environment with high illuminance.

ACKNOWLEDGEMENT

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Evaluation Method for Effects of Improvement of Indoor Environmental Quality on Productivity

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SUMMARY

A subjective experiment was conducted in a climatic chamber to evaluate the effects of improvement of indoor environmental quality on occupant's performance. The chamber was conditioned with the combination of: operative temperatures of 25°C or 28°C, illuminance of 750lx or 400lx and with or without traffic noise. The subjects performed triple-digit multiplication and voted satisfaction with indoor environment and the subjective symptoms of fatigue before and after each session of task. "Predicted-performance-vote", which estimates the effects of improvement of thermal, lighting and acoustic environment on productivity, was proposed in this study. In the environmental combinations of this study, subjects gave priority to improve thermal and acoustic environment rather than lighting environment. The self-estimated performance was related to occupants' satisfaction with indoor environment and was inversely related to fatigue. The results from the experiment indicated that subjects evaluated themselves well-performed when satisfaction with indoor environment was higher and fatigue was lower.

INTRODUCTION

In offices, indoor environmental qualities (IEQ) such as thermal environment, lighting environment, acoustic environment and IAQ may affect intricately on occupants' performance and productivity. In previous studies, the effects of environment on human responses and performance have been evaluated, and it was suggested that poor IEQ would increase the occupants' fatigue and decrease the level of satisfaction with indoor environment and performance [1]. In this study, a subjective experiment was conducted to consider the evaluation method for effects of improvement of indoor environmental factors on productivity by using not only performance but also subjective votes.

METHODS

Experimental Conditions

The subjective experiment was conducted in a climatic chamber at Waseda University between October 14th and November 11th in 2005. The chamber was conditioned as the conceivable environments in office with the combination of: operative temperatures of 25°C or 28°C, illuminance of 750lx or 400lx and with or without traffic noise. Eight conditions were named as "Control(28°C, 400lx, with traffic noise)", "T(25°C, 400lx, with traffic noise)", "N(28°C, 400lx, without traffic noise)", "L(28°C, 750lx, with traffic noise)", "TN(25°C, 400lx, without traffic noise)", "TL(25°C, 750lx, with traffic noise)", "NL(28°C, 750lx, without traffic noise)" and "TNL(25°C, 750lx, without traffic noise)". In "with traffic noise" conditions, the traffic noise, which was recorded near the heavy traffic street in Tokyo was played in the chamber from the speakers. The relative humidity was kept at same level in all conditions. The environmental conditions measured during the experiment are shown in Table 1. Subjects experienced these eight conditions in balanced order. Prior to the above-mentioned eight conditions, the subjects underwent a practice session under the same condition as Control.

Table 1. Environmental conditions

	Air temp [°C]	Operative temp [°C]	Relative humidity [%RH]	Equivalent sound level [dBA]	Illuminance [lx]	CO_2 [ppm]
Control	28.3 (0.4)	28.5 (0.4)	44 (3)	63	405 (15)	941 (50)
T	25.2 (0.6)	25.2 (0.6)	44 (1)	64	404 (12)	875 (88)
N	27.4 (1.3)	27.9 (0.8)	43 (2)	51	401 (13)	825 (50)
L	27.9 (0.8)	28.2 (0.4)	41 (1)	64	773 (17)	825 (138)
TN	25.6 (0.8)	25.7 (0.7)	43 (3)	51	401 (16)	792 (93)
TL	25.1 (1.3)	25.1 (1.4)	44 (3)	64	772 (28)	862 (87)
NL	27.7 (2.0)	28.0 (1.2)	42 (3)	51	757 (24)	798 (132)
TNL	26.1 (1.4)	26.4 (1.4)	42 (3)	51	752 (20)	963 (100)

( ) Standard deviation

Subjects

Ten male college-aged subjects participated in the experiment with their age of 22.1±1.3, height of 170.1±3.7cm, and weight of 61.5±5.8kg. Subjects were exposed in the climatic chamber in groups of 3 or 4 on the same time of the day in the successive experimental weeks to avoid any influence of time. All subjects were volunteers and they were paid at a fixed rate for their participation. In order to keep their motivation at the same level, they were informed that higher performers of the task could earn bonus. Subjects wore a uniform business suit with an insulation value of 0.93 clo [2].

Measurements

During the experiment, subjects performed thirty minutes of triple-digit multiplication tasks on paper four times in a day. The task performance for each subject was evaluated by his Z-score as a normalized performance to precisely reflect the performance change of each subject. The Z-score was calculated from the equation (1).

Z-score $$S_{A,i} = \frac{x_{A,i} - \bar{x}_A}{s_A} \qquad (1)$$

where,

$x_{A,i}$: number of correct answers during the session i for subject A [-]

$\bar{x}_A$: average number of correct answers of the subject A among all sessions [-]

s_A : standard deviation for the number of correct answers of the subject A among all sessions [-]

Satisfaction with indoor environment was asked to evaluate the perception of the environment. It was measured using a scale with rating of: -1="clearly unsatisfied", -0="just unsatisfied", +0="just satisfied", +1="clearly satisfied".

To evaluate the feeling of fatigue, subjects filled in the sheets of "Evaluation of Subjective Symptoms of Fatigue", which was suggested by the working group for occupational fatigue of the Japan Society for Occupational Health [3]. This evaluation method is used in the field of science of labor and ergonomics in Japan. It consists of three categories: group I consists of 10 terms on "drowsiness and dullness", group II consists of 10 terms on "difficulty in concentration", and group III consists of 10 terms on "projection of physical disintegration". Based on Yoshitake's method, the rate of complaints was calculated by Equation (2).

$$\textit{Rate of complaints}\ [\%] = \frac{\textit{number of selected symptoms of all subjects}}{\textit{number of terms concerned} \times \textit{number of subjects}} \times 100 \qquad \ldots(2)$$

"Predicted-performance-vote", which estimates effects of improvement of thermal, lighting and acoustic environment on productivity, was introduced in this study. The questionnaire sheet is shown in Figure 1. Subjects predicted and answered in this questionnaire how they thought their performance at task would increase if thermal, lighting and acoustic environment were improved. First, a subject marked on a scale (0: worst performance, 100: best performance under the environment assumed thermal, lighting and acoustic environments were all improved) how he could perform well in the office under same condition with the experiment. This value is named "predicted performance before environmental improvement (PP-0)". Second, he chose one of the environmental factors which they wanted to improve the most, and marked on a scale how they could perform well under the environment assumed to be improved (PP-1). Then, he chose another environmental factor which they wanted to improve the next, and marked on a scale how he could perform well under the environment assumed that two environmental factors of his chose were improved (PP-2). If all the environmental factors were selected to be improved, subject marked his predicted performance on the scale at 100. To estimate the effects of improvement of indoor environmental factors on performance simply, subjects were asked to answer predicted-performance-vote at the end of the day.

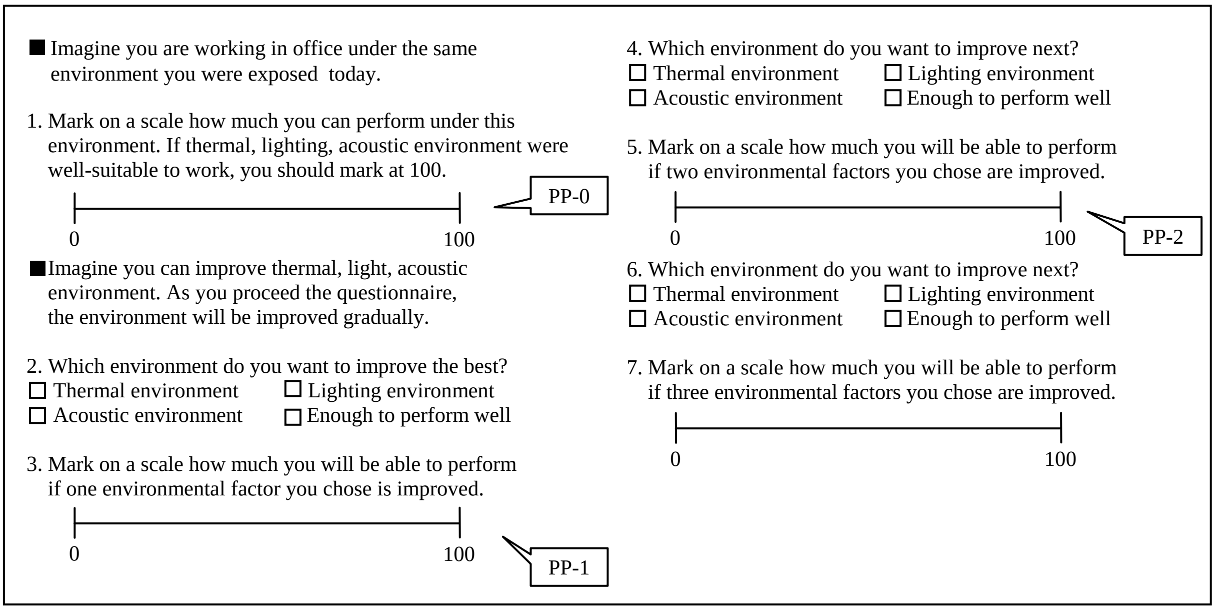

Figure 1. The questionnaire sheet of predicted-performance-vote

Experimental Procedure

Procedure of the experiment is shown in Figure 2. Duration of the experiment lasted for 210 minutes. First, subjects answered questionnaire in an anteroom. After entering the climatic chamber, they seated in the chair and voted satisfaction with indoor environment and fatigue on a PC screen. They rested in a sedentary position for 45 minutes to adopt themselves to the environment. Then, they performed thirty-minutes of multiplication task four times, and voted satisfaction with indoor environment and fatigue before and after the tasks. They were asked to answer predicted-performance-vote at the end of the day.

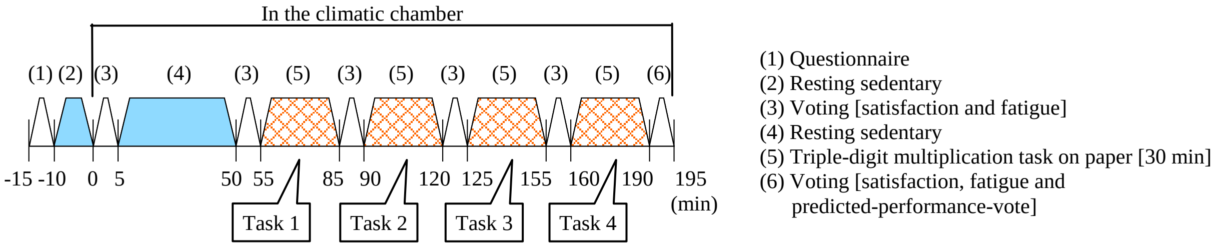

Figure 2. Procedure of subjective experiment

Statistical Analysis

The comparisons between the environmental conditions were analyzed using one-way ANOVA. When a statistically significant result was found, post hoc comparisons were made by using Fisher's protected LSD. The level of significance was set at $p<0.05$. The comparisons between PP-0 in Control and that in the others were analyzed by paired t-test with the level of significance of $p<0.05$. For the analysis of correlation, Pearson's product moment correlation coefficients were calculated.

RESULTS

Performance

The performance of multiplication task was evaluated by Z-score. The results of Z-score are shown in Figure 3. At the 3rd session of multiplication task, the Z-score in Control was significantly lower than that in L, TN, TNL ($p<0.01$), and TL ($p<0.05$).

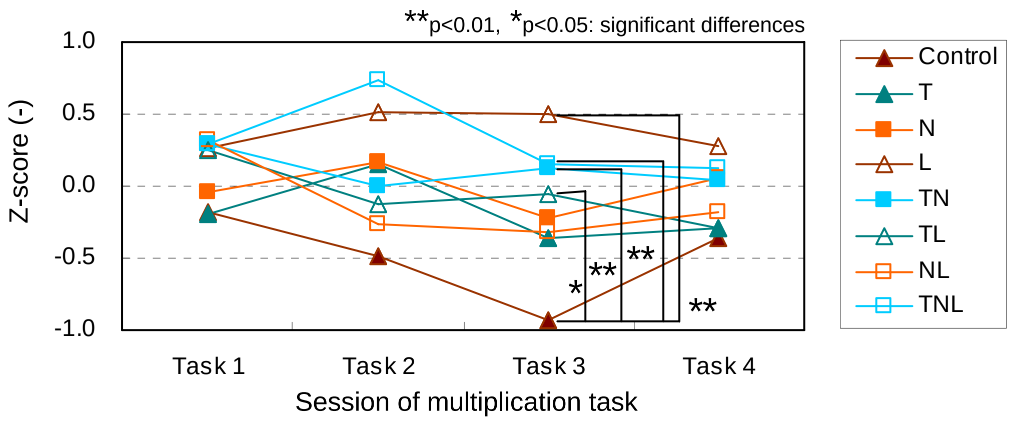

Figure 3. Z-score of multiplication task

Satisfaction with Indoor Environment

Subjects voted their satisfaction with indoor environment six times a day, before and after each session of resting and task. The results of the votes of satisfaction with indoor environment are shown in Figure 4. They were satisfied with environments of TN and TNL, and dissatisfied with those of all other conditions.

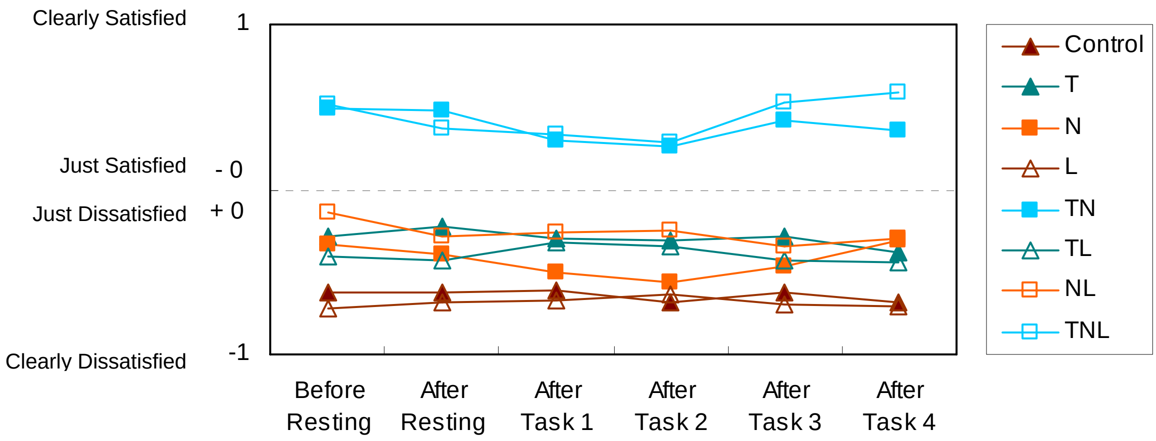

Figure 4. The votes of satisfaction with indoor environment

Fatigue

The results of the general rate of fatigue are shown in Figure 5. The general rate of fatigue was increased as the experiment proceeded. The general rate of fatigue in the “28°C” conditions (Control, N, L, NL) was higher than that in the “25°C” conditions (T, TN, TL, TNL), and that in the conditions “with traffic noise” (Control, T, L, TL) was higher than that in the conditions “without noise” (N, TN, NL, TNL).

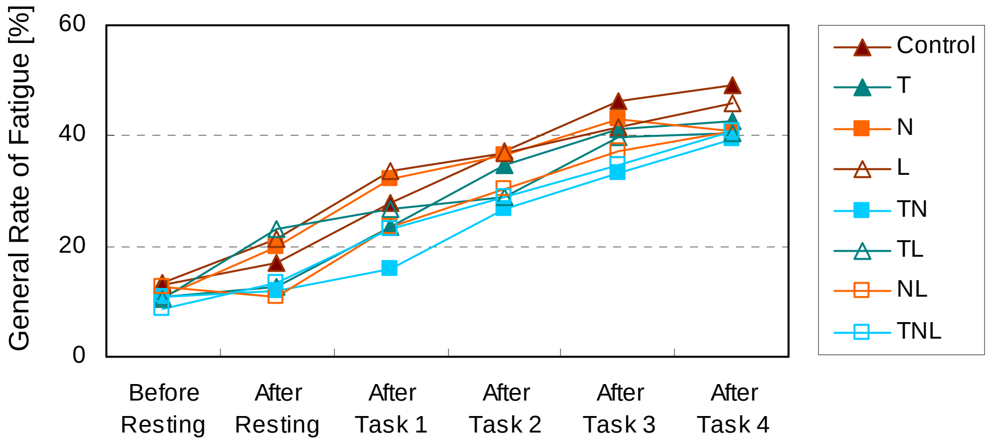

Figure 5. The general rate of fatigue

Predicted-Performance-Vote

The results of a part of the predicted-performance-vote are shown in Table 2. PP-0 is the predicted performance in the office under the same condition with the experiment. PP-1 is the predicted performance after one environmental factor was assumed to be improved. PP-0 in each condition was in the order from TNL, TN, NL, T, TL, N, L and Control. It indicates that the conditions considered to be well-suited to office work in this study were in that order. The subjects gave priority to improve thermal and acoustic environment rather than lighting environment in the environmental combinations of this study. PP-1s in Control and L were lower than those in all other conditions. Control and L were the conditions that combination of poor thermal and acoustic environment, one of the two was left to be improved when PP-1 was voted. The subjects estimated their predicted performance highly under the condition that both of thermal and acoustic environments were assumed to be improved. In the environmental combinations of this study, the supposed improvement of thermal and acoustic environment affected predicted performance more than that of lighting environment.

Table 2. A part of the predicted-performance-vote

		Control	T	N	L	TN	TL	NL	TNL
Predicted Performance before environmental improvement (PP-0)		56.1 (16.0)	66.7 (20.3)	75.7 (9.6)	61.9 (15.4)	81.2 (13.0)	76.5 (9.4)	71.0 (19.6)	86.5 (11.0)
Environmental factor chosen to improve	Thermal	5	0	10	4	5	0	10	7
	Lighting	0	0	0	0	3	0	0	0
	Acoustic	5	10	0	6	1	10	0	1
	Enough to perform well	0	0	0	0	1	0	0	2
Predicted Performance after one environmental factor was assumed to be improved (PP-1)		81.7 (13.0)	93.3 (9.2)	93.4 (7.7)	83.6 (11.4)	92.7 (6.1)	96.6 (3.5)	93.6 (7.5)	97.4 (5.0)

() Standard deviation

To evaluate the effects of improvement of environmental factors on self-estimated performance, PP-0 in Control was compared with PP-0s in all other conditions. The results of the effects of difference of environmental factor on PP-0 are shown in Figure 6. PP-0s in N, TN, TL, NL, TNL were significantly higher than that in Control (p<0.01). When one environmental condition was differed, the environmental factor whose difference affected PP-0 was acoustic, thermal and lighting environment in that order. The combinations of the two environmental factors whose differences affected PP-0 were acoustic and thermal, thermal and lighting, acoustic and lighting in that order.

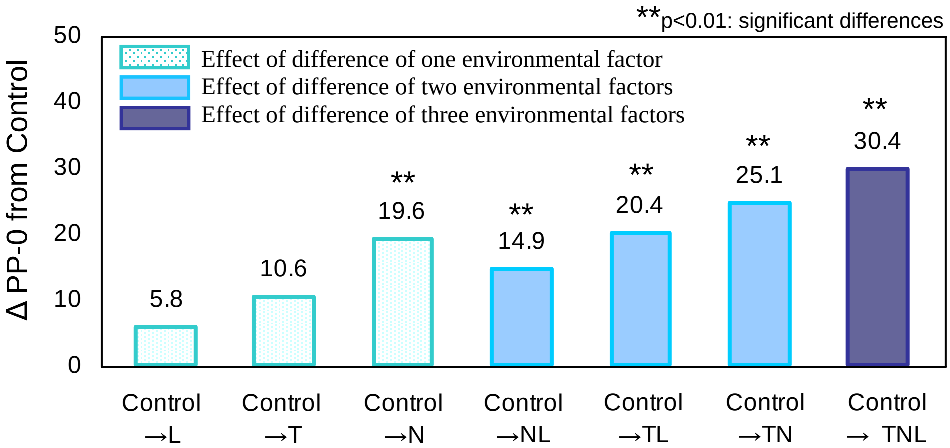

Figure 6. Effects of difference of environmental factors on PP-0

Relationship between Satisfaction with Indoor Environment or Fatigue and Performance

To evaluate how much satisfaction with indoor environment and fatigue levels affect task performance, the relationship between satisfaction with indoor environment or general rate of fatigue and the Z-score of multiplication task were examined. The data set of satisfaction with indoor environment by the interval of 0.2 was averaged, the data set of general rate of fatigue by the interval of 10% was averaged, and corresponding Z-score was also averaged. The relationships between satisfaction with indoor environment or general rate of fatigue and the Z-score are shown in Figure 7. The Z-score was weakly related to subjects' satisfaction with indoor environment (r=0.43) and was inversely weakly related to fatigue(r=-0.42).

Relationship between Satisfaction with Indoor Environment or Fatigue and PP-0

To evaluate how much satisfaction with indoor environment and fatigue levels affect self-estimated performance, the relationship between satisfaction with indoor environment or general rate of fatigue and PP-0 were examined. The data set of satisfaction with indoor environment by the interval of 0.2 was averaged, the data set of general rate of fatigue by the interval of 10% was averaged, and corresponding PP-0 was also averaged. The relationships between satisfaction with indoor environment or general rate of fatigue and PP-0 are shown in Figure 8. PP-0 was related to subjects' satisfaction with indoor environment (r=0.80) and was inversely related to fatigue(r=-0.87).

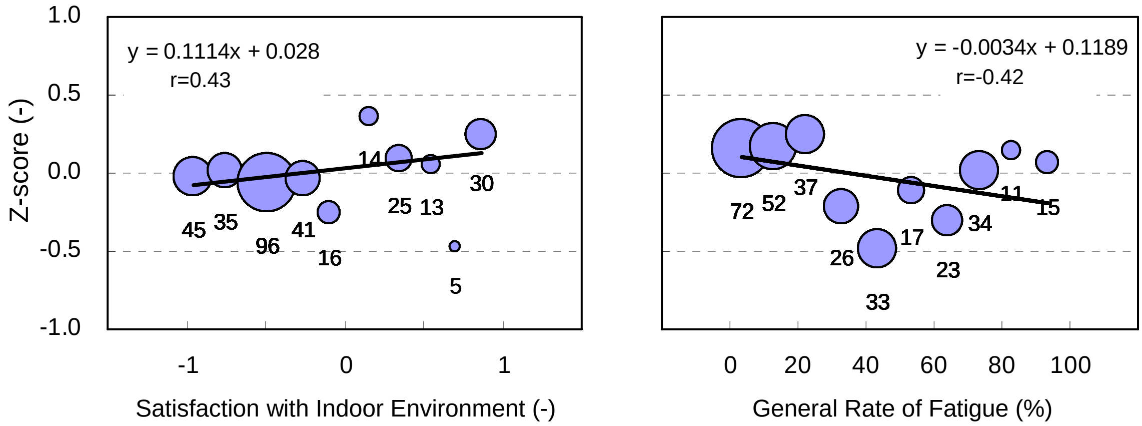

Figure 7. Relationship between satisfaction with indoor environment or fatigue and Z-score

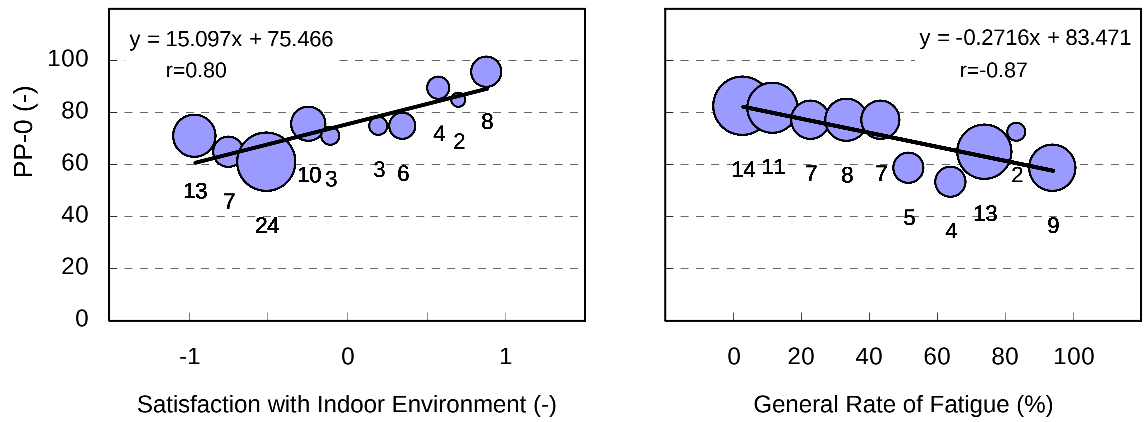

Figure 8. Relationship between satisfaction with indoor environment or fatigue and PP-0

DISCUSSIONS

It was shown that the environments in TN and TNL were evaluated as satisfied, and those in all other conditions were evaluated as dissatisfied by the subjects. The general rate of fatigue in the conditions of "28°C" was higher than that in the conditions "25°C" and that in the conditions "with traffic noise" was higher than that in the conditions "without noise". These results suggested that the effects of improvement of thermal and acoustic environment on subjects' satisfaction with indoor environment and fatigue levels were higher than that of lighting environment in the environmental combinations of this study.

Clausen and Wyon [4] conducted a subjective experiment to investigate people's priorities to IEQ and whether individual choice of working environment affects work performance and individual perceptions of the indoor environment. It was reported that the possibility to individually choose which parameters to improve within a given budget had a positive effect on acceptability with the overall indoor environment [4]. In our experiment, there was not obvious change of task performance between the conditions. On the other hand, the results of predicted-performance-vote showed that subjects gave priority to improve thermal and acoustic environment rather than lighting environment, and effects of improvement of thermal and acoustic environment on predicted performance were higher than that of lighting environment. Predicted performance can be practical as self-estimated performance under a certain environmental condition, and subjects also seemed to evaluate how the environmental condition was well-suited to perform task by predicted performance. It is possible to figure out the environmental factors occupants want to improve, and they can simply estimate the effects of improvement of environmental factors on their performance by using the predicted-performance-vote.

The task performance and PP-0 was related to occupants' satisfaction with indoor environment and was inversely related to fatigue. The relationships with task performance were rather weak in this experiment, however, strong relationship between task performance and fatigue was reported for the long time exposure [5]. Predicted-performance-vote is possible to simply estimate the effects of IEQ on output-side of productivity for short term study.

CONCLUSIONS

A subjective experiment was conducted to evaluate the effects of improvement of indoor environmental quality on occupant's performance. Subjects were exposed eight conditions and performed thirty-minutes multiplication task four times in each condition.

1) The effects of improvement of thermal and acoustic environment on subjects' satisfaction with indoor environment and fatigue levels were higher than that of lighting environment in the environmental combinations of this study.
2) The results of predicted-performance-vote indicated that the conditions considered to be well-suited to office work were in the order from Control, L, N, TL, T, NL, TN and TNL.
3) The Z-score and PP-0 were related to subjects' satisfaction with indoor environment and were inversely related to general rate of fatigue. Subjects evaluated themselves well-performed when satisfaction with indoor environment was higher and fatigue was lower. To promote increase of productivity in the office, it is effective to reduce occupants' dissatisfaction with IEQ and fatigue.
4) It is possible to figure out the environmental factors occupants want to improve, and they can simply estimate the effects of improvement of environmental factors on their performance by using the predicted-performance-vote.

ACKNOWLEDGEMENT

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The Health Impact Of Space Planning Policies In Relation To Walking And Exercise In The Workplace

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SUMMARY

This paper presents an evidential review of current knowledge on the issue of walking versus sedentary behaviour in office environments. It considers the medical evidence in relation to cardio-vascular activity and the long-term effect of sedentary compared to mobile work routines. The paper then goes on to consider the concept of 'embedded health' within buildings, reflected in the way that buildings are designed and managed. It highlights in particular the role of the facilities manager in enabling such embedded health. The argument in the paper is that health activities cannot be assumed to occur simply by the provision of health facilities. Professionals, with busy working lives, working long hours, inevitably neglect health activities. It is therefore incumbent on employers to 'design in' healthy behaviour as part of the daily work routine. The author argues that current space planning practices are intent on achieving quite the reverse outcome.

INTRODUCTION

A mounting body of evidence is highlighting the widespread problem of obesity, particularly in developing countries. According to a recent World Health Organization report [1] "obesity poses one of the greatest public health challenges for the 21st century, with particularly alarming trends in several parts of the world, including the WHO European Region. More than 75% of all deaths in the European Region are caused by non-communicable diseases, the highest proportion in the world. Coronary heart disease (CHD) is the most common cause of premature death, alone accounting for 16% and 12% of all premature deaths in men and women, respectively". This problem is influenced to a significant extent by work routines and behaviours. The office worker population represents a significant proportion of the working population, particularly with the emergence of the knowledge worker and the increased growth of the service sector. In this paper we consider the impact that facilities design and space planning has on the health of individuals. In particular it considers the cardio-vascular health impact, largely arising from walking. The following hypothesis is considered;

'The cardio-vascular health of office workers is significantly affected by the spatial design, adjacencies and routines imposed by facilities management policies'.

One example where office workers may consciously choose to take the 'healthy option' in the working environment is the 'lift or stairs' scenario. Such a decision may impact on travel time to a destination. Furthermore it may not be an individual decision but may be influenced by the consensus of a group. The dilemma then involves a number of competing values, some of which may be in relation to productivity (i.e. the organization); some to the group (i.e. interests of the group) and some to personal wellbeing (or indeed reluctance to exercise).

However, many of the 'embedded' health opportunities of facilities are prevented: no such choice exists (compared to the lift or stairs dilemma). Individuals cannot choose to take the more energetic route to an office coffee area, if the position of such a facility has been positioned in order to minimize travel distance. Indeed, evidence suggests that unless individuals are forced to travel to a more remote facility (through lack of choice) individuals will invariably choose the most proximal facility. The question then arises as to whether the facilities manager should socially engineer behaviours that force individuals to be more mobile. This paper firstly considers current medical evidence in relation to workplace behaviour and health. It then goes on to consider the potential impact that space planning and service delivery can have on human energy expenditure during the course of a working day.

THE MEDICAL EVIDENCE RELATING TO HEART DISEASE AND EXERCISE AMONGST WORKERS

The evidence suggesting an association between vigorous exercise and the apparent protection it affords against heart disease stems back as far as the ancient Greeks two thousand years ago. Both Hippocrates and Gallen argued that a lack of exercise was detrimental to human health. Paffenbarger *et al* [2] document how such evidence continued to develop throughout history culminating in a vast body of evidence in modern day epidemiology. By the *Middle Ages,* the Italian physician Bernardini Ramazzini (1690-1731), founder of modern day epidemiology, studied the health of various tradesmen. He came to the conclusion that messengers and similar 'fleet footed' trades managed to avoid many of the occupational hazards that other trades such as tailors and cobblers experienced. Ramazzini (cited in [2]) stated: 'let tailors be advised to take physical exercise at any rate on holidays. Let them make the best use they can of some one day, and so to counteract the harm done by many days of sedentary life'.

Today a wealth of research has been undertaken to uncover the relationship between physical activity and health, particularly in relation to heart disease. This research has covered many continents, social groups, age bands, ethnic groups and specific genders. Much of this has been informed by the work of the Morris Group who by 1973 were starting to show clear evidence that an association existed between exercise levels and the occurrence of coronary heart disease (CHD). An initial study was undertaken by Morris *et al* [3] involving 9400 men aged 45-64 years. All of the participants were civil servants with no history of CHD. The study showed a reduction by more than a half for CHD incidence, non-fatal and fatal, with moderately vigorous or vigorous exercise. The study focused on leisure based activities since there was considered to be insufficient cardiovascular activity in the workplace to differentiate between the subjects studied. In a later study Morris *et al* [4] followed middle management civial servants for non-fatal and fatal diseases. The subjects were initially required to complete five minute logs over a two day period. This was then followed up some years later by detailed mail-back questionnaires on their health habits and health status. Having traced the subsequent occurrence of heart disease, Morris *et al*[4] identified two categories of worker: that undertook 'vigorous' exercise; those that 'moderately vigorous exercise' for whom there was a demonstrable association between vigorous exercise and coronary heart disease (CHD). Vigorous exercise consisted of sports undertaken twice or more a week, including swimming, fast regular walking (>4 mph) or cycling. The findings suggested that short bouts of regular exercise were most effective in combating heart disease. Intensity, followed by frequency are more important than duration: most exercises being less than 20 minutes in duration.

The early studies by the Morris Group focused primarily on leisure activities of office workers. However more recent work has considered the health effects of activity during the work period as well as activities associated with getting to and from work. For example, a study extending from six to sixteen years in Osaka, Japan (see [5]) involving over sixteen thousand Japanese men between 35 to 60 years considered the effect of walking to work, in relation to the risk of hypertension. The findings suggested that the activity of walking to work was associated with a decreased risk for incident hypertension, even after adjustment for age, body mass index and other factors.

CARDIO-VASCULAR HEALTH IN THE WORKPLACE

For many organizations seeking to promote healthy living, it is not possible to legislate against ‘unhealthy modes’ of transport to and from work: this, despite the increasing efforts to promote ‘green policies’ and discourage driving by car to work. However, many organizations, as well as encouraging gym membership (either on-site or through a gym membership scheme) have also introduced exercise in the workplace. Using such an ‘interventionist’ approach, Chan *et al*[6] recruited participants from five workplaces, where jobs were moderately or highly sedentary. A recognized exercise regime was introduced to the workplaces and it was hypothesized that this exercise regime would increase participants’ daily ambulatory activity. This was determined using pedometers. The pedometers served a three-fold purpose; (1) for feedback on baseline and increasing levels of activity during the programme; (2) as a motivational device and (3) to objectively measure changes in physical activity. Changes in body weight index (BMI); waist girth and heart rate were measured during the study. Participant’s heart rate as well as waist girth and body weight reduced significantly during the intervention period although there were no significant changes in systolic or diastolic blood pressure.

So far, we have considered only the effects of exercise on the effects of cardio-vascular health. However, the psychosocial work environment has also be considered as a possible source of stress and hence hypertension. In a study by Unden *et al* [7] a study group of 148 working men and women from seven occupational groups including teachers, musicians, policemen, engineers and saw mill workers were examined. They underwent 24 hour recordings of electrocardiograms (ECG) to determine cardiovascular health. Psychosocial work characteristics were determined by means of the following parameters: social support at work; work demand; decision latitude; and skill direction. The results suggested that mean heart rates are significantly higher in people reporting low social support at work. This applied to each of the seven occupational study groups considered.

The willingness of employees to participate in workplace exercise was the subject of a study by Waikar and Bradshaw [8]. The attention in this study of sedentary workers was not on cardiovascular health but on musculoskeletal exercise to overcome problems associated with working long hours on VDT display screens. However, many of the findings are pertinent to the present discussion, since they provide insight into the propensity of individuals to engage in exercise during working hours. In attempting to design a suitable exercise regime, workers were asked to comment on:

- The ease of comprehending exercise instructions
- Degree of difficulty of the exercise
- Degree of privacy when exercising
- How exercises are initiated
- Length of exercise breaks

- Level of embarrassment of exercises
- Places where exercises are completed
- Types of exercise

The results indicated that almost two-thirds of respondents do not exercise on their own during work houirs to relieve pain and discomfort. However, two-thirds indicated that they would be willing to participate in formal exercise programmes. Around forty per cent suggested that embarrassment is not an issue when it comes to exercise and one-third of the respondents prefer to exercise in a group. ‘Types of exercise’ and ‘degree of difficulty of exercises’ were the two most important categories for the participants.

FACILITIES MANAGEMENT AND ITS ROLE IN CARDIOVASCULAR HEALTH

In a paper entitled ‘The escalating pandemics of obesity and sedentary lifestyle’, Manson *et al* [9] provide a blueprint for action on the part of health care professionals in curbing the obesity epidemic associated with sedentary work. It suggests that only a few minutes of a clinician’s time will facilitate more effective intervention related to obesity. However, in view of the previous discussion it is clear that intervention in the form of health programmes in organizations are replete with difficulties. Many people are reluctant to participate in exercises as a group, in part due to embarrassment. Moreover, heavy work schedules often deter employees from taking time out.

For these reasons, this paper suggests that exercise be designed into the daily routine of employees. This could in principle be crafted into a worker’s daily activity without any ongoing intervention on the part of the organization or indeed by any planned approach on the part of the employee. In this sense the activities could be seen as being ‘embedded’ into the work, by virtue of the design of the facility. It is worth considering typical design and facilities management planning decisions that will impact on the amount of energy expended by employees in the course of the day. These include:

- Vertical circulation paths between individuals and resources that necessitate walking
- Horizontal circulation paths between individuals and resources that necessitate stair climbing or descent

The type of resources that might be sought and necessitate walking include printing facilities, photocopying facilities, archives, coffee areas, cafeterias and wash rooms. The location of people and resources within a facility as well as the overall size of the facility and layout efficiency will determine the likely amount of expended energy in a given day. Facilities managers, tasked with optimizing the effectiveness of an organization by efficient space planning will typically be concerned with the following:

- Minimizing travel time thus liberating ‘dead’ time to enable effective work to be done
- Minimizing the disruption to ‘neighbours’ when circulating
- Maximizing space use efficiency, thus eliminating the amount of unused space
- Maximizing productive interaction, both formal and informal
- Maximizing convenience, providing resources at hand with minimal interruption
- Maximizing use of costly equipment such as printers through sharing and networking

Many of these objectives are consistent with attempts to reduce the sedentary behaviour of employees. For example, the increased recognition of the role of informal interaction in organizations (with the advent of the flattened organization) provokes a need to ‘mingle’.

Often, this involves moving to informal meeting areas such as coffee bars or kitchens. However, the association of ‘interactions’ with some form of consumption activity may itself undermine any such benefits. Moreover, the increasing practice in facilities management of supplying the catering needs on location (e.g. at a business meeting), whilst stimulating the socialization element, will also obviate the need to walk. Pressures to make more efficient use of expensive office equipment has stimulated more widespread use of shared resources such as printers. This in turn may necessitate that employees get up from their workstations and move around. Furthermore, with the advent of the ‘activity-settings’ workplace solution (a term originally coined by Stone and Luchetti [10]), employees are encouraged to migrate from one work area to another, to attain a work environment suited to the task, whether it be private study, collaborative working or meetings.

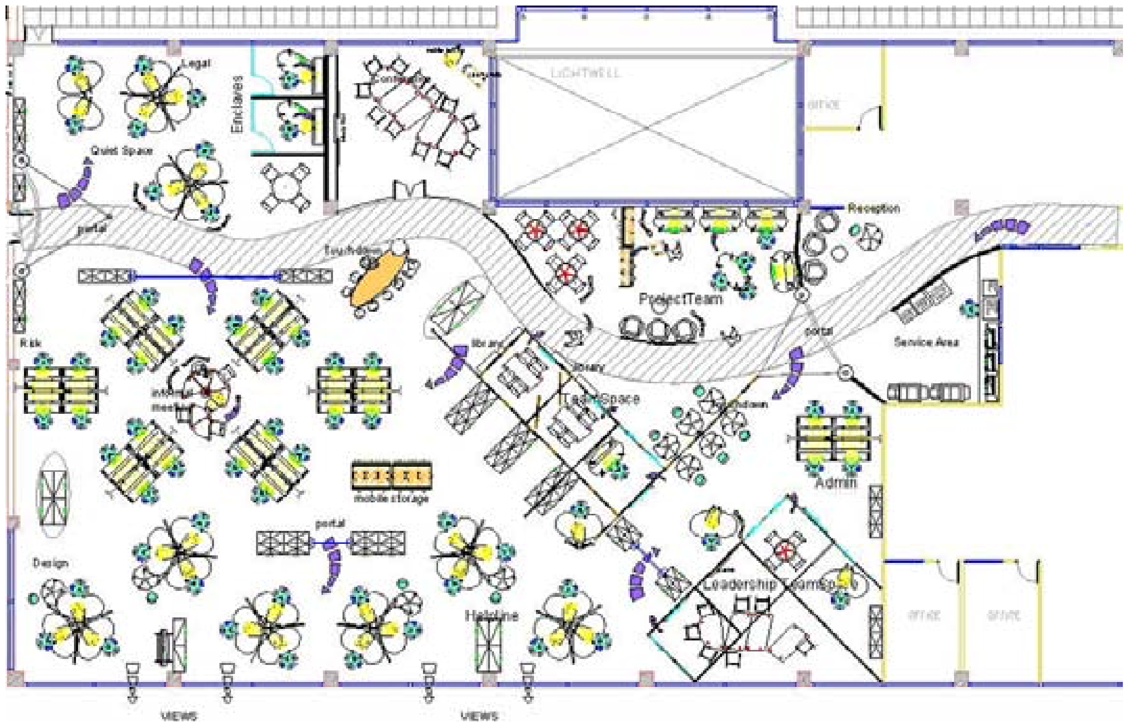

Figure 1. Modern ‘activity-settings’ space plan.

THE FACILITIES MANAGER’S DECISION RULE

Undoubtedly, the overriding rule used by facilities managers, which in turn determines the ‘embedded’ health of a building layout, is the ‘stacking’ and ‘blocking’ plan. Both techniques are routinely used as part of a ‘computer-aided facilities management’ (CAFM) package. They incorporate algorithms that enable the minimization of travel times, based on ‘affinity diagrams’ or ‘interaction matrices’. These matrices indicate the relative strength of relationships that exist or are sought between departments and resources/services within a facility. The optimized stacking plan indicates how departments and individuals are dispersed throughout the building to enable efficient functioning of the organization. Those parts of the organization that require proximity to one another are typically located on the same floor, since a floor separation is detrimental to such intense and regular exchanges.

It is however, the intention in the remainder of this paper to consider the implications of these decision rules used by facilities managers. The assumptions upon which they are based are challenged and the author suggests a more flexible contingency approach to space planning that embraces the concept of 'embedded' health in buildings.

IS TRAVEL TIME 'WASTED TIME'?

Implicit in space planning systems widely used today is the belief that the necessity to travel (walk) is inherently bad. This, despite the growing evidence that the sedentary nature of work brought about by reliance on the car and the computer have 'designed out' exercise from the normal work routine. In particular, the ubiquity of email has circumvented the need to walk to a colleague's workstation, even if only a few metres away. From an organizational perspective, travel time is universally considered to be undesirable. Not only is it perceived as 'lost' time: it also is seen to hamper 'processes', that are seen to be affected by layout. However, some of these assumptions need to be challenged.

- Work involving one-to-one communication can be undertaken whilst walking. Little is known about how the activity of walking affects the richness or productivity of such interactions. Indeed it is possible that an active behaviour may positively affect the interaction.
- Adjacency to support 'process' is significantly less important in modern work environments where most knowledge exchange is played out in a virtual environment (e.g. exchange of documents). Thus the imperative for close proximity to support process is less evident.
- Organisations are seeking to stimulate 'non-routine' rather than 'routine' interactions whereby individuals exchange ideas and approaches with people in different departments or areas. This cross-fertilisation thus challenges 'the way we do it here' mentality and encourages innovation and inter-group collaboration.

Quite at odds with organizational desires to increase efficiency, compact designs may only be producing organizational 'straight jackets' imposed by space plans.

DOES 'EMBEDDED' HEALTH IN BUILDINGS CONFLICT WITH ACCESSIBILITY?

One of the potential conflicts between the development of 'embedded' health in buildings is the need to satisfy accessibility requirements. Travel distance and vertical separation (i.e. stairs) are potential impediments to accessibility. Accessibility (or just access) refers to the ability to reach desired goods, services, activities and destinations (collectively called opportunities). Accessibility can be affected by (modified from [11]):

- *"Mobility*, that is, physical movement. Mobility can be provided by walking, lifts and escalators.
- *Mobility substitutes*, such as telecommunications and delivery services (including delivery of post and catering services)..
- *Transportation system connectivity*, which refers to the directness of links and the density of connections in the path network.
- *Land use (spatial use)*, that is, the geographic distribution of activities and destinations. The dispersion of common destination increases the amount of mobility needed to access goods, services and activities, reducing accessibility."

Accessibility is affected by a number of competing factors including time, money (in the case of transport usage), security, discomfort and risk. Each of these issues needs to be considered in relation to the ‘embedded’ health in terms of the challenging physical requirements of the workplace.

DISCUSSION

This paper has presented an argument for the creation of ‘embedded’ health in buildings. In other words, an environment which coerces occupants into physical activity, as part of their everyday work routine. One such example is stair walking (i.e. not taking the lift). It recognizes the fact that direct intervention involving scheduled workplace exercise activities are often difficult to implement, largely as a result of apathy and embarrassment on the part of employees (particularly in Western societies). Furthermore, provision of specialized gym facilities may fail to attract those employees most in need of exercise and who are unable to commit time to exercise activities, because of their busy working day.

The paper also argues that facilities management decisions in relation to stacking and blocking have a fundamental influence on the ‘embedded’ health of buildings. This profession needs to carefully reexamine their assumptions in relation to space utilization and travel time. Is travel distance something that should always be minimized? Accessibility and the ‘embedded’ health of buildings need not be conflicting objectives. Travel distance is only one of the factors that affect the accessibility of environments. A contingent approach, rather than a ‘one-size fits all’ approach may be most appropriate (for example, the choice of taking the lift or stairs). Undoubtedly we are only just beginning to wake up to the immensity of the obesity challenge in modern workplaces. This paper has attempted to highlight the pervasive effect of facilities management decisions in this context. The questions raised in this paper present the framework for a research project currently being formulated, to understand the impact of space planning on the cardio-vascular activity of office employees.

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References should be numbered consecutively in the order in which they are first mentioned in the text and identified in the text, tables or figures and legends by Arabic numbers in square brackets [1]. If a publication has four or fewer authors, all the authors are listed. If there are more than four, list the first three authors and add "et al".

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On the Effect of Thermal Environment on Meat and Carcass Quality in Swine Raised on Privately-Owned Pig Farms

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SUMMARY

Environmental temperature in which perform commercial hybrids on privately-owned family farms in the Banat area (România) has a strong impact on the valorisation of feed and particularly on energy, as there is an optimal temperature at which feed is valorised with maximum efficiency resulting, upon slaughter, in quality carcass ranging at a level of objective graduation depending on lean meat percentage. Fat deposits are thicker in swine bred at low temperatures (35.2%) compared to swine bred during fattening at 16-18^{0}C, the average of lumbar fat being 2.9 cm, of abdominal fat being 1.9 cm, and of dorsal fat being 3.1 cm. The percentage of lean meat in commercial hybrids on family exploitations is 56.06%, and high quality carcass represent 79%, with a meat: fat ratio of 1.8/1, i.e. neatly superior values compared to animals exploited in sheds with non-controlled temperature and having access to the paddock during winter.

INTRODUCTION

During their life, swine are exposed to different variations of temperature as they are exploited in different systems (intensive, semi-intensive, and extensive), and on pig farms thermal conditions are not always constant. Even in the sheds ensuring constant temperature there are temperature variations due either to improper mechanical control or to the variation of external environmental factors.

Research [2, 3, 5] concerning fattening swine physiology in different thermal environments were carried out on animals bred in constant thermal environments or, in some cases, on animals moved from a constant environment to other environments with different temperatures, showed the impact of air temperature on some physiological indices in swine (body surface temperature, body temperature, breathing rate, and pulse rate).

Environmental temperature influences at the same time all fundamental biological processes: global energetic metabolism, growth and development processes, carcass quality, and pork quality [1].

Temperature is the environmental factor that exerts the strongest influence on fundamental biological processes – metabolic processes – and, through them, on all the rest of biological processes and phenomena.

The dependence on temperature of metabolic processes intensity reflects its dependence on chemical reaction speed they rely on. It is known that a raise in temperature determines a raise of kinetic energy of the molecules that react and, therefore, a raise of the chemical reaction rate. In the area of thermal neutrality, energy expense is minimal, which characterises basal metabolism. When the temperature of the environment decreases until it reaches the critical point, there is a quick and intense raise of energetic metabolism, i.e. a defence reaction against cold.

As growth and development are biological processes depending strictly on metabolism and displaying simultaneously, optimal temperature ensuring their maximum rate varies depending on the different stages of swine life. In general, it is admitted that growth rate in commercial hybrids is lower during cold seasons, which means that environmental temperature must be controlled as swine are homoeothermic animals and sweat little.

Experiments carried out by different authors [5, 6] show that commercial swine have, during the growth and fattening periods, an average daily gain 2-4 g higher when shed temperature was 25^0C compared to higher temperatures (>25^0C), when there is a decrease of weight gain; this means that the larger the body weight, the more the animal suffers because of high temperatures.

Though they practice heating sheds for fattening swine, environmental temperature has a strong impact on valorising feed and, particularly, energy. For each category of swine there is an optimal temperature at which feed is valorised with maximum efficiency and, as a result, carcass and meat quality meets EU and objective grading system requirements. Maximum growth rate in swine is reached at temperatures between 15 and 23^0C in an intensive system, and 12 and 17^0C in a family system. Commercial hybrids are strongly affected by temperature changes than fattening swine. At low temperatures, part of the energy is used in thermal regulation through burning intensification and heat release [3].

Most research show that, the more temperature differs from the thermal comfort area, the poorer the valorisation of feed is, sudden changes and particularly the dropping of temperature lead to an increase of feed consumption and to a decrease of swine's performance with all the implications on meat quality, i.e. low-quality carcass upon slaughter (classes O and P).

METHODS

Our experiment lasted for 90 days. We took under study two lots of commercial swine [(Large White x Landrace) x Duroc], as follows:

- the first lot, made up of 100 heads, was raised at an environmental temperature of 8-10^2C, in a shed with paddock, with no heating source, on a straw litter, and was put to fattening when weighing 40.7 kg at the age of 70±2 days;

- the second lot, made up of 100 heads, was raised at an environmental temperature of 16-18^0C, in a shed with paddock, but without any access to it, with temperature regulating facilities, full pavement over 40% of the box area and grill over 60% of it, and was put to fattening when having the same weight and age as the first lot.

During the study period we monitored the average daily gain for the 90 days of fattening and specific consumption at different temperatures, aiming at developing a fattening technology for family-owned farms in the Banat area (Romania) that have such shed with paddocks, analysing carcass and meat quality and classifying carcass and meat depending on the EUROP objective graduation.

We measured fat layer thickness of carcasses in the 100 heads of each lot and measured the percentage of meat per carcass, as well as the meat: fat ratio in order to point out the impact of temperature on the raising-fattening process and on carcass quality to be able to implement the most convenient fattening system for privately-owned family farms in the Banat area, taking into account the fact that these last years these farms have no longer bred swine, preferring to sell the grains to larger pig farms, though they are not always satisfied with the prices they can get. Implementing such fattening technologies could be an alternative source of income for these farms, particularly if we take into account the fact that they own the infrastructure (sheds) and, for the surplus of grains, they can take in commercial hybrids for fattening, thus obtaining profit.

RESULTS

During the 90 days of fattening, the two lots of animals were fed mixed feed specific to the fattening period and having the nutritious value mentioned in Table 1.

Table 1. Nutritious value of feed supplied to commercial swine

Metabolisable energy	Matter	Raw protein	Raw cellulose	Raw fat	Salt
Cal/kg	%	%	%	%	%
3093.02	84.12	16.21	3.67	2.56	0.43

Swine were fed at leisure, feed being prepared with the available grain (maize, barley, and wheat in shares of 36.50%, 10%, and 20%), and purchased protein plants (sunflower 5%, soy 11%, meat flour 6%), microelements, calcium carbonate, salt, and protein-vitamin-mineral mixture (grower).

The average daily gain in the first lot fattened at a temperature of 8-10^{0}C varied between 595 and 709 g with a feed conversion index of 3.65-4.22 kg of feed for 1 kg of gain.

The second lot reached an average daily gain of 631-817 g for the 3 fattening months with a feed conversion index of 2.68-3.10 kg of feed for 1 kg of gain. The temperature of 16-18^{0}C in the shed of the second lot had a positive impact on growth rate and a lower specific consumption of feed per 1 kg of gain. The swine in the lot fed at a temperature of 16-18^{0}C needed a lower amount of energy for thermal regulation, most of the energy being used for growth and development.

In the first lot there were no shivering during the fattening period even at temperatures of 8-10^{0}C, because they were exploited on straw litter which ensured some kind of protection from cold: results were almost as good as in swine exploited in sheds with temperatures of 16-18^{0}C, with expenses somewhat higher because of the energy consumption for maintaining temperature constant (Table 2).

At the age of 160±2 days, the two lots reached an average weight of 100.34 kg (Lot I) and 105.83 kg (Lot II): the 100 heads in Lot I being slaughtered and classified in accordance with the objective graduation of the carcass, they ranged as shown in Tables 3 and 4.

Table 2. Average daily gain and specific consumption depending on temperature

Lot	Fattening period (days)	Average daily gain (g)	Specific consumption (kg of feed/kg of weight)	Shed temperature
I	70-100 101-130 131-160	505 684 709	3.65 3.93 4.22	8-10^{0}C
II	70-100 101-130 131-160	631 723 817	3.10 2.90 2.68	16-18^{0}C

Table 3. Swine carcass classification in Lot I fattened at 8-10^{0}C

Class	Class (lean meat)	Number of carcasses	%	Lean meat (%)
E	= 55%	5	5.0	56.30
U	50-55%	36	36.0	53.80
R	45-50%	32	32.0	49.70
O	40-45%	20	20.0	44.50
P	< 40%	7	7.0	39.60
Total		100	100	50.59

Results of classifying carcasses of commercial swine in the two lots are shown in Figure 1. We can see that swine fattened at low temperatures have a higher percentage of fat per carcass compared to the swine fattened at a temperature of 16-18^{0}C, having 56.06% lean meat compared to 50.59% in Lot I.

Table 4. Swine carcass classification in Lot II fattened at 16-18^{0}C

Class	Class (lean meat)	Number of carcasses	%	Lean meat (%)
E	= 55%	14	14.0	56.80
U	50-55%	65	65.0	54.20
R	45-50%	19	19.0	49.80
O	40-45%	2	2.0	44.80
P	< 40%	-	-	-
Total		100	100.0	56.06

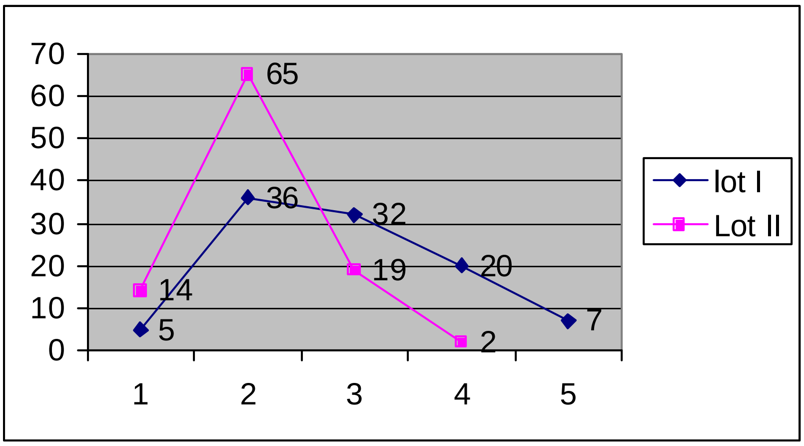

Figure 1. Results of carcass classification in the two lots depending on shed fattening temperature

Measurements made concerning the thickness of the fat layer in the carcasses of the two lots (cervix – between vertebrae 6 and 7, thorax – between vertebrae 19 and 20, lumbar – at the fore extremity of the *gluteus medius*, lumbar – at the middle of the *gluteus medius*, lumbar –

at the hind extremity of the *gluteus medius*, abdominal – at the sternum, abdominal – at the belly button, abdominal – at the side) shows that the average of lumbar measurements in Lot I is 3.7 cm, of abdominal ones is 2.0 cm, and of dorsal ones is 3.1, which shows that low fattening temperatures result in a thickening of the fat layer (Table 5).

Table 5. Thickness of the fat layer in commercial swine carcasses fattened at different temperatures

Lot	Average of lumbar measurements (cm)	Average of abdominal measurements (cm)	Average of dorsal measurements (cm)
I	3.7	2.0	3.8
II	2.9	1.9	3.1
Differences ±	+0.8	+0.1	0.7

Cutting carcasses showed the meat: fat ratio in Lot I (1.43: 1) and in Lot II (1.8: 1) (Table 6).

Table 6. Meat: fat ratio in the two lots

Lot	Lean meat (%)	Fat (%)	Bones (%)	Total (%)
I	50.59	35.20	14.21	100
II	56.06	31.13	12.81	100
Differences ±	+5.47	-4.07	-1.40	

Studies carried out to establish the influence of temperature on commercial swine hybrids raised and fattened show that at low temperatures (8-10^{0}C) the maximum average daily gain was 709 g with a conversion index of 4.22 kg of feed per 1 kg of gain. Animals exploited in environments with controlled temperature (16-18^{0}C) reach a maximum average daily gain of 817 g with a conversion index of 2.68 kg of feed per 1 kg of gain, as they are no longer forced to consume energy for thermal regulation, all of it being converted into weight gain.

CONCLUSIONS

Animals raised at low temperatures produce fatter carcasses than animals raised at higher temperatures (16-18^{0}) and have a close gain rate. The percentage of lean meat per carcass in hybrids raised at high temperatures is 56.06%: thus, 79% of their carcasses range in higher quality classes, compared to only 41% in the carcasses of animals raised at low temperature. The meat: fat ratio also records higher values in commercial hybrids exploited in optimal thermal conditions.

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Health and wellbeing in a deep plan air-conditioned commercial property

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SUMMARY

The purpose of this paper is to assess the health and wellbeing implications of working in a deep plan office building in a temperate climate and illustrate how office design can add to or detract from productivity.

The authors have carried out post occupancy evaluation surveys on employees before and after the move from a naturally ventilated, shallow plan office to a deep plan, air-conditioned office. The workforce and workload remained constant, which enabled true comparison between building types. The public sector employer specified an environmentally friendly building that incorporates health and wellbeing factors.

It is anticipated that in future, deep plan buildings will become increasing expensive to service and maintain. In this study there was no discernable increase in productivity and a significant number of correspondents preferred the shallow plan old building.

INTRODUCTION

This research tracks public sector employees as they move from an old office building into a purpose-built, environmentally sound, office building. It hypothesises that the attempts to change the image and effectiveness of an organisation by changing workspace layouts has been ineffective and considers the consequent implications for individuals working in that organisation. It looks at the likely relationship between health and wellbeing of individuals and productivity.

BACKGROUND

Productivity and the workplace

The Commission for Architecture and the Built Environment (CABE) in the UK reports [1] that differences in productivity can be as high as 25 per cent between comfortable and uncomfortable staff. Individuals react differently to different stimuli, but the most important factors in achieving health and comfort are air quality, temperature, overall comfort, noise and lighting. Most of these environmental factors can be readily and easily measured in any office, particularly in an era with the widespread use of Building Management Systems (BMS) to control mechanical and electrical building services plant.

Clements-Croome and Baizhan [2] however draw a distinction between 'comfort' and 'wellbeing', inferring that although comfort is an important factor in productivity, wellbeing is a 'prime requisite'. They suggests that productivity depends on:

> *"Good concentration, technical competence, effective organisation and management, a responsive environment and a good sense of well-being."*

Putting personal environmental control into occupants' hands can also be important for both comfort and productivity [3]. CABE suggests that variances in individual preference and the growing importance of staff autonomy both point to the value of introducing a means of personal control to the greatest degree consistent with efficient operation of the air conditioning, lighting and related building systems. Wyon [4] has indicated that a good indoor environment may only satisfy eighty per cent of an occupants' perceived level of comfort, with individual control satisfying the remaining twenty per cent. It has also been suggested in a number of studies that in order to improve productivity, firstly comfortable conditions have to be provided (and conditions should be improved periodically) and, secondly, occupants' requirements have to be met rapidly.

Workspace Development

Historically in the UK, public sector workspace has developed reactively and in an *ad hoc* manner, often inhabiting impressive buildings designed to reflect status and support top-down government structures. As a result of the wide scale expansion in the public sector workforce between 1945 and 1950, purpose-built structures were designed to accommodate whole government organisations under one roof [5]. Internally, these were organised into corridor offices [6] and continued to reflect status according to grade, and the horizontal and vertical boundaries of the hierarchical structure [5]. These slab blocks in the 'modernist' style were identified by Duffy in his seminal work 'The Responsible Workplace' as the most flexible form to accommodate the work styles, hive, club, club and den he classified [6].

More recently there are a range of workplace and cultural changes affecting the way the public services operate; the geography of an organisation; and redesign of workspace has been introduced to reinforce a culture of internal communication and break down traditional hierarchy [1]. The aim of creating a flexible working environment where staff would feel valued and well motivated, whilst improving the efficiency of space usage was a key aspect of the reform and modernisation agenda.

One aspect of modernisation is the use of the workplace to drive business change [5] which may be achieved through: *efficiency* - making economic use of real estate and driving down occupancy costs; *effectiveness* - using space to support the way that people work, improving output and quality; and *expression* - communicating messages both to the inhabitants of the building and to those who visit it, to influence the way they think about the organisation [1].

Sir Anthony Turnbull speaking of the refurbishment of the UK government's Treasury Building summarised these aims as:

> *'It has prompted communication, both formal and informal and has encouraged flexible ways of working. Above all it has fostered a feeling of self-confidence and presented an attractive image to the talent we need to recruit.'* [1]

THE STUDY

The Properties

This study examines the employees of two office developments within a city centre in the North East of England. In 2005 the participants moved from a 1960s office building to a new, adjacent, building that had been completed the previous year.

The 1960s building is of concrete frame and cladding construction and is a shallow floorplate building, 14 storeys in height. The building is single glazed with steel frame windows. Heating is by central gas fired boiler with perimeter radiators. All the radiators are fitted with thermostatic valves, allowing some level of occupant temperature control. Social facilities in the form of 'tea points' are provided on each floor.

No air conditioning system is installed within the building and the working areas are naturally ventilated by way of opening windows. Lighting to all work areas is provided by luminaires recessed into suspended ceilings, with additional task lighting in the form of desk lamps. No specific noise control measures are used within any of the buildings working areas, although meeting and interview rooms are constructed to provide a high level of acoustic performance.

The majority of floors within the building are open plan in nature, although a small number of departments retained cellular offices. Floors within the building each housed approximately fifty staff.

The 2004 building is a deep plan floorplate building constructed with a steel frame and double glazed steel framed windows. It is divided into an east and west wing with seven and four working levels respectively. The majority of floors within the building are open plan with minimal cellular offices constructed for senior staff.

Heating is by gas fired central boiler and radiators, cooling by passive chilled beams. Mechanical ventilation is provided through floor diffusers. A BMS automatically controls the internal environment.

Environmental assessment

The Building Research Establishment's Environmental Assessment Method (BREEAM) is the most widely used environmental assessment tool in the UK and is a voluntary scheme that aims to quantify and reduce the environmental burdens of buildings by rewarding those designs that take positive steps to minimise their environmental impacts. Projects are assessed using a system of credits that are grouped within the following categories: Management, Health and Wellbeing, Energy, Transport, Water, Materials and Waste, Land use and Ecology and Pollution. The assessment process results in a report covering the issues assessed together with a formal certification giving a rating on a scale of 'Pass', through to 'Excellent'. One of the deciding factors for selection of the 2004 building was that it is a BREEAM 'Excellent' building.

Research methods

This research carried out evaluations of comfort and wellbeing (as perceived by occupants) to determine a relationship between these factors and productivity.

Historical methods of productivity research are seen as largely scientific, and conducted in controlled, limited environments [7]. There have been a number of more recent studies considering the interactive and subjective nature of a 'real world' workplace (such as Leaman and Bordass [8]) that utilise occupants' self-assessment of workplace quality and personal productivity levels.

Building performance, and the effect this has on occupant satisfaction and productivity is often measured through the use of post-occupancy evaluation surveys (POE). This form of evaluation came to prominence during the 1980s, with the rise of the facilities management (FM) discipline demanding more information on the buildings they manage, and the development of the Office Environment Survey, conducted to address increasing reports of incidences of sick building syndrome within the UK.

POEs were used in this study and were carried out in 2002 and 2006, before and after the move. The POEs were identical apart from identifying those employees who had moved from the 1960s building. The research has the benefit therefore of drawing upon two identical surveys of the same staff in two offices. In contrast to many research projects trying to relate staff productivity and satisfaction over a number of case studies [9] this has the benefit of having the staff as a 'constant', notwithstanding some turnover between the two surveys.

The questionnaire design allowed respondent staff to indicate their views on a wide variety of issues using 'tick box' responses and the opportunity to provide a written commentary, this data was subsequently analysed to provide quantitative data to compare with the previous office. More interesting in many resects was the qualitative data which was gathered by allowing staff the opportunity to comment on the issues being measured.

Response data from the second survey are reported below with the comparable results from the first survey in brackets.

The second survey was administered to 400 (143) with a response rate of 41% (47). Of the respondents 61% (70) worked in open plan offices occupied by 8 or more persons. 53% (48) of the respondents described themselves as professional or managerial staff, which clearly implies that individual offices are a rare commodity in both buildings. Some 84% of the respondents in the second survey had worked in both offices.

RESULTS

Health and Wellbeing

It is possible to have an environmentally sound building, as defined by a recognised scheme such as BREEAM, but fail to address any health or wellbeing issues for occupants (as BREEAM is not prescriptive, the points may be obtained in other sections, such as Transport). BREEAM attaches considerable weight to health and wellbeing and includes a variety of indicators such as openable windows, proximity to windows to allow a view out and the provision of occupant controlled blinds to control glare. In this instance the building scored very well in the 'Health and Wellbeing' section but lost points on the 'view out' of a window due to the shape of the building. Credit was given for installing openable windows but in practice these are locked shut.

In the previous office, although the building did not have air conditioning and staff described it as *'smelly and stuffy'*, they did have an element of control over their environment to the extent that they had access to opening windows and could open them at will.

In contrast, the staff in the modern, open-plan, building have little or no control over their working conditions. An uncomfortable environment might only be improved by phoning the FM team, requesting more heating or cooling, then waiting for a sensory indication that their request had been dealt with. The situation may be exacerbated by a perception of longer response times that are common due to the practical difficulty in matching the different preferences of groups or individuals.

Layout

People tend to prefer working at low densities rather than at high densities due to them being given more freedom in the workplace. Open-plan layouts typically involve problems of raised noise, visual distraction and reduced privacy as well as an inability to control an individuals' environment referred to above.

There were complaints from staff that they could not concentrate on their work due to noise and distractions from colleagues. For example *'the noise levels make it very difficult to concentrate when people are talking to one another'*. The design of the building has attempted to compensate for this problem by providing 'quiet rooms ' which could be booked for concentrated works, however *'when in quiet rooms you can hear every word from adjoining rooms and the kitchen'*. The 'quiet' rooms have well insulated walls and soundproof doors, however sound is easily transmitted across the suspended ceiling.

Analysis of the questionnaires indicates how important these factors are in practice. The problems do not arise from a switch from cellular to open plan offices as, in the sample 10% fewer staff were working in open plan accommodation. The difficulties arise in the number of people per floor and the configuration of the office. Staff have moved from narrow floor plans with light and ventilation on both sides, to a deep plan building with up to 100 people in a large open space (where it may be 20 metres to the nearest window). In these circumstances staff feel more subjected to control of their environmental conditions via the FM team and the BMS and subject to noise coming from all directions.

There may also be a layout/social aspect to the working conditions revealed by anecdotal evidence from staff. In the previous office each floor had its own social breakout space in the form of a ‘tea-point’. This fostered a sense of community amongst the team working on the floor. In the new building the tea-making / lounge area is equal in area, but is accommodated in two large (anonymous) spaces shared by the whole building, thus losing the team ownership or the social space.

Personal control and response time

Individual occupants need systems to provide comfortable environment and also require systems to respond quickly to avoid their discomfort [10]. It is stated that the occupants become healthier, happier and more productive the more rapid the response times become and that an occupants’ tolerance threshold can be widened by a rapid response.

One of the major findings of the study relates to the control of environment afforded to staff. In the old building the staff were close to windows and although in poor condition were openable to improve ventilation. They had individual control of the ceiling lights and because the standard of lighting was perceived to be poor, had been provided with task lighting on an ad hoc basis. In the new building there was a complete absence of individual control exacerbated by an absence of commissioning of the building services during a rushed occupancy programme.

As a result parts of the building were too cold or too hot, the lighting generally was too bright and the windows, although openable, were locked shut to avoid extraneous natural ventilation compromising the BMS. Staff complained constantly to the FM team with requests to adjust temperatures at the local level (the BMS was addressable down to four workstations). Staff perceptions were of a lack of control over their environmental conditions, an unseen intermediary (the FM team) has to be phoned or e-mailed to request changes, and then there was never any certainty that action had been taken. Understandably, this was a considerable impediment to productivity.

Building commissioning

It is now three years since the building was occupied and the FM team have spent much of that time identifying and correcting defects, for example the location of sensors next to heat sources, which jeopardised the effective running of the BMS. The lighting was found to be well over the 400-lux design level and has been adjusted to a more comfortable specification. Low humidity and the carpet specification had conspired to create a chronic static electricity problem, to such an extent that at one stage the FM team considered issuing gloves to all the staff! (this has now been ‘cured’ by increasing the relative humidity levels).

Any of the problems could have been avoided if the building had undergone a comprehensive commissioning of the building services installation during the hand-over phase. In the event, completing the work was rushed to avoid penalty payments by the developer. It is ironic that the rush to move into the building transferred the commissioning process from the specialist installers to the building occupants.

Discussion

Reviewing the data the most remarkable aspect is the extent to which the new building fails to meet the aspirations of the staff that moved from a building, which by common consent was accepted as providing a poor working environment.

The survey revealed that 35% of respondents felt that their productivity was reduced in the new building whilst only 15% felt that productivity had increased.

Overall, most of the respondents considered that environmental conditions had improved, but in this instance environmental conditions include the ‘newness’ and appearance of the building as well as matters of temperature and humidity.

The most telling statistic is the response to the question ‘In respect to your overall satisfaction with your workspace, given the choice would you prefer to return to your physical conditions at the previous building’. 24% of staff would prefer to return to the conditions of the ‘poor’ building.

CONCLUSIONS

Organisations are aware of their responsibility to provide a healthy productive workplace for their staff. In the move to the new building the employers felt they were moving from a building that was of poor quality to a new building with excellent environmental credentials.

In the short term, benefits were lost due to inadequate commissioning of the building services that made the new building very uncomfortable for a significant number of staff for a considerable time.

In the long term it may be questioned whether the move to the new building was a good idea in principle. The old building could have been refurbished with new windows, solar shading and high efficiency boilers. The building would have had a new lease of life as a light, airy and flexible workspace.

The new building, whilst employing relatively efficient heating and cooling plant, has a considerably larger carbon footprint as well as higher energy costs. The shape of the building means that it is inflexible as regards work styles [6] and cannot be occupied without the extensive mechanical services.

In future energy costs will increase ahead of the rate of inflation and will become more significant in relation to overall office accommodation costs. An environmentally sensitive refurbishment may have not only retained the embedded energy in the 1960s building but also have provided a healthier working environment.

ACKNOWLEDGEMENTS

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Workers' Behavior and Thermal Sensation in Task-conditioned Office

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SUMMARY

Field survey in the M-Office which installed Task/ambient conditioning systems (hereinafter referred to as the “TAC”) was conducted in 2005 and 2006. It was intended to investigate the influence of the worker's behavior and task-conditioning to worker’s comfort sensation. In this survey, immediately thermal environment and worker’s behavior were measured, and questionnaire to occupants who worked as usual were conducted. We found that both activity level of occupant and exposed thermal environment is greatly different one by one. It was suggested that an increase in metabolic rate according to worker’s behavior influence on thermal and comfort sensation certainly. It is necessary to recognize the metabolic rate to estimate the occupant’s declaration in a field survey. In an actual office where occupants stay long terms, the tendency that occupant prefer isothermal airflow was seen. It was thought that adjustment performance with wide amount of task air-conditioning airflow and operationality was desirable to cover occupant’s sense integrally or to connect with occupant’s comfort and satisfaction. Based on the knowledge obtained from this investigation, needs for TAC were discussed.

INTRODUCTION

To stop the environmental destruction that progresses globally, the energy reduction is demanded in an architectural industry. In Japan, it is recommended by the Government to set the air temperature in an office building to be 28 °C in summer. TAC is assumed to be one of the solutions to it. By installing TAC, total energy can be lowered by moderating the amount of energy to ambient zone and giving intensive to the task zone. Moreover, the improvement of the comfort and satisfaction is expected by individual adjustment to create the desirable environment and by direct supply of a fresh air to the breath region. In this survey, it was intended to investigate the influence of worker's behavior and task air-conditioning to worker’s comfort sensation. For that purpose, immediately thermal environment and worker’s behavior, like a sit-down state, walking distance, and metabolic rates, were measured, and questionnaire to occupants who worked as usual for thermal sensation, comfort sensation, and so on were conducted.

METHODS

Outline of the M-Office and TAC

Figure 1 shows perspective of the M-Office, both interior view and exterior view [1]. It has a site area of 4,782 m^2 and a gross area of 19,169 m^2, and four floors above ground and one below ground. Structure of the M office is SRC.
Figure 2 shows the outline and layout of TAC in the M-Office. There is a unit of under floor air-conditioning (100 CMH) per two occupants, and a unit of task air-conditioning (25 CMH) per one occupant. The partition connects under floor, so air for floor-supply is used as for it from task unit. All occupants can handle the task unit for adjusting air volume between 0% and 100%, and air direction between 0° and 45° of the task air-conditioning by an individual.

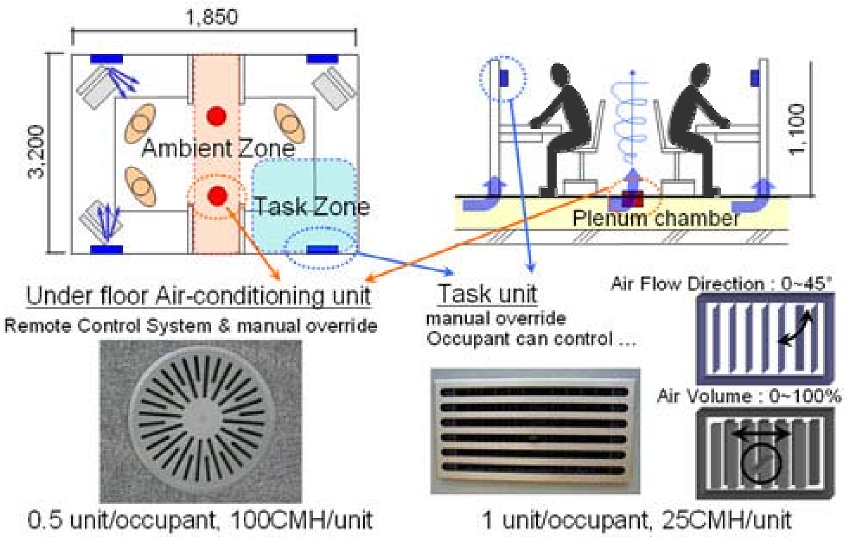

Figure 1. Perspective.

Figure 2. Walking distance and Metabolic rate.

Measuring methods

Field survey was conducted for five days in summer, between period, winter of fiscal year 2005, and summer of 2006. Twenty people who work at the M-Office, namely twelve males and eight females, participated in this survey. It was investigated that the influence of the worker's behavior and task air-conditioning to worker’s comfort sensation. Table 1 shows measuring items in this investigation. In the investigation in 2006, immediately thermal environment, temperature of airflow from task-conditioning, sit-down state, number of steps, and metabolic rate were measured at any time in work hours to understand the worker’s activity state and the thermal environment workers were exposed. Only the investigation of sit-down state was also measured in fiscal year 2005. Questionnaire to occupants who worked as usual for thermal sensation, comfort sensation, and so on were conducted once in the morning and once in the afternoon. At that timing, air temperature, relative humidity, radiant temperature, and airflow velocity at the position of subject were measured. Questionnaire was done only for three days, and it was made conditions in the use of task air-conditioning. Occupants cannot use task air-conditioning in the 1st day (TASK-OFF), must open fully the shutter of task air-conditioning in the 2nd day (TASK-ON), and can use task air-conditioning freely in the 3rd day (TASK-FREE).

Table 1. Measuring items.

Measuring items	Measuring equipment	Subject number	Interval	Timing
Air temperature	C-C Thermocouples	8 people	1 sec	declaring
Relative humidity	RH sensor	8 people	1 sec	declaring
Radiant temperature	Directional Radiometer	8 people	1 sec	declaring
Airflow velocity	Heated Anemometer	8 people	1 sec	declaring
Immediate air temperature	Thermohygrometer	20 people	1 min	any time
Airflow temperature	Thermohygrometer	20 people	1 min	any time
Opening level of task-conditioning shutter	Eye Observation	20 people	1 hour	-
Number of steps	Pedometer	20 people	1 hour	any time
Metabolic rate	Accelerometer	8 people	1 min	any time
State of seating	Thermometer	20 people	1 min	any time
Sensation vote	Questionnaire sheet	20 people	1 min	declaring

RESULTS

Worker's behavior

Working hours in M-Office is 9:00-18:00. The clerical worker was targeted in the measurement of 2006, and worker whose type of job in the measurement of 2005 is coexistence of the business employment, the research employment, and the clerical work. All data that will be shown are the average of five days on the weekday. But, absentee and occupant who sat only by ten minutes or less weren't included in the average.

1) Seating rate, Seat occupancy rate

Whether occupant seated or not was judged by the change of the surface temperature of the seats in one minute [2]. So there is a possibility that too short sitting down is not counted. The definition of seating rate and seat occupancy rate are as follows.

seat occupancy rate = (the number of the occupants seated) ×(total number of the attendance)
seating rate = (hour while the occupants seated) ×(Work hours)

Figure 3 shows temporal change of seat occupancy rate, and Figure 4 shows seating rate that was the average of twenty occupants. The same tendency was seen at three terms of fiscal year 2005. Excluding the break in daytime, seat occupancy rate, that shows the rate of person exposed to the task air-conditioning at given times, in 2005 hover nearly 40%, that in 2006 hover at 60-80%. In other measurement, seat occupancy rate hovers at 30-50% [3]. It becomes 0-20% at lunchtime, but 80% immediately after lunchtime. Seating rate in summer is 37%, that in between period was 35%, that in winter was 39%, that of yearly average is 37%. But that in summer 2006 is high, 63%. It is found that the rate of hours occupants stay in task-zone during work hours baries by occupational category and the content of work.

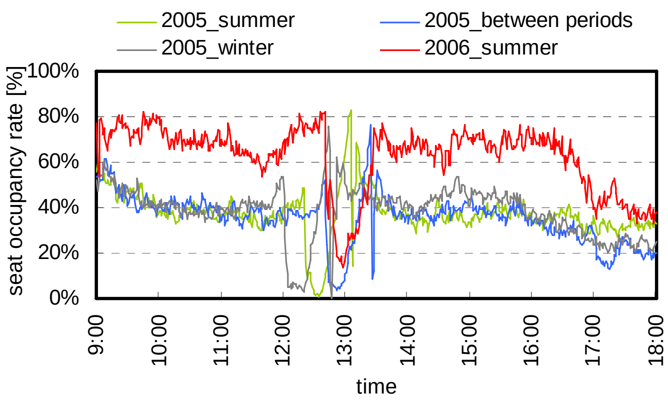

Figure 3. Seat occupancy rate.

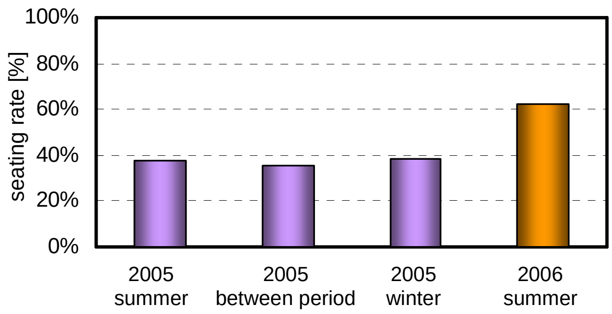

Figure 4. Seating rate.

2) Sitting down, Getting out of the seat

Figure 5 shows the penchant for sit-down. The rate of the number of 1-5 minute sitting down is the highest, nearly 50%. In the mean while, the rate of the number of 1-5 minute getting out of the seat is the highest, nearly 70% in 2005, 86% in 2006. Number of times shows occupant doesn’t leave one's desk easily when occupant sits down once. It is found that workers are not frequently repeating sit-down and leaving-out, so it is thought that metabolic rate while occupants are sitting down is sort of steady.

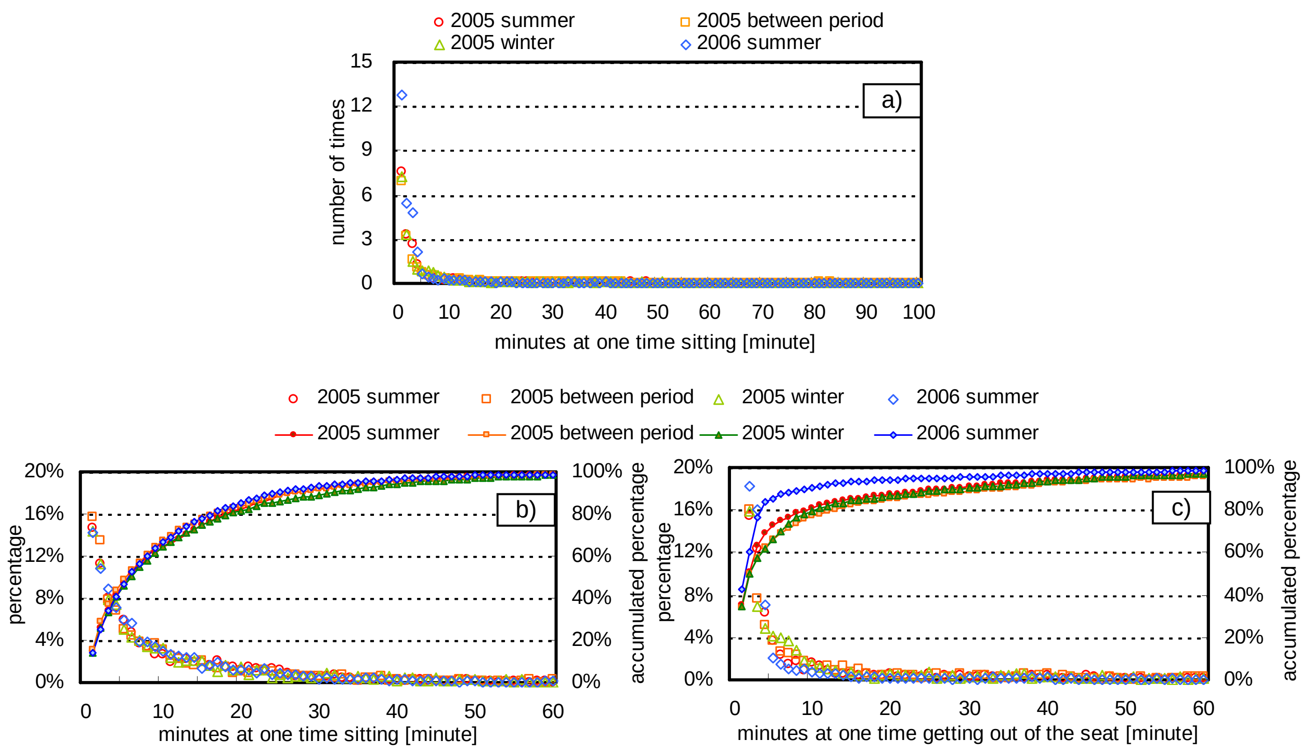

Figure 5. Penchant for sit-down.

3) Worker's behavior and thermal environment

Figure 6 shows worker's behavior and exposed air temperature of M2, one of the male subject, at TASK-OFF condition, that task conditioning system was not allowed to use. It also shows the declared time zone and timing he declared. Metabolic rate in this paper indicates the instant mean value per minute measured by omnidirectional accelerometer, that is a physical activity logging system. When a strong movement is momentarily done, a high value is measured, like 3.0 met or more. Almost activity level of occupant can be recognized. It was found that both activity level of occupant and exposed thermal environment is greatly different one by one.

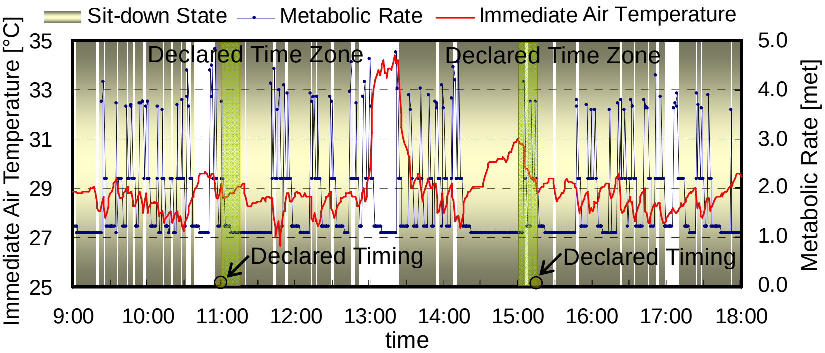

Figure 6. Worker's behavior and Exposed air temperature.

Figure 7 shows number of steps and metabolic rate that is mean value of all occupants. Table 2 shows five days average of metabolic rate while seating and that while leaving from the seat of each occupant. It is found that the number of steps in an hour was about 300 steps or more in work hours. Average of metabolic rate through work hours is 1.4 met, that while seating is 1.3 met, and that while leaving from the seat is 1.7met. Average of metabolic rate, which shows activity level, while seating is to each occupant. Metabolic rate measured in this survey could be one of the useful basic design value.

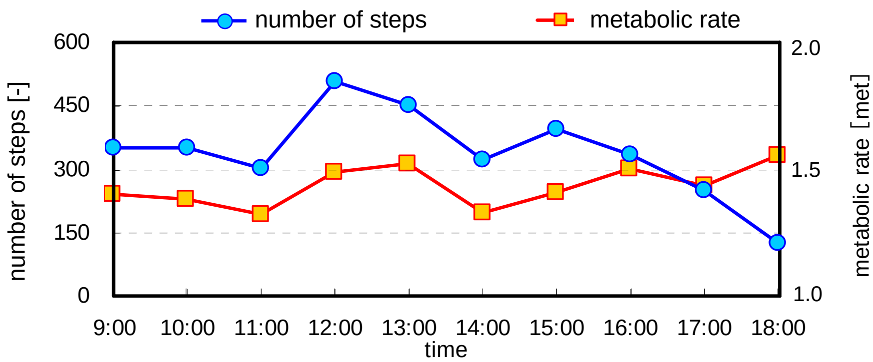

Figure 7. Number of steps and Metabolic rate.

Table 2. Metabolic rate [met] while seating and while leaving from the seat.

	M1	M2	M3	M4	F1	F2	F3	F4	Ave.
Work hours	1.4	1.7	1.3	1.4	1.3	1.4	1.3	1.7	1.4
While seating	1.3	1.5	1.2	1.2	1.2	1.2	1.2	1.4	1.3
While leaving	1.5	2.3	1.5	1.8	1.6	1.5	1.4	2.2	1.7

Occupant's sensation vote and metabolic rate

Figure 8 shows relation between declaration and metabolic rate. Standard new effective temperature (hereinafter referred to as the "SET*") was derived from questionnaire about amount of clothes in the morning and environmental measurement at declaration timing, namely air temperature, relative humidity, radiant temperature, and airflow velocity at the position of subject. But difference was not found by the classification by SET* between conditions. So the declaration was classified from the viewpoint of metabolic rate that was 30 minutes average before occupant declared, metabolic rate is less 1.2 met and is 1.2 met or more, so declarations in this figure are mixed with TASK-OFF, TASK-ON, TASK-FREE condition.
In case metabolic rate is 1.2 met or more, the declaration of thermal sensation votes (TSV) increased that on the hot side, and that of thermal comfort sensation votes (TCSV) increased that on the uncomfortable side. It was suggested that an increase in metabolic rate according to worker's behavior influence on thermal and comfort sensation certainly.

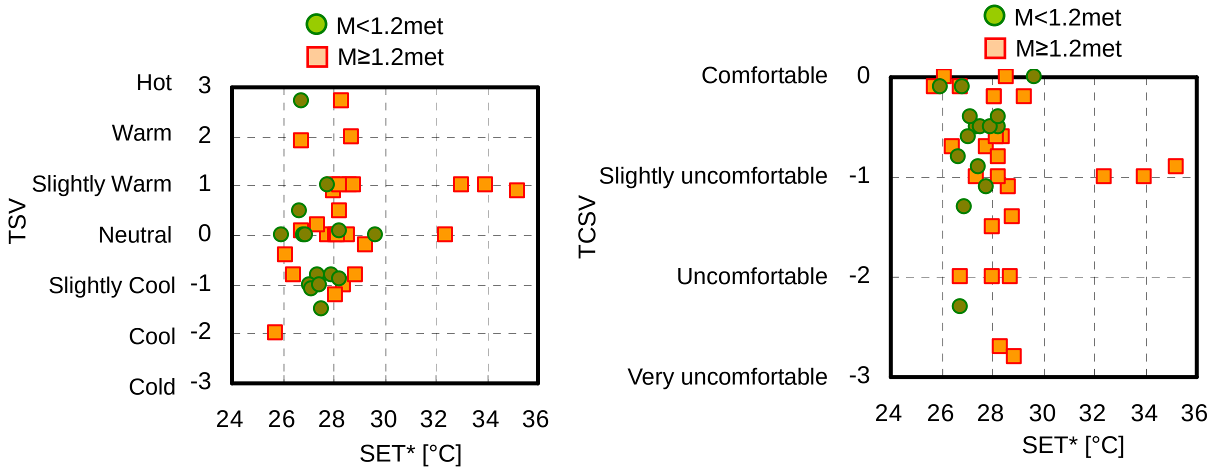

Figure 8. Relation between Metabolic rate and Declaration.

Occupant's sensation vote and temperature of airflow from task air-conditioning

Figure 9 shows relation between Δt and declaration of twenty occupants (Δt-TSV, Δt-TCSV, Δt-ECSV, Δt-EAV). Table 3 shows voting scales. The definition of Δt in this paper is as follows, and Δt as isothermal is between -1 °C to 1 °C, Δt as non-isothermal is more or less than 1°C.

$$\Delta t = \text{(temperature in task-zone)} - \text{(temperature of airflow from task air-conditioning)}$$

Temperature in task-zone and temperature of airflow from task-conditioning was averaged in time zone occupants declared, 11:00-11:30 and 15:00-15:30. The declaration in TASK-FREE condition, that occupants could operate task-conditioning systems freely, targeted only the occupants whose opening rate of the task air-conditioning shutter was 50% or more because of searching the influence that the task air-conditioning gives to sense of occupant. For TASK-ON condition, task-conditioning systems were forced to use without any occupants' control.

When airflow was isothermal, thermal sensation votes (TSV) were almost installed between from -1 to 1. Thermally or environmental comfort sensation votes (TCSV, ECSV) were positioned on the comfort side, and environmentally acceptable votes (EAV) were on the acceptable side. Meanwhile airflow is non-isothermal, both the hot side and the cold side declarations were seen with TSV. The declarations of the uncomfortable side were seen with TCSV, ECSV, and the declarations of the unacceptable side were seen with EAV. In an actual office where occupants stay long terms, the tendency that occupant prefer isothermal airflow was seen.

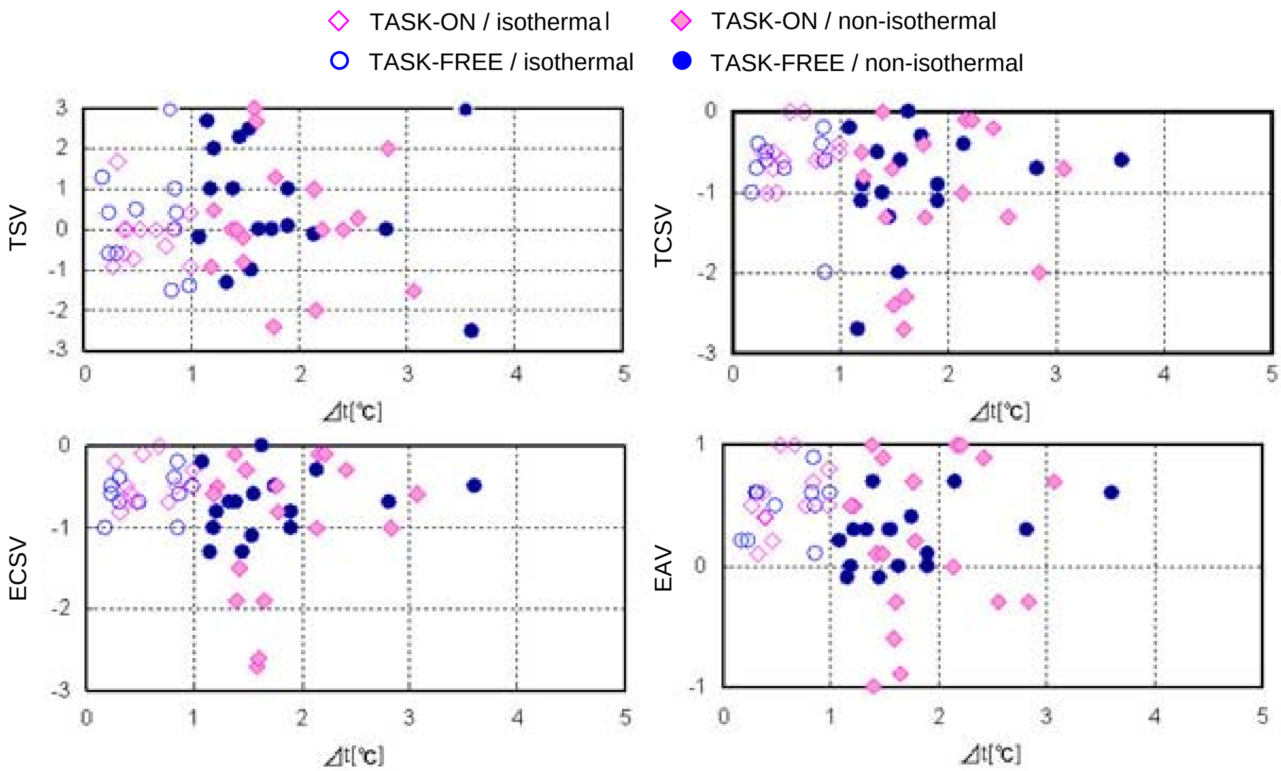

Figure 9. relation between *⊿t* and declaration.

Table 3. Voting scales

Declaration items	3	2	1	0	-1	-2	-3
Thermal sensation	hot	Sl. Warm	Warm	Neutral	Sl. Cool	Cool	Cold
Thermal and Environmental comfort sensation	-	-	-	Comfortable	Sl. Uncomfortable	Uncomfortable	Very Uncomfortable
Environmental acceptable sensation	-	-	Acceptable	-	Unacceptable	-	-

DISCUSSION

We found that the rate of hours occupants stay in task-zone during work hours is few by the type of job, and workers are not frequently repeating sit-down and leaving-out, so it is thought that metabolic rate while occupants are sitting down is sort of steady. But while leaving from the seat, both activity level of occupant and exposed thermal environment is greatly different one by one. It was confirmed that an increase in metabolic rate according to worker's behavior influence on thermal and comfort sensation certainly. It is necessary to recognize the

occupant's situation, especially like metabolic rate and exposed air temperature, to estimate the occupant's declaration in a field survey.
In an actual office where occupants stay long terms, the tendency that occupant prefer isothermal airflow was seen. It was thought that adjustment performance with wide amount of task air-conditioning airflow and operationality was desirable to cover a lot of occupants of sense integrally or to gain occupant's comfort and satisfaction. There is a report which is research of causing the delay in changing the metabolic rate and the skin temperature [4]. It is thought that air-conditioning at space that occupant fit in is very important for gaining occupant's satisfaction. It is possible to decrease dissatisfaction by the task air-conditioning. TAC is an air-conditioning system considering energy balance of task-zone and ambient-zone totally. How to use the building and office changes depending on the occupational category and the content of work, so it is desirable to design HVAC system and to decide operation considering usage. Therefore it is needed to design the HVAC system which can change the operation balance of ambient air-conditioning supply and task air-conditioning supply on some level. Environmental load reduction may be realized without losing occupant's comfort.

ACKNOWLEDGEMENT

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12 June 2007 at 10:00 - 11:30

A04
Criteria for thermal environment

Revised Swedish Guidelines for the Specification of Indoor Climate Requirements released by SWEDVAC

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SUMMARY

The Swedish indoor climate guidelines issued by Swedvac has been thoroughly revised. The new document is adapted to harmonize with related international, European and Swedish standards. The document comprises thermal climate, indoor air quality, sound and light.

The new R1-document is mainly intended to provide guidance for the specification of indoor climate requirements. It is recommended for use as a reference document in the very early phase of planning a new building or the reconstruction of an existing building. The document provides guidance regarding planning of measurements for verification of indoor climate quality, which is an issue to consider already when the indoor climate requirements are being specified.

INTRODUCTION

In the beginning of the nineties the first version of the Swedish indoor climate guideline document "R1" was published by Swedvac. In May 2006 a thoroughly revised version was released [1]. The new document has been updated with respect to knowledge that has evolved over the years and it is adapted to harmonize with related international, European and Swedish standards. The document comprises thermal climate, indoor air quality, sound and light; all indoor climate factors which are influenced by the installation systems in a building, and consequently should influence the design of the building services.

The starting point is that requirements and recommendations from Swedish authorities shall define the basic indoor climate requirements. As in the original R1-guideline, there is an opportunity to select a higher quality level. However, the system for classification of the indoor climate has been radically rearranged in the new version. Thermal climate and indoor air quality are divided into two quality classes each, and sound comprises three quality classes. Only one set of guideline values are specified for light.

The main objective of the new R1-document is to provide guidance for the specification of indoor climate requirements. As illustrated in Figure 1 it is intended as a reference document in the earliest phase of planning a building project. The document does neither provide guidelines for the details in the HVAC-design process, nor for the contracting or administration of buildings in operation. However, it is mandatory that the indoor climate requirements specified using the guidelines should be verified by measurements. The document provides guidance regarding the planning of such measurements, which is an issue to consider already when the indoor climate requirements are being specified.

The objective of the present paper is to provide a summary of the revised Swedish indoor climate guidelines, “R1”.

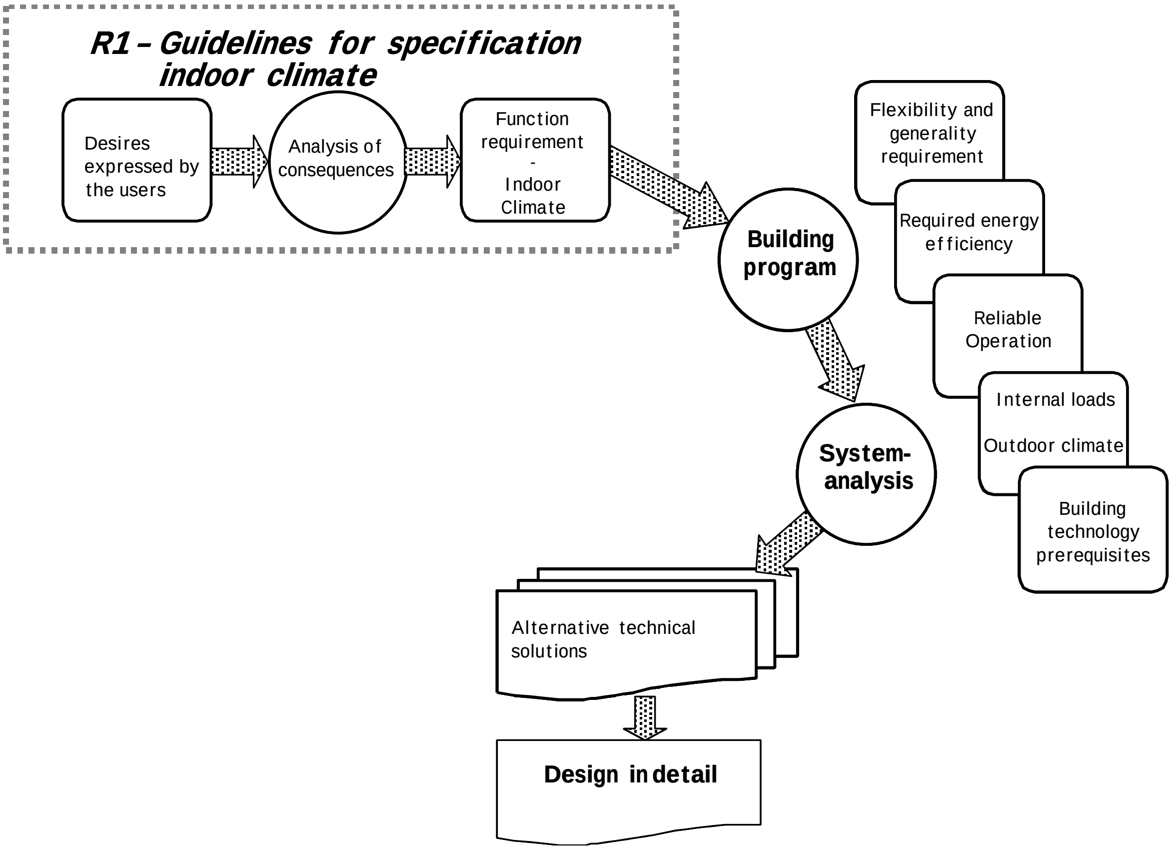

Figure 1. Illustration of the transformation of occupant’s desires to an unambiguous indoor climate requirement specification, and further towards selection and design of feasible technical solutions.

BASIS FOR THE GUIDELINES

Thermal climate

The guidelines regarding thermal climate are based on the ppd/pmv approach specified by the standard SS EN ISO 7730:2006 [2]. The ppd-index, defined in this standard, is used as a parameter to find a suitable target interval for the operative temperature. The starting point is to aim at a climate that is *normally perceived as comfortable*. According to the R1-guideline this corresponds to a ppd-index not exceeding 10%. However, the climate requirement is not explicitly expressed using the value of the ppd-index, since such an approach is deemed dubious and not feasible in practice. The requirement must be specified in a manner that enables a rather straightforward method of verifying compliance by measurements. The operative temperature is judged as a feasible measure, the ppd-index is not.

Some standards and guidelines suggest classification of the thermal climate based on the ppd-index. For example, SS EN ISO 7730 suggests 6%, 10% and 15% as the maximum allowable ppd-index values in three alternative quality classes. A ppd-index of 6% leads to a very narrow temperature range close to the “optimal operative temperature”. For example in an office during summer conditions the temperature range would be about 24.5±1.0°C. The precision of the pmv/ppd approach suffers from uncertainties of the assumed input data, especially regarding the clothing and activity of people, as well as the air velocity. For example, a small change of the met-value from 1.2 to 1.3 may correspond to a change of the preferred operative temperature of more than 0.5°C. Changing the assumed clo value from 0.5 to 0.6 (e.g. by assuming a thin long-sleeved shirt instead of a t-shirt being worn) has a similar effect on the predicted temperature preference. Thus, it can be questioned whether the narrow temperature interval ±1.0°C in practice reflects a higher quality level than the interval 24.5±1.5°C, resulting from calculations using a ppd-index value of 10%. According to the R1-guideline for offices and other premises the higher quality class is obtained by individual temperature control, not by trying to keep the temperature as constant as practically possible.

The R1-guideline does not comprise a third thermal quality class, as e.g. SS EN ISO 7730 does, allowing ppd-index values up to 15%. Such high ppd-values exceeds the recommendation of ppd<10% made by the Swedish Work Environment Authority, and cannot be applied: -A basic principle of the R1-guideline is that not only mandatory requirements, but also the recommendations made by the Swedish authorities shall be met. Thus, the R1-guideline comprises two thermal quality classes, denoted TQ1 and TQ2.

In residential buildings the occupants shall have the possibility to influence the room temperature in both climate classes. The higher quality class is in this case instead obtained by reducing high temperatures during hot summer days. This shall not be interpreted as a recommendation to install mechanical cooling in residential buildings, but rather implies extensive thermal insulation and sun shading.

Values to minimize local thermal discomfort follows the SS EN ISO 7730 closely, but with a few modifications justified by recommendations from Swedish authorities regarding, e.g. floor temperatures. In quality class TQ1 the requirements for reduction of the risk of local discomfort are more sharp than they are in quality class TQ2.

Indoor air quality

According to the R1-guideline, not only the mandatory requirements, but also the recommendations given by the Swedish authorities shall be met (The National Board of Housing, Building and Planning, the National Board of Health and Welfare, the Swedish Work Environment Authority). It is a matter of prescribing minimum outdoor airflow rates, and maximum acceptable concentrations of various pollutants. The R1-guideline does not explicitly prescribe requirements regarding emissions from building products. However, it is mandatory that this issue be addressed by quality assurance in line with some established method. In this respect the R1-guideline refers to, and implies the use of either the Finnish system, administered by The Building Information Foundation [www.rts.fi], or the recommendations made by SP The Swedish Technical Research Institute, in connection to the quality system for “P-marking” of indoor environments [3].

The R1-guideline provides a set of recommendations for target values regarding maximum acceptable pollutant concentrations. The set of target values is a synthesis of previously established guidelines [4, 5, 6], and the implementation in Sweden of the European directive for outdoor air quality (published by the Swedish Environmental Protection Agency) [7]. The recommended target values are the same for both indoor air quality classes, AQ1 and AQ2. The distinction between the quality classes is based on the risk for annoyance due to odors, primarily originating from bio-effluents. In the basic quality class, AQ2, odors must be accepted, at least shortly after entering the building. In the higher of the two classes, AQ1, this risk shall be substantially reduced. The concentration of carbon dioxide (above that outdoors) is used as the quality measure.

Sound and noise
The requirements regarding acoustic quality are based on two Swedish national standards [8, 9]. One of these standards addresses sound in residential buildings and the other addresses premises like offices, schools etc. According to the R1-guideline it is mandatory to consider noise from installations, e.g. the ventilation system. Target values are given both for A-weighted and C-weighted noise levels. However, it is recommended to consider all acoustic quailties comprised in the two standards, e.g. outdoor noise, reverberation, sound insulation etc. The R1-guideline comprises the three sound quality classes NQ1, NQ2 and NQ3.

Light
The R1-guideline prescribes that the lighting system shall be planned and verified according to the standard SS EN 12464-1 [10]. This standard does not comprise a distinction between various light quality classes, and neither does the R1-guideline. Target values are presented for the average illumination, the evenness of the illumination, the unified glare rating index (UGR-index) and the average color rendering index (Ra-index). Furthermore, recommendations regarding the reflectance of various surfaces and the daylight factor are made.

Other factors
In addition to thermal climate, air quality, sound and light, a requirement specification prepared according to the R1-guideline also addresses the risk for legionella growth in water systems, radon in tap water and electromagnetic fields.

TARGET VALUES

This section provides a summary of target values presented in the R1-guideline.

Table 1 concerns thermal climate requirements in offices and similar premises. The guideline prescribes that the ppd/pmv concept shall be used in order to determine a suitable target interval for the operative temperature, considering the clo and met values estimated for the activity the building is planned to house. As can be seen in the table the guideline also comprise recommendations on how to consider deviations from this interval, e.g. during the hottest summer days. According to the R1-guideline the designing engineer shall clarify to the commissioner the consequences of suggested technical solutions, e.g. by an estimation of the percentage of the occupied hours the indoor temperature will rise above the maximum value of the target interval.

Table 2 indicates the corresponding information for residential buildings. Table 3 summarizes target values to minimize the risk of local thermal discomfort. The criterion for reducing the risk of draught is in each quality class given at two alternative temperatures.

Table 1. Example of target values regarding the operative temperature in an office room. The same values are applicable in both quality classes, TQ1 and TQ2. In TQ1 (but not in TQ2) there is an additional requirement of individual temperature control in single rooms and/or smaller zones of large rooms.

Parameter	Target values	Deviations	Note
Operative temperature			
Winter	20.0-24.0ºC	It shall be possible to maintain the indoor temperature below 23ºC	1.0 clo / 1.2 met
Summer	23.0-26.0ºC	At outdoor conditions above 27ºC the indoor temperature should be about 3ºC below the outdoor temperature	0.5 clo / 1.2 met

Table 2. Target values for operative temperature in residential buildings. Note that the difference between quality class TQ1 and TQ2 concerns the prevalence of high temperatures during hot summer days.

Parameter	Target values	Deviations	Note
Operative temperature			
Winter	20.0-24.0ºC	It shall be possible to maintain the indoor temperature below 23ºC	1.0 clo / 1.2 met
Summer	23.0-26.0ºC	TQ1: Indoor temperatures above 26ºC accepted during short periods (hot summer days) TQ2: Indoor temperatures above 28ºC accepted during short periods (hot summer days)	0.5 clo / 1.2 met

Table 3. Target values for minimizing local thermal discomfort.

Parameter	TQ1	TQ2
Floor temperature	22-26ºC	20-26ºC 16-27ºC in rooms not intended for children
Vertical temperature difference	<2ºC	<3ºC
Radiant temperature asymmetry	Warm ceiling <5ºC Cold wall <10ºC	Warm ceiling <5ºC Cold wall <10ºC
Air velocity (draught risk)	<0.10m/s at 20ºC <0.15m/s at 26ºC (Draught rating < 10%)	<0.15m/s at 20ºC <0.25m/s at 26ºC (Draught rating < 20%)

As mentioned above the distinction between the two air quality classes AQ1 and AQ2 is based on the risk of annoyance due to odors caused by bio-effluents. Thus, in quality class AQ1 the concentration of carbon dioxide may not rise above 800 ppm (about 400 ppm above the outdoor concentration) more than temporarily, while in quality class AQ2 the limit is set to 1000 ppm (about 600 ppm above the outdoor concentration). In this respect class AQ2 corresponds to advice given by Swedish authorities.

In both air quality classes AQ1 and AQ2 the aim shall be to maintain the pollutant concentrations below the values given in Table 4.

Table 4. Target values regarding maximum acceptable pollutant concentrations. All values are applicable to both air quality classes, AQ1 and AQ2.

Substance	Designation	Maximum concentration	Reference
Radon	Rn	100 Bq/m^3	FiSIAQ [4]
Carbon monoxide	CO	2 mg/m^3	SP, FiSIAQ [3, 4]
Nitrogen dioxide	NO_2	40 µg/m^3	WHO, ISIAQ-CIB, Swedish EPA [5, 6, 7]
Ozone	O_3	50 µg/m^3	FiSIAQ [4]
Formaldehyde	HCHO	50 µg/m^3	SP, FiSIAQ [3, 4]
Particulate matter <10µm	PM10	40 µg/m^3	FiSIAQ, Swedish EPA [4, 7]
Particulate matter <2.5µm	PM2.5	15 µg/m^3	FiSIAQ, ISIAQ-CIB, Swedish EPA [4, 6, 7]

Furthermore, the R1-guideline prescribes that the outdoor airflow rate shall be selected according to recommendations given by Swedish authorities. In office buildings and other similar work premises this means at least 7 l/s per person plus 0.35 l/s per m^2 floor area. In residential buildings the corresponding values are 0.35 l/s per m^2 floor area or 0.5 air changes per hour, but at least 4 l/s per person. However, the requirement to limit the carbon dioxide concentration is valid also for residential buildings, which typically leads to a higher airflow rate than 4 l/s per person.

The entire occupied zone shall be efficiently ventilated without occurrence of stagnant zones. This shall be verified, e.g. by measurement of the local ventilation index (>90%) or the air change efficiency (>40%). A recommendation is to carry out this verification according to applicable Nordtest medhods [11, 12]

If air filters are needed these should preferably at least be of class F7, and they should be quality assured, e.g by the Swedish “P-mark” system for air filters [13], administered by SP Technical Research Institute of Sweden.

Table 5 gives examples of target values for thee selected indoor environment types regarding noise from installations. As mentioned above the R1-guideline specifies target values regar-ding noise from installations, but recommends that also other acoustic qualities be considered, with reference to two Swedish standards [8, 9].

Table 5. Examples of recommended target values regarding noise from installations.

	Office room			Classroom			Residential room*		
Class	NQ1	NQ2	NQ3	NQ1	NQ2	NQ3	NQ1	NQ2	NQ3
LpA, dB(A)	35	35	35	26	30	30	22	26	30
LpC, dB(C)	55	-	-	45	45	50	42	46	50**
LpAFmax, dB(A)	-	-	-	-	-	-	27	31	35
Note.	-	-	-	-	-	-	No tones	No tones	No tones

* Less stringent values specified for kitchens

** Applies to bedrooms

In addition to the illumination parameters presented in Table 6 the R1-guideline specifies that the illumination in no single point may be lower than 70% of the average illumination over a workplace-surface. The corresponding value for surrounding surfaces and corridors etc is 50%. The daylight factor is recommended to be at least 1%. Furthermore, light sources shall be located with respect to the location of the occupants in order to avoid glare. It is recommended to shade sources of strong luminance, e.g. windows. Guidance regarding the reflectance of various surfaces are given: ceiling: 0,6-0,9; walls 0,3-0,8; floors 0,1-0,5; writing table 0,2-0,6.

Table 6. Examples of recommended target values regarding light.

	Office room	Classroom	Waiting room
Average illumination, lux	500	300*	200
UGR-index	<19	<19	<22
Ra-index	>80	>80	>80

* Classroom for adults, 500 lux

DISCUSSION

The Swedish R1-guideline for specification of indoor climate requirement has been revised. The guideline is adapted to Swedish conditions and regulations, although it has been of priority also to harmonize the document with respect to European standards and guidelines.

The main purpose of the guideline is that it shall be used as a reference document in the earliest planning phase of the building process. The guideline provides a basis for the communication between primarily the commissioner and the HVAC-engineer, but it can also serve as a common basis for the united efforts by all participants in the building process, e.g. also the architect, the tenant and the contractor.

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A field study of the performance of the Dutch Adaptive Temperature Limits guideline

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SUMMARY

Field measurements and recording of occupant thermal responses and behavioral actions in 4 buildings clearly demonstrates adaptation: at higher outdoor temperatures, higher indoor operative temperatures are judged as comfortable by the occupants. The comfort votes however didn't relate well with the climate types Alpha and Beta as determined according to the Dutch ATG guideline. But when the climate types were restructured on the basis of the occupants' perceptions the comfort temperatures match very well with the climate types. Furthermore, compared to the former GTO-guideline a higher cooling capacity is needed to keep the indoor temperature strictly below the ATG-limit, but when the temperatures are allowed to exceed this limit for some 5% of the time, both GTO and ATG lead to almost similar capacities.

INTRODUCTION

In 2004 the Dutch ISSO-74 Adaptive Temperature Limits guideline (ATG) was introduced for the design and assessment of thermal comfort in buildings [1,2,3]. To gain experience with the ATG-method a field study in 40 office rooms in 4 buildings during 5 weeks in the summer of 2005 was conducted. The main focus was to assess the performance of the ATG-method in relation to the perceptions of the occupants in real buildings and to the former GTO-method [4] (Weighted Temperature Exceeding Hours).

The ATG-method was developed as an alternative for the former GTO-guideline, which is based on the analytical PMV model. According to the Government Buildings Agency guidelines an adequate thermal indoor climate had to comply with $-0.5<PMV<0.5$ and this limits were allowed to be exceeded for a maximum of 100 hours a year. The time during which the calculated PMV exceeds the limit of +0.5 is then weighted proportionally to the PPD (table 1). It was found that a building with medium thermal mass had a mean PMV of approximately 0.7 at 100 hours exceeding the PMV limit of 0.5 and thus a PPD of 15%. So the weighting factor is 1.5 and the limit of the GTO was determined at 150 (weighted) hours.

Table 1: PMV, PPD and weighting factor

PMV	PPD	weighting factor
0	5	0
0,5	10	1.0
0,7	15	1.5
1,0	26	2.6

Experiences with the GTO-method for a 15 year period have lead to the need for a method that could communicate more clearly than the somewhat abstract "150 weighted exceeding hours". Furthermore consultants started to use the GTO method in a sort of "adaptive" way. Buildings with sealed windows were designed on the basis of 0 (zero) GTO, while in buildings with operable windows more than 150 weighted temperature hours were allowed, although no research evidence was available to support this approach. This changed when [5] was published; the data of this study formed the basis of the ATG (Adaptive Temperature Limits) guidelines. Figure 1 shows the ATG limits for 90, 80 and 65% acceptability of the indoor thermal conditions.

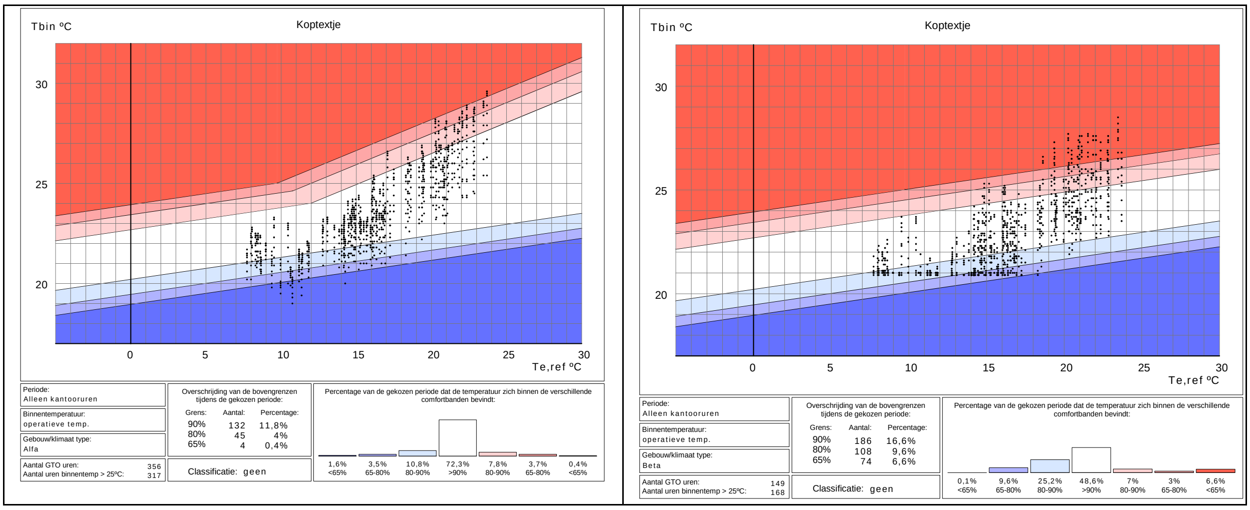

Figure 1: Alpha (left) and Beta (right) type building/climate. Limits of operative indoor temperatures for 90%, 80% and 65% acceptability, as a function of the weighted outdoor temperature Te,ref.

It is assumed that the indoor operative temperature is never allowed to exceed the limits. Whether this is justifiable also is a subject of the current study.

In this study naturally ventilated buildings were defined as “buildings with operable windows and ceiling fans within small single- or dual occupant offices that afford high degrees of adaptive opportunity” and were called Alpha buildings in the ATG guideline. Air conditioned buildings, defined as “sealed centrally air-conditioned buildings with open plan floor layouts” that provide minimal adaptive opportunities and where the occupants are presumed to have no option to open/close windows” were called Beta buildings or climates. The definitions appeared to be unpractical in the Netherlands, because most buildings have operable windows and a variety of room or group sizes (single cellular offices, group offices, landscaped offices) and most buildings combine operable windows with various types of HVAC systems (e.g. all air systems, passive façade ventilation systems, induction units, peak cooling).

As an attempt to overcome the problem to assign the correct comfort limits to a certain building type a flow chart (figure 2) had been developed to distinguish between Alpha and Beta on the basis of the theoretical aspects “sealed façade/operable windows”, clothing adaptation possibilities”, “mechanical cooling/no mechanical cooling” and “temperature control”.

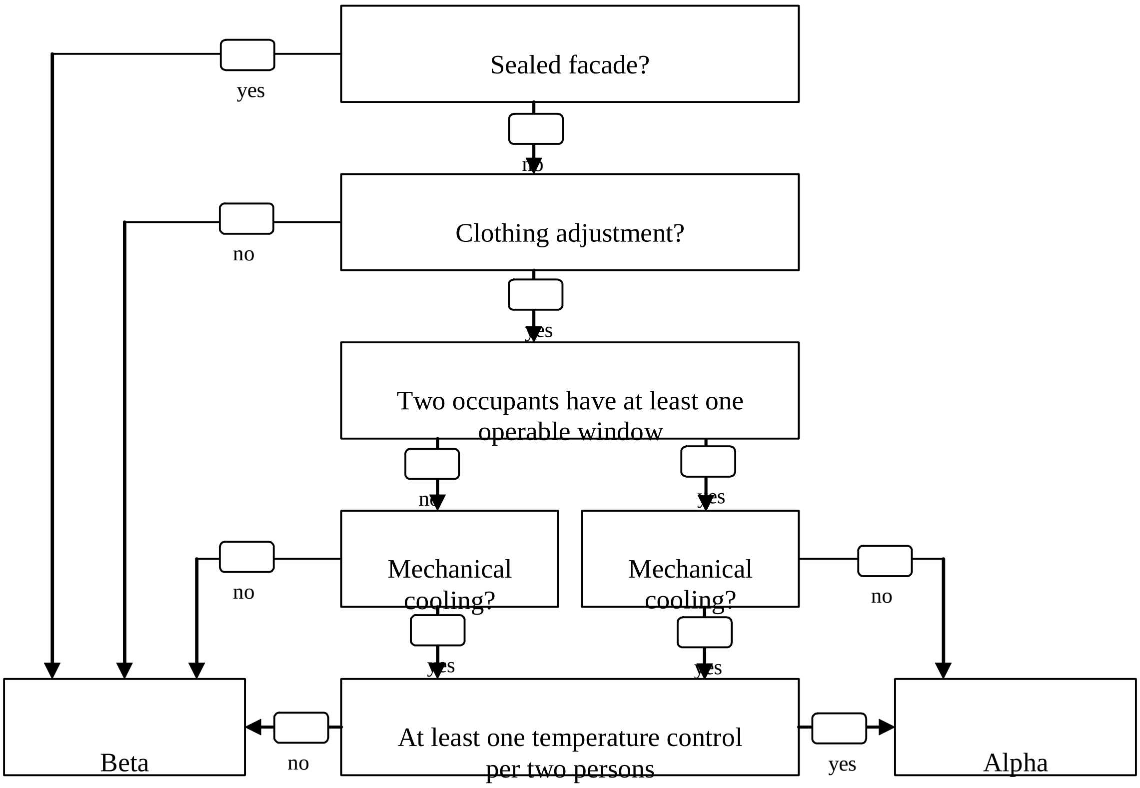

Figure 2: Diagram for determining type of building/climate Alpha or Beta

METHODS

Four office buildings in the Netherlands were monitored in the summer of 2005. See table 2 for details.

Table 2: Details of the four studied buildings

Building	**A**	**B**	**C**	**D**
GFA (m^2)	48400	11800	12550	9000
Number of occupants	1200	280	325	250
Ventilation system	MV No cooling	MV No cooling, cooling	Air-conditioning (induction)	MV Peak cooling
Operable windows	Yes	Yes	Yes	Yes
Room size (p)	1-4	1-10	1-2	1-4
SAM (kg/m^2)	60	62	62	60
IHL (W/m^2)	20	20	20	20
SGF	0,19	0,21	0,22	0,20
Climate type	Alpha	Alpha/Beta	Alpha/Beta	Alpha

GFA=Gross Floor Area, MV= Mechanical Ventilation, SAM= Specific Active Mass, IHL= Internal Heat Load from occupants, office equipment and lighting, SGF= Solar Gain Factor, Climate type according ISSO 74, p 16 and figure 2.

In each building physical measurements were taken in 10 office rooms during a period of at least 5 weeks. Air temperature, globe temperature, relative humidity, air speed and outdoor temperature were measured with a 15 minute interval and operative temperatures and running mean outdoor temperatures were calculated. Twice a day thermal responses of the occupants were monitored by means of a web-based questionnaire, including thermal sensation expressed on the seven-point ASHRAE-scale, thermal preference, acceptability and satisfaction. Also the clo-value, metabolism and adaptive opportunities, including control of operable windows, temperature controls, operation of sun-shading were recorded.

RESULTS

The use of adaptive opportunities show mixed results. The sun shading is open most of the time, despite the fact that the observations were made in mid summer with sunny weather most of the time (Table 3).

Tabel 3: Position of sun shading according to the respondents.

Building	**A**	**B**	**C**	**D**
Open	78%	91%	65%	86%
Half closed	5%	3%	19%	11%
Closed	17%	6%	16%	3%

On the other hand table 4 shows that the operable windows are used on a very regular basis, but there are large differences between the buildings. In building A the occupants were instructed not to open the windows when they felt warm and buildings C and D are mechanically cooled. Further results show that the doors of the office rooms were open most of the time, temperature adjustment knobs of the cooled buildings were seldom used and lighting in the buildings was switched on most of the time in three of the buildings. In building A lighting was off 24% of the time due to an automated, light dependent lighting system.

Table 4: Position of operable windows according to the respondents and measured minimum operative temperature at opening the windows.

Gebouw	**A**	**B**	**C**	**D**
Closed	56%	14%	74%	61%
Just open	10%	43%	10%	38%
Open	34%	43%	16%	2%
$T_{op,\,min}$ (°C)	20,9	21,5	20,6	18,9

The average, self-administered, clo-value was 0,67 ± 0,10 clo and the metabolism was 1,24 ± 0,14 met (72 ± 8 W/m^2).

The thermal comfort votes, represented by the linear regression line through the operative temperatures at which the respondents voted -1, 0 or 1 on the ASHRAE-scale ánd the respondents also voted “no change” on the preference scale, were plotted against the running mean outdoor temperature and compared to the comfort temperatures according to the ATG guideline. When the Alpha and Beta climates were defined according to the scheme of figure 2, no close agreement was found (figure 3).

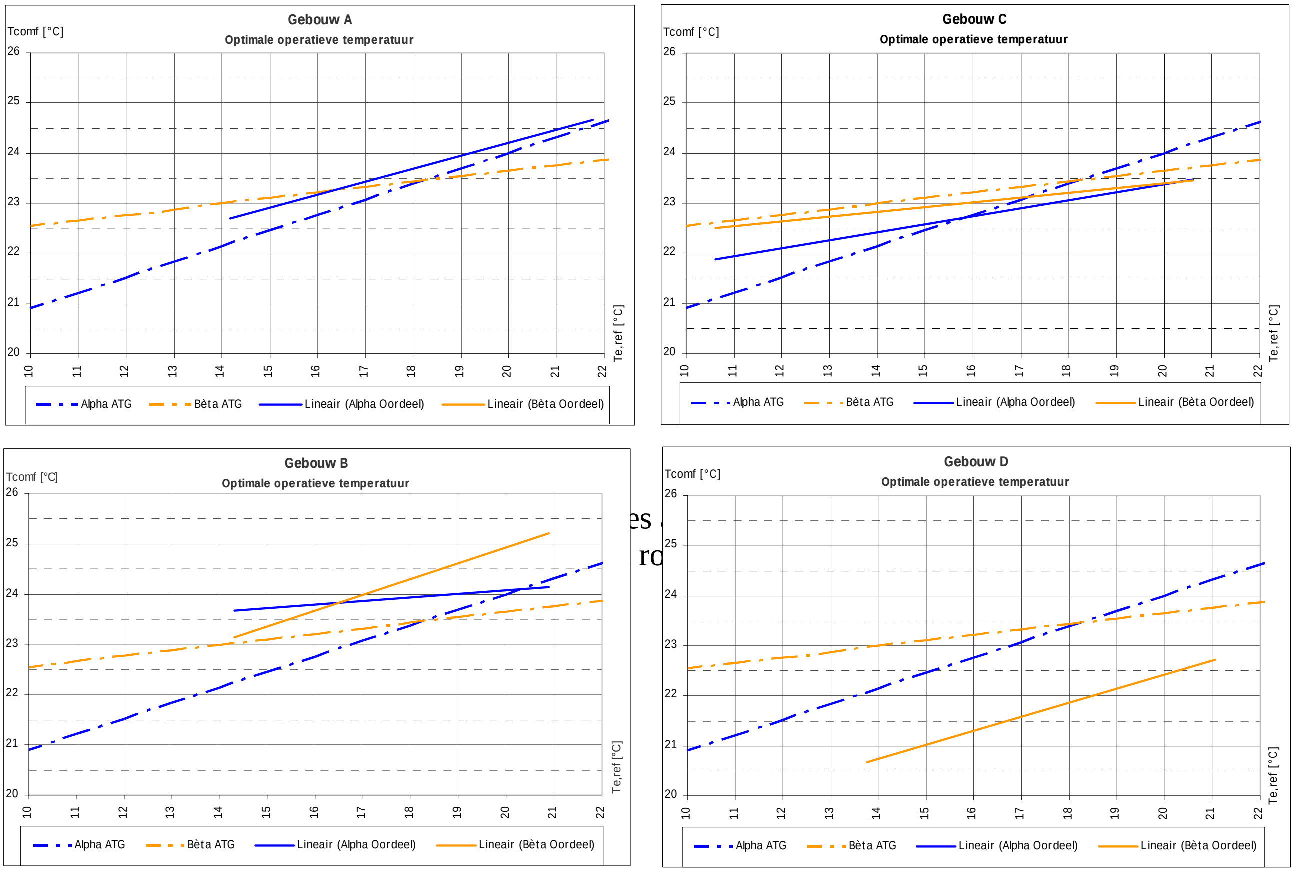

Figure 3: The perception of comfort temperature plotted against the running mean outdoor temperature for Alpha climate (solid blue line) and Beta climate (solid yellow line). The dotted blue and orange lines are the comfort temperature according the ATG-guideline for Alpha and Beta climates respectively, *according to the scheme of figure 2*. The comfort temperatures are represented by the linear regression line through the operative temperatures where the respondents voted -1, 0 or 1 on the ASHRAE-scale ánd were they also voted “no change” on the preference scale.

In figure 4 the same perceptions of thermal comfort are plotted against the running mean outdoor temperature for Alpha and Beta climate types, but now the distinction between Alpha and Beta is defined the perceptions, the slopes of the regression lines through the comfort votes in each room, rather than according to the scheme of figure 1. Again the dotted blue and orange lines are the comfort temperatures for Alpha and Bèta climates.

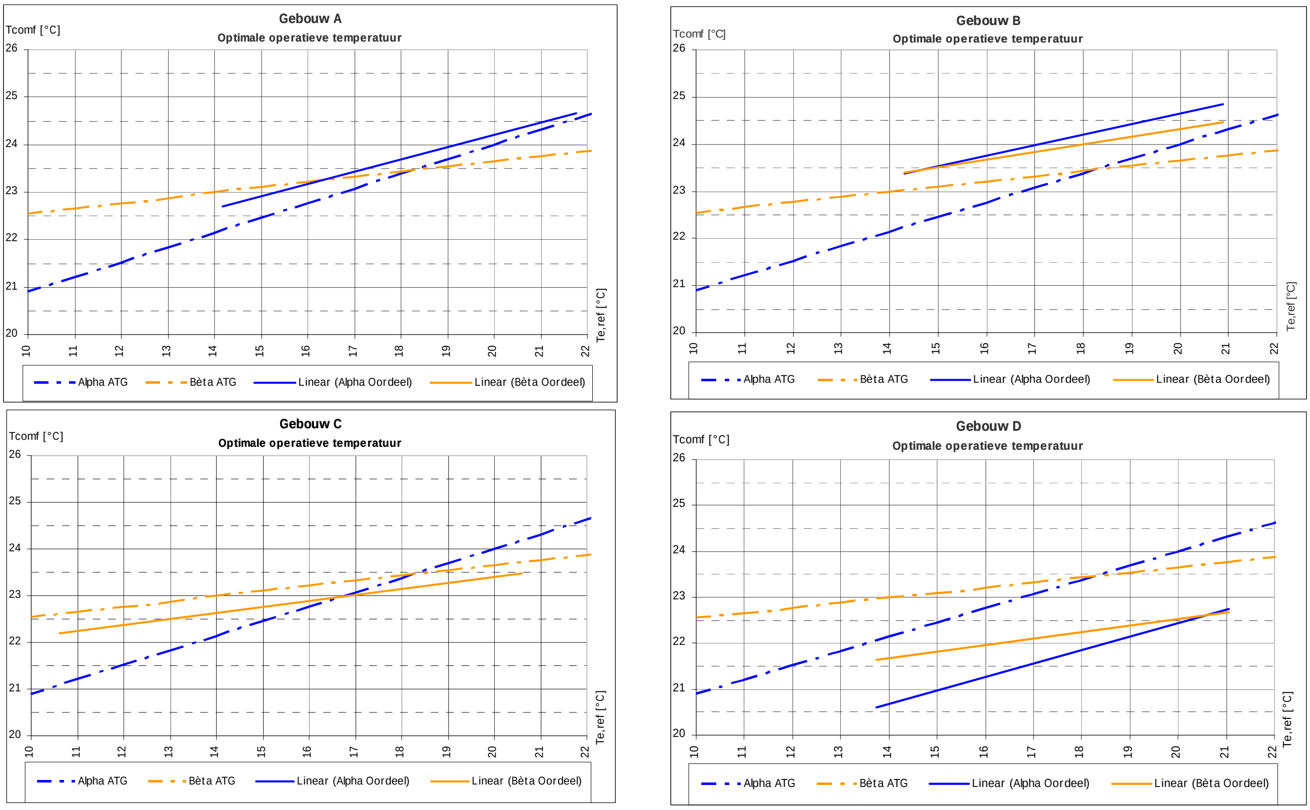

Figure 4: The perception of comfort temperature plotted against the running mean outdoor temperature for Alpha climate (solid blue line) and Beta climate (solid yellow line). The dotted blue and orange lines are the comfort temperature according the ATG-guideline for Alpha and Beta climates respectively, *according to occupants' perceptions*. The comfort temperatures are represented by the linear regression line through the operative temperatures where the respondents voted -1, 0 or 1 on the ASHRAE-scale ánd were they also voted "no change" on the preference scale.

Next, the measurements and occupants' perceptions and actions were used to calibrate temperature simulations to be able to compare the behavior of the GTO en ATG guidelines. The results of new simulation calculations show that to meet the ATG limits strictly, a substantial rise of the cooling capacity is needed, compared to the GTO-criterion, in particular for the Beta climate type (59% more cooling capacity), but also for the Alpha climate type (+14%). One of the goals of the introduction of the ATG-guideline was to allow more room for the design of passive buildings by less stricter temperature limits and these results counteracts with this intension.

DISCUSSION

This field study clearly shows adaptation: at higher outdoor temperatures, higher indoor operative temperatures are judged by the occupants as comfortable. This relation is demonstrated even in the relative small sample of only 10 office rooms in each single office building. But in buildings B, C (Alpha) and D (Alpha) the comfort votes didn't relate well with the climate types of the ATG guideline, when determined on the basis of the scheme in the Dutch guideline (figure 2). This flow chart was developed in an attempt to distinguish between Alpha and Beta climates on theoretical considerations, but not on research data. However when the climate types were restructured on the basis of the occupants' perceptions of thermal comfort (the slopes of the regression lines through the comfort votes in each room) the comfort temperatures according the ATG-guideline for Alpha and Beta climates and those according to occupants' perceptions show a good agreement. This indicates that the distinction between Alpha and Beta climates should be made on the basis of the dependence of the operative indoor temperature on the running mean outdoor temperature or to what extent the indoor climate is "free-running". The dependence of the indoor climate on the outdoor climate is the result of the combined effects of building characteristics (e.g. thermal mass, thermal isolation, external heat load), type of HVAC system (ventilation, mechanical cooling) and usage (internal heat load, operable windows, use of sun shading). Specifically the cooling capacity of the HVAC-system seems to influence the voting behavior and is subject for further research.

Furthermore it should be noticed that the adaptive opportunity of temperature control doesn't seem to have an effect on the dependence of the operative indoor temperature from the running mean outdoor temperature and should not be taken into account in the scheme in figure 2. All this is evidence that the scheme to determine between Alpha and Beta climates should be revised and that topics like group size, clothing adjustment and temperature control should be reconsidered and information concerning the properties of HVAC systems (cooling capacity) should possibly be added.

The higher cooling capacity that is needed to comply to the ATG-limits compared to the GTO-guideline is caused by allowing 150 weighted hours to exceed the PMV=0,5 limit in the GTO-method, whereas the ATG-guideline doesn't allow the limits to be exceeded at all. These arbitrary choices treat the ATG-guideline stricter than the GTO-guideline. However, when the temperature is allowed to exceed the ATG-limit for 2,5% to 7,5% of the time both GTO and ATG show similar results. Further research is needed to asses the acceptability to allow the temperature limits in the ATG-guideline to be exceed to occasionally.

CONCLUSIONS

This study clearly shows the effects of adaptation: at higher outdoor temperatures higher indoor temperatures are judged to be comfortable by the occupants. Furthermore the degree of adaptation according to the occupant's perceptions complies very well with the Dutch ATG guideline (ISSO 74), provided the climate classification is made on the basis of the occupant perceptions, rather than on the basis of the flowchart in ISSO 74. This clearly should be subject of further research. Adaptive opportunities are being used by the occupants, but in some cases differently than foreseen during the design. Sun shading is used sparsely, but opening windows is used regularly, even at relative low indoor temperatures. The ATG-method is judged stricter compared to the GTO-method because 150 weighted exceeding hours are allowed whereas in the ATG-method no exceeding of the limits is allowed. It seems reasonable to allow temperatures to exceed the limits a small percentage of time.

ACKNOWLEDGEMENT

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Thermal Comfort, Perceived Air Quality and Intensity of SBS Symptoms During Exposure to Moderate Operative Temperature Ramps

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SUMMARY

The objective of the presented research work was to study the effects of moderate operative temperature drifts on human thermal comfort, perceived air quality, and intensity of SBS symptoms. Experimental subjects (52, 50% female) were seated in a climatic chamber and exposed to operative temperature ramps with different slope, direction and duration during two related experiments (±0.6 K/h, ±1.2 K/h, +2.4 K/h, +4.8 K/h). The studied temperature ranges were 22-26.8ºC (light clothing - 0.5 clo) and 17.8-25ºC (heavier clothing - 0.7 clo). Exposure to steady temperatures (24.4, 21.4ºC) corresponding with a neutral thermal sensation was included in the experimental design. Results of the experiment showed that all tested operative temperature ramps were perceived by sedentary subjects when the exposure time exceeded four hours. No significant effect on SBS symptoms related to local irritation of mucous membranes was found, while the intensity of headache, well feeling and concentration ability was significantly higher at the end of the exposure to the temperature ramps. A linear relation between perceived air quality and temperature (enthalpy) was found.

INTRODUCTION

The fact that about one third of the primary energy production in developed countries is consumed by heating, ventilating and air conditioning of non-industrial and residential buildings initiated extensive development of new HVAC systems that allow indoor climate control with reduced energy demand. Some of these systems are associated with moderate drifts of indoor temperatures, also during the building occupancy period. Allowing indoor temperatures to drift rather than to keep them steady, which is common practice in most air-conditioned buildings, may be a feasible means to reduce the building energy demand. The results of previous research suggest that slow temperature ramps up to ±0.5 K/h (23-27ºC) have no influence on the width of the comfort zone as established under steady-state conditions [1, 2]. Berglund and Gonzalez [1] also concluded that temperature limits for 80% acceptability of the thermal environment, particularly the upper limit, defined by ASHRAE Standard 55 [3] might be conservative in case of drifting temperatures. Knudsen et al. [4] addressed the possibility to use the PMV/PPD model [5] (developed and intended for use mostly under steady-state conditions) to predict thermal sensation during temperature ramps up to ±5 K/h. On the other hand, an increase of indoor temperatures in a building has been shown to increase symptoms of fatigue, headache and difficulty in thinking clearly [6]. A field study by Mendell et al. [7] showed significant effects of temperature on the prevalence

of SBS symptoms, even within the comfort zone. Elevated intensity of SBS symptoms may be expected to negatively affect the performance of occupants' mental work.
In the view of the fact that expenses incurred in employee salaries exceed building energy and maintenance costs by a factor of 80 to 100, it becomes clear that reduced energy consumption should not be achieved at the expense of occupants' comfort, productivity or even health [8]. Although previous studies conducted in the climate chambers examined a large range of temperature ramps, their focus was mostly on establishing temperature limits for acceptable thermal comfort. The objective of this study was to extend the knowledge on the effect of moderate operative temperature drifts to cover not only human thermal comfort, but also the intensity of SBS symptoms and perceived air quality.

METHODS

Fifty-two healthy subjects (mean age: 23.7±4.4 (yrs); mean height: 176.6±9.4 (cm); mean weight: 74.7±11.9 (kg); 50% female), recruited college students, were seated in a climatic chamber and exposed to different conditions during two related experimental series; see Table 1. The climate chamber (5 m wide, 6 m long and 2.5 m high) was developed to control accurately the thermal environment [9]. Ventilation system provided approx. 170 L/s of fresh air (air change rate 9 h^{-1}). That corresponds to about 28 L/s per person when considering 6 persons seated in the chamber at the time. Air and radiant temperatures, which were identical during all exposures, air velocity and air humidity were measured continuously at the center of the chamber at 0.6 m above the floor. The water vapor pressure 1.53 kPa (corresponding to 50% RH at 24ºC) was kept constant during all exposures.

In both experimental series, subjects were divided into groups of six. Each group came to an experimental session on the same week-day. Each group experienced one condition a week and the order of the conditions was randomized (Latin Square Design). Subjects were seated at separate workstations consisting of a desk, a chair and a PC. Each session started with an acclimatization period (30 min.). Two times per hour, subjects filled in a questionnaire that included a 7-point thermal sensation interval scale, scales to assess the acceptability of the thermal environment and perceived air quality, a scale to assess odor intensity (Figure 1a) and Visual Analogue Scales (VAS) to assess the intensity of selected specific and general SBS symptoms [10]. A scale to assess self-estimated performance was also included in the questionnaire. Subjects' performance was measured by simulated office work tasks but these results are not included in this paper. Both questionnaires and office work tasks were presented to subjects in an Internet browser [11]. Subjects wore their own clothing during all experimental sessions. Clothing composition was determined during pre-sessions in which subjects were exposed to constant operative temperature 24.4 and 21.4ºC (50% RH, 2 hours) for experiment 1 and 2 respectively. They were also allowed to move around the chamber casually, so that their metabolic rate was approximately 1.2 met.

A commercially available statistical software package was used to analyze the data. Data with normally distributed residuals were tested with Linear Mixed Effects model. Kolmogorov-Smirnov goodness of fit test was used to test whether residuals were normally distributed. In case of not normally distributed residuals, a non-parametric Friedman Rank Sum test was applied.

The percentage of subjects dissatisfied with their thermal condition was determined by calculating the percentage of votes that were placed in the "negative" part of the thermal acceptability scale (from "just unacceptable" to "clearly unacceptable") for each vote.

RESULTS

Results of physical measurements and mean values of clothing insulation for examined conditions are summarized in Table 1. Measured air velocity was lower than 0.1 m/s during all experimental sessions.

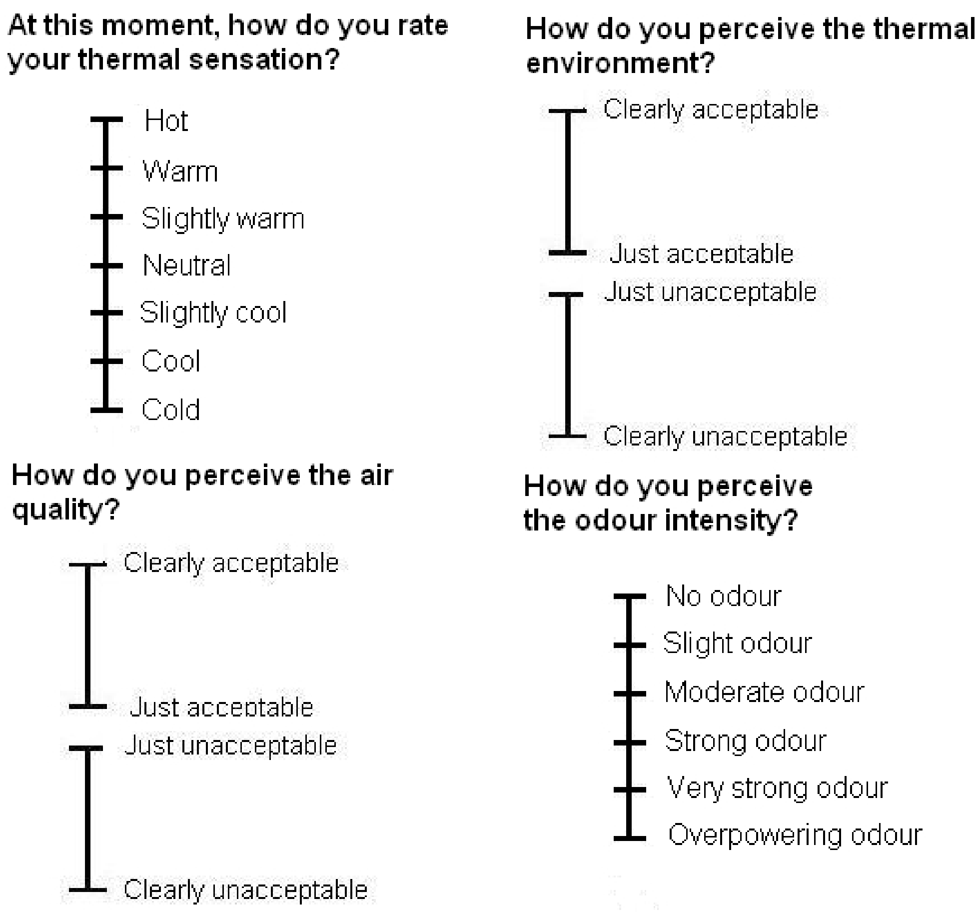
Table 1. Experimental conditions and results of physical measurements (mean±SD)

Experimental design				Physical measurements			
Temp. ramp, (K/h)	Temp. range, (ºC)	Clothing insulation, (clo)	Exposure duration, (h)	Start operative temperature, (ºC)	Final operative temperature, (ºC)	Hum. ratio, ($g/kg_{d.a.}$)	Measured clothing insulation, (clo)
Pre-session (Experiment 1)							
0.0	24.4	0.5-0.6	2	24.4±0.07[1]	-	9.2±0.2	0.54±0.18
Experiment 1							
0.0	24.4		4	24.4±0.05[1]	-	9.5±0.7	0.55±0.17
+2.4			2	22.2±0.17	26.4±0.08	9.2±0.2	0.53±0.13
+1.2	22.0-26.8	0.5-0.6	4	22.2±0.09	26.4±0.08	9.2±0.2	0.56±0.13
+0.6			8	22.2±0.03	26.8±0.02	9.2±0.2	0.54±0.17
+4.8			1	22.1±0.12	26.4±0.05	9.2±0.3	0.52±0.13
Pre-session (Experiment 2)							
0.0	21.4	0.9-1	2	21.4±0.07[1]	-	9.1±0.2	0.78±0.19
Experiment 2							
0.0	21.4		6	21.5±0.08[1]	-	9.5±0.2	0.74±0.18
+0.6	19.0-23.8		8	19.2±0.15	23.7±0.02	9.0±1.5	0.75±0.2
+1.2	17.8-25.0	0.9-1	6	18.3±0.26	24.8±0.06	9.5±0.4	0.74±0.17
-0.6	23.8-19.0		8	23.7±0.32	19.9±0.07	9.5±0.2	0.73±0.18
-1.2	25.0-17.8		6	25.2±0.21	19.2±0.19	9.4±0.3	0.74±0.17

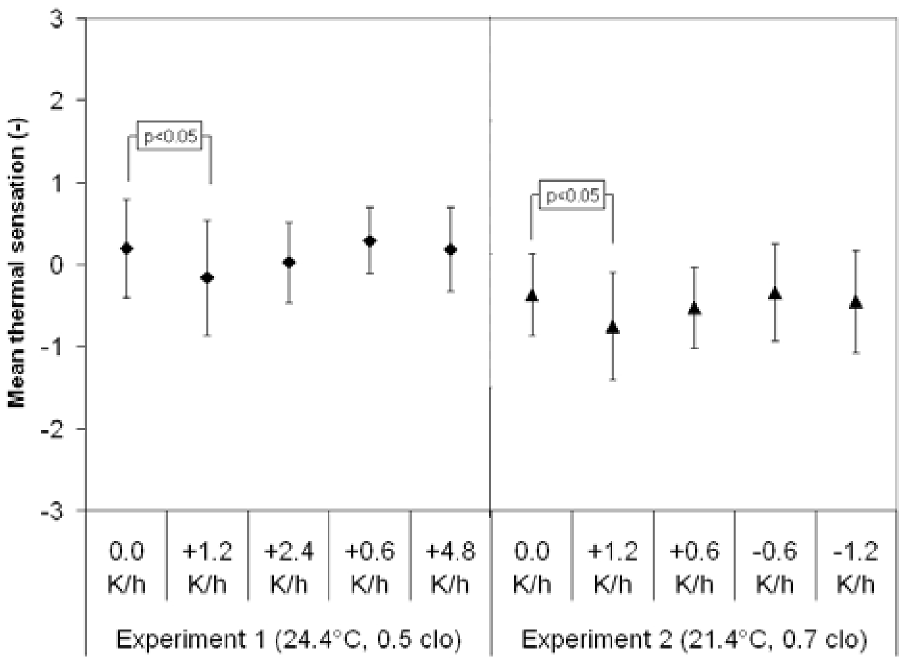
[1]In case of exposure to constant temperature mean value for whole condition is given;

Thermal sensation

Comparison of the mean thermal sensation from conditions with constant temperature (24.4 and 21.4ºC) to data from the same temperature level reached by different ramps is depicted in Figure 1b. The analysis showed, that mean thermal sensations differed significantly only in case of comparison to +1.2 K/h ramp at both temperature levels.



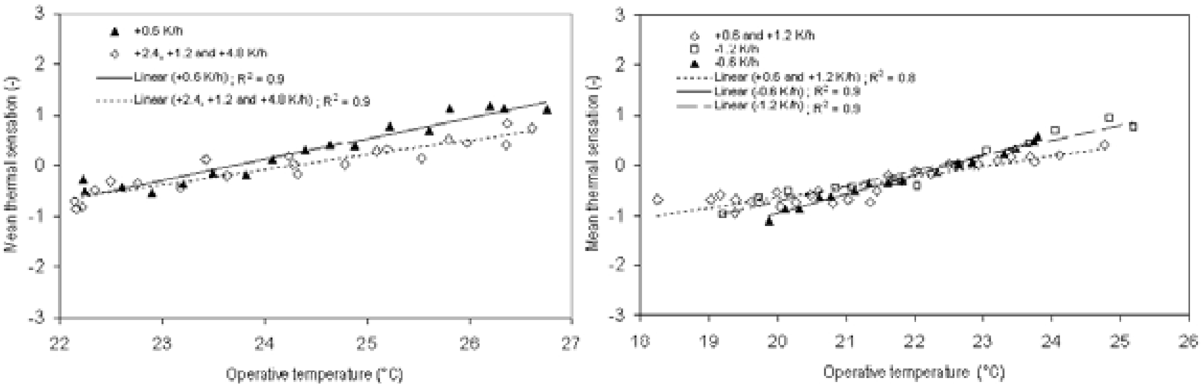
a)



b)

Figure 1. a) Interval and acceptability scales; b) Comparison of mean thermal sensation at the level of thermal neutrality reached by temperature ramps to data from exposure to constant temperature, vertical bars indicate ± SD

A linear relation between thermal sensation and operative temperature was observed in all studied ramps. The data obtained in experiment 1 are depicted in Figure 2a. Analysis with Linear Mixed Effects model showed that an intercept and slope of the linear relation did not differ significantly for +2.4, +1.2 and +4.8 K/h ramps. With these ramps, subjects perceived operative temperature the same way regardless the ramp they were exposed to. Common linear fit for +2.4, +1.2 and +4.8 K/h ramp is indicated by dash line in Figure 2a. The development of thermal sensation was different in +0.6 K/h ramp (Figure 2a – solid line). During this exposure, thermal sensation for the same level of operative temperature was higher. This difference tended to be larger for higher temperatures. Analysis showed significant difference in both slope and intercept ($p<0.001$) when +0.6 K/h ramp was compared to +2.4, +1.2 and +4.8 K/h ramps.



a) b)

Figure 2. Mean thermal sensation votes as a function of operative temperature: a) experiment 1 - 22-26.8ºC, 0.5 clo, b) experiment 2 - 17.8-25ºC, 0.7 clo

Figure 2b presents a linear relation between operative temperature and mean thermal sensation in experiment 2. Analysis of the data showed no significant difference in slope or intercept of the regression line for +0.6 and +1.2 K/h ramp. Common linear fit for +0.6 and +1.2 K/h ramp is indicated by the dashed line. In case of decreasing ramps, the thermal sensation changed with temperature significantly faster than in increasing ramps. The solid line in Figure 2b represents a linear fit for -0.6 K/h ramp data. Its slope is significantly different ($p<0.0001$) from the one of the line representing increasing ramps (+0.6 and +1.2 K/h). The slope of -1.2 K/h ramp differed both from the slope of increasing ramps ($p<0.05$) and the slope -0.6 K/h ramp ($p<0.01$).

Percentage of dissatisfied with thermal environment

Figure 3 shows the percentage of dissatisfied with the thermal environment (calculated from thermal acceptability data) as a function of mean thermal sensation. The curve indicating Predicted Percentage of Dissatisfied (PPD) [5] was included in the figure for comparison. As it can be seen from the figure, observed data follows rather closely the PPD curve.

Acceptability of air quality and intensity of odour

The results from exposures to constant temperature indicated that subjects needed approximately 120-150 minutes to adapt their olfactory senses to the pollution level in the climatic chamber. During exposure to temperature ramps, the acceptability of the air quality depended linearly on the operative temperature. This relationship was statistically significant for +0.6 K/h ramp in experiment 1 and -1.2 K/h ramp in experiment 2 (Figure 4a). The results seem to be in agreement with results obtained by Fang et al. [6]. The effect of adaptation observed under steady-state conditions could not be studied with temperature ramps as the acceptability of the air quality was affected both by changing temperature (enthalpy) and different initial temperature level.

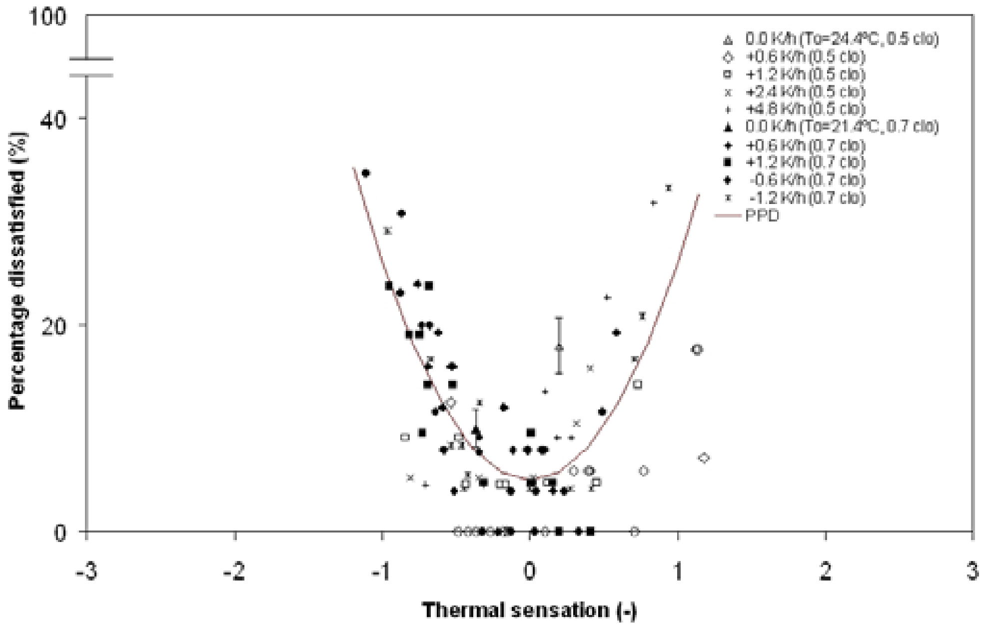
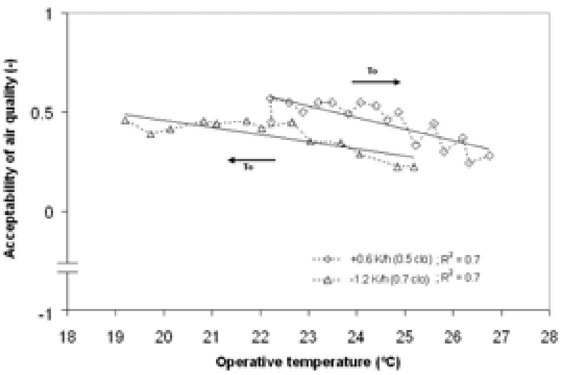
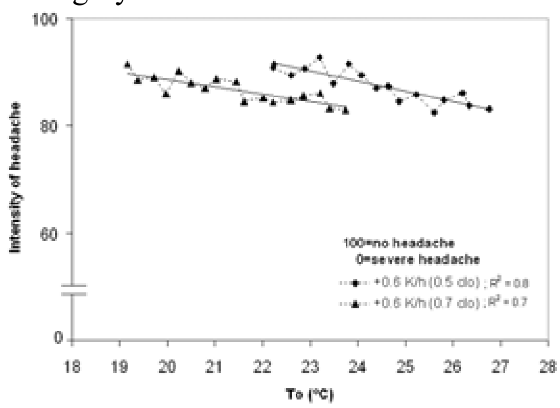


Figure 3. Percentage of subjects dissatisfied with the thermal environment; in case of exposure to constant condition the mean percentage of dissatisfied ± SEM calculated for this condition is given

No significant change in the intensity of odour was observed in experiment 1 and 2. Overall means were 0.8 ± 0.6 and 0.7 ± 0.5 for experiment 1 and experiment 2 respectively, which corresponds to a little bit less than "slight odour" category on the scale.





a) b)

Figure 4. a) Acceptability of air quality as function of operative temperature, arrows indicate direction of the temperature ramp; b) Development of intensity of headache for +0.6 K/h temperature ramp in experiment 1 (0.5 clo) and experiment 2 (0.7 clo)

Intensity of SBS symptoms

Increasing temperature ramps increased the intensity of headache, aggravated concentration ability and decreased self-evaluated performance. An example of the development in intensity of headache for the +0.6 K/h ramps is depicted in Figure 4b. On the other hand, -1.2 K/h ramp had a positive effect on the ability to concentrate, self-evaluated performance and decreased fatigue. It also decreased mouth and skin dryness. Table 2 summarizes results of the analysis of data regarding intensity of SBS symptoms for all tested conditions. The results clearly show that general symptoms were more affected than specific symptoms related to local irritation of mucous membranes (nose, eyes, mouth).

Table 2. Development of intensity of SBS symptoms during exposure to operative temperature ramps and constant temperature; see legend below table

Ramp (K/h)	Nose	Mouth Dryness	Skin Dryness	Eye	Smart. eyes	Gritty eyes	Head-ache	Well feeling	Fatigue	Con. ability	Self-eval. perform.
Experiment 1											
0.0	NS	NS	NS	NS	NS	NS	↓	NS	NS	NS	NS
+0.6	NS	NS	NS	NS	NS	NS	↓	↓	↓	↓	↓
+1.2	NS	|	NS	|	↓	NS	↓	↓*	NS	↓	↓
+2.4	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS
+4.8	NS	NS	NS	NS	NS	NS	↓	↓	NS	↓	↓*
Experiment 2											
0.0	|*	NS	NS	NS	NS	|*	NS	|	|*	NS	↓
+0.6	NS	NS	NS	NS	NS	NS	↓	↓	NS	↓	↓
+1.2	↓	NS	NS	NS	NS	NS	|*	NS	|	↓	↓
-0.6	NS	↑	NS	NS	NS	NS	|	↓	|	↓	↓
-1.2	NS	↑	↑*	NS	NS	NS	NS	NS	↑*	↑	↑*

Legend: vertical line (|) indicates that there was a significant change of the symptom along the condition, but no consistent trend of mean values was observed; arrow (↑) indicates that symptom became significantly less severe along the condition; arrow (↓) indicates that symptom became significantly more severe along the condition; NS = no significant change of symptom was observed; asterisk (*) indicates that statistical analysis showed either low level of significance (0.05 > p > 0.01) or result was very close to significance (0.1 > p > 0.05)

DISCUSSION

No significant effect of temperature ramps was observed for SBS symptoms related to local irritation of mucous membranes, while general symptoms like intensity of headache, well feeling and concentration ability were significantly affected in most of the ramps. The results are in agreement with previous research [13]; An increase of the operative temperature increases also the intensity of a particular symptom. However, it should be noted that the slope of the symptom increase is much lower than shown previously; about 2% in intensity per 1ºC increase in temperature in comparison to 12% increase in intensity of SBS symptoms per 1ºC increase temperature above about 22.5ºC as estimated by Seppänen and Fisk [14]. Moreover, the general level of the symptom intensity was rather low in the present study.

The observed relationship between the acceptability of air quality and operative temperature (respective enthalpy of the air) suggests that in case of insufficient ventilation, temperature increase can cause further degradation of perceived air quality. Poor air quality may, in the end, result in increased intensity of SBS symptoms, decrease overall comfort and productivity of the occupants.

Subjects dressed in summer clothing (approx. 0.5 clo) immediately perceived operative temperature ramps +2.4 K/h, +1.2 K/h and +4.8 K/h in the range 22-26.8ºC. The perception of the thermal environment was the same regardless of the ramp they were exposed to. Temperature ramp +0.6 K/h was recognized with about 3 hours delay. Afterwards, the same operative temperature was perceived as warmer than in case of steeper ramps. However, a higher thermal sensation was not accompanied by higher dissatisfaction with the environment.

During exposure to +0.6 K/h ramp at lower temperatures (19-23.8ºC, 0.7 clo) subjects did not perceive given operative temperature differently from +1.2 K/h ramp. This results suggests it may not be the temperature ramp itself, but rather the combination of a temperature level above 24.4ºC and time of the exposure that influenced the thermal sensation of the subjects. Although the design of the experiment did not allow separation of the combined effect of rising temperature and time, the fact that subjects spent 8 hours in the climatic chamber could result in fatigue, which consequently influenced thermal sensation. The results of the present study seem not to be in conflict with findings of previous research [1]. Subjects did not distinguish slow temperature increase +0.6 K/h for the first 3 to 4 hours of the exposure. However, as the exposure continued, a linear relation between thermal sensation and temperature was observed. A higher level of clothing insulation seemed to enlarge the delay period for flat ramps (+0.6 K/h) and even introduce it for steeper ramps (+1.2 K/h).

A comparison of the percentage dissatisfied with thermal environment obtained in the present study with PMV/PPD model did not consistently show that acceptability range specified by current standards could be extended when subjects were exposed to temperature ramps [3, 12]. During exposure to the +4.8K/h ramp, the percentage of dissatisfied increased faster than predicted by the model. On the other hand, subjects accepted the slightly warm environment during the +0.6 K/h ramp more readily than the model would predict. The data from ±1.2, +2.4 and -0.6 K/h ramps followed quite well relationship given by PMV/PPD model. A rather high percentage of dissatisfied during the exposure to constant temperature 24.4ºC can not be fully explained. It is possible that as thermal acceptability data were not adjusted for higher initial metabolic rate, higher dissatisfaction in the beginning of the ramp influenced mean value given in Figure 3.

CONCLUSIONS

- Increasing operative temperature had a slight but significant negative effect on general SBS symptoms, such as intensity of headache, well feeling or fatigue. Similarly for self-evaluated concentration ability and performance.
- No general effect (negative or positive) on specific symptoms related local irritation of mucous membranes (nose, eyes, mouth) was observed.
- Adaptation to indoor air quality occurred after approx. 2 hours during exposure to steady temperature. A linear relationship between acceptability of air quality and temperature (enthalpy) was observed in temperature ramps.
- A linear relationship between mean thermal sensation and operative temperature was observed in all studied ramps. Flat ramps (±0.6 K/h) were realized by sedentary subjects with 3 to 4 hours delay (depending on the level of clothing).
- A relationship between mean thermal sensation and the percentage of thermally dissatisfied subjects was in fairly good agreement with prediction by PMV/PPD model.

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Derivation and analysis of the indoor Wet Bulb Globe Temperature index (WBGT) with a human thermal engineering approach — Part 1. Properties of the WBGT formula for indoor conditions with no solar radiation.

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SUMMARY

Based on a heat balance equation between humans and the environment, the authors developed a theoretical derivation of the indoor WBGT formula, which was originally established empirically by Yaglou and Minard. We demonstrated that the coefficients of wet bulb temperature T_w and globe temperature T_g (= air temperature T_a), in the indoor WBGT formula contain variables such as the metabolic activity, the clothing insulation and the wind velocity, and that these three coefficients do not remain strictly constant. We also considered the thermophysiological significance of the WBGT based on the derived formula. In addition, we performed a numeric examination and indicated irrational results of the WBGT.
The newly derived formula is WBGT = $0.85T_w + 0.20T_g$ (= T_a) or WBGT = $0.80T_w + 0.20T_g$ (= T_a) which differs from Yaglou et al's original formula WBGT = $0.7T_w + 0.3T_g$ (= T_a) in the coefficients.

INTRODUCTION

In 1957 Yaglou and Minard invented Wet Bulb Globe Temperature (WBGT) [1] as an index to evaluate thermal sensation and heat stress during training. The empirical index, developed for the prevention of heat stroke in U.S. soldiers, is now standardized by National Institute for Occupational Safety and Health (NIOSH) and International Organization for Standardization as ISO-7243. Although its use is recommended in many countries for the evaluation of the thermal environment during labor and outdoor sports, its background has hardly been discussed from the perspective of the heat transfer theory.

The original WBGT formula suggested by Yaglou and Minard has two versions depending on the usage environment [1].
For indoor conditions with no solar radiation;

$$WBGT = 0.7T_w + 0.3T_g \tag{1}$$

For conditions with solar radiation;

$$WBGT = 0.7T_w + 0.2T_g + 0.1T_a \tag{2}$$

where T_w is (natural) wet bulb temperature[°C], T_g is globe temperature[°C] and T_a is air temperature[°C].
Humans produce metabolic heat in the body and release or receive heat in accordance with the physical law of heat transfer. Given that a thermal sensation arises when heat produced in the body is transferred between humans and the environment, evaluation indexes of sensation, including those developed based on empirical rules or experimental data, must be in accordance with the law of heat transfer.

From the above-mentioned perspective, we derive theoretically WBGT, which was originally suggested empirically, employing the heat transfer theory based on the heat balance equation between humans and the environment. Then, based on the derived theoretical formula, we clarified the structures of the three constant coefficients of wet-bulb temperature, globe temperature and air temperature that define the original WBGT formula. Furthermore, we performed an examination using real values to present the characteristics and applicable conditions of the WBGT [2].

METHODS

First, we derive the indoor WBGT formula (1) and its analog by expanding heat balance equations between humans and the environment, and those of the wet bulb and the globe. The heat balance between humans and the environment is expressed as follows:

$$M = C + R + E_{sk} + E_{res} + W + S \qquad (3)$$

Where M is metabolic rate per unit body surface area[W/m^2], C is convective heat loss [W/m^2], R is radiative heat loss[W/m^2], E_{sk} is evaporative heat loss[W/m^2], E_{res} is evaporative heat loss from respiration[W/m^2], W is external mechanical work[W/m^2] and S is rate of heat storage[W/m^2].

The heat losses through convection C, radiation R, evaporation from the skin E_{sk} and respiration E_{res} [3] in Equation (3) are expressed by Equations (4), (5), (6) and (7).

$$C = h_c F_{cl}(T_{sk} - T_a) f_{cl} \qquad (4)$$
$$R = h_r F_{cl}(T_{sk} - T_r) f_{cl}\, f_{ref} \qquad (5)$$
$$E_{sk} = LR h_c F_{pcl}(P_{sk} - P_a) f_{cl} \qquad (6)$$
$$E_{res} = 0.0014M(35 - T_a) + 0.0173M(5.87 - P_a) \qquad (7)$$

where h_c and h_r are human's convective and linear radiative heat transfer coefficients [W/(m^2°C)], F_{cl} is thermal efficient factor[N.D.] [4], F_{pcl} is permeation efficiency factor[N.D.] [5], T_{sk} is mean skin temperature[°C], T_a is air temperature[°C], T_r is mean radiant temperature[°C], f_{cl} is clothing area factor[N.D.] [6], f_{ref} is effective radiation area ratio of human body[N.D.], LR is Lewis relationship (= 16.5) [°C /kPa], P_{sk} is water vapor pressure at skin temperature[kPa] and P_a is water vapor pressure in air[Pa].

Though Yaglou et al's paper supposes the use of the natural wet bulb temperature measured by August psychrometer, the general heat balance equation for the wet bulb is given in the form of Equation (8). Because the sensing part of the wet bulb thermometer is very small, about 6mm in diameter, the convective heat transfer coefficient h_c' is approximately 24.4~31.0W/m^2°C in normally air-conditioned rooms with wind velocities of 30~50cm/s. Meanwhile, the linear radiative heat transfer coefficient hr′ in the normal indoor environment is about 4~5W/m^2°C, an order of magnitude less than h_c', and therefore can be ignored. On the assumption that h_r'/h_c' is nearly equal to zero, Equation (8) leads, through Equation (9), to Equation (10).

$$h_c'(T_w - T_a) + h_r'(T_w - T_r) + LR\, h_c'(P_w^* - P_a) = 0 \qquad (8)$$
$$(T_w - T_a) + \frac{h_r'}{h_c'}(T_w - T_r) + LR(P_w^* - P_a) = 0 \qquad (9)$$
$$(T_w - T_a) + LR(P_w^* - P_a) = 0 \qquad (10)$$

where, h_c' and h_r' are wet bulb's convective and linear radiative heat transfer coefficients [W/(m^2°C)], T_w is (natural) wet bulb temperature[°C], T_r is mean radiant temperature[°C] and P_w* is saturated water vapor pressure on wet bulb[kPa].

As far as the thermal environment surrounding human work is concerned, the relationship of the saturated vapor pressure of water P_w* to the wet bulb temperature T_w on the psychrometric chart is sufficiently linear, as shown in Figure 1, and can be approximated as follows:

$$P_w^* = .T_w + . \qquad (11)$$

where κ and ζ are constants of linear approximation of saturated water vapor pressure to wet bulb temperature[kPa/°C] (See Figure 1).

During work in a hot environment, the mean skin temperature is within a range of 35~40°C. Therefore, by treating the skin temperature in the same way as the wet bulb temperature, the relationship of the saturated vapor pressure of water on the skin P_{sk} to the skin temperature T_{sk} can be linearly approximated by Equation (12).

$$P_{sk} = .^*T_{sk} + .^* \qquad (12)$$

where κ* and ζ* are constants of linear approximation of saturated water vapor pressure to mean skin temperature[kPa/°C].

In the forth coming numerical examination, we will adopt the following values shown in Figure 1: κ = κ*= 0.279kPa/°C and ζ= ζ*= – 4.03kPa. The vapor pressure of water on the skin surface P_{sk} is expressed by Equation (12) ′, accompanied with μ that indicates the degree of saturation on the skin surface. Here we expand equations with μ preserved, but in the later stages, we will suppose μ=1 because WBGT index is designed for use within the thermal limitation where the vapor pressure of water on the skin surface is almost saturated in consequence of promoted sweating[7].

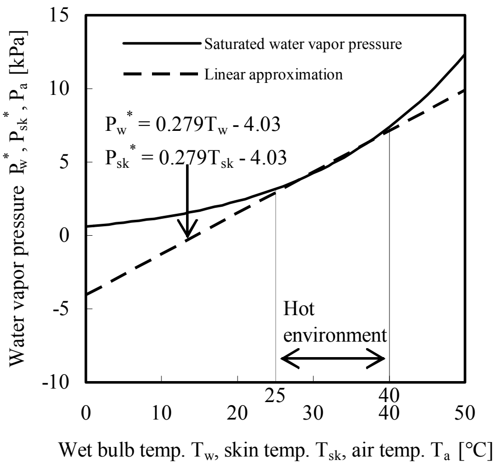

Figure 1 Linear approximation of saturated water vapor pressure to skin and air temps.

$$P_{sk} = \mu(.^*T_{sk} + .^*) \qquad (12)'$$

where μ is saturated ratio of water vapor pressure on skin surface[N.D.].

Meanwhile, the steady-state heat balance after sufficient exposure of the globe thermometer to the indoor environment with little influence of solar radiation for actual measurement is expressed by Equation (13).

$$h_c''(T_g - T_a) + h_r''(T_g - T_r) = 0 \quad (13)$$

where h_c'' and h_r'' are globe's convective and linear radiative heat transfer coefficients [W/(m^2°C)].

RESULTS

Substituting Equations (4)~(7) and Equations (10)~(13) into Equation (3) to eliminate C, R, E_{sk}, E_{res}, P_{sk}, P_a, P_w* and T_r and arranging the yielded equation, we obtain Equation (14).

$$\begin{aligned}
&\left[(h_c + h_r)F_{cl}f_{cl} + \mu\kappa^* \cdot LRh_cF_{pcl}f_{cl}\right]T_{sk} \\
&+\left[(\mu\zeta^* - \zeta)LR \cdot h_cF_{pcl}f_{cl} - M(0.8494 + 0.0173\zeta) + W + S\right] \\
&= \left[h_cF_{pcl}f_{cl}(1 + \kappa \cdot LR) + 0.0173M\left(\kappa + \frac{1}{LR}\right)\right]T_w \qquad (14) \\
&+\left[F_{cl}f_{cl}(h_c'' + h_r)\right]T_g \\
&+\left[F_{cl}f_{cl}(h_c - h_c'') - h_cF_{pcl}f_{cl} + M\left(0.0014 - \frac{0.0173}{LR}\right)\right]T_a
\end{aligned}$$

The convective heat transfer coefficients of the human body [8], the wet bulb [9] and the globe [10] show a significant difference for the same wind velocity according to the difference between their representative parameters, whereas their radiation heat transfer coefficients h_r, h_r' and h_r'' are nearly equal. Furthermore, in the indoor environment, which is free from the influence of solar radiation, T_r and T_a can be regarded as the same, and the relationship $T_r = T_a = T_g$ makes it possible to arrange Equation (14) into Equation (15).

$$\begin{aligned}&\left[(h_c + h_r)F_{cl}f_{cl} + \mu\kappa^* \cdot LRh_cF_{pcl}f_{cl}\right]T_{sk} \\ &+\left[(\mu\zeta^* - \zeta)LR \cdot h_cF_{pcl}f_{cl} - M(0.8494 + 0.0173\zeta) + W + S\right] \\ &= \left[h_cF_{pcl}f_{cl}(1 + \kappa \cdot LR) + 0.0173M\left(\kappa + \frac{1}{LR}\right)\right]T_w \\ &+\left[(h_c + h_r)F_{cl}f_{cl} - h_cF_{pcl}f_{cl} + M\left(0.0014 - \frac{0.0173}{LR}\right)\right]T_g\end{aligned} \tag{15}$$

Writing the coefficient of the skin temperature T_{sk} as ξ, the other terms on the left-hand side as η and the coefficients of T_w and T_g on the right-hand side as α and δ, respectively, in Equation (15), we obtain Equation (20). Dividing the first term on the left-hand side of Equation (20) by ξ, we obtain Equation (21), a form based on the skin temperature T_{sk}.

$$\left[(h_c + h_r)F_{cl}f_{cl} + \mu\kappa^* \cdot LRh_cF_{pcl}f_{cl}\right] = \xi \tag{16}$$

$$\left[(\mu\zeta^* - \zeta)LR \cdot h_cF_{pcl}f_{cl} - M(0.8494 + 0.0173\zeta) + W + S\right] = \eta \tag{17}$$

$$\left[h_cF_{pcl}f_{cl}(1 + \kappa \cdot LR) + 0.0173M\left(\kappa + \frac{1}{LR}\right)\right] = \alpha \tag{18}$$

$$\left[(h_c + h_r)F_{cl}f_{cl} - h_cF_{pcl}f_{cl} + M\left(0.0014 - \frac{0.0173}{LR}\right)\right] = \delta \tag{19}$$

$$\xi T_{sk} + \eta = \alpha T_w + \delta T_g \tag{20}$$

$$T_{sk} + \left(\frac{\eta}{\xi}\right) = \left(\frac{\alpha}{\xi}\right)T_w + \left(\frac{\delta}{\xi}\right)T_g \tag{21}$$

The right-hand sides of Equations (20) and (21) both take the same form as Yaglou et al's WBGT formula (1) does. In particular, Equation (21) expresses the relationship of the skin temperature T_{sk}, the wet-bulb temperature T_w and the globe temperature T_g in a form based on the temperature on the human side.

We show the results of examination using concrete values of the α/ξ and δ/ξ in Figures 2 to 4, changing the combination of parameters.

Figure 2 shows the variation of the α/ξ and δ/ξ values in response to that of metabolic rate with clothing insulation and indoor wind velocity held constant. The vertical axis indicates the α/ξ, δ/ξ and $\alpha/\xi + \delta/\xi$ values, and the horizontal axis indicates metabolic rate. Responding to metabolic rates of 1 to 4met, the α/ξ value ranges from 0.81 to 0.88 and the δ/ξ remains around 0.21. These values are close to 0.8 and 0.2 rather than to 0.7 and 0.3, the coefficient values in the original formula (1). The sum of α/ξ and δ/ξ ranges from 1.02 to 1.09, slightly exceeding 1.0. Since α/ξ and δ/ξ represent a prorated proportion, their sum must remain at 1.0 constantly. However, as seen in Figure 2, both values increase as metabolic rate increases.

Similarly, Figure 3 shows the variation of the α/ξ and δ/ξ values in response to that of wind velocity (horizontal axis) with metabolic rate and clothing insulation held constant. The α/ξ and δ/ξ values are almost constant at 0.85 and 0.21, roughly 0.85 and 0.21 as in Figure 2, respectively. However, in Figure 3, unlike Figure 2, the δ/ξ value decreases and the sum of α/ξ and δ/ξ approaches 1.0 as wind velocity increases.

Figure 4 shows the variation of the α/ξ and δ/ξ values in response to that of clothing insulation (horizontal axis) with metabolic rate and wind velocity held constant. The α/ξ value is 0.86 and the δ/ξ value ranges from 0.18 to 0.23. As clothing insulation increases, the α/ξ value remains almost constant and the δ/ξ value increases, and consequently, the sum of α/ξ and δ/ξ increases.

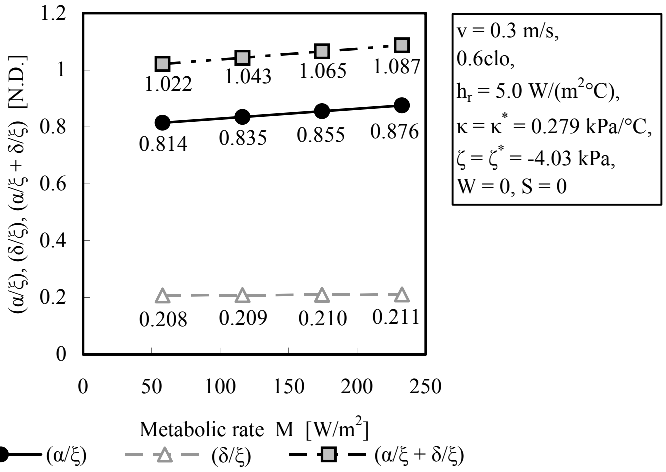

Figure 2 Weighting factors in Eq.(21) in case of variable metabolic rate under const. clothing insulation and air velocity.

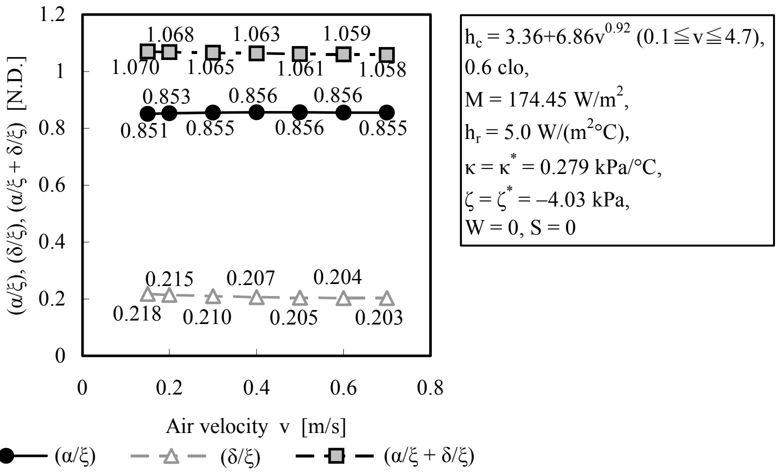

Figure 3 Weighting factors in Eq.(21) in case of variable air velocity under const. clothing insulation and metabolic rate.

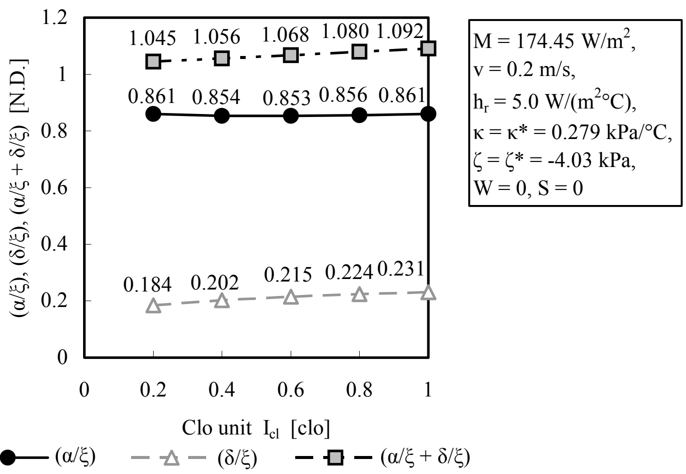

Figure 4 Weighting factors in Eq.(21) in case of variable clothing insulation under const. air velocity and metabolic rate.

DISCUSSION

We consider the general properties of Equation (21).
Seeing the structure and constituent elements of the coefficient of air temperature Ta in Equation (14) preceding Equation (21), we understand that the value of the coefficient is either very small or negative depending on the values of the constituent elements. This means that air temperature makes a small contribution to human heat stress during work in a hot environment compared to wet bulb temperature (humidity) and globe temperature (radiant heat), which qualitatively agrees with Yaglou et al's original formula (1) in which the coefficient of Ta is small compared to the other coefficients.
Comparing the coefficients of wet bulb temperature T_w and globe temperature T_g in Equation (14) with each other, assuming the values of their constituent elements, we understand that both of the coefficients are positive and that the coefficient of T_w is much greater than that of T_g. This agrees qualitatively with the original WBGT formula (2) in which the coefficient of T_w is 0.7 and that of T_g is 0.2.
Next, we consider the thermo-physiological implication of the two formulas through Equations (20) and (21), which are derived for indoor use.
In 1923, Houghten and Yaglou developed the Effective Temperature [11]. Afterward, Yaglou focused his attention on the mean skin temperature, speculating about its close connection with thermal sensation and comfort [12]. These facts provoke the supposition that the WBGT value on the left-hand sides of Equations (1) and (2) is meant for the mean skin temperature and that the coefficients of T_w and T_g, 0.7 and 0.3, on the right-hand side merely represent a prorated proportion. More specifically, the left-hand side of Equation (21) can be considered to express human heat stress caused by heat loads from the environment as the degree of increase in the mean skin temperature.
Using Equation (20), we consider the possibility that the left-hand sides of Equations (1) and (2) express the quantity of heat, not the mean skin temperature.
As is clear from Equations (16) and (17), the left-hand side of Equation (20) has the unit of heat quantity W/m^2. This means that α and δ on the right-hand side are not dimensionless but have the unit $W/m^2°C$. Comparing Equation (20) with the definitional equation of the WBGT (1), we understand that α and δ correspond to $0.7W/m^2°C$ and $0.3W/ m^2°C$, respectively. Viewed in this light, Equation (20) possibly expresses something like the body heat content of a person working, under certain conditions of wind velocity and clothing, in an environment characterized by an air temperature of Ta (= a mean radiant temperature of T_r = a globe temperature of T_g) and a wet bulb temperature of T_w. It is unlikely, however, that Equation (20) expresses the heat content of the human body because the coefficient of T_{sk} represents the heat transfer between the skin and the environment. Thus, the physical meaning of $0.7W/m^2°C$ and $0.3W/m^2°C$ remains unclear. The values of α and δ in Equations (18) and (19), corresponding to the coefficients on the right-hand side of the WBGT formula (1), are $9.8\sim28.8W/m^2°C$, or 13~41 times larger than $0.7W/m^2°C$, and $2.7\sim5.0W/m^2°C$, or 9~17 times larger than $0.3W/m^2°C$, respectively, showing an order of magnitude difference. Based on these results, it is reasonable to consider 0.7 and 0.3 in Equation (1) to be dimensionless and to merely represent a prorated proportion.
Furthermore, we will examine the variation range of the second term η/ξ on the left-hand side by substituting different state values on the assumption that external work W=0 and heat storage S=0.
Since WBGT is an index that indicates the thermal limitation, S is unlikely to be strictly equal to 0 at every moment. Our assumption that S=0 is based on the thought that the body temperature is not raised by continuous heat accumulation throughout the working hours but is put into a certain cycle through work and rest. The NIOSH and ISO-7243 have provided

their recommended proportions of work time to all working hours, typically 8 hours, based on WBGT.

We calculated the value of η/ξ, changing metabolic rate from 1 to 4met, clothing insulation from 0.2 to 1.0clo and indoor wind velocity from 0.15 to 0.7m/s. The calculated values were all negative, ranging from –15.6 to –1.4°C, which is in accordance with the recommended WBGT limits for heat stress given at 20~33°C, temperatures lower than the skin temperature. The negative values are also partly attributed to the assumption that external work W and heat storage S are equal to zero. The ratio of the absolute value of η to metabolic rate M, ranging from 1 to 4met, or $|\eta|/M$, is approximately 0.78 and external work is about 0.2M at most[3], so heat storage S needs to be as large as 0.5~0.6M for η/ξ to be positive.

CONCLUSIONS

The most important points for a thermal index is that it be in accordance with thermo-physiological and psychological reactions and that the heat balance be maintained. The index must qualitatively and quantitatively reflect the thermal reactions of humans with high accuracy.

WBGT has been used for almost 50 years. However, despite its frequent use, the index has infrequently been theoretically analyzed. In this study, we derived theoretically the WBGT formula, which was originally developed empirically, based on the heat transfer equation. We clarified the internal structures of the three constant coefficients of wet-bulb temperature, globe temperature and air temperature, and we performed an examination using real values. Based on the derived formulas and the obtained results, we presented the characteristics of WBGT as an index.

Examination by the derived formula confirmed the following points:

1-The coefficients of wet-bulb temperature T_w, globe temperature T_g and air temperature T_a in WBGT formula, considering their elements and structures, are not strictly constant but variable depending on metabolic rate, clothing insulation, wind velocity and other such factors.

2-An examination using real values showed the possibility of the new formula WBGT = $0.85T_w + 0.20T_g$ (= T_a) or WBGT = $0.80T_w + 0.20T_g$ (= T_a), which differs from Yaglou and Minard's original formula WBGT = $0.7T_w + 0.3T_g$ (= T_a) in the coefficients.

3-WBGT can be used more accurately by specifying the values of the coefficients of T_w, T_g and T_a, substituting human and environmental conditions into Equations (14) and (15) derived based on the heat balance equation concerning the human body.

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Velocity Preference of Tropically-Acclimatized People after medium-term (90 minutes) exposure to Local Air Movement at the Face

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SUMMARY

The existing draft guideline [1] is based on laboratory studies, in which participants were exposed whole-body to each condition for 15 minutes. However, the understanding is limited on acceptable air velocity of locally applied air flow over a longer duration.

This study explores human perception of a group of 24 people in the Tropics with 90-minute (medium-term) facial exposure to local air movement with the aim of examining the local velocity preference and how this preference changes over time. The subjects were exposed to local airflow at six velocity levels, at temperatures between 21.0 to 26.0°C.

The study shows that the subjects preferred a range of local air movement during the 90 minutes of facial exposure, with the optimum velocities identified to be between 0.2 and 0.6 m/s within the experimental conditions. These optimum velocities are relevant to the design of personalized ventilation in practice.

INTRODUCTION

The existing draft guideline [1] is based on laboratory studies [2,3], in which participants were exposed whole-body to each condition for 15 minutes. Numerous studies reported the decrease of thermal sensation or the increase of draft sensitivity either with whole-body or local exposure to air flow [4,5,6]. However, the understanding on acceptable air velocity of locally applied air flow over a longer duration is limited. The understanding of acceptable local air velocity/flow rate is indispensable for the design of personalized ventilation. This is especially so for tropically-acclimatized people who has been shown to demonstrate a preference for some air movement over a short-term (15-minutes) exposure.

This study explores human perception of a group of 24 people in the Tropics with 90-minute (medium-term) facial exposure to local air movement with the aim of examining the local velocity preference and how this preference changes over time.

METHODS

The experiment was conducted in a controlled indoor air quality (IAQ) chamber with dimensions of 6.6m (L) x 3.7m (W) x 2.6m (H) in Singapore. (The detailed description of the experimental chamber can be found in [7]). Interior partitions divided the chamber into six workstations, each being equipped with a personalized air terminal device to supply conditioned outdoor air. Twenty-four tropically acclimatized subjects, 12 males and 12 females, participated in the experiments and responded to questionnaires on their thermal and draft sensations using visual-analogue scales. (The details of personalized air terminal devices, the human subjects, the questionnaires and experiment procedure can be found in [8].

The subjects were exposed to predefined local air movement with velocities ranging from 0.15 to 0.90 m/s, and localized (personalized ventilation) airflow temperatures of 21.0 and 23.5 °C at room temperatures of 23. 5 and 26.0 °C (refer to Table 1). Ambient-local temperature (T_a-T_f) is used to denote each condition. Each combination was maintained for 90 minutes of facial exposure, during which the subjects performed normal office work and responded to the questionnaires at the 5th, 10th, 15th, 30th, 45th, 60th, 75th and 90th minutes during the experiments.

Table 1. Experimental conditions

Room Temperature T_a (ºC)	26.0		23.5
Target Temperature T_f (ºC)	21.0	23.5	21.0
Target Velocity V_f (m/s)	0.15, 0.30, 0.45, 0.60, 0.75, 0.90		

RESULTS

The percentage dissatisfied is plotted against air velocities at different times in Figure 1 through Figure 6 to examine the optimum velocities. Quadratic regression is applied in the figures. The curves have similar shape, in which percentage dissatisfied initially decreases with the increasing air velocity and beyond a certain velocity (optimum velocity) the percentage dissatisfied begins to increase. This scenario illustrates that to a certain extent, air movement may be considered pleasant by the subjects during the 90-minute duration of exposure.

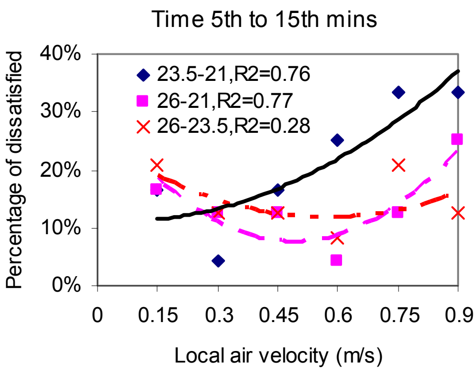

Figure 1 Percentage dissatisfied from exposure time 5th to 15th minutes

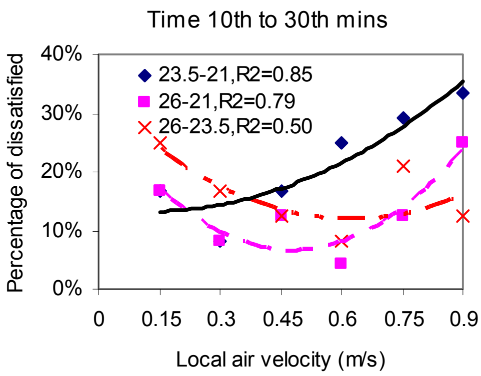

Figure 2 Percentage dissatisfied from exposure time 10th to 30th minutes

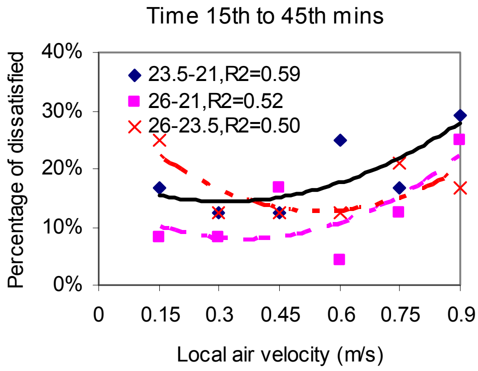

Figure 3 Percentage dissatisfied from exposure time 15th to 45th minutes

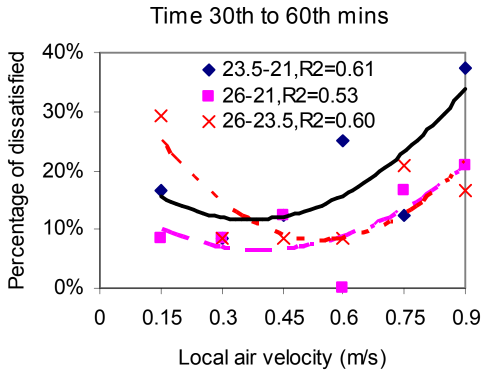

Figure 4 Percentage dissatisfied from exposure time 30th to 60th minutes

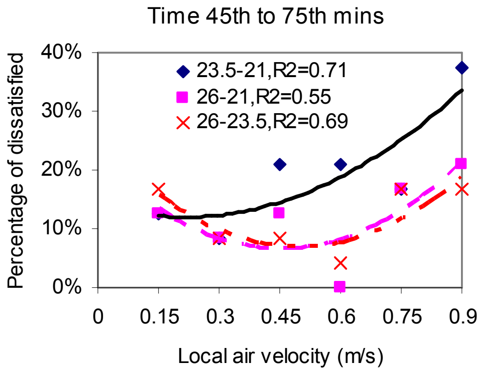

Figure 5 Percentage dissatisfied from exposure time 45th to 75th minutes

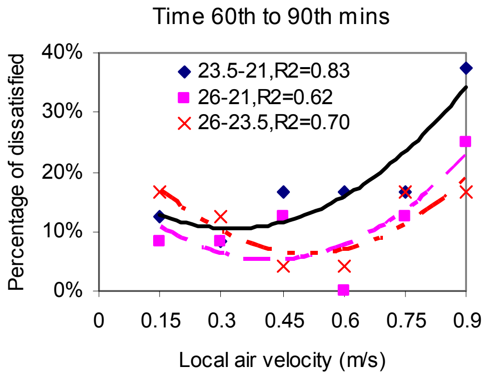

Figure 6 Percentage dissatisfied from exposure time 60th to 90th minutes

The optimum velocity is defined as the velocity value associated with the lowest percentage of dissatisfied. Applying the first derivative on the regression lines for percentage dissatisfied, the velocities with minimum percentage dissatisfied are calculated and depicted in Figure 7. The optimum velocities for the condition 26-23.5 range between 0.5 and 0.6 m/s, and that for condition 26-21 range between 0.35 and 0.5 m/s. The optimum air velocities (not shown) for the 15th and 30th minute for the 23.5-21 condition are unreasonable low compared with the values at the 45th minute and beyond; however, the reason is unknown.

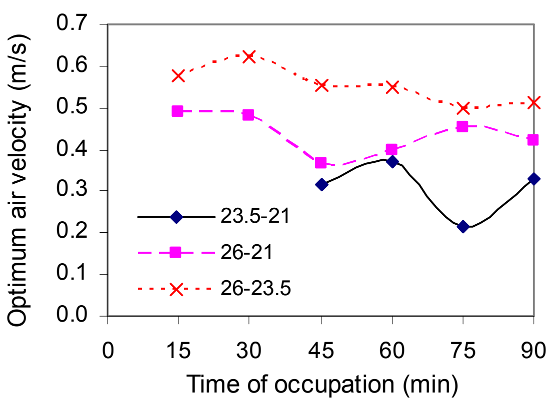

Figure 7 Optimum air velocity during 90 minutes

DISCUSSION

This study has identified the optimum velocities for a 90-minute (medium-term) facial exposure to local air movement and found the optimum velocities to fluctuate within the range between 0.2 and 0.6 m/s. Expectedly higher optimum velocities occur with higher background temperature (26^{o}C) and T_a-T_f conditions.

In a similar but separate study involving short-term exposures of 15 minute duration to localized air movement achieved through personalized ventilation, it had been demonstrated that tropically acclimatized subjects prefer facial air movement within the range 0.3 and 0.6 m/s under the same set of temperature conditions of 23.5-21.0, 26.0-23.5, 26.0-21.0 [9]. The optimum velocity ranges for 90-minute exposure identified in this study and that for 15-minute exposure are comparable.

As the optimum velocity study shows that the subjects preferred air movement to varying magnitudes at different sets of conditions during the 90 minutes' exposure to local air flow, the air movement perception and preference of the subjects were examined. These are depicted in Figure 8 and 9.

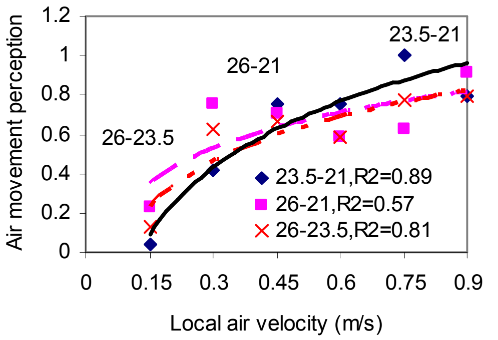

Figure 8. Logarithmic regression of air movement perception at the 90th minute (Y-axis: -3=Much too still; -2=Too still; -1=Slightly still; 0=just right; 1=Slightly breezy; 2=Too breezy; 3=Much too breezy)

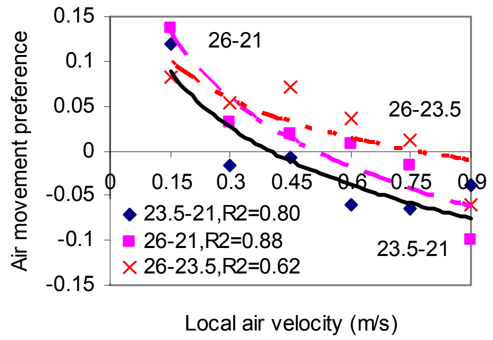

Figure 9. Logarithmic regression of air movement preference at the 90th minute (Y-axis -1=Less air movement; 0=Same air movement; 1=More air movement)

Preference of air movement was neutral around 0.45 m/s, 0.5 m/s, and 0.75 m/s for the 23.5-21, 26-21 and 26-23.5 conditions respectively. Below and above these velocities, subjects preferred more air movement respectively. It was interesting note that subjects perceive low localized air velocities to be "just right", and that at neutral air movement preference, they perceive the conditions to be between "just right" and "slightly breezy". The preference is consistent to the observation of optimum velocities that local air movement is preferred until to a certain limit, which is relevant to the temperature conditions.

CONCLUSION

The key findings of this air movement perception study of medium term exposure to local air movement of tropically acclimatized people are as follows:

1. optimum velocities for 90-minute (medium-term) facial exposure to local air movement were found to be within the range 0.2 and 0.6 m/s over the time
2. optimum velocity ranges for 90-minute exposure identified in this study and that for 15-minute exposure are comparable
3. higher optimum velocities occur with higher background temperature (26^{o}C) and T_a-T_f conditions
4. preference of air movement was neutral around 0.45 m/s, 0.5 m/s, and 0.75 m/s for the 23.5-21, 26-21 and 26-23.5 conditions respectively
5. at neutral air movement preference, the conditions were perceived to be between “just right” and “slightly breezy”.

ACKNOWLEDGEMENT

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The Substitution Of Comfort Pmv Values By A New Experimental Operative Temperature

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SUMMARY

Problems following the application of optimal operative temperatures estimated on the basis of PMV and the necessity to apply correct values into the new Czech Government Directive No. 523/2002 Code led to experiments based on the physiological human body response instead of on solely man's feelings in a defined environment. On the basis of experiments on 32 subjects (university students) it has been possible to estimate: a) total balance of hygrothermal flows between the human body and environment, b) the optimal operative temperature as a function of the subject's activity, c) the thermoregulatory range for each optimal operative temperature, i.e. maximal (category C_{max}) limited by the beginnings of sweating, minimal (category C_{min}) limited by the beginnings of shivering (category C can be applied to the natural ventilated buildings), optimal (comfort level – category A) defined by time constant 0.368 (can be applied to air conditioned buildings) and submaximum (decreased comfort level – category B) defined by time constant 0.632 (can be applied to buildings with basic air conditioning systems).

INTRODUCTION

The provision of optimal hygrothermal conditions, i.e. first of all optimal operative temperature (air being calm and air temperature reaching radiant temperature) is the principal condition for healthy living of a man in building interior. Optimal operative temperature so far is to be calculated from PMV (Predicted Mean Value) (see e.g. EN ISO 7730 Moderate Thermal Environment) estimated on the basis of positive feeling of 80% of present persons. The feelings of a man are very subjective values being impacted by many other factors in addition to hygrothermal conditions, e.g. by indoor interior colors, a man's mood etc. Furthermore, as a result of the way of experimental estimation of PMV and proved by other experimental works (see Fishman, Pimbert 1979, Newsham, Tiller 1995), it is valid approximately for the neutral zone only. The more leave the neutral zone the more the real values leave the values calculated from PMV. And more: the higher man's activity the higher the difference. The application of high activity values is then impossible in practice.
This may be due to three main reasons. The first are the assumed steady state laboratory conditions used in the derivation of the PMV equation. The second is the oversimplified approach to the assessment of the metabolic rate of the occupant. The occupants rarely sat in the room for a long period, say one hour, without moving around. The third is the sensitivity of PMV to clo values (Croome et al. 1993). It can be concluded that the PMV equation over predicts the neutral temperature by as much as 2K and under predicts the comfort requirement when air temperature deviates from neutrality.
And maybe the most important: from PMV system no thermoregulatory ranges can be estimated.

These are the reasons we decided to estimate optimal operative temperatures on the basis of the physiological response of human organism.

MATHEMATICAL MODEL OF THE PHYSIOLOGICAL BODY RESPONSE

The total heat rate production and its distribution into individual components during heat exchange between the human body and the environment are shown in *Fig.1*, where q_m=M-W = metabolic heat (see Jokl 1989). q_{res} and $q_{ev,d}$ are the components of the heat rate from the organism by respiration and by skin moistening (evaporation), with the human body being in the thermal neutral zone. The heat flow q_{dry} represents the component transferred from the organism through the clothing layer with a total thermal resistance $R_{t,wa}$(q_{dry}= q_c+q_r). The regulatory process within the neutral zone is achieved mainly by vasodilatation and vasoconstriction changing the body's internal resistance into the thermoregulatory and adaptational heat flux q_{tr} a q_a to the skin surface. q_{tr} a q_a is the heat flux regulating the instantaneous value of the skin temperature during the subject's interaction with environment, q_{tr} is the organism's immediate response to the changes of the microclimate or metabolic heat changes; q_a is the reaction shift due to adaptation to heat in summer and cold in winter. q_{tr} + q_a may be negative (heat loss) or positive (heat gain). It is the transient heat flow – even in the thermal neutral zone – that is called "quasi-stationary", to be differentiated from the hyperthermia and hypothermia zone.
q_{tr} + q_a represents the rates of heat storage or heat debt accumulation. When the body is in a steady-state thermal balance with the environment, these therms are equal to zero. But it is possible to consider the state of the subject in the neutral zone by non-steady-state conditions due to periodical changes of metabolic heat rate, q_m, or short thermal excitations in time followed by changes of internal thermal resistance of body within the neutral zone.
The temporary characteristics of each non-steady process are determined, in addition to the thermal resistances $R_{t,i}$ and $R_{t,wa}$ by the human body heat capacity, C_t. The values characterizing the heat exchange are: T_{sk}, T_{core} and T_g. The internal thermal resistance, $R_{t,i}$ also determines the changes in thermoregulation and the adaptational heat, q_{tr} + q_a, which is necessary for maintaining the skin temperature within physiological values if the core temperature should remain constant (T_{core} = 36.7 ± 0.4 °C).
The heat flow balance, as presented in the model shown in *Fig.1* can be expressed by a thermal flux equation at the boundary: subject-environment. Thus (if heat conduction is neglected):

$$q_{dry} = \frac{1}{R_{t,wa}}(T_g - T_{sk}) = q_m - q_{res} - q_{ev} + q_{tr} + q_a = q_i - q_{sw} \quad [\mathrm{Wm}^{-2}] \qquad (1)$$

where q_{ev}=$q_{ev,ins}$ +$q_{ev,sens}$= $q_{ev,ins}$+q_{sw} [W.m^{-2}]
q_m-q_{res}-$q_{ev,ins}$= q_i [W.m^{-2}]
q_{sw}= 0.6(q_m-58.14) [W.m^{-2}]
= the quantity of excreted sensible but mostly invisible sweat, it was estimated by weighing during experiments as a mean value for the whole range.
Heat flux within the human body can be represented as (see model in *Fig.1*):

$$q_m - q_{res} + q_{tr} + q_a = G_{t,ti}(T_i - T_{sk}) = \frac{1}{R_{t,ti}}(T_i - T_{sk}) \quad [\mathrm{Wm}^{-2}] \qquad (2)$$

where $G_{t,ti}$ is total body thermal conductance, which could be expressed by Equation 3:

$$G_{t,ti} = \frac{(q_m - q_{res})}{(T_i - T_{sk})} + \frac{(q_{tr} + q_a)}{(T_i - T_{sk})} = G_{t,m} + G_{t,i} \quad [\mathrm{W.m^{-2}.K^{-1}}] \qquad (3)$$

where $G_{t,i}$ is the internal thermal conductance and $G_{t,m}$ is the metabolic thermal conductance.

The thermoregulation and adaptational heat flux first affects the skin temperature, T_{sk}. The internal thermal resistance value, $R_{t,i} = 1/ G_{t,i}$, characterizing the vasodilatation and vasoconstriction process, can be calculated from the equation:

$$R_{t,i} = \frac{(T_i - T_{sk})}{(q_{tr} + q_a)} \qquad [W^1 m^2 K] \qquad (4)$$

EXPERIMENTAL ESTIMATION OF MATHEMATICAL MODEL PARAMETERS

An experiment lasting several years was undertaken in the climatic chamber from which the parameters in Equation (1) and (4) could be identified.
The experimental subjects were male university students each of them underwent six experiments lasting about three hours at four levels of activity: (1) sitting in a chair, (2) sitting on a bike-ergometer without pedaling, (3) pedaling on a bike-ergometer with a 40 W load and (4) pedaling on a bike-ergometer with a load of 1 W per kg body mass (as long as he was able to do it). Metabolic heat production during each activity was measured by indirect calorimetric method. Mean skin temperature, heat rate and body water loss were estimated continuously during each experiment.

Two sets of clothing were used by the subjects: lightweight (pajamas) and a heavier one (anti-g suit for fighter pilots). The results of anti-g suit experiments will be presented is a separate report.

There were no differences between air temperature and surface wall temperatures – it could be supposed globe temperature equals operative temperature. Six temperatures were chosen (29±3 °C and 14±3 °C, which determine temperature ranges where some of the subjects started to leave the neutral zone and appeared to begin sweating or shivering). The originally chosen range of temperatures 8, 11, 14, 17, 20, 23, 26, 29, 32°C was found not to be necessary and thus they was reduced.

Within the comfort range the relative humidity was maintained corresponding to a partial water vapour pressure from 700 to 1850 Pa. The beginning of sweating and shivering was always assessed by the same person. Experiments were carried out during all seasons, thus reflecting the seasonal adaptation effect on maximal and minimal thermoregulatory heat, i.e. it was possible to determine adaptational heat. But it was evident that it can be neglected (Jokl, Moos 1992) being lower as 0.2 °C (it is within the range of experimental faults of measuring the temperatures). The same finding has been described by other authors (see Fanger 1970). The results were only accepted from subjects within the thermal neutral zone with the thermoregulatory heat constant.

The graph construction of $T_g = f(q_m)$

The measured values are plotted as $q_{dry} = f(q_i - q_{sw})$ in *Fig.2*, where for optimal valuesthe linear equation - $q_{dry} = q_i - q_{sw}$ [$W.m^{-2}$] representing equilibrium is valid. The application of this graph into the practice is very difficult; useful is the relationship $T_g = f(q_m)$. Therefore the linear relationship from *Fig.2* has been transferred into the graph in *Fig.3* by plotting a regression line through points limited by equation – $q_{dry} - (q_i - q_{sw}) = \pm 4.8$ [$W.m^{-2}$] in *Fig.3*. The value of ±4.8 $W.m^{-2}$ of the regression line is the minimal thermoregulatory heat, i.e. represents maximal vasoconstriction in human body and can be estimated from the minimum value of human body internal thermal conductivity (see Fig.4) which equals 9.07 W/m^2K (for core body temperature T_i= 36.6 °C, skin temperature T_{sk}= 30.5 °C and q_m= 45.7 W/m^2).

$$q_{tr,min} = G_{t,i,min}(T_i - T_{sk}) = 1.57(36.6 - 30.5) = 9.6\ Wm^{-2},\ tj. \pm 4.8$$

$$\text{where}\quad G_{t,i,min} = G_{t,ti,min} - G_{t,m,min} = 9.07 - 7.5 = 1.57\ Wm^{-2}K$$

$$G_{t,m,min} = \frac{q_m}{T_i - T_{sk}} = \frac{45.7}{36.6 - 30.5} = 7.5\ Wm^{-2}K$$

The estimation of thermoregulatory range

The widest thermoregulatory range, i.e. from optimum up to the beginning of visible sweating can be estimated by plotting the regression line into the points of beginning of sweating. But for comfort lower values are necessary, without visible sweating occuring. This area is between the line of optimum and the tangent from origin (being an intersection of the line of optimum and the regression line of beginning of sweating) to the area of beginning of shivering (see *Fig.5*). These tangents are analogous to the thermoregulatory range of C category according to CR 1752-1998.

For A and B categories account must be taken that the human is a thermoregulatory mechanism in the surrounding environment balancing the operative temperature changes by thermoregulatory heat flows in the human body so that an equilibrium can be reached, this at three levels (analogically to technological mechanisms) (see *Fig.6*):

- level A, corresponding to the time constant 0.368 $\Delta T_{o,tr,max}$
- level B, corresponding to the time constant 0.632 $\Delta T_{o,tr,max}$
- level C, corresponding to the time constant 1.000 $\Delta T_{o,tr,max}$

Level A is valid for the building interiors with the highest requirements and can only be attained by air conditioning systems application. The level C is valid for the building interiors with the lowest requirements, usually only naturally ventilated. Level B covers the remaining buildings where air conditioning is necessary only in some cases.
The time constant according to control theory characterizes the system response, (human organism response) to the operative temperature changes and equals to the product of system thermal resistance R and its thermal capacity C:

time constant = R.C
where R= $R_{ti} + R_{twa}$ [$W^{-1}.m^2.K$]

For a complete list of values see *Table 1*.

A COMPARISON OF OPTIMAL VALUES AND THERMOREGUATORY RANGES WITH ACCEPTED VALUES

The proposed optimal temperatures and their thermoregulatory ranges were compared with the values according to ISO 7730 (Moderate thermal environments ISO 7730-1984 (E)), CR (1752) (1998) and ISO/DIS 7730 (2003), ANSI/ASHRAE Standard 55-2004.

The comparison of the above proposed operative temperatures with the values by CR 1752 and ISO/DIS 7730 is presented on *Table 2*. There is agreement of operative temperatures for 50 W/m^2, 70 W/m^2 and 80 W/m^2. For higher activities the values differ: the higher the activity, the higher the operative temperature difference. There is agreement with experiments (sitting persons in the neutral zone) on which PMV value is based.

The comparison of the proposed optimal operative temperatures with the values according to ISO and ANSI/ASHRAE is presented in *Table 3*. There is an evident agreement for low activities (graph is also based on ISO 7730).

RESULTS

From the mathematical model (*Fig.1*) the role of various heat flows produced by the human body as it interacts with the environment is evident. All the heat flows must be in mutual equilibrium the human body to stay homoiotherm. The application of this graph in practice is very difficult and T_g=f(q_m) is a more useful relationship (*Fig.3*). This equilibrium is the basis for optimal operative temperature estimation (*Fig.2*). The experimental data on the beginning of sweating and the beginning of shivering enable the thermoregulatory ranges to be estimated (*Fig.5*). The thermoregulatory area is between the line of optimum and the tangent from pole, defined as the intersection of the line of optimum and the regression line of beginning of sweating, to the field of beginnings of sweating (upper limit, level C_{max}) and to the field of beginning of shivering (lower limit, C_{min}). It is interesting and in agreement with human feelings, that the thermoregulatory field for cold is smaller than the thermoregulatory field of the warm area – human body is more sensitive to temperature decrease in cold area. The question is how to sub-divide the thermoregulatory range into categories (A, B and C). Instead of qualified assumption, it is proposed to base the categories on control theory. The human body does behave as any other system to which control theory can be applied. It is proposed that human body time constant is used to differentiate the categories. These values were used: time constant $0.368\Delta T_{0,tr,max}$ (A), $0.632\ \Delta T_{0,tr,max}$ (B) and $1.0\ \Delta T_{0,tr,max}$, which correspond to categories A, B and C (*Fig.3*). Category A can be applied to air conditioned buildings and category C to natural ventilated buildings. As a result, two previously separate methods of assessment can be merged, those for air conditioned buildings, based on PMV, and for natural ventilated buildings, based on a mean monthly outdoor temperature. The results have been compared with the values according to ISO 7730, CR (1752) (1998), ISO/DIS 7730 (2003) (*Table 2* and *3*) and with ANSI/ASHRAE Standard (*Table 3*). But, most importantly, it was possible to base the new Czech Government Directive No. 523/2002 Code (*Table 3*) on the new findings, which were used to derive the compulsory microclimatic condition for workplaces in the Czech Republic.

DISCUSSION

The optimal operative temperatures derived from PMV values (from the 1970's) are now not fully acceptable. It is more precise to use optimal operative temperatures based on the physiological human body response and not based on the man's feelings only. It has been proved by experimental works (Fishman, Pimbert 1979) and shown when ISO values have been applied in practice. The higher the man's is activity, the higher the discrepancy in the optimal temperature. Because of this discrepancy, the new Czech Government Directive No. 523/2002 Code is based on the presented values and not on ISO/DIS 7730, which is based on PMV. The absence of adaptation to heat and cold, for example resulting from a stay in heated rooms in winter and in air-conditioned cars in summer, in the directive results in the same optimal operative temperatures for winter and for summer; temperatures are differentiated only by various clothing.

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Table 1. Optimal operative temperatures and thermoregulatory range as a function of man's activity q_m.

°C	50	70	80	100	120	150	180
sweat	**32.2**	**31.4**	**31.1**	**30.3**	**29.6**	**28.5**	**27.4**
sweat -opt	5.6	6.8	7.5	8.8	10.1	12	14
sweat-opt (0.5)	5.5	6.5	7	8.5	10	12	13.5
Max	**29.3**	**27.9**	**27.2**	**25.8**	**24.4**	**22.3**	**20.2**
max C – opt	2.7	3.4	3.7	4.3	4.9	5.9	6.8
max C - opt (0.5)	2.5	3	3.5	4	4.5	5.5	6.5
max B (0.632)	28.3	26.7	25.9	24.3	22.6	20.2	17.7
max B – opt	1.7	2.1	2.3	2.7	3.1	3.7	4.3
max B - opt (0.5)	1.5	2	2	2.5	3	3.5	4
max A (0.368)	27.6	25.8	24.9	23.1	21.3	18.6	15.9
max A - opt	1.0	1.2	1.4	1.6	1.8	2.2	2.5
max A -opt (0.5)	1	1	1	1.5	1.5	2	2.5
opt	**26.6**	**24.6**	**23.6**	**21.5**	**19.5**	**16.5**	**13.4**
min A(0.368)	25.9	23.7	22.6	20.4	18.2	14.9	11.6
min A - opt	-0.7	-0.9	-1	-1.1	-1.3	-1.5	-1.8
min A - opt (0.5)	-0.5	-0.5	-0.5	-1	-1	-1.5	-1.5
min B (0.632)	25.4	23.1	21.9	19.6	17.3	13.8	10.4
min B- opt	-1.2	-1.5	-1.6	-1.9	-2.2	-2.6	-3.1
min B - opt (0.5)	-1	-1	-1.5	-1.5	-2	-2.5	-3
min	**24.7**	**22.2**	**21**	**18.5**	**16**	**12.3**	**8.6**
min C - opt	-1.9	-2.4	-2.6	-3	-3.5	-4.2	-4.8
min C - opt (0.5)	-1.5	-2	-2.5	-3	-3	-4.0	-4.5

Table 2. The comparison of experimentally found operative temperatures with the values in CR 1752 and ISO/DIS 7730 (2003) in categories A, B, C (0.5 clo, 1.2 met).

Category	**CR + ISO/DIS**	**Jo/Ka**
A (air conditioning)	24.5 ± 1.0	24.6 + 1.0 – 0.5
B (air conditioning and natural ventilation)	24.5 ± 1.5	24.6 + 2.0 – 1.0
C (natural ventilation)	24.5 ± 2.5	24.6 + 3.0 – 2.0

Table 3. The comparison of experimentally found operative temperatures with the values in ISO and ANSI/ASHRAE (clothing 0.5).

q_m [W/m²]	50	70	80	100	120	150	180
[met]	0.86	1.20	1.38	1.72	2.07	2.59	3.10
Jo/Ka [°C] (B)	26.6 +1.5/-1.0	24.6 +2.0/-1.0	23.6 +2.0/-1.5	21.5 +2.5/-1.5	19.5 +3.0/-2.0	16.5 +3.5/-2.5	13.4 +4.0/-3.0
ISO 7730 (1984)	26.6±1.5	24.5±1.5	23.6±2.0	22.3±2.0	20.6±2.5	18.5±2.5	16.4±2.5
CR 1752, ISO/DIS 7730 (2003) [°C] (B)	-	24.5±1.5	23.5±2.0	-	-	-	-
ANSI/ASHRAE 55 (1992) $T_{o\ active} = T_{o\ sedentary} - 4.5\ (met - 1.2)$ [°C]	-	24.5	23.7	22.2	20.6	18.2	16.0

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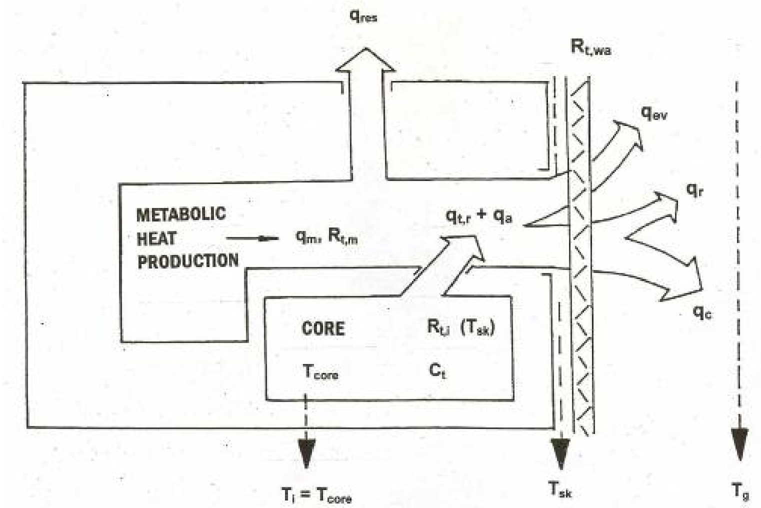

Figure 1. Total heat rate production and its distribution in individual components during heat exchange between the human body and the environment (q_m metabolic heat, q_{res} respiration heat, q_{tr} thermoregulatory heat, q_e evaporative heat, q_c convective heat, q_r radiant heat, $R_{t,wa}$ total thermal resistance of clothing, R_t total internal thermal body resistance, C_t thermal body capacity, T_i deep body temperature, T_{core} core body temperature, T_{sk} skin temperature, T_g globe temperature.

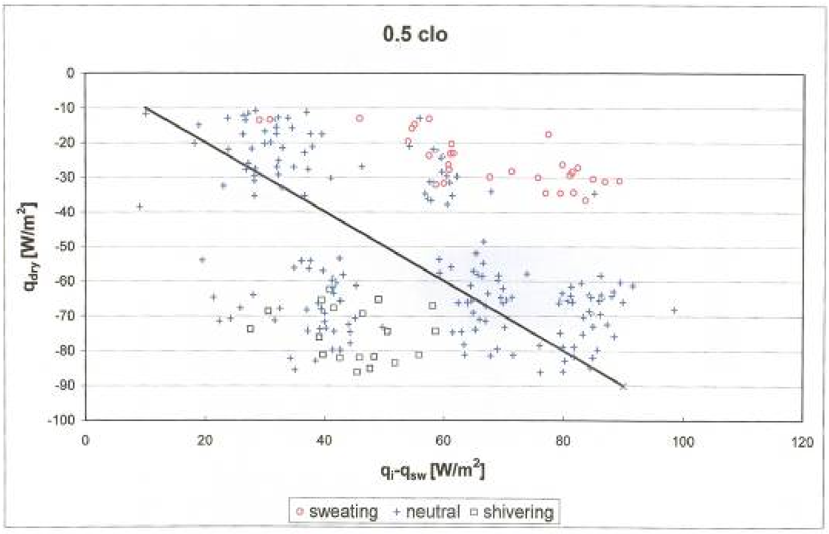

Figure 2. The graph of the relationship $q_{dry} = f(q_i - q_{sw})$ for clothing 0.5 clo, points from experiment. Optimal values are on the line - $q_{dry} = q_i - q_{sw}$.

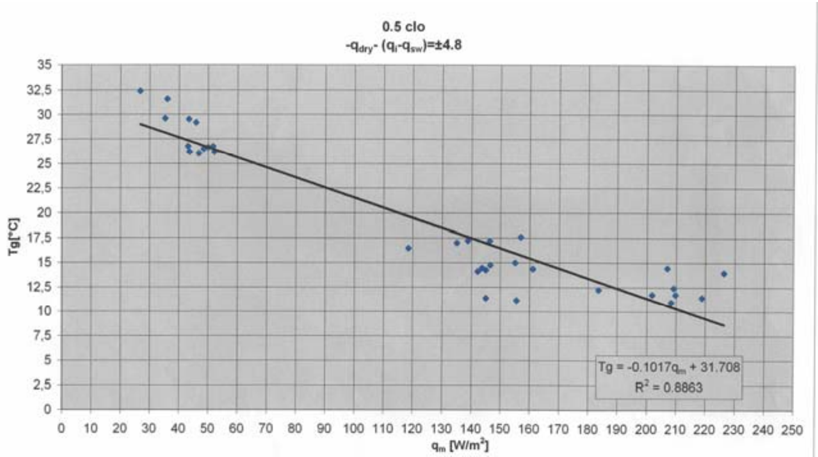

Figure 3. The graph of the relationship $T_g= f(q_m)$ for optimal values within the range $-q_{dry}-(q_i - q_{sw})= \pm4.8$, where the value ±4.8 W/m^2 represents the minimal thermoregulatory heat.

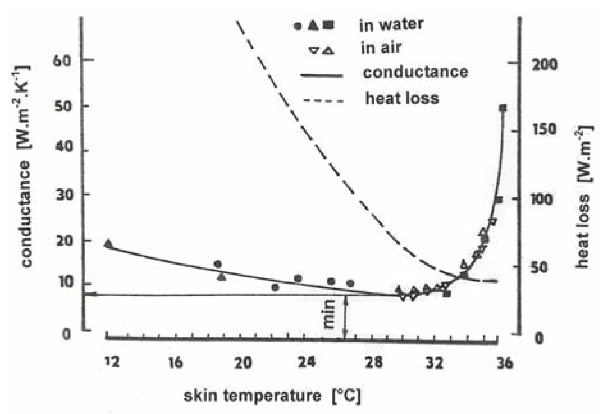

Figure 4. Human body thermal conductance $G_{t,ti}$ and human body heat loss as a function of skin temperature T_{sk} for resting subject during day (q_m=45.6 – 57.4 W.m^{-2}) (Burton, Bazett 1936, Du Bois et al. 1952, Lefevre 1898, Liebmaster 1869) (from Itoh et al. 1972).

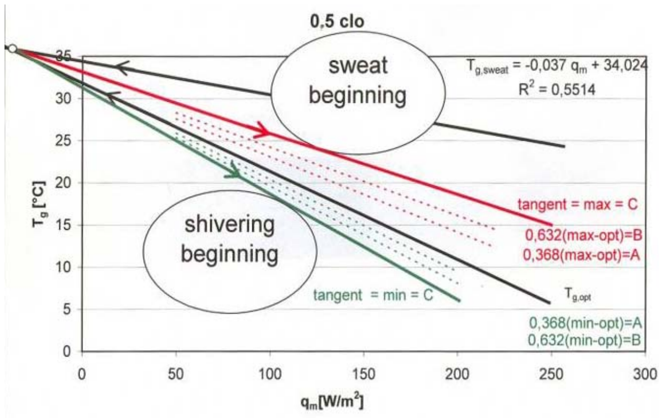

Figure 5. The way how to get the thermoregulatory ranges. See text for explanation.

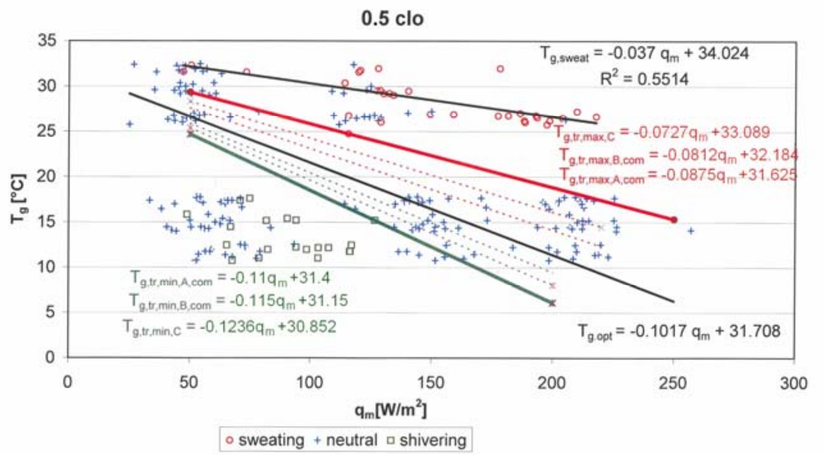

Figure 6. The graph of the relationship $T_g= f(q_m)$ with the regression line of beginnings of sweating and the thermoregulatory range for levels (categories) A, B, C for warm (towards beginnings of sweating) and for cold (towards beginnings of shivering).

Modeling human thermal comfort

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SUMMARY

Thermal comfort standards determine indoor conditions in buildings as well as the energy consumption for heating and cooling purposes. Existing thermal comfort standards are based on steady-state thermal conditions, and according to recent research these standards can not describe thermal comfort accurately enough with transient boundary conditions. In this paper, a method based on Hui (2003) is presented to calculate local and overall human body thermal sensations in time dependent and non-uniform. In addition, local and overall thermal comfort predictions can be performed based on these thermal sensation values.

INTRODUCTION

Due to increasing awareness of energy consumption and environmental issues, modern buildings have better thermal insulation and ventilation heat recoveries. This lead also to other kind of demands in design phase, especially when low temperature levels are utilised. So far most human thermal comfort models are based on estimates assuming steady-state conditions. However, this often leads to underestimations of local cold or hot surfaces. In addition, that kind of models do not take into account variable conditions.

It is important for people that they feel comfortable with the environment when they are inside buildings. This is regardless of their role (employer, employee, resident) in a specific environment. Therefore, a large amount of research has been conducted in order to establish how the variables creating the environment in a building should be tuned so that the users feel comfortable.

Two widely used standards - ANSI/ASHRAE Standard 55-1992 (Thermal Environmental Conditions for Human Occupancy) [1] and ISO 7730 (Moderate thermal environments - Determination of the PMV and PPD indices and specification of the conditions for thermal comfort) [2] - present a necessary method (originally presented by Fanger in 1972 [3]) for evaluation of moderate thermal environment. When using this method, human's thermal sensation is related to the thermal balance of a body as a whole. This balance is influenced by occupant's physical activity and clothing, as well as the environmental parameters: air temperature, mean radiant temperature, air velocity, and air humidity (ISO 7730). The method is derived for steady-state conditions and - due to treating a human body as whole - is does not allow estimations in any spatially non-uniform conditions.

The aim of this paper is to present a method how to predict human thermal sensation and comfort under transient and non-uniform boundary conditions.

METHODS

The transient and non-uniform thermal environment as well as our body including our clothing are affecting to our thermal sensation and thus our comfort. Since neither the physiology nor the thermal comfort are not uniform to the whole human body, a detailed model is needed in order to estimate realistic thermal sensation. The model used here for estimating thermal sensation and comfort is based on Hui [4]. Basically, the model calculates local human body part's thermal sensations and local thermal comfort, and based on that information also the overall thermal comfort can be estimated.

Human Body Model

For the estimation of thermal sensation and comfort, a human body model interacting with thermal environment is needed. The body model used here is based on Smith's model [5]. In this model the human body is divided to 15 parts, and each body part has bone, muscle, fat and skin layers as well as blood circulation. The blood circulation and body core temperature is controlled by a human body control model which acts rather close as the real body "control systems". For example, when human body core temperature rises above its neutral value, vasodilation occurs and cardial output increases dramatically. Nearly 100% of this increase goes to the skin tissue. For this development, a state of maximum vasodilation is achieved when core temperature reaches 37.2°C. At this state, the total skin blood flow rate may be as much as seven times its basal value. This increase in cardiac output is distributed to the individual body parts according to surface area.[5] A more detailed description of that model is presented by another paper by Tuomaala et al. in this congress.

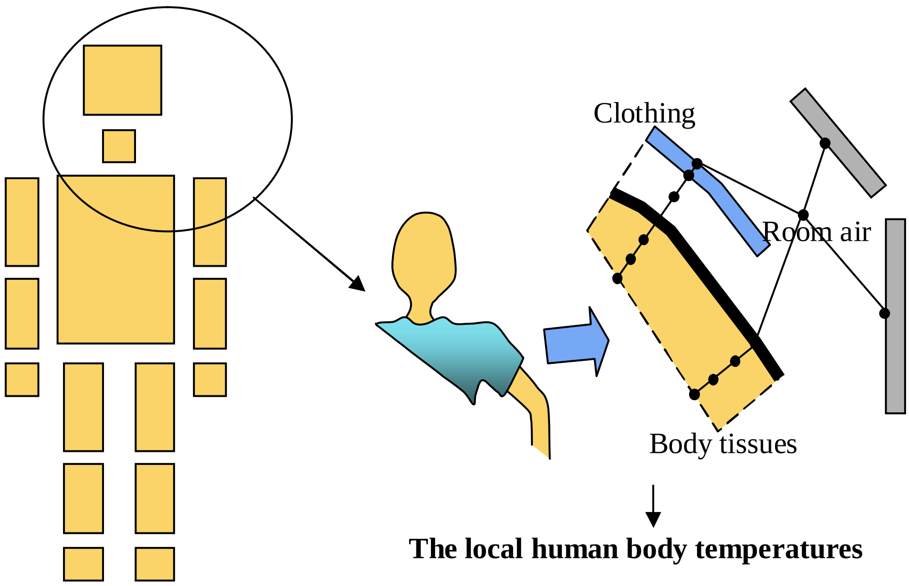

Figure 1. The human model is divided to 15 parts.

Thermal sensation and comfort prediction model

The model used in this study is based on Hui's study [4] which includes results from 109 human subject tests that were performed under non-uniform and transients conditions in the UC Berkeley Controlled Environmental Chamber. In those experiments, local body surfaces of the subjects were independently heated or cooled while the rest of the body was exposed to a warm, neutral or cool environment. Skin temperatures, core temperature, thermal sensation and comfort responses were collected at one- to three-minute intervals. [4] The Figure 1 shows the flow chart how the overall thermal comfort is calculated.

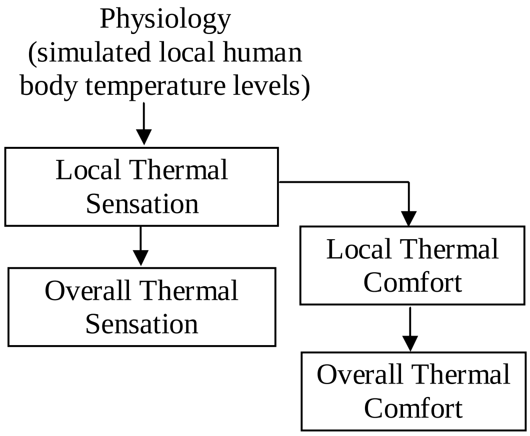

Figure 1. Overall thermal comfort calculation flow chart.

The local sensation model is a function of skin and core temperature and their rates of change (Eq. 1). The model has a sub-model to each body part, and together they capture the asymmetry of thermal conditions.

$$S_l = f\left(T_{skin}, \frac{dT_{skin}}{dt}, T_{core}, \frac{dT_{core}}{dt}\right) \quad (1)$$

Local comfort is predicted from local sensations and average of all body's local sensations, and the overall sensation model integrates the local sensations. The whole body comfort model integrates the local comfort values. Overall thermal sensation is modelled as a weighted average from the local sensations:

$$S_o = \frac{\Sigma weight_i \cdot S_{l,i}}{\Sigma weight_i} \quad (2)$$

where *weight* is a function between the difference of local sensation and geometrical average of all sensations.

weight can be calculated for each body part:

$$weight_i = a \cdot (S_{l,i} - \overline{S_l}) \quad (3)$$

where *a* is a factor depending on local and overall sensation, $\overline{S_l}$ average value from all local thermal sensation factors.

For some body parts (e.g., chest and back) the weight is higher than for other parts. The weight is higher either due to body part size or sensitivity. Some body parts are not behaving similarly if the local sensation is higher or lower compared to average thermal sensation. One body part may be more important to determine cold than warm sensation. As the *weight* is a function between the difference of local and geometrical average of all sensations it assigns larger weights when the local thermal sensation is opposite to rest of the body's thermal sensation.

Sensation can have values from -4 to 4. Value 4 corresponds very hot, 3 hot, 2 warm, 1 slightly warm and 0 neutral.

Local comfort is a function of local sensation and overall sensation:

$$C_{li} = f(S_{l,i}, S_o) \quad (4)$$

and overall thermal sensation is a function of local thermal comfort. Overall thermal comfort is evaluated from two rules:

•Rule 1: Overall comfort is the average of the two minimum local comfort votes unless Rule 2 applies.

•Rule 2: If the following criteria are met the overall thermal comfort is average of the two minimum and the maximum comfort vote.
Criteria are:
•the second lowest local comfort vote is higher than -2.5
•the person has some control over his/her thermal environment or the thermal conditions are transient

If both hands or both feet comprise the two most uncomfortable body parts ignore the second lowest hand or foot comfort value, and use the third lowest comfort vote as the second lowest vote in both Rules 1 and 2.

Comfort can have values from -4 to 4 and value 4 corresponds to very comfortable, 2 comfortable and 0 neutral.

RESULTS

When the body surface temperatures and core temperature are close to neutral conditions, local thermal sensations are close to zero, thus no big thermal sensations. The local and overall comfort values are close to 2, indicating comfortable thermal conditions, Figure 2. It should be noted that according to [4] the very comfortable conditions (value 4) was only recorded in transient conditions when thermal stress is suddenly removed.

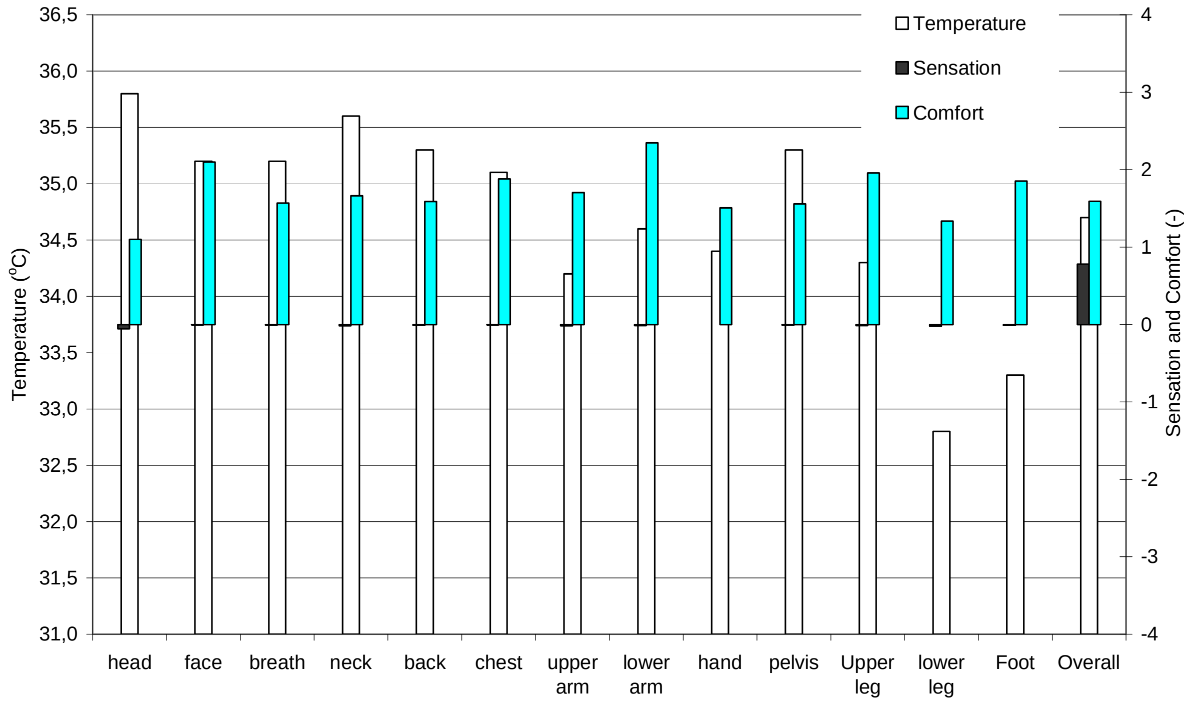

Figure 2 Local sensations and comfort values close to thermal neutral area.

When body temperatures are decreasing in cold environments, the sensation and comfort values are negative. If body part temperatures are decreasing by 4 degree the local thermal comfort is lower than -2 indicating uncomfortable cold, Figure 3. In calculations for Figure 3 it was assumed that the core temperature is not changed.

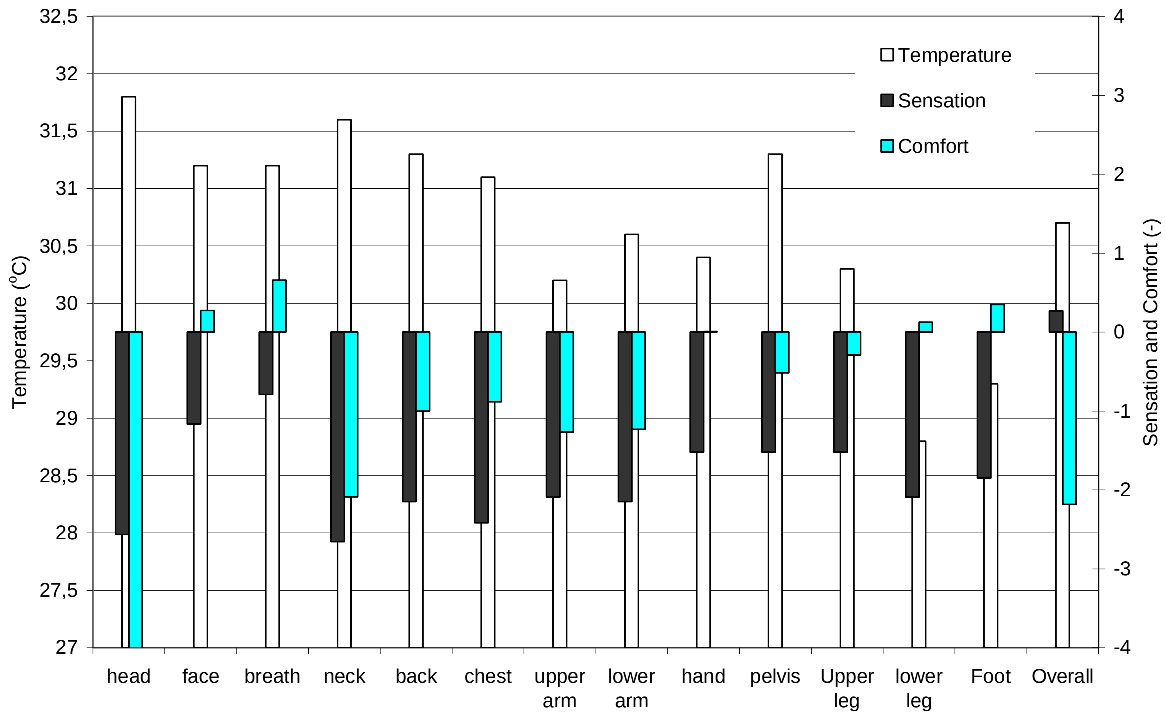

Figure 3 Local sensations and comfort values when body surface temperatures are in average 4 degrees lower than thermal neutral area. Core temperature was not changed.

Head of human body is sensitive to temperature changes. If all other body parts are kept in thermal neutral area, but head, face, breath and neck temperature is decreasing 2^{o}C lower than thermal neutral temperatures it is affecting overall comfort dramatically, Figure 4.

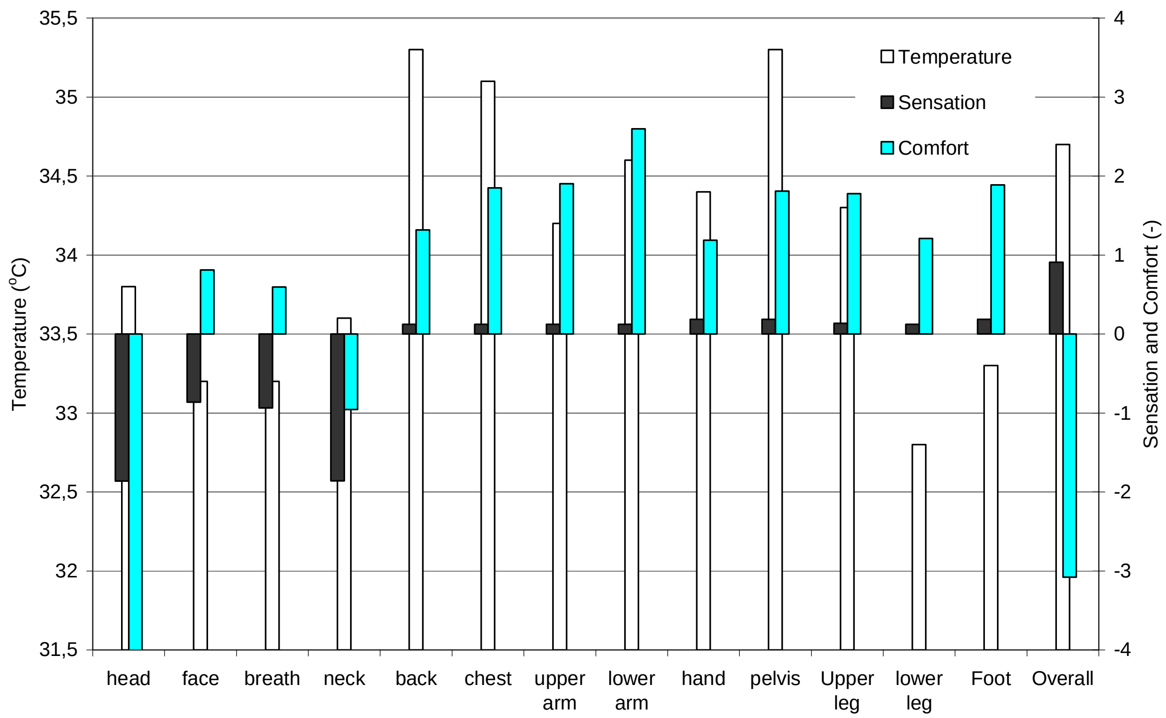

Figure 4 Local sensations and comfort values when head are is 2 degrees Celsius lower than the thermal neutral area, other body parts remained close to thermal neutral conditions. Core temperature was not changed.

DISCUSSION

The model based on Hui's study [4] gives new understanding about the thermal comfort and how changes in thermal environment are effecting on human thermal sensation. The new knowledge is valuable in designing and developing new indoor environments. It would also help us to find out the most suitable renovation alternatives for old buildings in respect to indoor environment comfort.

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A concept for utilizing detailed human thermal model for evaluation of thermal comfort

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SUMMARY

This paper presents a concept for more realistic and reliable evaluation of thermal indoor environment. A detailed thermal model of a human body, interacting with building simulation model, makes it possible to predict transient tissue temperatures of a human body as well as local skin temperature levels in realistic working and living conditions. Using these temperature distributions of a human body as input data for a new thermal comfort prediction model (Hui 2003), both local and overall estimations of thermal comfort of a human body can be obtained. In the future, these estimations of thermal comfort can be used when analysing links between alternative technical solutions vs. productivity and health issues.

INTRODUCTION

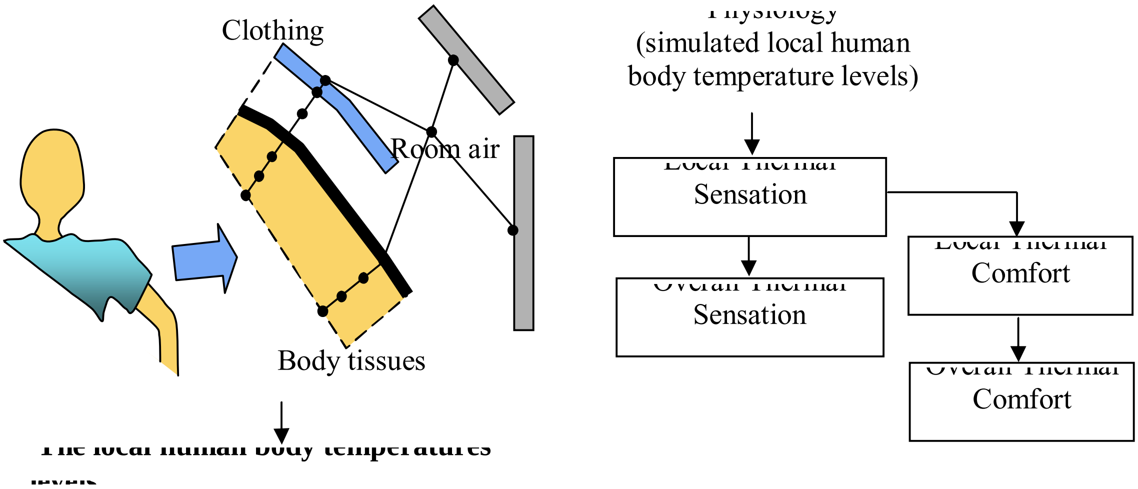
Two widely used standards - ANSI/ASHRAE Standard 55-1992 (Thermal Environmental Conditions for Human Occupancy) [1] and ISO 7730 (Moderate thermal environments - Determination of the PMV and PPD indices and specification of the conditions for thermal comfort) [2] - present a necessary method (originally presented by Fanger in 1972 [3]) for evaluation of moderate thermal environment. When using this method, human's thermal sensation is related to the thermal balance of a body as a whole. This balance is influenced by occupant's physical activity and clothing, as well as the environmental parameters: air temperature, mean radiant temperature, air velocity, and air humidity (ISO 7730). The method is derived for steady-state conditions and - due to treating a human body as whole - is does not allow estimations in any spatially non-uniform conditions.

There is a very old and simple experiment described by English philosopher John Locke in his 1690 *Essay Concerning Human Understanding*. A person places one hand in a basin of warm water and the other in a basin of cool water. After a short time, both hands are placed together in a third basin of water, which is at intermediate temperature. The hand previously in warm water feels cool and the hand previously in cool water feels warm even though they are actually at the same temperature. In a similar way, people are probably more often exposed to spatially non-uniform and transient temperatures than to thermal environments that are uniform and stabile. We experience transient and non-uniform temperature conditions when moving between spaces - e.g., from indoors to outdoors, from shun to shade, and when occupying spaces with widely varying temperatures. [4]

The aim of this paper is to present a concept how to predict and evaluate human thermal sensation and comfort under transient and non-uniform boundary conditions. The concept includes guide-lines for predicting transient thermal behavior of a detailed human body model, and an estimation method for quantifying thermal sensation and comfort of a human body.

METHODS

Since we experience transient and non-uniform temperature conditions in our real living and working environment, both human body and surrounding indoor environment need to be modeled. In addition, both human body and indoor environment models need to be detailed enough for estimating transient thermal behavior of individual body parts and local skin temperature levels. After obtaining estimations of local skin temperature levels, as well as body core and local skin temperature change rates, a detailed prediction of both local (body part) and overall (whole body) thermal sensation and comfort can be conducted.



(i) (ii)

Figure 1. The two main phases when predicting human thermal comfort: (i) simulation of the local human body temperature levels, and (ii) quantifying local and overall thermal sensation and comfort by Hui's method [4].

Human Body Model

In the first phase of this evaluation concept, the human thermal simulation model - having a true physical inter-action with a surrounding space model - estimates human body tissue and skin temperature levels. The human thermoregulatory system senses the thermal state of the body, interprets these thermal signals, and responds accordingly. Because the body's thermoregulatory response is determined using feedback from the thermoreceptors, it is appropriate to model the human thermal system using two interactive systems: (i) the passive or controlled system, which includes the body tissues, internal organs, circulatory and respiratory systems, and (ii) the control system, which initiates and controls the physiological responses of the body. [5]

<u>Human Body Model - Passive system</u>

A model of the human thermal system ought to be sufficiently detailed and flexible enough to be easily modified for different types of applications. For this reason, a finite-difference method with free description of the thermal calculation network generation will be used to develop the new thermal model. When using this technique in the passive system defined above, the human body is discretized into a number of smaller nodes (with thermal mass) and inter-nodal connections (thermal conductance). This network represents tissue sections, blood vessels, segment of the respiratory track, or portion of an internal organ. Energy, momentum, and mass balance equations of the human thermal model calculation network, incorporating the environmental conditions, can then be solved for the unknown variable of interest (e.g., variations of the skin temperatures around the body). The control system uses thermal input in the form of local tissue temperatures to determine the appropriate thermoregulatory responses. Possible responses include sweating, shivering, and the increase or decrease in blood flow rates to the skin. How the thermal state of the passive system changes, and to what degree, depends on the thermoregulatory responses initiated by the control system. A state of thermoneutrality exists when core and mean skin temperatures are 36.8°C and 33.7°C, respectively. [5]

Because of simplicity, the individual 15 human body parts (see Figure 2) are approximated by concentric rectangulars describing the human body tissue type distribution presented in Table 1. The basic idea is to model the body parts mimicking the true anatomy as a single bone core surrounded by muscle, fat, and skin layers. Further divisions are then made in the angular and axial directions until each body part is composed of small tissue elements.

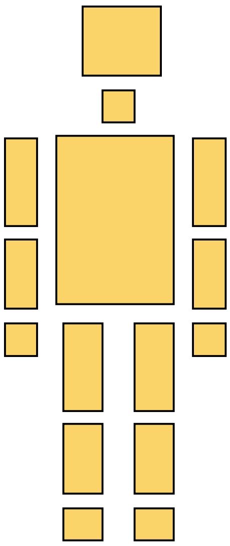


Figure 2. Adopted multi-element thermal model of human body divided into body parts.

Human Body Model - Control System

As core temperature rises above its neutral value, vasodilation occurs and cardial output increases dramatically. Nearly 100% of this increase goes to the skin tissue. For this development, a state of maximum vasodilation is achieved when core temperature reaches 37.2°C. At this state, the total skin blood flow rate may be as much as seven times its basal value. This increase in cardiac output is distributed to the individual body parts according to surface area. [5]

Table 1. Tissue distribution [5].

Body part	Tissue type	Tissue mass [kg]
Head	Brain	1.3975
	Bone	1.7770
	Muscle	0.4519
	Fat	0.2823
	Skin	0.1874
Neck	Bone	0.2330
	Muscle	0.5811
	Fat	0.0710
	Skin	0.031
Torso	Abdomen	11.1444
	Lung	2.9188
	Bone	4.1688
	Muscle	11.9822
	Fat	9.0135
	Skin	1.2305
Right or left upper arm	Bone	0.4730
	Muscle	1.5798
	Fat	0.2400
	Skin	0.1808
Right or left thigh	Bone	1.4816
	Muscle	2.5386
	Fat	1.1342
	Skin	0.3414
Right or left forearm	Bone	0.2661
	Muscle	0.8825
	Fat	0.1343
	Skin	0.1010
Right or left calf	Bone	0.6550
	Muscle	1.1196
	Fat	0.5080
	Skin	0.1527
Right or left hand	Bone	0.1641
	Muscle	0.1407
	Fat	0.1493
	Skin	0.0925
Right or left foot	Bone	0.4120
	Muscle	0.2149
	Fat	0.3468
	Skin	0.1265

As mean skin temperature falls below its neutral value, vasoconstriction occurs. Skin blood flow, and therefore, cardiac output, decreases. At a state of maximum vasoconstriction, assumed to occur when mean skin temperature falls to 10.7°C, the total skin blood flow rate may be as low as one eight of its basal value. As before, this change in cardiac output is distributed among the body parts according to surface area. [5]

Regional blood flow rates and skin blood vessel radii have been calculated for a hypothetical state of thermoneutrality and for states of maximum vasodilation and maximum vasoconstriction. This establishes the range over which skin blood vessel radii can vary, with respect to the model. Core temperatures above 36.8°C govern the effects of vasodilation while mean skin temperatures below 33.7°C govern those of vasoconstriction.

Control equations for skin blood vessel radii have been developed as functions of core and mean skin temperatures. For core temperatures less than or equal to 36.8°C, skin blood vessel radii experience no dilation and assume their values at thermoneutrality. At a core temperature of 37.2°C, skin blood vessels are maximally dilated. Between these two core temperatures, skin blood vessel radii vary linearly with core temperature.

Thermal sensation and comfort prediction model

Most existing thermal comfort models are applicable only in uniform, steady-state thermal environments. However, Hui (2003) has developed predictive models of local and overall thermal sensation and comfort derived from results from 109 human subject tests that were performed under non-uniform and transients conditions in the UC Berkeley Controlled Environmental Chamber. In these tests, local body surfaces of the subjects were independently heated or cooled while the rest of the body was exposed to a warm, neutral or cool environment. Skin temperatures, core temperature, thermal sensation and comfort responses were collected at one- to three-minute intervals. [4]

The local sensation model is a function of skin and core temperature and their rates of change (see Equation 1). The temperatures of skin and core represent stable conditions and the rate of change transient conditions respectively. The model has a sub-model to each part and together they capture the asymmetry of thermal conditions.

$$S_l = f\left(T_{skin}, \frac{dT_{skin}}{dt}, T_{core}, \frac{dT_{core}}{dt}\right) \quad (1)$$

Local comfort is predicted from local sensations and average of all body's local sensations, the over all sensation model integrates the local sensations. The whole body comfort model integrates the local comfort values. Overall thermal sensation is modelled as a weighted average from the local sensations:

$$S_o = \frac{\Sigma weight_i \cdot S_{l,i}}{\Sigma weight_i} \quad (2)$$

where *weight* is a function between the difference of local sensation and geometrical average of all sensations.

weight can be calculated for each body part:

$$weight_i = a \cdot (S_{l,i} - \overline{S_l}) \qquad (3)$$

where a is a factor, $\overline{S_l}$ average value from all local thermal sensation factors. The values for a are written in Table 2.

Table 2. Coefficient a for calculating overall thermal sensation. [4]

Body Part	S_l-S_o< 0	R^2	S_l-$S_o \geq 0$	R^2
Head	-0.13	0.82	0.21	0.57
Neck	-0.13	0.71	0.23	0.68
Chest	-0.23	0.90	0.23	0.70
Back	-0.23	0.81	0.24	0.80
Pelvis	-0.17	0.85	0.15	0.73
Upper arm	-0.10	0.69	0.14	0.42
Lower arm	-0.10	0.69	0.14	0.42
Hand	-0.04	0.29	0.04	0.23
Thigh	-0.13	0.38	0.26	0.30
Calf	-0.13	0.38	0.26	0.30
Foot	-0.09	0.10	0.14 (1 foot)	0.20
Upper back	-0.16	0.86	0.26	0.88
Lower back	-0.21	0.84	0.35	0.88

For some body parts e.g. chest and back the weight is higher than for other parts. The *weight* is higher either due to body part size or sensitivity. Some body parts are not behaving similarly if the local sensation is higher or lower compared to average thermal sensation. One body part may be more important to determine cold than warm sensation. As the *weight* is a function between the difference of local and geometrical average of all sensations it assigns larger weights when the local thermal sensation is opposite to rest of the body's thermal sensation.

Thus, the local thermal sensation can be presented as:

$$S_{l,i} = 4\left(\frac{2}{1+e^{-C1_i (T_{i,skin,local} - T_{i,skin,local,set}) - K1_i \left[(T_{i,skin,local} - T_{i,skin,local,set}) - (\overline{T}_{skin} - \overline{T}_{skin,set})\right]}} - 1\right)$$
$$+ C2_i \frac{dT_{i,skin,local}}{dt} + C3_i \frac{dT_{core}}{dt} \qquad (4)$$

The coefficients $C2_i$ and $C3_i$ are presented in Table 3.

Table 3. Coefficients $C2_i$ and $C3_i$ for calculating local dynamic thermal sensation. [4]

Body part	$C2_i$ $\left(\frac{dT_{i,skin,local}}{dt} \le 0\right)$	$C2_i$ $\left(\frac{dT_{i,skin,local}}{dt} \ge 0\right)$	$C3_i$	R^2
Back	88	192	-4054	0.73
Chest	39	136	-2135	0.61
Face	37	105	-2289	0.74
Hand	19	46	0	0.90
Foot	109	162	0	0.55
Neck	173	217	0	0.80
Breath (cheek)	68	471	0	0.92
Head	543	90	0	0.64
Pelvis	75	137	-5053	0.86
Lower arm	144	125	0	0.77
Upper arm	156	167	0	0.74
Lower leg	206	212	0	0.85
Upper leg	151	263	0	0.94

RESULTS

Transient tissue and skin temperatures can be simulated. Table 4 presents one distribution of local skin temperatures, and corresponding local thermal sensation values are also presented. Local thermal sensation can have values from -4 to 4, where value 4 corresponds to “very hot”, 3 to “hot”, 2 to “warm”, 1 to “slightly warm”, and 0 to “neutral”.

Table 4. A basic distribution of local skin temperatures with a human body core temperature of 37.4°C, and corresponding predicted local thermal sensations.

Body part	Skin temperature [°C]	Local thermal sensation [-]
Head	35.7	0.76
Neck	35.5	1.58
Torso	35.1	1.34
Right upper arm	34.1	1.64
Left upper arm	34.1	1.64
Right forearm	34.5	2.58
Left forearm	34.5	2.58
Right hand	34.3	0.99
Left hand	34.3	0.99
Pelvis	35.2	1.37
Right thigh	34.2	1.92
Left thigh	34.2	1.92
Right calf	32.7	1.39
Left calf	32.7	1.39
Right foot	33.2	2.23
Left foot	33.2	2.23

DISCUSSION

The presented human thermal model seems very promising when predicting tissue temperatures under transient and non-uniform boundary conditions. However, further development, testing and validation are still needed until this human body model can be utilized for producing reliable thermal data for thermal sensation and comfort prediction model under extreme boundary conditions.

The presented concept (i.e., linking interacting human body and building simulations together with thermal sensation and comfort prediction model) offers a new mechanism for evaluating interactions and connections for example between thermal properties of individual building components, design and dimensioning building service devices and systems, human thermal comfort and sensation, productivity, etc. This way, well being indoor environments can be improved in a controllable way.

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THE THERMAL ENVIRONMENT LEVEL ASSESMENT BASED ON HUMAN PERCEPTION

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SUMMARY

A new way of the thermal level environment assessment based on "operative temperature thermal level" and new units "decitherm" is introduced, which allow the feelings of man to be followed. Simultaneously the fact that the operative temperature decrease is felt the more unpleasantly the lower are temperatures is also taken into account. The assessment in decitherms also allows the comparison (based on numerical values comparison) with the noise level in decibels and odor level in deciodors and the whole environment level assessment (based on weighted constituent levels adding). Also syndrome SBS can be estimated, the effectiveness of heating and cooling appliances and even the condition of human organism threat by overwarming (hyperthermia) or undercooling (hypothermia).

INTRODUCTION

The paper presents a new way of the thermal environment level assessment being able to give a true picture of the thermal environment level as perceived by man. Thus this way respects also the fact that the operative temperatures decrease is perceived the more uncomfortable the temperatures are lover, e. g. the decrease by 3 °C from 10 °C to 7 °C is always more unpleasant than the decrease from 20 °C to 17 °C. The distribution of 15-day moving means of mortality against temperature is shown in Fig. 1. Simple correlation gave a coefficient of r= -0.64. Mortality expressed as a quadratic function of temperature ($Y= 17.69-1.149T+0.025T^2$) produced a multiple correlation coefficient of 0.72 and *t*-values of -14.0 and 12.2 for temperatures T and T^2, respectively. These appeared to be high enough to justify the preference for the polynomial model over linear. Minimum mortality was predicted by the quadratic equation at 23° C.

From Fig. 1 it appears that mortality is accelerated below a 'critical' mean temperature of about 18° C and especially those below 16° C. On the warm side, temperatures did not reach sufficiently high values to enable identification of particularly critical thresholds (see Auliciems and Skinner 1989). There is also a physiological background of perceiving cold and warm in different way: special sensors for cold and warm in the skin (see Fig. 2) and special evaluation centres in the brain (Fig. 3). New way originates from human body physiology where Weber-Fechner law is valid:

$$R=k.\log S \qquad (1)$$

where R… the human body response
S… the stimulus of the environment causing the response
k… the coefficient

PROPOSAL OF NEW EVALUATION SYSTEM

At the thermal condition of environment this law can be applied as

$$L_{th} = k_{th} \cdot \log\left(\frac{T}{T_{threshold}}\right) \quad [dTh] \qquad (2)$$

where L_{th}…operative temperature thermal level [decitherm], [dTh]
T…operative temperature [°C]
$T_{threshold}$… threshold operative temperature, in this case optimal operative temp. [°C]

Equation (2) corresponds with the formula for noise assessment with the so called acoustic pressure level

$$L_p = 20 \cdot \log\left(\frac{P}{P_0}\right) \quad [dB] \qquad (3)$$

Where the ratio of acoustic pressures forms the stimulus, P is the acoustic pressure within the space investigated, P_0 is the acoustic pressure perceived lower limit 20 μPa. The unit decibel (dB) has been used for acoustic pressure level, the unit decitherm (dTh) for operative thermal level.
The analogical equations also exist for odor constituent of the environment (Jokl 1998, 2000), the so called deciodors, based on carbon dioxid, so called decicarbdiox (dCd) and on total volatile organic compounds (TVOC), so called decitvoc (dTv).

CORRESPOND NEW UNITS TO HUMAN FEELINGS?

The equation (2) for thermal environment assessment seems to be reasonable, both from the human body physiology aspect and from the theory of similarity (Kline 1965, Kožešník 1983), however, its approval by an experiment is necessary, i.e. that decitherms describe the man's perception.
The perception of the thermal environment level can be described by ASHRAE scale (ANSI/ASHRAE 55-1992) (cold, cool, slightly cool, neutral, slightly warm, warm, hot). These should be proportional to thermal level, i.e. it must be proved that

$$L_{th} = k_1 \cdot (AV) \quad [dTh] \quad (4)$$

where AV… the values of ASHRAE scale
k_1… coefficient of proportionality

Of course, Weber-Fechner law (2) is valid simultaneously, so it must be proved that

$$AV = k_2 \cdot \log\left(\frac{T}{T_{opt}}\right) \qquad (5)$$

to be $L_{th} = k_1 AV$.

After substitution

$$L_{th} = k_1 \cdot AV = k_1 \cdot k_2 \cdot \log\left(\frac{T}{T_{opt}}\right) = k_{th} \cdot \log\left(\frac{T}{T_{opt}}\right) \quad [dTh] \qquad (2)$$

See fig. 4 where the results of experiments by Fishman and Pimbert (1979) are presented. There is the relationship between operative temperatures (axis x) and averages votes ASHRAE (axis y). Average values ASHRAE have been estimated in the range ±0.2°C for each temperature, e.g. for 20 °C: 19.8 °C to 20.2 °C. The average activity was 80 W/m^2. The feelings of 26 subjects were registered 8 times per day (between 9.30 and 16.30 hour) for the whole year, i.e. 54080 values were registered altogether. From the graph is evident that the ASHRAE and calculated PMV values differ (the lowest difference is near neutral physiological state for which the PMV was derived). The relationship (6) can be estimated for ASHRAE values from the graph (*Fig. 4*):

$$AV = 14.469 \log T - 19.172 = 14.469 \log T - 14.469 \log 21.14 = 14.469 \cdot \log\left(\frac{T}{21.14}\right) \qquad (6)$$

This can be rewritten in a general form

$$AV = k_2 \cdot \log\left(\frac{T}{T_{opt}}\right) \qquad (5)$$

and therefore it is valid
$L_{th} = k_1 AV$
and it is evident that decitherms correspond to human feelings and that operative temperature do not correspond to human feelings.

THE OPERATIVE TEMPERATURE THERMAL LEVEL ESTIMATION

Now thermal levels can be estimated for operative temperatures optimal and long-term and short-term tolerable.

Optimal operative temperatures

Optimal operative temperatures are determined by the neutral physiological state of man at his activity and clothing. The experimental estimation on this basis being very difficult (however in this case values from Jokl and Kabele 2006 experiments have been available) the optimal values were estimated by the votes of the subjects satisfied with the evaluated environment (or dissatisfied – in the range 10% to 30% related to the requested environment quality) (Fanger 1970, EN ISO 7730).
To optimal operative temperatures (analogically as by noise and odors) correspond dTh=0 because log1=0.

Long-term tolerable operative temperatures

Long-term tolerable operative temperatures have ranges too. Their beginning is identical with optimum upper limit and their end is determined by the operative temperature on an average skin temperature level because by higher operative temperatures: there is the danger of human body hyperthermia with the body temperature increase. Range in dTh: 46-90, see item Scale.
The long-term tolerable operative temperatures can be admitted in warm environment only: the disturbed thermal equilibrium is balanced by sweating. In cold environment sweating corresponds shivering which does not exist within most people (shivering caused by nerves cannot be taken into account). Thus shivering cannot be respected as a protective mechanism of the human body. Therefore only short-term values can be taken into account in cold environment.
The operative temperature can be estimated for the value minus 46 dTh, the beginning of hypothermia of human body.
It is interesting that the area of long-term values is the area of sick building syndrome as well (it is out of optimal values being long-term tolerable at the same time).

Short-term tolerable operative temperatures

Short-term tolerable operative temperatures are also determined by their limits. The beginning they have in warm environment is identical with maximal long-term tolerable values, in cold environment with minimal values of optimum. The end in the warm is before the threshold of pain (cca 42 °C, 135dTh)(pain is the same criterion for noise). Range in dTh: 91-134, see item Scale. In cold environment there is a new limit: if a man is able to work continuously (e.g. during the whole shift) during the chosen operative temperature. The limit in Czechia is 10 °C, in USA 15 °C only.

Intolerable operative temperatures

Intolerable operative temperatures are characterized only by their beginning identical with the end of short-term tolerable values. Range 135dTh and more, see item Scale.

SCALE OF OPERATIVE TEMPERATURE THERMAL LEVELS

It is determined by the basic equation (2) in which

$$k_{th} = \frac{135}{\log\left(\frac{42}{T_{opt}}\right)} \qquad (7)$$

Thermal levels, analogically to operative temperatures, are optimal, tolerable and intolerable (*Table 1*).

Optimal values are very pleasant, pleasant and acceptable (optimal admissible from directives point of view). Tolerable are long-term and short-term tolerable.

APPLICATION

Two examples have been chosen:

- Application to experimental values (Jokl, Kabele 2006)
- Application to optimal operative temperatures in an airliner cabin starting also from experimental values in (Jokl, Kabele 2006) but transformed into Czech Directive.

Application to experimental values based on neutral physiological state of man (*Table 1*)

The following values have been chosen from the experimental results (Jokl, Kabele 2006):

a) For 1.2 Met (70 W/m^2), 0.5 clo (optimal operative temperature 24.5 °C)

$$L_{th} = 576.718 \cdot \log\left(\frac{T}{24.5}\right) \quad [\text{dTh}]$$

The skin temperature determining the upper limit of long-term tolerable values:

T_{sk}= -0.0276q_m+35.7= 34 °C (Fanger 1970)

where q_m= 70W/m^2 (activity).

The scale of operative temperature thermal levels is presented in *Fig. 5*.

b) For 1.72 Met (100 W/m^2), 0.5 clo (optimal operative temperature 21.5 °C)

$$L_{th} = 464.22 \cdot \log\left(\frac{T}{21.5}\right) \quad [\text{dTh}]$$

The skin temperature determining the upper limit of long-term tolerable values: 33 °C

The scale of operative temperature thermal levels is presented in *Fig. 5*.

c) For 2.1 Met (120 W/m^2), 0.5 clo (optimal operative temperature 19.5 °C)

$$L_{th} = 405.144 \cdot \log\left(\frac{T}{19.5}\right) \quad [dTh]$$

The skin temperature determining the upper limit of long-term tolerable values: 32.5 °C
The scale of operative temperature thermal levels is presented in *Fig. 5*.
In all three cases following the corresponding thermal levels it can be assessed which measured operative temperatures in an interior are very pleasant, pleasant, acceptable or only long-term or short-term tolerable or intolerable.

Application to optimal operative temperatures in an airliner cabin
Optimal operative temperature in an airliner cabin is 22 °C (dTh=0). The corresponding scale of operative temperature thermal levels for activity 1.2 Met and clothing 0.75 clo ($L_{th} = 480.724 \cdot \log\left(\frac{T}{22}\right)$) is presented in *Fig. 6*. At the flight beginning operative temperature can be decreased to 21 °C (dTh=-10). After some flight time, when passengers make themselves comfortable by putting off their jackets (decrease to 0.5 clo) and start to walk in the cabin (increase to 1.72 Met) it should be increased to 23 °C (dTh=9). The maximal admissible operative temperature increase can be 25.5 °C (31 dTh). In the case of some failure the long-term tolerable values are closing at 33.76 °C (89.4 dTh).

A NEW PROSPECT: THE ASSESMENT OF THE EFFECT THE THERMAL LEVEL ON THE TOTAL ENVIRONMENT

Perhaps the greatest advantage of the new decitherm unit is possibility of a new type microenvironment evaluation. First each constituent is assessed separately, and then its effect on the whole environment. Decitherms can be also a new basis for a constituent mutual interaction study.
The paper by Rohles et al. (1989) can be used for this purpose. Various constituents have different effects on the resulting environment; e.g. our health is more threatened by cold than by aeroions.
The preliminary results by Rohles are presented in *Table 2*. The operative temperature seems to be very important (16%), the corresponding hygrothermal constituent has the greatest impact (30%), it is followed by illumination (24%), acoustic (22%), toxic (10%), odor (8%) and aerosol (6%) constituents.
The influence of acoustic, odor and hygrothermal constituents on the overall environment can be expressed as follows:

$$L_{acoustic} = AC20\log\left(\frac{P}{20}\right) = \frac{22}{100} 20\log\frac{P}{20} \quad [dB] \qquad (8)$$

$$L_{odor} = OD(50\log\left(\frac{\rho_{TVOC}}{50}\right) + 90\log\left(\frac{\rho_{CO_2}}{480}\right)) = \frac{8}{100}(50\log\left(\frac{\rho_{TVOC}}{50}\right) + 90\log\left(\frac{\rho_{CO_2}}{480}\right)) \quad [dCo] \qquad (9)$$

$$L_{th} = HT(480.7\log\left(\frac{T}{22}\right) = \frac{30}{100}(480.7\log\left(\frac{T}{22}\right) \quad [dTh] \qquad (10)$$

where dCo = deciodors.

CONCLUSION – THE BENEFITS OF DECITHERMAL SCALE

The advantages of new proposed evaluation system can be summed up in the following items:
1. The undoubtable benefit of using the decithermal scale is that it gives a much better approximation to human perception of the thermal environment level compared to the operative temperature in grad Celsius.
2. The new decitherms values also fit in numbers very well with the dB values for sound and odors, i.e. so they can be compared to each other.
3. The new decitherms values allow the estimation of operative temperature impact on the whole environment.
4. The new units, decitherm and deciodor, can be a new basis for a constituent mutual interaction study.
5. Decitherms can be estimated by direct measurement: thermometers can be calibrated directly in new units, i.e. grad Celsius scale can be completed by decitherm scale.
6. Decitherms allow assessing the feasibility of the thermal environment level, i.e. to which extent it is pleasant or unpleasant.
7. Decitherms allow defining newly the ranges of optimal, short-term and long-term tolerable environment condition.
8. Decitherms allow a new definition of Sick Building Syndrome (SBS) caused by thermal environment condition – it equals to long-term tolerable values.
9. Decitherms allow estimating the condition of human organism threat by overwarming (hyperthermia) or by overcooling (hypothermia) as a result of long-term tolerable values overcoming.
10. Decitherms allow assessing the effectiveness of heating and cooling devices by a new way – by estimation to which extent they can satisfy optimal environment level for a user.

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LIST OF TABLES

Table 1. Recommended values of operative temperature thermal levels.

Optimal	Very pleasant	0-20 dTh
	Pleasant	21-30 dTh
	Acceptable (admissible from prescription point of view)	31-45 dTh
Tolerable	Long-term tolerable	46-90 dTh
	Short-term tolerable	91-134 dTh
Intolerable		135 dTh and more

* 22-6 h
** 6-22 h

Table 2. The impact of some constituents and their parts on perceived total environment level (according to Rohles et. al. 1989).

Constituent (or part of it)	Impact (%)	Constituent factors
Hygrothermal	30.1	HT = 0.30
globe temperature	15.8	
air streaming	7.2	
air humidity	7.1	
Odor	7.5	OD = 0.08
Toxic (tobacco smoke only)	9.9	TX = 0.10
Aerosol	6.6	AE = 0.06
Acoustic	21.9	AC = 0.22
loudness	8.7	
noisy distraction	8.6	
pitch of sounds	4.6	
Lighting	24	LI = 0.24
Brightness	11	
Glare	7.9	
Shadows	5.1	

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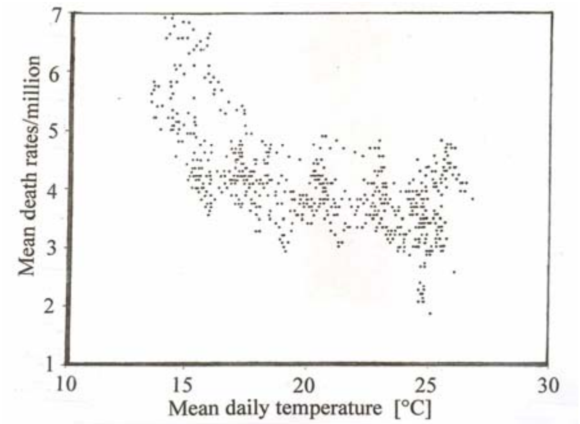


Figure 1. Distribution of 15-day means of deaths during 1984-1985 in Brisbane against temperature daily means (Auliciems, Skinner 1989).

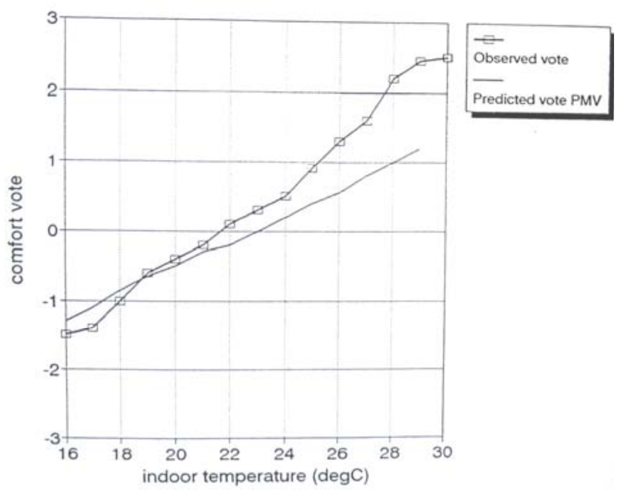
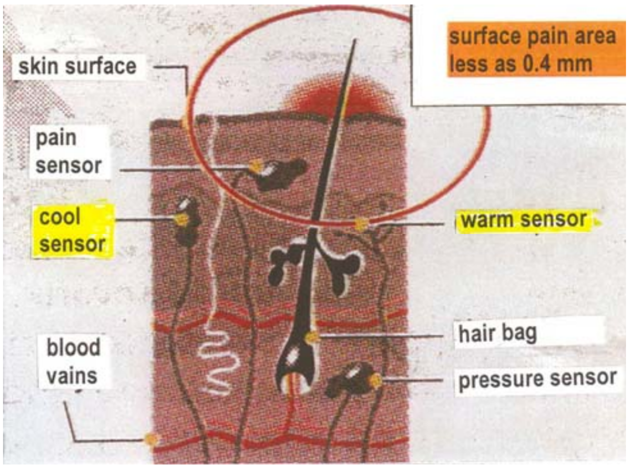


Figure 4. Comfort votes (ASHRAE scale) in relationship to indoor temperature. Activity 80 W/m^2, clothing 0.64 up to 0.82 clo. (Fishman and Pimbert 1979).



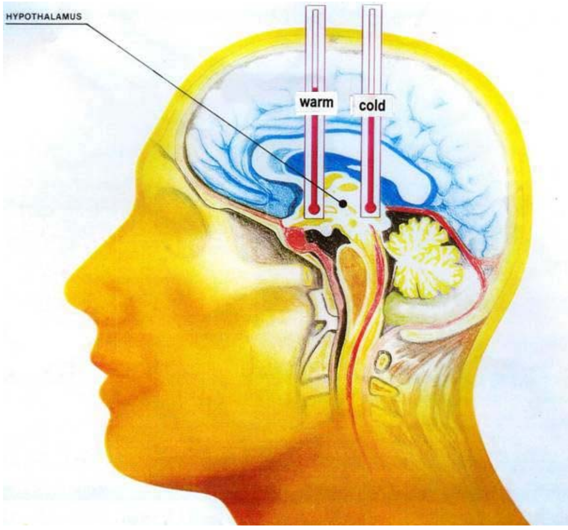


Figure 2. Skin receptors.

Figure 3. Centres for cold and warm in hypothalamus.

a) 1.2 Met, 0.5 clo

b) 1.72 Met, 0.5 clo

c) 2.1 Met, 0.5 clo

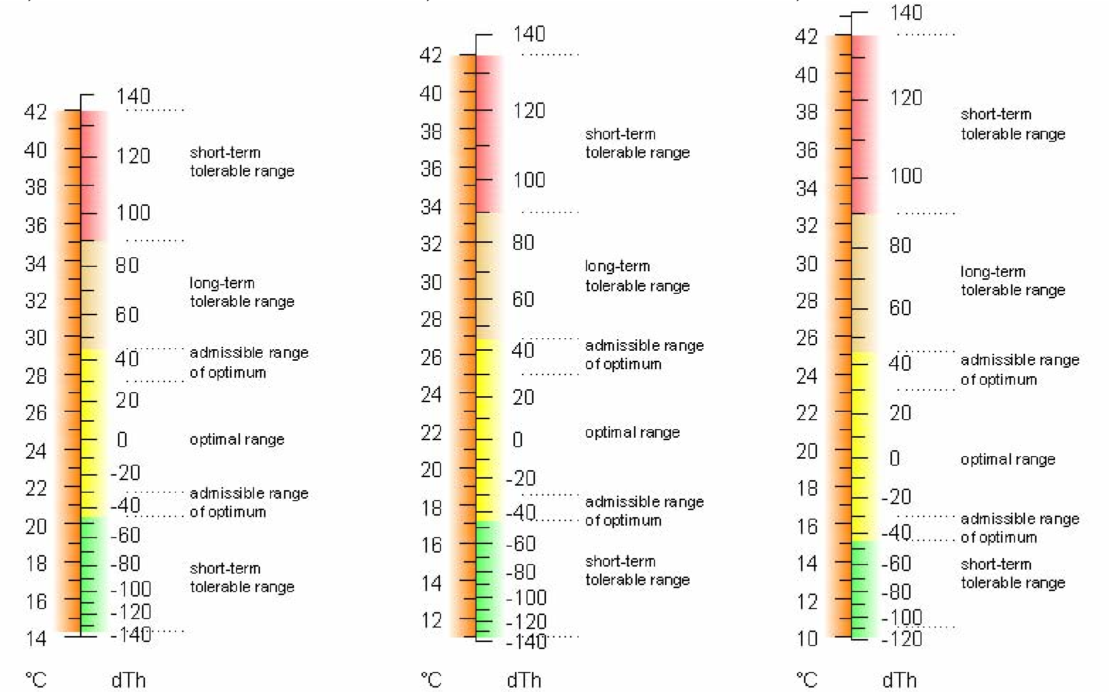


Figure 5. Scale of operative temperature thermal levels for 1.2 Met, 0.5 clo (optimal operative temperature 24.5 °C), 1.72 Met, 0.5 clo (optimal operative temperature 21.5 °C) and for 2.1 Met, 0.5 clo (optimal operative temperature 19.5 °C).

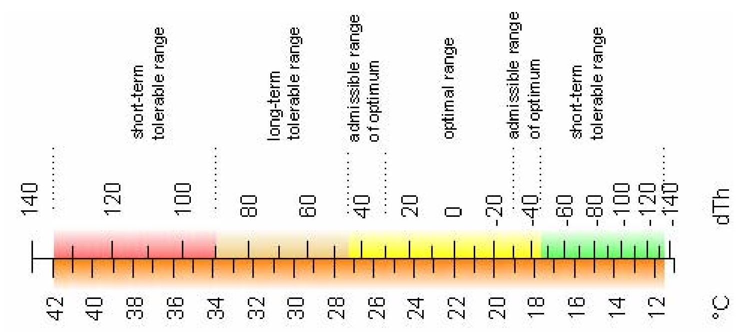


Figure 6. Scale of operative temperature thermal levels for an airliner cabin.

CFD validation of the thermal comfort in a room using draft rates

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SUMMARY

Air temperature and velocity are the two main factors affecting the thermal comfort indoors. These two values can be easily obtained using computational fluid dynamic (CFD) simulations together with the turbulence kinetic energy value. This paper evaluates methods of calculating thermal comfort indices using CFD. Simulated results are compared against experimental data measured in a purpose build full-scale model room. The results show that CFD data can reliably predict thermal comfort values.

INTRODUCTION

Computational fluid dynamics (CFD) has become a popular tool for evaluating air distributions in a buildings and it is becoming common practice to simulate different designs alternatives before building a full or reduced-scale physical model. Quite often CFD is used alone to evaluate thermal comfort conditions in a room at design stage and when this is the case, it becomes important that both the user and method are validated against some form of measured data.

The main objective of this work is to validate how closely simulated results represent measured thermal comfort conditions in a room. Experimental data for CFD validation purposes is readily available [1-3]. However, as mentioned earlier, CFD is often used for initial design instead of building a physical model. In many cases the final design of the physical model can be very different from the earlier simulated design.

Air velocities are traditionally measured using omni-directional hot-sphere anemometers which record an average air speed (V). This averaged air speed reading consists of a time averaged unidirectional velocity component as well as an unidirectional turbulent fluctuation. The comparable result from a CFD simulation is a directional velocity vector (Vv) which includes a turbulent kinetic energy component. This has being addressed by Koskela et al [4] and a method has been developed to correct the vector value to a simulated speed value (Vo). The directional results obtained from CFD simulations are being reported to have smaller values than the omni-directional results obtained using a hot-sphere anemometer. Thermal indices such as the Draft Rate (DR) defined in ISO 1994 [5] are based on omni-directional results where the effect of the proposed correction for velocity on DR are evaluated.

METHOD

Test room

The test room had a floor area of 2.83m x 4.725m and a height of 3m and was surrounded by a cavity in which the temperature could be controlled. The room had insulated walls, ceiling and floor to reduce the heat flow through the surfaces. A bulkhead 300mm high and 600mm deep was built inside the room to accommodate the supply and extract ductwork. An air handling unit (AHU) was located adjacent to the test room which provided a temperature and volume controlled supply of air to the room.

Supply and exhaust of air into and from the test room was provided via two 100mm ϕ ducts located in the bulkhead as shown in figure 1. Two dummies [6] also as shown in figure 1, were used to provide a potentiometer controlled heat load into the space. The dummies were absent in case 1.

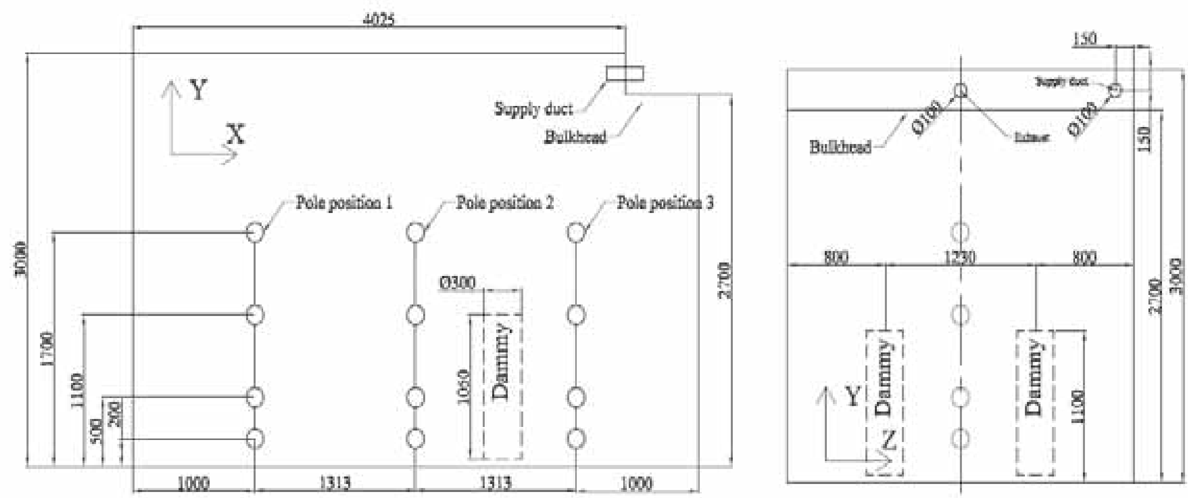

Figure 1. Test room plan and dimensions from Z and X-directions

Measurement equipment

The supply and extract volume flow rate was measured using calibrated MSD flow measurement units whose accuracy was ±5%. Temperatures were measured using PT 100 temperature probes that were calibrated to an accuracy of ±0.3K over the temperature range 15°C to 35°C. Omni-directional hot-sphere anemometers were used at various locations to simultaneously record air velocity to an accuracy of ±0.02m/s. The velocity and temperature sensors were fixed to a pole. Each pole had 4 sensors to measure air velocity and 4 for temperature (figure 2). The sensors were fixed 200mm away from the pole to reduce any effect the pole may have on the flow. Light-weigh paper strips were placed immediately below the sensors to visualise the air flow pattern in the room in addition to using a smoke tracer system. The heat load from the dummies was controlled using potentiometers and the power to them recorded using current meters.

Figure 2. Velocity and temperature measurement equipment

Test Procedure

Two different experimental test cases were used to study the conditions and flow fields in the room as shown in Table 1.

Table 1. Specification of the test cases

Case	Supply air (l/s)	Supply air temperature (°C)	Air extraction rate (l/s)	Dummy heat load (W)
1	50	23	50	0
2	50	19.2	50	240

In both tests the cavity air temperature was controlled at 23°C to minimize the heat exchange between the cavity space and the test room. The supply air temperature and volume were controlled to the specified rates and prior to any temperature and velocity measurements being recorded, the room was allowed to stabilize. A stable condition was assumed when the temperature difference across the room was less than ±0.5°C. In case 2, the desired room temperature under a heat load was controlled by reducing the supply air temperature.

Once the room had stabilised the measurement poles were placed in the room at the predefined locations as shown in figure 1. The test room was allowed to stabilize for a further 3 minutes before measurement was started. Each measurement period lasted for 3 minutes with recordings being taken at 1 second intervals and averaged. Each measurement was repeated 5 times to ensure time independent results.

Computational model

Several commercial CFD codes are available which can be used to simulate room air conditions. In this work ANSYS CFX 10 [7] was used. The continuity, momentum and energy equation were used to calculate the 3-dimentional flow field and heat transfer. The finite control-volume method was implemented for the spatial discretisation of the domain.

The geometry and boundary conditions used in the simulations represented the test room. The vertical surfaces of the dummies were defined as surfaces with a heat flux. The K-ε turbulence model was initially used because it has been regarded as industry standard for many years and its robustness and reliability has being reported in many studies [8-10]. One of its weaknesses is its ability to predict convection flows. In the most types of indoor spaces, convection flows are present and the STT turbulence model has a reputation for predicting these realistically [11] although comparisons are difficult to find. In this work CFD simulations have been made using both the K-ε and STT turbulence models and results reported.

Method used for calculating Draft Rates

The basic equation to calculate the DR is presented in ISO 7730 [5]. Turbulence intensity cannot be obtained directly from CFD results but a method to calculate turbulence intensity is presented by Koskela et al [4].

RESULTS

The velocity and temperature results were shown not to be time dependent. The variation between the results taken over five 3 minute measurement periods was less than the accuracy of the equipment used.

The average velocities recorded at the 12 measurement points are shown in Table 2 and as can be seen, simulated and experimental results show fairly good agreement. The SST simulated values show better agreement in the non-isothermal case 2. The values reported are average values taken over 5 measurements.

Table 2. Average velocity values for experiments and simulated cases

	Experiment	Simulation SST		Simulation K-E	
Case	V	Vv (m/s)	Vo (m/s)	Vv (m/s)	Vo (m/s)
1	0.20	0.16	0.20	0.18	0.23
2	0.21	0.20	0.23	0.18	0.20

The simulated and experimental velocity readings for the isothermal test (case 1) are shown in figure 3. Measurements recorded at different locations show a similar trend with regions of high and low velocities agreeing with simulated results. The biggest difference being equal to, or less than 0.1 m/s.

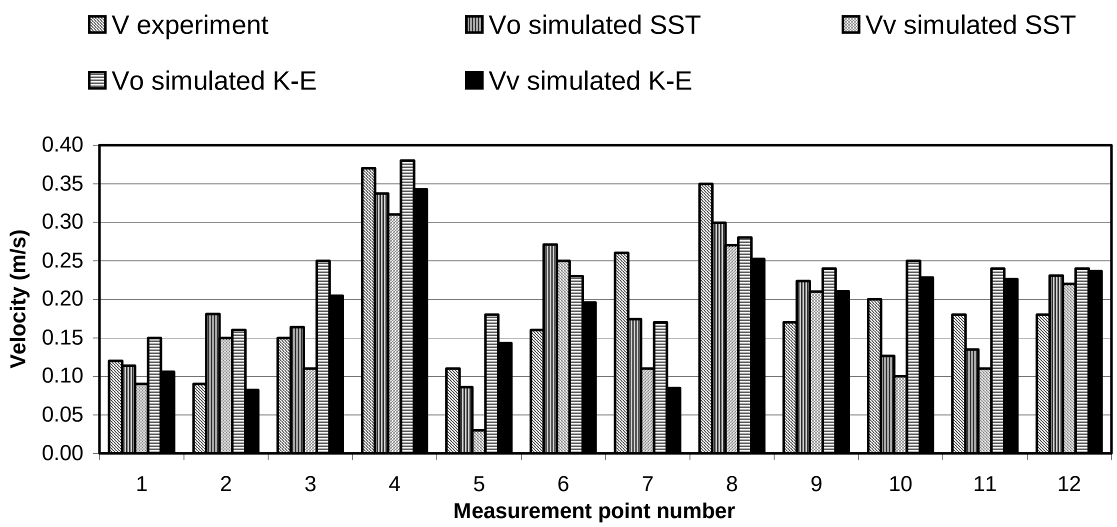

Figure 3. Velocity readings for experimental and simulated data for the isothermal test case 1

The experimental and velocity readings for the non-isothermal test (case 2) are shown in figure 4. Again in this case, experimental and simulated results agree closely but the difference between the results using the two different turbulence models is becoming evident. Both turbulence models seem to have difficulties accurately simulating high and low velocities.

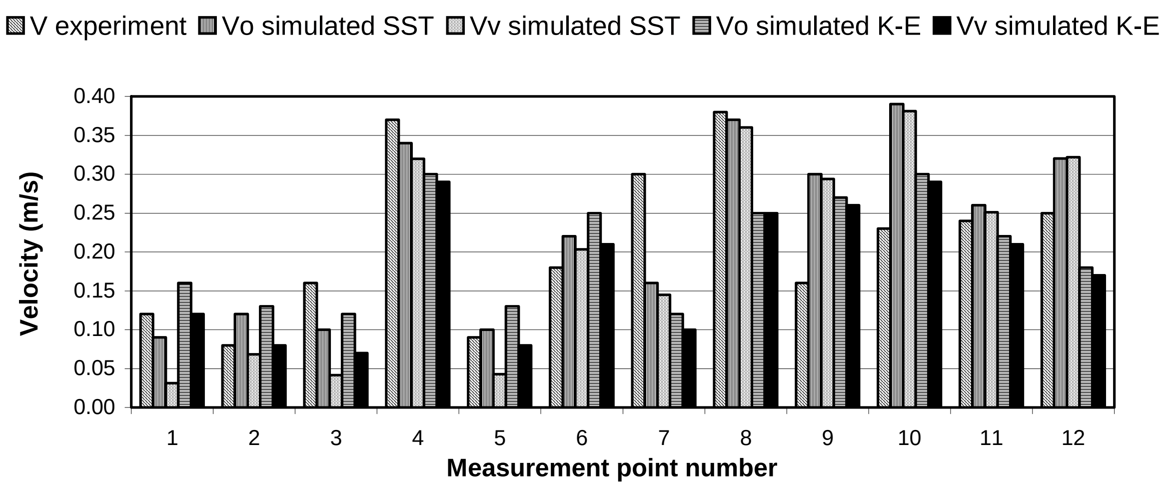

Figure 4. Velocity readings for experimental and simulated data for non-isothermal test case 2

The average draft rate (DR) results shown in Table 3 were obtained using the method reported by Koskela et al [4]. The DR values are important because in most cases, thermal comfort predictions are based on this value.

Table 3. Average Draft Rate percentage values obtained from the measured and simulated values.

Experiment	SST		K-E	
V	DR Vv	DR Vo	DR Vv	DR Vo
18	15	18	15	17

As can be seen from the table, the percentage difference between all the average DR values is small. The omni-directional simulated values agree closely with the measured data. The vector values are lower as would be expected since the correction is only applied to the higher values. Figure 5 shows the measured and simulated draft rates at various locations in the room and as can be seen, the results are in line with earlier results.

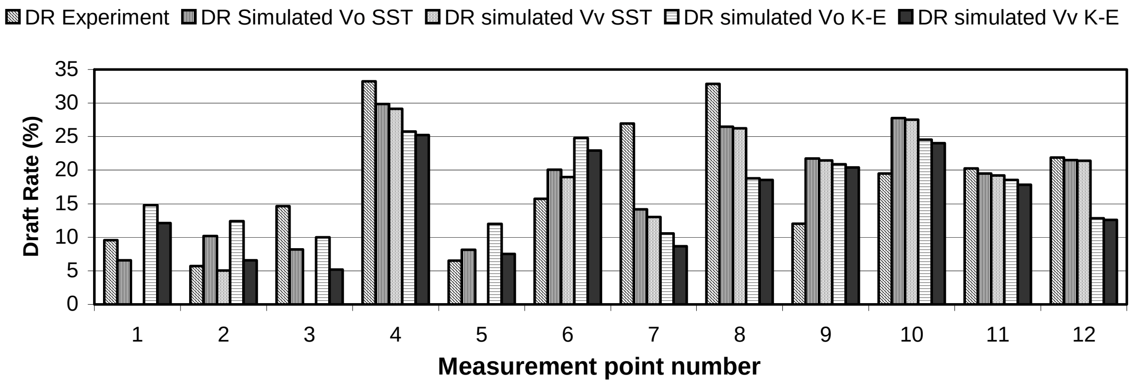

Figure 5. Draft Rate values for the experimental and simulated data

DISCUSSION

The purpose of the work was to study the capability of CFD to predict the comfort conditions in a room under different air flow scenarios. Experimental work was carried out in a full-scale test room and the same geometry and boundary conditions then used in a computational model. The experimental results are in good agreement simulated values although some differences are evident.

The results show that CFD is a useful tool to predict comfort conditions at design stage. In this study a simple room geometry was used which produced a complicated flow scenario. Both turbulence models were able to simulate the flow reliably but the performance of the SST model was better.

All new room designs have to guarantee specified levels of comfort. Common target levels of either DR 15 or 20 are specified for new designs. In addition, a maximum velocity can also be specified. The results of this study were compared with the DR 15 and DR 20 threshold values. Table 4 shows the number of points that are above these defined threshold values. CFD simulations using the SST model seem to agree slightly better with the experimental results although the differences are very small. Omni-directional CFD values seem to have very little effect on the DR.

Table 4. Measurement points above threshold values

Treshold	Experiment	Simulation SST		Simulation K-E	
DR	V	Vv (m/s)	Vo (m/s)	Vv (m/s)	Vo (m/s)
15	7	7	7	6	6
20	5	5	5	4	3

Great care has to be taken when results are presented and comparison made. The methods used to obtain experimental results must clearly stated to clarify whether the results are vector values or omni-directional values. In the present study there was little notable difference on the DR but for validation, velocity and temperature correction is highly recommended.

CONCLUSIONS

- The work showed that CFD can be reliably used to evaluate the thermal comfort provision in a new design.

- CFD simulations using a K-ε turbulence model have slightly better agreement with experimental velocities particularly in isothermal test cases. In non isothermal cases, SST turbulence models perform better.

- Correcting simulated velocity results to an equivalent omni-directional value has negligible effect on any subsequent DR evaluation. Use of a correction is however recommended when experimentally measured omni-directional velocity results are used for validation purposes.

ACKNOWLEDGEMENT

The financial and other support of Halton Oy is acknowledged and this enabled the work to be carried out.

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Effect of Moderately Hot Environment on Productivity and Fatigue Evaluated by Subjective Experiment of Long Time Exposure

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SUMMARY

In this study, a subjective experiment was conducted in a climatic chamber to clarify the effect of moderately hot environment on productivity, especially on the relationship between performance and fatigue. The climatic chamber was conditioned at operative temperature of 25.0°C (insulation value of 0.93 clo), 28.0°C (insulation value of 0.93 clo), and 28.0°C (insulation value of 0.57 clo). Subjects were exposed in the environment for six hours. The performance at 28.0°C condition was lower than at 25.0°C condition. Performance decreased as the sessions proceeded, while the total rate of subjective symptoms of fatigue increased. Correlation between personal rates of complaints of fatigue and performance in Z-score was observed with correlation coefficient of -0.76. From the linear regression model, increase in 10% of fatigue corresponded to the decrement in standard number of correct answers by 1.7%. These results implied that thermal environment effected on performance and fatigue and subjects performed worse when they felt high level of fatigue.

INTRODUCTION

It was reported that subjects exposed to moderately hot environment condition worked as much as exposed to thermally neutral conditions for short time exposure [1], [2]. However, it was shown that the subjects felt more tired in hot environment conditions than in thermally neutral conditions, so it was suggested that the evaluation of fatigue feeling was useful to evaluate productivity in thermal environment from the assumption that performance would eventually decrease when workers were tired. But there was little clear scientific evidence for the relationship between the level of fatigue and performance. The objective of this study is to clarify the effect of moderately hot environment on productivity, especially on the relationship between performance and fatigue. Subjective experiment for six hours exposure was conducted in this study.

METHODS

Experimental Conditions

Subjective experiment was conducted in a climatic chamber to clarify the effect of moderately hot environment on productivity, especially on the relationship between performance and fatigue. This experiment was conducted from July 13 to August 10, 2005. As effort for energy conservation, Japanese government has recommended to modify the cooling temperature setting in offices and to also modify the dress code to keep the workers be comfortable in summer. From this perspective, the conditions were set by the combination of operative temperature in the chamber and the amount of clothing insulations [3] as: 25.0°C with 0.93 clo, 28.0°C with 0.93 clo, and 28.0°C with 0.57 clo. A practice at 25.0°C with 0.93 clo was conducted prior to these experimental conditions, and these conditions were balanced for order. Experimental conditions are shown in Table 1. Air temperature was measured by Type T thermocouples, relative humidity was measured by Thermo Recorder RS-11(ESPEC), desktop illuminance was measured by digital illuminometer T-10 (MINOLTA), equivalent sound level was measured by sound level meter NL-31 (RION), and the concentration of CO_2 was measured by IAQ monitor MODEL2332 (KANOMAX). Air temperature, relative humidity, and the concentration of CO_2 were measured at 1.1 meters from the floor every one minute. Desktop illuminance was measured on each desk prior to the experiment. And sound level was measured at 1.1 meters from the floor every one second.

Table 1 Environmental conditions

Conditions	Operative Temperature [°C]	Relative Humidity [%RH]	CO_2 Concentration [ppm]	Desktop Illuminance [lx]	Equivalent Sound Level [dBA]
Practice	25.0 (0.16)	47 (1.2)	841 (49)	730 (43)	59.5
25°C with 0.93 clo	25.0 (0.30)	48 (0.9)	843 (113)	679 (38)	57.9
28°C with 0.93 clo	28.3 (0.42)	48 (2.9)	880 (117)	710 (43)	57.6
28°C with 0.57 clo	28.1 (0.34)	48 (3.2)	910 (61)	682 (39)	58.0

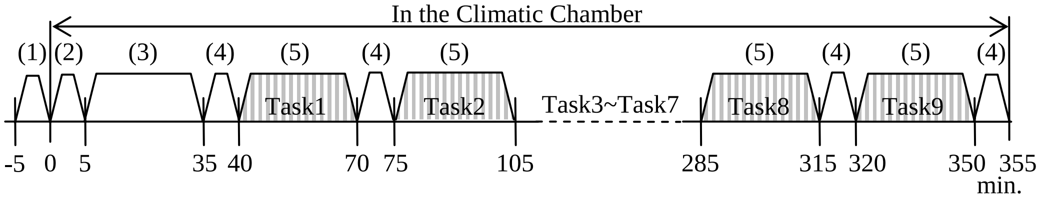
( ) Standard Deviation

Subject

Fifteen collage-age male subjects participated in this experiment. For the analysis, data of fourteen subjects were available except for the data of one subject whose data was partially missing. The anthropometrical data of the fourteen subjects were age of 20.8 ± 1.6, height of 171.7 ± 7.0cm, and weight of 62.3 ± 10.9kg. To keep the motivation of the subjects at same level, they were informed to be paid a bonus depending on their performance. Every subject participated in this experiment on same day of the week and same time of the day in the successive weeks to avoid the day and/or time influence.

Experimental Procedure

The experimental procedure is shown in Figure 1. First, subjects changed into suit coat, long-sleeved shirt, T-shirt, necktie, socks, suit trousers and leather shoes (without suit coat and necktie at 0.57 clo conditions), and then answered questionnaire on paper in the anteroom. After that, subjects entered the chamber, and then reported first thermal sensation in the chamber and their feeling of fatigue on PC and took flicker test. Subjects rested for 30 minutes in sedentary position to adapt themselves to the environment. Then, subjects were assigned to work on nine sessions of one-digit addition tasks on PC. In each session, the subjects worked for 30 minutes on the task and they reported thermal sensation, their feeling of fatigue and their evaluation of the productivity on PC and took flicker test.



(1) Change clothes and questionnaire,
(2) Vote, evaluation of subjective symptom of fatigue and flicker test
(3) Sedentary
(4) Vote, evaluation of subjective symptom of fatigue and of productivity, and flicker test
(5) Addition task

Figure 1 Experimental procedure

Measurement Items

Thermal sensation votes, thermal acceptance, and comfort sensation votes were asked to evaluate the perception of the thermal environment. Subjective votes on thermal environment are shown in Figure 2.

Satisfaction level, suitability for work, and effect of thermal environment on workability when subjects were exposed to the thermal environment were asked in order to evaluate the productivity which was evaluated by the subjects. Subjective votes on productivity are shown in Figure 3.

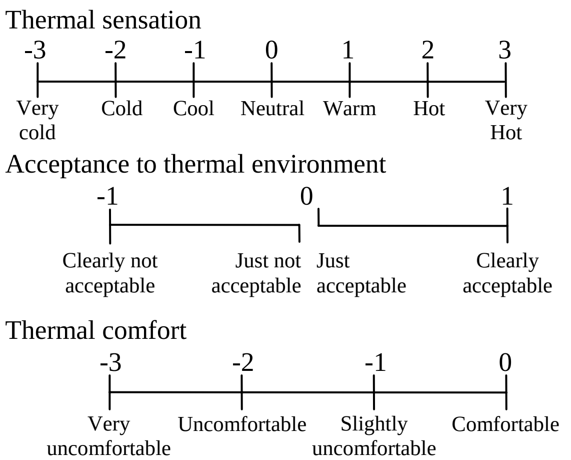


Figure 2 Subjective votes on thermal environment

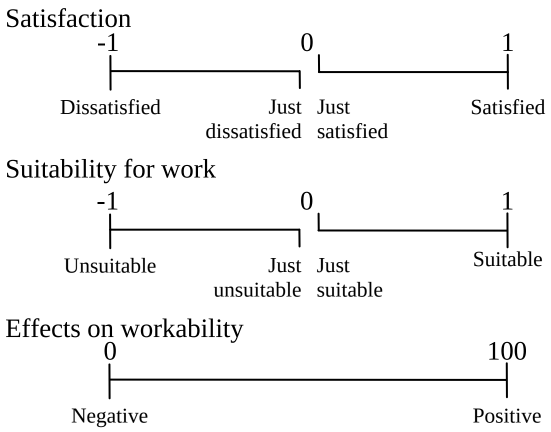


Figure 3 Subjective votes on productivity

To evaluate the feeling of fatigue, subjects filled in the questionnaire of “Evaluation of Subjective Symptoms of Fatigue [4]” on PC. It consists of three categories: group I consists of 10 terms on “drowsiness and dullness”, group II consists of 10 terms on “difficulty on concentration”, and group III consists of 10 terms on “projection of physical disintegration”. Based on Yoshitake’s method, the rate of complaints was calculated by the equation (1). By the order of rate of complaints among three categories, three types of fatigue feeling were suggested: “I>III>II” for general pattern of fatigue, “I>II>III” for typical pattern of mental work and overnight duty, and “III>I>II” for typical pattern of physical work. “General rate of complaints” is defined as the rate of complaints about all thirty symptoms.

$$Rate\ of\ complaints\ [\%] = \frac{number\ of\ selected\ symptoms\ of\ all\ subjects}{number\ of\ terms\ concerned \times number\ of\ subjects} \times 100 \ \ldots(1)$$

Subjects were assigned to work on nine sessions of one-digit addition tasks which were programmed in computers. In each session, subjects worked on the task for 30 minutes. Performance was evaluated by the number of correct answers per session in terms of Z-score (standard score). Z-score was calculated from the equation (2)

$$\text{Z-score } S_{A,i} = \frac{x_{A,i} - \bar{x}_A}{s_A} \qquad \ldots(2)$$

where, $x_{A,i}$ is the number of correct answers during the session *i* for subject *A*, $\bar{x}_A$ is the average number of correct answers of the subject *A* among all sessions, and s_A is the standard deviation for the number of correct answers of the subject *A* among all sessions.

To evaluate fatigue from physiological responses, flicker tests were taken at the first vote, after the rest in sedentary, and after each task session. In each test, flicker values were measured for five times, and the average of three flicker values without the highest and the lowest values was calculated. Then, the flicker value was evaluated by the change rate from a value of flicker value measured after sedentary.

Statistical Analysis

The comparisons between the environmental conditions were analyzed by using one-way ANOVA with the level of significance of $p<0.05$. When a significant difference of $p<0.05$ was found, Fisher's protected LSD was used for further analysis. For analysis of correlation between fatigue level and corresponding Z-score of task performance, Peason's product moment correlation coefficients weighted by number of subjects were calculated.

RESULTS

Perception of thermal environment

The results of thermal sensation, thermal acceptance and thermal comfort votes are shown in Figure 4, 5 and 6. From these results, subjects evaluated the conditions of 28.0°C with 0.93 clo and 28.0°C with 0.57 clo as hotter, less acceptable, and less comfortable than the condition of 25.0°C with 0.93 clo ($p<0.01$, $p<0.05$).

Productivity evaluated by the subjects

Satisfaction level of the thermal environment reported by the subjects is shown in Figure 7. Suitability of the thermal environment of the conditions for work reported by the subjects is shown in Figure 8. Effect of thermal environment of the conditions on workability reported by the subjects is shown in Figure 9. From the results, the environment of 28.0°C was evaluated as dissatisfied, unsuitable for work, and negative on workability, while that of 25.0°C was evaluated as satisfied, suitable for work, and positive on workability. Subjects evaluated the conditions of 28.0°C with 0.93 clo and 28.0°C with 0.53 clo as less satisfied, less suitable for work, and less positive than the condition of 25.0°C with 0.93 clo ($p<0.01$, $p<0.05$).

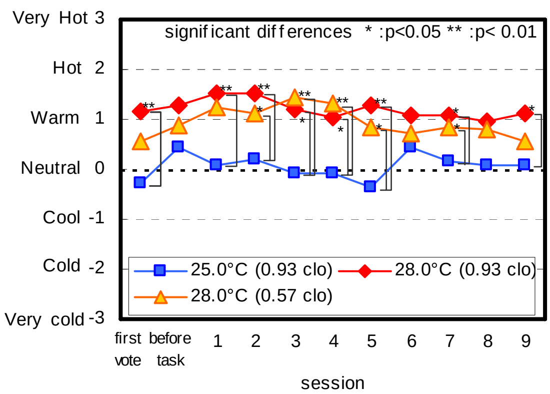


Figure 4 Thermal sensation

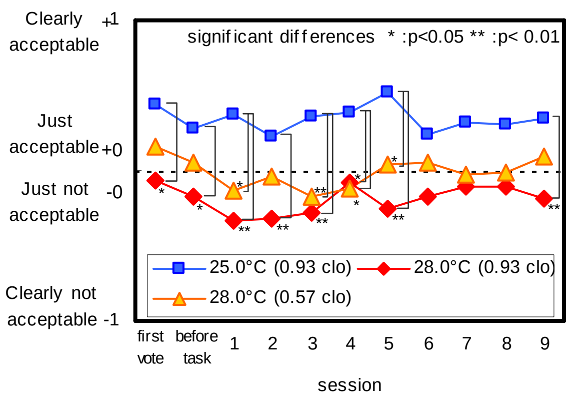


Figure 5 Thermal acceptance

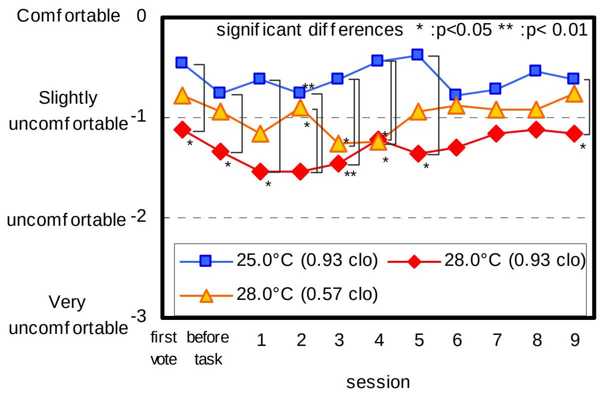


Figure 6 Thermal comfort

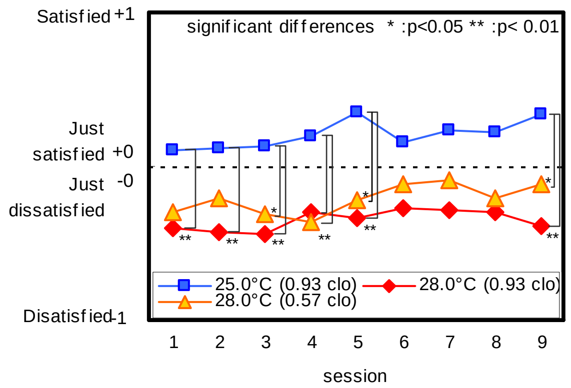


Figure 7 Satisfaction level of the thermal environment

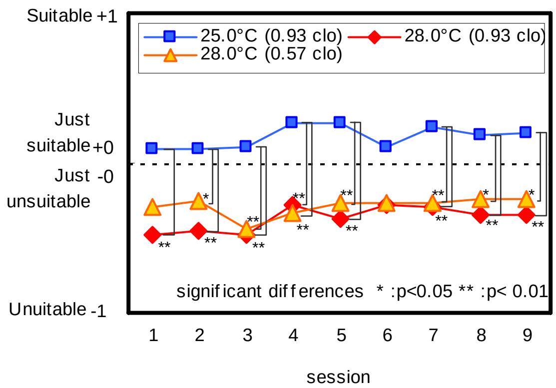


Figure 8 Suitability of the thermal environment

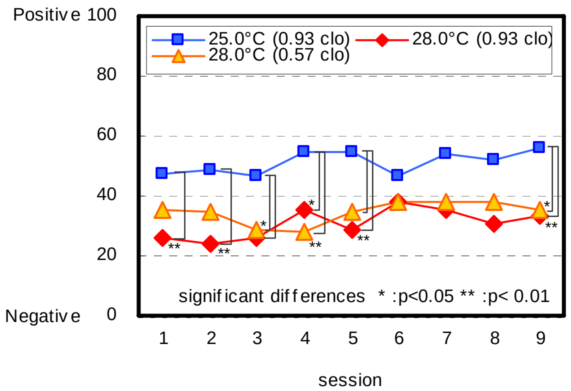


Figure 9 Effect of thermal environment on workability

Fatigue

The results of "Evaluation of Subjective Symptoms of Fatigue" are shown in Figure 10. The general rate of complaints tended to increase over the time in all conditions.

To evaluate fatigue after entire task sessions, the general rate of complaints of before and after tasks, i.e. the rates of complaints after sedentary and that after session 9, were calculated. The results are shown in Table 2. After the tasks, the general rate of complains increased, and the type of the fatigue feeling the subjects felt was the typical pattern of mental work and overnight duty (I>II>III) from the order of the categories in all conditions.

The change rate of flicker value is shown in Figure 11. The change rate of flicker value of after resting in sedentary, i.e. before task, was compared with the sessions after that. The increase rate of flicker value tended to decrease over the time in all conditions.

Table 2 The result of "Evaluation of Subjective Symptoms of Fatigue" before and after tasks

	Conditions	Rate of Complaints[%]				Category
		I	II	III	T	
Before task	25.0°C (0.93 clo)	17	11	9	12	I > II > III
	28.0°C (0.93 clo)	23	21	7	17	I > II > III
	28.0°C (0.57 clo)	20	4	12	12	I > III > II
After task	25.0°C (0.93 clo)	45	29	21	31	I > II > III
	28.0°C (0.93 clo)	45	43	22	37	I > II > III
	28.0°C (0.57 clo)	47	35	23	35	I > II > III

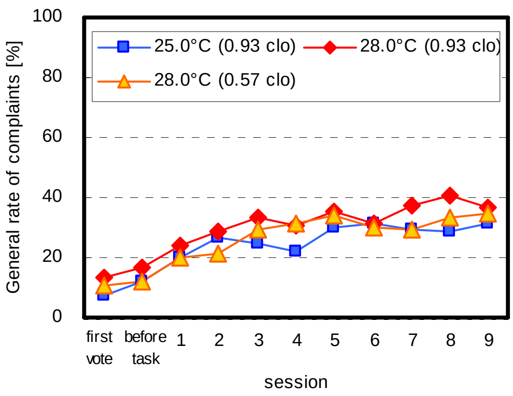


Figure 10 General rate of complaints

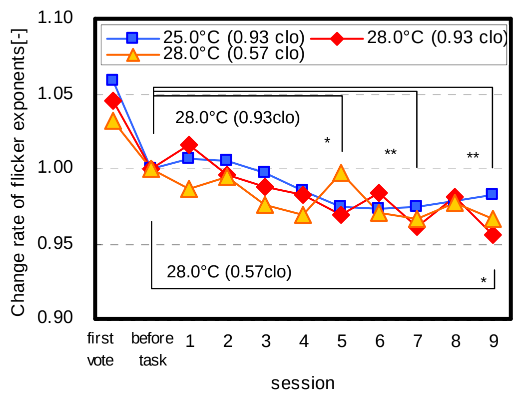


Figure 11 Change rate of flicker exponents

Performance

The result of Z-score is shown in Figure 12. The performance decreased as the sessions proceeded. The performance of the first session was compared with the sessions after that. Z-score did not changed significantly at 25.0°C with 0.93 clo condition over the time. However, the performance was significantly lower after six session ($p<0.01$, $p<0.05$) at 28.0°C with 0.93 clo condition. Also, the performance was significantly lower after the third session ($p<0.01$, $p<0.05$) at 28.0°C with 0.57 clo condition. The performance at 28.0°C condition was not significantly lower than at 25.0°C condition.

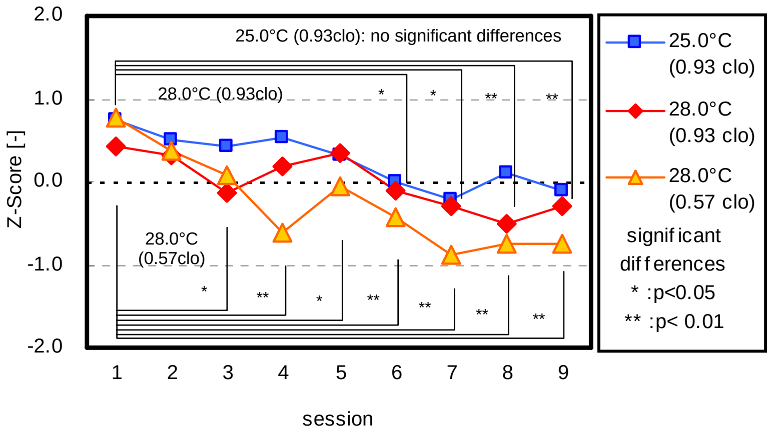


Figure 12 Task performance in Z-score

To determine the effects of thermal environment on fatigue, fatigue were analyzed in terms of the personal rates of complaints of fatigue, which are calculated by equation (1) whose number of subjects is 1. The relationship between the subjective votes on thermal environment and the personal rate of complaints of fatigue are shown in Figure 13. The data set of thermal sensation by the interval of 0.5 was averaged, and corresponding Z-score was averaged. Similarly, thermal acceptance the interval of 0.2 and thermal comfort by the interval of 0.2 were calculated. Circle shown in Figure 13 describes the number of corresponding subjects. The correlation coefficients of fatigue and thermal sensation was 0.44, thermal acceptance was -0.91 and thermal comfort was -0.93. From the result, fatigue increase when the subjects evaluated the thermal environment as hot, unacceptable, and uncomfortable.

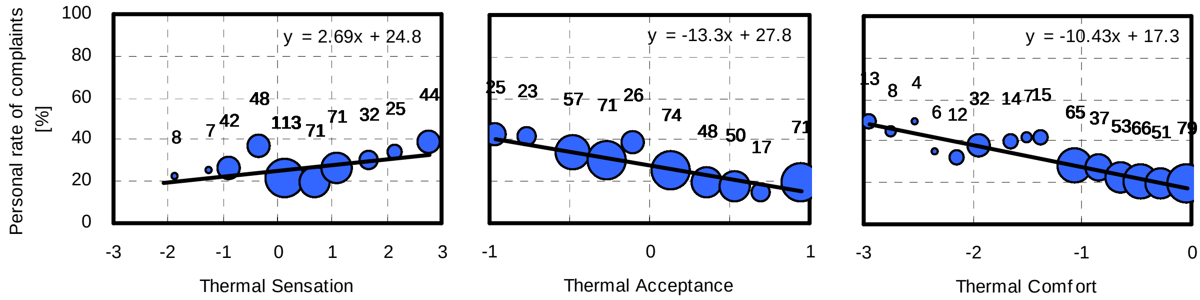


Figure 13 The correlation of the subjective vote on thermal environment and personal rate of complaints of fatigue

To determine the relationship between the level of fatigue and performance, personal rates of complaints of fatigue were analyzed. The relationship between personal rates of complaints of fatigue and task performance is shown in Figure 14. The date set of personal rates of complaints by the interval of 10% was averaged, and corresponding Z-score of task performance was calculated. The correlation between personal rates of complaints of fatigue and performance was observed (r = -0.76). In order to standardize the number of correct answers which included individual differences, standard number of correct answers was calculated by the equation (3).

$$\text{Standard number of correct answers } T = \frac{\sum_{i=1}^{N}\left(\bar{x}_A + S_{A,i} \times s_A\right)}{N} \qquad \ldots(3)$$

where, $\bar{x}_A$ is the average number of correct answers of the subject A among all sessions, $S_{A,i}$ is the number of correct answers in Z-score, s_A is the standard deviation for the number of correct answers of the subject A among all sessions, and N is the total number of data.

From the linear regression model obtained from the results, increase in 10% of fatigue corresponds to the decrement in the standard number of correct answers by 1.7%.

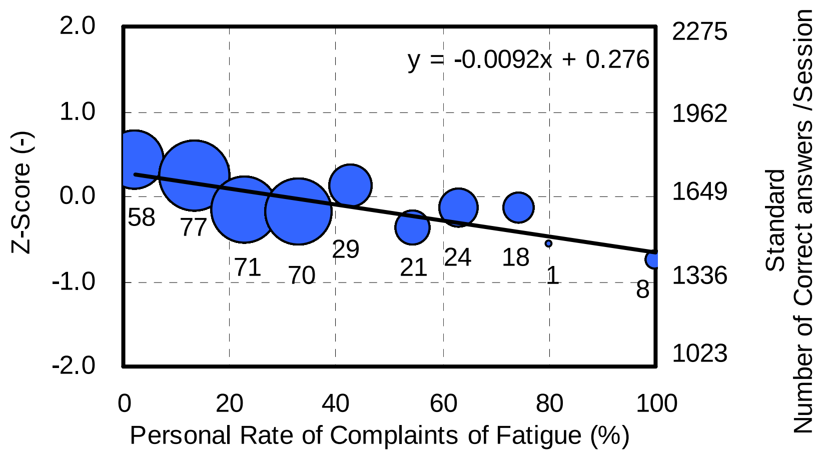


Figure 14 The correlation of the personal rates of complaints of fatigue and performance

DISCUSSION

It was suggested to evaluate productivity not only with task performance but also with human responses such as feeling of fatigue and cerebral blood oxygenation in the short exposure (2-3 hours) studies [1], [2]. There was an assumption that the high level of fatigue would lead the workers to perform worse. In this study, six hours experiment was conducted, and the results showed that there was a relationship between the subjective votes on thermal environment and the level of fatigue. Also, a relationship between the level of fatigue and performance was observed. Increase in 10% of fatigue corresponds to the decrement in standard number of correct answers of one-digit addition by 1.7% in this experiment. The results, higher level of fatigue had negative effect on task performance, supported the earlier assumption and implied the possibility that to measure subjective votes, productivity would be evaluated with higher dimensional accuracy.

CONCLUSION

To examine the effect of thermal environment on performance and fatigue, and the relationship between performance and fatigue, subjective experiments were conducted and the following results were obtained.

1. The general rate of complaints of fatigue tended to increase over the time in all conditions.
2. The performance decreased as the sessions proceeded. The performance at 28.0°C condition was not significantly lower than at 25.0°C condition.
3. The personal rate of complaints of fatigue increased when subjects evaluated the environment as hot, unacceptable, and uncomfortable. From the results, there might be effects of thermal environment on fatigue.
4. The correlation between personal rates of complaints of fatigue and performance was observed ($r = -0.76$). From the linear regression model obtained from the results, increase in 10% of fatigue corresponds to the decrement in the standard number of correct answers by 1.7%

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Thermal comfort evaluation in workplaces in Brazil: the case of furniture industry

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SUMMARY

Then main objective of this work is to create subsidies for an analysis of Predicted Mean Vote (PMV) from ISO 7730 (1994) and from the new version of the norm, from 2005, for thermal comfort evaluation in Brazilian workplaces. The furniture industry in the city of Itatiba, São Paulo State, in Brazil, was chosen for the study. A survey among the workers was carried out through questionnaires, collecting data about thermal sensation, clothing and worker's activities. Dry and wet bulb temperatures, as well as globe temperatures were measured, simultaneously to the questionnaires. The Software Conforto 2.03 was used to estimate thermal resistance of the clothes, metabolic rates, mean radiant temperature, relative humidity and the correspondent PMV for the set of environmental and personal variables. The declared votes were compared with those obtained by calculating the PMV. The obtained results were compared to the patterns specified by the standards ISO 7730, 1994 and only 4,3% of the answers are in the limits of PMV recommended for thermal comfort. An analysis is performed about the work conditions in this industry, considering the workplace conditions, especially the poor ventilation.

INTRODUCTION

The growth of industrial processes and the diversification of the economic activities show that most of urban population lives for a great number of hours in the working environments. In hot climates, like in Brazil, there is an increasing interest about the possibility of mechanical air conditioning. But this technique for achieving comfortable thermal environments is very expensive and energy consuming. The artificial climates for human occupation are designed with the objective to provide comfortable environments and also that the thermal ambient can be adapted so that each individual will be in a condition of thermal comfort.

Thermal comfort studies are based in the thermal balance of the body. With the objective of establishing optimum comfort criteria for the greatest of people Ole Fanger [1] performed studies in controlled chambers in Denmark. Fanger's model was adopted as the basis for the international standard ISO 7730 (1984), and updated in 1994 [2]. This standard establishes the determination of the PMV and PPD indices and specification of the conditions for thermal comfort in moderate thermal environments.

With the advancement of thermal comfort evaluation studies, many researches have been accomplished in different regions around the world. Some of these researches question the defined limits established by ISO 7730 in regions with tropical climate. Most of the accomplished studies evaluate ambient where the occupants sedentary activities, like offices and schools.

Thermal discomfort is a common feature in the Brazilian work environments and that certainly affects the efficiency and the workers' productivity. Most of the complaints refer to discomfort due to heat, but there are also complaints due to cold environments. These occur mainly in places where cold or frozen products are kept or manipulated. Designers of buildings and ventilation systems, as well as those responsible for work safety and health can find few practical tools available for thermal comfort evaluation in the built environments.

In Brazil, there are not available standards for thermal comfort evaluation adequate for the work conditions in the country, and especially, for no sedentary activities, as those developed in most industrial workplaces. In the national context, there is only a norm by the Ministry of Labour that regulates the exposition to extreme temperatures.

Due to this lack of information, in the last years some researchers are trying to overcome this problem, and carrying out some field studies about thermal comfort evaluation in the different climate conditions of the country.

With the aim of finding the PPD of students in school rooms, in the process of development of their normal activities, Xavier & Lamberts [3], have measured the environmental variables, simultaneously to the survey among the users with questionnaires about their thermal sensation. The study was carried out in a Technical School in the State of Santa Catarina, south region of the country, from April to July, in the morning and afternoon, with 25 sets of measurements. The authors adopted a constant metabolic rate of 70 W/m2 (1,2 met). It was obtained the following equation relating the percentage of dissatisfied (PI) with the operative temperature (T_o):

$$PI = 1{,}4151.T_o^2 - 65{,}772.T_o + 784{,}22 \qquad (1)$$

The obtained temperature of neutrality was 23,24ºC, corresponding to a minimum percentage of dissatisfied equal to 19,96%.

Gonçalves [4] carried out a study in the city of Belo Horizonte, State of Minas Gerais, in the Southeast Region of the country. His object of study was to find the intervals of comfort temperatures for undergraduate students and compare the results with those of International Standards. In the survey 570 people were interviewed. In his conclusions, the author states that Fanger’s model [1] can be applied to this population, although the obtained PPD for neutrality condition – about 27%, is much higher than Fanger’s result of 5%.

In the same year, Xavier, [5] analysed the variables which could influence thermal comfort of people in school and office buildings in three cities of the country: Florianopolis, at the South, Brasilia, with an extremely dry climate, especially in winter, and Recife, at the northeast of the country, located at seaside, with a hot humid climate. Besides the six influence variables (environmental and personal ones) from the PMV model, the author researched also habits like lifestyle, physical activities, stress, age, sex and body form. Results show that, for sedentary activities, metabolic rate is not only function of activity, but also of age and body mass. The author also found a high Percentage of Dissatisfied for neutrality temperature – 25%.

Gouvea and co-authors [6] accomplished a research with the main objective of creating subsidies for an analysis of PMV for thermal comfort evaluation in Brazilian workplaces. The clothing industry in the city of Amparo, São Paulo State, was chosen for the study. A survey among the workers was performed through questionnaires, collecting data about thermal sensation, clothing and worker's activities. Dry and wet bulb temperatures, globe temperatures and air speed were also measured, simultaneously to the questionnaires. A probit analysis was also performed which allowed to identify the temperature of neutrality. By using the average values of environmental parameters and personal variables of the surveyed population, the neutrality temperature was calculated through the PMV/PPD model. In this case the temperature of neutrality was 22,8 ºC, corresponding to 5% dissatisfied people. The probit analysis showed nearly 22,5 ºC for this temperature, corresponding to 16% dissatisfied.

Then main objective of this work is to create subsidies for an analysis of Predicted Mean Vote (PMV) for thermal comfort evaluation in Brazilian workplaces where moderate activities are performed. In this paper, results of a field research carried out also in the region of Campinas, State of São Paulo, Brazil, are presented. The evaluations followed the ISO Standards – ISO 7730 [2] which establishes the determination of the PMV and PPD indices and specification of the conditions for thermal comfort in moderate thermal environments, the prescriptions of ISO 7726 [7] regarding the instrumentation and methods recommended for the measurement of the environmental variables. Also the standards ISO 8996 [8] and ISO 9920 [9] dealing respectively with Ergonomics-determination of metabolic heat production and Ergonomics-estimation of the thermal insulation and evaporative resistance of a clothing ensemble were adopted.

METHODS

Two furniture industries in the city of Itatiba, State of Sao Paulo, Brazil, were chosen to the study. The city is 23º01'00'' South latitude and 46º50'00'' West longitude, at an altitude of 760 m. Its climate is temperate, with 20,6º C average annual temperature, oscillating between 18 ºC average minimum and 25 ºC average maximum. Average relative humidity is 72,4%. The city has a population of approximately 100.000 inhabitants, and the main economic activity is the furniture industry.

Two companies were chosen for analysis, here called companies A and B, with different sizes and characteristics. Company A is a modern one, but still maintains some old machines together with those with modern technology. Their production is mainly of special furniture, designed for specifically orders. Production sectors are well organized with defined spaces for each one. All the powder generated is collected through an exhaustion system. The polishing and woodworking sectors were analyzed. Company B is smaller, with old equipment. Sectors are separated in two sheds: one for the production and the second for polishing and painting. Some eolic exhausters are installed, but there are no openings for air renovation. The two sheds were analyzed.

Data were collected through visits to the companies for measurement of the environmental parameters. The questionnaires were applied to all the users of the buildings. For each day of measurement three questionnaires were applied throughout day: one in the morning and two in the afternoon. The daily pre-established schedules for fulfilling of the questionnaires were: 9:00 a.m., 2:00 p.m and at the end of the working period. The fulfilling was carried through by the employees without the interference of the researcher.

The measurements had been carried through in 20 days in the months of March, April, July and August 2003. Through the application of the questionnaires 914 valid answers for comparison with ISO 7730 were collected. (1994), 719 sets of answers in the company A and 195for the company B. The analyzed population was composed only of male individuals. Four values of thermal insulation of the clothing had been found: 0,55, 0,56, 0,70 and 0,89 clo. Seven activity levels were obtained, according to the description and the interviews: 1.45, 1.71, 2.05, 2.31, 2.48, 2.65 and 3.34 met as shown in Table 1.

Table 1. Metabolic rate for the different activities in the two companies A and B

Activity	Met
Clerk Production Supervisor	1,45
Electrical fitter Press worker	1,71
Expedition Cutting machine operator Hammer worker Glass worker	2,31
Machine Polisher	2,05
Hand working Polisher	2,48
Machine Woodworker Maintenance Mechanician	2,65
Mounter	3,34

Thermal sensation and preference are presented in Tables 2 and 3. It is observed that for the evaluated interval, 37.96% of the workers consider the environment as thermally neutral (0) and 31.51% consider the environment hot (+2). With regard to the preference, 55.58% consider the environment as thermally neutral (0) and 28.45% manifested the desire that the environment could be little colder. The answers supplied for the workers, with regard to the sensation and preference, do not present a symmetrical distribution. While 529 workers (57.88%) consider the environment between slightly hot (+1) and very hot (+3), 373 workers (40.81%) would only desire that the environment could be between little more cold (+1) and much more cold (+3). It is observed that 288 workers (31.51%) consider the environment hot, approximately the same number - 260 (28.45%) who desired that the environment was little colder (+1) and only 11 workers (1.20%) desired the environment much colder (+3).

Table 2 – Distribution of workers thermal sensation

Scale	Sensation	Total occurrences	Occurrence Frequency (%)
+ 3	Hot	66	7,22
+ 2	Warm	288	31,51
+ 1	Slightly warm	175	19,15
0	Neutral	347	37,96
- 1	Slightly cool	31	3,39
- 2	Cool	5	0,55
- 3	Cold	0	0
No answer		2	0,22
Total		914	100,00

Obtained data were compared with the specifications of ISO 7730 [2] which considers and environment comfortable when the PPD is below 10%, corresponding to the PMV between -0,5 and +0,5. Only 39 results are in accordance to these specifications, as shown in Graph 1. Tables 4 and 5 show the distribution of the workers sensation and preference for the interval of PMV between - 0,50 and +0.50. It is also observed that with regard to the thermal sensation, 21 workers (53.8%) considered the environment as neutral (0), 6 workers (15.4%) considered the environment hot (+2) and 8 workers (20.5%) considered slightly cold (-1). When asked about the preference in relation to the environment, 24 workers (61.5%) did not desire a change (neutral 0) and 11 workers (28.2%) wished that the environment could be a little colder (+1).

Table 3 – Distribution of workers thermal preferences

Scale	Preference	Total occurrences	Occurrence Frequency (%)
- 3	Much hotter	2	0,22
- 2	Hotter	9	0,98
- 1	Warmer	19	2,08
0	Neutral	508	55,58
+ 1	Cooler	260	28,45
+ 2	Colder	102	11,16
+ 3	Much colder	11	1,20
No answer		3	0,33
Total		914	100,00

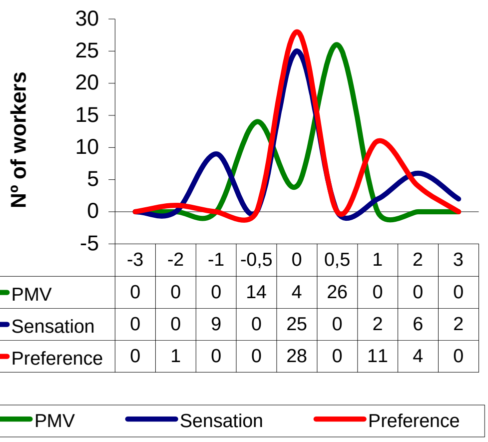

Figure 1. PMV, thermal sensation and preferences for PMV between -0.5 and +0.5

Since the measured valued of air speed were very close to zero in the first stage measurements, this parameter was not measured throughout the whole period, since the powder present in the air could damage the sensor. Most of the tasks were performed in standing position, so it was calculated the relative air speed due to the movement of the body in relation to air. This velocity was calculated as a function of the metabolic rate, in accordance with the equation proposed in the standard ISO 7730 [2].

$$v_{air} = 0,3\ (M-1) \text{ for } M>1 \text{ met.} \qquad (1)$$

By doing so, 44 results for PMV between - 0,5 and + 0,5 were obtained, as it is shown in Graph 2.

Tables 4 and 5show the distribution of the sensation and the preference of the workers for the interval of PMV between - 0,50 and + 0.50, after the correction for air velocity. One observes that with regard to the thermal sensation, 25 workers (56.8%) considered the environment neutral (0), 6 workers (13.7%) considered the environment hot (+2) and 9 workers (20.5%) considered the slightly cold environment as (-1). When asked about the preference in relation to the environment, 28 workers (63.6%) did not wish any change (neutral 0) and 11 workers (25%) desired a little colder (+1) environment.

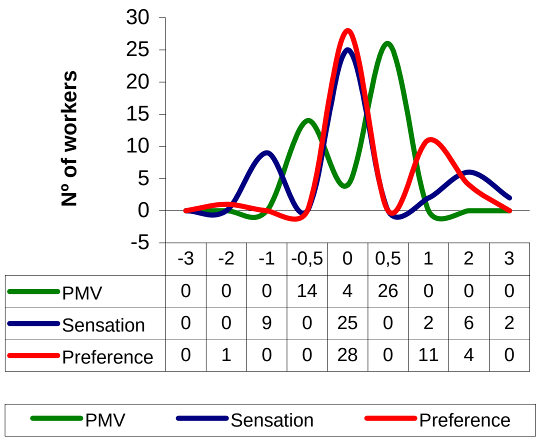

Figure 2: PMV, thermal sensation and preferences for PMV between -0.5 and +0.5.

Table 4. Workers thermal sensation for PMV between –0,50 and +0,50, after air-speed correction

Scale	Sensation	Total occurrences	Occurrence Frequency (%)
+ 3	Hot	2	4,5
+ 2	Warm	6	13,7
+ 1	Slightly warm	2	4,5
0	Neutral	25	56,8
- 1	Slightly cool	9	20,5
- 2	Cool	0	0
- 3	Cold	0	0
Total		44	100

Table 4. Workers thermal preference for PMV between –0,50 and +0,50, after air-speed correction

Scale	Preference	Total occurrences	Occurrence Frequency (%)
-3	Much hotter	0	0
-2	Hotter	1	2,3
-1	Warmer	0	0
0	Neutral	28	63,6
+1	Cooler	11	25,0
+2	Colder	4	9,1
+3	Much colder	0	0
Total		44	100

DISCUSSION

Obtained data in this field research were compared with the standards specified by ISO 7730 [2] and it was verified that, from a total of 914 sets of answers from the survey, only 39 answers (4.3%) are in the limits of PMV recommended for thermal comfort. From the total of 20 days of the survey, only in six days of measurement the recommendations for thermal comfort were reached, in some schedules. Considering the relative air speed caused by the movement of the body, an addition of 0,5% in the amount of answers that in accordance to the recommendations for thermal comfort was observed.

In this research, it was verified that there is little available information for the determination of the metabolism of the Brazilian worker, as well as data on the thermal insulation of the clothing. These values of personal variables are of extreme importance for studies about thermal comfort evaluation; so the estimate of these variables for the attainment of the PMV eventually influenced in the final results. The lack of ventilation and renewal of internal air can be considered as one of the factors that contributed for the discomfort of the users in relation to analyzed environments. Another important factor is that some of the activities performed by the workers were of medium intensity, for which Fanger's method is not the most indicated.

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Comfort Ranges Drawn up Based on the PMV Equation as a Tool for Evaluating Thermal Sensation

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SUMMARY

When dimensioning closed rooms according to the thermal sensation solving the relevant equations is necessary for a prediction of the PMV and for the determination of the surface temperature of clothing without using expensive simulation programmes in many cases, since they are not available for every designer.
Thus we considered the drawing up of thermal comfort ranges as necessary plotting the PMV ranges corresponding to the categories *A*, *B* and *C* of the general thermal comfort according to the standard CR 1752.
We have determined the terminating lines on both sides of the comfort range plotted for this purpose by fixing the values PMV = ±0.2, ±0.5 and ±0.7 using Fanger's equation on the t_{lev}-t_{ks} plane. When drawing up comfort ranges for different applications we have considered the changes in relative humidity and air velocity as well.

INTRODUCTION

During the last decades dimensioning methods have been developed enabling an even more exact calculation and evaluation of the temperature sensation of a person staying in a room. A forecast of thermal comfort is already prescribed by standards as well, however they do not determine any simple calculation method of the parameters PVM and PPD. Comfort ranges given depending on air temperature and on the radiant temperature of the surfaces surrounding the room, which could help the dimensioning do not contain references on PVM values being fixed in CR 1752 as preconditions of thermal comfort. Therefore it necessary to develop a comfort range demonstrating the connection between the expected temperature sensation corresponding to the building quality categories A, B and C, the air temperature and surface temperatures.

METHODS

When evaluating the thermal environment the environmental parameters playing the most important role are the air temperature (t_a), mean radiant temperature of the surrounding surfaces (t_{mrt}), relativeair movement (v_r), partial water vapour tension of air (p_a) as a result of relative moisture content (φ) and personal parameters, i.e. thermal insulation ability of clothing (I_{cl}), ratio of body surfaces being naked and covered with cloth (f_{cl}) and activity level (M/F_{Du}).
Criteria related to the general thermal comfort in CR 1752 are determined by the operative temperature (t_{op}) on one side and by the expected values of the temperature sensation (PVM) on the other side.

Local discomfort factors have an important role as well during evaluating the thermal environment, which are the followings: draft (i.e. medium air speed and turbulence intensity), vertical differences in air temperature, radiant temperature asymmetry and surface temperature of the floor.
Forecast of thermal comfort can be numericated by the parameters of the expected temperature sensation (PVM) and PPD. The parameter PPD gives the ratio of those being unsatisfied with the thermal environment, thus it is a characteristic value of evaluating the thermal environment quality.
Numerication of the expected temperature sensation can be determined with the help of the comfort equation written by Fanger. This thermal equilibrium equation written on the basis of thermal equilibrium of the human body is an equation defining the identity of heat produced by the human body and the heat amount transferred to the environment.
Definition of thermal comfort sensation is based on the assumption that a person feel his/her environment - in which he/she stays - as comfortable, if he/she can give over the heat produced by him/her under such conditions being appropriate to evaluate the environment as pleasant. In this case heat production and heat transfer is in equilibrium and PMV = 0.
Temperature sensation depends on thermal load (L_h) being the difference between the internal heat production and the heat amount transferred to the environment.
In case of comfort sensation the value of heat load $L_h = 0$ and the expected temperature sensation PMV = 0 (neutral). Equation determined by Fanger for the expected temperature sensation:

$$PMV = \left(0{,}352 \cdot e^{-0{,}042\left(\frac{M}{F_{Du}}\right)} + 0{,}032\right)\left\{\frac{M}{F_{Du}}(1-\eta) - 0{,}35\left[43 - 0{,}061\frac{M}{F_{Du}}\cdot(1-\eta) - p_a\right] - 0{,}42\left[\frac{M}{F_{Du}}(1-\eta) - 50\right] - 0{,}0023\frac{M}{F_{Du}}(44 - p_a) - 0{,}0014\frac{M}{F_{Du}}(34 - t_a) - 3{,}4\cdot 10^{-8}\cdot f_{cl}\left[(t_{cl}+273)^4 - (t_{mrt}+273)^4\right] - f_{cl}\cdot\alpha_c\cdot(t_{cl}-t_a)\right\} \quad (1)$$

Value η in the equation is the efficiency of mechanical work and t_{cl} the surface temperature of clothing, for which equation (2) is relevant.

$$t_{cl} = 35{,}7 - 0{,}032\frac{M}{F_{Du}}(1-\eta) - 0{,}181\cdot I_{cl}\cdot\left[3{,}4\cdot 10^{-8}\cdot f_{cl}\cdot\left[(t_{cl}+273)^4 - (t_{mrt}+273)^4\right] + f_{cl}\cdot\alpha_c\cdot(t_{cl}-t_a)\right], \quad (2)$$

Value α_c in the above equation is the heat-transfer coefficient for convection:

$$\alpha_c = \max\left\{2{,}05(t_{cl}-t_a)^{0{,}25};\quad or \quad 12{,}1\sqrt{v_r}\right\}, \quad (3)$$

Air velocity (v_r) is influenced by the activity level and by the air state built up in the room, the relative air speed caused by the ventilation and by the natural air flows arising near to cold or warm surfaces. Equation (1) does not take into consideration the turbulence intensity caused by the air flow, limited by CR 1752 together with air speed depending on the air temperature.
For the forecast of the general temperature sensation equations (1) and (2) related to the calculation of PMV are to be calculated during dimensioning - according to CR 1752 - with the given parameters by the designer for the resting points in the room.
Therefore it is necessary to draw up a thermal comfort range supporting the design demonstrating at the same time the PMV ranges corresponding to the categories A, B and C of the general thermal comfort according to the standard CR 1752.

For the forecast of PMV and for determining the surface temperature of the clothing the solving of the equations (1) and (2) and their calculation is necessary even several times within a room, since PMV values are changing in the room.
Calculations can not be performed by calculators or with EXCEL software with results of satisfying accuracy and within an acceptable period because of the multiple iteration steps.
For performing the calculations we have applied the MAPLE software calculating iteration steps within an acceptable time and gives the results with an adequate accuracy.

RESULTS

When realising the comfort range drawn up with the determination of PMV lines corresponding to the building quality categories A, B and C of the standard the following sequence were followed:

1. The standard MSZ CR 1752 gives increasing PMV ranges (±0.2; ±0.5; ±0.7) to the different thermal environment categories (A, B and C).

2. Writing these extreme PMV values into the PMV equation derived from Fanger's comfort equation the corresponding mean radiant5 temperatures can be determined for the given air temperature. During calculations determination of the t_{cl} value is also necessary depending also on air temperature and on the mean radiant temperature, thus the calculation can be performed with multiple iteration with the help of a computer.

3. As a result of the calculations six PMV = constant lines are given and by drawing them into a diagram the boundaries of the comfort range can be determined from two sides (Fig. 1).

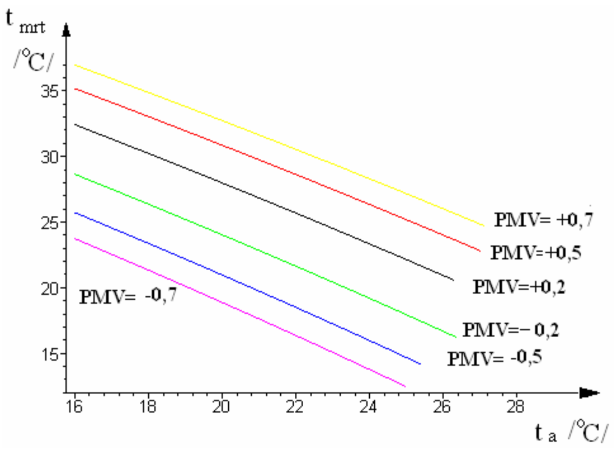

Fig.1. PMV = ±0.2; ±0.5; ±0.7 curves corresponding to the building quality categories A, B and C depending on the air temperature and the mean radiation temperature with p_a = 7.5 Hgmm, v = 0.1 m/s, M/F$_{Du}$ = 58 W/m^2, I_{cl} = 1.0 clo and f_{cl} =1.15

4. Closing the other two sides of the comfort range is possible only when meeting the condition that the ratio of convective heat transfer is fixed for different thermal environment categories, i.e. by fixing the value of "m" within all of the dry heat transfer. Thus based on

data in the literature the value of “m” was taken for 0.6 and 0.4 for the categories B and C and for 0.55 and 0.45 for the category A.

5. With the help of a computer programme developed directly for this purpose we have determined the points on the PMV = constant lines for which the above preconditions are met.

The drawn up comfort range (Fig. 2) is closed on every sides, however it is relevant only for given p_a and v. The diagram can follow the changes in air speed and partial water vapour tension if the ranges corresponding to the possible values of the variables are drawn up together. The solution is given by drawing together the ranges corresponding to v_{min} and v_{max} and $p_{a,min}$ and $p_{a,max}$.

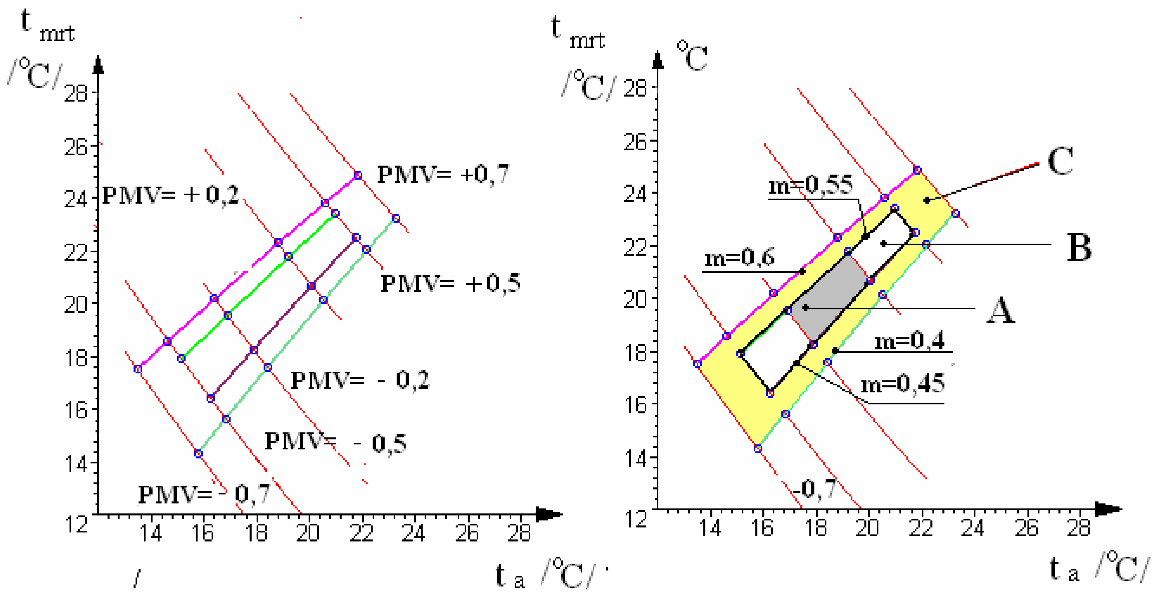

Fig. 2. Drawing up the lines for m = constant and PMV = ±0.2; ±0.5; ±0.7 by marking the ranges A, B and C

Table 1. Air velocity and partial water vapour tension values of the three room groups related to the different categories

Building types :	GROUPS	Cate-gory	Cooling season (summer)				Heating season (winter)			
			p_a (Hgmm)		v (m/s)		p_a (Hgmm)		v (m/s)	
			min.	max.	min.	max.	min.	max.	min.	max.
Offices, auditoriums, conference halls, school, restaurants, coffee-shops	**I.**	**A**	8,7	14,2	0,05	0,18	7,0	12,4	0,05	0,15
		B	5,1	15,6	0,05	0,22	5,1	14,2	0,05	0,18
		C	5,1	15,6	0,05	0,25	5,1	14,2	0,05	0,21
Nurseries	**II.**	**A**	8,7	14,2	0,05	0,16	7,0	12,4	0,05	0,13
		B	5,1	15,6	0,05	0,20	5,1	14,2	0,05	0,16
		C	5,1	15,6	0,05	0,24	5,1	14,2	0,05	0,19
Store-rooms	**III.**	**A**	8,7	14,2	0,05	0,16	7,0	12,4	0,05	0,13
		B	5,1	15,6	0,05	0,20	5,1	14,2	0,05	0,15
		C	5,1	15,6	0,05	0,23	5,1	14,2	0,05	0,18

Together with the operative temperature corresponding to the building quality categories also maximum air speeds are limited. Rooms marked in the standard CR 1752 can be classified into three groups of Table 1 according to the values t_{op}. and v_{max}. Supposed minimum value of air velocity is 0,05 m/sec in every cases taking into accout slow natural air flow and little movements during working activities.

The standard contains no information about the impact of changing φ on the offset of comfort range. Original researches and experiments composing a basis of these requirements correspond to a value of φ = 50%. Other standards are relevant for the drawing up of comfort ranges following the changes in partial water vapour tension. The diagrams of ASHRAE STANDARD 55 (1994) follow the impact of changing relative water vapour tension (φ) well and partial tension of water vapour can be clearly determined for winter and summer periods in cases where comfort sensation is guaranteed. According to our assumptions values given by the diagram correspond to the building quality categories B and C which values are shown in Table 1. Assumed value of the relative humidity in category “A” is φ = 40 ÷ 60%.

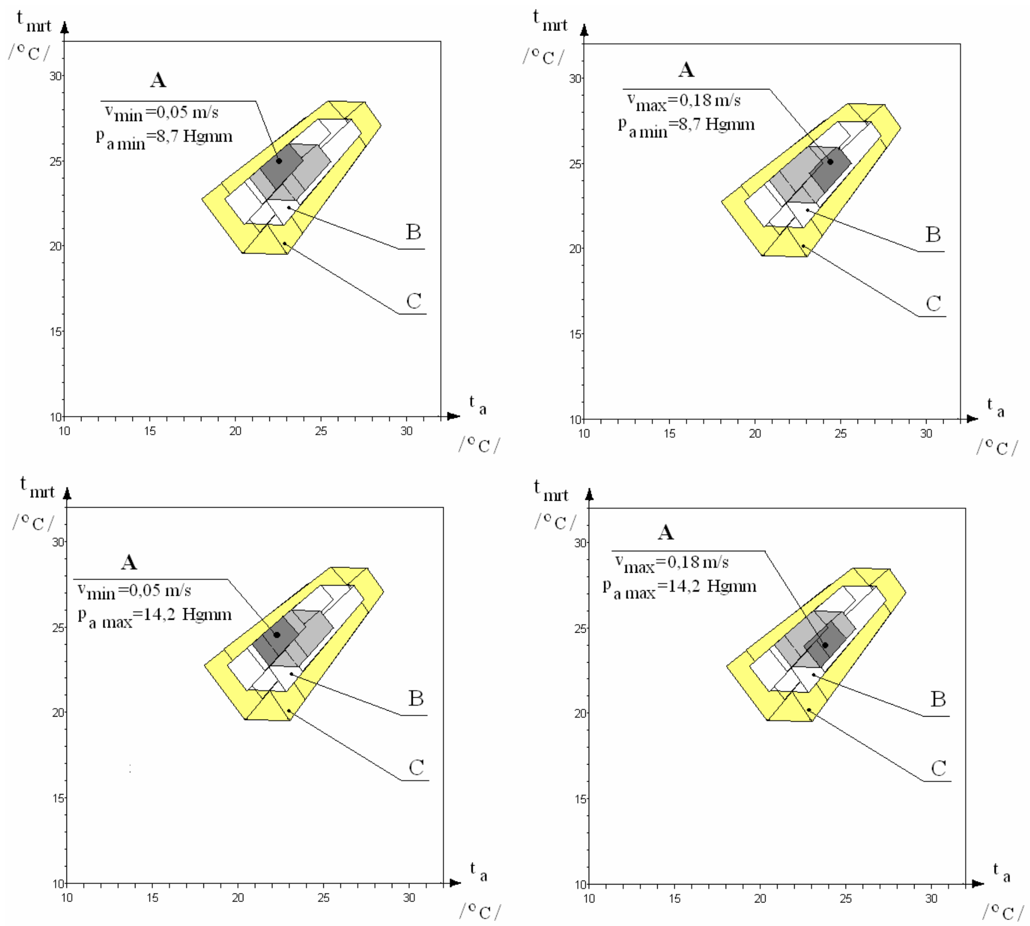

Fig. 3. Building quality categories of group 1 for summer on the plane t_a - t_r with markings for the ranges corresponding to the values of air speed and partial water vapour tension

Parameters classified into three groups depending on the type of the building are shown in Table 1, having limitations related to the clothing according to the standard, for the cooling season: I_{cl} = 0.5 clo, f_{cl} = 1.1; for the heating season: I_{cl} = 1.0 clo, f_{cl} = 1.15. The standard determines also the activity levels of the groups, for the 1st group: M/F_{Du} = 1.2 met, for the second M/F_{Du} = 1.4 met and for the third group M/F_{Du} = 1.6 met. When running the MAPLE software with the variables of Table 1, $p_{a,min}$ and $p_{a,max}$ and v_{min} and v_{max} four figures are the result drawn onto each-other giving so possible comfort areas related to a certain category.

Fig. 3 shows the building quality categories of group 1 for summer season. Rectangles within category A in the figures are relevant to changing humidity and air speed.

CONCLUSIONS

In the course of the investigations of comfort ranges the followings can be stated:

1. Fundamental difference between the groups is the activity level, the increase of which pushes the ranges nearer to the origin.
2. With increasing air velocity the ranges corresponding to the categories move to the direction of increasing t_a.
3. With increasing relative humidity, i.e. increasing partial water vapour tension the ranges move nearer to the origin, i.e. the same comfort sensation can be reached with a lower air temperature (t_a) and lower mean radiation temperature (t_{mrt}).
4. Temperature boundaries of the ranges drawn up for summer season look being appropriate related to the practical experiences in spite of the fact that in the time of the development of Fanger's PMV equation building service engineering was limited mainly to the heating and cooling had a smaller importance.
5. Ranges developed for the winter season - taking into consideration also the values of partial water vapour tension higher than those in our everyday experiences - contain low air and medium radiation temperature value pairs because of the position of the parameter lines PMV = -0.7 and PMV = -0.5. In spite of the correct mathematical processing of the PMV equations it is uncertain, if the low value pairs guarantee the expected comfort sensation in the practice.
6. With drawing up the line $t_{mrt} = t_a$ it is clear that the comfort range moves to the direction of higher wall temperatures, i.e. from the point of view of the comfort it is more desirable to guarantee a radiation temperature slightly higher than the room temperature.

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Some universal indoor environmental requirements of the seniors from Northern-East to Southern-West Europe

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SUMMARY

The attitudes can reveal the hidden reasons behind the customer choices. Referred interview studies identified typical indoor environmental requirements of the residents and made international comparison of the feedback from both young families and senior citizens.

Good housing conditions have positive influence on people's wellbeing and they should be archived without energy usage in vain. Even if most the elderly can manage well in nominal indoor air conditions, some of them might complain of draft or low room temperatures. The aged housing of the studied elderly lacked proper heating and good indoor air. Senior citizens' feedback was quite similar when comparing response from Finland and from Spain.

For the differences in the indoor air quality needs control possibility could be a solution. The acceptance of energy saving technology was low if the basic needs were not satisfied comfortable enough or the control set disturbs the routines of daily life, which in this case concerned the restrictive technology of room temperatures and in comparison that of home appliances in the sake of energy efficiency.

INTRODUCTION

It is a well-known fact that better housing conditions influence people's health. Normally with this is meant such old problems as bad heating, draft due to leaking structures, lack of tap water, shower or toilet, large amounts of hard maintenance work due to materials wearing out easily and being difficult to clean or maintain otherwise, etc. As known, many accidents happen in homes [1]. Even then, the way of building matters.

Even if most the elderly can manage well in nominal indoor air conditions, some of them might complain of draft or low room temperatures easier than people on average, most obviously due to retarded vital functions and motions or immobility. The gender differences in thermal comfort are often underestimated. Latest research carried out by Karjalainen (in press) showed significant gender differences in thermal comfort, temperature preference, and the use of thermostats. Women were less satisfied with room temperatures than men, preferred higher room temperatures than men, and felt both more uncomfortable when cold and uncomfortable when hot than men. Men used thermostats in the households more often than women, although women were more critical of their thermal environments.

Healthy building design and construction has paid special attention to health, safety and security, and functionality. Green building is a building concept promoting sustainable development by energy labelling such as American LEED (www.usgbc.org), British BREEAM (BRE Environmental Assessment Method), (www.bre.co.uk/) or Finnish WWF

Green office (www.wwf.fi/greenoffice/), etc. The prime aim is an ecological building process and buildings, and energy efficiency. However, sustainable solutions can be many.

Many house owners favour ecological and energy efficient technology. Unexpectedly, the professionals – not homebuilders, consumers and housing associations – retard the implementation of new products in housing sector. The traditions of the conservative building sector have several reasons for this [4, pp. 9, 108−109], cf. e.g. [5]. For example, prejudices against new and loyalty to workable professional paradigms play their own roles, short-term financial arguments or existing long-term contracts might prevent the building provider from picking new technology or architects are afraid for their reputation if the new products turn out to perform in an impropriate manner.

The indoor air quality was important for the elderly. The seniors expressed they clear opinion of it by favouring mechanical ventilation during the Elderly housing survey (Table 1). They named also the lack of good indoor air when asked for the three most important matters their current homes were lacking (Table 5). Expectedly, an older person wants to have warm housing without draft more often than a younger person does. Often, the elderly housing is aged too and needs repair and better indoor air, which was obvious from the result of the Elderly housing survey carried out in Western Central Finland. An exception of the need of low indoor air velocity is the summer time in the Mediterranean countries, where people and the elderly particularly suffer from the periods of high temperatures. There can be a need of proper cooling and even the motion of air – unwanted in Northern climates – mitigates the hot feeling and will not necessarily considered that bad in the Southern parts of Europe.

Indoor air quality was also the sixth most important option picked by the respondents of the interview study carried out in the Pirkanmaa region in Finland, when it was studied the residents' opinions of the smart home qualities [6]. Out of 39 smart home facility options, the top of five before the indoor environmental quality were about safety: fire-fighting equipment (55 per cent of the respondents), water leakage detector (55 per cent), automatic on/off system of home appliances (48 per cent), distant alarm receiver (41 per cent) and smart home security system (36 per cent). The respondents belonged to all adult age groups. Interestingly, also office workers valued indoor air quality control highly according to the study of intelligent building technology “What Office Tenants Want” by the BOMA and the ULI in USA. Furthermore, office workers in Finland complained of poor indoor air quality not only in the other high quality offices, but also in the intelligent office buildings according to the Intelligent Buildings Survey [7].

As there were personal differences in the needs concerning indoor air quality, personal control possibility could be a solution. The earlier Elderly housing survey gave evidence that the elderly could not combine together the indoor air quality and its controls [3]. Instead, many of them mentioned only good indoor air quality to be important for them. That was rational thinking. Actually, as mentioned earlier good indoor air was important for women, while men rather than women use the control devices [2]. Latest research by Karjalainen (in press) shows significant gender differences in thermal comfort, temperature preference, and use of thermostats among all aged subjects. Females were less satisfied with room temperatures than males. Females feel both uncomfortable cold and uncomfortable hot. Males use thermostats in households more often than females, if only females were more critical of their thermal environments.

Table 1. The shares of those who were willing to have smart home features for building services at home according to the Elderly housing survey (in 1995, n=315, the percentages of the respondents) [3] [1].

Air-conditioning, domestic water	[1]
Mechanical ventilation	61
Water leakage alarm (alarming when water on floor)	48
Cooling in summer time	33
Electric metering tap (closing automatically)	24
Photoelectric tap (opening when hands are under the tap, automatic closing)	24
Irrigator for greenhouse	8
Other	3

[1] Pick of several choices of multiple-choice questions possible

Table 2. Lackings of the current homes during the Elderly Housing survey (the numbers of the respondents) [3].

SURROUNDINGS (altogether)	70
Services too far away, poor transportation facilities or two distant location	35
Disturbing traffic	11
Other problems	24
APARTMENT (altogether)	169
Unfunctional (the kitchen was mentioned 25 times among them)	40
Too little space (the bath room was mentioned 12 times among them)	37
Too little rooms (the entrance was mentioned 8 times among them)	22
Too many levels	13
Bad shape	13
Too much space	8
Too much maintenance	6
Too high housing costs	5
Too high heating costs	4
Other	21
MISSING SPACES (altogether)	143
Storaging rooms, cellar	67
Sauna bath	18
Washing room	17
Balcony	11
Second toilet	10
Other	20
MISSING OR POOR QUALITY EQUIPMENT (altogether)	100
Air-conditioning	25
Extra heating unit (fire place etc.) or poor quality heating (mentioned 7 times)	23
Elevator	18
Poor soundproofing	9
Windows unfunctional or in a poor shape	9
Other	16
UNCLASSIFIED (altogether)	19
Bad neighbours	4
Renovation irritates	3
Other	12

METHODS

Urala (2005) has found that the respondents' attitudes have revealed statistically significant differences in both liking and purchase intentions. The study of attitudes can reveal the hidden reasons behind the customer choices, which often remain unclear even for the individuals themselves. The surveys referred in this paper had almost the same themes and used quite often similar questions [8], [10]. It has been possible to identify typical indoor environmental requirements of the end users, as well as to make comparisons within the international context between various customer groups based on feedback from both young families and senior citizens.

Within the EU 5th FP project Demulog (Living Conditions and Preferences of Families in Housing Need, Table 3) the numerical analysis were carried out (using the SPPS program). The qualitative responses to the open ended questions from the EU 5th FP project Elderathome (The Prerequisites of the Elderly for living at home: Criteria for Dwellings, Surroundings and Facilities, QLK6-CT-200-00405, Table 4) have been grouped, listed and summed up. Results from two corresponding Finnish studies were used as reference values (Table 5). All samples in each country were dominated by residents who dwell in the block s of flats and in most cases in cities as well as their demography represented quite well the national averages cf. [10].

Table 3. Summary of the cases of the Demulog project (n=294).[1]

Year	*Location*	*Target groups*	*Sample*
1. Autumn 2000 2. Spring 2000 3. Winter 2000-2001	1. Finland, Helsinki 2. France, Vienna 3. the United Kingdom, Peterborough	1. The subjects from 500 applicants for social housing provided by VVO[2]. 2. The subjects from some 1,500 households applying for social housing from OPAC Vienna[3]. 3. The subjects from the applicant lists of the Nene Housing Society Ltd in the Peterborough City[3] and the Peterborough City Council Housing Department.	1. 103 2. 101 3. 90

[1] Personal face-to-face interviews, the same questionnaire for the interviews used in each country; [2]VVO Group was a private social housing provider and an owner of 35,000 properties in 72 municipalities; [3]a private social housing provider.

Table 4. Summary of the Elderathome study from Finland and Spain (n=206).[1]

Year	*Location*	*Selected case groups*	*Sample*
Spring and summer 2003	Finland Helsinki	Subjects aged over 55 years from three cases: - 32 members of an association of active seniors[2], - 28 tenants of two housing real estate companies in Puotila housing area[3] and - 46 randomly accessed elderly living in Vuosaari housing area[4] from the data bases of the Population Register Centre.	106
Autumn 2002	Spain Barcelona and neighbouring towns	Subjects in the age of over 55 years from 15 cases living in regular housing, in senior housing, in private and municipal service homes for elderly.	100

[1] Personal face-to-face interviews, the same questionnaire for the interviews used in each country; [2] A group of seniors, who plan a senior house of their own ; [3] The elderly were the majority and their houses need repairs.; [4] The largest residential area in Helsinki, where population was growing rapidly, because the recent new development since 1960's.

Table 5. Summary of the Elderly housing survey carried out in Finland [3].[1]

Year	*Data collection location*	*Target groups*	*Sample*
1995	The city of Tampere, a few municipalities in the Pirkanmaa region	A randomly accessed subjects aged 55 to 70 years by the Population Register Centre.	over 300[2]

[1] Posted questionnaire; [2] the answering percentage of 32 per cent

ADJUSTMENT OF INDOOR AIR MEASURES FOR ENERGY SAVING

The Demulog interview studied the user requirements of energy saving options carried out by the smart home technology, such as saving energy by reduced room temperatures or limiting the use of electricity for home appliances to certain periods of the day. The families in Finland (71 per cent of the responses) and in France (55 per cent) responding to the Demulog interview oppose an energy saving system using a restrictive technology [8]. Only in the United Kingdom 58 per cent was in favour of such savings. The preferable room temperatures turn out to be as follows [8]:

- A large majority (> 80 per cent) accepts lower temperatures when nobody was at home or during the night.
- A small majority (from 50 to 57 per cent) accepts lower temperature in the bedrooms but not in other living spaces, and a third (35 to 45 per cent) wants to have the same temperature all over the dwelling.
- A large majority (95 per cent in Finland, 93 per cent in France, 72 per cent in UK) wants as high or higher temperatures in the bathroom and toilet as in the living areas.

Nevertheless, the respondents did not oppose all other means of saving energy as the limitations of room temperatures. The acceptance of low energy consuming home appliances was very good in Finland (in most cases 99 per cent, n=106), good in the United Kingdom (in most cases from 91 to 96 per cent or more, n=90) and rather good in France (in most cases from 64 to 70 per cent, n=101). The acceptance of water saving measures was good (Finland 69 per cent, France 70 per cent, the United Kingdom 87 per cent), but the acceptance of a composting latrine was not that good (Finland 19 per cent, France 18 per cent, the United Kingdom 29 per cent).

The lesson of the result of the Demulog interview teached that if residents seem in practical terms flexible in timing during the daytime, still, when they were without any compulsory ties they do not feel free. The daily routines of the society were still usually followed. Changing the timing of domestic work is not necessarily possible.

The elderly can be considered as citizens without responsibilities and with free time, and thus flexible in the timing of daily activities. However, also they follow the common timing of daily routines. The travel experiments which have offered the elderly the chance to use public transportation with a reduced charge have resulted in a failure because those trips were not made to a great extent beyond the peak hours (Discussions at the NECTAR (Network on European Communications and Transport Activities Research) conference "Transport Innovations, Competitiveness and Sustainability in Information Age". no 4 in Israel). Furthermore, Fox (2004) found that seniors use computers on a typical day similarly to the younger computer users.

IMPORTANCE OF GOOD INDOOR ENVIRONMENT

It was interesting to find out how similar the response from the two geographically and climatically very separated countries, Finland and Spain was when it concerns the lacking qualities of present housing of seniors. Within the EU 5th FP project the Elderathome such lacking qualities mentioned during the Finnish interviews as the lack of elevators or the need for other types of easy access out, needs for kitchen repair, poor soundproofing and indoor air quality, difficulties in using the bathtub or the wish to have a balcony (or two) and more natural light, were common to all studied cases of senior citizens. Similarly to the earlier Finnish studies, the indoor air quality turned out to be important.

The Spanish respondents paid special attention on such shortcomings as (in order of importance) [9]:
Basic contraction issues
- Dwelling was not equipped with central or electric heating. (This was a frequent problem in old quarters.)
- The surface outside the entrance was not horizontal and on a different level with the surface of the entrance.
- No elevator.

The Spanish respondents paid special attention on such best qualities of housing as (in order of importance) [9]:
Positioning of the house in relation to the sun.
- Natural light.
Outside space.
- To have a terrace, a balcony, a little space outside where they can have some plants.
Neighbourhood issues
- Services in the neighbourhood and near to the urban centre. No one was worried about the house technical aspects. They thought that was too expensive and difficult to understand.

In Finland, the respondents to the Elderathome interview reported problems with the HVAC measures and technology in their current homes quite as often as problems with sounds and noises, after the major problems of mobility and spatial needs (Table 6). The result was quite same as gained earlier during the Elderly Housing survey (Table 2). When asked about the ideal future home for old ages the importance of the qualitative HVAC technology guaranteeing the good indoor environment turned out not to be that important by the Finnish respondents (Table 7). Similar result can be found from the summary of the Spanish responses.

The sample in Spain had the male majority or men were over presented compared to the national average in that age group, and the Finnish sample had the female majority or over presentation compared to the national average in that age group. Despite gender differences in the two samples, when comparing them in their entirety this study did not show any differences between genders in the needs and wishes for the indoor environmental requirements. However, during this study too the gender differences were found in the preferences of the choices between different control technologies [12].

DISCUSSION

The concept of sustainability as a whole necessitates both the energy conservation activity and the human acceptance of it, because sustainability equals both the human wellbeing and integrity of the nature. According to the results of the Demulog project, the possibilities to use smart home control for energy saving were dependent on which home electronics and appliances were in concern and which types of means were used for the energy saving. The quality of the technical application counts in the acceptance of using energy saving measures. On one hand, a large majority accepted lower temperatures when nobody was at home or during the night, and on the other hand, a higher temperature in the bathroom and the toilet than in the living areas. A small majority accepted a lower temperature in the bedrooms but not in the other living spaces. The acceptance of energy saving technology was low if the basic needs were not satisfied comfortable enough, which in this case concerned the room temperatures. Using technology which saves energy and is ecological was also in favour. If the control did not disturb every-day life, the daily rhythm or the routines of the activities, the control was acceptable. To prevent electricity consumption peaks by changing the time schedule of the domestic work does not seem to be possible largely.

Table 6. The three worst shortcomings of present dwelling in the studied Finnish elderly homes (the numbers of the respondents).

Problem type	Problem	n
Mobility	Entering the house: no elevator, too many steps or too small elevator cage	22
	Moving indoors: two floors and steps upstairs	5
Need of space	Too big apartment, garage and sauna bath in vain	8
	More space needed, too small rooms	8
	Missing space	22
Sound and noise	Poor soundproofing	7
	Noise from traffic disturbing or metro disturbing	3
HVAC	Bad air-condition or bad indoor air, draft	4
	Cigarette smoke in the balcony, cooker hood	2
	Personal control on air-conditioning needed	1
	Too hot in the summer time	1
Bathroom	Bathtub difficult to use, change of the tub to shower	7
Maintenance	The courtyard in the winter time overwhelming because of snow, caretaker takes care of removing snow and sweeping up and residents not allowed	8
	Too much work, too much to clean	2
	Other	4
Renovation	Old building needs renovation	4
	Planned renovation, future boiler room, bathroom or kitchen renovation, damages due to damp in the basement, bad structures, cold floor	7
	Renovation took too long time/renovation disturbing	2
Costs	Energy consumption	1
	Too high rent, dwelling (housing) is expensive	2
Electricity	Location of lighting switches could be better	2
Interior design	Thresholds, darkness, kitchen cabinets too high	4
Other		15
No complains	Satisfied	6

Both the Finnish and the Spanish elderly respondents kept the indoor environmental qualities as important. Both groups had problems with the HVAC measures and technology in their current homes but other factors than good building service technology counted less when they told about their "dream homes" for old ages. The Elderly housing survey [3] tested the importance of the HVAC measures against a limited number of factors, against those of the building services and then the importance of the HVAC technology emphasized (Table 1). The same was the case with the other Finnish study of the residents' opinions of the smart home qualities which also found the relevance of the indoor air requirements important for the residents [6]. All studies have concluded that the current homes of the elderly tend to be in such a shape that they need renovation, and the residents have problems with the HVAC measures.

This study found no gender differences in relation to the importance of the indoor air quality measures. However, the samples were small − although well representative [10], compared to that of Karjalainen (n=1000) [2], which showed gender differences in relation to the indoor air measures. Most importantly, the studies were not otherwise fully comparative either. The set of the questions were different. Importantly of all, the method of analysis was different. This analysis has reached up to the comparison of two sample as a whole − the one with female majority and the other with female majority, on the basis of the response to the qualitative open ended questions and without any statistical comparison between gender factors. In stead, Karjalainen has accomplished careful scientific analysis on the gender differences [2].

Furthermore, the comparison of this study was made between two nationalities. Then, the cultural differences and climatic factors may influence the result. It can be concluded though that cultural differences often have less influence on the satisfaction of such basic needs as the indoor air quality [10]. However, the cultural influence on the standard of housing might have an effect on the requirements for the indoor air quality as was seen from the results of the Demulog study. The English were more ready to accept the lower temperatures than the Finnish or French respondents. When comparing the Finns and the Spaniards, the comparison was made between two nations where the housing habits were quite similar (most housing in the blocks of flats) despite climatic differences.

Table 7. Important qualities in an ideal home building, especially, if the life as a senior citizen was in concern in Finland (the numbers of the respondents).

Wish type	Wish	n
Services	All services available or near by	34
	Health service near by	6
	Services - cleaning, services – shopping	7
	Other	5
Mobility	Easy entrance (wheel-chair taken into account in design)	34
	Easy access indoors	6
	Good (public) transportation	11
Need of space	Sauna	6
	Balcony, balcony also next to bed room	5
	Enough space, applicable size, own room, missing rooms	10
Interior design	Functional lay-out and for domestic work, materials easy to take care	12
	Peaceful dwelling,	7
	Other	3
Bathroom	Bath easy to use, free access to bath, shower (no bath tub)	3
	Bigger bathroom, spacious enough and of good quality	3
	Floor heating, safety system in bathroom	2
HVAC	Mechanical air conditioning, good indoor air or air conditioning and no draft	3
Electricity	Safety system in electricity	1
Sound and noise	Good sound proofing	1
Maintenance	No winter maintenance – snow	1
Surrounding	Peacefulness, safe, sea near by, etc.	20
Neighbourhood	Friends, communality, help available, safety	17
Not classified	Not many floors	7
	Other	19

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Derivation and analysis of the outdoor Wet Bulb Globe Temperature index (WBGT) with a human thermal engineering approach — Part 2. Properties of the WBGT formula for outdoor conditions with solar radiation

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SUMMARY

The authors present a theoretical derivation of the WBGT formula for outdoor conditions that was originally developed from the results of experiments on human subjects, based on a heat balance equation between the human body and its outdoor environment. The three coefficients of wet bulb temperature T_w, globe temperature T_g, and air temperature T_a were expressed in almost the same way as in the indoor WBGT formula, but they contain a new element characterizing solar radiation. In addition, we calculated the coefficients in the theoretically derived formula, changing the amounts of metabolic activity, clothing worn, wind velocity, and solar radiation. We obtained the new formula WBGT = $0.84T_w + 0.30T_g - 0.08T_a$, characterized by a negative coefficient of air temperature T_a, as an alternative to the original outdoor formula WBGT = $0.7T_w + 0.2T_g + 0.1T_a$. Finally, we indicated the characteristics of the WBGT as an index as well as instructions for use.

INTRODUCTION

The Wet Bulb Globe Temperature (WBGT [1]) has been developed from experiments by Yaglou and Minard in the U.S. in 1957 for the prevention of heat stroke in soldiers. The index is standardized by the National Institute for Occupational Safety and Health (NIOSH) and the International Organization for Standardization (ISO) as ISO-7243. Therefore the index is now frequently used as an evaluation index for thermal environment during work and outdoor sports, and its use is recommended in many countries including Japan.

The WBGT was developed based on experiments with humans, but its background has not been discussed from the perspective of the heat transfer theory. The physical heat exchange between humans and the environment provides physiological and psychological reactions. Hence the properties of indices, even those based on human experiments, can be examined from the viewpoint of physical heat balance.

In this paper, we carry out theoretical derivation and analysis of the WBGT for outdoor conditions employing the heat transfer theory based on the heat balance equation between human body and an outdoor environment. Then, based on the derived theoretical formula, we clarify the structures of the three constant coefficients of wet-bulb temperature, globe temperature and air temperature that define the original WBGT formula for outdoors. In addition, the derived formula for outdoors is compared with that for indoor conditions derived in another paper [2], and we consider points of difference and similarity between two formulae. Furthermore, we perform an examination using actual values to present the characteristics and applicable conditions of the WBGT.

METHODS

Yaglou and Minard have suggested the original WBGT formula (1) for outdoor conditions.

$$WBGT = 0.7T_w + 0.2T_g + 0.1T_a \qquad (1)$$

where WBGT is wet bulb globe temperature [°C], T_w is (natural) wet bulb temperature [°C], T_g is globe temperature [°C] and T_a is air temperature [°C].

The heat balance equation (2) between the human body and an outdoor environment is expressed as follows:

$$M = (C + R) + E_{sk} + E_{res} + W + S \qquad (2)$$

where M is metabolic rate per unit body surface area [W/m^2], C is convective heat loss [W/m^2], R is radiative heat loss [W/m^2], E_{sk} is evaporative heat loss [W/m^2], E_{res} is convective and evaporative heat loss from respiration [W/m^2], W is external mechanical work [W/m^2] and S is rate of heat storage [W/m^2].

The heat transfer equations by convection C, radiation R [3], evaporation E_{sk} and respiration E_{res} [4] are given as the following equations.

$$C = h_cF_{cl}(T_{sk} - T_a)f_{cl} \qquad (4)$$

$$R = h_rF_{cl}[(T_{sk} + 273) - \lambda(T_{gr} + 273)]f_{cl}f_{ref} - F_{cl}(H_d + H_s + H_r)f_{cl} \qquad (5)$$

$$E_{sk} = LRh_cF_{pcl}(P_{sk} - P_a)f_{cl} \qquad (6)$$

$$E_{res} = 0.0014M(35 - T_a) + 0.0173M(5.87 - P_a) \qquad (7)$$

where h_c and h_r are human's convective and linear radiative heat transfer coefficients [W/(m^2°C)], F_{cl} is thermal efficiency factor [N.D.] [5], T_{sk} is mean skin temperature [°C], T_a is air temperature [°C], f_{cl} is clothing area factor[N.D.] [6], λ is long-wave radiation coefficient [N.D.] [3], T_{gr} is ground temperature [°C], f_{ref} is effective radiant area factor [N.D.], H_d, H_s and H_r are direct, scattered and reflective solar radiation absorbed into body surface [W/m^2], LR is Lewis relationship (= 16.5) [°C/kPa] , F_{pcl} is permeation efficiency factor [N.D.] [7], P_{sk} is water vapor pressure at skin temperature [kPa] and P_a is water vapor pressure in air [Pa].

λ in Equation (4), physical quantities expressing the properties of radiation specific to the region, date and time, are defined by the following equations:

$$\lambda = (1 + \Omega)^{0.25}(1 - U_r)^{0.25} \qquad (7)$$

$$\Omega = (T_{sky} + 273)^4U_r / (T_{gr} + 273)^4(1 - U_r) \qquad (8)$$

where Ω is ratio of atmospheric radiation to long-wave radiation from the ground to the human body [N.D.], T_{sky} is hypothetical sky temperature [°C], U_r is sky view factor for the human body [N.D.].

Also, Ω in Equation (7) expresses a ratio of atmospheric radiation to long-wave radiation from the ground to the human body. U_r in Equations (7) and (8) denote configuration factor between the human body and the sky, and is referred to as sky view factor for the human body [3].In this study, f_{ref} = 1 in equation (4).

Even in the outdoor environment with solar radiation, the wet bulb thermometer is essentially used for measurement in shaded areas, so there is no need to modify the general heat balance equation (9) for the wet bulb [2]. Because the sensing part of the wet bulb thermometer is very small, on the assumption that h_r'/h_c' is nearly equal to zero, Equation (9) leads to Equation (10).

$$h_c'(T_w - T_a) + h_r'(T_w - T_r) + LR\, h_c'(P_w^* - P_a) = 0 \qquad (9)$$

$$(T_w - T_a) + LR(P_w^* - P_a) = 0 \qquad (10)$$

where h_c' and h_r' are convective and linear radiative heat transfer coefficients of wet bulb [W/(m^2°C)] , T_r is mean radiant temperature [°C], P_w* is saturated water vapor pressure on wet bulb [kPa].

As far as a hot environment surrounding human work is concerned, the relationship of the saturated water vapor pressure P_w^* to the wet bulb temperature T_w on the psychrometric chart is sufficiently linear, and can be approximated as follows:

$$P_w^* = \kappa T_w + \zeta \qquad (11)$$

where κ and ζ are constants of linear approximation of saturated water vapor pressure to wet bulb temperature [kPa/°C].
During work in a hot environment, mean skin temperature is within a range of 35-40°C. Therefore, by treating skin temperature in the same way as the wet bulb temperature, the relationship of saturated water vapor pressure on skin P_{sk} to skin temperature T_{sk} can be linearly approximated by Equation (12).

$$P_{sk}^* = \kappa^* T_{sk} + \zeta^* \qquad (12)$$

where κ* and ζ* are constants of linear approximation of saturated water vapor pressure to mean skin temperature [kPa/°C].
In the forth coming numerical examination, we will adopt the following values shown in Reference [2]: $\kappa = \kappa^* = 0.279$ kPa/°C and $\zeta = \zeta^* = -4.03$ kPa.
Water vapor pressure on skin surface P_{sk} is expressed by Equation (12)′, accompanied with μ that indicates the degree of saturation on skin surface. Here we expand equations with μ preserved, but in the later stages, we will suppose μ = 1 because the WBGT index is designed for use within thermal limitation where water vapor pressure on skin surface is almost saturated in consequence of promoted sweating.

$$P_{sk} = \mu (\kappa^* T_{sk} + \zeta^*) \qquad (12)'$$

where μ is saturated ratio of water vapor pressure to skin temperature [N.D.].
The globe thermometer is basically intended for measuring heat radiation, and steady-state heat balance after sufficient exposure of the globe thermometer to an outdoor environment with solar radiation for actual measurement is expressed, with reference to heat transfer equation by radiation for the human body (4), as follows:

$$h_c''(T_g - T_a) + h_r'' F_{cl}'' [(T_g+273) - \lambda''(T_{gr}+273)] f_{cl}'' - F_{cl}'' (H_d''+H_s''+H_r'') f_{cl}'' = 0 \qquad (13)$$

where h_c'' and h_r'' are convective and linear radiative heat transfer coefficient of globe [W/(m^2°C)], λ″ is long-wave radiation coefficient of globe [N.D.], F_{cl}'' is thermal efficiency factor of globe [N.D.], f_{cl} is clothing area factor of globe [N.D.], H_d'', H_s'' and H_r'' are direct, scattered and reflected solar radiation absorbed into globe [W/m^2].
For the globe thermometer, $F_{cl}''=1$ and $f_{cl}''=1$. In this study, $\lambda''=\lambda$.

RESULTS

Substituting the relevant equations including Equations (3) ~ (8) and (10) ~ (13) into heat balance equation of the human body (2) gives the outdoor WBGT formula (14) in conformity with the indoor WBGT formula [2].

$$\begin{aligned}
&\left[(h_c + h_r)F_{cl}f_{cl} + \mu\kappa^* LRh_c F_{pcl} f_{cl}\right]T_{sk} + \left[F_{cl}f_{cl}\left(H_d'' + H_s'' + H_r''\right)\frac{\lambda h_r}{\lambda'' h_r''} - F_{cl}f_{cl}\left(H_d + H_s + H_r\right)\right.\\
&\left. + \left(\mu\zeta^* - \zeta\right)LRh_c F_{pcl} f_{cl} - M(0.8494 + 0.0173\zeta) + W + S\right]\\
&= \left[h_c F_{pcl} f_{cl}(1 + \kappa \cdot LR) + 0.0173M\left(\kappa + \frac{1}{LR}\right)\right]T_w\\
&+ \left[h_r F_{cl} f_{cl}\frac{\lambda}{\lambda''}\left(1 + \frac{h_c''}{h_r''}\right)\right]T_g\\
&+ \left[F_{cl}f_{cl}\left(h_c - h_c''\frac{\lambda h_r}{\lambda'' h_r''}\right) - h_c F_{pcl} f_{cl} + M\left(0.0014 - \frac{0.0173}{LR}\right)\right]T_a
\end{aligned} \qquad (14)$$

Writing the coefficient of T_{sk} and the constant term on the left-hand side of Equation (14) as ξ and η, respectively, and the coefficients of T_w, T_g and T_a as α, β and γ, we obtain equations (15) ~ (20).

$$\left[(h_c + h_r)F_{cl}f_{cl} + \mu\kappa^* LRh_c F_{pcl}f_{cl}\right] \equiv \xi \tag{15}$$

$$\begin{aligned}&\left[F_{cl}f_{cl}\left(H_d'' + H_s'' + H_r''\right)\frac{\lambda h_r}{\lambda'' h_r''} - F_{cl}f_{cl}\left(H_d + H_s + H_r\right)\right.\\ &\left. + \left(\mu\zeta^* - \zeta\right)LRh_c F_{pcl}f_{cl} - M\left(0.8494 + 0.0173\zeta\right) + W + S\right] \equiv \eta\end{aligned} \tag{16}$$

$$\left[h_c F_{pcl}f_{cl}\left(1 + \kappa \cdot LR\right) + 0.0173M\left(\kappa + \frac{1}{LR}\right)\right] \equiv \alpha \tag{17}$$

$$\left[h_r F_{cl}f_{cl}\frac{\lambda}{\lambda''}\left(1 + \frac{h_c''}{h_r''}\right)\right] \equiv \beta \tag{18}$$

$$\left[F_{cl}f_{cl}\left(h_c - h_c''\frac{\lambda h_r}{\lambda'' h_r''}\right) - h_c F_{pcl}f_{cl} + M\left(0.0014 - \frac{0.0173}{LR}\right)\right] \equiv \gamma \tag{19}$$

$$\xi T_{sk} + \eta = \alpha T_w + \beta T_g + \gamma T_a \tag{20}$$

Dividing both sides of Equation (20) by ξ yields Equation (21), a form based on the skin temperature T_{sk}.

$$T_{sk} + \left(\frac{\eta}{\xi}\right) = \left(\frac{\alpha}{\xi}\right)T_w + \left(\frac{\beta}{\xi}\right)T_g + \left(\frac{\gamma}{\xi}\right)T_a \tag{21}$$

Equation (21), on removing the physical quantity related to solar radiation, corresponds to the WBGT formula for indoor use [2].

We examine the varying properties of the variable coefficients in Equation (21) that correspond to the constant coefficients in the original WBGT formula (1) for outdoor conditions with solar radiation, substituting concrete values. We calculated the value of η/ξ, α/ξ, β/ξ and γ/ξ, changing metabolic rate from 1 to 4 met, clo unit from 0.2 to 1.0 clo and outdoor air velocity from 1.0 to 5.0 m/s with global solar radiation set at 600 W/m^2.
The calculated values of η/ξ ranged from –10.0 to 2.3. If air velocity, clo unit and metabolic rate were low, the η/ξ value was positive.
Figures 1 to 3 show the results of calculation of the α/ξ, δ/ξ and γ/ξ values. Figure 1 shows the variation of each coefficient in response to the variation of metabolic rate M with clo unit and air velocity held constant. The horizontal axis indicates metabolic rate, and the vertical axis indicates the α/ξ, β/ξ, γ/ξ and α/ξ + β/ξ + γ/ξ values. Responding to metabolic rates of 1 to 4 met, α/ξ ranges from 0.81 to 0.86, β/ξ is constant at 0.33 and γ/ξ is approximately –0.12. Thus, the values of α/ξ, β/ξ and γ/ξ are roughly 0.84, 0.33 and –0.12 rather than 0.7, 0.2 and 0.1, the coefficients in the original formula (1). The sum of α/ξ, β/ξ and γ/ξ ranges from 1.02 to 1.07, slightly exceeding 1.0. It is remarkable in the numerical examination in Figure 1, also in the following examinations in Figure 2 and 3, that the mean value of the coefficient γ/ξ of T_a is negative. The value shifts to the positive side when air velocity exceeds a certain value.
Figure 2 shows the variation of the coefficients in response to the variation of air velocity (horizontal axis) with metabolic rate and clo unit held constant. The values of α/ξ, β/ξ and γ/ξ are, respectively, 0.84 ~ 0.82, 0.33 ~ 0.24 and –0.12 ~ –0.02, as in Figure 1. In Figure 2, unlike Figure 1, the α/ξ and β/ξ values decrease, when the sum of the three coefficients approaches 1.0.
Similarly, Figure 3 shows the variation of the coefficients in response to the variation of clo unit with metabolic rate and air velocity held constant. The values of α/ξ, β/ξ and γ/ξ are 0.86

~ 0.84, 0.24 ~ 0.29 and –0.08 ~ –0.05, respectively. As clothing insulation increases, the α/ξ values decreases and the β/ξ and γ/ξ values increase. The sum of α/ξ, β/ξ and γ/ξ increases gradually from 1.02 to 1.07.

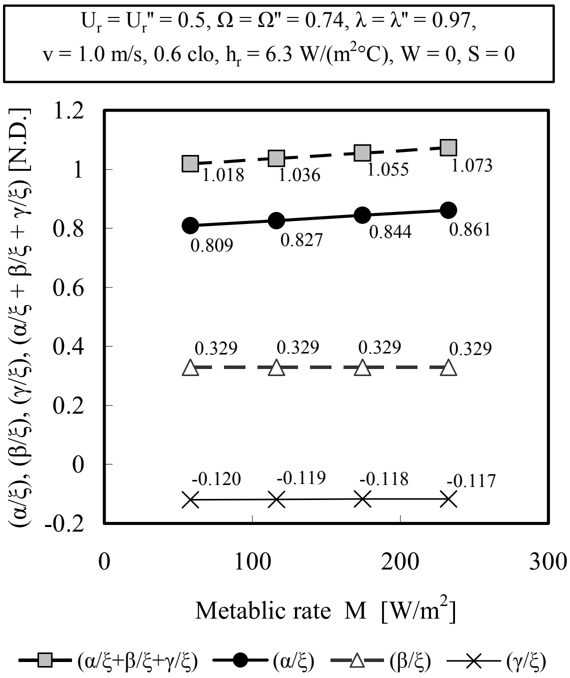

Figure 1 The variation of each coefficient in eq.(21) in case of variable metabolic rate with constant clo unit and air velocity.

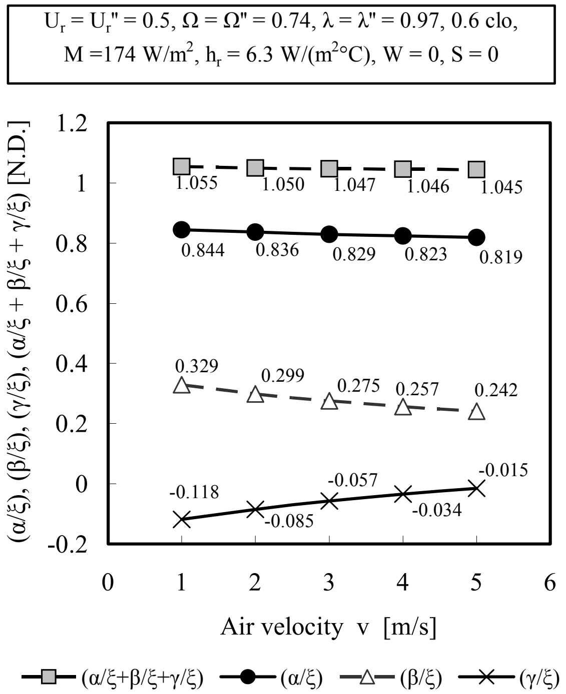

Figure 2 The variation of each coefficient in eq.(21) in case of variable air velocity with constant clo unit and metabolic rate.

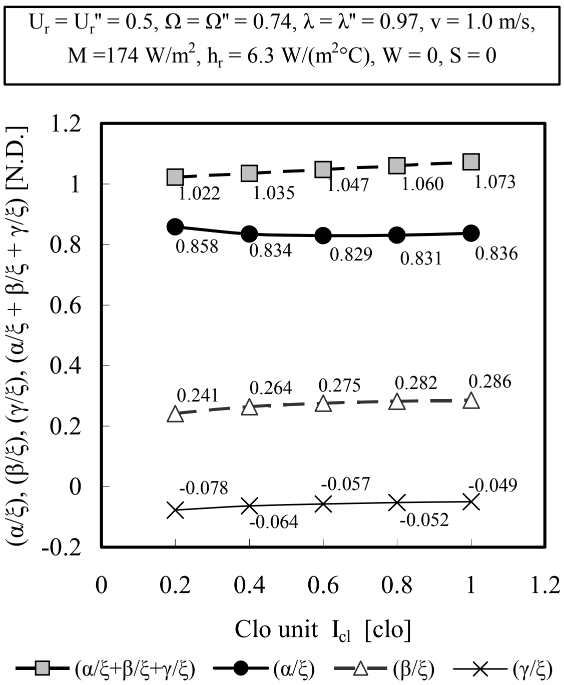

Figure 3 The variation of each coefficient in eq.(21) in case of variable clo unit with constant metabolic rate unit and air velocity.

DISCUSSION

We consider points of difference and similarity between the derived formula for outdoor conditions and that for indoor conditions [2]. The derived formula for indoor conditions in Reference [2] is shown as follows:

$$\begin{aligned}
&\left[(h_c + h_r)F_{cl}f_{cl} + \mu\kappa^* \cdot LRh_cF_{pcl}f_{cl}\right]T_{sk} \\
&+\left[(\mu\zeta^* - \zeta)LR \cdot h_cF_{pcl}f_{cl} - M(0.8494 + 0.0173\zeta) + W + S\right] \\
&= \left[h_cF_{pcl}f_{cl}(1 + \kappa \cdot LR) + 0.0173M\left(\kappa + \frac{1}{LR}\right)\right]T_w \\
&+\left[F_{cl}f_{cl}\left(h_c'' + h_r\right)\right]T_g \\
&+\left[F_{cl}f_{cl}\left(h_c - h_c''\right) - h_cF_{pcl}f_{cl} + M\left(0.0014 - \frac{0.0173}{LR}\right)\right]T_a
\end{aligned} \tag{22}$$

Comparing the coefficients of T_w, T_g and T_a in Equation (14) with those in Equation (22), the coefficient of first term on the left- and right-hand side is the same between two equations. The coefficient of second term on the left-hand side is the same constituent elements between two equations except solar radiation term. Meanwhile, the second term on the right-hand side in equation (14) includes the long-wave radiation coefficient, and third term in equation (14) includes the elements with respect to globe thermometer. Other elements of the coefficients in two equations are equally common. Examination using concrete values in Equation (22) provides that coefficient values in two equations were nearly equal.

Examination of the derived formula confirmed that the coefficients of T_w, T_g and T_a in the derived WBGT formula, considering their structures, are not strictly constant but variable depending on metabolic rate, clothing insulation, air velocity and other such factors.

We calculated the concrete values of each coefficient in Equation (14), changing metabolic rate, clothing insulation and air velocity. The calculated values of η/ξ ranged from –10.0 to 2.3. We consider the effect of solar radiation on the η/ξ qualitatively. Assuming that $\mu = 1$ in Equation (16), the inequality, which turns into $\eta \geq 0$, is expressed as follows:

$$\frac{F_{cl}f_{cl}\left[\left(H_d'' + H_s'' + H_r''\right) - \left(H_d + H_s + H_r\right)\right] + W + S}{M} \geq 0.781 \tag{23}$$

Equation (23) shows that $\eta \geq 0$ is obtained if the sum of the solar radiation H, external work W and heat storage S for the metabolic rate M is greater than 0.781. The difference between the first and second terms in the left-hand side of Equation (23) represents the difference between solar radiation absorbed by the globe thermometer and that absorbed by the human body. This difference, dependent on solar absorptance, projected area factor and configuration factor, is positive because solar absorptance of the human body is generally less than that of the globe thermometer. This indicates that in an outdoor environment with solar radiation, the η/ξ value may be positive even without external work or heat accumulation, and also that in an environment without solar radiation, the η/ξ value cannot be positive without them.

The calculation showed the coefficients α/ξ, β/ξ and γ/ξ of T_w, T_g and T_a to be 0.81 ~ 0.86, 0.25 ~ 0.33 and -0.12 ~ –0.02, respectively, roughly 0.84, 0.30 and –0.08 rather than 0.7, 0.2 and 0.1 the coefficients in the original formula. The coefficient γ/ξ of T_a is negative value for low air velocities and positive for high air velocities, and that lower air velocities resulted in larger negative values. Negative values, seen for low air velocities, mean a relatively small influx of dry heat by convection and radiation from an environment and the cooling of skin due to sweat evaporation, whereas positive values, seen for high air velocities, indicate a relatively large influx of dry heat from the environment which, negating the effect of sweat evaporation, raises the skin temperature T_{sk}.

The meteorological conditions included in the outdoor WBGT formula depend on the region, date and time. In particular, solar radiation and air velocity vary significantly depending on the place and hours of work or exercise. Consequently, it is safer for the use of the WBGT to adopt the coefficients of T_w, T_g and T_a specific to the region, date and time than to adopt the constant coefficients suggested by Yaglou et al.

CONCLUSIONS

We derive theoretically the outdoor WBGT formula, which was originally developed empirically, based on the heat balance equation between the human body and an outdoor environment. We clarify the inner structures of the three constant coefficients of wet-bulb temperature, globe temperature and air temperature, and we perform an examination using real values. Examination of the outdoor WBGT formula using concrete values provides the new formula $WBGT = 0.84T_w + 0.30T_g - 0.08T_a$, compared to the original formula $WBGT = 0.7T_w + 0.2T_g + 0.1T_a$ of Yaglou and Minard. Based on the obtained results, we present the characteristics of the WBGT as an index as well as instructions for use.

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Effective Thermal Insulation of Body Segments by Summer Clothing

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SUMMARY

The aim of this study is to create a database of the effective thermal insulation provided to the body segments by various types of summer clothing ($R_{cle\mathrm{i}}$). In a climatic chamber with calm air conditions ($v \leq 0.15$ m/s), $R_{cle\mathrm{i}}$ was obtained for 52 garments and 97 ensembles. In the measurements, a female thermal manikin that had twenty segments was arranged in a sitting posture. In the chamber, operative temperature was set to 27.0°C, 25.0°C, and 24.0°C for a nude manikin, a manikin clothed in a garment, and one clothed in an ensemble, respectively. $R_{cle\mathrm{i}}$ were calculated for the nine segments—the chest, back, abdomen, buttock, upper arms, forearms, thighs, legs, and feet. The common regression for all the body segments, $R_{cle\mathrm{i}.en} = 0.784\ \Sigma R_{cle\mathrm{i}.gr}$, was proposed for estimating the $R_{cle\mathrm{i}}$ for the ensembles ($R_{cle\mathrm{i}.en}$) from the sum of the components' $R_{cle\mathrm{i}}$ ($\Sigma R_{cle\mathrm{i}.gr}$).

INTRODUCTION

In order to use human thermal models with multiple segments, it is essential to obtain data on the thermal insulation provided to each body segment by a type of clothing. In previous studies [1, 2, 3, 4], these data were measured for several ensembles. However, the authors were unable to find the sufficient database for estimating the thermal insulation provided by various types of clothing. The aim of this study is to create a database of the effective thermal insulation provided to body segments by various types of summer clothing ($R_{cle\mathrm{i}}$).

METHODS

In order to express the thermal insulation of a clothing, three kinds of insulation—total insulation (I_T), intrinsic insulation (I_{cl}), and effective insulation (I_{cle})—are used [5] [NOTE1]. For expressing the thermal characteristics of a clothing, the usage of I_{cl} is recommended [5, 6] since it is the actual thermal resistance for a temperature difference between the skin and clothing surface. However, it has the following characteristics. (1) I_{cl} for the same clothing changes depending on the airflow around the human body and the body motions. (2) At present, it is difficult to set adequate heat transfer characteristics for the clothing surfaces in each segment. Considering these characteristics and the intended application of the clothing thermal insulation data on the various body segments as an input to the human thermal models with multiple segments, the use of I_{cl} instead of I_{cle} is considered to be less advantageous. Therefore, the present authors have described the clothing thermal insulation in terms of the effective thermal insulation.

In a climatic chamber with calm air conditions ($v \leq 0.15$ m/s), the effective thermal insulation of the body segment i, denoted by ($R_{cle\mathrm{i}}$) [1], was measured for summer clothing garments and ensembles. $R_{cle\mathrm{i}}$ is the hypothetical thermal resistance that expresses the effect of the addition/removal of clothing. It is calculated as follows.

$$\begin{aligned} R_{cle\mathrm{i}} &= R_{dress\mathrm{i}} - R_{nude\mathrm{i}} \\ &= \frac{Q_{dress\mathrm{i}}}{Tsk_{dress\mathrm{i}} - To_{dress\mathrm{i}}} - \frac{Q_{nude\mathrm{i}}}{Tsk_{nude\mathrm{i}} - To_{nude\mathrm{i}}}, \end{aligned} \tag{1}$$

where R is the total thermal resistance [m^{2o}C/W]; Q, sensible heat loss [W/m^2]; Tsk, skin temperature [ºC]; and To, operative temperature [ºC]. The suffixes are as follows: *dress*i is the value for segment i when the body is clothed and *nude*i is the value for segment i when the body is not clothed.

With regard to the thermal insulation for the overall body, all total insulations are calculated by the global method [7] from the data on the segments. Then, the effective thermal insulation I_{cle} is calculated as follows.

$$\begin{aligned} I_{cle} &= I_T - I_a \\ &= \frac{\sum\left(Q_{dress\mathrm{i}} \cdot W_\mathrm{i}\right)}{0.155\sum\left[\left(Tsk_{dress\mathrm{i}} - To_{dress\mathrm{i}}\right)\cdot W_\mathrm{i}\right]} - \frac{\sum\left(Q_{nude\mathrm{i}} \cdot W_\mathrm{i}\right)}{0.155\sum\left[\left(Tsk_{nude\mathrm{i}} - To_{nude\mathrm{i}}\right)\cdot W_\mathrm{i}\right]}, \end{aligned} \tag{2}$$

where I_a is the total thermal insulation when the body is not clothed [clo], and W_i is the weighting coefficient for the data on segment i that is set in accordance with area ratio [N.D.].

Table 1 Summary of garments (example)

Symbol	Product	Weight [g]	Material† [%]	Color
UW1	Short	79	C: 100	Blue
UW2	Panty	29	C: 100	Skin color
UW3	Brassiere	37	C: 100	Skin color
UW6	Socks (middle length)	61	C: 100	White
SU3	Camisole	98	C: 100	Light blue
SU4	Short sleeve T-shirt	185	C: 100	White
SU12	Short sleeve shirt	201	C: 60, PE: 40	Checked light blue
SU16	3/4 sleeve blouse	179	PE: 84, PU: 16	Light green
SU17	Long sleeve blouse	190	PE: 59, C: 34, PU: 7	White
SU18	Ensemble summer sweater (inner, set with SU19)	113	A: 100	Brown
SU19	Ensemble summer sweater (outer, set with SU18)	172	A: 100	Brown
SU25	One-piece dress	201	C: 95, PU: 5	Grey
SU30	Long sleeve work wear	234	PE: 60, C: 40	Grey
SD6	Trousers	449	C: 60, PE: 40	Grey
SD7	Trousers	440	C: 55, R: 40, PU: 5	Blue
SD8	Skirt	262	C: 68, PE: 28, PU: 4	Black
SD17	Working trousers	327	PE: 60, C: 40	Grey
SU26	Short sleeve T-shirt	170	C: 69, PE: 31	Grey
SD12	Short pants	187	C: 69, PE: 31	Grey
SU27	Short sleeve pajama	159	C: 100	Pink
SD14	Short pants of pajama	111	C: 100	Pink

† "C" is cotton, "PE" is polyester, "PU" is polyurethane, "A" is acrylate, and "R" is rayon.

Table 2 Summary of ensembles (example)

No.	Ensemble	Weight [g]
19	Short (UW1), socks (UW6), short pants (SD12), short sleeve T-shirt (SU26)	497
26	Short (UW1), trousers (SD6), short sleeve shirt (SU4), working wear (SU30)	947
36	Short (UW1), short sleeve pajama (SU27), short pants (SD14)	349
91	Panty (UW2), trousers (SD7), brassiere (UW3), 3/4 sleeve blouse (SU16)	685
102	Panty (UW2), socks (UW6), trousers (SD7), brassiere (UW3), camisole (SU3), 3/4 sleeve shirt (SU16), summer sweaters (SU18 and SU19)	1129
116	Panty (UW2), socks (UW6), skirt (SD8), brassiere (UW3), camisole (SU3)	487
151	Panty (UW2), socks (UW6), working trousers (SD17), brassiere (UW3), camisole (SU3), long sleeve blouse (SU17), working wear (SU30)	976
167	Panty (UW2), one piece dress (SU25)	230

Table 1 shows examples of the garments that were studied. Table 2 shows examples of the ensembles that were studied. For the measurements, a female thermal manikin that had twenty segments was arranged in the sitting posture. The manikin had an electrical heating source that was located just below its surface. The *Tsk* of each segment was maintained such that it conformed to the typical skin temperature distribution of human subjects who are in a thermally neutral state in a uniform environment [8]. The average *Tsk* for the overall body was 33.8°C. In the chamber, *To* was set to 27.0°C, 25.0°C, and 24.0°C for a nude manikin, a manikin clothed in a garment, and one clothed in an ensemble, respectively. The relative humidity was within 40 ± 20%. The data in the steady state were used for calculating $R_{cle\mathrm{i}}$ and I_{cle} in Eqs. (1) and (2). In these conditions, the measured Q from the overall body ranged from almost 30 to 70 W/m^2. I_a was 0.81 clo (= 0.13 m^{2o}C/W). $R_{cle\mathrm{i}}$ values were calculated for nine segments—the chest, back, abdomen, buttock, upper arms, forearms, thighs, legs, and feet. Measurements were carried out two or three times for each type of clothing. Two or three data sets were then obtained for each type of clothing. From these data sets, two that satisfied the condition of the difference in their $R_{cle\mathrm{i}}$ values being less than 0.031 m^{2o}C/W (= 0.2 clo) were selected. The average of these two selected data sets was then obtained as the final output of the measurements.

RESULTS AND DISCUSSIONS

Tables 3 and 4 show the examples of the obtained results. The $R_{cle\mathrm{i}}$ values for 52 garments and 97 ensembles were obtained through the selection of the data sets. In this selection, the rejections were mainly attributed to the large difference in $R_{cle\mathrm{i}}$ in the abdomen and thighs. We consider the reasons for the large differences in these segments to be as follows. In the sitting posture, there was a strong curvature along these two segments on the front side of the body. Since the clothing on these segments was located inside this curvature, the thickness of the air layer inside the clothing was largely influenced by the magnitude of the curvature. Therefore, a slight difference in the posture resulted in a large difference in $R_{cle\mathrm{i}}$. In addition, since many garments overlapped on these segments, the errors in the manner of dressing appeared were more apparent in these segments compared with the others. Even in the selected data, with regard to some ensembles in which the legs were covered only by pantyhose, $R_{cle\mathrm{i}}$ in the legs took negative values (≥ –0.005 m^{2o}C/W). We consider that the level of accuracy in the measurements was insufficient for detecting the effect of the addition/removal of the pantyhose.

Table 3 R_{clei} and I_{cle} values for garments (example)

Symbol	Chest	Back	Abdomen	Buttock	Upper arms	Fore-arms	Thighs	Legs	Feet	Overall
					R_{clei} [m^2°C/W]					I_{cle} [clo]
UW1	0	0	0.113	0.065	0	0	0.021	0	0	0.06
UW2	0	0	0.030	0.042	0	0	0	0	0	0.03
UW3	0.042	0.014	0	0	0	0	0	0	0	0.03
UW6	0	0	0	0	0	0	0	0.021	0.031	0.04
SU3	0.068	0.048	0.064	0.028	0	0	0	0	0	0.09
SU4	0.113	0.121	0.123	0.059	0.076	0	0	0	0	0.19
SU12	0.134	0.137	0.106	0.068	0.144	0	0.028	0	0	0.24
SU16	0.117	0.132	0.088	0.053	0.086	0.040	0.002	0	0	0.18
SU17	0.116	0.124	0.105	0.059	0.075	0.085	0.001	0	0	0.21
SU18	0.086	0.091	0.088	0.030	0	0	0	0	0	0.08
SU19	0.074	0.103	0.046	0.036	0.068	0.023	0	0	0	0.14
SU25	0.099	0.028	0.119	0.058	0	0	0.066	0	0	0.15
SU30	0.156	0.132	0.090	0.053	0.130	0.073	0.011	0	0	0.24
SD6	0	0	0.143	0.092	0	0	0.043	0.084	0	0.12
SD7	0	0	0.121	0.055	0	0	0.032	0.050	0	0.09
SD8	0	0	0.110	0.064	0	0	0.075	0.020	0	0.10
SD17	0	0	0.129	0.077	0	0	0.061	0.079	0	0.13
SU26	0.113	0.118	0.120	0.046	0.075	0	0	0	0	0.14
SD12	0	0	0.143	0.047	0	0	0.037	0	0	0.06
SU27	0.133	0.131	0.110	0.055	0.104	0	0.004	0	0	0.18
SD14	0	0	0.118	0.059	0	0	0.045	0	0	0.07

Table 4 R_{clei} and I_{cle} values for ensembles (example)

No.	Chest	Back	Abdomen	Buttock	Upper arms	Fore-arms	Thighs	Legs	Feet	Overall
					[m^2°C/W]					[clo]
19	0.094	0.107	0.402	0.130	0.063	0	0.045	0.010	0.024	0.25
26	0.215	0.185	0.332	0.156	0.181	0.072	0.072	0.098	0	0.50
36	0.110	0.105	0.308	0.156	0.089	0	0.063	0	0	0.24
91	0.135	0.123	0.175	0.085	0.068	0.031	0.028	0.040	0	0.27
102	0.262	0.273	0.316	0.161	0.144	0.042	0.056	0.065	0.026	0.47
116	0.094	0.060	0.218	0.122	0	0	0.094	0.042	0.034	0.27
151	0.294	0.259	0.323	0.163	0.172	0.153	0.082	0.080	0.025	0.60
167	0.100	0.044	0.127	0.096	0	0	0.083	0	0	0.19

With regard to the value for the overall body, I_{cle} ranged from 0.02 clo to 0.24 clo for the garments and from 0.13 clo to 0.60 clo for the ensembles. Fig. 1 shows a plot of I_{cle} for ensembles ($I_{cle.en}$) against the sum of the components' I_{cle} ($\Sigma I_{cle.gr}$). The slopes of the regressions of $I_{cle.en}$ by $\Sigma I_{cle.gr}$ (indicated in Fig. 1) are slightly less than those reported by McCullough et al. ($I_{cle.en}$ = 0.76$\Sigma I_{cle.gr}$ + 0.079, $I_{cle.en}$ = 0.84$\Sigma I_{cle.gr}$) [9]. The Difference in the posture (standing in study [9] and sitting in the present study) and difference in the clothing type (only summer clothing is used in our study) are possible explanations for this discrepancy. In order to ascertain the reasons for this discrepancy, it is necessary to measure

I_{cle} of the same clothing as used in our study in the standing posture and also to measure I_{cle} of various types of clothing.

Table 5 shows the ranges of the calculated R_{clei} for the garments and ensembles. R_{clei} was large in the trunk and was relatively small in the periphery of the limbs. Fig. 2 shows a plot of

Table 5 Ranges of calculated R_{clei}

		Chest	Back	Abdomen	Buttock	Upper arms	Forearms	Thighs	Legs	Feet
						[m²°C/W]				
Garments	Min	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
	Max	0.162	0.137	0.150	0.092	0.144	0.085	0.075	0.084	0.034
Ensembles	Min	0.082	0.042	0.127	0.075	0.000	–0.001	0.019	–0.005	0.000
	Max	0.315	0.303	0.405	0.216	0.181	0.153	0.133	0.098	0.037

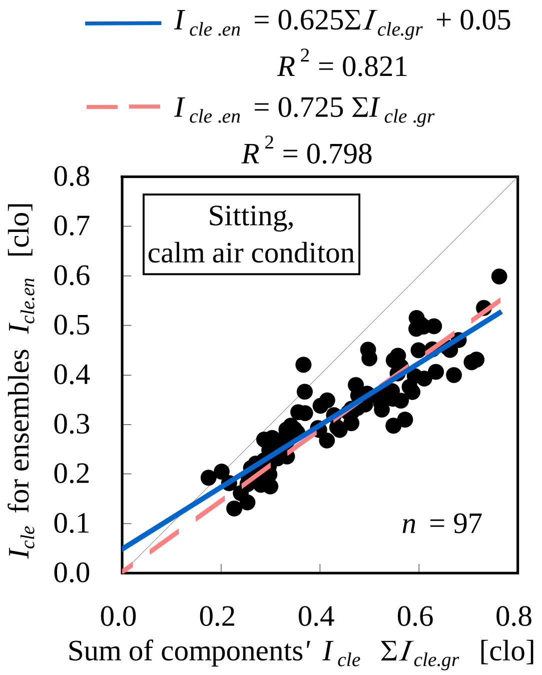

Fig. 1 $I_{cle.en}$ vs. $\Sigma I_{cle.gr}$ for the overall body

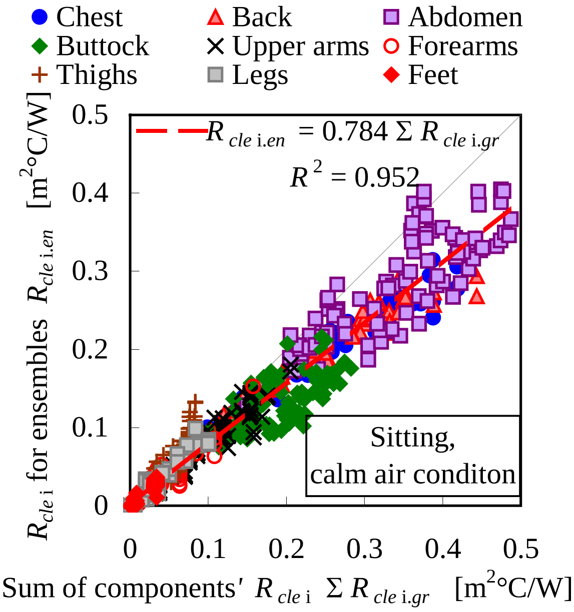

Fig. 2 $R_{clei.en}$ vs. $\Sigma R_{clei.gr}$ for nine segments

Table 6 Slope (A), intercept (B), standard deviation of error (sd) for regressions of $R_{clei.en}$ by $\Sigma R_{clei.gr}$

$R_{clei.en} = A\Sigma R_{clei.gr} + B$		Chest	Back	Abdomen	Buttock	Upper arms	Fore-arms	Thighs	Legs	Feet
$B \neq 0$	A	0.667	0.672	0.659	0.525	0.823	0.838	1.042	0.886	0.840
	B	0.0248	0.0259	0.0572	0.0309	–0.0015	–0.0007	0.0002	–0.0016	–0.0002
	sd	0.012	0.014	0.036	0.023	0.011	0.008	0.014	0.006	0.004
	A					0.763				
	B					0.0049				
	sd	0.015	0.017	0.041	0.032	0.012	0.009	0.020	0.008	0.005
$B = 0$	A	0.761	0.773	0.817	0.678	0.811	0.829	1.045	0.862	0.833
	sd	0.016	0.018	0.039	0.024	0.011	0.008	0.014	0.006	0.004
	A					0.784				
	sd	0.016	0.019	0.041	0.031	0.012	0.008	0.022	0.007	0.004

the R_{clei} values for the ensembles ($R_{clei.en}$) against the sum of the components' R_{clei} ($\Sigma R_{clei.gr}$) for nine segments. In each segment, $R_{clei.en}$ tends to be slightly less than $\Sigma R_{clei.gr}$. The plot exhibits an approximately similar relationship for the all segments. Table 6 shows the slope (A), intercept (B), standard deviation of error (sd) [9] [NOTE2] for the regressions of $R_{clei.en}$ by $\Sigma R_{clei.gr}$ for each segment and the values common to all the segments. In the individual regressions with an intercept, large sd values were observed for the abdomen (0.036 m^2°C/W) and buttock (0.023 m^2°C/W). Many overlaps of the garments explain the larger sd in these segments in comparison to the sd in the other segments. When an sd of 0.041 m^2°C/W (slightly larger sd than that in the abdomen of 0.036 m^2°C/W) is regarded as acceptable, the common regression ($R_{clei.en} = 0.784\Sigma R_{clei.gr}$) can be used for the nine segments.

Fiala et al. [10] developed a clothing insulation model of in which the intrinsic thermal insulation of a body segment provided by the clothing ensemble ($R_{cli.en}$) is expressed by the sum of the components' intrinsic clothing thermal insulation of the body segment ($\Sigma R_{cli.gr}$). However, the definition of clothing insulation in their model is different from that used in our study; the slope of our regression (0.784) is much less than unity (used in their model). The overlap of garments compresses the air layer inside the clothing. This can explain why the slope of our regression becomes less than unity. In the previous study of Sakoi et al. [2], the relations between $R_{cli.en}$ and $\Sigma R_{cli.gr}$ were different among the body segments. However, their definition of clothing insulation was different from that in the present study and their data set was smaller. On the other hand, the variety of clothing types was large as compared with the summer clothing used in the present study. Therefore, the accuracy of our common regression may reduce significantly when the regression is applied for estimating the R_{clei} provided by the types of clothing other than summer clothing.

The slope of the common regression for the nine segments becomes 0.784 and is larger than the slope of regression for the overall body ($I_{cle.en} = 0.725\Sigma I_{cle.gr}$ in Fig. 1). With the insulation for the overall body, it is reported that the effect of addition/removal of clothing garments becomes less when the clothing is worn nonuniformly [9]. The fact that the slope in the common regression for nine segments is larger than that for the overall body is explained by the same reason, i.e., the clothing distribution is more uniform for segments than it is for the overall body.

CONCLUSIONS

In a climatic chamber with calm air conditions ($v \leq 0.15$ m/s), the effective thermal insulation provided to body segments (R_{clei}) by summer clothing was measured. R_{clei} of 52 garments and 97 ensembles was obtained for nine segments—the chest, back, abdomen, buttock, upper arms, forearms, thighs, legs, and feet. In each segment, R_{clei} for the ensembles ($R_{clei.en}$) tends to be slightly less than the sum of the components' R_{clei} ($\Sigma R_{clei.gr}$). Since the plot of $R_{clei.en}$ against $\Sigma R_{clei.gr}$ exhibits an approximately similar relationship for all the segments, the common regression for all the body segments, $R_{clei.en} = 0.784\Sigma R_{clei.gr}$, was proposed for estimating $R_{clei.en}$ from $\Sigma R_{clei.gr}$.

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NOTE

[1] The total thermal insulation I_T is defined as the actual thermal resistance for a temperature difference between the skin and the operative temperature of the surrounding environment. It includes the thermal resistance of air (including the effect of radiation) in addition to that of the clothing itself. Therefore, it does not express the heat transfer characteristics of the clothing itself, but that of the clothing and the environmental conditions together [5, 6]. The intrinsic thermal insulation I_{cl} is the actual thermal resistance for a temperature difference between the skin and the clothing surface. It expresses the heat transfer characteristics of the clothing itself. The effective clothing insulation I_{cle} is the hypothetical thermal resistance that expresses the effect of the addition/removal of clothing. It is defined as the difference between the total thermal insulation when the body is clothed (I_T) and the total thermal insulation when the body is not clothed (I_a). Since this hypothetical resistance I_{cle} includes the effect of the environmental conditions as well as the clothing itself, it changes with the environmental conditions [5].

[2] The standard deviation of error (*sd*) [9] is calculated as follows.

$$sd = \sqrt{\frac{\sum(\text{predicted value} - \text{measured value})^2}{\text{number of data}}}$$

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Expression of the radiative heat exchange for the human body and its application to modifying the original WBGT for outdoor environment

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SUMMARY

In order to calculate mean radiant temperature in an outdoor environment, we had to calculate a hypothetical sky temperature from directly measurement or an empirical formula of atmospheric radiation expressed as functions of daily mean air temperature, cloud amount, etc. The aim of this research is to propose new expression for simply calculating the radiative heat exchange in an outdoor environment. In this paper, we defined the long-wave radiation coefficient based on the ratio of atmospheric radiation to long-wave radiation from the ground to the human body and derived linearized radiative heat exchange equation in an outdoor environment to be expressed only in the ground temperature and solar radiation. We qualitatively analyzed the long-wave radiation coefficient and the radiative heat transfer coefficient. Finally, based on measured data in an outdoor environment, we proposed the empirical formulae for calculating the long-wave radiation coefficient using solar radiation or the ground temperature.

INTRODUCTION

It is necessary to represent accurately heat exchanges by long-wave and short-wave radiations between the human body and an outdoor environment in order to evaluate thermal and comfort sensations of humans in outdoor environments. The authors have already proposed a series of effective radiant temperatures as mean radiant temperature in an outdoor environment [1] [2]. One is only based on the direct radiation between the human body and the surroundings [1], and the other is based on the multiple radiations between the human body and the surroundings [2]. The effective radiant temperature based on the direct radiation requires determining a hypothetical sky temperature by use of the empirical formula of downward atmospheric radiation defined as functions of daily mean air temperature, cloud amount, etc. Therefore, the calculation of the effective radiant temperature requires not only measured data but the meteorological data. The effective radiant temperature based on the multiple radiations requires calculating incidence factor to describe the influence of multiple radiations by use of linear simultaneous equation. Consequently, it is difficult for evaluation of an outdoor environment to calculate these temperatures on-site.
In this research, to simply calculate a radiative heat exchange in an outdoor environment, we define new long-wave radiation coefficient based on the ratio of atmospheric radiation to long-wave radiation from the ground to the human body, and propose new expression of the heat exchange by radiation between the human body and an outdoor environment considering solar radiation and the ground temperature. We qualitatively analyze the long-wave radiation coefficient, and propose concrete formulae for calculating the long-wave radiation coefficient by use of measured data in an outdoor environment.

METHODS

In an outdoor environment, heat exchange by long-wave radiation occurs between humans and the ground, the sky and body surfaces such as buildings. Besides, the outdoor radiative environment contains short-wave radiation, specifically, direct solar radiation, scattered solar radiation and reflected solar radiation from the ground and buildings.
The heat exchange R between humans and the environment containing short-wave and long-wave radiation is given as Equation (1). We assume the buildings' surface temperature is equal to the ground temperature and only consider reflected solar radiation from the ground.

$$\frac{R}{A_{cl}} = \varepsilon_{cl}\varepsilon_{gr}\sigma\left[(T_{cl}+273)^4-(T_{gr}+273)^4\right](1-U_r)f_{ref} + \varepsilon_{cl}\varepsilon_{sky}\sigma\left[(T_{cl}+273)^4-(T_{sky}+273)^4\right]U_r f_{ref} - (H_d+H_s+H_r) \quad (1)$$

$$H_d+H_s+H_r = a\{f_p I_{DN} + I_{SH}U_r + \rho_{gr}I_{TH}(1-U_r)\}f_{ref} \quad (2)$$

where R is radiative heat exchange [W], A_{cl} is surface area of clothed human [m^2], ε_{cl} is emissivity of clothed human body [N.D.], ε_{gr} is emissivity of ground [N.D.], σ is Stefan-Boltzmann constant (=5.67 * 10^{-8}) [$W/(m^2K^4)$], T_{cl} is mean surface temperature of clothed human [°C], T_{gr} is ground temperature [°C], U_r is sky view factor for the human body [N.D.], f_{ref} is effective radiant area factor [N.D.], ε_{sky} is emissivity of sky [N.D.], T_{sky} is hypothetical sky temperature [°C], H_d, H_s and H_r are direct, scattered and reflected solar radiation absorbed into body surface [W/m^2], a is absorptivity of human surface [N.D.], f_p is projected area factor [N.D.], I_{DN} is direct solar radiation to normal plane [W/m^2], I_{SH} is scattered solar radiation to horizontal plane [W/m^2], ρ_{gr} is reflectivity of ground [N.D.], I_{TH} is global solar radiation to horizontal plane [W/m^2].
Figure 1 shows atmospheric radiation and the long-wave radiation from the ground in an outdoor environment. Atmospheric radiation maintains a stable condition during any season. Long-wave radiation from the ground varies with the climate condition of day. Broken line in Figure 1 demonstrates the ratio of atmospheric radiation to long-wave radiation form the ground. The ratio is distributed between 0.5 and 0.7.

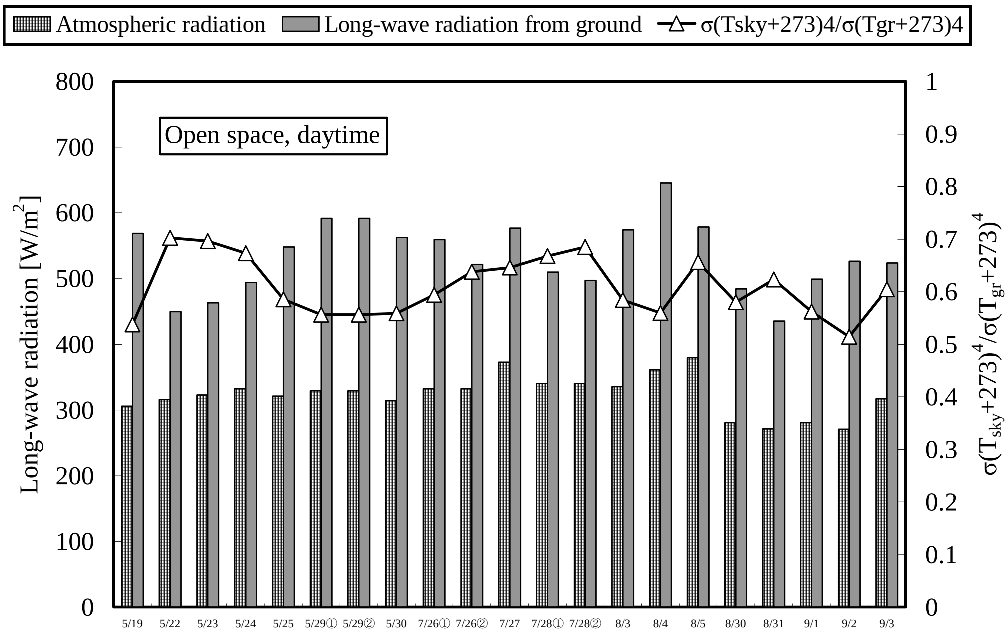

Figure 1. Long-wave radiations in an open space in the daytime of Sapporo city

We define the ratio Ω of atmospheric radiation to long-wave radiation from the ground to the human body as equation (3), and try to simplify the radiative heat exchange equation(1) between the human body and an outdoor environment by use of the ratio Ω.

$$\Omega = \frac{U_r \sigma (T_{sky} + 273)^4}{(1 - U_r)\sigma (T_{gr} + 273)^4} \qquad (3)$$

where Ω is ratio of atmospheric radiation to long-wave radiation from the ground to the human body [N.D.].
U_r in equation (3) denote configuration factor between the human body and the sky, and is referred to as sky view factor for the human body.

RESULTS

Substituting equation (3) into equation (1) to eliminate T_{sky} yields equation (4).

$$\begin{aligned}
\frac{R}{A_{cl}} &= \varepsilon_{cl}\sigma\left[(T_{cl}+273)^4 - \left\{(T_{gr}+273)^4(1-U_r) + (T_{sky}+273)^4 U_r\right\}\right] f_{ref} - (H_d + H_s + H_r) \\
&= \varepsilon_{cl}\sigma\left\{(T_{cl}+273)^4 - (1+\Omega)(T_{gr}+273)^4(1-U_r)\right\} f_{ref} - (H_d + H_s + H_r) \\
&= \varepsilon_{cl}\sigma\left\{(T_{cl}+273)^2 + (1+\Omega)^{\frac{1}{2}}(1-U_r)^{\frac{1}{2}}(T_{gr}+273)^2\right\} \\
&\quad \times\left\{(T_{cl}+273) + (1+\Omega)^{\frac{1}{4}}(1-U_r)^{\frac{1}{4}}(T_{gr}+273)\right\} \\
&\quad \times\left\{(T_{cl}+273) - (1+\Omega)^{\frac{1}{4}}(1-U_r)^{\frac{1}{4}}(T_{gr}+273)\right\} f_{ref} \\
&\quad - (H_d + H_s + H_r)
\end{aligned} \qquad (4)$$

Defining equation (5) makes it possible to define linear radiative heat transfer coefficient h_r in an outdoor environment as equation (6). λ in equation (5) expresses the long-wave radiation properties of the human body specific to the region, date and time, and is referred to as long-wave radiation coefficient.

$$(1+\Omega)^{\frac{1}{4}}(1-U_r)^{\frac{1}{4}} \equiv \lambda \qquad (5)$$

$$h_r \equiv \varepsilon_{cl}\sigma\left\{(T_{cl}+273)^2 + \lambda^2(T_{gr}+273)^2\right\}\left\{(T_{cl}+273) + \lambda(T_{gr}+273)\right\} \qquad (6)$$

where λ is long-wave radiation coefficient [N.D.], h_r is human’s linear radiative heat transfer coefficient [W/(m^2°C)].
The use of the radiative heat transfer coefficient h_r provides the linearized radiative heat exchange between the human body and an outdoor environment as follows:

$$\frac{R}{A_{cl}} = h_r\left\{(T_{cl}+273) - \lambda(T_{gr}+273)\right\} f_{ref} - (H_d + H_s + H_r) \qquad (7)$$

Finally, convective and radiative heat exchange equation considered clothing insulation R_{cl} is shown as follows:

$$\begin{aligned}
\frac{C+R}{A_{sk}} &= h_c(T_{sk} - T_a)F_{cl}f_{cl} \\
&\quad + h_r\left\{(T_{sk}+273) - \lambda(T_{gr}+273)\right\} f_{ref}F_{cl}f_{cl} - (H_d + H_s + H_r)F_{cl}f_{cl}
\end{aligned} \qquad (8)$$

where A_{sk} is skin surface area [m^2], C is convective heat loss [W], h_c are human’s convective heat transfer coefficient [W/(m^2°C)], T_{sk} is mean skin temperature [°C], T_a is air temperature [°C], F_{cl} is thermal efficiency factor [N.D.], f_{cl} is clothing area factor[N.D.].

If the long-wave radiation coefficients λ were preliminarily calculated in each region and time, the radiative heat exchange could be calculated using solar radiation and ground temperature.

DISCUSSION

We qualitatively analyze λ and h_r. λ means a correction factor to calculate the heat exchange by long-wave radiation using ground temperature only. Substituting equation (3) into equation (5) gives equation (5)'.

$$\lambda = \sqrt[4]{1-\left\{1-\frac{\sigma\left(T_{sky}+273\right)^4}{\sigma\left(T_{gr}+273\right)^4}\right\}U_r} \qquad (5)'$$

Because the term in brace on the right-hand side of equation (5)' is positive, the long-wave radiation coefficient λ increases with the decrease of the sky view factor for the human body U_r and is close to 1. Figure 2 shows the variation of λ in response to the variations of U_r and $\sigma(T_{sky} + 273)^4/\sigma(T_{gr} + 273)^4$. The long-wave radiation coefficient λ decreases with close to an open space, which means the increase of the sky view factor for the human body U_r, and the decrease of $\sigma(T_{sky} + 273)^4/\sigma(T_{gr} + 273)^4$.
Figure 3 shows the variation of linear radiative heat transfer coefficient h_r in response to that of the ground temperature T_{gr} and the long-wave radiation coefficient λ. The ground temperature T_{gr} in an outdoor environment ranges approximately from 0 to 50 °C and the radiative heat transfer coefficient h_r varies with the variation of λ. Therefore, it is difficult to regard h_r as a constant value in an outdoor environment unlike an indoor environment. The limitation of a temperature condition could regard h_r as a constant value.

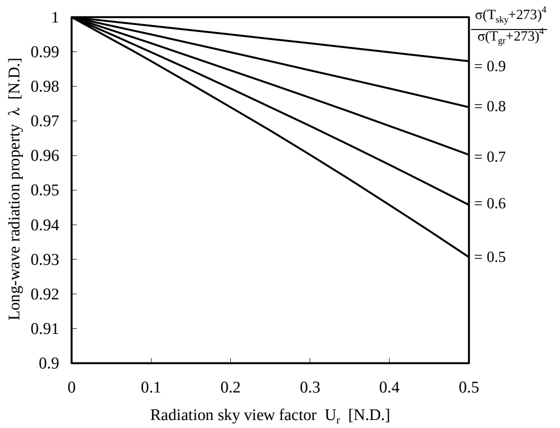

Figure 2. Variations of long-wave radiation coefficient λ

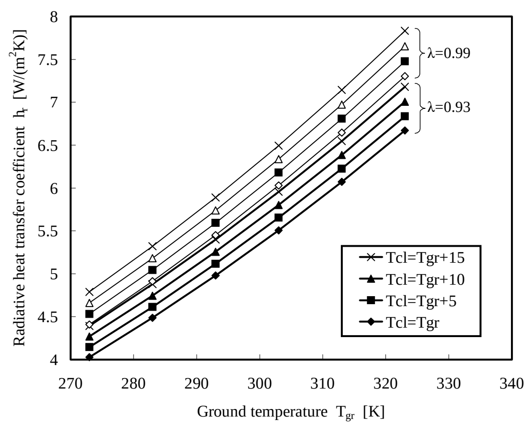

Figure 3. Variations of radiative heat transfer coefficient h_r

We determine concrete formulae for calculating the long-wave radiation coefficient λ using the data of human experiments. Human experiments for evaluating thermal and comfort sensation in an outdoor environment were carried out from 1999 to 2004 in Sapporo and Nagoya city of Japan [3]. The experiments were conducted on the roof of a six-storied building which could be considered an open space, in a courtyard surrounded by four buildings in Sapporo city, and in a street space near a three-storied building in Nagoya city. The sky view factor for the human body U_r of the open, courtyard and street space is, respectively, 0.5, 0.16 and 0.41. Experimental time was 20 or 30 min, and mean value during an experiment was used. We simultaneously measured ground temperature and solar radiation, and used for calculating the long-wave radiation coefficient λ. Atmospheric radiation was calculated from empirical formula of downward atmospheric radiation defined as functions of daily mean air temperature, cloud amount, etc.[4].

Figure 4 shows the variation of the long-wave radiation ratio $\sigma(T_{sky}+273)^4/\sigma(T_{gr}+273)^4$ in equation (5)' in response to that of global solar radiation. Rise in the ground temperature caused by increased solar radiation decreases the long-wave radiation ratio. There is no regional difference between Sapporo and Nagoya city. However, the range of the ratio is approximately 0.1 in response to same solar radiation. Therefore, the long-wave radiation ratio Ω of atmospheric radiation to the long-wave radiation from the ground to the human body depends on the season, solar radiation and sky view factor for the human body.

Figure 5 shows the variation of the long-wave radiation ratio in response to that of the ground temperature after sunset. Because the ground surface is exchanged for only the sky after dark, there is a proportional relation between the two. There is no difference in regions or spaces which has different sky view factor for the human body. Long-wave radiation ratio ranges from 0.64 to 0.9.

The relationship between global solar radiation and long-wave radiation coefficient λ in an open space of Sapporo city is shown in Figure 6. Increased solar radiation decreases the long-wave radiation coefficient λ, as in Figure 4. Therefore, the long-wave radiation ratio defined as a function of solar radiation or ground temperature makes it possible to calculate the long-wave radiation coefficient λ using solar radiation, the ground temperature and sky view factor for the human body U_r.

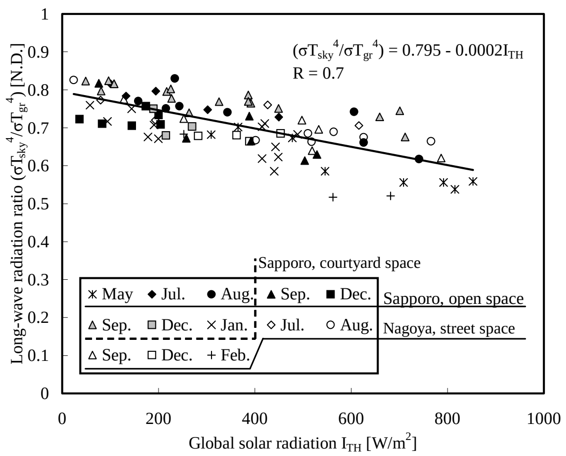

Figure 4. Relationship between global solar radiation and long-wave radiation ratio $\sigma(T_{sky}+273)^4/\sigma(T_{gr}+273)^4$ in daytime

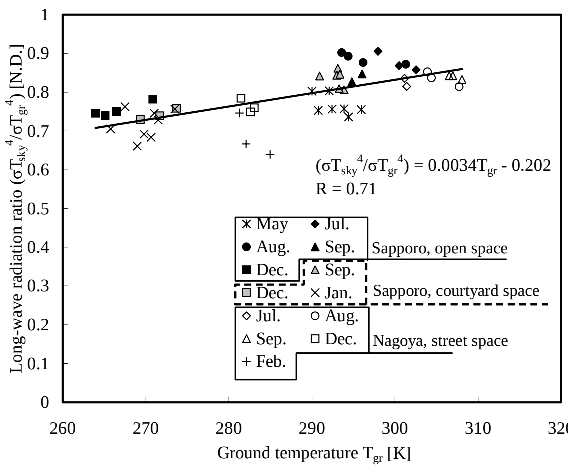

Figure 5. Relationship between ground temperature T_{gr} and long-wave radiation ratio $\sigma(T_{sky}+273)^4/\sigma(T_{gr}+273)^4$ at night

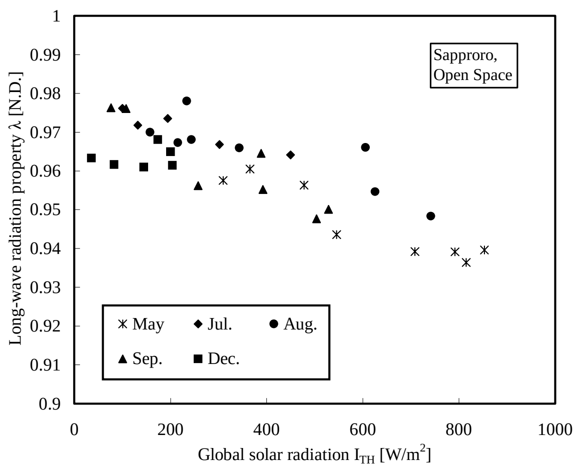

Figure 6. Relationship between global solar radiation and long-wave radiation coefficient λ

Equation (9) shows an empirical formula of the long-wave radiation ratio based on global solar radiation derived from Figure 4. Equation (10) shows an empirical formula of the long-wave radiation ratio based on the ground temperature derived from Figure 5.

$$\text{At daytime: } \frac{\sigma\left(T_{sky}+273\right)^4}{\sigma\left(T_{gr}+273\right)^4} = 0.795 - 0.0002 I_{TH} \qquad (9)$$

$$\text{After sunset: } \frac{\sigma\left(T_{sky}+273\right)^4}{\sigma\left(T_{gr}+273\right)^4} = 0.0035 T_{gr} - 0.202 \qquad (10)$$

Calculating λ using equations (9), (10) and sky view factor for the human body U_r makes it possible to calculate the heat exchange by radiation between the human body and an outdoor environment using λ, the ground temperature and solar radiation. However, these empirical formulae can be applied in moderate season because there are seasonal differences in Figure 4. Empirical formulae considered seasonal differences must be determined with detailed data seasonally.

CONCLUSIONS

In order to simply calculate a heat exchange by radiation between the human body and an outdoor environment, we propose a linear expression formula of radiative heat transfer based on a long-wave radiation coefficient of the human body. We qualitatively analyze the long-wave radiation coefficient and the radiative heat transfer coefficient, and derive concrete formulae for calculating the long-wave radiation coefficient from solar radiation, the ground temperature and the sky view factor for the human body using human experiments' data in an outdoor environment.

The linear expression formula can be applied to the modification of the original WBGT for an outdoor environment [5], based on a heat balance equation between the human body and an outdoor environment [6].

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Appropriate indoor climate for environmentally sustainable Supermarkets – Measurements and Questionnaires

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SUMMARY

Half the electricity use in supermarkets derives from the display of refrigerated food. Climate influences the performance of the cabinets and their energy use as well as the thermal comfort of people and the temperature quality of the food. This work is an interdisciplinary study of perceived comfort in relation to measured local climate. The aim is to define an appropriate climate and to improve energy efficiency with maintained or even better climate for goods, staff and customers. Perceived comfort depends on thermal balance of the whole body. ISO7730 is valid only for situations where people spend a longer time. This study involves temporary situation of supermarket shopping. Perception data have been obtained by means of a questionnaire based on ISO10551. The results show that according to ISO7730 people should be dissatisfied with their thermal comfort. The questionnaires (ISO10551), however, indicate that the thermal conditions are acceptable.

INTRODUCTION

In supermarkets there are three different categories that must be considered when a good climate should be established; goods, staff and customers. They are three different categories with incompatible requirements depending on many diverse factors. When installing energy efficient systems or improving existing installations it is important that it will be done without affecting any instance of the categories negatively. Refrigerated food is to be stored according to temperature intervals given by legislation which is harmonized within the EU. Staff in supermarkets is spending longer time in the environment and can adjust their dress-code according to the existing environment and their own activity. Customers on the other hand are and will be dressed in accord to the outdoor conditions which means that during summer while they are light-dressed they can be to cold in the supermarkets. However, the customers are in transit and are spending a shorter periods of time in the supermarkets compared to the goods and staff. The wish from the merchandiser on the other hand is to keep customers, as long as possible, within the sales area in order to sell as much as possible.

Supermarkets are large energy users and energy savings can be done through energy efficient equipment and systems but that is not enough as an argument for the merchandiser. There is a need of further arguments for the merchandisers and people involved when choosing the systems and technical solutions for supermarkets so that the energy use can be lowered with a result of less impact on the environment. An interdisciplinary analysis provides further arguments for the merchandiser to select energy efficient systems in the supermarket. The analysis highlights from an environmental and a financial point of view, advantages of energy efficient systems, improved thermal comfort and improved temperature quality of the food in the supermarkets. As defined by ANSI/ASHRAE Standard 55-2004, Thermal Environmental conditions for Human Occupancy, thermal comfort is “that condition of mind which expresses satisfaction with the thermal environment” [1]. The hypothesis is that energy efficient display cabinets and system solution, as well as increase knowledge, results both in an improved thermal comfort and better temperature quality of the food. The thermal comfort problem in supermarkets especially in front of display cabinets have been studied by several researchers by using CFD-modelling, e.g. Foster [2] and Foster and Quarini [3]. Their results also show that cold air spillage can be a problem. Cold air from the cabinet falls out of the cabinet on to the floor, causing the customer's feet to become cold, commonly known as the 'cold feet' effect. Local discomfort can be caused when one particular part of the body is exposed by unwanted cooling. A high vertical temperature difference between head and ankle is an example of local discomfort. Fang et al [4] studied both in laboratory and in controlled field experiments in an office room the effect of temperature and humidity on the perception of indoor air quality. Conclusions were that air temperature and humidity have a significant impact on both the immediate and the adapted perception of indoor air quality. Decreasing the indoor air temperature and humidity improved the perceived air quality significantly, the acceptability of air increased linearly with decreasing enthalpy of air. The study included different combinations of three levels of temperature (18 °C, 23 °C and 28 °C) and three levels of humidity (30%, 50% and 70%).

This paper presents measured air temperatures along with questionnaires answered by mainly customers in front of vertical cabinets during summer and winter. The objective of this paper is to present results from field measurements and describe how staff and customers are judging the thermal environment (temperature and overall indoor environment) by variations, summer and winter, in front of open vertical display cabinets. The long-term aim of the project is to define the desired indoor climate that would satisfy both people (staff and customers) and the quality requirements of the food. When the desired indoor climate is declared the aim is to improve energy efficiency in supermarkets with a maintained or even better climate for goods, staff and customers.

METHODS

Field measurements have been carried out in three different supermarkets, denoted A, B and C in Sweden. The size of the three supermarkets differs where A is the largest and C the smallest supermarket. In the three different supermarkets questionnaires have been collected in order to find out how the indoor climate was judged by the people in the supermarket. On the same time as people have responded to the questionnaires physical measurements on indoor parameters have been performed continuously. Questionnaires and measurements have been collected during a whole day (opening-hours) for summer (2005) and winter (2006) conditions in front of vertical display cabinets. In supermarket A, 425 people (34 staff) responded in supermarket B, responded 378 people (47 staff) and in supermarket C responded 405 people (49 staff). Totally 532 responded during the summer and 676 during the winter. Answers received were from 48% male and 52% female.

Presented in this paper is from physical measurement air temperatures in front of display cabinets. Together with questionnaires have people answered how they perceive, evaluate and prefers the temperatures. The presentation from the questionnaire also includes answers on how the overall indoor environment is judged.

SUPERMARKETS

Refrigerated food can be displayed in horizontal or vertical display cases/ cabinets with different design. The vertical display cabinets are common but are however more sensitive to infiltration and are also large users of electrical energy. Two common open vertical display cabinets as shown in Figure 1 are included in the study; (a) A Vertical open display cabinets for roll-in and back-loading of goods. It is mainly for storage of refrigerated food, dairy-goods. The cool air is distributed as an air curtain in the front of the cabinet and the rear is closed by curtains. The cold room behind is separated by curtains, the cold room is used for storage of goods. From the back is the cabinet loaded with new goods. (b) A Vertical open display cabinet, front-loaded, for storage of refrigerated food. The cool air is distributed through perforated plates in the rear and as an air curtain in the front of the cabinet. New goods are filled up from the front. One way for the merchandiser to increase the sales area and display refrigerated food in a larger volume for the customers is to use Cold rooms, Fig. 2. The cooled air inside the room is kept separated from the warmer air outside the room. Customers are transported through the room while purchasing the food. Inside the Cold room refrigerated food can not only be stored in vertical display cabinets but also as examples be kept on shelves, boxes and pallets on the floor.

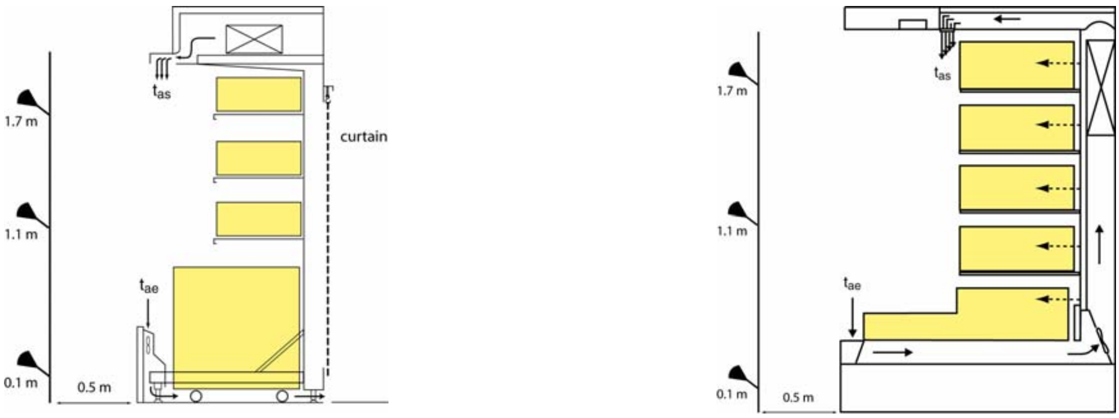


Figure 1: Open vertical display Cabinets and rack positioned 0.5m in front of the cabinet. The sensors are on levels 0.1m (ankle), 1.1m (abdominal) and 1.7m (head) (a) Roll-in and back loaded for Supermarket A, B and C (denoted A1, B1 inside a Cold room and C1) and (b) Front loaded for Supermarket A, B and C (denoted A2, B2 and C2).

OBJECTIVE MEASUREMENTS – PHYSICAL PARAMETERS

During the field measurements environmental parameters were collected from a rack with sensors vertical on three levels above the floor (Figure 1), recommended heights according EN ISO7726 [5]. The sampling intervals were 6 times/ hour and in each supermarket two different vertical display cabinets were measured. The refrigerated foods in the vertical cabinets were keeping temperatures between 0 °C and +8 °C. The Annual mean outdoor temperature for locations of the Supermarkets were 6 °C and during the measurements average temperatures for summer-condition was15 °C and winter-condition around 0 °C. Supermarket B has a Cold room for displaying refrigerated food for customers. In previous work by Axell and Lindberg [6] measurements shows that supermarket B uses the energy most efficient, measures as total energy consumption per square meter total or sales area, A and C have similar energy performance using (energy/sales are).

a)

b)

Figure 2: Photos from the field study a) Cold room in supermarket B b) Supermarket A and questionnaires answered in front of vertical display cabinet, A1. It is a Roll-in and back loaded vertical display cabinet

SUBJECTIVE MEASUREMENTS - QUESTIONNAIRES

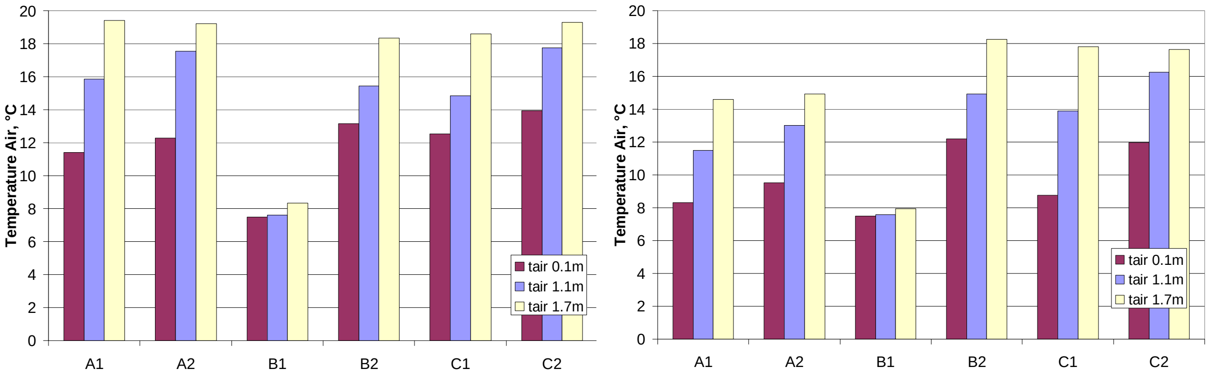
The thermal sensation can be predicted but existing thermal comfort standards, e.g. ISO7730 [7] are based on experiences where people spend longer time. The standard ISO10551 [8] covers the construction and use of judgment scales and proposes a set of specifications on direct experts assessment of thermal comfort/discomfort expressed by persons subjected to various degrees of thermal stress. This approach has been done in order to supplement the objective measurements with the aim of receiving reliable and comparative data on the subjective aspects of thermal comfort/discomfort. Judgment scales as proposed in the standard have used and questioned in following order;

- Perception scale - the scale of Perception is a 7-degree two-pole scale, comprising a central indifference point and two times 3 degrees of increasing intensity. The central point of indifference corresponds to the absence of hot and cold. Neutral = 4, very cold = 1 to very hot = 7.
- Evaluative scale - the Evaluative scale is a 5 degrees scale with a point of origin indicating the absence of the effect, and 4 degrees of increasing intensity of the comfort. Comfortable = 1 and 4 degrees of increasing intensity of the effect, extremely uncomfortable = 5 and
- Scale of preference - the Thermal preference scale is a symmetrical 7-degree scale bipolar comprising a central point of indecision and two times 3 degrees of increasing intensity. Preference scale 1-7, where neutral = 4, much colder = 1 to much warmer = 7.

The questionnaires have been answered by staff and customer standing 0.5 m in front of the vertical cabinet where also the rack has been positioned (Figure 1 and Figure 2). The persons interviewed have been informed briefly about reasons for the questionnaire. Together with backgrounds parameters such as activity, clothes, age etc other physical parameters concerning indoor environment have been collected from the responders. A total of five indoor environment parameters (e.g. air temperature, humidity and air velocity) with above three judgments scales have been questioned. Answers from questionnaire in this paper concerns the temperature and an overall question on how the acceptability is of the indoor environment (local climate) on a personal level.

RESULTS

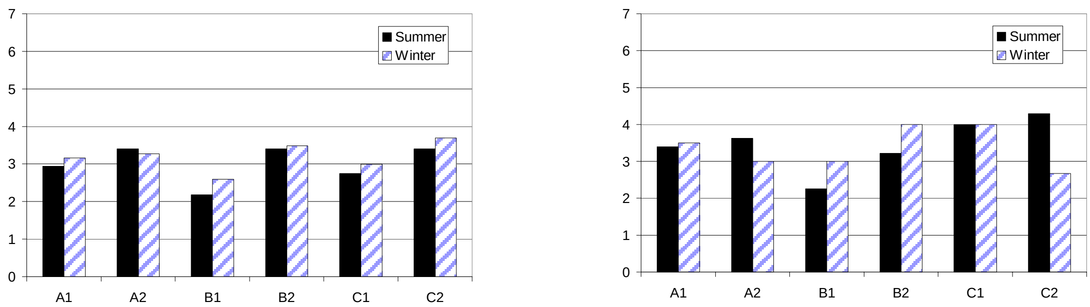
Depending on the outdoor conditions during the year there will also be an influence on the indoor environment. Ambient temperatures are depending on where and when it is measured as shown in Figure 3 with different levels presented during summer and winter conditions. The ambient temperature in front of the vertical cabinets is lower at all levels in all supermarkets during summer compared with winter. For supermarket B, B1, measured in a Cold room the temperature is keeping 8 °C on all three levels during summer and winter. In supermarket A there is a larger difference between summer and winter measurements compared with supermarket B, which also is using the energy more efficient.



a) b)

Figure 3. Average values, air temperatures (tair), during a) Summer and b) Winter. Levels on measurements 0.1, 1.1 and 1.7 m above the floor. For positions, A1-C2, see Figure 1.

The temperature is felt colder for the staff compared with the customers. For example when evaluated in front of A2 the average value for staff was 0.3 colder compared with the customers. Position A2 is measured where two vertical display cabinets are facing each others, during winter the temperature is perceived colder for staff (3.27) compared with customers (3.0). Inside the cold room, B1, lowest temperatures is evaluated from all positions for summer and winter.



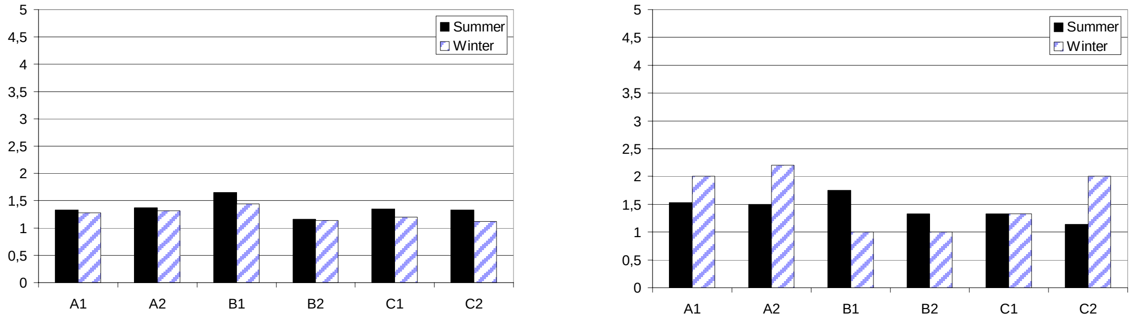
a) b)

Figure 4. Average values during Summer and Winter. For positions, A1-C2, see Figure 1.
a) Customer and b) Staff.
Question answered regarding the Temperature "How do you feel at this precise moment?"
Perceptions scale 1-7 where neutral = 4, very cold = 1 to very hot = 7.

The temperature, see below Table 1 and diagrams in Figure 5, is found to be more uncomfortable for staff than customers (comfortable = 0). During summer and winter more than 60 % of the customer and staff answered comfortable, an exception is B1, inside the cold room where 57 % answered comfortable during summer conditions. During summer 57 % of the customers answered comfortable compared with 70 % during winter conditions.

Table 1. Average values of preference scale for Customer and Staff during Summer and Winter (s/w). For positions, A1-C2, see Figure 1.
Question answered regarding Temperature "Do you find it….?" Scale 1-5, where comfortable = 0 and 4 degrees of increasing intensity of the effect, extremely uncomfortable = 5)

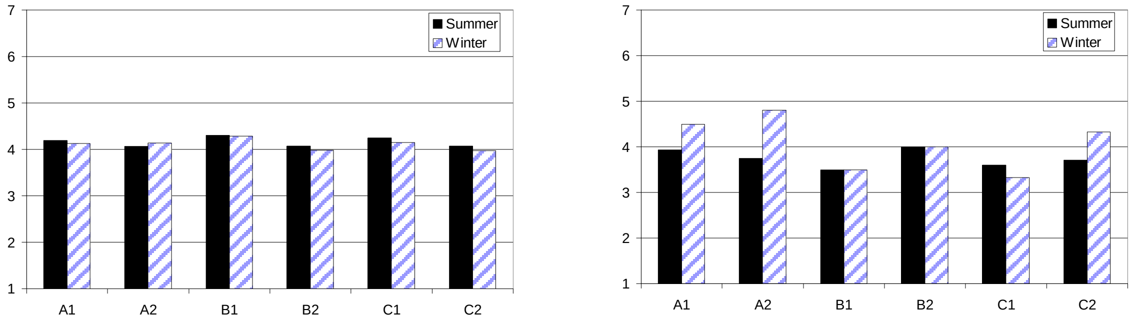
	A1 s/w	**A2 s/w**	**B1 s/w**	**B2 s/w**	**C1 s/w**	**C2 s/w**
Customer	1.33/ 1.28	1.37/ 1.32	1.65/ 1.44	1.16/ 1.14	1.35/ 1.20	1.33/ 1.12
Staff	1.53/ 2.00	1.50/ 2.20	1.75/ 1.80	1.33/ 1.00	1.33/ 1.33	1.14/ 2.00



a) b)

Figure 5. Average value during Summer and Winter. For positions, A1-C2, see Figure 1.
a) Customer and b) Staff.
Question answered regarding the Temperature "Do you find it ....?
Evaluative scale 1-5, where comfortable = 1 and 4 degrees of increasing intensity of the effect, extremely uncomfortable = 5)

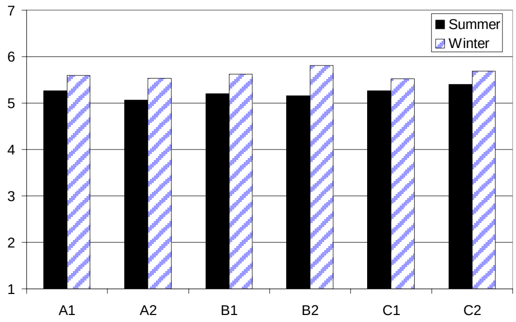
The temperature is preferred to be neither warmer nor colder (4=neutral) for customers. Variation of the preferred temperature is larger for the staff as shown in b) Figure 6 below.

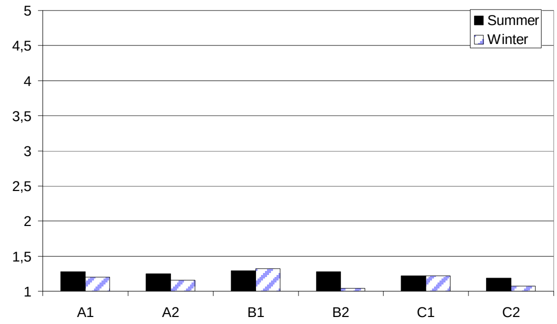


a) b)

Figure 6. Average value during Summer and Winter. For positions, A1-C2, see Figure 1.
a) Customer and b) Staff.
Question answered regarding Temperature "Please state how you would prefer to be now…"
Preference scale 1-7, where neutral = 4, much colder = 1 to much warmer = 7.

When finally judgements about the indoor environment was asked in the questionnaire it was better during winter-conditons for customers as shown in a) Figure 7 below. Followed was the question on how the indoor envrionment was found. Average value was below 1,5 for all cases (1= comfortable, 2= slightly uncomfortable).





a) b)

Figure 7. Average value during Summer and Winter. For positions, A1-C2, see Figure 1.
a) Question answered "How do you judge the indoor environment at this precise moment?" (Scale of 7 degrees from very bad = 1 to very good = 7, 4 = neutral)
b) Question answered "Do you find the indoor environment …..? (Scale 1-5, where comfortable 1 and 4 degrees of increasing intensity of the effect, extremely uncomfortable = 5

DISCUSSION

The cabinets are cooled and kept cold by chilled air. The chilled air is circulating, through the cold cooling-coil and as an air-curtain in the front (sometimes also from the rear) of the cabinet. The air-curtain works as a barrier between the cold air in the cabinet and the warm ambient air outside the cabinet. The variations in the outdoor condition seem to have the strongest influence on the climate condition outside the display cabinet during summer. During summer are ambient temperatures and enthalpy higher and the air-curtain in the cabinet will be weaker. Due to frost growth on the cooling-coil, the air-flow will be lower, the air-curtain weaker and cold air falls easier out from the cabinet. The cold air will lead to colder ambient, higher temperature differences (head-ankles) and warmer temperatures in the cabinets so that the temperature quality of the goods will be influenced. In order to maintain the same temperatures for the cabinets the cooling-demand for the cabinets will be higher and in order to maintain the same ambient temperature outside the cabinets the heating demand will increase for the heating system which often is in combination with the ventilation system. It is of importance how the ventilation system together with the other systems in the supermarket is controlled in order to condition the air where and most needed. There exist a large need of knowledge and further arguments for the merchandiser and people involved with supermarkets in order to choose efficient systems, the systems in the supermarket includes both the system for storage of chilled and frozen goods as well as the heating and ventilation system. A lack of knowledge for people seems to exist since they believe that goods are kept cold when the ambient areas outside the refrigerated cases are cold. The cabinets are not suppose to cool the customers, the are supposed to keep the good cold. The physical measurements and questionnaires presented will be used in order to define a suitable indoor climate for goods, staff and customers in supermarkets. In this paper physical measurements (temperatures) are presented along with answers from questionnaires (regarding temperatures and indoor environment). Further physical parameters along with answers from the questionnaires concerning humidity, illumination, air velocity, background parameters (age, gender etc) are to presented in future publications. The relation between the subjective and objective data will be used to define a desirable indoor climate.

CONCLUSION

The results show that according to ISO7730 people should be dissatisfied with the thermal comfort, especially during summer. Analysis from physical data, Figure 3, indicates that in all supermarkets especially customers during summer are dissatisfied. Temperature differences (between ankle and head) are higher than 5K. According ISO7730, local discomfort caused by vertical air temperature differences, at least 20% are dissatisfied. However customers in front of the cabinets perceive the temperatures to be in average 3.1 (3 = neutral). Inside the cold room, B1, for customers the temperature is perceived lowest (2,18 for summer resp. 2,59 for winter). During summer customers evaluated the temperature as more uncomfortable compared with the winter. Customers are wearing less clothes during summer compared with winter season which most probably is the reason for that answer. However as answer on how the overall indoor environment in the supermarkets during summer and winter was the average value higher than 5 (4 = neutral, 5 = slightly good). During winter the indoor environment was judged as better than summer conditions. However Customers answered 0.78 higher for summer and 0.45 higher for winter compared with staff. The staff answered an average value of 4.85 for summer respectively 5.18 for winter conditions. The questionnaires however, indicate that customers find the indoor environments as comfortable to slightly uncomfortable. When asked how the temperature is preferred an average of 4.1 was given from customers and 3.9 for staff, see Figure 6. Results from previous presentations on field measurement in this study shows that supermarket A, which use the energy less efficient, also have highest difference on the enthalpy measured in front of the display cabinet. With a lower temperature and humidity the energy use can be lowered. For appropriate indoor air climate and environmentally sustainable supermarkets it is of importance that all systems in the supermarket are optimized in order to run efficient.

ACKNOWLEDGEMENT

We gratefully acknowledge the financial support from the Swedish Energy Agency and participation from the project group consisting of a number of industrial partners.

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12 June 2007 at 15:00 - 16:30

A05
Pollutants in the indoor air

Health risk based ETS targets for design of room environments

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SUMMARY

This paper outlines the Australian Standard 1668.2: 2002 ETSHI risk methodology and extends it to give a means of calculating the numerical death risk from exposure of specific age/gender groups to defined Environmental Tobacco Smoke, (ETS) exposures.

Acceptable risk exposure is discussed (with numbers) and actual mortality statistics are presented for comparison.

A 'smoking permitted office' situation is analysed and some conclusions are drawn.

This paper (and the underlying research) was not funded by 'pro' or 'anti' tobacco interests.

INTRODUCTION

The Environmental Tobacco Smoke Harm Index (ETSHI) was introduced in the 2002 edition of Australian Standard 1668 Part 2. (Appendix 1 of Supplement 1)[1].

This author chaired the Standards Australia committee responsible for AS1668.2: 2002. He also chaired the working group that developed the ETSHI methodology.

The ETSHI is an estimate of a significant part (over 50%) of the mortality risk associated with exposure to environmental tobacco smoke in an environment that is ventilated and that is, or is not, fitted with a device for removal of some components of airborne environmental tobacco smoke (ETS) which contribute to the risk.

The ETSHI is a specific aggregate of the mortality risks associated with lung cancer (LC, more specifically: malignant neoplasm of trachea, bronchus and lung), and ischaemic heart disease (IHD) - and these diseases are said to be the causes of at least one half of ETS related deaths.

In AS1668.2: 2002 the development of the ETSHI equations is presented clearly so that the underlying principles may be applied by ventilation system designers to systems other than those that exactly fit the situations covered by its Appendix.

The ETSHI methodology has been extended by this author to give estimated deaths from lung cancer and ischaemic heart disease that are specific to gender and age groups.

Neither the ETSHI methodology nor the extended version described herein takes into account the morbidity (non-fatal disease) effects of ETS exposure e.g., childhood respiratory disease, asthma exacerbations. Also, the methodology does not deal with adverse effect of ETS exposure on child development.

It appears to be reasonably well accepted that Lung Cancer (LC) and Ischaemic Heart Disease (IHD), together, account for at least half of the deaths caused by ETS thus it may be appropriate to view (x)*(ETSHI) as a reasonable estimate of ETS caused deaths – where x is a number in the 1.0 to 2.0 range.

ETSHI methodologies should be used with caution - appreciating the underlying medical and scientific issues and the limitations of the methodology would seem essential.

VENTILATION SYSTEMS

The ETSHI methodology allows for the influence of ventilation system design on ETS mortality to be estimated though it is probably best used for sensitivity analysis i.e. for determining how variation of ventilation parameters varies outcomes.

In the modelled case of a single 'smoking permitted' enclosure in which non smokers are present the parameters that may be varied are : ETS generation, time of exposure of non-smokers, ventilation rate, air recirculation rate, filtration of particulate and gas phase carcinogens and filtration of particulate and gas phase contributions to IHD.

The ventilation model assumes an occupied enclosure with fully mixed air conditions thus the alleged benefits of 'displacement' systems are not accounted for. Similarly the influence of local concentrations etc. of ETS in the enclosure are not modelled.

Manipulation of the model quickly reveals a limitation of technology i.e. that ETS, in a space, has the capacity to influence health *before* it is taken to an air cleaning system.

THE ENVIRONMENTAL TOBACCO SMOKE HARM INDEX

The development of the equations of the ETSHI is presented fully in AS1668.2 Sup.1: this should be referenced if detail is required.

An example of ETSHI calculation is provided later in this paper.

In developing the index it was asserted that an index of the mortality risk associated with ETS exposure can be represented by an aggregate of the risk of death from ETS induced LC and risk of death from ETS induced IHD by considering the following:

- The mortality risk from ETS induced IHD is correlated with short-term, medium-term and, perhaps, long-term exposures to ETS. This correlation is, likely, linear in the range of exposures up to that represented by typical spousal exposure. At higher exposures, the relationship is better correlated with that of smoking.
- For smoking-induced IHD it is reasonable to assert a dose response relationship corresponding to the expected response of a statistically log-normal (i.e. Gaussian Distribution) population. Accordingly, the dose vs cumulative risk curve, for smoking, is a modified sigmoid (viz. integral of the log-normal distribution) shape with the curve asymptotic to {Relative Risk = 1.0} at the low end and asymptotic to {Relative Risk = a higher value, AS1668.2:2002 estimates 2.0} at the high end

- The IHD effect(s) of ETS exposure is (are), at least partially, correlated with the tendency of exposure to enhance human blood platelet aggregation. An initial estimate is that 50% (likely range 30% to 80%) of the mortality contribution of ETS lies in its enhancement of blood platelet aggregation.
- For ETS induced LC, the nature of the disease and the dose response evidence suggests that it is reasonable to assess mortality risk as proportional to average medium to long-term exposure with no nil-effect dose.
- The LC causing effect of ETS is predominantly related to ETS particulates, though ETS gas phase components may contribute.
- The relative extent to which the particulate component of ETS is responsible for LC deaths can be estimated with reasonable accuracy. An initial estimate is 80% (likely range 70% to 95%) of risk attributable to particulate and the balance attributable to gas phase components.
- Published NHMRC (National Health and Medical Research Council, Australia) estimated ETS induced spousal LC death rates and IHD death rates are appropriate.

For LC and IHD a dose response reference point, consisting of an outcome and its associated dose, was determined. RSP was the chosen indicator of ETS concentration or exposure. A dose response relationship was asserted for LC and IHD.

From this information, a steady state ventilation/air treatment/dose/risk relationship was calculated. The development of the Index was presented in AS1668.2 in an unreduced form so that the constituent logic and variables were not obscured.

A simple model was used with one enclosure containing mixed air, and one air treatment assembly taking air from the enclosure, treating it, and returning it to the enclosure.

Allowance was made, in respect of each challenge, for two-component cleaning action of the air treatment unit.

For LC and IHD deaths the NHMRC has reported data for non-smokers attributable to spousal smoking . This is 'official' health authority data for Australia. This or other data may be used in the ETSHI approach.

In calculation of the Index, it was, recommended that not less than 80% of the LC inducing effect of ETS be attributed to the particulate phase of ETS and that fine particulate air filters rated in accordance with the requirements of AS1668.2 be considered effective in removal of the particulate phase of ETS.

Similarly, it was recommended that approx. 50% of the IHD inducing effect of ETS be attributed to the characteristic of ETS that enhances blood platelet aggregation. Neither demonstrated means of removal of this characteristic nor a test specification for apparatus for removal of this characteristic from air are not known to exist.

The ETSHI estimates apply to where the stated daily ETS exposure has been repeated every day for an indefinite period prior to the time of risk assessment.

The ETSHI is the sum of (a) the population average estimated mortality outcome for LC, and (b) the estimated mortality outcome for IHD is based on data applying to an age sixty-five person.

Given the limited accuracy of the data in the model etc. it was considered appropriate to consider the simple arithmetic sum of the two mortality outcomes to be a useful index.

VALIDITY OF THE ENVIRONMENTAL TOBACCO SMOKE HARM INDICES

The validity of ETSHI, or like approaches to other hazards, is to be assessed by users.

AS 1668.2: 2002 Sup. 1 provides full detail of the ETSHI methodology and all of the underlying assumptions, data and references so this assessment is facilitated.

One opinion of the validity of the approach is known to this author. In a report[2] to the Cancer Council of New South Wales (an Australian State) James Repace, a noted ex-USEPA anti-smoking activist, found that his risk calculation was 'consistent with the Environmental Tobacco Smoke Harm Index published in Australian Standard 1668.2:2002 Supplement 1'.

The ETSHI was developed without reference to James Rapace's work and he refers, in his paper, to calculations of his published in 1998. The Standards Australia and Rapace approaches were developed independently.

The example calculated in AS1668.2:2002, for a smoking permitted office situation, gives an ETSHI of 225 deaths per million persons per year using mean NHMRC ETS death estimates.

James Rapace evaluated an essentially identically situation (assumptions vary in minor respects) and calculates a death rate of 244.

The similarity of these results is remarkable.

ACCEPTABLE RISK

Few reputable persons or government/other agencies have chosen to state acceptable mortality risks for exposure to hazards.

It is more common to depend on less solid criteria such as the Precautionary Principle and the ALARA (As Low As Reasonably Achievable) exposure approach.

There is a quite useful and nuanced discussion of the precautionary principle in the EC document[3] 'Communication from the commission on the precautionary principle'.

In reference 4 Mark Tweeddale refers to UK government HSE (Health and Safety Executive) proposed risk criteria. These, in turn, draw upon RSSG (UK Royal Society Study Group) information.

The HSE/RSSG information, which in this paper ETS hazard is referred to, is interpreted as proposing these risk criteria (with various caveats applicable).

- An 'upper bound' criteria of 10 fatalities per year per million persons for a normal population where a risk above this level would be regarded as 'unacceptable in essentially all circumstances except perhaps for activities entered into voluntarily with full knowledge of the risks'

- A lower bound of 1 fatality per year per million persons where a risk below this level may 'legitimately be regarded as trivial by the decision maker'.

These proposed risk criteria may be viewed against actual mortality rates. Some approximate mortality rates for Australia are shown below. The actual rates have year-to year scatter, longer term time trend and gender dependence.

Homicide: 22 deaths per million persons per year
Suicide: 127, Firearm accident: 2, Machinery accident: 5, Drowning: 18
Fire: 6, Air transport: 4, Motor vehicle: 95, Birth trauma: 2, Intestinal infection: 3
Septicaemia: 22, Viral diseases: 8, Malignant neoplasm of colon: 183
Malignant melanoma of skin: 48, Malignant neoplasm of female breast: 286
Malignant neoplasm of prostate: 245, Alcohol dependence: 12,
Drug dependence: 17, Multiple sclerosis: 4, Epilepsy: 12,
Cerebrovascular disease: 708, Pneumonia: 101, Influenza: 4, Asthma: 49

THE <u>EXTENDED</u> ENVIRONMENTAL TOBACCO SMOKE HARM INDICES

The ETSHI was developed within the Australian Standardisation process. The extended ETSHI indices (e-ETSHI), described here, have been developed outside this framework.

Although the ETSHI is a useful indicator of overall societal mortality risk from ETS exposure it does not allow examination of the effect of gender and age on the risk.

e-ETSHI provides this information. As reasoned above (for ETSHI) it may be appropriate to view (x)*(e-ETSHI) as a reasonable estimate of ETS caused deaths, for specific age/gender groups – where x is a number in the 1.0 to 2.0 range.

Given that 'relative risk' is at the core of the ETSI methodology the extension to calculating e-ETSHI is simple – merely a matter of proportioning in accordance with mortality tables[5] - so it is not treated here in detail. A hypothetical example will suffice:
If,

- the population average death rate for a disease, from all causes, is 100 per million persons per year, and
- the gender/age group subset death rate for a disease from all causes is 70 per million persons per year
- the population average death rate for a disease – from a particular cause - is, 30 per million persons per year.

Then,

- the gender/age group subset death rate for a disease from a particular cause is (70/100) * (30) = 21 per million persons per year.

AN EXAMPLE

The example in AS1668.2:2002 is typical of an office ETS exposure situation with these parameters (an alternative e.g. a bar is equally easily analysed): smoking is permitted, 100 people are present, each person is present for 8 hr per day, 1/3 of those present are active smokers, each smoker consumes 2 cigarettes per hr, outdoor air supply is 10 litres per sec. per person (l/s/pp), recycle (return) airflow is 50 l/s/pp, 35% (hot DOP) return air filter is fitted, 80% of the cancer risk is contained in the particulate phase of the ETS, air treatment for the removal of factors affecting ischaemic heart disease is not installed.

When calculating the ETSHI or e-ETSHI a consideration is the base judgement of severity of mortality risk set by the responsible health authority.

In Australia the peak health body (the NHMRC) has dealt with uncertainty by publishing data corresponding to situations that represent Highest Plausible risk, Estimated risk and Lowest Plausible risk.

For the example the ETSHI corresponding with the estimated risk is 225 deaths per million people per year.

For the example e-ETSHI values corresponding to the mean, lowest plausible and highest plausible risk assessments are stated in the following tables for the various disease, gender and age groups.

Table 1. RISKS: USING NHMRC MEAN ESTIMATES

Disease =>>	LC	LC	IHD	IHD	LC	IHD	Both	Both	Both
Gender =>>	M	F	M	F	M+F	M+F	M	F	M+F
Age	------------Deaths per million persons per year------------								
||||				||||					
0	0	0	0	0	0	0	0	0	0
5	0	0	0	0	0	0	0	0	0
10	0	0	0	0	0	0	0	0	0
15	0	0	0	0	0	0	.1	0	0
20	0	0	.1	0	0	.1	**.1**	**0**	**.1**
25	.2	.2	.7	.1	.2	.4	**1**	**.3**	**.7**
30	.4	.4	1.3	.2	.4	.8	**1.8**	**.6**	**1.2**
35	2.8	2.5	5.2	1	2.6	3.1	**8**	**3.6**	**5.8**
40	5.1	4.6	9	1.8	4.9	5.4	14.2	**6.5**	10.3
45	22.2	11.1	23.2	5.3	16.7	14.2	45.4	16.5	30.9
50	39.3	17.6	37.4	8.8	28.5	23.1	76.7	26.4	51.6
55	109	38.7	88.7	26.2	73.9	57.5	197	65	131
60	178	59.8	140	43.6	119	91.9	319	103	211
65	280	85.8	263	108	183	185	543	194	369
70	381	111	387	172	246	280	768	284	526
75	477	120	669	378	299	523	1147	498	823
80	573	129	952	583	351	767	1526	712	1119
85	524	92	1986	1548	308	1767	2511	1640	2076

(M=male, F=female, LC=lung cancer, IHD=Ischemic heart disease)

Table 2. RISKS: USING NHMRC 'LOW END OF PLAUSABLE RANGE' EST.

Disease =>>	LC	LC	IHD	IHD	LC	IHD	Both	Both	Both
Gender =>>	M	F	M	F	M+F	M+F	M	F	M+F
Age	------------Deaths per million persons per year------------								
0	0	0	0	0	0	0	0	0	0
5	0	0	0	0	0	0	0	0	0
10	0	0	0	0	0	0	0	0	0
15	0	0	0	0	0	0	0	0	0
20	0	0	0	0	0	0	**0**	**0**	**0**
25	0	0	0	0	0	0	**.1**	**0**	**.1**
30	.1	.1	.1	0	.1	0	**.2**	**.1**	**.1**
35	.7	.6	.4	0	.7	.2	**1.1**	**.7**	**.9**
40	1.4	1.2	.7	.1	1.3	.4	**2.1**	**1.4**	**1.7**
45	6	3	1.8	.4	4.5	1.1	**7.8**	**3.4**	**5.6**
50	10.7	4.8	2.9	.6	7.7	1.8	13.6	**5.5**	**9.5**
55	29.7	10.5	6.9	2	20.1	4.4	36.6	12.6	24.6
60	48.8	16.3	10.9	3.4	32.5	7.1	59.7	19.7	39.7
65	76.4	23.4	20.5	8.4	49.9	14.4	96.9	31.8	64.4
70	104	30.5	30.1	13.4	67.2	21.8	134	43.9	89.1
75	130	32.9	52.1	29.4	81.6	40.8	182	62.3	122
80	156	35.3	74.2	45.4	95.9	59.8	230	80.8	155
85	143	25.1	154	120	84.1	137	297	145	221

(M=male, F=female, LC=lung cancer, IHD=Ischemic heart disease)

Table 3. RISKS: USING NHMRC 'HIGH END OF PLAUSABLE RANGE' EST.

Disease =>>	LC	LC	IHD	IHD	LC	IHD	Both	Both	Both
Gender =>>	M	F	M	F	M+F	M+F	M	F	M+F
Age	------------Deaths per million persons per year------------								
0	0	.1*	0	0	0	0	0	.1*	0
5	0	0	0	0	0	0	0	0	0
10	0	0	0	0	0	0	0	0	0
15	0	0	.2	0	0	0.1	.2	.1	.1
20	0	0	.4	.1	0	.2	**.4**	**.2**	**.3**
25	.5	.4	1.9	.4	.4	1.1	**2.4**	**.8**	**1.6**
30	.9	.8	3.5	.7	.9	2.1	**4.4**	**1.5**	**3**
35	5.6	5.1	13.1	2.6	5.3	7.8	18.7	**7.7**	13.2
40	10.3	9.3	22.7	4.6	9.8	13.6	33	14	23.5
45	44.4	22.2	58.2	13.3	33.4	35.7	102	35.7	69.2
50	78.6	35.3	93.7	22	57	57.9	172	57.4	114
55	218	77.5	222	65.7	147	144	440	143	292
60	357	119	351	109	238	230	709	229	469
65	560	171	660	271	366	466	1221	443	832
70	763	223	970	433	493	701	1733	657	1195
75	955	241	1678	947	598	1313	2634	1189	1911
80	1147	259	2387	1461	703	1924	3534	1721	2628
85	1049	184	4980	3881	616	4430	6029	4065	5047

(M=male, F=female, LC=lung cancer, IHD=Ischemic heart disease)

* odd value due to an aberration in mortality data

COMMENT ON THE EXAMPLE

Viewed in the context of the above discussion of acceptable risk (say, 1 to 10 deaths per million persons per year) The ETSHI index of 225 indicates that the examined situation is probably unsafe.

In the three tables of e-ETSHI (above) the situations wherein the population e-ETSHI is less than 10 deaths per million persons per year (and the population is aged less than about 18 years, and thus respiratory organs are normally well developed) are printed in **BOLD.** In examining these tables it is wise to recollect the above developed likely relationship: death rate = (e-ETSHI) * (x) where 1 <= x <= 2.

Examination of the tables allows a number of conclusions to be drawn

- Exposure to ETS can be quite dangerous
- The main risk from ETS exposure is death from heart disease, lung cancer risk though significant is not as high
- Exposure rapidly becomes more dangerous as one ages - it is wise to be young
- It is wise to be female – you have a lower ETS death risk (though there may be some data aberrations contributing to the lower lung cancer estimate)
- The risk of lung cancer death from ETS exposure increases monotonically with age
- The risk of Ischemic Heart Disease from ETS exposure increases up to age 80 and then decreases. Perhaps the cohort of susceptible people has been largely exhausted by age 80
- There may be an age range (somewhere in the larger 20 to say 45 years range) where the sheer robustness of human construction allows that ETS exposure will present a death risk, in defined circumstances, that is within some definition of acceptability. This age range seems to be greater for females. Perhaps this may tempt the morally challenged to justify the predominance of young female employees in service jobs in smoking permitted bars and the like?

The relative prominence of Ischemic Heart Disease in the ETS exposure risk situation, as seen above, has an interesting prophylactic consequence: it allows something to be done to reduce the risk of ETS exposure.

It is difficult to see how Lung Cancer risk of ETS exposure can be reduced other than by exposure reduction/avoidance.

However, with Ischemic Heart Disease, these measures could be augmented by medicines such as aspirin during periods of ETS exposure - the ‘not young’ may be wise to dose with such anticoagulants before being exposed to ETS. This effect could be easily incorporated in the e-ETSHI model.

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FLIES (Flanders Indoor Exposure Survey): The influence of contaminants in ambient air on the indoor air quality and children's exposure

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SUMMARY

This study presents the first Flemish monitoring campaign on indoor air quality in children's living environment. The main aim of the study was to determine for sensitive groups of children the indoor environment exposure a result of contaminants that occur in the outdoor and indoor air, and to quantify the relative importance of ambient and indoor-source related exposure to total personals exposure of children.
The monitoring campaign was performed in January – February 2006 in 50 dwellings, and 24 other children's micro-environments. Indoor and outdoor concentrations (7-day averages) of 15 pollutants were measured: MTBE, benzene, trichloroethene, tetrachloroethene, ethylbenzene, m+p xylene, styrene, o-xylene, 1,2,4 trimethylbenzene, p-dichlorobenzene, TVOC, NO_2, formaldehyde, acetaldehyde and PM.
Since MTBE has no indoor source (petrol additive), the slope of indoor versus outdoor MTBE (F_{INF} = 0.86; 95 % C.I.: 0.59-1.13) could be used to assess the infiltration of other gases. Using this indirect technique, it was estimated that, on average, the relative importance of indoor sources to total indoor concentrations indoor varied from 85 % for formaldehyde to 34 % for traffic related components such as benzene. The MTBE infiltration technique was not suitable for PM and NO_2.
Typical children's personal exposure was modeled based on typical time patterns and median indoor concentrations of different micro-environments. It revealed that on average, the indoor-source exposure attributed from 32 % (benzene) to 80 % (formaldehyde) to personal exposure, whereas ambient exposure accounted for 66 % (benzene) to personal exposure. Exposure experienced during transport consisted of 2 % (tetrachloroethene) to 15 % (MTBE).
It is concluded that the relative importance of indoor sources or ambient sources is pollutant-dependent and it is advised to take this pollutant-dependent behaviour into account when stipulating policies aiming at reducing exposure to children.

INTRODUCTION

Whereas ambient air quality in Flanders has been monitored for years, indoor air quality in Flanders has not been investigated systematically in a quantitative way in the past. However, people spend up to 90 % of their time indoors, which indicates the relevance of indoor air quality to respiratory health.
Air pollutants present in the indoor environment are the result of (1) product emissions such as furniture, consumer products, cleaning materials,… and (2) infiltration of ambient substances. Knowledge on the relative importance of these two terms is crucial for stipulating a good indoor air quality policy.
The aim of the study was (1) to make an inventory of indoor concentration in Flemish dwellings and other micro-environments in which people spend a large part of their time, (2) to assess if traffic density affected indoor and outdoor concentrations and (3) to determine the

personal exposure of children and, (4) to differentiate the personal exposure in an ambient and non-ambient fraction. It is hypothesized that traffic density affects the concentration of typically traffic-related compounds such as MTBE and benzene and not for other compounds Children were selected as sensitive groups because they experience a higher air pollutant load given their 50 % larger body-weight rescaled or 35 % larger long-surface rescaled exposure than adults.
The selection of the investigated pollutants was based on 3 criteria : 1) only pollutants with possible outdoor sources were selected (biological agents and ozone not retained) and 2) only pollutants for which health effects have been proven in the past (e.g. based on WHO health criteria). Terpenes, pesticides and bromated flame retardants were not measured in the first Flemish Indoor Exposure Survey because of too specific and demanding sampling techniques.

METHODS

Selection of measuring locations

The monitoring campaign was performed in January-February 2006 in the eastern part of Flanders in 50 dwellings of volunteers. The dwellings were selected based the proximity to traffic: 12 houses in rural background (RB) areas (< 50 cars passing/day), 24 houses in urban background (UB) area (< 500 cars/day) and 12 houses in urban hot spot (HS) area (>15000 cars/day). The emphasis on dwellings in UB areas is because of the 75 % of Flemish houses are of this type.
In the houses, 2 indoor locations, namely living room and bedroom, and 2 outdoor locations, namely front door and back door (if applicable) concentrations were sampled. The set of 50 houses existed of a heterogeneous mix of dwelling types: detached dwellings (12), semi-detached dwellings (n =3), connected dwellings (n = 23), flats at ground floor (n = 1) and flats above the ground floor (n = 7). Additionally, indoor and outdoor air was sampled at schools or daycares (n= 10), transport (n= 9; car, public transport, wandering) and sport and leisure infrastructures (n =7). The number of sampled dwellings was smaller for PM (10 living rooms, 34 bedrooms and 19 dwelling outdoor locations) because of logistical reasons.

Questionnaires

Information regarding possible indoor sources (presence of consumers products, cleaning products, furniture, heating devices, …), dwelling characteristics (ventilation types and frequency, isolation, single or double glass windows, presence of an attached garage …) was obtained by means of a detailed questionnaire. The second part of the questionnaire investigated the children's time patterns (7-day diary with 1 hours intervals for each child).

Sampling and analysis techniques

Except for particulate matter, all pollutants were measured by diffusive sampling techniques. For VOC's Radiello samplers were used. After 7-days sampling, the samplers were extracted with carbon disulfide containing an internal standard. Analysis were performed on a HP6890 gas chromatograph hyphenated with a HP5975 mass spectrometer. Total volatile organic compounds (TVOC) was measured in full scan-modus, while the selected individual compounds were measured in selected ion monitoring mode. External standards were used for calibration. Aldehydes were sampled with diffusive dosimeters manufactured by SKC. Mechanism of the dosimeters is chemisorption with dinitrophenylhydrazine. After the sampling period, the aldehydes were desorbed with acetontrile and analyzed by LC-UV. NO_2

was sampled with diffusive monitors IVL (Swedish Environmental Institute) and analysed by IVL. Particulate matter was measured by Grimm dust, Buck or Aeromini monitors. The latter was provided with a weather proof enclosure and was used outdoors. Grimm and Buck monitors were used indoors.

Infiltration of ambient pollutants

The total indoor concentration of a substance can be expressed with equation 1 [1] which included a left term related to outdoor concentration and a right term representing the contribution of indoor sources:

$$C_{indoor} = \frac{Pa}{a+k} C_{outdoor} + C_{indoor\ sources}, \qquad (1)$$

The infiltration factor (F_{INF}) can be derived in the absence of indoor sources, assuming the following relationship between concentrations indoor (C_i) and outdoor (C_o) [2,3]:

$$F_{INF} = \frac{C_i}{C_o} = \frac{Pa}{a+k}, \qquad (2)$$

with P: penetration factor (-); a is the air exchange rate (/h) and k is the deposition, removal or sorption rate (/h). In this study, MTBE was selected as a tracer with only outdoor sources to calculate F_{INF} and thus, in a next step, to discriminate the outdoors generated from indoor generated fractions of other pollutants:

$$C_{ig,x} = C_{i,x} - F_{INF} \times C_{o,x}, \qquad (3)$$

with ig = indoor generated and x = substance x
This method is analogous to the principle used in other studies using other tracers (e.g.SO_4 as tracer for PM2.5) to determine outdoor to indoor infiltration of PM [4,5]

Personal exposure

The typical children's personal exposure was modeled based on time activity patterns obtained through the questionnaires and concentrations of each micro-environment. Total personal exposure can be split up in a 1) ambient exposure (directly and infiltrated), 2) indoor sources and 3) exposure experienced during transport. The latter micro-environment is difficult to separate in a ambient and indoor generated fraction under the current set-up. Thus, personal exposure to substance x was calculated as:

$$T_x = A_x + N_x + TR_x, \qquad (4)$$

Ambient (A_x) and non-ambient (N_x) exposure were respectively calculated as :

$$A_x = \sum_j t_{o,j} \times C_{o,j,x} + \sum_j t_{i,j} \times F_{INF} C_{o,j,x}, \qquad (5)$$

$$N_x = \sum_j t_{i,j} \times C_{i,j,x}, \qquad (6)$$

with $t_{o,j}$ and $t_{i,j}$: time fraction spent respectively outdoor and indoor at µ-environment j.
Exposure experienced during transport (TR_x) (I or O) was calculated as a separate term. Total personal exposure to x is the sum of A_x, N_x and TR_x.
The variability in exposure was assessed based on Monte Carlo simulations (Crystall Ball software) by accounting for variability in dwelling indoor concentrations, which is the micro-

environment with the highest variation in concentrations and the most important compartment with respect to time budget. Variations in other micro-environments were not accounted for because they affected to a much lesser extent the exposure and because a distribution on concentrations could not be established because of the limited dataset for micro-environments other than dwellings.

Statistical analyses

The statistical analyses were performed using the statistical tool package Statistica (version 7). The non-parametric Kruskal-Wallis Anova test and non- parametrical correlation analyses were applied because of not normally distributed datasets.

RESULTS

A summary of indoor and outdoor concentrations in 50 Flemish dwellings is given in Table 1. Concentrations of gases show a very high variability between different houses (n=50), both indoors and outdoors. The most abundant gases in both indoor and outdoor environments were toluene, NO_2, formaldehyde and acetaldehyde.
Indoor concentrations generally exceeded corresponding outdoor concentrations (except for NO_2 and PM10). The target value of the Flemish Indoor Decree of 11/06/2004 [6] of 200 µg/m² is exceeded in 95 % of the investigated indoor environments. The Flemish target values for formaldehyde (10 µg/m³) and for benzene (2 µg/m³) were exceeded in nearly 50 % of the cased. The Flemish intervention limits for benzene and formaldehyde were exceeded in respectively 3 and 1 indoor dwelling locations. The Flemish Indoor Decree does not define intervention limits for TVOC's.

Table 1: Indoor (living room and bedroom) and outdoor (front door and backdoor) concentrations in 50 Flemish dwellings

	Dwelling Indoor (n=100)			Dwelling Outdoor (n=86)		
	median	min	max	median	Min	max
		µg/m³			µg/m³	
MTBE	0,54	0,12	33,47	0,35	0,08	1,27
Benzene	2,14	0,70	23,71	1,58	0,69	3,21
Trichloroethene	0,16	0,03	1,21	0,12	0,03	0,26
Toluene	8,13	1,29	122	3,06	1,12	12,27
Tetrachloroethene	0,26	0,06	52,13	0,20	0,03	6,48
Ethylbenzene	1,09	0,20	7,37	0,50	0,16	1,38
m-+p-Xylene	2,33	0,43	17,52	1,12	0,31	3,56
Styrene	0,21	0,01	2,95	0,06	0,01	1,94
o-Xylene	0,86	0,14	6,14	0,41	0,12	3,48
1,2,4-Trimethylbenzene	1,99	0,17	16,94	0,64	0,14	3,32
p-Dichlorobenzene	0,07	0,03	7,11	0,03	0,03	0,09
TVOC	491	138	2793	218	136	396
NO2	21,3	7,1	122,1	40,9	1,9	71,2
Formaldehyde	23,7	1,4	124,1	3,4	2,2	64,9
Acetaldehyde	21,8	1,1	65,2	13,8	0,4	63,2
PM10	11,7	2,3	58,0	33,8	17,8	73,7

Statistics in Table 1 include both living room and bedroom concentrations and front door and backdoor concentrations. On average, the ratio of bedroom to corresponding living room concentration was near 1 (average ratio varied from 0,80 for acetaldehyde to 1,51 for styrene). In general, backdoor concentrations were slightly (± 0,9) fold lower than frontdoor concentrations. This suggests that the dwellings act as a barrier for pollutants that are mainly formed at the street.

For some gases, outdoor concentrations were affected by the traffic density, concentrations for HS locations being highest, followed by urban background, and by rural background (Table 2). As expected, traffic related compounds (MTBE, benzene, toluene and NO_2) are higher in outdoor environment of HS compared to RB. For some traffic pollutants, higher indoor concentrations in HS than in RB and UB (e.g. NO_2) were observed. Compared to the outdoor concentrations traffic-related pollutants like toluene are no longer significantly different in the different locations, indicating an additional contribution from indoor sources. For other, more indoor generated pollutants (xylenes, …) no effect of traffic density indoor concentrations was observed. The distribution of PM sampling sites was not balanced enough over HS, UB and RB classes to allow a statistical comparison (data not shown).

Table 2: Mean outdoor and indoor concentrations for 3 traffic density classes (UB: urban background; HS: hot spot; RB: rural background) Statistical different values between location type classes (Anova, P<0,05) are marked with different letters (for gases without differences between any of the 3 groups are not marked with letters)

	outdoor			indoor		
	UB	HS	RB	UB	HS	RB
	µg/m³			µg/m³		
MTBE	**0,4^{AB}**	**0,6^{A}**	**0,3^{B}**	1,53	0,75	1,52
Benzene	**1,6^{A}**	**2,0^{B}**	**1,4^{A}**	**2,75^{A}**	**4,16^{B}**	**2,15^{A}**
trichloroethene	**0,13^{A}**	**0,12^{A}**	**0,07^{B}**	**0,22^{A}**	**0,28^{A}**	**0,12^{B}**
Toluene	**3,5^{AB}**	**4,3^{A}**	**2,5^{B}**	13,31	10,58	19,03
tetrachloroethene	**0,6^{A}**	**0,5^{A}**	**0,1^{B}**	**0,76^{A}**	**4,13^{B}**	**0,23^{C}**
ethylbenzene	0,5	0,7	0,5	1,45	1,47	1,66
m+p xylene	1,3	1,5	1,1	3,23	2,97	3,5
styrene	0,1	0,1	0	0,3	0,23	0,36
o-xylene	0,5	0,5	0,4	1,27	1,24	1,31
1,2,4trimethylbenzene	**0,8^{A}**	**1,0^{A}**	**0,4^{B}**	3,47	3,37	3,99
p-dichlorobenzene	0	0	0	**0,61^{A}**	**0,10^{B}**	**0,06^{B}**
TVOC	229,6	244,6	263,4	510,5	605,5	700,7
NO_2	**38,4^{A}**	**47,5^{B}**	**27,5^{A}**	**24,3^{A}**	**31,7^{B}**	**17,6^{C}**
formaldehyde	11,8	5,7	8,8	**34,5^{A}**	**36,9^{AB}**	**21,1^{B}**
acetaldehyde	18,5	23,1	15,4	17,9	24,4	16,4

The MTBE infiltration method (see above) can only be used under the assumption of absence of indoor sources of the tracer However, exceptionally high indoor/outdoor MTBE ratios were observed for 2 dwellings. In these two dwelling, MTBE indoor concentrations 7,6- 33 µg/m³ were measured, while concentrations were as low as 0,146 - 0,195 - µg/m³. The only potential MTBE indoor source was the presence of gasoline indoors. Indeed, the presence of a garage in the dwelling, or adjacent to the dwelling (with passage between garage and house) in these 2 houses suggests the presence of MTBE indoor source. In 6 other houses, indoor concentrations above 1 µg/m³ were measured; although the MTBE ratio was I/O remarkably lower than of the 2 above mentioned dwellings, I/O of these 6 dwellings was largely above the remainder part, and 5 of 6 of these houses had a garage in the dwelling (directly or adjacent with passage), suggesting MTBE indoor sources. Therefore, these houses were excluded and based on the filtered dataset (n = 42), the average F_{INF} factor was derived from the slope of indoor MTBE versus outdoor MBTE.

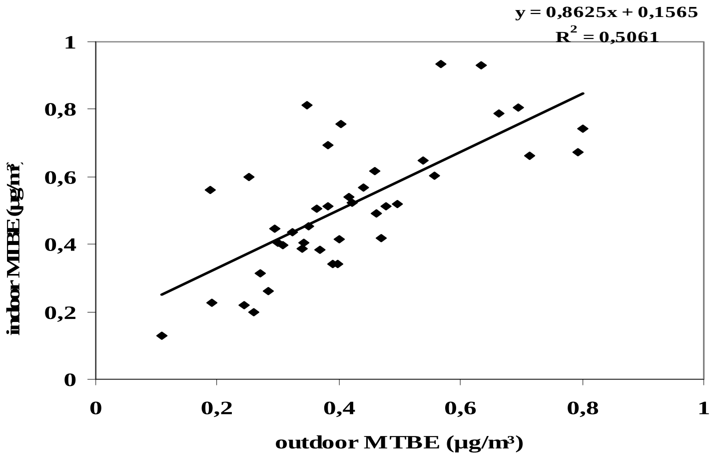

Figure 1: Indoor versus outdoor MTBE concentrations in 42 houses without indoor sources of MTBE. The average infiltration factor FINF is defined as the slope of indoor versus outdoor MTBE.

The indoor generated concentrations of pollutants (C_{ig}) were calculated for each dwelling individually, using an average $F_{INF,\ MTBE}$ factor (0,86; 95% CI: 0,59-1,13). It should be kept in mind that the individual $F_{INF,\ MTBE}$ values may differ between houses. However, dwelling type and ventilation frequencies, building properties (isolation, single versus double glass windows) did not statistically affect I/O MTBE ratio's. It was thus not feasible to stratify $F_{INF,\ MTBE}$ for building type and ventilation classes due to the limited dataset of houses in this study.

The pollutant with the largest median C_{ig}/C_i ratio was formaldehyde ($C_{ig}/C_i = 0.85$) and the pollutant with the lowest C_{ig}/C_i was benzene ($C_{ig}/C_i = 0.34$). For each component, rather large deviations from this median value was observed (e.g. formaldehyde P25 of $C_{ig}/C_i = 0.71$ and P75 of $C_{ig}/C_i = 0.91$; for benzene: P25 of $C_{ig}/C_i = 0.22$ and P75 of $C_{ig}/C_i = 0.55$). This is not surprising given the wide spread in possible indoor sources between individual dwellings. Notwithstanding the dominance of indoor sources to indoor formaldehyde concentrations pointed out by this indirect MBTE infiltration method, the statistical analyses between indoor sources, building and ventilation characteristics and measured concentrations did not reveal significant relationships for formaldehyde. Also for other pollutants, the statistical relations between indoor sources and measured concentrations were generally poor. The most important significant relations were the significant effects of (1) smoking on C_i and C_{ig} of toluene, (2) use of glue and stain removers on C_i and C_{ig} TVOC and (3) use of sealing products on C_i and C_{ig} benzene.
The MTBE infiltration method was not successful to estimate C_{ig} for NO_2 and PM10.

The typical exposure for children is calculated based on median concentration in each micro-environment and average time patterns. Typical children's exposures attributed 0,52 – 0,71 µg MBTE/m³, 1,9-2,7 µg benzene/m³, 0,17-0,23 µg trichloroethene/m³, 6,3-7,5 µg toluene/m³, 0,24-0,48 µg tetrachloroethene/m³, 0,9-1,4 µg ethylbenzene/m³, 2,0 – 2,9 µg m+p xylene/m³, 0,10-0,70 µg styrene/m³, 0,8-1,1 µg o-xylene/m³, 1,6-2,3 µg 1,2,4-trimethylbenzene, 0,1-0,9 µg p-dichlorobenzene, 410-522 µg TVOC/m³, 19-34 µg NO_2/m³, 15-25 µg formaldehyde/m³, 10-27 µg acetaldehyde/m³ and 8-13 µg PM10/m³. These ranges of typical exposure refer to variation in typical exposure for different age classes (0-2,5 years, 2,5-6 years, 6-12 years, 12-18 years) and proximity of the houses to traffic density (UB, RB, HS). The differences in typical exposure between different age classes and traffic density classes were rather small. It was assessed that the highest exposed children (P95) were 2-4 fold higher exposed than the median, typical exposed children for most gases. Exceptions were tetrachloroehtene (x11), and p-dichlorobenzene (x 50). These extremes were related to the extremely large range of indoor concentrations (Table 1).

The distribution of exposure over ambient (A: $A_i + A_{o,;,}$ with A_o, exposure to ambient pollutants directly in outdoor environment and Ai, exposure to ambient pollutants that have infiltrated indoors)), indoor source-related (N) and exposure during transport (TR) is given in Figure 2 for TVOC and for the 2 components with the most extreme A,N, TR distribution profiles (except for MTBE, NO_2 and PM).
For other pollutants, either ambient exposure dominated typical personal exposure (for MTBE, trichloroethene, tetrachloroethene), or indoor sources exposure (toluene, ethylbenzene, m+p xylene, styrene, o-xylene, 1,2,4-trimethylbenzene, p-dichlorobenzene) dominated the personal exposure. Exposure during transport was for most components below 5 % to the total personal exposure and varied from < 2% (tetrachloroethene) to 15 % (MTBE).

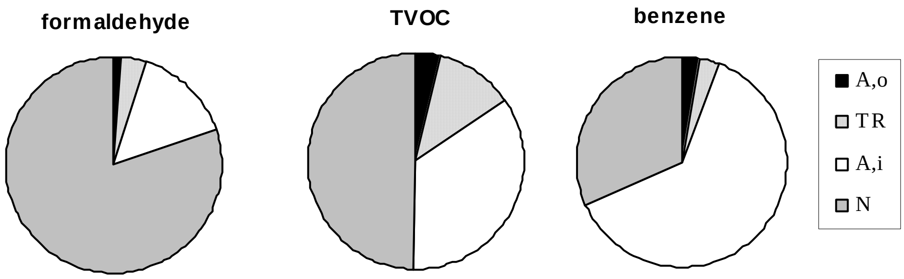

Figure 2: Distribution of typical exposure to formaldehyde, TVOC and benzene over ambient, non-ambient (indoor sources) exposure and exposure during transport.

DISCUSSION

Indoor concentrations of benzene, toluene, o-xylene, ethylbenzene, trimethylbenzene,… in Flemish dwellings were generally within ranges reported previously elsewhere [7,8]. In contrast, the low indoor-outdoor PM ratio of this study (mean I/O ratio: 0.3) are not in line with previous studies [9] and could not be explained.

The use of MTBE as infiltration tracer is relatively new, and this MTBE based infiltration method gives realistic values of F_{inf} for dwellings, in accordance with infiltration factors reported by others using other tracers (e.g. F_{inf} based on sulphate = 0,7 (90 % CI: 0,5-0,9) [5]. The intercept of the MTBE indoor versus outdoor graphs refers to small residual (background) MTBE concentrations in houses. Similar background concentrations were also found for SO_4 [10]. However, the MTBE based infiltration factor was not applicable to discriminate indoor and outdoor generated NO_2 and PM10 concentrations in this study. The inappropriateness of MTBE to estimate the infiltration of PM and NO_2 probably lies in different penetration factors and/or removal/sorption rates or, in the extreme low indoor-outdoor ratio's for PM10. The 95 % CI of F_{INF} (0.59-1.13) indicates that there is a further need to refine F_{INF} for different dwelling types with respect to ventilation. A larger dataset than the one of this study is needed hereto.

Children’s exposure is mainly determined by indoor dwelling exposure given the majority of time spent indoors. Variation in exposure between children is mainly due to variations in indoor dwelling concentrations (up to 100-fold), rather than variations in children’s time budgets, which are limited.

This study shows that the contribution of infiltrated outdoor air pollution to personal exposure is different among investigated pollutants. This is a point of attention in ambient air quality policies, to include the indoor exposures more explicit. Recommendations for precautionary measures to reduce or avoid exposure indoor exposure to certain gases, for example for formaldehyde, are difficult to make based on this study because only few clear source-concentrations-exposure relationships were found. For this, work on short-term and long-term emission sources and their relation to concentrations, using various time average measurements should be performed. This is best placed in the context of product policy.

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On the indoor air quality problem in residential areas: the Athens case

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ABSTRACT

The buildings' environment plays a very important role in health matters and the quality of life.

A series of experimental measurements were carried out in the residential sector of the greater region of Athens. Parameters influencing the indoor air quality such as particles, CO, CO2 and total organic compounds (TOCs) concentrations were monitored. Thermal and acoustic comfort as well as characteristics linked to the type, size and existing thermal insulation materials of the buildings were investigated.

INTRODUCTION

It is well understood that since 70 per cent of a humans' time is spent indoors, indoor air pollution may pose a serious threat to human health. Indoor air pollution can be caused by outdoor and/or indoor sources as well as anthropogenic activities. There are many studies on this topic which refer primarily to the air pollution aspect and secondarily on its impact on health. The evidence of indoor air pollution and its impact on public health is consistent in showing adverse health effects at exposures that are currently experienced by urban populations in both developed and developing countries. The range of health effects is broad, but is predominantly to the respiratory and cardiovascular systems. All population is affected but susceptibility to the pollution may vary depending on the age and health condition of the subjects. The risk has been shown to increase with exposure and there is little evidence to suggest a threshold below which no adverse health effects would be anticipated.

Recently the research interest is focused on the indoor concentration of pollutants such as particulate matter, carbon monoxide, carbon dioxide and total volatile organic compounds. Naturally the evaluation of the indoor environment is also affected by the outdoor regime.

The sources of indoor pollution and its impact has been a research topic in a great number of studies. In particular detailed research on the impact tobacco smoking on human health was discussed by Rushton (2004). Indoor air quality measurements performed in the work environment in Greece (Lagoudi et al, 1996, 1996a, Bernhard et al, 1995), followed by similar studies in schools (Synnefa et al, 2004, Chaloulakou and Mavroidis, 2002), hospitals (Santamouris et al, 1994) and residences (Baya et al, 2004, Alexopoulos et al, 2006, Lai et al 2006) report a number of pollutants including black smoke, nitrogen dioxide, volatile organic compounds, PM10 and PM2.5.
The present work takes into account both outdoor and indoor measurements as well as the type and characteristics of the buildings under investigation. The measurements

presented here extend to over seventeen private residences in a number of different locations in Athens.

EXPERIMENTAL CAMPAIGN

The experimental campaign incorporates automated instrumentation placed for short periods in and around seventeen private buildings. The details of these buildings are shown in table 1.

No	Cite	Location	Area (m2)	Building type	Inhabitants number	Specific remarks
1	Drafi	Rural	240	Detached house	2	Non smokers
2	Ekali	Suburban	100	Apartment	1	Non smokers
3	Ekali	Suburban	110	Apartment	3	Non smokers
4	Drafi	Rural	210	Detached house	2 (+1 child)	Non smokers
5	Ekali	Suburban	400	Detached house	4 (+1 child)	Non smokers
6	Ilioupoli	Urban	95	Detached house	1	Non smokers
7	Vironas	Urban	90	Apartment	1	Non smokers
8	Vironas	Urban	90	Apartment	2	Non smokers
9	Papagou	Urban	154	Detached house	4	Smokers
10	Nea Erithrea	Suburban	65	Apartment	1	Smokers
11	Koropi	Urban	104	Detached house	2	Non smokers
12	Pireas	Urban	147	Detached house	4	Smokers
13	Haidari	Urban	61	Apartment	2	Smokers
14	Haidari	Urban	61	Apartment	3	Non smokers
15	Brahami	Urban	150	Detached house	3	Smokers
16	Nea Ionia	Urban	110	Detached house	2 (+2 children)	Smokers
17	Nea Ionia	Urban	110	Detached house	3	Non smokers

The instrumentation included monitors of the determination of particulate matter with dynamic diameter 10, 2.5 and 1 μm (PM10, PM2.5, PM1), total volatile organic compounds (TVOCs), carbon monoxide (CO) and carbon dioxide (CO2).

The instrument used to collect the particulates is a lightweight personal sampling device consisting of a single-stage impactor and a final filter. Aerosol particles are sampled through the single-stage impactor to remove particles above the cut-point, which is either 2.5 or 10 μm aerodynamic equivalent diameter, depending on which a sampling head is chosen. Particles smaller than the cut-point are collected on a 37mm

diameter filter. The impactor uses the principle of inertial separation of airborne particles.
Measurements of carbon monoxide, carbon dioxide and TVOCs have been performed using a multigas analyser. The gas analyser is a highly accurate, reliable and stable quantitative gas monitoring system. By installing up to five filters in the analyser, it can measure the concentration of up to five component gases and water vapour in any air sample. Although the detection limit is gas-dependent, it is typically in the 10^{-2} ppb region. The analyser has a built-in pump system that allows samples to be drawn from up to fifty metres away.

RESULTS

The results are conveniently divided into two categories: homes without smokers and homes with smokers.

Non smoking residences

A guideline for the carbon dioxide concentration limit has been set by the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE). This allowance is 1000 ppm for schools, offices, and areas where people spend extended periods of time indoors. The Federal Standard for exposure limits of carbon monoxide in the home is 50 ppm. The Environmental Protection Agency (EPA) standard for ambient air is 9 ppm averaged over an eight-hour period and 35 ppm for one hour.
The mean measured concentrations of the carbon dioxide, carbon monoxide and total volatile organic compounds are shown in Figure 1. The average indoor concentration varied between 350 and 610 ppm for all residences without smokers, however measurements of the maximum indoor CO2 concentration showed that 85% of the buildings were below the 1000 ppm limit.

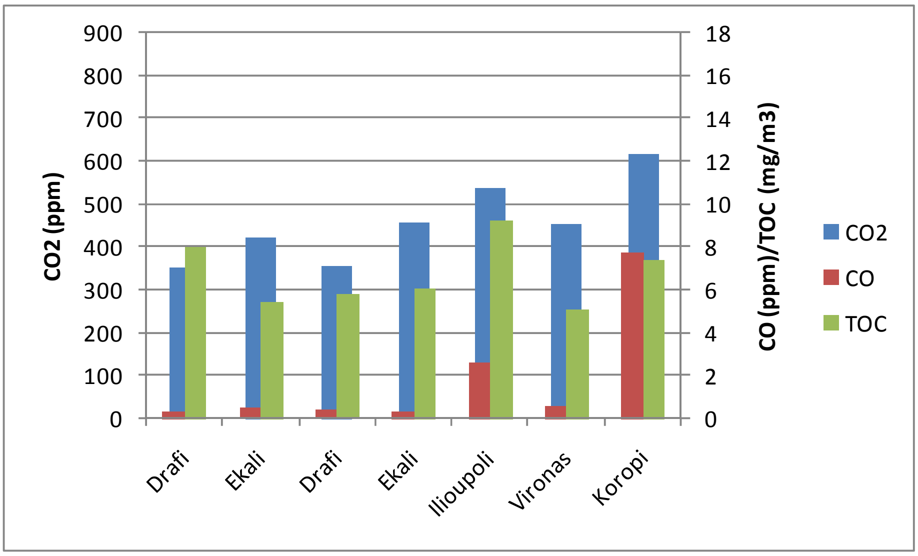

Figure 1. Mean concentration of carbon dioxide, carbon monoxide and total volatile organic compounds for houses without smokers

The mean carbon monoxide concentration did not exceed the 9 ppm limit in any of the residences tested. The maximum carbon monoxide concentration showed that approximately 16% of the buildings were above the limit.
The mean concentration of the total volatile organic compounds (TVOC's) in the non smoking homes was between 5 and 9 mg/m^3, while the absolute maximum measured concentration reached 15 mg/m^3. According to Molhave (1990) the maximum permitted concentration for TVOC's depends on the health consequences. It is considered that indoor concentrations below 0.2 mg/m^3 do not pose any threat for the human health, while concentrations between 0.2 to 3 mg/m^3 may cause discomfort when in combination with other environmental problems. Concentrations higher than 3 mg/m^3 have been shown to cause respiratory problems.
The existing air quality standards for PM_{10} entered into force on 1 January 2005 (Directive 1999/30/EC). The daily limit is 50 microgrammes per cubic metre (averaged over 24 hours) and the annual limit 40 microgrammes per cubic metre. This limit can be exceeded for up to 35 days a year taking account unusual and adverse weather.
The limit on $PM_{2.5}$ concentrations introduced by WHO Air Quality Guidelines is 25 micrograms per cubic metre, 24-hour mean. On the contrary, limits on PM1 concentrations do not exist.

The measured concentrations of PM10, PM2.5 and PM1 in the non-smoking residences are shown in figures 2-4. On average the concentration of PM10 was approximately 42.5 μg/m3 and the mean concentration of PM2.5 and PM1 was 31.8 μg/m3 and 25.2 μg/m3 respectively.

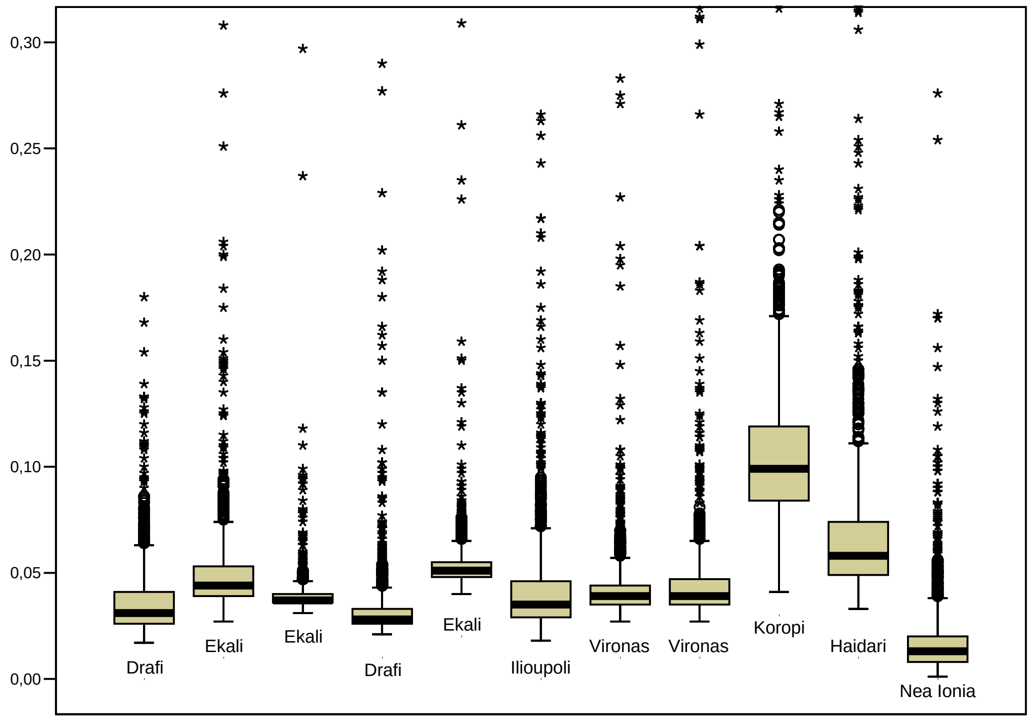

Figure 2. PM10 concentrations in the non-smoking residences (in mg/m^3)

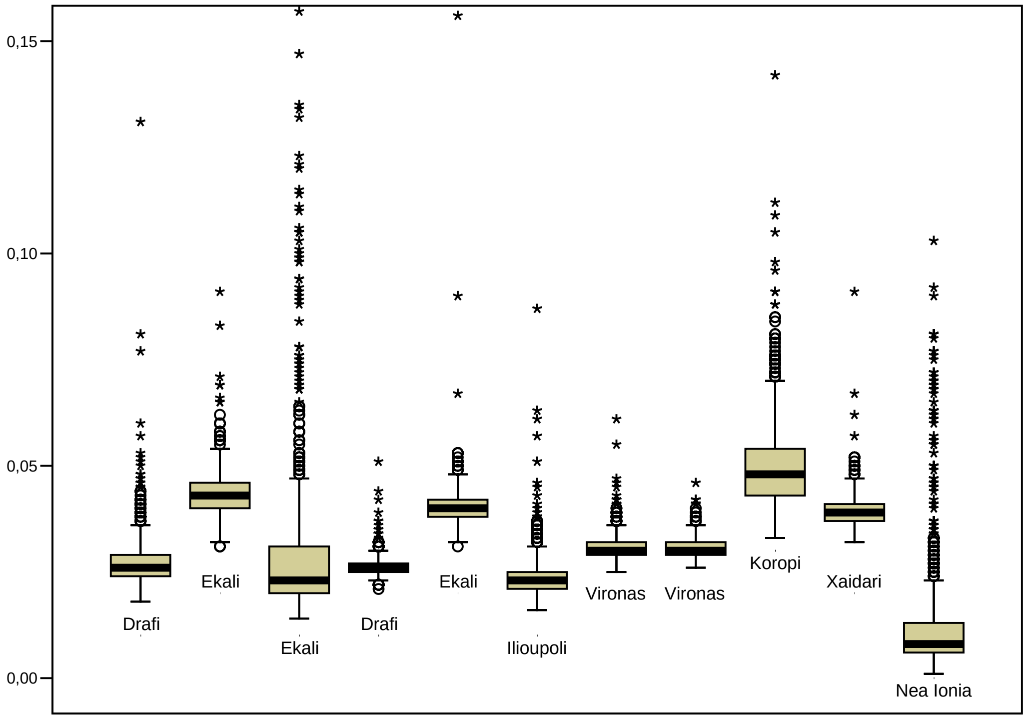

Figure 3. PM2.5 concentrations in the non-smoking residences (in mg/m^3)

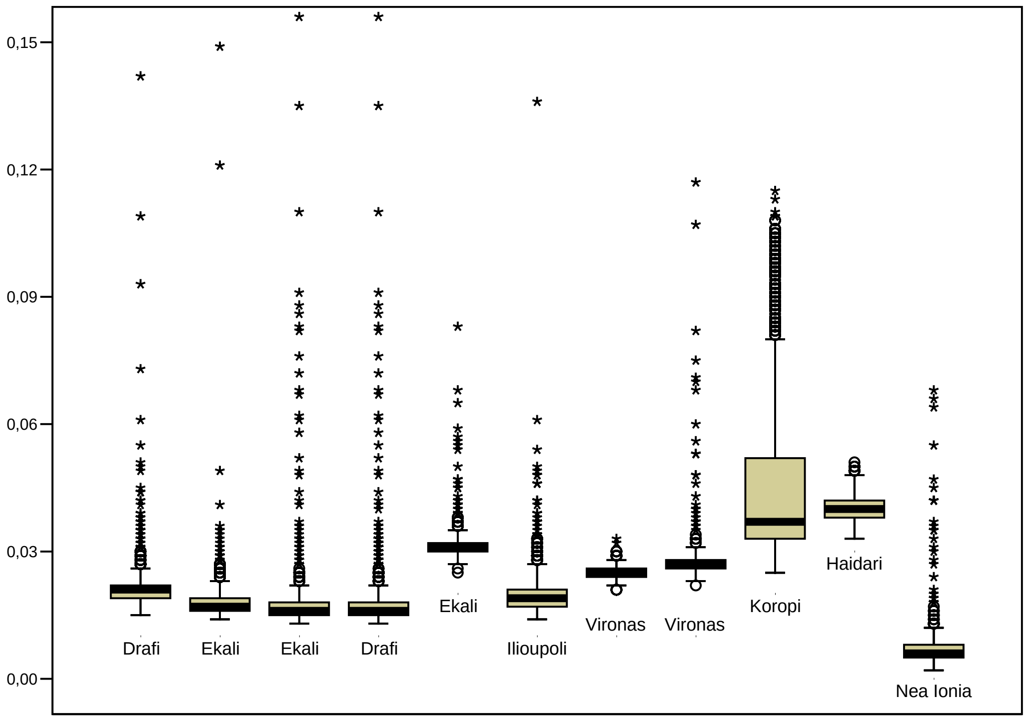

Figure 4. PM1 concentrations in the non-smoking residences (in mg/m^3)

In the majority of the non smoking residences (approximately 72%) the concentrations of PM10 measured were below the 50 µg/m3 limit, while 28% presented PM10 concentrations ranging from 50 to 90 µg/m3. Measurements

performed in dwellings in Beijing found that the mean indoor PM_{10} concentration was 109.9 μg/m3, (Houyin, Z. et al, 2005)
Measurements of the PM2.5 concentrations, however, showed that only 18% of the residences were below the limit of 25 μg/m3, 63% of the buildings ranged from 25 to 50 μg/m3 and 19% of the buildings reached 70 μg/m3. Measurements performed in 20 residences in California found that the median $PM_{2.5}$ concentrations were 32.2 μg/m3, (Sawant et al, 2004).
It is important to mention here that the PM1 concentrations showed that 71% of the residences were below 30 μg/m3, while 29% were between 30 and 70 μg/m3.

Smoking residences
Tobacco smoking is a major source of indoor carbon dioxide and contributes highly to indoor TVOC's concentration (Santamouris et al., 2006).
The mean measured concentrations of the carbon dioxide, carbon monoxide and total volatile organic compounds are shown in Figure 5. The average indoor concentration varied between 450 and 810 ppm for all residences with smokers however, measurements of the maximum indoor CO2 concentration showed that approximately 33% of the buildings were above the 1000 ppm limit.

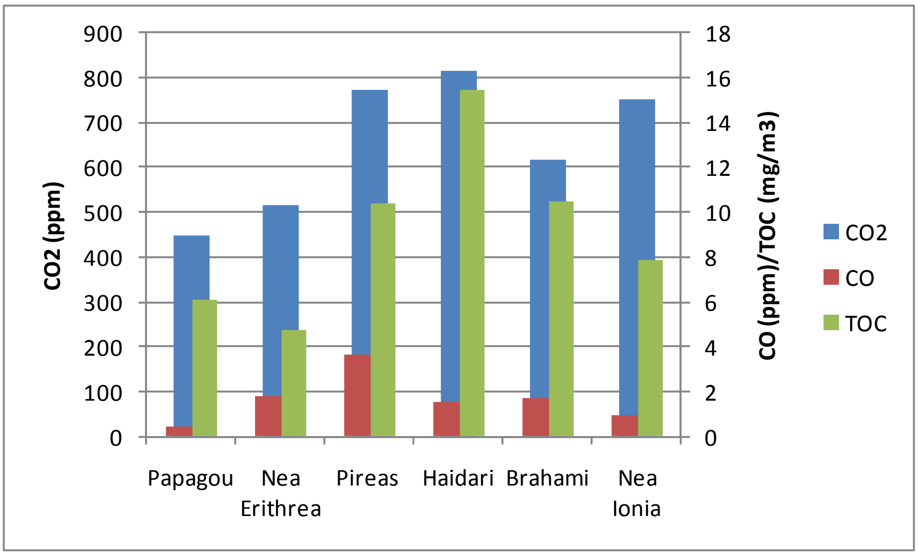

Figure 5. Mean concentration of carbon dioxide, carbon monoxide and total volatile organic compounds for houses with smokers

The mean carbon monoxide concentration did not exceed the 9 ppm limit in any of the residences tested. On the contrary, the maximum carbon monoxide concentration showed that approximately 50% of the buildings were above the limit.
The mean concentration of the total volatile organic compounds (TVOC's) in the smokers' homes was between 5 and 15 mg/m^3, while the absolute maximum measured concentration reached 37 mg/m^3.
The measured concentrations of PM10, PM2.5 and PM1 in the smokers' residences are shown in Figures 6-8. On average the concentration of PM10 was approximately 105 μg/m3 and the mean concentration of PM2.5 and PM1 was 224 μg/m3 and 131 μg/m3 respectively.
In the case of residences with smokers the majority of the buildings present very high PM10 and PM2.5 concentrations. In approximately 67% of the buildings the PM10

and PM2.5 concentrations exceeded 100 µg/m3, while in 33% of the buildings were 50 µg/m3.

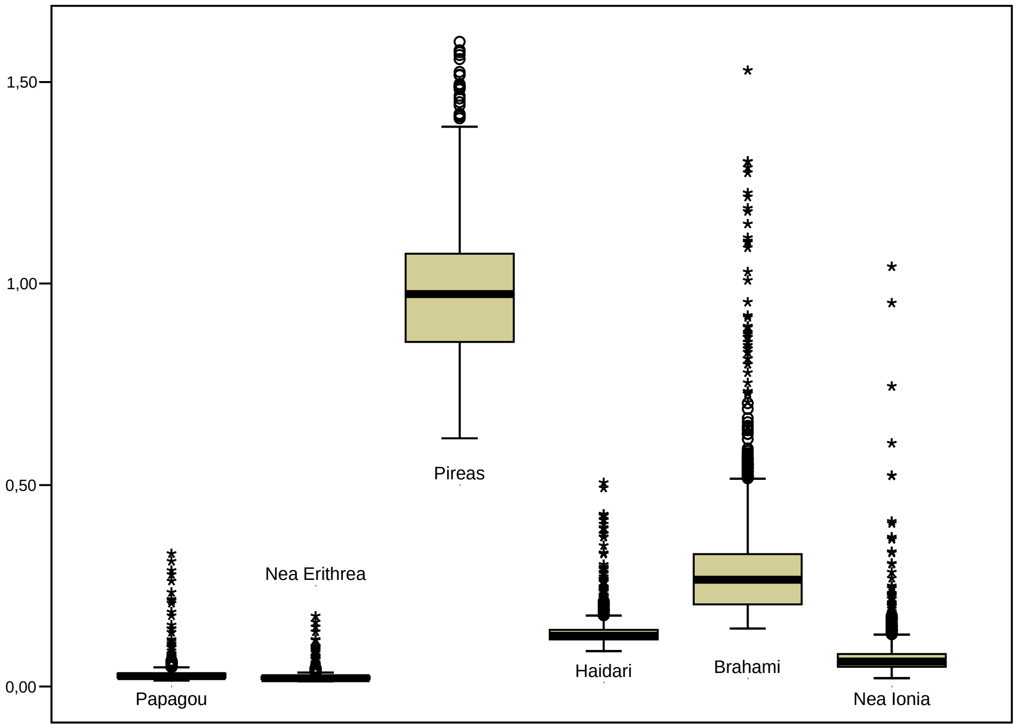

Figure 6. PM10 concentrations in the smoking residences (in mg/m^3)

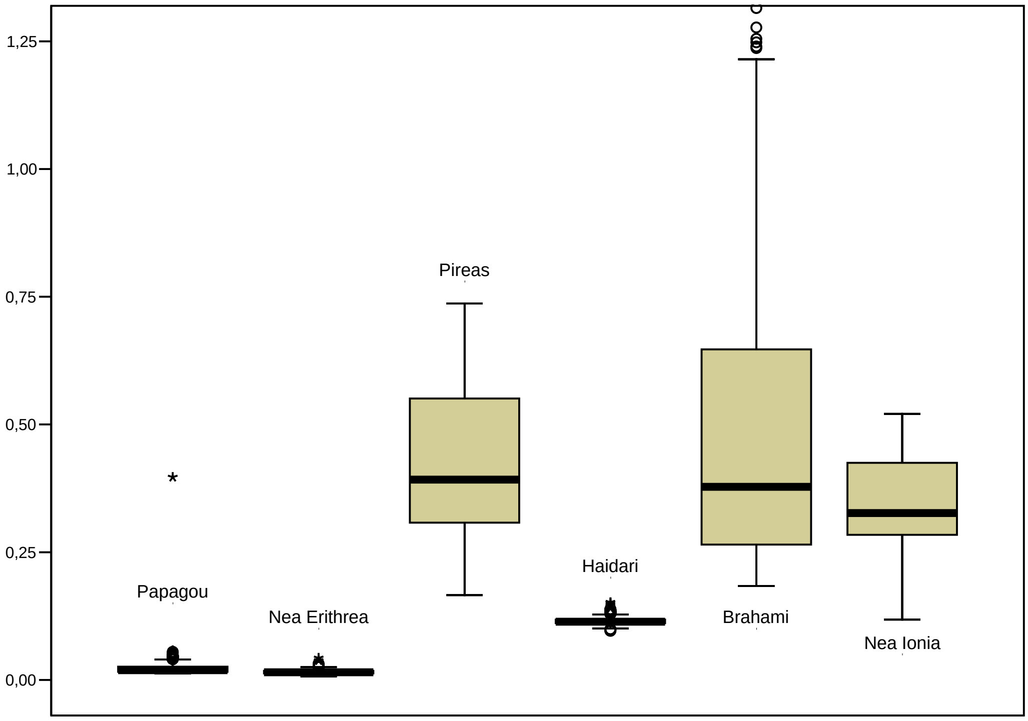

Figure 7. PM2.5 concentrations in the smoking residences (in mg/m^3)

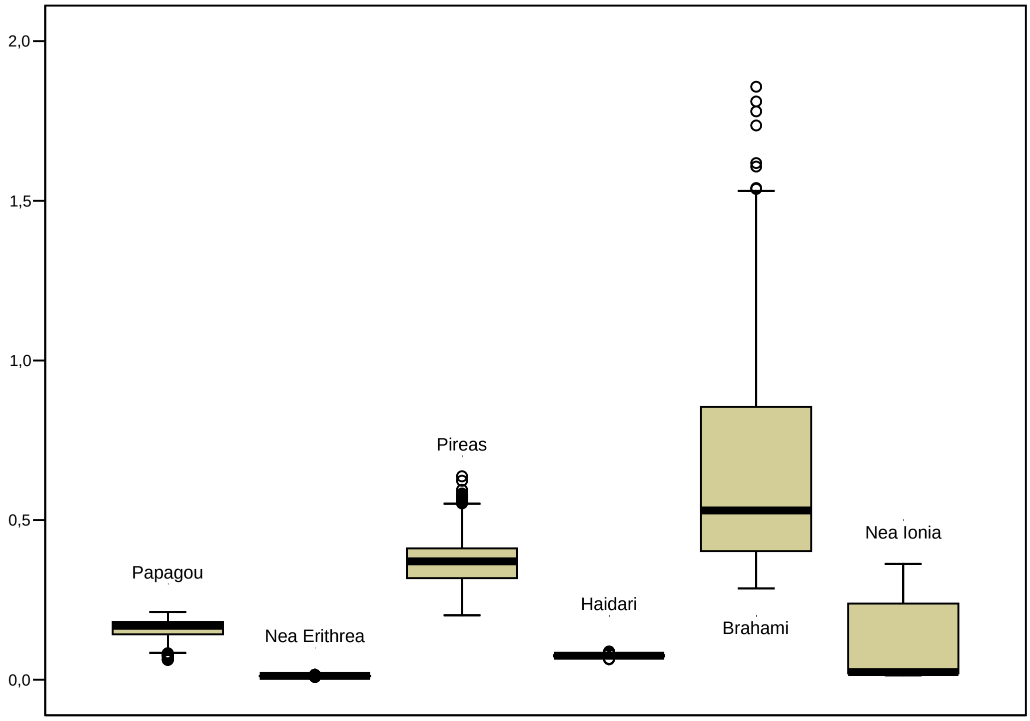

Figure 8. PM1 concentrations in the smoking residences (in mg/m^3)

Measurements of the PM1 concentrations however showed that only 17% of the residences were below 50 µg/m3 and in 83% of the buildings the concentrations measured were higher than 100 µg/m3. The above levels are very high and may pose a serious threat for the health of the residents.
For the PM2.5 almost 24 % of the dwellings present a concentration below 50 µg/m3, while the 64 % are below 100 µg/m3. The ratio between PM2.5/PM10 varies between 0.2 to 0.8. Measurements performed in 20 residences in California found that the median $PM_{2.5}$ concentrations were 32.2 µg/m3, (Sawant et al, 2004).

CONCLUSIONS

Measurements of the indoor concentration of CO2, CO, TVOC's and PM10, PM2.5and PM1 quality in seventeen buildings have been performed in the greater area of Athens, Greece. The main conclusions are :

- CO2 concentration measurements showed that in houses with smokers the 1000 ppm limit was exceeded in more cases than in houses without smokers. On average, buildings without smokers were found to have lower CO2 concentrations than the 600 ppm limit.
- Concerning the average CO levels, the 9 ppm limit is not exceeded in any case. Maximum carbon monoxide concentrations increase highly as a function of smoking levels inside the dwellings.
- The mean total volatile organic compounds (TVOC's) measured in all homes was above 5 mg/m^3. The absolute maximum measured concentration for residences without smokers did not exceed the limit of toxicity of 25 mg/m^3 while this was not the case for the houses with smokers.

- In the majority of the non smoking residences the concentrations of PM10 measured were below the acceptable limit however the PM2.5 concentrations showed that only 18% of the residences were below the acceptable limit. The highest measured PM2.5 concentration did not exceed 70 μg/m3. PM1 concentrations showed that 71% of the residences without smokers were below 30 μg/m3, while 29% were between 30 and 70 μg/m3.
- All the smokers' residences had concentrations of PM10 and PM2.5 exceeding the acceptable limits while in most cases the measurements approached the 100μg/m^3 mark. PM1 concentrations in only 17% of the residences with smokers were below 50 μg/m3 and in 83% of the buildings the concentrations measured were higher than 100 μg/m3.

In conclusion it is obvious that indoor pollution in homes has a major significance in the health and well being of their inhabitants. There is an unmistakable effect of tobacco smoking in the indoor air quality of dwellings and measurement clearly show non-smokers homes to have a cleaner environment with fewer adverse effects due to indoor air pollution.

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Production and Characterization of Air Suspendable, Dusts Containing Common Indoor Allergens for Inhalation Exposure Research

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SUMMARY

Controlled milling of natural source materials has led to establishment of an allergen carrier particle sample bank for common, indoor, allergen proteins. Allergen "reference" powders are obtained from controlled processing of spent dust mite culture, roach colony fras, and cat and dog fur samples from pet grooming establishments. Due to wide variations in parent material particle size, shape, hardness or tensile strength, different milling prescriptions are required to produce appreciable yields of suspendable particles in the 0.5 - 20 µm size range with substantial mass fractions less than 12 µm. Characterization data includes: mass fraction and number particle size distributions determined by optical scattering and inertial impactor methods; particle morphology as determined by scanning electron microscope examination; allergen concentrations determined by immuno-assay enzyme-linked analysis (ELISA); inhalation exposure risk analysis of the powders as reservoir dusts. Certified quartz powders (99% < 10 microns) and Bacillus Thuringiensis spores serve as reference powders.

INTRODUCTION

Epidemiological studies report significant increases in health care and hospitalization of patients for respiratory system related diseases, such as asthma, over the last 35 years [1-3] According to the Center for Disease Control and Prevention, approximately 5 - 6 % of all Americans suffer from asthma and approximately 5000 people die each year of asthma related complications. The economic burden of this illness in the United States is estimated at $12.7 billion dollars per year[1]. The allergic response disease trends appear to be mirrored in urban societies globally[2].

Allergic response and asthmatic disease symptoms may be triggered in genetically predisposed individuals and developed in non-atopic individuals by repeated exposure to allergens. The link between allergic response diseases and common indoor allergens is based on epidemiologic investigations of allergic response related hospital visitations, environmental measurements of protein allergen levels in building reservoir dusts of asthmatic populations, and specific IgE antibody levels in identified allergic and asthmatic populations [4, 5].

Indoor allergens are most often specific protein structures associated with cat and dog fur and saliva, cockroach and dust mite droplets and body parts, mice urine, as well as some fungi. These sources contribute allergens to the reservoir dusts of building surfaces, either in standalone particulates or, upon mechanical and microbial disintegration of parent particles, as adherents to "inert" dust particles. When disturbed and aerosolized by human activity the allergen containing reservoir dusts become available for inhalation exposure with resultant allergic sensitization and bronchial hyperresponsiveness disease development, i.e., asthma, in susceptible individuals. Protein allergens have molecular weights in the tens of thousands of Daltons and sizes in the nanometer range, but are associated with micron sized “carrier” particles in building dusts (See Figure 1).

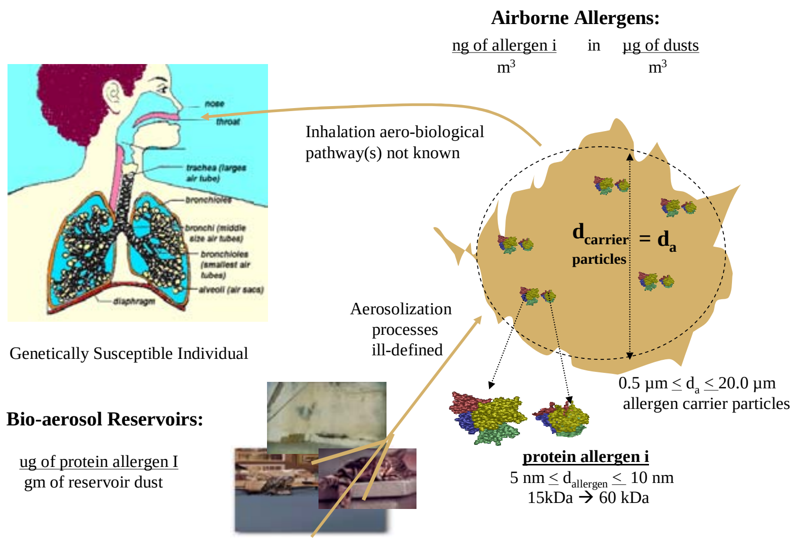

Figure 1: Allergens are “Carried” by Micron-Size Particles in Indoor Environments

The lack of quantified aerobiological exposure pathways obtained under controlled parameter conditions is limiting our understanding of the “distribution of inflammation and airway obstruction” processes associated with allergic and asthmatic responses to indoor environment allergens. The allergen inhalation exposure problem is a subset of the general indoor air problem of exposure to PM_{10} and $PM_{2.5}$ contaminant containing particles. Pesticide and insecticide containing carrier particles in indoor environments present a similar challenge[6, 7]. The disparity in particle resuspension parameters reported in estimating inhalation exposure dosages is perhaps most indicative of the current state of knowledge [8, 9].

Limited particle-size-resolved data reduces greatly our knowledge of what is “actually deposited in upper and lower respiratory tracts…” and the aerobiological pathways leading to inhalation exposure[3, 10, 11]. As a result, epidemiologic risk factors, for inhalation based sensitization, are based on allergen concentrations in reservoir dust samples for bulk samples of particles sizes < 150 microns (See Table 1). A more quantitative description of the resuspension and depositions processes of naturally occurring allergen carrier particles is required.

Table 1: Risks relative to allergen concentration in reservoir dusts[12]

Risk of Sensitization for Atopic Children	Mite Group 1 µg/g	Fel d 1 µg/g	Can f 1 µg/g	Bla g 1 U/g	Bla g 2 µg/g
High	>10	1-8	1-8	>8	>1
Medium	2-10	8-20	8-20	1-8	0.08-0.4
Low	<0.3	<0.2 or >20	<0.2 or >20	<0.6	<<0.08

Site-to-site property variations in reservoir dust samples, low mass fractions of samples having particle sizes with low settling velocities, and the limited sensitivity of standard ELISA techniques for determining allergen concentrations of a sample, restricts the usefulness of field collected samples.

A set of allergen "reference" dusts with reproducible characteristics with substantial allergen concentrations for ELISA measurement allergen tracking - but derived from natural source materials -is needed. These powders are then utilized to establish reference resuspension and aerodynamic behavior, filtration susceptibility and protein denaturing properties of the allergen proteins in natural material carrier particles.

The objectives of this specific effort are to:

1.) determine the feasibility of establishing reproducible methods of making samples of air suspendable allergen carrier particles in the 0.5 to 20 micron size range from natural allergen source materials;
2.) characterize the resultant powdered samples with respect to size distribution parameters, particle shapes, allergen concentration, and respiratory system deposition probability properties.

METHODS

Spent mite dust culture is provided by Indoor Biotechnologies Corporation (M. Chapman) and is the source of the initial dust mite target allergen (Der p 1). Laboratory roach colonies of both German and American and the resultant fras is the source of the roach allergens (German: Bla g 1, Bla g 2; (American: Per a 1 -cross reactive to Bla g 1). Cat and dog fur parent materials from pet grooming established are utilized as sources of target canine (Can f 1) and feline (Fel d 1) allergens.

Considerable differences in parent source material particle sizes, shapes, hardness and/or tensile strength necessitate very different sample milling prescriptions to obtain air suspendable powders. Organic based materials are easily over milled, producing large mass fractions of micron and sub-micron particles with significant agglomerating tendencies. For particles sizes of interest in indoor air quality, one wants to mill samples so that the resultant powder has significant mass fractions between 1 and 20 microns, with a large mass fraction < 12 microns. It is these particles that are most frequently observed in investigations of airborne PM_{xx} investigations, contain identified protein allergens, are most susceptible to iterative deposition and resuspension during occupant activities. During the milling process, sample temperatures must be kept low, so that thermal denaturing of any indigenous protein allergens does not occur. Depending on the size characteristics and physical properties of the parent source material, the starting material may be subjected to some initial shredding.

Optical particle counting, inertial separation and scanning electron microscope techniques are implemented to characterize the allergen-containing powders for their size distribution, shape and surface roughness. Specific allergen contents (Der p 1, Bla g 1, Bla g 2, Can f 1, Fel d 1) of the "reference dusts" are determined by enzyme linked, immunoassay techniques (ELISA). The particle size resolved allergen content distribution allows potential respiratory system delivery profiles to be quantified. Figure 2 displays the overall allergen, carrier particle powder production process.

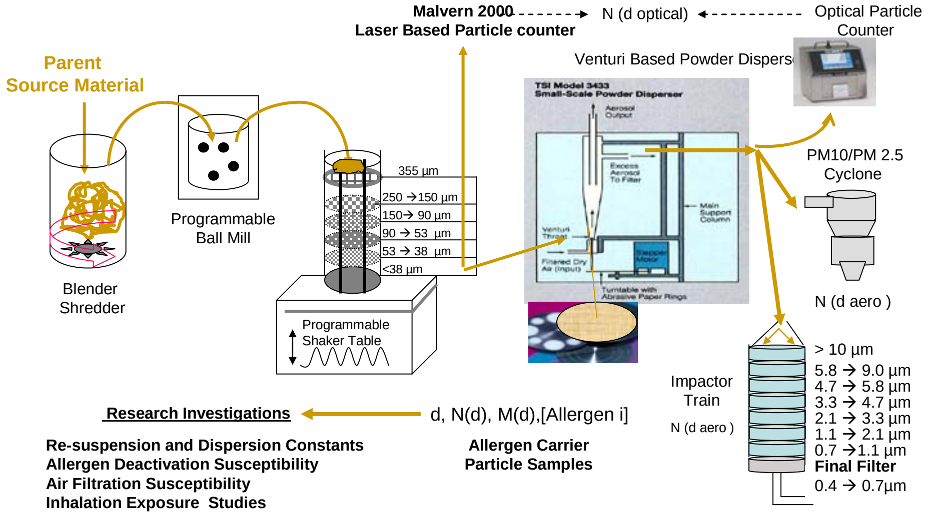

Figure 2: Allergen - Containing Carrier Particle Powder Production

RESULTS

The milled powders are subjected to optical particle size analysis using a Malvern Mastersizer 2000 which employs a He-Ne laser and blue light sources for scattering. The system is configured for detection of 0.058 to 878 microns in 64 size bins. Samples are generally subjected to sonic probe or sonic bath treatment immediately before analysis to minimize small particle agglomeration tendencies. Table 2 shows the overall particle size characteristics of the milled dust samples as well as that determined by the Malvern Mastersizer 2000 for the certified quartz reference powder and the Bacillus Thuringiensis spore powders provided by the US Army CHPPM.

Table 2: Particle Size Properties of Milled, Quartz Dust, and B.T. Spores

Sample Type	Mastersizer 2000 Determined Diameters (microns)				
	Volume Mean	Area Mean	Volume 0.X mass fraction below d (µm)		
	D[4,3]	D[3,2]	D[v,0.1]	D[v,0.5}	D[v,0.9]
Quartz Ref.	4.14	2.08	0.91	2.98	9.07
B.T. Spores	11.14	3.14	1.15	8.77	24.50
Dust Mite (I)	13.75	5.82	2.23	12.93	25.84
Dust Mite (II)	13.00	3.48	1.63	12.24	25.41
Roach Fras (I)	22.26	2.54	2.65	18.74	47.18
Roach Fras (II)	30.39	3.38	1.06	17.84	77.49
Dog Fur (I)	26.46	2.88	1.75	13.93	62.27
Cat Fur (1)	37.56	2.89	1.78	19.1	91.95

To insure consistency, mass mean sample determinations were calculated and compared to the instrument based volume mean samples, D[v,0.5] values. The specific gravity of the dust mite samples were determined experimentally, but the other allergen dust samples specific gravities were estimated. The specific gravity of the quartz reference dust samples were provided with the certification documentation from the vendor.

ELISA determined specific allergen content of the powders are shown in Table 3. These concentrations are generally high relative to the mixed dust samples from field studies, allowing the samples to be used in particle resuspension, filtration and protein denaturing studies while avoiding "inert" dust sample masking of the allergen carrier particle behavior. Extensive examination of the mite powders in size bins in the 1 to 10 μm range revealed about 25% variation in allergen concentrations around the mean shown. Other samples have not yet been examined for particle size allergen content variation in the milled powders.

Table 3: ELISA Allergen Concentrations of Samples

Sample Type	Mass Fraction d < 12 μm	D[v,0.5]= 50% sample d cutoff (μm)	Calculated Mass Mean (μm)	ELISA Allergen i μg i/gm dust
Quartz Ref.	0.96	2.98	4.47	NA
B.T. Spores	0.69	8.77	10.32	NA
Dust Mite (I)	0.47	12.93	14.84	4092 Der p 1
Dust Mite (II)	0.5	12.24	14.03	4180 Der p 1
Roach Fras (I)	0.36	18.74	24.02	448 Bla g 2
Roach Fras (II)	0.45	17.84	25.19	340 Bla g 2
Dog Fur (I)	0.47	13.93	20.66	3.4 Can f 1
Cat Fur (1)	0.39	19.1	22.08	75.6 Fel d 1

Some samples were analyzed with and without powder disaggregating sonic treatment (Table 4) to determine the agglomerating tendencies of the milled samples, relative to the quartz reference dusts and the US Army provided Bacillus Thuringiensis spore samples. The quartz samples are not expected to agglomerate significantly whereas it is know that pure spore samples display agglomerating behavior.

Table 4: Agglomerating Tendencies of Samples

	Mass Fraction	Charcteristic Diameters (microns)			
Sample ID	d<12 microns	Mass Mean	Volume Mean	D[v,0.1]	D[v,0.5}
Quartz Reference wo*	0.95	4.73	4.38	0.95	3.14
Quartz Reference w	0.96	4.47	4.14	0.91	2.98
B.T. Spores wo Sonic	0.53	14.72	13.64	2.49	11.54
B.T. Spores w Sonic	0.69	10.32	11.14	1.15	8.77
Dust Mite (I) Culture wo	0.05	52.73	48.85	19.54	43.15
Dust Mite (I) Culture w	0.02	51.63	47.83	15.30	41.69
Milled Cat Powder wo	0.11	32.99	50.83	10.44	45.89
Milled Cat Powder w Probe	0.43	18.4	17.04	1.00	14.74
Milled Cat Powder w Bath	0.37	19.76	18.34	1.16	16.61

* w = with sonic treatment, wo = without sonic treatment

As expected the quartz showed little aggregation tendency whereas the B.T. spores showed significant aggregation as evidenced by the shift to a higher mass fraction of particles below 12 μm upon sonication and lower calculated mass mean, volume mean and D[v,0.x] diameters. The ~50 μm mass mean, nearly spherical, parent, mite culture particles, showed little aggregation tendencies, although the resultant milled dust displayed some aggregation propensities. Cat fur milled samples displayed the largest aggregation tendencies, as exemplified by the large shifts in mass fractions less than 12 microns and characteristic diameter properties of the samples when some form of sonication is utilized - the sonic probe treatment being more intense than the sonic bath. The milled cat fur material is significantly more aggregating than any of the other milled powders from the other natural material allergen source materials. Subsidiary experiments with the cat and dog fur powders revealed a high electrostatic susceptibility relative to the other materials, again, with the cat fur showing the greater susceptibility.

Reservoir Contaminant Inhalation Exposure Propensities of Samples

Contaminant inhalation exposure risks for a given contaminant carrying reservoir dust can vary greatly for two reservoir dusts having the same bulk contaminant concentration. The variations are due to:

1.) particle size dependencies of surface-to-air aerosolization and resuspension;
2.) differences in contaminant carrier particle size distribution with the bulk dust sample;
3.) differences in particle size dependencies of the carrier particle contaminant mass fraction concentrations;
4.) particle size dependency of respiratory system deposition probabilities;
5.) settling and deposition velocities of the contaminant carrier particle sizes.

Aerosolization and resuspension properties (3 above) are disturbance and reservoir surface type dependent and are considered in latter studies performed with the sample dusts. One can then define a Reservoir Contaminant Inhalation Exposure Propensity (RCIEP) of a reservoir dust contaminated with species i as the joint probability of property probabilities related to 2 - 5 above. The RCIEP(d) for particles of size d having a mass fraction $f_{N,i}(d)$ particles in reservoir particles of size d + d(d), having a bulk sample normalized $f_i^I(d)$ chemical contaminant mass fraction, a respirable deposition probability RSDF(d) and a settling time factor of ST(d) in a size bin around around diameter d would be:

$$RCIEP_i(d) = RSDF_i(d)* f_{N,i}(d) * f_i^I(d) * ST_i(d) . \quad (1)$$

$f_{N,i}$ (d) is the number fraction of particles of diameter d in the reservoir dust, i.e., the number of particles in dust of size d containing contaminant “i” normalized by the total particle count in unit mass of reservoir dust.

$f_i^I(d)$ is the average concentration of contaminant i in size d normalized by bulk dust sample contaminant concentration.

RSDF(d) is a specific respiratory system location deposition probability associated with aerosol particle of size d and as calculated by International Commission on Radiological Protection (ICRP) are equations describing the particle size dependencies of these probabilities [13, 14]

ST(d) is the 8 foot fall, air suspension time of a particle of the given size, d, normalized by the time it would take a 5 μm particle of unit density to settle 8 feet at standard air conditions (~1 hour) and as calculated by standard settling velocity equations in still air.

In practice, it is difficult to distinguish between non-contaminant carrying, reservoir, dust particles of diameter d and contaminant carrier particles of contaminant d in reservoir d. The two particle types are indistinguishable.. "$f_i^I(d)$" is then the mass fraction of total reservoir dust contaminant mass measured found in the size range around diameter d, so that the final form becomes.

$$RCIEP_i(d) = RSDF(d) * f_N(d) * f_i^I(d) * ST(d). \qquad (2)$$

Summing over all available particle bins for which data is available:

$$RCIEPi,_{\text{reservoir dust}} = \Sigma_d\ [RSDF(d) * f_N(d) * f_i^I(d) * ST(d)]. \qquad (3)$$

A reservoir dust consisting of equal numbers of pure 1 ($f_i(d) = 1$)micron spores has a much greater respiratory deposition RCIEP index than a corresponding 5 micron spore dust, partially because of a greater respiratory system deposition probability, but mainly because of its greater air "hang time," ST(d) property, relative to 5 micron particles.

The milled samples can now be characterized according to their overall RCIEP for inhalation as well as thoracic and alveolar regional depositions. These index estimates assume the mass fraction of contaminant in the milled samples are not particle size dependent, that is $f_{i,}^I(d)$ scales with the mass fraction of the sample represented by the number particle fraction in that size bin. Preliminary analysis of size resolved dust mite samples indicates about 25% variation in allergen concentrations throughout the 1 to 10 micron regime. But this particle size variation of contaminant concentrations is not included in the analysis presented here.

Table 5: Reservoir dust contaminant inhalation exposure probabilities

Sample ID	Mass Fraction d< 12 microns	Σ_d RCIEP Indices Inhalation	Thoracic Deposition	Alveolar Deposition
Quartz Ref.	0.96	0.65	0.52	0.35
B.T. Spores	0.69	1.19	1.09	0.96
Dust Mite (I)	0.47	0.60	0.51	0.42
Dust Mite (II)	0.50	0.50	0.60	0.51
Roach Fras (I)	0.36	0.68	0.60	0.53
Roach Fras (II)	0.45	0.79	0.94	0.88
Dog Fur (I)	0.47	1.10	1.05	0.93
Cat Fur (1)	0.39	1.10	1.03	0.95

The results indicate that subtle differences in the particle size distributions of the reservoir dusts can have a significant impact of the contaminant inhalation exposure probability indices, even assuming a particle size independent contaminant mass fraction contaminant distribution. These subtle differences appear in the detailed investigation of the sample specific particle size distributions of the powders.

SUMMARY

Air suspendable particles with ELISA traceable specific allergen contents have been processed from controlled milling of naturally occurring materials. Sufficient quantities of powdered particulate matter is obtained to fully characterize the particles size distributions, particle morphologies and specific allergen contents. Given a specific natural material source, the powder properties of various production batches are reproducible within ~25%. The powders are being utilized to investigate the surface-to-air aerosolization, resuspension, filtration and allergen denaturing susceptibility properties of indoor allergen carrier particles. The powders are amenable to utilization in inhalation exposure chambers for controlled particle size - allergen concentration investigations.

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Simulation of moisture and microbial problems in building

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SUMMARY

Clean indoor air is one of the most important factors for the welfare of the society. To understand and study the complex causality of different factors affecting indoor air quality, a databank and an IAQ-simulator have been developed. Databases have been developed to connect knowledge of material emissions to the indoor air measurement results. This allows screening of possible indoor air contaminant sources, provides reference data for both indoor air and material emission measurements and enables statistical evaluations of measured indoor air data when connected with diagnosed health effects. IAQ Simulator offers a new testing tool for building product manufacturers to develop and test materials in multilayer structures. The developed simulator can be incorporated to the "Indoor Climate" classification and CEN when drafting emission measurement standards for structures. The simulator offers also a new tool to study the complicated causal connections of building structure, materials and unexpected exposure to water and microbes, especially during the use of the buildings. The methodology and the simulator system generate new possibilities for advanced research. The use of molecular biological tools connected with IAQ simulator can give new opportunities for estimating the role of microbes and their interactions in indoor air and material related problems.

INTRODUCTION

Clean indoor air is one of the most important factors for the welfare of the society. Clean indoor air is also an important product of building industry. This has also been manifested in the Essential Requirement No 3 (ER 3) "Health, hygiene and the environment" of the Construction Products Directive (89/1106/EEC). In a studies of moisture damaged or SBS buildings the definition of moisture damage has had wide interpretation. The exposure to mould growth or moisture damage has been estimated by using visual inspections i.e. walk through by an expert, questionnaires to occupants and building maintenance staff as well as environmental sampling (e.g. Bornehag et al. 2001, Nevalainen et al. 1998).

The indoor climate contains partly the same VOC's as the outdoor air and partly VOC's emitted from building materials, paint, furniture etc. (Wolkoff, 1995) The VOCs, especially some terpenes might have an important role on indoor air quality. In very high concentrations in indoor air they might cause some sensory irritation. MVOC are special compounds produced by the different microbes growing in different building materials. The definition of MVOC, however, is not clear. VOC patterns in buildings suffering moisture problems can be difficult to analyse and detect. Mould fungi and microbes produce microbial volatile organic compounds (MVOC). There are several types of compounds, which may be evaporated from wet materials and microbes. Typical MVOC:s are e.g. geosmin or 3-octanol, 3-methyl-2-

butanol, 2-isopropyl-3-methoxypyrazine, 1-octen-3-ol, 2-octen-1-ol, 2-methyl-1-propanol (Claeson A. -S. and Sunesson A.L., 2005, Scleibinger at al., 2005, Wilkins et al., 2000, Korpi et al. 1998, Sunesson et al., 1997). The samples of MVOC are collected using Tenax probes and then analysed in laboratory using a gas chromatograph equipped with a flame ionisation detector (FID) and a mass selective detector (MSD). Lighter or heavier volatile hydrocarbons are normally collected by active-carbon filters so as SKC 226-09 or Tenax filters so as SKC 226-35-03 and the samples are analysed in laboratory using a gas chromatograph equipped with a flame ionisation detector (FID) and a mass selective detector (MSD).

There are suitable and adequate methods for the assessment of airborne viable and non-viable microorganisms, but the methods for viable microorganisms often do not allow long enough sampling times and thus underestimate the true number of viable fungi. Non-viable micro-organisms can be analyzed quantitatively by using a real-time PCR-method. PCR, Polymerase Chain Reaction, is a molecular biological technique where DNA can be extensively amplified in order to allow its detection. With real-time PCR it is possible to detect not only the presence of certain DNA like in traditional PCR, but also its quantity. In PCR, the DNA sequences of the organisms to be analyzed need to be known in order to have specific primers for the reaction. The disadvantages of PCR are that sample preparation from different matrices may be difficult, and the presence of inhibitors may interfere with the PCR reaction.

The most important factors causing mould and microbial growth are moisture, temperature and the exposure time (Viitanen and Salonvaara 2001). Also nutrients and pH of substrate are important. For the function of the building, several factors are affecting same time: materials, structure, ventilation, environmental exposure, humidity, air pressure and leakages etc. Modelling of the building physics has been developed to evaluate the complex system of building (Ojanen et al 2001). For evaluate the effects of different factors on the development of a moisture and microbial damage, an IAQ simulator was developed at VTT.

METHODS

PCR

With the real-time PCR method, the quantification of the overall fungal load as well as the specific species *Stachybotrys chartarum* and *Aspercillus versicolor* were studied. Both *S. chartarum* and *A. versicolor* are considered to be indicator organisms in damp buildings, and they have been shown to be able to produce toxins while growing on building materials. For the analyses, pieces of gypsum board, mineral wool, and test wall of the IAQ simulator were contaminated with fungal spores. Spores of 13 different fungal species were used. Different sampling techniques were also studied, including homogenisation of material samples, non-destructive surface sampling with a cotton swab, and air sampling with glass fibre filters. DNA was extracted from the samples with a commercial kit (FastDNA Spin kit for Soil, Qbiogene). Real-time PCR was performed with LightCycler (Roche Diagnostics, Gmbh). A universal fungal primer pair with DNA-binding SYBRGreen I dye was used for the detection of fungi, and TaqMan hydrolysis probe assays with specific probes were used for the detection of *S. chartarum* and *A. versicolor*. Conventional cultivation method using Potato Dextrose Agar (PDA, Difco) plates was used as a reference method.

MVOC

Analytical methods for the determination of volatile compounds produced by microbes (MVOCs) on building materials were developed. Specific MVOCs were clarified for

Penicillium roquefort and *Stacchybotrus chartarum* microbes during their growth on gypsum wall samples. The growth was monitored with a carbon dioxide detector. Passive sampling methods proved to be more reliable for the follow-up in comparison to active sampling. The VOC profile for the microbes was found to be affected by the mould species and air supply. The repeatability of MVOC measurements was by most affected by the mold growth rate. The change in carbon dioxide concentration during growth could not be detected when an air exchange of ~$0.4h^{-1}$ was coupled to the test chamber.

DATABANK

Chemical indoor air quality measurements have been carried out for more than two decades. Until now, the interpretation of the results has been very difficult due to the lack of scientifically proven dependence between VOCs and health effects. In this respect VTT material emission and indoor air quality databases provide a powerful tool. The databases have been developed 1) to enable to connect the knowledge of material emissions to the indoor air measurement results for screening the possible indoor air contaminant sources, 2) to provide reference data for both indoor air and material emission measurements, 3) to enable statistical evaluations of measured indoor air data when connected with diagnosed health effects.

These databases include, in addition to the actual measured contaminant concentrations and emission factors and other source related data, also information of the health effects and odour characteristics of the compounds. Connecting the data in the indoor air database with diagnosed health data it has been possible to demonstrate a statistically significant correlation between newly diagnosed asthma and indoor air TXIB concentration. We have also been able to identify other compounds and chemical groups that have statistically significant correlation to other indoor air related effects and symptoms.

IAQ simulator

An IAQ simulator was developed. An example of a structure was studied using a dividing wall structure between the different air spaces (Figure 2). The materials were chosen among the actual dividing wall structures used in Finland (a gypsum board as a wind barrier, frames from sawn spruce, fiberglass as the insulation material, the PE-film as damp-proofing and painted gypsum board as an inside wall material). In to the wall, different hatches for microbial growth can be introduced. The wall has a hole on top of it and a tube inside the fibreglass to direct the additional moisture inside the structure. This hole and tube also makes it possible to measure the humidity inside the structure. The amount of water needed to raise the humidity in the wall is calculated based on the materials used, their amounts and the final RH wanted. The microbes and VOC were collected from the air and from the wall component. The RH, temperature, influence of air pressure and air exchange was measured simultaneously.

RESULTS

PCR

All fungal species tested representing 13 indoor fungal genera could be quantified with the real-time PCR using universal fungal primers and SYBRGreen I chemistry. Only slight variations in the efficiency of DNA amplification were observed between the DNA samples of different fungal genera, except for DNA of *Fusarium* species, which amplified the most effectively. The practical detection limit of the PCR assay was determined by adding a known amount of spores into a cotton swab and analysing the swab extract by the real-time PCR. Approx. 100 spores of *Penicillium roqueforti* could be detected from a cotton swab, indicating a detection limit of 100 cfu (colony forming units)/g or 1 cfu/cm^2, if the sampling area is considered to be 100 cm^2, as recommended by the Ministry of Social Affairs and Health in Finland (Table 1). Ten times higher amount was needed for the detection of *S. chartarum* spores. Similar detection limits (100-1000 spores) were obtained when glass fibre filters contaminated with *P. roqueforti* or *A. versicolor* were studied.

Table 1. Approximate detection limits of the real-time PCR assays including DNA extraction.

	Real-time PCR assay		
DNA sample	Universal fungal	*S. chartarum*	*A. versicolor*
Pure fungal DNA / reaction	1.5 pg - 150 ng	40 fg - 40 ng	1.3 pg - 130 ng
Pure spore suspension	1000-10 000 spores	1000 spores	1000-10 000 spores
Spores in cotton swab	100-1000 spores	100 spores	100 spores
Spores in glass fiber filter	100-1000 spores	nd	1000 spores

nd = not determined

S. chartarum and *A. versicolor* could also be detected with specific real-time PCR reactions using the TaqMan chemistry. The detection limit for *S. chartarum* spores form artificially contaminated mineral wool or gypsum board was approx. 1000 cfu/g or 20 cfu/cm^2, depending on the sampling technique. Both *S. chartarum* and *A. versicolor* specific PCR were able to detect approx. 100 spores from the cotton swab, the same detection limit as obtained for fungal-specific PCR (Table 1).

The sampling method, destructive homogenisation or non-destructive surface sampling, had only a minor effect on the PCR results. No differences were observed when *Penicillium* spores were used for the contamination of gypsum board. Homogenisation was found to be slightly more effective compared to the non-destructive cotton swab method when *S. chartarum* spores were harvested from a recently contaminated gypsum board. It is known that *S. chartarum* produces fewer spores, and the spores may not be as easily released from the mycelium as the spores of *Penicillium* species, which might explain this result.

MVOC

Higher VOC amounts were measured in the tests with *Penicillium Roquefort* compared to *Stacchybotrus Chartarum* (10-900 ng/ tube vs. 5-45 ng/tube, passive sampling four weeks, air exchange rate ~$0.4h^{-1}$). The least specific MVOC growth indicator for both mold species was butanol. Specific VOCs for *Stacchybotrus Chartarum* (not detected in the control chamber): 3-octanone, 1-octen-3-ol and methyl benzoate.

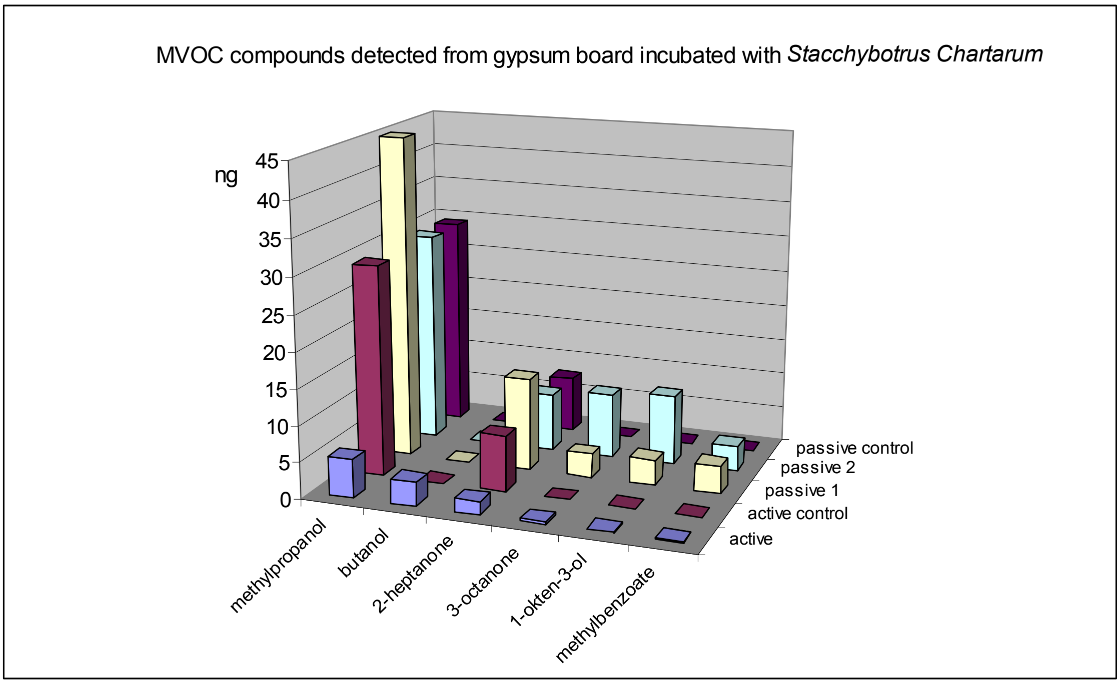

Figure 1. MVOC compounds detected from gypsum board incubated with *Stacchybotrus Chartarum* (incubation in glass chambers for 28 days, passive and active sampling on Tenax TA adsorbent tubes, results are given as the average amount from three different chambers).

Tests showed that temperature, humidity, air tightness, air velocity, air exchange rate and eventually background concentration were below the values set in the standard. The sink effect is a parameter that describes how much VOCs adhere to the walls of the test chamber. The ideal situation would be such that VOCs wouldn't adhere to the walls at all and they all could be sampled to test tubes, but in real life this objective is impossible to achieve, there are always some molecules stuck on the walls. The sink effect (recovery) didn't quite fit to the standard. With n-dodecane, the sink effects were almost as good as the standard demands, but tests with toluene did not succeed because the test method set in the standard wasn't suitable for the calibration of our FID. The results of the improvements are presented in Table 1.

Table 1 Test results of IAQ simulator 'Sissi' compared with ISO 16000-9 standard

Quantity	Standard	IAQ Simulator 'Sissi'
Temperature	23 ºC ± 2 ºC	23.5 ºC ± 0.5 ºC
Relative Humidity	50 % ± 5 %	49 % ± 1 %
Air tightness	leak <5 %	leak < 3 %
Background concentration	TVOC < 20 µg/m³, individual VOC < 2 µg/m³	TVOC < 20 µg/m³, individual VOC < 2 µg/m³
Recovery	> 80 % toluene and n-dodecane	73% n-dodecane
Air velocity	0,1–0,3 m/s	0,2 m/s
Air exchange rate	0,5 1/h ± 5 %	0,5 1/h ± 3 %

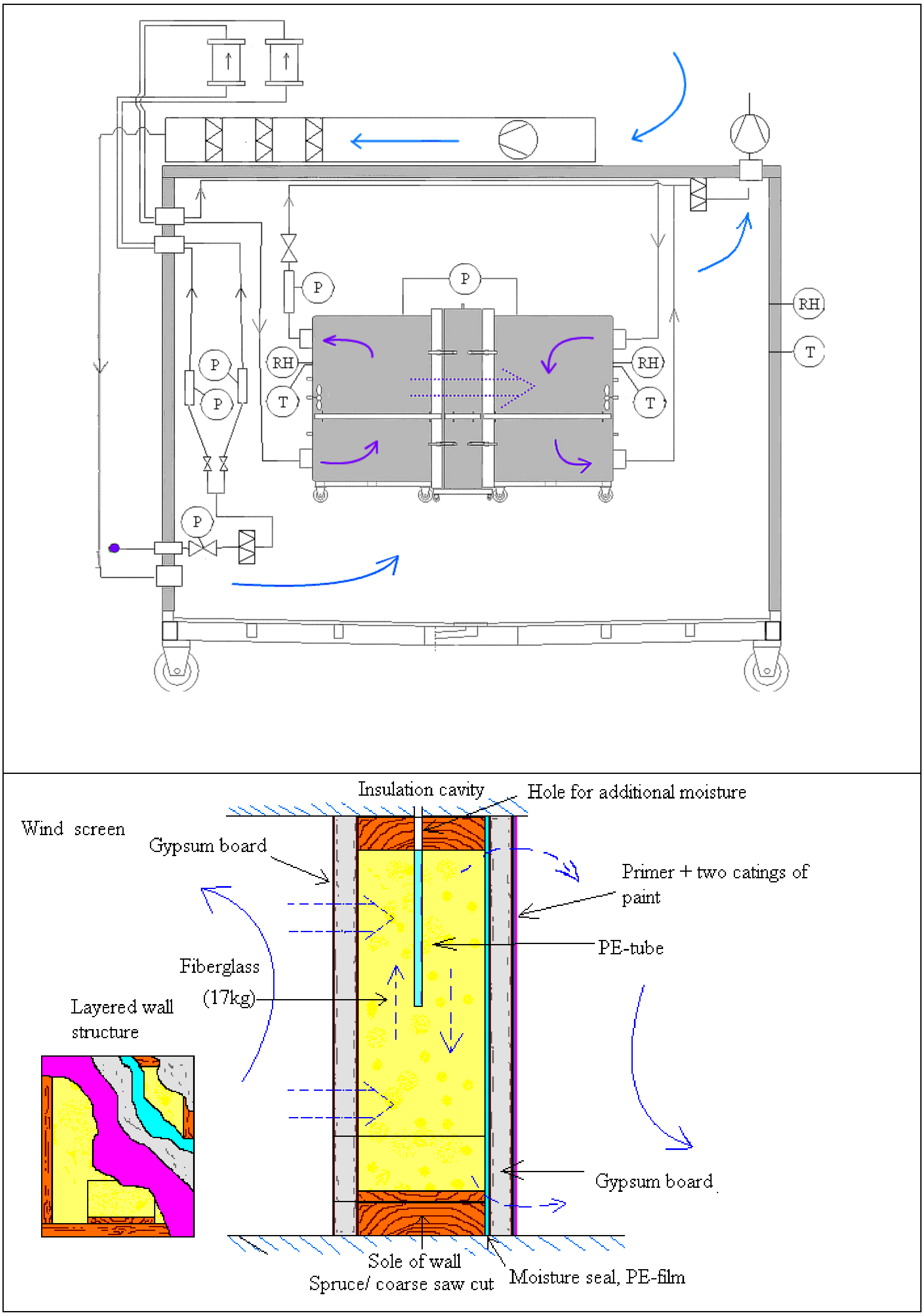

Figure 1 The principle of the IAQ simulator (top). The purple dot on the left of the protective room depicts the connection to the clean air source. An example of a structure of the dividing wall of the IAQ simulator (bottom).

DISCUSSION

IAQ Simulator offers a new testing tool for building product manufacturers to develop and test materials in multilayer structures. The developed simulator can be incorporated to the "Indoor Climate" classification and CEN when drafting emission measurement standards for structures. The simulator offers also a new tool to study the complicated causal connections of building structure, materials and unexpected exposure to water and microbes, especially during the use of the buildings.

Passive sampling methods proved to be more reliable for the follow-up in comparison to active sampling. The VOC profile for the microbes was found to be affected by the mould species and air supply. The repeatability of MVOC measurements was by most affected by the mold growth rate. The carbon dioxide concentration can not be used as a growth indicator when an air exchange is coupled to the test chamber.

The Ministry of Social Affairs and Health in Finland has given guidelines for the estimation of microbial risks in indoor environments by cultivation. Dry, “healthy” surfaces of building materials normally contain fungal spores <10 cfu/cm^2, when an area of 100 cm^2 is sampled with a cotton swab and the swab is analysed using the plate count technique. If the sampled surface of 100 cm^2 contains >1000 cfu/cm^2 fungi, or the amount of fungi is 100 times higher than that of the intact reference surface, there is evidence of abnormal fungal growth. The corresponding limits for abnormal fungal growth in building material samples and in air samples are 10 000 cfu/g and 500 cfu/m^3, respectively. The detection limits of the real-time PCR methods tested were under these risk limits, indicating that real-time PCR is a promising method for estimating the fungal risk in indoor environments. The advantage of PCR is that results could be obtained during one working day instead of 5-7 days required by cultivation methods.

ACKNOWLEDGEMENT

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Liabilities of Vented Crawl Spaces And Their Impacts on Indoor Air Quality in Southeastern U.S. Homes

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SUMMARY

This study documented that houses in the southeastern United States built on typical wall vented crawl spaces possess the following characteristics: 1) liquid water, water vapor and associated moisture issues, 2) mold spores, 3) measured holes between the crawl space and living space, 4) measured transmission of mold spores from the crawl space to the living space. When these characteristics exist together, the study indicates that contaminants (mold spores and moisture vapor) present in the crawl space are being transmitted through holes in the house floor and heating, ventilation and air conditioning (HVAC) system into the livable parts of the home, thereby exposing occupants to potentially harmful crawl space contaminants. These results confirm vented crawl spaces as important sources of mold species in the home environment. In order to reduce this exposure, closed crawl spaces in combination with thorough house and duct air sealing should be implemented. Such a system was found to be a robust intervention that reduced the moisture and indoor air quality problems associated with typical wall vented crawl spaces.

INTRODUCTION

Approximately 20% of new homes in the United States (200,000 per year) are built on vented crawl space foundations according to the U.S. National Association of Home Builders (NAHB). An estimated 26 million existing homes have vented crawl space foundations. Because adults and children today spend increasing amounts of time indoors, home environmental health is important for people's well being. Mounting evidence suggests that exposure to mold in damp buildings is an important risk factor for childhood asthma [1]. The strongest identifiable risk factor for the development of asthma appears to be exposure to environmental allergens, including indoor and outdoor pollutants [2]. Because vented crawl spaces in the mixed-humid climate of the southeast experience periodic high levels of moisture, they are very likely building areas where mold can be found. One remedy, in the form of properly closed crawl space standards, demonstrated houses will be notably drier, more energy efficient, and support less mold growth compared to houses built over vented crawl spaces [3]. However, it was previously unclear whether the presence of crawl space mold species resulted in exposure to occupants in houses. The overall purpose of this study was to evaluate the importance of typical wall vented crawl spaces as sources of mold species in the livable parts of the home environment. Forty-five homes in North Carolina were selected for mold species sampling and a building science evaluation to characterize the conditions of typical wall vented crawl spaces. This report will explore the influences of this foundation construction technique on mold species growth, indoor air quality, and house durability in houses located in the southeastern United States. From the results of these studies, we created a subsequent study of 36 new houses to validate the improved energy and moisture performance of the closed crawl space protocol established above compared to traditionally vented crawl spaces. Results from this study will be available in June of 2007.

METHODS

Mold species sampling

Prior to mold species sampling, the HVAC system was kept off for four hours. In each home, a minimum of two sets of samples were taken during the test using a Wilcoxon matched-pairs signed rank test. First, before the HVAC system fan was turned on, three samples were taken. One was taken near the return grill for the HVAC system, one in the crawl space and one outside the house. Then the system fan was turned on and allowed to run for at least five minutes before two additional samples were taken, one near the return grill and one at the closest supply air diffuser (or register) to the system fan. The supply diffuser sample air was isolated from the potential contaminant sources within the house, thus allowing characterization of the relative contribution of the HVAC system to the total bioburden within the house. The sampling was conducted by two trained indoor air quality technicians using Andersen two-stage cascade impactors, which collect and separate both non-respirable and respirable size particles. The sampler was connected to a vacuum pump calibrated to collect air samples at the rate of 0.5 liters per second. Equipment calibration was conducted at the beginning of sampling, at mid-day and at the end of the day. A sampling period of 3.5 minutes was used for the outdoor air sample and all samples collected within the houses. The sampling period for the crawl space samples was one minute. The collection medium used for impaction of mold spores was Malt Extract Agar, an aciduric mycologic medium designed for the collection of environmental fungi. After sampling, the culture plates were incubated at ambient temperature for 96 hours prior to enumeration and identification. Mold identification was accomplished by macroscopic examination of colony morphology and microscopic examination of fungal elements.

Characterization protocol

We collected data to better understand the thermal, moisture and air leakage data associated with each ventilated crawl space:

- We interviewed homeowners about how they operate house and crawl space and to determine any potential indoor air quality related health issues.
- We performed air leakage and zone pressure testing to quantify the"holes" between the house and outside, the crawl space and house, the HVAC system and crawl space.
- House characteristics such as house measurements, topography, HVAC and other equipment, and existing moisture control strategies.
- Crawl space characteristics such as evidence of past moisture problems (wood rot, condensation, mold growth, puddles on vapor barrier, etc.), wood moisture content to evaluate the potential of wood for supporting current mold growth, temperature

measurements of the ground, water pipes, ductwork, air handling cabinet and floor framing to assess surface condensation potential.
- Long-term temperature and relative humidity data for the central area of the crawl space compared to outside between July 2004 and August 2005.

Building pressure diagnostics

We measured detailed air leakage using a multi-pressure testing system connected to three different systems: 1) house envelope leakage measuring system, 2) crawl space to house leakage measuring system, 3) HVAC duct leakage measuring system. The testing order was as follows:

1) Baseline-HVAC system off, all windows and doors closed.
2) House leakage test only.
3) House, crawl space and duct leakage measuring systems run together.
4) House and duct leakage measuring systems run together.
5) Baseline - HVAC system on.

House description and moisture history

In order to document past or current moisture problems, we conducted a 100 point inspection which included house air temperature and relative humidity, crawl space air temperature and relative humidity, outside air temperature and relative humidity, crawl space surface temperatures, house framing wood moisture content, crawl space construction details, crawl space and exterior grading conditions and drainage systems. These data were collected using commercially available and calibrated wood moisture meters, spot radiometers, digital thermometers, and relative humidity meters.

RESULTS

The average number of vents per house was 13, with the maximum being 22 and the minimum being four vents. Sixty-seven percent of all vents were found open, 26% were partially open, and 7% were closed at the time of the data collection.

Moisture

In 33% of the homes, moisture was present on the ground vapor retarder, duct, and plumbing systems located in the crawl spaces. Seven percent of the homes had a leaking condensate drain for the HVAC system. Active plumbing leaks were found in 31% of the houses. Water was found inside 15% of the duct systems.

Although moisture was not always visible at the time of the testing, the presence of recent moisture accumulation in the crawl space was visible by the following means in Table 1.

Table 1. Moisture indications and percent of frequency found inside crawl spaces.

Drip line visible on ground	22%
Absence of ground vapor retarder	27
Absence of full coverage of ground vapor retarder	100
Discoloration on walls	49
Termite tunnels	4
Animals and insects	36
Dryer exhaust terminating in crawl space	16
Visible mold growth	62
Wood moisture readings at mold supporting levels (≥19 %)	67
Wood moisture meter readings at wood rot supporting levels (≥ 25 %)	36

Mold species

Mold species sampling provided an evaluation of the total number of breathable mold spores, reported in colony forming units per cubic meter of air and the most common species of mold growth found. Table 2 shows the summary of bioaerosol results (in colony forming units per cubic meter) by the possibility of transmission and sample location. Figure 1 displays the bioaerosol levels for houses with the possibility of transmission. Figure 2 illustrates bioaerosol levels for all houses by location.

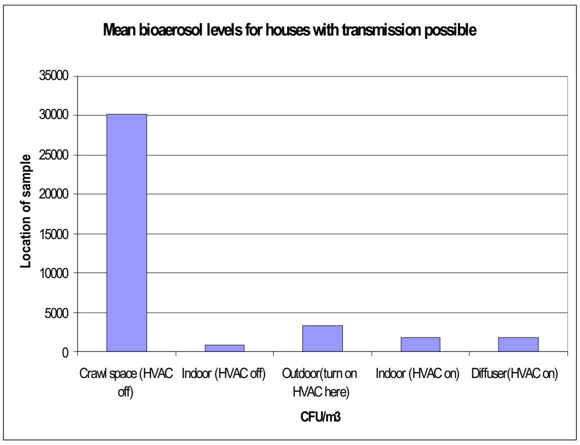

Figure 1. Mean bioaerosol levels for houses with transmission possible.

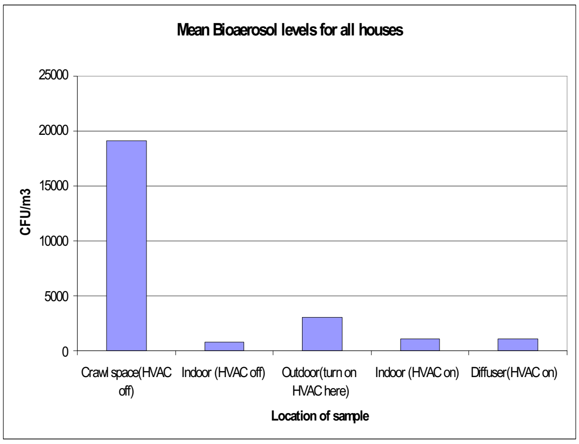

Figure 2. Mean bioaerosol levels for all houses.

Table 2. Summary bioaerosol results.

Sample	# Houses	Mean	Std. Dev.	Max	Min
Transmission possible	In CFU/m3				
Crawl space (HVAC off)	21	30163	16230	41146	1348
Indoor (HVAC off)	21	861	1233	5802	146
Outdoor(turn on HVAC here)	21	3235	3862	11756	349
Indoor (HVAC on)	21	1761	2425	11756	373
Diffuser(HVAC on)	21	1822	2607	11756	166
Transmission not detectable	In CFU/m3				
Crawl space(HVAC off)	10	161	508	1607	0
Indoor (HVAC off)	10	55	173	548	0
Outdoor(turn on HVAC here)	10	2033	2524	8418	40
Indoor (HVAC on)	10	176	556	1759	0
Diffuser(HVAC on)	10	1415	2282	11756	0
No transmission	In CFU/m3				
Crawl space(HVAC off)	14	16041	15144	41146	105
Indoor (HVAC off)	14	1323	3045	11756	71
Outdoor(turn on HVAC here)	14	3427	4630	11756	146
Indoor (HVAC on)	14	645	765	3219	124
Diffuser(HVAC on)	14	556	1101	4326	71
All homes	In CFU/m3				
Crawl space(HVAC off)	45	19102	18179	41146	0
Indoor (HVAC off)	45	825	1911	11756	0
Outdoor(turn on HVAC here)	45	3027	3836	11756	40
Indoor (HVAC on)	45	1061	1837	11756	0
Diffuser(HVAC on)	45	1062	2011	11756	0

Measured transmission

Initial assessment of transmission of crawl space air and its contaminants, including mold spores and moisture vapor, into the living space was determined to be present if two conditions held true. First, the concentration of the mold samples had to be higher in the living space once the HVAC system was turned on compared to the level of spores with the HVAC system off. Second, the mix and rank order of the indoor samples with the HVAC system running shifted to reflect the dominant mold species present in the crawl space sample and the rank order of species was different from the outdoor sample. If only one condition held, the house was classified as "transmission not detectable" and if neither condition held true, the house had no transmission.

Transmission of air and its contaminants was possible in 21 (47 %) of the houses characterized. In 10 (22 %) houses, transmission was not detectable or rather only one of the two conditions held true. No transmission was found in 14 (31 %) of the houses.

Measured holes between the crawl space and living space

Three leakage paths were measured: total house air leakage, air leakage between the living space and the crawl space, and air leakage between the HVAC duct system and the crawl space. See Table 3 for total house leakage testing documented.

Table 3. Measured house leakage.

CFM 50 per ft² of surface area *	M³/h/m² at 50 Pascals	Classification	Percent of houses tested
<0.25	<4.6	Minimal	0
0.26-0.45	4.7-8.2	Limited	24
0.46-0.60	8.3-10.9	Moderate	42
0.61-0.75	11-13.7	Excessive	20
>0.76	>13.8	Major	13
* Cubic Feet per minute at 50 Pascals			

The majority of the homes (69%) had 11% and 30% of the total house air leakage coming from the crawl space. The measured leakage between the HVAC duct system and the crawl space are shown in Table 4. Five homes were not classified for this test because they were unable to reach their target pressure.

Table 4. Classification of duct leakage.

CFM 25 per ft² of conditioned floor area as a percentage	M³/h/m² at 25 Pascals	Classification	Percent of houses tested
< 3%	<0.55	minimal	0
3.1-5%	0.56-0.91	limited	4
5.1-8%	0.92-1.46	moderate	9
8.1-12%	1.47-2.19	excessive	18
> 12%	>2.20	major	65

See Table 5 for the mean equivalent hole size for air leakage across the floor between the house and crawl space and house duct system and crawl space.

Table 5. Equivalent hole size by location.

Equivalent hole size in ft² (m²)	Mean	High	Low	NA*
House to crawl space	0.5 (0.046)	2 (0.19)	0.0	
Crawl space ducts	0.4 (0.04)	1.5 (0.14)	0.1 (0.01)	2
*NA indicates numerical data could not be calculated due to difficulty in reaching target pressure.				

DISCUSSION

This study—conducted during typical 12 month conditions—documents moisture characteristics of typical wall vented crawl spaces in the southeastern United States and measures the impact on living space mold sources. In some situations, the use of foundation vents to dry a crawl space may cause additional moisture. Of the houses in this study,

- 49% had moisture induced wall discoloration
- 62% had visible mold growth
- 67% had wood moisture meter readings at mold-supporting levels
- 36% had wood moisture meter readings at wood rot-supporting levels

Indoor air quality is compromised when moisture conditions exist in combination with air leakage between the house and crawl space and between the HVAC duct system and crawl space, as mold species can be delivered into the house through the air leaks. Therefore, both a moisture management strategy for the crawl space and an air sealing plan to reduce house and duct leakage should be incorporated into new and existing homes.

To demonstrate a protocol that will eliminate crawl space moisture problems and stop the total air leakage between the house and the crawl space, Advanced Energy also tested an intervention protocol on 12 similar sized homes in southeastern United States. This intervention study compared the standard vented crawl space design with a closed crawl space design. The closed crawl space design included a sealed ground vapor retarder that extended up the perimeter walls of the crawl space, air-sealed the perimeter wall between the crawl space and outside, air-sealed penetrations between the house and the crawl space, provided a source of conditioned air to the crawl space, and monitored the results [3]. The data from this study demonstrated that this closed crawl space protocol is a robust measure producing substantially drier crawl spaces (reducing conditions for mold, wood decay, and insects). These data also demonstrated reduced house space conditioning energy use by 15% to 18% annually as compared to the standard vented crawl space houses. Utilizing the results of these studies, we then created a study with 36 new houses to validate the improved energy, moisture, and indoor air quality performance of the closed crawl space protocol established above compared to traditionally vented crawl spaces. Results from this study will be available in June of 2007.

ACKNOWLEDGEMENT

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Questionnaire investigation of asthma/allergy symptoms among children and the indoor environment in homes in Sisimiut, Greenland

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SUMMARY

The aim of this project was to investigate the indoor environment and building characteristics in relation to with asthma and allergy among Greenlandic children. A cross-sectional questionnaire study involving 229 children in schools and institutions in Sisimiut was performed. The response rate was 23%. Of the children included,15% had doctor-diagnosed asthma. Rhinitis and eczema within the last 12 months was associated with; visible damp stains in the child's room; mouldy, earthy, unpleasant and tobacco smell and with suspected dampness in the construction.

Mould spots, suspected dampness in the construction, condensation on kitchen windows and stuffy air were observed significantly more often in houses constructed prior to 1971.

INTRODUCTION

Throughout the developed world the prevalence of asthma and allergies has increased over the last few decades [1]. No investigations have been made concerning the prevalence of asthma in Greenland. However, the reported frequency of hospital admissions due to asthma among Inuits in Greenland has previously been low [2]. The frequency of atopy in Greenland was analysed in 1987 and again in 1998 and an overall increase from 10% to 19% was found [2]. A main risk factor for developing asthma and allergy is genetic heredity. Due to the short time span of the increase, heredity cannot explain the increase during recent decades. Today children spend more than 90% of their time indoors [3] [4], and the indoor environment therefore has a strong influence on their health. Dampness and mould are indoor environmental factors of great importance for health [5] [6]. Today houses are built tighter, the result being a reduced ventilation rate accompanied by more dampness and mould problems indoors. In Sweden, an investigation has been made entitled Dampness in Buildings and Health (DBH), involving 14077 pre-school children. A similar study have been made in Bulgaria (ALLHOME), involving 4479 children [7]. In Sisimiut, an epidemiological cross-sectional questionnaire study including all children in schools and institutions was performed. The aim of this study was to investigate the association between indoor climate and building characteristics and asthma and allergy among Greenlandic children.

METHODS

A questionnaire including 81 questions divided into four parts, was sent to schools and institutions in Sisimiut for distribution to the children. The questionnaire used in the Swedish

DBH study was used as a template and adapted to Greenlandic conditions. The parts of the questionnaire analysed in this article concern the health of the child, and the housing conditions. The statistical programme SPSS was used. For evaluation of a significant association, the Pearson Chi-Square test was used. If frequencies below 5 were present, the Fisher's test of significance was applied. An association was considered to be significant if the significance level was $P < 0.05$.

Two diseases and four symptoms of asthma and allergy were used as evaluation criteria for associations between asthma/allergy and indoor environment. The two diseases were doctor-diagnosed asthma and rhinitis. The four symptoms were; wheezing, dry cough, eczema and non-doctor diagnosed rhinitis within the last 12 months.

RESULTS

In all, 1010 questionnaires were delivered and 229 valid questionnaires were returned. This results in a response rate of 23%. With this relatively low response rate no generalizations of the prevalence of symptoms in Greenland can be made. However the associations between asthma/allergic symptoms and indoor environmental factors are more valid.

Health of the child

In Figure 1 it is seen that eczema was the most frequently occurring symptom among children in Sisimiut.

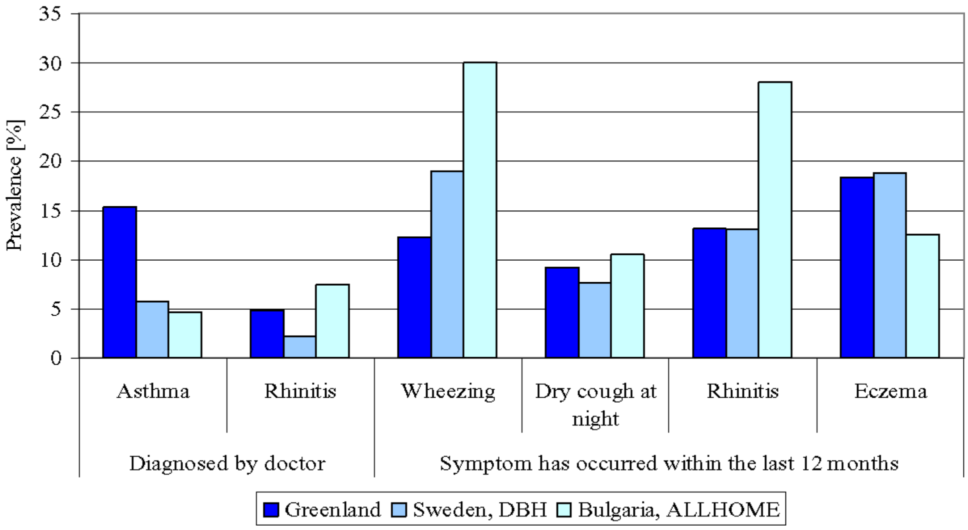

Figure 1: Occurrence of asthma/allergy among children in Greenland, Sweden and Bulgaria

Compared to the results from Sweden and Bulgaria, the prevalence of asthma is noticeably higher in Greenland and the prevalence of wheezing is noticeably lower.

Housing conditions of the child

One third of the children have lived in the same residence all their life; of the remaining children most have moved to the present residence within the last three years (2004-2006). Roughly half of the children lived in apartments followed by an almost equal amount of single and attached houses. The most common size of homes was 75-99m2. Most residences were erected during two distinctive year-intervals 1961-70 and 1984-1993. The majority of the families (75%) rented their residence. Exhaust air valves existed in most of the homes.

Only half of the homes had a cooker hood in the kitchen and one third had exhaust in the bathroom. The most frequent heating system was waterborne. The most common type of window was one with two-layer wooden frame. In most homes the flooring material was parquet and the surface material in the child's room was painted/unpainted wallpaper. 47 (21%) families reported that they had made considerable changes or extensions to their home during the child's life and in six cases that it was done due to problems with damp and mould in the residence. Visible damp stains were more often seen than visible mould. Visible mould and damp stains occurred considerably more often in bathrooms as opposed to the other rooms in the home. Drying of clothes took place mainly outside the residence but also inside in one third of the homes.

40 families (17%) suspected that they had humidity/mould problems within the floors, walls and ceiling, problems that were not visible from inside the residence.

Significantly more homes constructed before 1971 had visible mould spots inside, , especially apartments. The occurrence of condensation on the kitchen windows was slightly more frequent than condensation on the windows in the child's and parents' room. A significant association between condensation on windows in the kitchen and the year of construction was found. The possibility of condensation was increased when living in a building constructed before 1971. Stuffy bad air, tobacco smell, and dry air were the most common perceptions regarding the air quality. A significant association was found between condensation and dry air: the more condensation, the greater the number of complaints of dry air. A significant association was also found for stuffy air and houses constructed prior to 1971, as stuffy air was more often perceived in these homes.

Associations between symptoms and housing conditions

There were no significant associations between the type of residence and doctor-diagnosed asthma, wheezing, rhinitis and eczema within the last 12 months. A significant association was found between wheezing and houses built after 1993, as wheezing more often occurred here. No attempt to correlate doctor-diagnosed rhinitis and cough within the last 12 months and type of house and year of construction was made due to the low frequencies. There was a significant association between rhinitis, eczema, and visible damp stains in the child's room (see Table 1), as rhinitis and eczema occurred when visible damp stains were seen. Considering visible mould in all rooms (except bathrooms) the risk of asthma was increased. No significant association existed between visible mould and visible damp stains in the bathroom and doctor-diagnosed asthma, doctor-diagnosed rhinitis, wheezing, dry cough, rhinitis and eczema within the last 12 months.

Table 1: Associations between reported dampness in the child's room and asthma/allergy

		Visible mould			Visible damp stains			Suspicion of dampness in the construction		
		yes n=8	no n=203	P-value	yes n=11	no n=181	P-value	yes n=40	no n=131	P-value
Doctor-diagnosed	Asthma	2	27	0.337	1	27	0.500	8	20	0.491
	Rhinitis	0	9	0.680	0	9	0.579	2	7	0.647
Last 12 months	Wheezing	1	23	0.497	0	22	0.538	5	20	0.631
	Dry cough	2	18	0.191	0	18	0.326	7	10	0.071
	Rhinitis	3	24	0.057	4	20	0.035	11	12	0.002
	Eczema	0	37	0.148	0	38	0.046	6	24	0.696

A significant association between suspected dampness in the construction and rhinitis within the last 12 months existed see (Table 1), as rhinitis occurred when there was a suspicion of dampness. There were significant associations between unpleasant smell, mouldy smell, earthy smell, tobacco smell and rhinitis within the last 12 months (see Table 2). These smells were observed when rhinitis occurred. There were no associations between smoking in the home and doctor-diagnosed asthma, doctor-diagnosed rhinitis, wheezing, dry cough, rhinitis and eczema within the last 12 months.

Table 2: Associations between rhinitis within last 12 months and smells

Stuffy bad		
yes, n= 47	no, n=127	P-value
8	15	0.312
Unpleasant smell		
yes, n=14	no, n=147	P-value
4	14	*0.042*
Pungent smell		
yes, n= 7	no, n=147	P-value
2	14	0.107
Mouldy smell		
yes, n=19	no, n=143	P-value
9	13	*0.000*
Earthy smell		
yes, n=16	no, n=142	P-value
5	13	*0.005*
Tobacco smell		
yes, n=40	no, n=128	P-value
10	11	*0.005*
Dry air		
yes n=27	no, n=130	P-value
6	13	0.076

DISCUSSION

The occurrence of doctor-diagnosed asthma in Sisimiut was high, i.e. 15%. If this is true for the population as a whole, it would classify Greenland among countries having a high occurrence of asthma [1]. This would indicate that asthma has increased in Greenland when compared to previous studies of hospital admissions [2]. Comparing the study in Sisimiut to the Swedish DBH study [6] and the Bulgarian ALLHOME study [7], the occurrence of asthma is seemingly high. The occurrence of self-reported wheezing in Sisimiut was 11%, which is lower compared to the Swedish DBH-study and the Bulgarian ALLHOME-study. In another study in five cities (Adelaide, Sydney, West Sussex, Bochum, and Wellington) the prevalence of self-reported wheezing was between 20% and 40% [9]. The response rate to the questionnaire was 23% which means that tendencies can be indicated but generalizations cannot be made. Considering the low response rate, selection bias can be a problem. Families with symptomatic children could be more prone to participate, resulting in a higher prevalence of symptoms. Furthermore, the size of the data set is small and this makes the statistics more sensitive.

In cold climates, visible mould on indoor surfaces is not often found, except in bathrooms, as the humidity is low and the houses are well insulated [6]. The climate in Sisimiut is arctic and the observations of mould were generally low in rooms but considerably higher in bathrooms. These results comply with the climatic conditions. Visible mould was observed slightly less than visible damp stains in the homes. This accords with findings in the Swedish DBH-study [6]. Several people who reported observations of visible mould observed also visible damp stains. Rhinitis and eczema occurred when visible damp stains were observed in the child's room, and asthma occurred when visible mould in all rooms in the home (except bathrooms) were considered. This indicates that the indoor environment influences the health of the child.

Condensation on the kitchen window was more common in houses built before 1971. The same finding was reported in the Swedish DBH-study [6]. Condensation on the inside of windows is an indication of poor ventilation or poor glazing, and thus high indoor humidity. As the most common type of window was a two-layer glazing, it can be assumed that the type of window is not the primary reason for condensation. The association found may be related to the demands of ventilation strategies in newer houses as the buildings are tighter and ventilation strategies have to be implemented. A significant association between condensation and the perception of dry air was found. This finding has also been reported in the Swedish DBH-study and supports the suspicion that the sensation of dryness is linked to air pollutants, and thus to poor ventilation.

A significant association between the year of construction and wheezing was found. The risk of wheezing increased in houses built after 1993. In a study by Emenius et al. [8] it was found that the building age, or more likely, certain types of building construction, and indoor air humidity were associated with recurrent wheezing.

CONCLUSION

The occurrence of doctor-diagnosed asthma in Sisimiut was high. The likelihood of having had rhinitis within the last 12 months was increased: when visible damp stains were seen in the in child's room; when mouldy, earthy, unpleasant and tobacco smells were observed, and when the family had a suspicion of dampness in the construction. Significant associations of poor indoor environment were observed in houses constructed prior to 1971 as mould spots, suspicion of dampness in the construction, condensation on kitchen windows and stuffy air were observed.

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Thanks to Arne Villumsen for arranging the contact to the persons in Greenland and thanks to Kiril Naydenov for thoughtful comments on the project. Our greatest thanks go to the children and families in Sisimiut who participated in the studies.

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Indoor environment and building characteristics in homes of children with and without asthma and allergy in Sisimiut, Greenland

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SUMMARY

An epidemiological cross-sectional questionnaire study on housing and health, involving 229 children in Sisimiut, was initiated in May 2006. Based on this study, a nested case-control study was carried out in August 2006, covering inspections and physical measurements in the homes of 51 children. A tendency towards a lower average CO_2-concentration in the bedroom, at night, in the group of sick children (cases) compared to the group of healthy children (controls) was found. No significant associations or tendencies were seen between the two groups with regard to relative humidity and temperature at night. Condensation on windows in the kitchen was more common in homes built before 1971 than in newer houses. No significant association was found between condensation on the windows and asthmatic and allergic symptoms.

INTRODUCTION

Throughout the developed world the prevalence of asthma and allergies has increased over the last few decades [1]. In some countries the self-reported prevalence of asthma is above 25% [2]. No investigations have been made concerning the prevalence of asthma in Greenland. However, the frequency of atopy in Greenland was studied in 1987 and again in 1998. These studies showed an overall increase from 10% to 19% in one decade [3]. A main risk factor for developing asthma and allergy is genetic heredity. However, due to the short time span of the present dramatic rise in these illnesses, heredity alone cannot explain the increases; environmental factors must be the main cause [4] [5] [6] [7]. Studies have shown that children spend more than 90% of their time indoors [8] [9]. This means that the indoor environment in homes has a potentially strong influence on health. Studies show that dampness, mould and moisture are factors of importance for health [10] [6].

Due to increased focus on energy conservation, buildings are now more tight, with a resulting reduced ventilation rate. Bornehag and colleagues found that a low ventilation rate increased the risk of asthma and allergy in Swedish houses [10]. The same was indicated in studies by Öie et al.[11]and Emenius et al. [12].

The aim of this study was to investigate the indoor environment and building characteristics in regard to asthma and allergy in homes in the cold climate of Sisimiut.

METHOD

The inspection of homes was part of a larger investigation of Indoor Climate and Building Characteristics in Association with Asthma and Allergy in Sisimiut. The study was in two parts. The first phase, carried out in June and July 2006, was a cross-sectional questionnaire study covering all children in schools and day care institutions in Sisimiut [13]. Children were considered as "Cases" if they had: ever suffered from doctor-diagnosed asthma, doctor-diagnosed hay fever or allergic rhinitis. Children were also considered as "Cases" if they had had at least two of the following symptoms within the last 12 months:

1. An itchy rash, wheezing or whistling in the chest.
2. Dry cough at night, for more than two weeks, apart from having a cold.
3. Sneezing, or a runny or blocked nose, when the child did not have a cold or flu.
4. Sneezing, or a runny, or blocked nose, or itchy-watery eyes after having been in contact with furred animals or pollen.

Children defined as "Controls" had had none of the above problems. In total 86 children were grouped as "Controls", 56 children as "Cases" and 87 children could not be placed in either group.

The second phase reported here was a nested case control study, based on the responses from the first phase. The second phase was carried out in August 2006 and included 37 children who were randomly chosen for inspection in their homes. Some of the children visited were siblings and therefore information concerning some homes was given more than once in the inspections. In total, 34 dwellings spread over the whole Sisimiut village were inspected. Nineteen of the children were "Cases" and 18 were "Controls". The inspections of the children's homes were made by 4 inspectors in 2 groups. Each home was inspected for visible mould spots, odour, condensation on windows, building materials and type of dwelling. Type and location of the ventilation system as well as the heating system were also registered. A scaled floor plan of each dwelling was made and the volume of the child's bedroom was calculated. In the child's bedroom an indoor air quality monitor (IAQM), model PS31 by Sensotron, was placed measuring the CO_2 -concentration, relative humidity (RH) and temperature for at least 24 hours. The IAQM was calibrated prior to the measurements and each day the outdoor CO_2-concentration was measured. The body weight and location of people and pets sleeping in the dwelling, as well as the position of doors and windows were self-reported by the inspected families. The CO_2 -concentration can be used as an indicator of the air change rate (ACH), assuming that a high level of CO_2 involves a low ACH [14]. It was assumed that the air in the bedrooms was well mixed. Continuous measurements were analysed in the period from 23:00 to 05:00, during which time the RH, temperature and CO_2-concentration were measured at intervals of 1 minute and the average, minimum, maximum and standard deviations were calculated.

All data were entered into the statistical programme SPSS 11.5.1 for Windows. For evaluation of significant associations the Pearson chi square test was used. The Mann-Whitney U-test and Student T-test were used to evaluate the significance in the continuous variables of CO2, RH and temperature measurements. The Student T-test was used when the measurements were normally distributed, otherwise the Mann-Whitney U-test was used. An association was considered to be significant with the significance level, $P<0.05$.

RESULTS

The majority of children in the Case group lived in apartments (11 out of 16) whereas the majority of the Control group lived in attached houses (9 out of 12). Almost all houses in Sisimiut had sloped roofing with asphalt paper. The façades were almost exclusively made of wood, and the majority of the inspected houses had double-glazed windows. Examples of typical types of dwelling are shown in Figure 1.

(a) (b) (c)

Figure1. Types of dwelling in Sisimiut. Single family house (a), attached houses (b) and apartments (c).

The most common ventilation was natural ventilation (by opening windows and/or by use of ventilation slots in the outer walls) and a kitchen fan. Only 3 out of 37 homes had mechanical exhaust ventilation. During the inspections it was seen that some ventilation slots in the outer walls were closed by tape or cloths. For a description of the dwellings, see Table 2.

The child's bedroom

No statistically significant differences were found between Cases and Controls as regards nightly CO_2 -concentrations, temperatures and RH, although there was a tendency towards a slightly higher average CO_2 -concentration measured for the Controls. The RH and temperature did not deviate much between the Case and Control group. (Data for measurements from 23:00 – 05:00 and corresponding P-values are showed in Table 1. The values presented are the average, minimum and maximum of the average).

Table 1. Average concentration at night (23:00-05:00).

	Case	**Control**	**P-value**
CO_2 -concentration [ppm] (min-max)	1249 (934-1503)	1648 (1103-2056)	0.466
Relative humidity [%]	47.5 (44.2-49.9)	46.0 (43.4-47.9)	0.494
Tempertature [°C]	24.0 (23.5-24.3)	23.9 (23.4-24.3)	0.948

In 25 homes the child's bed was placed against the wall to the ground or to the outer wall. No condensation was seen on the windows in the child's room during the inspections. However, 8 dwellings (5 Cases) had a small amount of cracked paint on the window frame and 3 (1 Case) had a lot of cracked paint on the window frame. No moisture problems, such as moisture- or mould-spots or detached paint/wallpaper were observed behind furniture in contact with either the inner or outer walls. However, 1 case had indications of active mould spots. A sensation of stuffy odour in the child's room was registered in 4 homes (all in the Case group). Further registrations from the inspections are shown in Table 2.

In all children's bedrooms the ceiling was painted. Only 2 children (both "Cases") had PVC as flooring material in the bedroom. The most common flooring material was varnished parquet. The frequency of electronics in the children's rooms was evenly distributed between the groups. Fourteen children had a TV in their room, 11 had a stereo and 5 had a PC.

None of the dwellings had an obvious perception of tobacco smell during the inspections. One (Case) had an obvious chemical smell. Ten (6 Cases) smelled stuffy and 5 (all Cases) had an obvious stuffy, earthy or microbial smell.

Table 2. Type of dwelling in groups

Type of resident	**Case**	**Control**	**Total**
Single family house	5	4	9
Attached house	3	9	12
Apartment	11	5	16
Façade material			
Wood	14	17	31
Asbestos (Ethanite)	2	1	3
Ventilation			
Natural ventilation without kitchen fan	2	3	5
Natural ventilation with kitchen fan	16	13	29
Exhaust ventilation	1	2	3
Floor material in child's room			
PVC	2	0	2
Parquet (varnished)	14	16	30
Carpet (wall to wall)	1	1	2
Linoleum, cork or tiles	2	1	3
Air supply in child's room			
No special device for air supply	1	3	4
Slot in the outer wall	18	15	33
Windows in child's room			
Windows single glazed	1	2	3
Windows double glazed	16	14	30
Windows triple glazed	2	2	4
Weight of child			
0-12.5 kg	3	5	8
12.5-25 kg	6	2	8
25-37.5 kg	4	7	11
37.5-50.0 kg	5	4	9
Stuffy odour in child's bedroom			
No signs of odour or damage	9	11	20
Possible odour	6	7	13
Possible odour, certain	4	0	4
Mouldy odour in child's bedroom			
No signs of odour or damage	11	17	28
Possible odour	8	1	9
Moisture spots in child's bedroom			
No signs of odour or damage	15	12	27
Possible damage	3	2	5
Local indication, inactive, not significant	0	4	4
More indications, possible active	1	0	1

DISCUSSION

The investigation of the indoor environment and building characteristics in association with asthma and allergy in Sisimiut, covered a very small sample size. The findings of the study should therefore be seen only as indications or tendencies in the studied field.

All investigations where carried out in August which is the warmest month in Sisimiut. The measurements presented in this paper are not representative for the whole year. The change in outdoor conditions affects mainly the RH and CO_2 -concentration indoors as people do not ventilate as much, i.e. opening windows, when it is cold outside.

As humans produce CO_2, the CO_2-concentration will build up in an occupied room. When the room is ventilated the concentration will decrease. Assuming the air in the room is well mixed, the level of CO_2 can be used as an indicator of the ACH. The findings in this study show no statistically significant difference in the CO_2 -concentration between the Case and Control groups. The same was found in a study in Stockholm [12]. The opposite has been found in a study that indicated an increase in the prevalence of asthma and allergy for low ventilation rates [4].

The fact that the case group had a lower average CO_2 -concentration (1249 ppm) than the control group (1648 ppm) was also found in the Bulgarian ALLHOME-study [5]. This can imply that the case group ventilates more due to the awareness of indoor air quality and health. Another reason could be that parents with asthmatic or allergic children may tend to leave the door to the child’s bedroom open during the night because they want to listen to the breathing of the child if any complications should occur.

In cold climates, visible mould on indoor surfaces is not often found (except in bathrooms), as the humidity is low and the houses are well insulated [6].The climate in Sisimiut is arctic and the observations of mould were generally low in rooms but considerably higher in bathrooms. These results accord with the climatic conditions. Visible mould was observed slightly less than visible damp stains in the homes. This accords with findings in the Swedish DBH-study [6]. Measurements show no significant difference in the average temperature, RH and CO_2-concentration within the case and control groups.

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Horizontal Measurements of SO_2, NO_2 and SPM in Indoor Air and relationship to Outdoor concentration

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SUMMARY

Measurements of SO_2, NO_2, and Suspended Particulate Matter (SPM) of indoor air and correlation with outdoor air have been determined in an office of multistory mechanically ventilated building. The objective of the study was to evaluate the diversity of SO_2, NO_2 and SPM inside building and penetration of pollutants, mostly found outside, from outside to inside in an airtight building. Ground floor of the building was selected for study. Horizontal measurement of SO_2, NO_2 and SPM was done inside the building at four sampling stations. Simultaneously outdoor measurement too was carried out at two sampling stations to analyze the ratio of indoor to outdoor pollution. The results of measurement study illustrate the penetration of outside pollutants to inside the building. Activities of building users, occupants and visitors, play vital role manipulating the concentration of pollutants inside. No predominant sources of these pollutants were found inside.

INTRODUCTION

Indoor concentration measurements are required for determining the total personal exposure. It is used for providing information about building penetration factors, mainly through the coupling of simultaneously measured indoor and outdoor concentrations. Indoor microenvironments contain in general lower concentrations of outdoor pollutants such as sulphur dioxide (SO_2). However the presence of indoor sources of pollutants such as Nitrogen dioxide (NO_2), and Suspended Particulate Matter (SPM) can lead to increased indoor concentration of these (Moschandreas 1985). If there is a high ventilation rate, than the accumulation of indoor pollutants is reduced, but at the same time higher penetration of outdoor pollutants is expected within the indoor microenvironment (Hoppe and Martinac, 1998). Jetkins *et al* (1992) explained that urban people spend on average 87% times indoors and only a mere 6% outdoors. Due to the substantial differences in climatic conditions between two settings, bring enormously unusual pollution dispersion. The ratio between indoor and outdoor pollutant concentration (I/O) gives a method to analyze the origin and transport of indoor pollutant. Yocom (1982) pointed out that indoor-outdoor relationships are complex interactions of various factors like meteorological factors, indoor sources and sinks, pollutant depletions, filtrations and ventilations etc. Moreover, air pollutants indoors and outdoors differ in type, characteristics, concentrations and sources. It is therefore important to determine how outdoor air quality can affect indoor air quality.

A number of studies have been attempted to characterize the measurement of indoor air pollutants and its relationship with outdoor concentration. Most of the studies ended that level of indoor pollutants also depends on the outdoor scenario (Jones et al 2000, Kingham et al 2000, Molhave et al 2000). Koistinen et al (2004) concluded that residential and work environment contributes to indoor total particulate matter. In a measurement study of fine and coarse Michael *et al* (2002) pointed out the outdoor air as a major source of indoor coarse particles**.** Bae *et al* (2004) reported that Respirable Suspended Particles and NO_2 were predominately of outdoor sources. Riain *et al* (2003) executed a study on IO ratio in naturally ventilated building. Blondeau *et al* (2005) found IO ratio vary from 0.03 to 1.79 in school buildings.

The ratio between indoor and outdoor concentrations of SO_2, NO_2 and SPM provides information of source contribution to indoor pollutants whether pollutants are inside generated or penetrate from outside. In present study horizontal distribution of SO_2, NO_2 and SPM inside the office and penetration of outdoor pollution across the building envelope was discussed by measuring and analyzing IO ratio, when building was occupied and compare with the non-occupied moment. The objective of the study was to evaluate the infiltration of pollutants, mostly found outside, from outside to inside in an airtight building and diversity of SO_2, NO_2 and SPM inside building.

METHODS

Measurements were made for SPM, SO_2, and NO_2 for five months, with one-week frequency, from May to August and October at Capital Court Building of Delhi metropolitan area located on main road. The road in consideration is main road that links between southern part and western part of Delhi city. About 500-800 vehicles per hour with the average speed of 40 km h^1 run daily. Building is a multistory premises operated by mechanical ventilation system. The air conditioner maintains and controls overall indoors climate. The doorway of the office, which leads to the main entrance of the building, is connected directly to the main road. All windows of the office remained closed normally during the entire sampling period, excluding main entrance and another gate, right side to the main entrance. The opening of the main gate was>1m wide and opening of the right gate were <1 meter. Frequency to opening the main entrance was very numerous and opening of the right gate was occasionally only for building inhabitant and employees. The activities inside are normal office activities and computer work. Throughout the period of investigation, inside activities were recorded these are room volume, no of work stations, smoking activities, occupancy level during peak–non peak hours beside these prolonged entry of visitors, cleaning etc. Anomalous activity was also documented.

Site map

Office selected for study is comprised of Office, Meeting hall, Lobby, Cafeteria. Four sampling stations were installed inside office premises for horizontal measurement of SO_2, NO_2 and SPM. Two sampling stations were established outside the building at 20 feet height from the ground floor, to work out the indoor-outdoor ratio.

Instruments

Four indigenous handy samplers (APM 820 Envirotech Instruments Pvt. Delhi, India) were used to measure the SO_2, NO_2 and SPM concentration at four sampling sites. Eight hours air sampling (four hours average) was carried out. Eight hours continuous monitoring was also done outside using High Volume sampler (APM 460 Envirotech Instruments Pvt. Delhi, India). Particulate matter concentration was based on the weight accumulated on the filter papers during the 8-hour exposure period and concentration determined by gravimetric analysis. SO_2 and NO_2 analyzed by colorimetric method.

RESULTS

Sampling was carried with one-week frequency including weekend to evaluate the interruption of building occupants on indoor pollution. Total sixty samples of each from outside and 120 samples of each of inside air were collected. The IO ratio, in the simplest urging, depends ultimately on several parameters: outdoors pollution level C_o indoor pollution source S_{source}, source ventilation rate $Q_{source,}$ indoor pollutant sinks S_{sink} sink ventilation rate Q_{sink} as indicated by the equation

$$C_i/C_o=1+1/C_o\ (S_{source}/\ Q_{source} - S_{sink}/\ Q_{sink}) \qquad (1)$$

Where C_i is the indoor concentration and the fraction on the left of eq. (1) is essentially the IO ratio. Equation facilitates to explain database (Andy 2000). Figure-1 shows the horizontal distribution of SO_2 inside the building. It was vary from $\leq$3 to $\geq$17 μg/m^3 with maximum concentration was 16.8 μg/m^3 recorded in month of October. Repeated measure ANOVA revealed significant mean differences ($P<0.0001$) and difference at various sites too significant ($P<0.05$). Maximum elevation of SO_2 concentration was observed in October, while outdoor temperature goes down during this period. Cumulative SO_2 concentration was at site-4 *i.e.* 8 to 11 μg/m^3 in October. Figure-2 shows the indoor to outdoor ratio at different sites. There is no significant difference between indoor-to-outdoor ratios for SO_2 horizontally. Average IO ratio at various sites is 0.4472, slightly higher at site 4. Figure 3 illustrates the daily IO ratio in various months. Comparatively higher IO ratio for SO_2 (0.5015) in October and low weekend IO ratio is indicative of enhanced infiltration of outdoor SO_2, which contribute indoor source of SO_2 as well as decreased outside temperature.

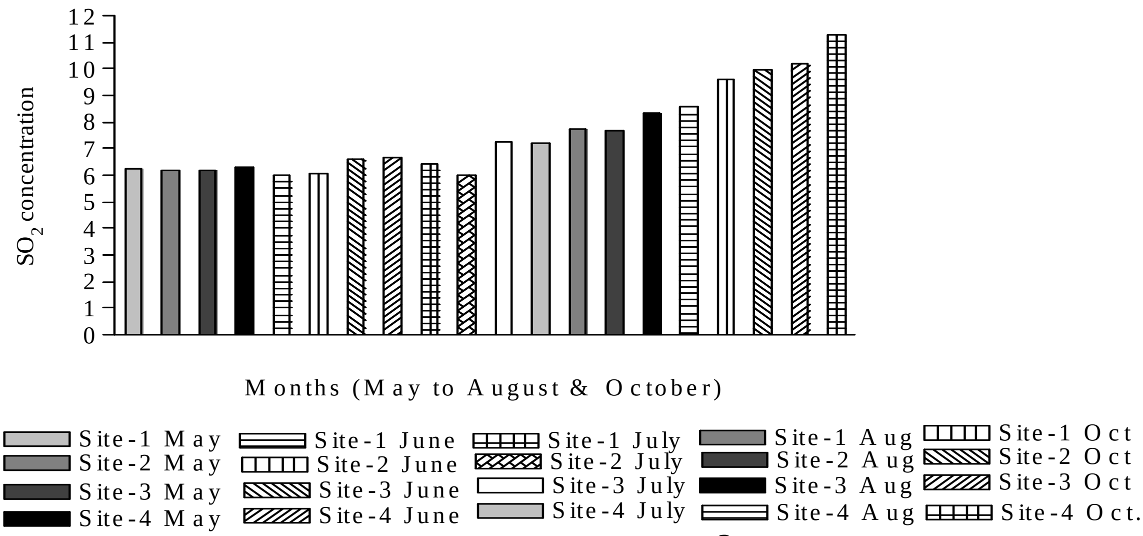

Figure-1:Horizontal distribution of SO_2 (μg/m^3).

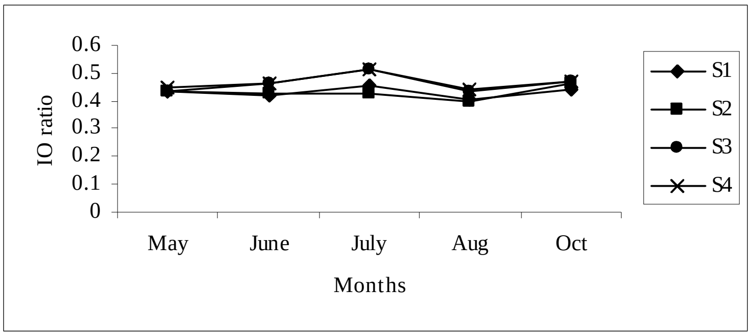

Figure 2: IO ratio for SO_2 at different sites

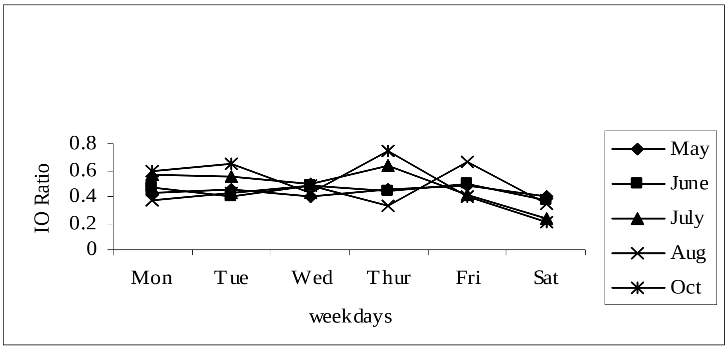

Figure 3: Daily IO SO_2 ratio in various months.

Figure-2 illustrates the horizontal variation of NO_2 across the building. It was fluctuated between 4 to >28 µg/m^3. Highest 28.8 µg/m^3 concentration was observed in month of October. It is quite higher corresponding to inside SO_2 concentration. Likewise SO_2, similar distribution pattern was followed by NO_2. Repeated measure ANOVA revealed significant mean differences ($P<0.0001$) and significant difference at various sites ($P<0.05$). Figure 5 shows the IO ratio for NO_2 at various sites (horizontally) and average peak IO ratio for NO_2 (0.467) was at site 4, which alike to SO_2. Figure 6 shows the daily IO ratio for NO_2. Relative higher IO ratio for NO_2 was observed in month of October. Peak IO ratio for NO_2 was 0.8810, which is very close to 1. Significant correlation was established between indoor and outdoor pollution levels of NO_2. It is indicative of increased net inflow from outside (S_{Source}).

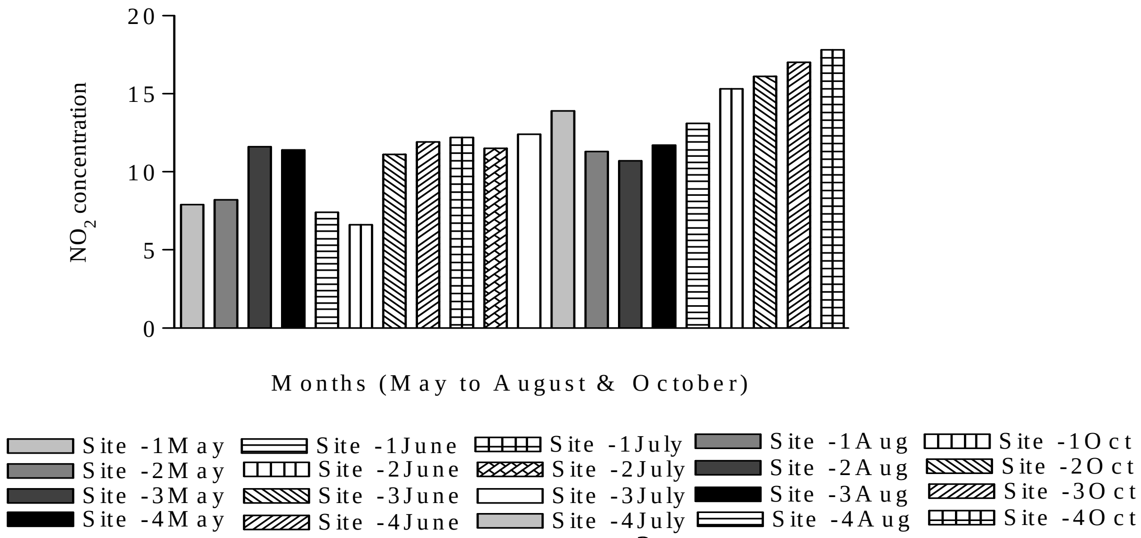

Figure-4: Horizontal distribution of NO_2 (µg/m^3)

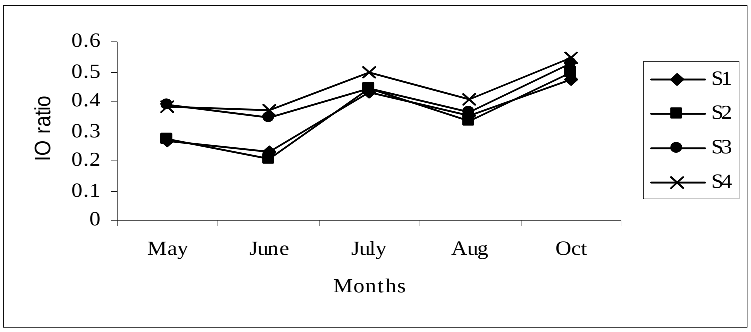

Figure 5: IO ratio for NO_2 at different sites.

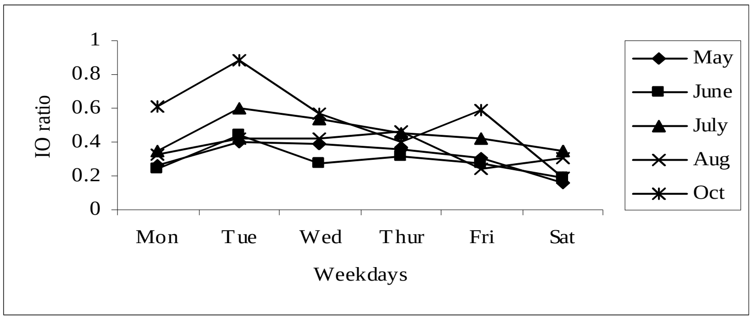

Figure 6: Daily IO ratio for NO_2 in various month

Particulate matter in the outdoor atmosphere is expected to penetrate interior spaces. It may be able to produce respiratory disease within the occupants. Figure-7 shows the horizontal distribution of suspended particulate matter (SPM) inside the building. Highest measured concentration was 70.5µg/m^3 in June. Some construction work was observed there during sampling in May-June. Elevated SPM concentration may be attributed to that only (increased of S_{source}). Lowest concentration was 20µg/m^3 in month of August it may be attributed to reduced particle concentration after rainy seasons. Likewise SO_2 and NO_2 repeated measure ANOVA revealed significant mean differences ($P<0.0001$) as well as significant difference in distribution pattern of SPM at various sites ($P<0.05$). Indoors horizontal SPM concentrations did not follow the pattern of SO_2 and NO_2. Figure-8 illustrates the IO ratio for SPM at various sites. Unlike SO_2 and NO_2 SPM ratio was relatively small. Average ratio for SPM was 0.1694 very low. It also indicates that SPM concentration was not greatly depended on outdoors pollution level (C_o). Figure-9 illustrates the slight elevated weekend ratio for SPM in the month of October than weekdays. It is also attributed by interruption of human being during weekend of October.

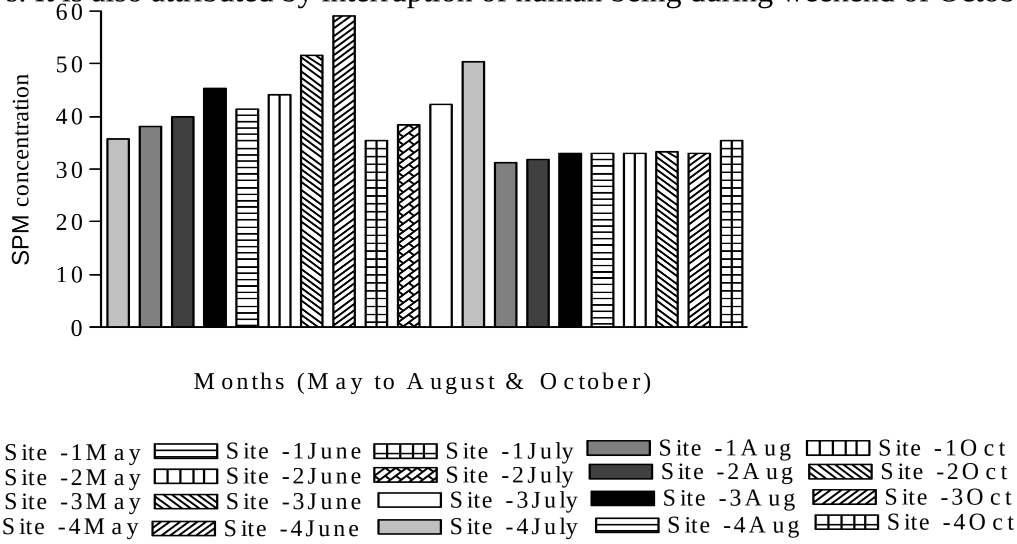

Figure 7: Horizontal distribution of SPM (µg/m^3)

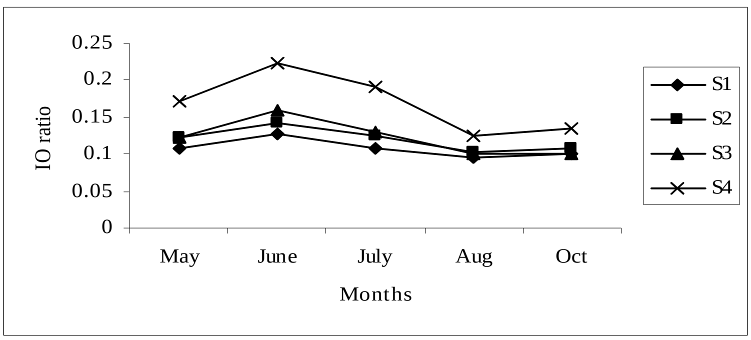

Figure 8: IO ratio for SPM at different sites.

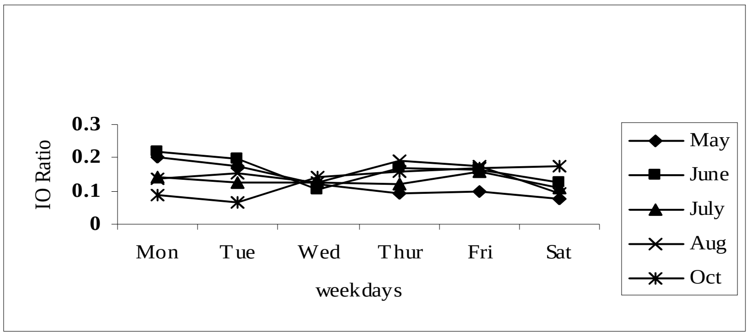

Figure 9: Daily IO SPM ratio

Results reveal that indoors SO_2 and NO_2 concentration were somewhat elevated in month of October at the time where ambient temperature gradually goes down. It depicts that temperature may have substantial role in increasing of net inflow of two gaseous from outside to inside. Simultaneously lower weekend concentration is indicator of confined interference of human activities. In general the pollutant concentration was lower inside than outside throughout the sampling period. The variation of SO_2 inside and outside was also smaller than SPM and $NO_{2.}$ Koponen *et al* (2001), Andy (2002) reported that variations of indoor pollutants were smaller and more gradual than that outdoors and also proved environmental factors to be strong manipulator of indoor-to-outdoor ratio. Peak IO ratio for all pollutants were observed during weekdays pinpointing the combined effect of increased occupants activities inside as well as heavy movement of visitors that leads to frequent opening of doors. Jhones *et al* (2000), Chan (2002) also observed that human activities and outside traffic manipulate the IO ratio. The IO ratio was higher at site four. It may be attributed to the two reasons, high occupancy level at this site and an addition door, which was facing the roadside.

DISCUSSION

Measurement of concentration of SO_2, NO_2 and SPM was done in an office during five-month period to study the horizontal flow of pollutants across the building and IO ratio with respect to outdoors concentration. Horizontal distribution pattern of measured pollutants varies across the building. IO ratio for SO_2, NO_2 and SPM were below one across the building demonstrating the influence of outdoor contaminants to indoor air quality. No predominant source was found inside the building. I/O ratio was somewhat lesser for SPM as compared to SO_2 and $NO_{2.}$ Low IO ratio for SPM may be due to the low penetration rate for SPM. Site four was identified as the really precious zone among all four-monitored points. This is characteristic of the exposure to exterior contaminants, occupant’s activities, and interference by operating the access from two sides. Prolonged entry of visitors and occupants’ activities during working days enhance the penetration of outside contaminants, which was restricted in weekends.

Horizontal measurement of pollutants encourages to examine dispersion of indoor pollutants at different positions in order to draw meaningful comparison at various spots throughout the building envelope and assessment of personal exposure to air pollutants.

ACKNOWLEDGEMENT

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Why Positive Olfactory Perception Is An Issue For Evaluation Of Buildings

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SUMMARY

The real estate market is changing rapidly and with it the evaluation of buildings. Requirements are being changed in order to meet market demand. Besides the location, infrastructure and the actual condition of a building's technical equipment a further parameter called the "Soft Factor" is required in the evaluation of a building's quality. High end equipped spaces, according to statistics, are more easily let and are therefore the key assets in higher cash flow portfolios. A factor that can significantly decrease the value of a building is low air quality. The quality of air, however, is not determined by standards, it is determined by perception. Although thermal comfort plays an important role, olfactory comfort is critical to this perception of air quality. This paper will show why the real estate market is adapting to this new paradigm. It will demonstrate why olfaction and the perception of "natural fresh" air play such an important role and why it is has become a key issue for the HVAC community.

INTRODUCTION

The ability to assess the Fair Market Value of a building or a portfolio of real estate assets has become particularly important since private equity and real estate funds have become major players in the European real estate market. They have introduced to Germany the internationally accepted standards for the evaluation of buildings. Other continental European countries have seen similar changes in evaluation parameters. For many years, the physical value of a building has played a major role in the evaluation process of a real estate asset. Today, the evaluation of an investment decision is focused above all on the evaluation of the discounted cash flow which is distinct from the physical or land value of the asset. The discounted net annual rent is the basis for the evaluation. The investor will typically choose an investment time horizon with the result that even the "life expectancy of the building" has little relevance to the calculation of value.

In this context the outfitting of a building can be of importance since the discounted cash flow calculation not only includes the actual rent but also an assessment of the sustainable market rent. The value of buildings with current or future vacancies therefore benefits from outfitting that create an additional value for potential users and make them easier to let. Outfitting features that are of little benefit to potential tenants (such as golden bathroom fixtures in an office building) no longer play any role in the valuation of a property. In other words, the substance of a building or the outfitting are incorporated in the cash flow calculation only if they increase the lease yield.

Employees are the largest expense of most companies and therefore these human resource issues are very important when analyzing companies. Nowadays, banks go as far as to look at these aspects in order to reveal the so called "hidden costs". These can have a significant impact on the banks' assessment of the creditworthiness or the interest rates they charge. Similarly, Chen [1] found that there is a strong awareness and growing concern over the "silent crises" of IAQ and its potential to cause large industry losses. These losses included both direct costs to insurers from paying health insurance and professional liability claims as well as the cost of litigation.

ENVIRONMENT

Increasingly, the business world is beginning to address the "soft factors" that have not traditionally been included in business economics or management science. More and more attention is being given to these soft factors - satisfaction vs. profitability, quality vs. productivity, qualitative advantages vs. financial ones, ethics vs. efficiency. Hidden costs with significant potential financial impact arise when these factors are not adequately measured or addressed. The impact of the environment is one of those soft factors.[2].

The impact of environment, air and its olfactory content has been known to science for at least 20 years. Frequently, the market fails to act on scientific evidence, sometimes as in the case of climate change perhaps "shutting the stable doors once the horse has already bolted". The recognition and acceptance of human resource issues and the associated costs has also lagged science. The HVAC community has clearly missed the opportunity to define air quality and its hedonic value [3] with respect to well-being in response to neurophysiological and psychological science. This shows quite clearly the impact of olfaction on humans since air quality encompassing olfaction plays a major role in well-being. Improving the well-being of room occupants will result in companies enhancing their performance. There are several studies that demonstrate the link between a qualitative working environment and the productivity of a company. [4]

STRESS

According to a European survey nearly one in three European workers is affected by work-related stress. In the European Union, over the last decade, work related stress has been consistently identified as one of the major workplace concerns – a challenge not only to the health of working people but also the healthiness of their organization. Studies suggest that between 50% and 60 % of all lost working days are related to stress. People are experiencing stress when they feel an imbalance between the demands on them and the personal and environmental resources that they have to cope with those demands. [5] One of the main factors of managing stress is promoting health. The perceived environment is one of the main factors which can enhance stress or reduce it. An improvement in workplace health can be a key ingredient of business efficiency and competitiveness.

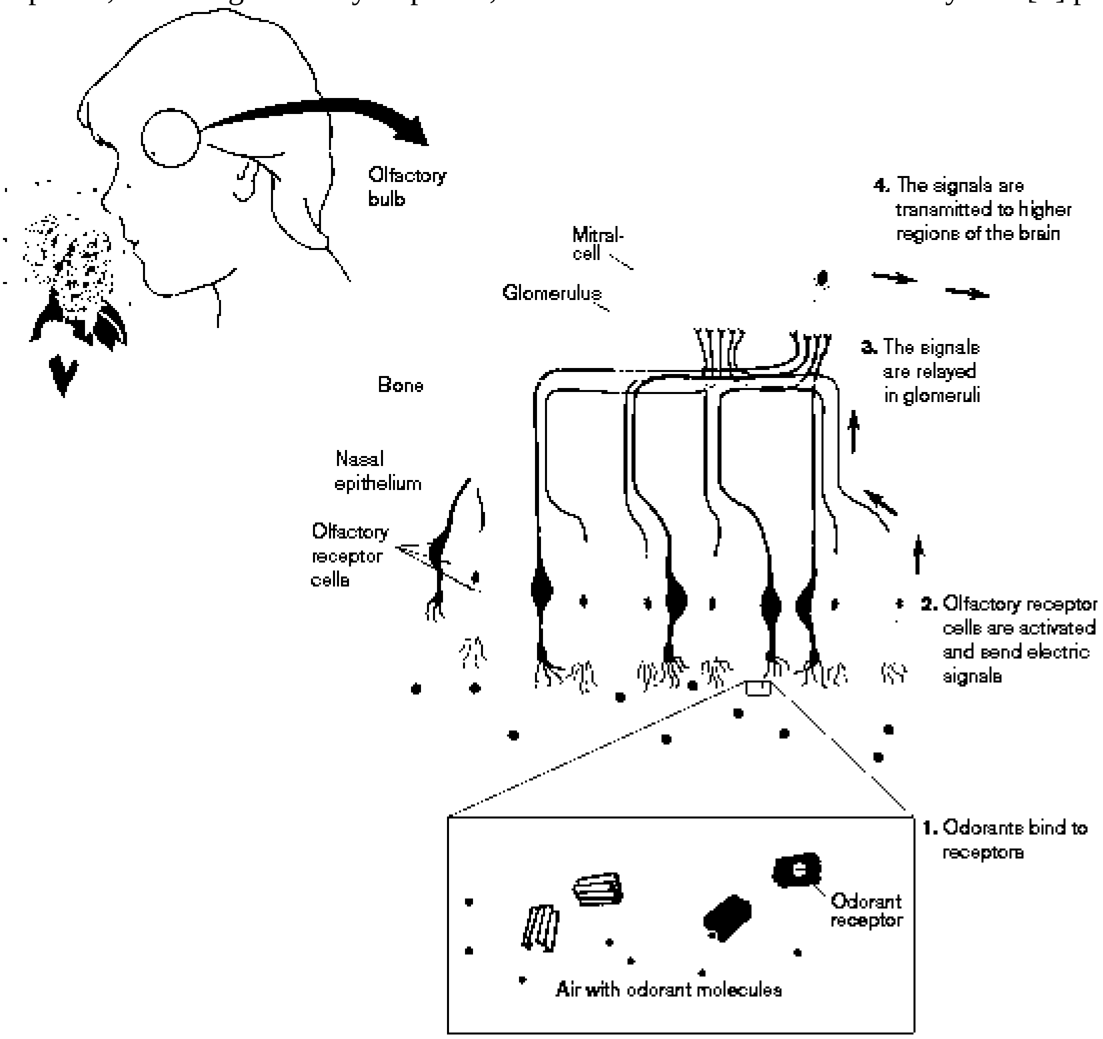
OLFACTORY COMFORT

Neurophysiological and psychological research in the field of chemical senses has shown that our well-being is controlled to a significant extent by the olfactory substances found in the indoor air. [6] Olfactory comfort is not only defined by the absence of negative olfactory substances but also as a condition of mind that expresses satisfaction with the olfactory environment created by positive stimulating substances in the indoor air. Buildings situated in metropolitan areas typically lack these positive olfactory substances. The substances are generally not found in sufficient quantities in the outside air and filters and cleaning mechanisms within HVAC systems remove them along with any negative substances. Therefore positive olfactory substances have to be restored in the indoor air. [7]

OLFACTORY SYSTEM

With every sniff, olfactory molecules pass through the nostrils to the olfactory epithelium. The millions of sensory (receptor) cells in the epithelium are connected via the sensory nerves and the olfactory bulb to the hypothalamus, other limbic system structures, the cortex and other sites which receive this information to coordinate and manage our abilities to learn, think, remember, respond and contemplate.

The limbic system is responsible for a person's range of feelings, his emotions, memories and the so-called effective tone of their entire behavior. A person cannot, therefore, block out any impulses, including olfactory impulses, that are transmitted to the limbic system.[8] picture 1



To date, findings from cerebral imaging have shown that olfactory function involves a complex and extensive neural network. Odor processing appears to be based on two main modes of processing. One is a serial processing with successive involvement of the primary and secondary olfaction areas, and the second is parallel processing which processes odor in the right and left hemisphere separately depending on the nature of the cognitive task. While areas located in the right hemisphere such as the orbital frontal cortex (OFC) and piriform cortex (PC) are more involved in memory and familiarity, those located in the left hemisphere, such as the OFC, insula, PC, amygdala, temporal lobe, and superior frontal cortex, participate more in the emotional response to odors. The piriform-amygdala region appears to be associated with the evaluation of emotional intensity and it is therefore activated more by unpleasant odors than by more mild odors. The role of the superior frontal cortex is to process the odors and their effect on emotional state and use this information to make personally relevant decisions. [9]

A number of major investigations in recent years have dealt with the effect of odor on electrical brain activity, as well the physiological effects of odors. Torii [10], Lorig [11] and Saito [12] carried out various studies to examine the effect of odors on the brain. They measured the contingent negative variation. Contingent negative variation (CNV) is an increasing negative shift of cortical electrical potentials associated with an anticipated response to an expected stimulus. Lorig [13] established that even low concentrations of odor that remain undetected bring about significant changes in EEG activity and behavior. Subsequently, Kobal [14] at the University of Erlangen demonstrated using electrical brain activity that cognitive processes can be influenced in a controlled manner by the administration of various odors to the right or left nostril. Kikuchi [15] explored the effect of five odorous substances on both heart rate and contingent negative variation. They found that pleasant odors activated the central nervous system increasing the heart rate variations, and calming odors decreased the heart rate.

In addition to the neurophysiological effects of odor substances, psychological effects can also play an important role. Clinical studies over the past 18 years have shown that certain odors promote relaxation and alleviate stress. Warm, Dember and Parasuraman[16] found that people who monitor routine tasks on video displays demonstrated improved performance when they received occasional whiffs of certain odors. These results show that pleasant olfactory substances can enhance task performance. Another study by Danuser suggests that performance deteriorations can be caused by environmental chemical odors and that unpleasant odors are distracters, capable of inducing sensory deprivation.

A number of techniques and scales have evolved to measure the different aspects of odor.. Zevon and Tellegen [17] and Watson [18] proposed that the structure of mood can be represented by two factors, which are called Positive Affect (PA) and Negative Affect (NA). These two factors can be monitored using the PANAS (Positive and Negative Affect Schedule). Another related scale is the OCA scale[19]. Electro-olfactograms (EOG) have also been used to provide evidence for the dominant role of the nervous system in olfactory desensitization, for the functional characterization of the olfactory epithelium, the specific topographical distribution of olfactory neurons, the expression of olfactory receptor neurons in response to exposure to odorants, or characterization of certain odorants as olfactory receptor antagonists.[20] Together with the recording of olfactory event-related potentials[21], magnetic source imaging [22], functional magnetic resonance imaging fMRI [23] [24], and appropriate psychophysical techniques[25], the human sense of smell can be described in its many facets at many different levels.

APPLICATIONS

The Kajima Building in Tokyo was one of the first big buildings to implement the addition of olfactory substances to their new premises in order to enhance the indoor environment. A comfort ratio increase of 24% (simple mean value) was achieved by implementing the olfactory environment.[26] The results in 1999 [27] of a building in Berlin where a new tenant conducted productivity research to satisfy themselves of the payback of higher rent that had been demanded based on the outfitting of the office building. The analysis that was based on their experience over the course of one year after moving to the premises is shown in Figure 1-3. This building was equipped to achieve the full potential of a "multisensual" building.

Real Estate Investor's Perspective

Building data			
Letting area		3,000	m^2
Basic Costs - **Low technology building**			
Property and building		€ 14,730,000	
Technical Equipment (without HVAC & R)		€ 2,450,000	
	Subtotal	**€ 17,180,000**	
Extra Costs - **High technology building**			
Property and building		€ 14,730,000	
HVAC & R		€ 400	per m^2
HVAC & R costs		€ 1,200,000	
Other technical equipment		€ 2,450,000	
	Subtotal	**€ 18,380,000**	
Additional cost for high technology building		**€ 1,200,000**	
Rent Income for low technology building		€ 22	per m^2 per month
Rent Income for high technology building		€ 26	per m^2 per month
Additional rent income for high technology building		**€ 144,000**	
PAYBACK PERIOD - IN CURRENT TERMS		**9.5**	**years**
D.C.F.YIELD - IN CURRENT TERMS		**6.7%**	
D.C.F.YIELD - IN CONSTANT TERMS		**4.1%**	

Assuming tax deductability and depreciation over 15 years.
Inflation set at 2.5%, tax rate at 43%, 15 year life of equipment

Tenant's Perspective

Low technology building

Building costs		
Rent	€ 22.00	per m^2 per month
Ancillary Costs (building)	€ 3.50	per m^2 per month
Subtotal Building costs	€ 918,000	p.a.
Employee Costs		
Employee productivity	100	Index
Sickness/Absenteeism	6%	days p.a.
Workforce (Full time equivalents)	120	FTE
Salaries	€ 3,100	per FTE per month
Other employees costs/benefits	38%	of salary
Office equipment & supplies	€ 200	per FTE per month
Subtotal Employee costs	€ 6,448,320	p.a.
Total building & employee costs	**€ 7,366,320**	

Tenant's Perspective

High technology building

Building costs		
Rent	€ 26.00	per m^2 per month
Ancillary Costs (building)	€ 4.75	per m^2 per month
Subtotal Building costs	€ 1,107,000	p.a.
Employee Costs		
Employee productivity	106	Index
Sickness/Absenteeism	4.2%	days p.a.
Workforce (Full time equivalents)	120	FTE
FTE reduction (productivity improvement)	(6.8)	FTE
FTE reduction (sickness reduced)	(2.0)	FTE
Salaries	€ 3,100	per FTE per month
Other employees costs/benefits	38%	of salary
Office equipment & supplies	€ 200	per FTE per month
Subtotal Employee costs	€ 5,973,800	p.a.
Total building & employee costs	**€ 7,080,800**	
Overall savings	**(€ 286,000)**	**p.a.**
equivalent to	**(3.9%)**	**of operating costs**

CONCLUSION

Changes in the real estate market have created new demands that have to be met in order to create more added value for buildings in particular for vacant buildings.. The competitive advantage of a high quality building can do much to increase return on investment. Buildings with lower standards are less likely to find tenants and will contribute to loss of productivity and absenteeism. Therefore an evaluation of buildings that includes the perception of indoor air and the environment of the occupants is necessary.

In order to fine tune the measurement of the indoor air quality, it is necessary to take into account the influence of the hedonic value of the environment on room occupants. Essentially, the hedonic value affects the well-being of the room occupants.

In order to understand olfactory comfort, we have to look into different psychological, neurological, physiological aspects. Our sense of smell lets us explore our chemical environment. Therefore the perception of an odor will determine our reaction - approach or avoidance, positive or negative - depending on the type and concentration of the odor.

DISCUSSION

An interdisciplinary scientific approach is necessary in order to create an environment which enhances well-being. Perception must play a major role in evaluating a building. Measurements should be based on neurophysiologic and psychological data such as the PANAS and OCA scales and whenever possible supported by research using electro-olfactograms.

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An Investigation into Effect of Air Distribution on the Indoor Air Quality

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SUMMARY

The air distribution is a realizing form of indoor flow field, which plays a vital role to indoor air quality in the design of Heating，Ventilation and Air Conditioning. The average temperature and air speed in working area are closely related to air distribution, and the main factors that affect such two indices are the form, position, quantity of air outlet as well as the velocity, temperature of air supply. This article discusses the effect of air distribution on indoor air quality.

INTRODUCTION

As the healthy environmental protection idea goes deep into the public gradually, how to raise indoor air quality becomes a problem of people's concern. People are also show a tremendous expectation that the air conditioning can improve the indoor air quality. The relevant expert also puts forward the thinking——"to the direction of the healthy air conditioning development". How to improve air quality in the design for air conditioning has become an important study topic in the modern air conditioning technique. This article will make an initial study that the air distribution affects on the indoor air quality in the design air condition system.

INDOOR AIR QUALITY

Concept of the indoor air quality

The formation of the indoor air quality concept is a gradually perfecting process. The indoor air quality differs from an indoor pollution. At first definition of the indoor air quality meant a series of pollutant concentration index. However, along with research of going deep into continuously, people discovered that a single pollutant concentration index can't reflect the indoor air quality accurately. People don't feel comfortable in the room with low pollutant concentration. So in a narrow sense, the indoor air quality can be regarded as the contaminative degree of smoke, dust, harmful air and microorganism. Speaking from the broad sense, the indoor air quality still include thermal environment parameter, such as temperature, humidity and flow speed etc. The indoor air quality is also closely related with the subjective feeling, mentality and the physiology conditioning of the occupants.

Main factors affecting to the indoor air quality

The factors affecting the indoor air quality mainly are the air conditioning system, indoor pollutant and deterioration of outdoor air. As figure 1:

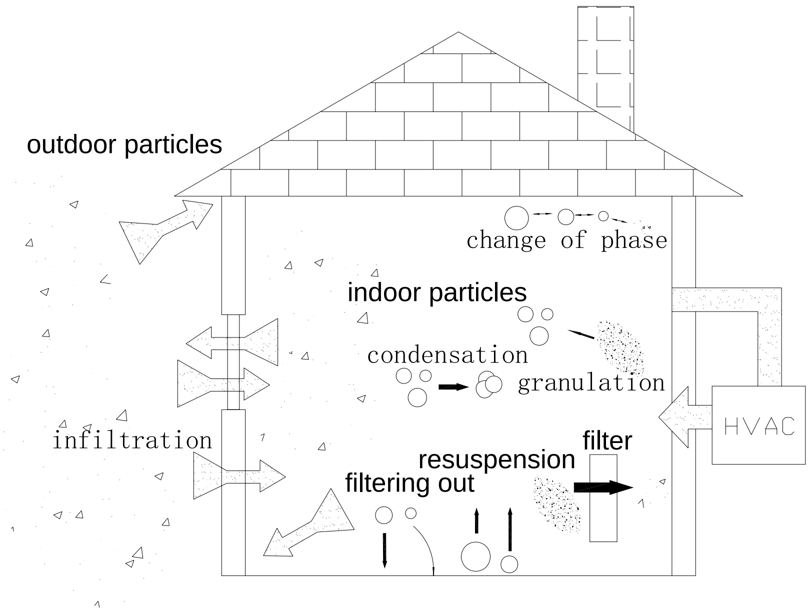

Figure 1

Generally speaking, the main factor affecting the indoor environment quality are: Temperature, relative humidity, air velocity, amount of fresh air, carbon monoxide, carbon dioxide, ozone, formaldehyde, benzene, respirable particles, the volatile organic compound , radon and etc.. Particularly in recent years a great amount of high business buildings has been built. The personnel's density increase continuously. A great amount of materials for decoration to be used, shortage of fresh air occupied, and unilateral pursuance for thermal comfort provided by air conditioning system, all of these worsen the indoor air quality.

Among them, the necessary parameter that insures thermal comfort in the environment is the temperature, relative humidity, the air velocity and etc. The sense of human body to the air environment not only lies on the chemical traces in the air, but also related with the temperature and humidity. So the suitable humidity can improve a people's requirement for air quality. The fresh air can dilute the indoor harmful air and make the air quality satisfied the request for work, living and health.

There are a great lot of kinds of pollutants influencing the indoor air quality, mainly have: inhalational particles, carbon monoxide, carbon dioxide, formaldehyde, volatile organic compound and etc..

The polluted outdoor air will impact the indoor air quality certainly. The outdoor air usually enters through fresh air system, doors, windows and fresh air syetem. At present in the air conditioning system, currently used air filter just trap the dust. It can't purge harmful gas and the germ. When outdoor air is polluted seriously, the volatile gas of organic compound and germ will send to air-conditioned rooms with the air conditioning. It makes indoor air quality grow worse.

THE VENTILATION AND AIRFLOW DISTRIBUTION

Ventilation processes

Ventilation process actually is an organized air renewal system. In order to altering the temperature, humidity and quality we replace the indoor air with fresh air.
At present there are tow modes of air distribution in the air-conditioned room. One is mixing ventilation based on dilution principle, another is displacement ventilation based on the thermal convection.

The typical scheme of mixing ventilation has ceiling and high-level side air supply. These two schemes of air distribution make the work area placed in recirculation zone. Handled fresh air is being sent in from the upper part of room with a certain speed. It merges with the hot contaminated air at first, and then is being sent into work area by backflow after mixing. Adopting this scheme of ventilation, the fresh air inhaled by indoor person has already been polluted during the period of ventilation. Setting out from the view point of air quality, the second scheme of air distribution would be better than the first one.

The devices of the displacement ventilation are often placed in the vicinity of floor. In this case the handled fresh air is being sent in directly to the work area at low speed (<0.5 m/s) and forming a thin layer as a air lake just above the floor because of its higher density. The supplied fresh air is spread out in the lower part of room, it dose not mix with the upper hot air due to low turbulence intensity of flow. When the fresh air comes into contact with hot surface (such as human body and hot equipment) and is being heated, then rising upwards by natural convection. At the same time, the polluted hot air is driven by fresh air flow and is departing from the work area to the upper part of room and finally is extracted by exhaust ventilation system. The indoor air has stratification phenomenon owing to the density difference of the air at the different levels. The denser clean air (lower temperature) sinks in a lower part of room space and the lighter muddy air (higher temperature) rise to the upper part in the room. Thus the indoor personals situated in the work area can inhale fresh air immediately.

Air distribution

The airs distribution is also called air current organization or air current distribution. It is a measure for reasonable air movement and organized flow of air in a ventilated room and for avoiding the formation of dead space and vortex. The mission of air distribution is to determine the location of air inlets and outlets according to the room geometry, technological process, personal position, character of pollutant and disperse conditions, and to make a reasonable air flow pattern. Thereby the temperature, humidity, velocity and cleanness of air within the work area could meet the technological and hygienic requirements.

According to the collocation of air inlets and outlets, there are the following schemes: side wall inlet and side wall outlet, ceiling diffuser, ejector system, ejector type grille, grille with direction blades, ceiling outlet, T-bar slot air diffuser, fan with backward curved blades, side wall register of horizontal and vertical louvers and shutters, floor extract grating, rectangular duct air supply and return opening with damper, round duct air supply and return openings with damper, mesh type air return opening, return grille and etc.

The effect of ventilation and air distribution on indoor air quality

We introduce the concept of efficiency of air exchange and efficiency of ventilation. The efficiency of air exchange is defined as the ratio of real resident time of air in a room to the theoretically shortest resident time of air. It shows the effectiveness of air exchange and is a criterion for assessment and is independent of air distribution.
The schemes of ventilation and air distribution will influence the effectiveness of air exchange, dilution and removal of pollutant from indoor; therefore it could be a different sense of occupants to the air quality.

Increasing the fresh air amount supplied alone, it could not dilute the pollutant concentration properly, because the different air distribution scheme can cause different effectiveness of pollutant removal. Hence a concept of ventilation efficiency has been introduced. Ventilation efficiency is defined as a ratio of the pollutant concentration at air outlet to the average indoor concentration of pollutant. It means the grade of indoor pollutant removal and is an important index for assessing room ventilation. One employ a suitable air distribution scheme under the same conditions (pollutant and its emission rate, quantity of air supplied), if such a scheme can maintain a stable pollutant concentration or can reduce the initial pollutant concentration more quickly, it is to be said that this scheme has a higher ventilation efficiency and is more favourable for raising the indoor air quality.

When designing ventilation and air-conditioning system of a building, one should adopt various schemes of air distribution according to the size and use function of the building. If there are two schemes of air distribution for choosing, one is high-level supply and low-level return and another is vice versa. Generally speaking, the latter scheme has advantage over the former in the respect of improving indoor air quality. In the case of low-level supply and high-level exhaust of air, the handled fresh air is sent to the lower part of room directly to the work area, and so called age of air has a minimum value. By virtue of the lower location of air inlet, lower speed of air discharged and higher density of fresh air supplied, the incoming air will spread out and occupy the whole lower part of room. As the age of fresh air is increasing, it comes into contact with hot source and then is being heated up. After that, hot air rise upwards due to natural convection, carrying pollutant and departing from the work area and finally discharging from the exhaust system.

The principles of design of air distribution system are as follows:

- To place the discharge grille as close as possible to the emission source of harmful matter or the region of higher concentration, and to remove the harmful matter quickly.
- To arrange the air supply diffusers as close as possible to the work area and avoid contamination of fresh air along the air low path.
- To make the air flow more uniform in the room space, reduce voetex and avoid excessive accumulation of harmful matter.

MATHEMATICAL ANALYZE OF EFFECT OF AIR DISTRIBUTION ON THE INDOOR AIR QUALITY UNDER DIFFERENT SCHEMES OF VENTILATION

3.1 The overall ventilation differential equation, its solution and application

According to the balance of pollutant matter, the increment of the indoor pollutant mass should be equal to the difference between the entering and exhausting amounts of pollution mass, namely

The increment of the indoor pollutant mass = entering pollutant mass + emitted pollutant mass － exhausting pollutant mass

Get a overall ventilation differential equation within d t time

$$\rho_{kr} V d\alpha_k = q dt \beta_k \rho_{ks} + k dt \rho_k - \Omega q dt \alpha_k \rho_{kr}$$

where: ρ_{kr} —— density of the pollutant gas in exhaust air

ρ_{ks} —— density of the pollutant gas in supply air

ρ_k —— the density of the indoor pollutant emitted

α_k ——concentration of the pollutant of exhaust air

β_k ——concentration of the pollutant air of supply air

q —— supply air rate

k —— emission of indoor pollutant source

V—— room volume

Ω —— ratio of densities of supply air to the exhaust air

Suppose the room volume is V, the initial concentration of formaldehyde in the room α_0 is 0.1341 mg/m^3, indoor emission quantity of formaldehyde $k_{甲}$ is 0.12 mg/m^3, and formaldehyde concentration of the supply air $\beta_{甲}$ is 0.1287 mg/m^3. (For energy saving, the fresh air mixes with the return air)

Introducing $\varphi = \rho_{ks}/\rho_{kr}; \theta = \rho_k/\rho_{kr}$, and solving the overall ventilation differential equation , we could get the relationship between indoor pollutant and volume V at anytime:

$$\alpha_{甲} = \left[\frac{\varphi}{\Omega}\beta_{甲} + \frac{\theta}{\Omega}\frac{k_{甲}}{V}\frac{1}{q}\right]\left[1 - e^{-\Omega\upsilon\frac{q}{V}t}\right] + \alpha_0 e^{-\Omega\upsilon\frac{q}{V}t}$$

Suppose the experiment as a ideal case, the mixing coefficient (also called ventilation efficiency) $\upsilon = 1$, ventilation process is isothermal process, i.e. $\varphi = \Omega = \theta$.

1. Fixing the room volume, the relation of indoor formaldehyde concentration to the supply air rate and time:

Suppose the room volume V=200 m^3, get the curves as shown in figure 2:

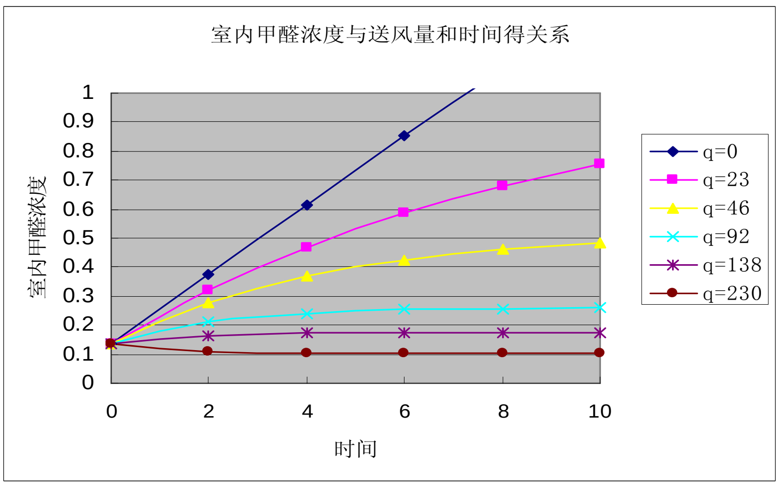

Figure 2

2. Fixing the supply air rate, the relation of indoor formaldehyde concentration to the

room volume and time:

Suppose the supply quantity q=92 m^3/h, get a group of curves shown in figure 3:

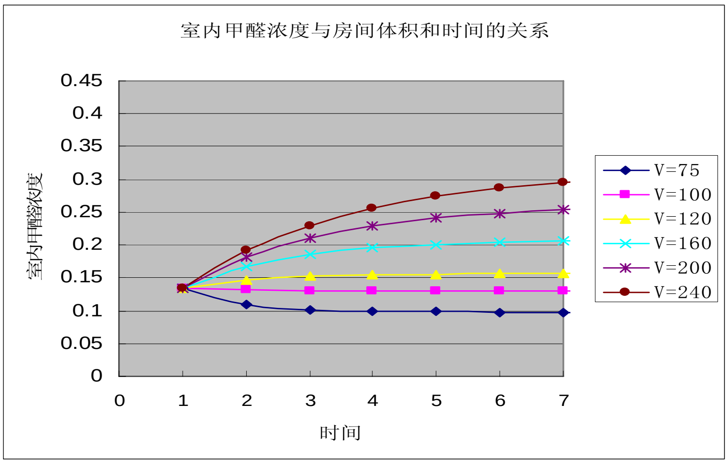

Figure 3

By analyzing above, for fixed supply air rate, the formaldehyde densities all attain a stable value after certain time. For the stuffy case (q=0), the formaldehyde concentration will rise straightly. The bigger the room is, the time is longer for attains a fixed value of formaldehyde. But the stabilized concentration value of formaldehyde is decided by the emitted quantity k of pollution source. It don't relate to the room volume.

CONCLUSIONS

From the analytical comparison above, we can conclude the following that: (1) the indoor air quality is not only decided by the size of the ventilation quantity, but also related closely to the air distribution. For the sake of getting better indoor air quality, we have to choice, dispose and control reasonably. (2) When designing a system we have to choose the suitable one according to the size and room shape, the amount of heat and humidity, position of pollutant source and concentration of pollutant in the room. (3) In engineering practice, the key problem is to keep the temperature, humidity, air velocity and pollutant concentration within a permissible region in the work area. So control of the air quality in the working area is not only the design pivot of the air distribution, but also the research direction of energy saving. (4)In the eyes of the control air quality, under supply is better when one building can use two types of air distribution: up supply and under supply.

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The relationships between the extent of mould problems and physical building characteristics in high-rise apartment buildings

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SUMMARY

This paper studies the extent of mould problems in a high-rise apartment complex with six buildings located in suburban of Seoul, Korea. The complex is composed of 466 households with different stories in buildings. The apartment buildings have occupants' complaints due to mould growth on interior surfaces right after the completion. The research team investigated the sizes of the mould infested areas in each household unit and analyzed them to find any possible relationships between mould extent and physical building characteristics, such as floors of the units, building orientation, number of exposed facades to the outside environment and so on. Although the plan, building materials, construction methods are the same in each apartment household, the difference of mould extent was found with building characteristics. One of the main findings is that the house units on the top floor have larger mould infested areas than the ones on lower stories.

INTRODUCTION

In many countries, microbial growth in indoor spaces has been identified as a main threat to deteriorate the level of Indoor Air Quality (IAQ). Earlier research works have shown that indoor mould growth is likely to occur when a combination of relatively high humidity and temperature on building material surfaces constitutes favorable conditions for mould germination [1]. However, in many cases, mould occurrence involves local, situational, and sometimes idiosyncratic aspects of a building during its operation [2]. Thus, in real situations, it is hard to find the exact causes and appropriate remediation actions, once mould problems are found on building surfaces.

In existing buildings, many studies have been devoted to collect data on the number of mould spores in indoor air using various air sampling methods. However, interpretation of the air sampling results is difficult and dramatically different depending on sampling methods. In the field study by Burge [3], the results of air sampling was not able to indicate fungal contamination in the building, even when visible mould growth appeared in the ventilation system. Miller [4] has concluded that air sampling can be a very useful tool for determining the extent of mould damage of a building only when the exercise of good judgment is provided. The other method is a visual inspection which is to investigate the size of the mould infested areas in existing buildings by experts. Most current guidelines for mould remediation depend on this method and focus on cleaning the affected areas. Although a visual inspection cannot detect hidden mould problems, many mould remediation guidelines have been published and used widely by various agencies and associations. [5, 6, 7, 8].

Despite decades' research on mould problems in buildings, it has not been studied that the relationships between various physical building characteristics and the extent of mould growth in apartment buildings. Thus, this study focuses on the extent of mould problems in existing buildings using a visual inspection method. The objective of this study is to find any possible relationships between mould infested areas and building characteristics, such as floors of households, building orientation, number of exposed facades to the outside environment and so on. One high-rise apartment complex is selected for this study.

METHODS

The selected apartment complex is located in a suburban area of Seoul, Korea, which has hot summer and severe winter. The complex is composed of six high-rise apartment buildings with different number of stories (10 to 24^{th} floors) depending on the lines. The total households are 466 in this complex. 211 units are facing southwest and 255 units are facing southeast. Figure 1 shows the site plan of the selected apartment complex. The structure of the buildings is concrete. Most walls in living areas of the units were finished with wallpaper on the concrete walls except balconies where paint was applied on the concrete walls. The complex has 3 different floor plans in terms of floor area (80, 100, and 160 m^2). Figure 2 gives a typical floor plan of the largest unit with 4 bedrooms, 2 bathrooms, and 4 balconies. Smaller units are composed of 3 bedrooms, 1 bathroom, and 3 balconies.

The construction of the complex was finished in 2000. However, the occupants have reported mould infestations inside the buildings after moving in. In August of 2004 our research team has visited the complex and collected information on the severity of mould infestations in all 446 units. The mould infested areas and the location of the mould problems were investigated in each room in the units. Basic building characteristics were also recorded including floors, the face of the units, number of exposed walls to outside, and so on. In this paper the sizes of mould infested areas are analyzed with each building characteristic to find any possible relationships.

Figure 1. The site plan of the selected apartment complex with six buildings.

Figure 2. A typical floor plan of a unit (floor area of 160 m^2) in the selected apartment complex.

RESULTS

From the visual inspection of all 466 units in the apartment complex, 240 units (about 51.5%) are found with mould infested areas. The largest infested area is found in the unit located on the top floor (24th floor) of the building #5 with 10.84 (m^2). The distributions of the mould areas have been studied for each building as well. Here we introduce two building cases (building #2 and #4) that are the biggest buildings in this complex.

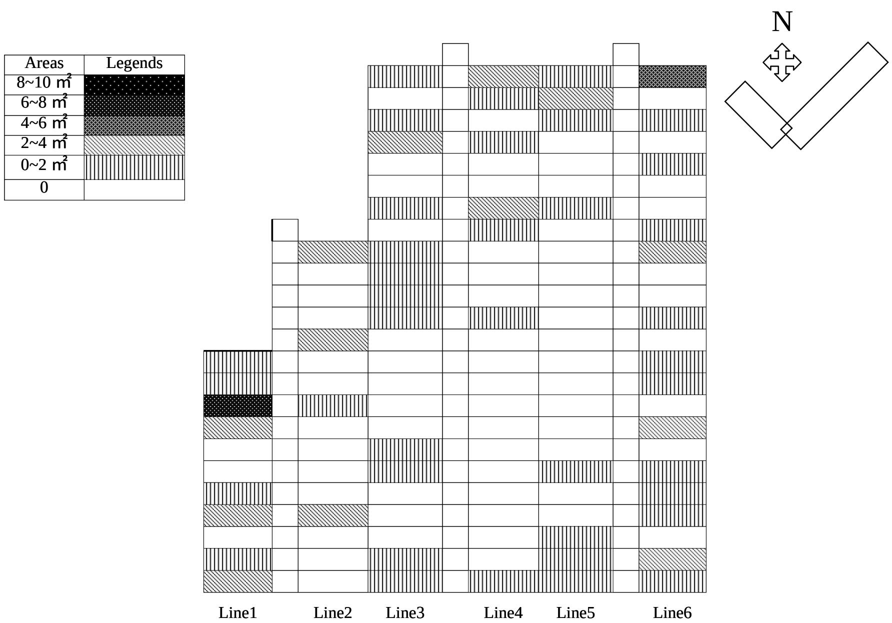

Figure 3. Distribution of mould infested areas in building #2.

Figure 3 shows the distribution of the extent of mould infested areas in building #2. In the figure the areas are grouped in 6 categories according to the mould infested areas from 0 (no mould growth) to 10 m^2. In this particular building all units on the top floor and the bottom floor have mould problems (except one unit on the first floor). The units in the center of the building have much less mould infested areas than the units in the perimeter of the building.

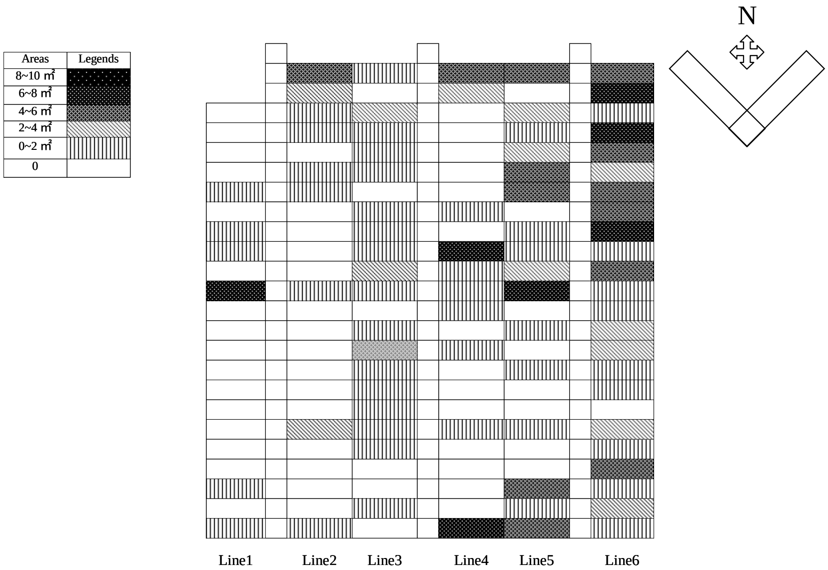

Figure 4. Distribution of mould infested areas in building #4.

Figure 4 shows the distribution of the mould infested areas in building #4. In this particular building the units on the top floor shows larger mould infested areas than other units. However a few units in the middle floors have severe mould infested areas as well. The units on the right side of the building seem to have larger mould area than others. But, the difference between the average mould infested areas in the units facing southeast (1.022 m^2) and southwest (1.104 m^2) is not significant in the analysis of all 466 units in the complex.

The relationships between the floors of the units and the size of the mould infested areas are investigated in the selected apartment complex. Since the lines in the apartment buildings have different top floors, we analyzed mould infested areas in the units according to lines with the same highest top stories. Figure 5 presents the average mould infested areas in the units in the lines with top floor of 24. The average infested area in the units on the top floor is 3.738 m^2, which is 3 times greater than the average value of 1.016 m^2 from all 466 units. The units on the 7th floor have the smallest mould infested area of 0.131 m^2. In this case the mould infested area in the first floor has a larger area than the ones in the middle floors. Thus the units on the bottom floor and the top floor show larger mould infested areas than the ones on middle floors. Similar trend is found in other units in the lines with different top stories, i.e., top story of 10, 15, 19, and 22.

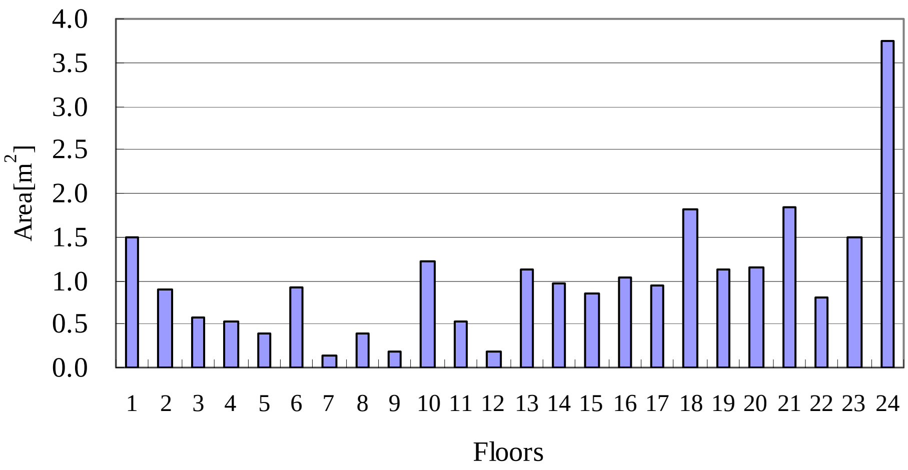

Figure 5. Distribution of mould infested areas in the units that are located in the lines with the top story of 24.

Table 1 Mould infested areas in the units located in the bottom floor and the top floor [m^2].

		BR1	BR2	BR3	Bal.1	Bal.2	Bal.3	Bal.4	Bath1	Bath2	LR	Total & Avg.
Bottom floor units (24)	Area	2.250	6.020	0.200	8.510	4.840	3.720	0.000	0.039	1.016	0.000	26.595
	Avg.	0.094	0.251	0.008	0.355	0.202	0.155	0.000	0.002	0.042	0.000	1.108
Top floor units (24)	Area	2.620	0.000	0.000	19.000	28.630	10.870	0.000	0.685	0.843	0.000	62.648
	Avg.	0.109	0.000	0.000	0.792	1.193	0.453	0.000	0.029	0.035	0.000	2.610

(BR: Bed Room, Bal: Balcony, Bath: Bath Room, LR: Living Room)

Table 1 summarizes the mould infested areas with average values in each room of the units located in the bottom floor and the top floor in all six buildings. The average area for the bottom floor is 1.108 (m^2) that is similar to the average value in all units (1.016 m^2). However, top floor units shows over 2 times higher value of 2.610 (m^2) than the ones in all units. Balconies in the top floor unit have much larger mould infested areas compared to the ones on bottom floor (See Bal.1~Bal.3). Thus, balconies on the top floors are the most vulnerable space for mould growth in these specific apartment buildings.

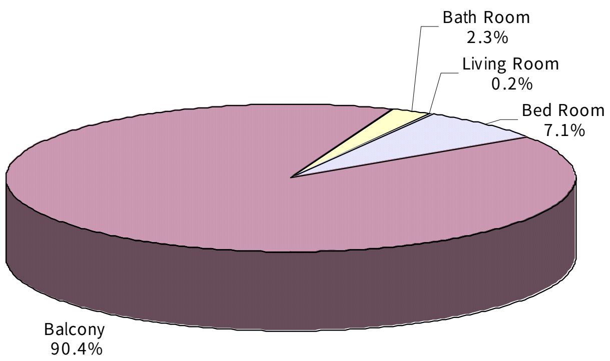

Figure 6. Percentage of mould infested areas in rooms.

Apartment units are typically composed of bedrooms, bathroom(s), living and kitchen areas, and enclosed balconies in a high-rise apartment in Korea. In this particular apartment complex, the balconies are enclosed with concrete walls and windows which are exposed to the environment. Thus, the balconies can be considered an interior space without heating or cooling systems. Figure 6 shows the percentage of mould infested areas in each room. Although the walls in balconies are finished with paint on concrete, balcony accounts for 90.4% of the total infected areas. Bedrooms and bathrooms are account for 7.1% and 2.3% of total area, respectively. It is thought that the severe mould growth on balcony walls is due to the condensation and heat bridge effects on the line joints and window sills. Further research will be conducted to find the main causes of mould growth in the balcony space.

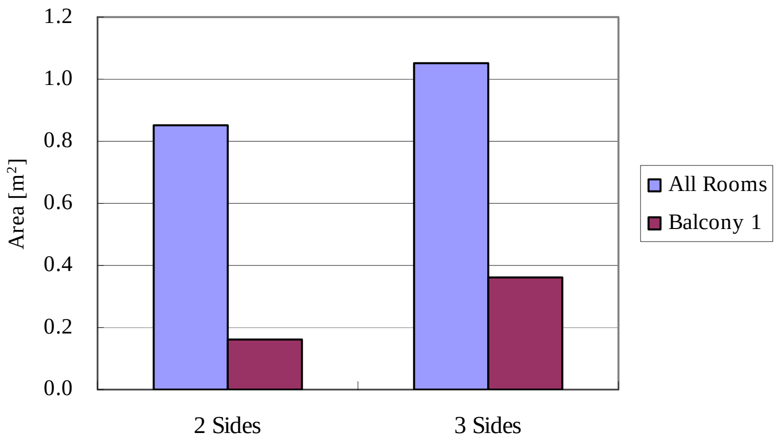

Figure 7. Average mould infested areas in the units with 2 or 3 exposed facades.

Figure 7 shows the average mould infested areas in the units with 2 or 3 exposed facades. This apartment complex has 136 units with 2 exposed facades and 282 units with 3 exposed facades. In this analysis the units located on the top and bottom floors are not included to exclude additional effect of the roofs and ground. The units with 3 exposed facades have an average mould infested area of 1.05 (m^2) which is larger than an average area of 0.85 (m^2) for the units with 2 exposed facades. In the analysis of individual room, balcony 1 accounts for most increase of mould infested area in the units with 3 exposed facades as shown in the figure.

DISCUSSION

This study investigated the mould infested areas in all 466 households in a high-rise apartment complex. In the study of house units located in the first floor and the top floor revealed that the units on the top floor have over 2 times larger average mould infested areas compared to the average value of all units. Especially, the balconies on the top floors are the most vulnerable space for mould growth.

In the analysis of mould infested areas and the number of exposed facades, the units with 3 exposed facades showed larger mould infested areas than the one in the units with 2 exposed facades. Particularly, balconies in the units with 3 exposed facades need special attention to reduce the extent of mold growth. In conclusion, the balconies in the units located on the top floor with 3 façades should be carefully examined to avoid mould favorable environmental conditions.

In this particular apartment buildings balcony spaces showed most severe mould growth in terms of the sizes of the mould infested area, which accounted for 90.4% of the total infested area. It is interesting that mould can grow well on the painted concrete walls in the balconies. According to Baughman[9] and Nielsen [10], concrete is one of the least favorable building materials for mould to grow due to not sufficient amount of nutrients. However, if favorable temperature and humidity are provided for enough time, mould spores on the concrete walls can germinate and grow as reported in the case study by Moon [2]. Thus, it is possible that concrete walls in balcony may have mould infestations due to condensation, heat bridge effects, or not enough thermal insulation. Designers and architectural engineers should pay special attentions to the environmental conditions in balconies.

At this stage of the research no investigations have been conducted to find the main causes of mould growth in the balcony spaces. This will be done as a future research by looking at the details of the buildings and measuring the environmental conditions including surface temperature and relative humidity.

ACKNOWLEDGEMENT

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On prediction of the mold fungus formation probability on critical building components in residential dwellings

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SUMMARY

In buildings, favorable growing conditions for mold fungi can occur and cause fungus infestation. The danger for the occupants of dwellings lies in the spreading of pathogens through microorganisms. Mold fungi can occur not only on the surface of external walls, but also inside construction parts. A prerequisite for preventing mold fungus is the knowledge of the transient building physical boundary conditions under which fungus growth takes place. The decisive parameters of influence like temperature, humidity and substrate have to be available over a certain period of time simultaneously. Presently common evaluation methods do not take into account the transient physical boundary conditions or lack information on mold fungi species and their favorable growing conditions.
The problem is uniquely suited for analysis by a numerical simulation model that accounts for coupled heat, air and moisture transport. A Combined Heat, Air, Moisture and Pollutants Simulation (CHAMPS) model has been supplemented by features to account for the local climate next to the internal surface of the building parts (micro climate) which is essential for predicting the formation of mold fungus under realistic climatic conditions. The paper presents recent advances in CHAMPS modeling as implementation of boundary layer equations for a wall-room coupling and a study on extension to multi-zone simulation.

Keywords: Mold fungi, ventilation, air leakage, hygrothermal properties,
micro climate, building envelope, single and multi zone building simulation

INTRODUCTION

Fungi play a significant role in the indoor environment and their implications range from unacceptable musty smells and defacement or structural damage of interior finishes, to health effects and, indirectly, legal proceedings. There is little doubt that the hygrothermal conditions of the building envelope are one key element in prevention of fungal growth. In the USA, fungal problems frequently occur in a considerable part of the housing stock; in the UK the indications were that 12% was seriously affected, whereas a further 9.5% had slight condensation problems (Sanders and Cornish, 1982). Moisture problems on this scale are not just typical of the temperate climate, and may be found in warmer areas too. In the coastal region of Israel, containing more than 50% of the country 's population, about 45% of the dwellings suffered from condensation and mold growth, with 20% having severe problems (Becker, 1984). The origin of serious fungal growth is surface condensation or interstitial condensation of water vapor, most of it being generated by the normal domestic activities of the occupants. The general impact of planning and design is revealed from statistical analyses of post-occupancy surveys showing that location and orientation of dwellings and occupancy density (occupants/volume ratio) affect risks of indoor fungal growth, which is in line with theoretically anticipated trends (Adan, 1994).

There is a lack of understanding the interaction between outdoor climate, exterior wall constructions (including hygrothermal materials response and air infiltration) and the indoor climate, in relation to heating, moisture production, ventilation and occupancy behavior. Therefore, with funding support from the Syracuse Center of Excellence in Energy and Environmental Systems, the Combined Heat, Air, Moisture and Pollutants Simulation (CHAMPS) transport model has been developed (*http://beesl.syr.edu/champs.htm*).

Quantification of the mold risk and danger for the occupants of dwellings requires modeling of mold spores and MVOC (Microbial Volatile Organic Compounds) transport to the indoor air. The CHAMPS model currently accounts for heat, air and moisture flow in combination with transport and emissions of MVOC in building envelope systems. Understanding the micro climate next to the wall surface (the interior building physical boundary conditions) as one of the most decisive parameters for mold growth, its major affecting factors as the physical processes in the boundary layer and the wall-room interaction need to be investigated.

SCOPE AND METHODOLOGY

The aim of this work is to develop and to apply exemplary a new method that makes it possible to predict and to reduce the probability of mold fungus formation through the following major steps:

1. The infiltration air flows through wall leakages and the convective and diffusive transport of MVOC have to be taken into account for a better quantification of the mold risk inside the envelope construction. A more detailed description of the air and MVOC model and a simulation example can be found in Grunewald et al. 2007.
2. Based on boundary layer models, functionality shall be added to the CHAMPS model to account for the air velocity dependence of surface heat and vapor transfer (exchange or film) coefficients.
3. The CHAMPS model needs to be extended to consider the wall-room interaction (CHAMPS multizone). On the basis of calculated zone temperatures, relative humidities and air fluxes, it shall be possible to predict the input for determination of the local microclimate conditions in much higher precision.
4. The CHAMPS model shall be enhanced by models which describe the mycelium growth rates as function of local relative humidity and temperature. Knowing the microclimatic conditions, the mold growth model can be invoked to determine the conditions under which mycelium growth probability is little.

MODELING THE MICROCLIMATE NEXT TO THE WALL SURFACES

The microclimate next to the interior wall surfaces is determined by the indoor climate, in relation to heating, moisture production, ventilation and occupancy behavior and the design of the building envelope, particularly with respect to thermal bridges, moisture transport and air tightness. The heat and moisture transfer coefficients that govern the transport at the room-wall interfaces depend on the local air velocity. Equations for description of these dependencies can be derived from the set of boundary layer equations under simplifying assumptions. The work to be done can be cut into following areas:

- selection and review of appropriate modeling equations to account for wall-room boundary layers to describe air velocity, diffusion processes and temperature drop / humidity increase next to the wall surfaces,
- experimental and numerical verification of transfer coefficient dependencies for typical building materials,
- development of simplified models to estimate the local air velocity from the indoor air climate conditions under consideration of building geometry and room setup (layout of tables, chairs, other furniture),
- example simulation of typical situations that can lead to mold damages for quantification of a reference microclimate by CHAMPS.

Boundary layer modeling

The aim of this chapter is to highlight briefly the background and assumptions of the boundary layer models. A more detailed derivation of the boundary layer theory can be found in fundamental books for heat and mass transfer (for example in Jischa, 1982 or in Incropera et al. 2007).

The boundary layer equations are derived from a full set of balance equations concerning total mass, linear momentum, internal energy and partial species masses of a fluid phase. Assuming steady, incompressible, laminar 2D flow with constant fluid properties and negligible viscous dissipation, the boundary layer equations can be written as system of coupled differential equations for mass (continuity), momentum, energy and species of a fluid phase.

The solution of these equations is simplified by the fact that for constant properties, conditions in the velocity boundary layer are independent of temperature and species concentration. The solution can be derived by the method of Blasius, 1908, which is based on the similarity of the velocity profiles in the boundary layer stream for a flat plate in parallel flow (see Figure 1).

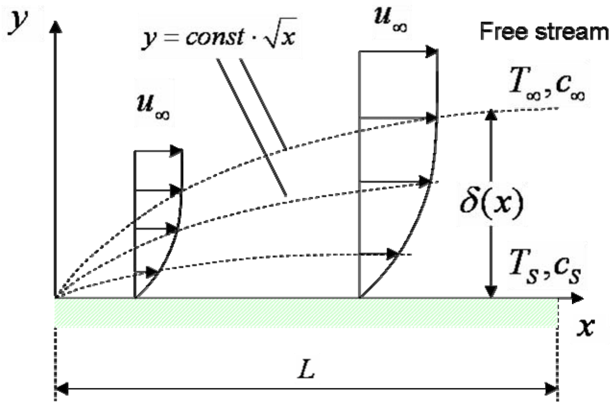

Figure 1: Similarity solution of boundary layer equations for a flat plate in parallel flow.

With respect to the fact that the boundary layer thickness varies proportionally to square roots of the distance measured from the leading edge, the viscosity and the air free stream velocity, new similarity variables for transformation of the coordinates x, y and velocities u, v can be introduced. Substituting these expressions yields a set of ordinary, non-linear differential equations of 2nd and 3rd order which can be numerically integrated. The numerical solution for different Prandtl and Schmidt numbers allows to establish the dependency between the surface transfer coefficients and the free stream air velocity which is given by the local Reynolds number Re_x. For practical applications, average transfer coefficients over a certain characteristic length L are of interest. The constants C and m account for different Reynolds numbers and different geometry like cylinders, squares, hexagon and vertical plate.

mass transfer coefficient $$h = \frac{\lambda}{L} \cdot C \cdot Re^m \cdot Pr^{1/3} \quad (1)$$

heat transfer coefficient $$h_m = \frac{D}{L} \cdot C \cdot Re^m \cdot Sc^{1/3} \quad (2)$$

The evaluation of equations (1) and (2) is depicted in Figure 2. The parameters listed below have been used for the calculation.

1. Input parameters

Thermal conductivity of air	λ =	0.0263	W/mK
Specific heat capacity of air	c_p =	1000	J/kgK
Density of air	ρ =	1.25	kg/m^3
Viscosity of air	ν =	1.55E-05	m^2/s
Specific gas constant of water vapor	R_v =	462	J/kgK
Reference temperature	T =	20	°C
Characteristic length	L =	0.50	m
Slope constant velocity profile	C =	0.664	
Exponent	m =	0.5	

2. Calculated parameters

Vapor diffusivity in air	D(T) =	2.61E-05	m^2/s
Prantl-Number	Pr =	0.74	
Schmidt-Number	Sc =	0.59	

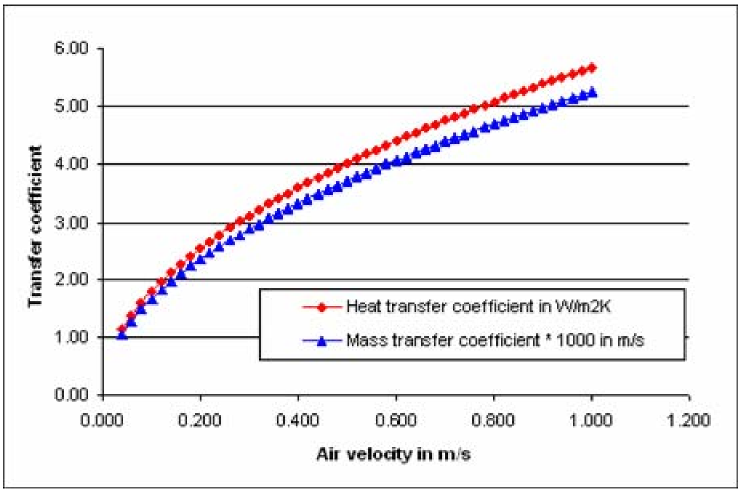

Figure 2. Heat and mass transfer coefficients of a flat plate with parameters as listed above.

Simulation example

The CHAMPS multizone model shall be exemplarily applied to a reference case. It will be quantified how the combined effects vary with outdoor climate conditions including humidity, temperature, sun radiation, rain penetration and wind pressure and the indoor climate conditions in relation to the occupancy behavior.

For quantification of reference microclimate conditions, a typical situation that leads to critical mold risk has been selected. A brick masonry wall of 360 mm thickness is subjected to natural climate conditions (location Amsterdam: temperature, humidity, wind, rain, sun and heat radiation) and sinusoidal indoor climate ($T = 22 \pm 2$°C, RH = 60 ± 5 %, maximum after 215 days). The wall is not insulated and the furniture in the living room is inappropriately arranged. A sofa has been placed next to the wall leaving a small air gap of 2.5 cm and covering the air gap by a pillow or something similar prevents proper ventilation. Figure 3 illustrates the construction and room setup.

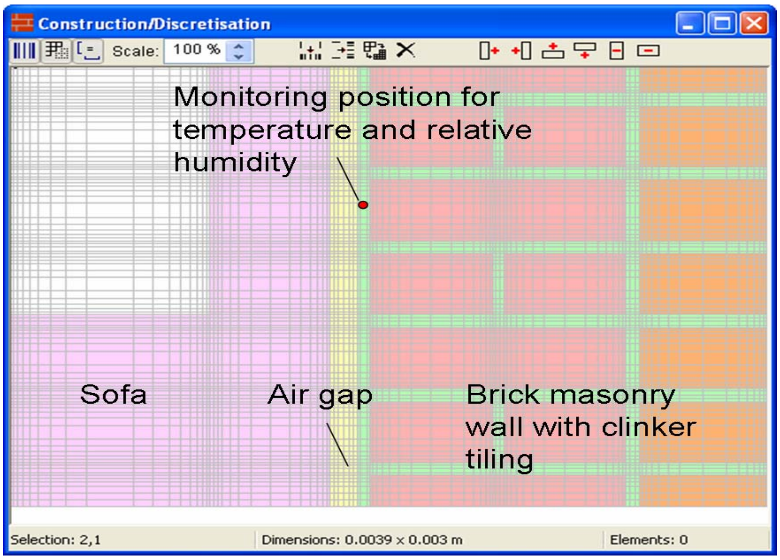

Figure 3: Construction and room setup of the sofa / brick wall simulation example. The red circle indicates the monitoring position for temperature and relative humidity on output.

The simulation is conducted for one year starting at the 1st of July. The sofa acts as thermal insulation which results in an temperature drop and humidity increase within the air layer. The temperature and relative humidity fields after 100 d simulation time (mid of October) are shown in Figure 4. At this time the outside temperature is low (T = 5°C) and condensation of water vapor diffusing through the sofa already went on. The blue color in Figure 4 indicates relative humidity next the wall being almost close to RH = 100 %.

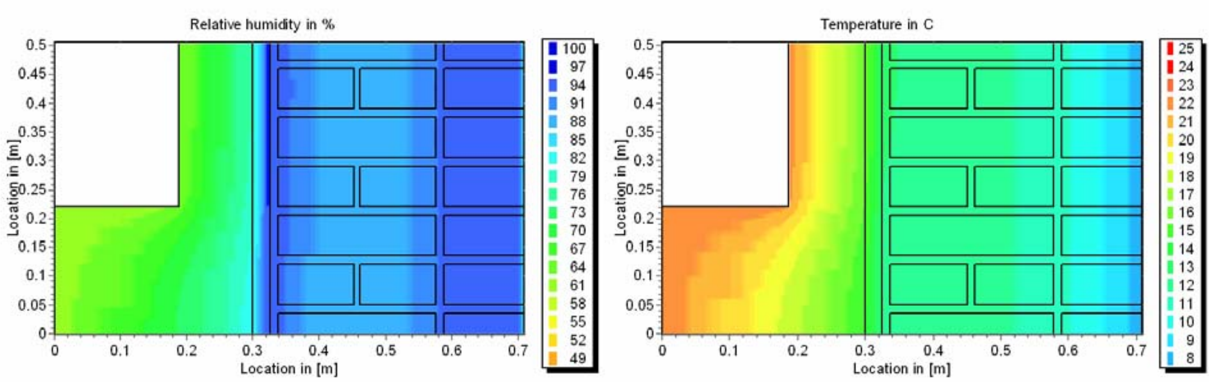

Figure 4: Fields of relative humidity (left) and temperature (right) after 100 days (mid of October).

In wintertime the situation gets even worse. This can be seen in the Figure 5 which depicts the climate at the monitoring positions. From 80 days (beginning of October), the relative humidity is constantly increasing which results in surface condensation and which forms favorable conditions for mold growth.

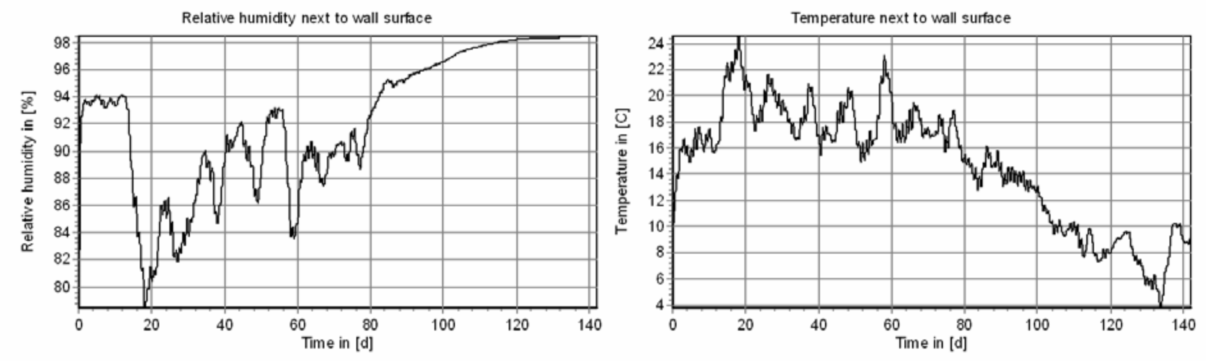

Figure 5: Courses of relative humidity (left) and temperature (right) at the monitoring position; time = 0 is the 1st of July.

This case represents the worst case since the air in the gap beneath the sofa was assumed to be still. The formation of a velocity profile has not yet been taken into consideration. Further simulations are being carried out to quantify the influence of the air velocity in combination with the boundary layer equations described above.

INTERFACING OF MULTIZONE AND ENVELOPE MODELS

The technical possibilities for interfacing CHAMPS multizone and CHAMPS envelope systems (BES) simulation models have been investigated. The focus was on the time integration method, the data handling and data exchange between the different models and codes and the technical details of the interface. Within the scope of this work, a simple representative multi-zone model was implemented. The conclusions drawn from the discussion of this representative model will however be valid for other, more complex multi-zone simulation models as well. For the CHAMPS–room interface the important indoor boundary conditions (related to the conditions in adjacent rooms/zones) can be formulated for each conserved quantity in terms of a heat and vapor flux density.

Discussion

The aforementioned flux quantities need to be formulated depending on the surface conditions of the building component/enclosure and the conditions in the zone adjacent to it. When considering a coupled, simultaneous simulation of multizone and CHAMPS models and the exchange of quantities between these models, one can formally classify the possible methods into *mass/energy-conservative* and *non-conservative* methods. Conservative methods ensure mass/energy conservation within the tolerances and errors of the numerical solution. For non-conservative methods mass/energy conservation may be violated depending on certain conditions.

For the conservative methods the calculated flux quantities need to be consistently used in the multizone model and CHAMPS model. Several conservative and non-conservative methods have been investigated:

- Fully coupled solution, balance equation systems are solved simultaneously
- Decoupled/staggered solution with iterative coupling
- Decoupled solution with imposed interface fluxes
- Non–conservative coupling, most flexible method

Implementation and Results

Of all methods mentioned above, only the last one has been found to be practical, if the indent is to couple complex simulation systems. An interface is defined at each wall surface where temperature, relative humidity and exchange coefficients are stored. Each model uses that data during the given integration interval to calculate boundary fluxes. Additional knowledge about the other coupled models is not required. The performance of the simulation largely depends on the interchange interval lengths. This is depicted in Figure e 6, where simulation times for a one year simulation of a single zone coupled to a one dimensional wall construction is shown, as function of the interchange interval length.

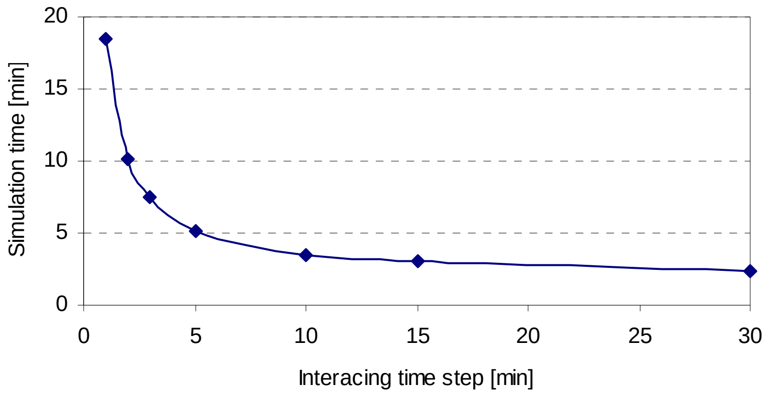

Figure 6. Simulation performance depending in interchange interval length

Once the exchange time step becomes smaller than the time step used by either model, the performance will greatly reduce. For exchange time steps larger than 30 minutes the simulation times do not change significantly. Long interchange intervals are therefore desirable, however, as mentioned above, accuracy (mass/energy conservation) might still be a problem with this method, particularly for larger interchange intervals. To investigate this, several simulation cases have been run, of which one will be briefly discussed here.

An empty room with a volume of 40 m^3 is modeled, ventilated at about 5 ach (0.06 m^3/s), with constant inlet relative humidity of 45% and admittedly a very simple heating schedule; begin of April the inlet temperature is turned down from 25 C to 18 C, and mid-September it is increased again to 25 C. The room has one outside brick wall (surface area 15 m^2), which is exposed to cold and rainy climate in northern Germany. Since the relative humidity is of special interest when determining growing conditions for mold fungi, it is used indicator for the quality of the coupled simulation.

While unexpected, the non-conservative method proved to be fairly accurate, even for larger coupling intervals, as can be seen in Figure 7. Regardless of the interchange interval lengths, from 5 minutes to 12 hours, the calculated relative humidity in the zone does not seem to differ much between the different simulations.

However, Figure 7 shows clearly the problem with larger exchange intervals. While the simulation results for 5 and 30 minute intervals are nearly the same, the relative humidity calculated with 3 hour interval shows already small steps. With 12 hour intervals the steps become too large, to be acceptable. Hence, the maximum permissible time step for this simulation is 30 minutes to one hour. This limit is process dependent and will vary depending on the simulation conditions.

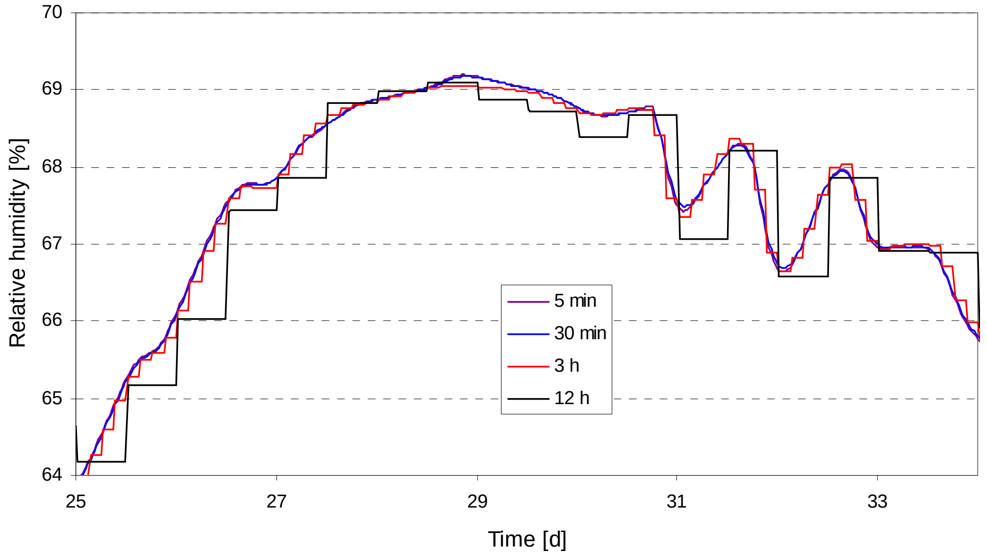

Figure 7. Stepwise change of zone RH for larger exchange intervals

CHAMPS simulations typically require time steps below one hour, yet for performance reasons, the interchange interval lengths should be above 15 minutes. Therefore an exchange time step between 30 minutes and one hour is the recommended compromise between performance and accuracy.

CONCLUSION AND ACKNOWLEDGEMENT

It is necessary that one can state how long and how often which humidity state may act on a building component, before a mould fungus is formed. Further research work is being conducted in the frame of the CARTI CoE project at the Syracuse University with support from the Syracuse Center of Excellence in Energy and Environmental Systems.

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MCS/IEI and Personal Exposures of VOCs in Office Workers

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SUMMARY

In our previous research[1], office workers are relatively frequent in self-reported symptoms of MCS/IEI compare to construction worker who highly exposed to VOCs compounds. However in that research, the subjects were too small to say it clearly. Based on that research results, self reported symptom surveys to 110 office workers and personal exposure concentration measurement were conducted to 13 of people. VOCs exposure levels were measured within office, house and other places for a week by Passive Sampling Method. In this study, it was found that the number of office workers who met the operational criteria of MCS/IEI was 23.6% similar to 24.5% of previous result, and average of TVOC concentration was appeared high state by order of office, house, and personal exposure.

INTRODUCTION

Background

A relation between new houses' high VOCs concentration with MCS/IEI and SBS has been a social issue in Korea now. According to such reason, the study about IAQ in house is proceeding actively in Korea. However, people can be exposed to chemical compounds in various places not newly-built house only. In that reason, a people's daily life exposure to VOCs must be studied.
In our previous research, office workers were relatively frequent in self-reported symptoms of MCS/IEI compare to construction worker who highly exposed to VOCs compounds. However in that research, the subjects were too small to say it clearly and limited to construction worker.
As the next step, in this research, self reported symptom surveys to 110 office workers who are not construction workers were conducted. And personal exposure concentration measurement were conducted to 13 office workers. VOCs exposure levels were measured within office, house and whole day.

Objectives

This study aims to analyze how office workers are exposed to chemical compounds in their daily life, and to know the relation between concentration and their MCS/IEI. This research will show:
(1) the office workers' self reported symptom of MCS/IEI;
(2) the office workers' VOCs exposure levels for a week and VOCs concentration levels in their office and house;
(3) the relation between self reported symptom and VOCs exposure level;

(4) perform the office workers' health risk assessment by benzene.

METHODS

Survey method

Self reported symptom surveys using a questionnaire were completed from 3rd August, 2005 to 7th September, 2005. Subjects were organized in the following job categories: 110 office workers except construction worker and 112 students as the control group.
The questionnaire was same as that of the previous research[1] to compare both research results.
Operational MCS criteria developed by Iowa College[4] was used in this research. Table 1 lists the contents of this criteria.

Table 1. Operational MCS criteria

A. Routine or normal levels of exposure to chemical agents/substances(eg, gasoline, hair spray, paint, perfume, and soap)caused respondent to feel ill

B. Sensitivity (or illness after exposure) is reported to ≥2 of the following:
1. Smog/air pollution
2. Cigarette smoke
3. Vehicle exhaust/fumes
4. Copiers, printers, and office machines
5. Newsprint
6. Pesticides, herbicides, and fertilizers
7. New buildings
8. Carpeting and drapery
9. Organic chemicals, solvents, glues, paints, and fuel
10. Cosmetics, perfumes, hair spray, deodorants, and nail polish
11. Other

C. Symptoms are reported from ≥2 of the following categories:
1. Constitutional (eg, fever, night sweats, fatigue, weight loss, and weight gain)
2. Rheumatologic (eg, joint pain and muscleaches)
3. Neurologic (eg, headaches, sensory loss, tingling, and paralysis)
4. Cardiovascular (eg, palpitations)
5. Gastroenterological (eg, gas, bloating, and abdominal pain)
6. Dermatologic (eg, rash and blisters)
7. Pulmonary (eg, shortness of breath, cough, and wheezing)
8. Cognitive (eg, confusion, difficulty concentrating, and memory loss)

D. Symptoms lead to a behavioral change in ≥1 of the following ways:
1. Wearing a mask, gloves, or special clothes
2. Changing one's lifestyle to minimize chemical exposure
3. Moving to a new home/location
4. Use of special vitamins, supplements, or diets
5. Use of oxygen, antifungal agents, or neutralizing injections/drops

This table gives the operational definition or MCS. The criteria require that a person report illness from chemical sensitivity, report sensitivity to 2 or more types of incitants, have symptoms in at least 2 organ systems, and manifest evidence of impairment or behavioral change in response to the perceived sensitivity.

Personal exposure concentration measurement was conducted to 13 office workers. Each of 13 subjects got three tubes, for 24 hours, for office, and house. Total 39 measurement tubes were distributed, and got back.
Measurement method is Passive Sampling Method using the passive charcoal tube. Tubes installed in office and house collected VOCs during the time subjects kept their residence. When subjects did not dwell, they kept the tube in an aluminum tube bag. The tube for 24 hours was always worn or set around a subject. We made subjects to write their activities in their diary during the measurement period to guess variables may influenced on the VOCs concentration.

Analysis method

We analyzed 39 charcoal tubes for VOCs concentration, 3 travel blank tubes for the pollution from the process of keeping, transportation, and installation, and 3 blank tubes for the pollution of charcoal tubes themselves. Total 45 tubes were analyzed. The result is the mean concentration for a week, and its unit is $\mu g/m^3$. VOCs were extracted from a charcoal tube using the 2 mℓ carbon disulfide as a solvent, and analyzed by Gas Chromatograph (GC).
We analyzed six kinds of VOCs by the Indoor Air Quality Control Law in Korea; that is, benzene, toluene, ethylbenzene, m.p-xylene, o-xylene, and styrene. TVOC were calculated by converting VOCs from hexane to hexadecane into molecular weight of toluene.

RESULTS

Self-reported symptoms of MCS/IEI

Table 2. Subjects' socio demographic characteristics in self-reported symptoms survey

		Office workers (n=110)	Students (n=112)
Gender	Male	39.1% (n=43)	51.7% (n=62)
	Female	60.9% (n=67)	43.1% (n=50)
Age	Under 30s	71.8% (n=79)	97.3% (n=109)
	30s	17.3% (n=19)	2.7% (n=3)
	40s	6.4% (n=7)	-
	50s and over	4.5% (n=5)	-
Highest level of education	High school graduate	10% (n=11)	83.9% (n=94)
	University graduate	90% (n=99)	16.1% (n=18)
Length of service	Less than 5 years	71.8% (n=79)	-
	More than 5 years ~ Less than 10 years	10% (n=11)	-
	More than 10 years	18.2% (n=20)	-
Medical history	Existence	23.6% (n=26)	25% (n=28)
	Nonexistence	76.4% (n=84)	75% (n=84)

Table 2 lists subjects' socio demographic characteristics. The gender distribution was almost same to the previous research[1]. The percentage of under 30s was highest in both male and female groups. As the result, 26 office workers (23.6%) and 18 students (16.1%) were classified as MCS patients. This result was about the same as that of office workers group in previous research. 26 office workers (24.5%) were MCS patients in previous research. Therefore, it proved that office workers group had more MCS patient candidates than exterior and interior workers group. It was not that they were construction related office workers, but that they had usual characteristics of office work.

Personal VOCs exposure measurement

Subjects' socio demographic characteristics

Table 3 lists 13 subjects' socio demographic characteristics. 6 men and 7 women were selected. They were all late 20s except one. 6 people worked less than 1 year, the longest length of service was 5 years 2 months, and the average of length of service was 2.07 years. The age of office building was from 3 years to 20 years. Its mean was 10.5 years. The average of subjects' work hours was 10.1 hours, the longest hour was 13 hours, and the shortest hour was 8 hours. The age of house that can influence on the VOCs concentration was 7.8 years on average, but 8 houses of 13 were less than 5 years. Subjects spent most of their time in house or office, and used computer for many hours in office.
From self reported symptom surveys to these 13 people, 4 of them answered they have oversensitive matters. Especially, subject F was classified as a MCS patient.

Table 3. Subjects' socio demographic characteristics in personal exposure measurement

Subject	Gender	Age	Length of service	Work Hours per day	Age of office building	Ventilation type (office)	Permission of smoking (office)	Commuting method	House type	Age of house
A	Female	24	1year	11	5	Natural	X	Bus	apt.	14
B	Male	27	2years	11	18	Mechanical	X	Subway	apt.	5
C	Male	29	3years	8	4	Mechanical	X	Bus	multiplex	3
D	Female	23	1year	10	3	Mechanical	X	Bus	highrised apt.	3
E	Female	26	1year	9	3	Mechanical	X	Subway	apt.	30
F	Male	27	3years	12	5	Natural	X	Walking	multiplex	1
G	Female	31	3years	13	18	Mechanical	X	Car	highrised apt.	3
H	Female	27	3years	10	20	Mechanical	X	Walking	row house	4
I	Female	31	6years	9	14	Mechanical	X	Car	apt.	30
J	Female	27	4years	8	18	Mechanical	X	Car	apt.	6
K	Male	28	1year	10	14	Mechanical	X	Subway	studio	1
L	Male	27	1year	11	9	Mechanical	X	Walking	studio	1
M	Male	29	1year	9	6	Mechanical	X	subway	studio	1

(apt. : apartment house)

Personal VOCs exposure

Table 4. Individual VOCs concentrations of whole daily life ($\mu g/m^3$)

	Benzene	**Toluene**	**Ethyl benzene**	**M,P-Xylene**	**O-Xylene**	**Styrene**	**TVOC**
A	n.d	24.95	9.70	1.49	3.48	0.96	122.19
B	n.d	30.28	16.31	0.82	0.75	4.67	67.85
C	n.d	28.68	14.93	1.22	5.45	0.78	84.02
D	n.d	46.98	10.08	20.84	0.78	6.38	96.62
E	9.43	47.20	6.53	11.40	0.78	4.20	103.95
F	12.07	122.89	8.41	21.34	1.14	6.68	314.17
G	8.86	35.58	9.66	19.87	0.78	6.82	197.54
H	5.64	62.20	12.31	20.08	1.19	7.07	405.71
I	n.d	17.60	3.17	4.73	0.77	2.05	74.97
J	2.99	26.97	6.87	12.60	1.31	4.77	101.36
K	3.30	64.27	9.15	16.36	1.57	5.04	162.03
L	8.09	118.36	17.69	44.99	2.69	17.58	348.67
M	n.d	72.16	9.12	14.62	1.85	5.06	362.84

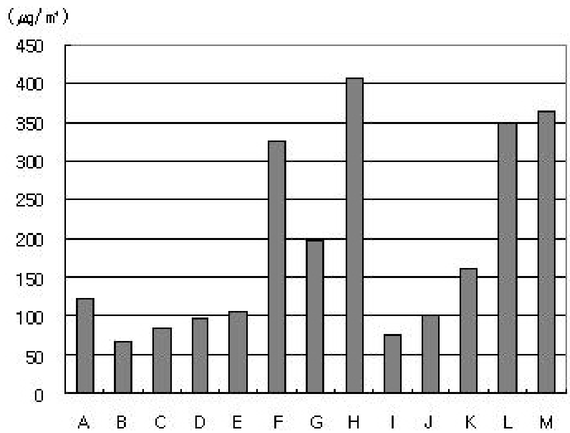

Figure 1. Persoanl TVOCs exposure level

Figure 1 displays the average of personal TVOCs exposure. It means the average of total values of 7 days, including transportation and outdoor space as well as house and office. Table 4 lists the concentrations of individual VOCs.

VOCs exposure in house

Figure 2 displays each subjects' VOCs exposure in their house. Though the target guideline of TVOC announced by WHO is 300 $\mu g/m^3$, the VOCs exposure in house exceeded it in case of C, F, G and L. F's multiplex and L's studio were newly-constructed houses less than 1 year. Table 5 lists the concentrations of individual VOCs in house.

Table 5. Individual VOCs exposure level in house ($\mu g/m^3$)

	Benzene	Toluene	Ethyl benzene	M,P-Xylene	O-Xylene	Styrene	TVOC
A	17.49	22.50	4.24	5.42	2.23	3.04	87.58
B	23.48	33.71	7.97	12.72	2.24	4.94	85.74
C	n.d	14.18	2.78	2.98	1.99	1.60	454.85
D	7.52	51.42	14.97	30.36	1.85	9.19	143.43
E	8.51	30.46	3.42	4.92	1.52	2.50	44.33
F	34.47	736.92	44.46	143.92	5.25	41.46	1487.69
G	6.53	29.39	10.01	17.52	2.02	6.41	961.98
H	5.64	43.81	16.61	2.46	5.88	1.47	96.83
I	14.62	17.46	3.25	4.73	1.81	2.32	94.24
J	8.03	13.02	2.50	2.96	1.71	1.96	24.90
K	n.d	91.37	13.71	25.21	1.44	7.50	219.76
L	n.d	135.45	17.41	47.70	1.98	3.10	385.30
M	n.d	95.83	14.82	22.97	1.74	8.10	191.37

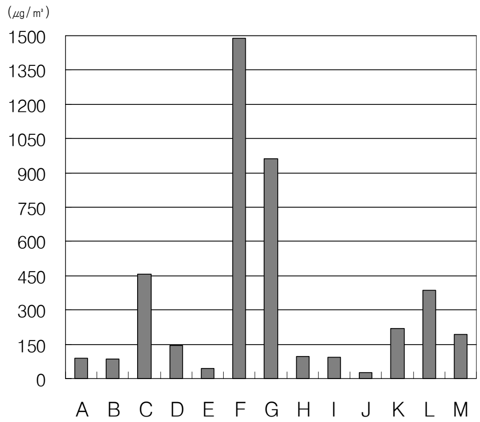

Figure 2. Personal TVOCs exposure level in house

VOCs exposure in office

Table 6. Individual VOCs exposure level in office ($\mu g/m^3$)

	Benzene	Toluene	Ethyl benzene	M,P-Xylene	O-Xylene	Styrene	TVOC
A	12.74	94.36	13.66	20.12	3.97	8.62	485.09
B	0.50	63.66	33.25	2.84	2.16	9.92	158.21
C	n.d	40.46	24.53	2.44	10.36	2.19	86.73
D	10.85	76.80	9.20	21.19	2.67	8.50	144.18
E	9.30	50.43	6.94	11.84	2.45	5.29	116.82
F	19.62	500.83	33.89	85.09	5.84	26.17	1135.19
G	22.18	59.05	11.89	23.71	2.97	11.56	990.91
H	27.63	71.45	11.65	14.97	2.39	6.08	180.42
I	n.d	53.39	8.80	10.46	3.11	5.01	382.77
J	9.43	26.86	6.46	11.15	1.60	4.79	696.37
K	23.03	44.84	6.26	8.85	5.37	4.33	98.95
L	7.36	40.82	10.39	19.01	3.68	6.69	96.83
M	20.83	144.96	11.57	21.09	2.93	8.21	2126.26

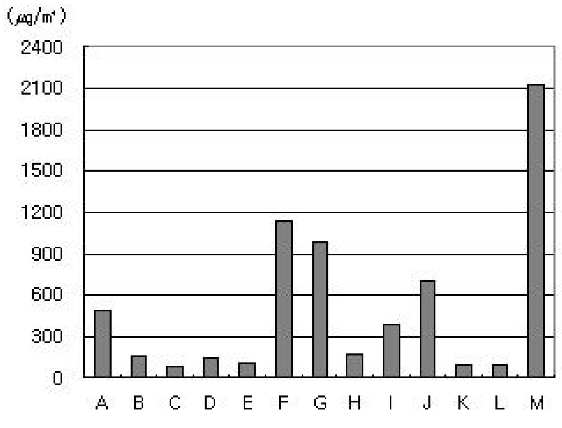

Figure 3. Personal TVOCs exposure level in office

Figure 3 displays each subjects' TVOCs exposure in their office. TVOCs exposure concentrations in office exceeded 300 $\mu g/m^3$ in subject A, F, G, I, J, and M. Especially, M's concentration in office was considerably high. It was 2100 $\mu g/m^3$. After interview, we assumed it caused by a lot of office machine including computers and printers, and the lack of ventilation. M's TVOCs concentration was 7 times higher than WHO target guideline. VOCs concentrations in office of F and G were exceeded 900 $\mu g/m^3$ also. F's office space was small, crowded, and had many office machines. Besides, smoking was permitted after regular office hours, and there was no mechanical ventilation. Table 6 lists the concentrations of individual VOCs in office. Toluene concentration in office was relatively high same to house.

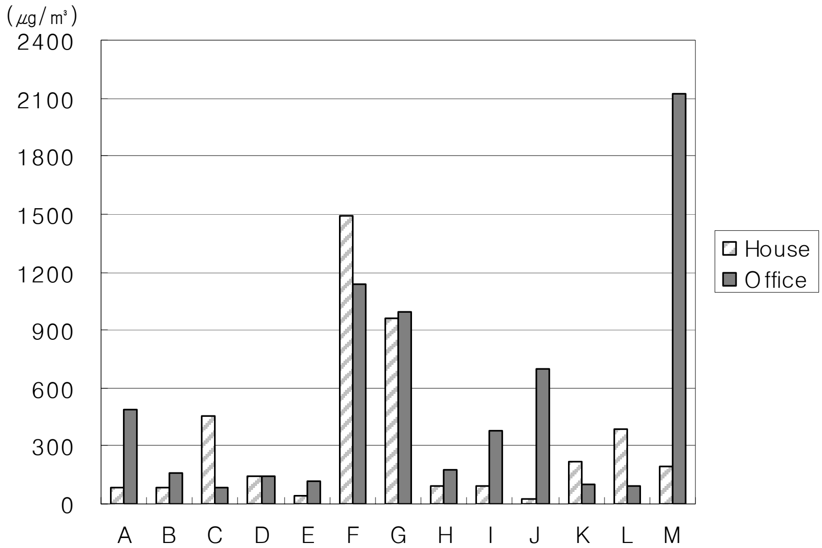

Figure 4. Comparison of TVOCs exposure levels in house and office

Figure 4 displays comparison of TVOCs exposure level in house and office by subjects. Almost all of subjects' TVOCs exposure level in office were higher than in house except C, F, K and L. Therefore, we can assume VOCs concentrations in office are higher than house because of office machines et al.

The relation between self-reported symptoms of MCS/IEI and VOCs concentrations

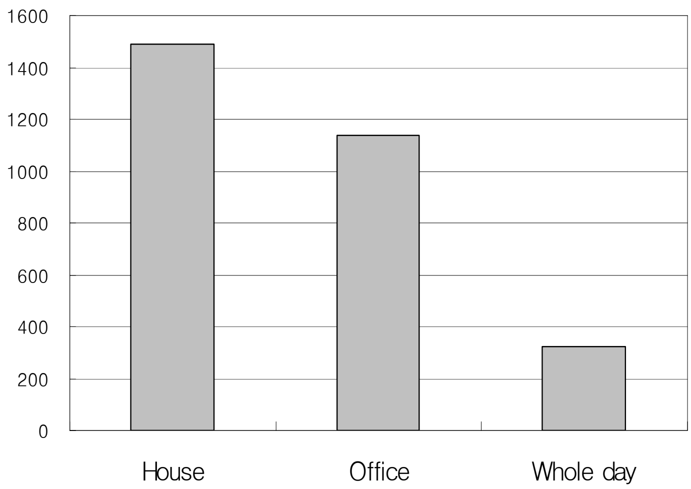

Figure 5. F (MCS patient)'s TVOCs concentrations in daily life

The subject who classified as a MCS patient was F. Figure 5 displays F's TVOCs exposure concentrations in daily life. F has been exposed to high TVOCs concentrations in whole day as well as house and office. Especially, TVOCs concentrations in house and office were about 4~5 times higher than WHO target guideline, 300 ㎍/㎥. These values were much higher than other subjects' values. F went out 4 nights a week. This life style could be one of the causes for high TVOCs concentrations. Toluene concentration exceeded Japanese guide line[5] (260 ㎍/㎥) in both of house and office. Therefore F's risk is serious in the side of individual

VOCs concentrations, too. From this result, we could assume the relation between VOCs exposure concentrations and MCS/IEI risk. Higher exposure make higher risk of MCS/IEI.

Health risk assessment by benzene

Risk assessment method

Mean of benzene exposure concentrations in whole day were used to assess the health risk. We referred to IRIS data of US EPA and other researches for the estimating benzene unit risk. The unit risk means excess carcinogenic probability that can be generated when a healthy adult was exposed to one pollutant(unit concentration: 1 $\mu g/m^3$) during whole life(the average span of a man's life is 70 years in this research). It is calculated through the cancer potency drawn by a dose-response assessment, the adult average weight, and a daily respiratory quotient (equation 1). Table 7 lists the unit risk in this research. A daily respiratory quotient was 20 m^3/day and The adult average weight was 60 kg.

Table 7. The cancer potency and unit risk of benzene

Carcinogens	Dose-response model	Cancer potency ((mg/kg/day)$^{-1}$)	Unit risk (($\mu g/m^3$)$^{-1}$)
Benzene	Low-dose linearity utilizing maximum likelihood estimates (Crump, 1994)	6.66×10^{-3} ~ 2.34×10^{-2}	9.57×10^{-6}

Unit risk (($\mu g/m^3$)$^{-1}$)

$$= \frac{\text{cancer potency}(\text{mg/kg/day})^{-1} \times \text{daily respiratory quotient}(\text{m}^3\text{/day})}{\text{wight}(\text{kg}) \times \text{unit conversion coefficient}(1000\text{ug/mg})} \quad (1)$$

The theoretical number of deaths was assumed through multiplying the carcinogens concentrations of environment by the unit risk and the number of exposure population.

The theoretical cancer mortality rates

$$= \frac{\text{benzene concentration } (mg/m^3) \times \text{benzene air unit risk}((mg/m^3)^{-1})}{70(\text{year}) \times \text{exposure population}} \quad (2)$$

Health risk assessment result

Table 8. The theoretical number of deaths by benzene

	concentration ($\mu g/m^3$)	unit risk	individual risk (mean)		Annual Population Risk (person/year)
			annual	lifetime	
whole daily life	7.20	9.57×10^{-6}	9.83×10^{-7}	6.89×10^{-5}	2.62

Subjects' benzene exposure was 0.1~34.47 $\mu g/m^3$. Benzene concentrations were appeared high state by order of office, house, and whole day.
We needed the carcinogenesis affecting to human body by benzene for assuming the theoretical number of deaths. We used the cancer potency offered by US EPA, and referred to Rinsky's research[6] and Crump's research[7] for the cancer potency of benzene. Table 8 lists the theoretical number of deaths by benzene. An individual risk of 13 subjects was 1.37×10^{-8}

~ 1.65×10^{-6}, and its mean was 9.83×10^{-7}. If office workers are 2,664,000 (from the National Statistical Office in Korea, 2005), the theoretical number of deaths by benzene is 2.62.

CONCLUSION

We performed self reported symptom surveys of office workers, and measured personal exposure VOCs concentration. Based on these results, we performed the carcinogenic risk assessment by benzene. The conclusions are as follows.

First, after self reported symptom surveys of office workers, it was found that office workers group had more MCS patient candidates than exterior and interior construction workers group, or students group.

Second, after VOCs exposure concentrations measurement, it was found that VOCs exposure level in office was higher than VOCs exposure level in house.

Third, the subject who exposed to high VOCs concentrations in both office and house had symptoms of MCS patient.

Finally, The Korean theoretical number of deaths by benzene was 2.62.

From these research results, we can say that continuous exposure to low level of VOCs (eg. office) has more risk of MCS/IEI than instantaneous exposure to high level of VOCs (eg. interior or exterior construction worker).

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Noise measurements in a bloc of flats

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SUMMARY

This study was carried out in collaboration with the district heating firm RADET SA from Bucharest, Romania, in order to support the acoustic renovation of the thermal stations, to meet the modern requirements of the indoor acoustic comfort. The measurement sequence consists in the recording of the sound pressure level in a bloc of flats: inside the thermal station placed at the underground level of the building and inside the apartments. The noise level was 70 dB inside the thermal station and over 44 dB inside the apartments. The most affected rooms are those adjacent to the thermal station and those adjacent to the indoor chimney of the thermal station. These values of the noise levels overtake the standardized recommended maximum value. Multiple solutions should be considered in order to reduce the sound transmission from the thermal station towards the apartments.

INTRODUCTION

The building acoustics is accorded today an enhanced importance. The developing methodology of energy consumption and certification of the buildings considers the acoustic comfort as one of the criteria of the indoor environment quality, alongside with the indoor air quality or thermal comfort. The noise inside the apartments has two main sources: the outdoor and indoor noise sources. Among the indoor noise sources, the thermal station is maybe the most important one. Inside the thermal station the main noise generating sources are the boilers and the pumps [1], [2].

This study was carried out in collaboration with the district heating firm RADET SA from Bucharest, Romania, in order to support the acoustic renovation of the thermal stations, to meet the modern requirements of the indoor acoustic comfort [3], [4]. In this article we present first the measurement protocol, then the measured values of the sound pressure level inside the thermal station and in different apartments, and finally the airborne noise transmission towards the apartment adjacent to the thermal station.

METHODS

The measurements were carried out in a bloc of flats in Bucharest, Romania. A 1.6 MW thermal station is places at its underground level, producing the thermal energy necessary for four blocs. The space of the thermal station is a rectangular shape room of 70 m^2 surface and 4.6 m height (Figure 1). Apartment number 1 is adjacent to the thermal station while the apartments number 6,10,14 and 18 are above the thermal station and the apartments 7,11,15 and 19 are above the apartment number 1.

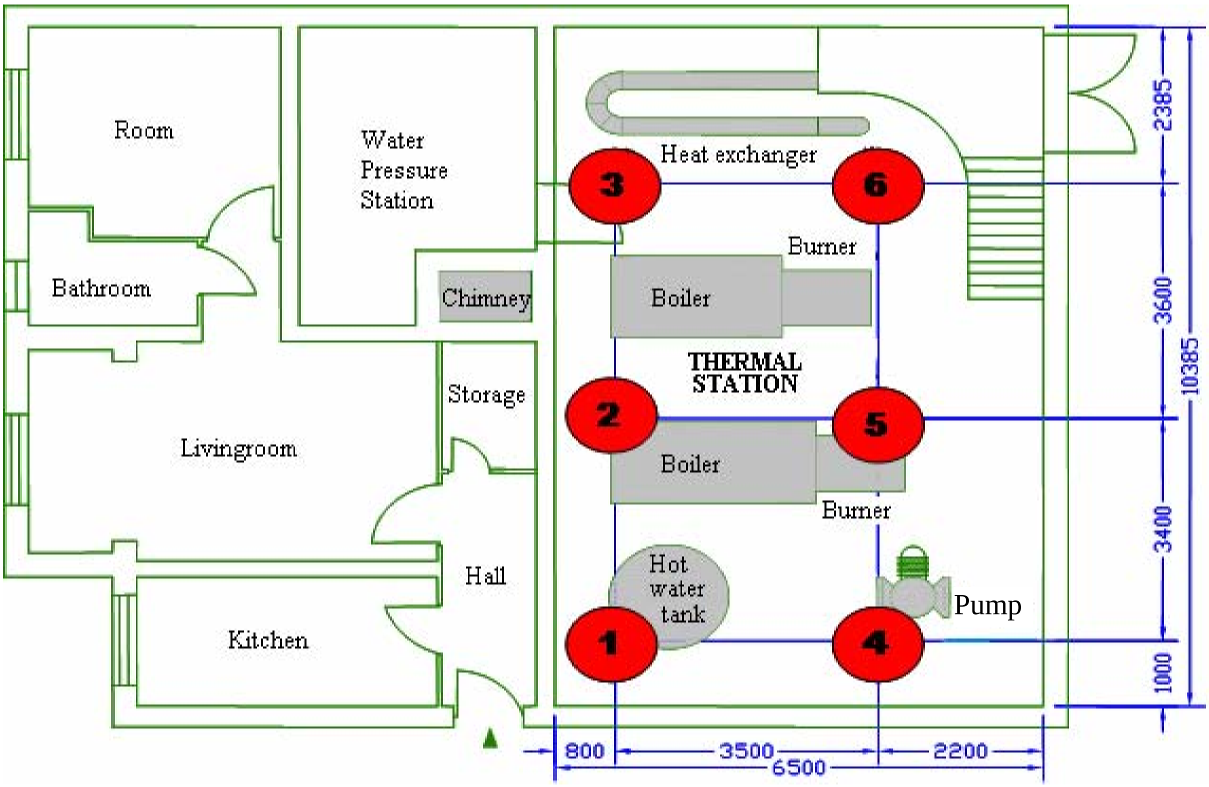


Figure 1. Ground level plan - Apartment no1 in the left and Thermal station in the right

The main equipments of the thermal station are: two boilers (1 MW and 0.6 MW) a tubular heat exchanger for hot water preparation, a hot water hank and pumps. The horizontal part of the chimney is under ground, under the apartment number 1 and the vertical part of the chimney has common walls with the thermal station, the water pressure station and all the apartments.

Sound pressure level recordings were carried out in three conditions:

- when neither the pumps nor the boilers were working,
- only the pumps were working, and
- both the pumps and the boilers were working at full capacity.

Each measurement sequence consists in the recording of the noise level inside the thermal station in six different points (Figure 1) and inside the apartments in the center of each room. All measurements were carried out at 1,4 m height from the floor of the thermal station of the floor of the rooms. One recording represents the equivalent sound level measured during one minute. The measurements were carried out during both daytime (12:00-14:00) and nighttime (22:00-24:00), when outdoor noise was reduced.

The measurements were carried out by means of the 2260 Investigator from Bruel&Kjaer. The visualization of the recorded data and their download were achieved by means of software Noise Explorer Type 7815.

RESULTS

Sound pressure level measurements inside the thermal station

The sound pressure level measurements inside the thermal station during daytime revealed the following values (Figure 2):

- when neither the pumps nor the boilers are working, the pressure level varies (it was measured in six different points) between 45 and 60 dBA,
- when only the pumps are working, the noise level varies between 52 and 63 dBA, and
- when both the pumps and the boilers are working at full capacity, the noise level inside the thermal station varies between 55 and 77 dBA.

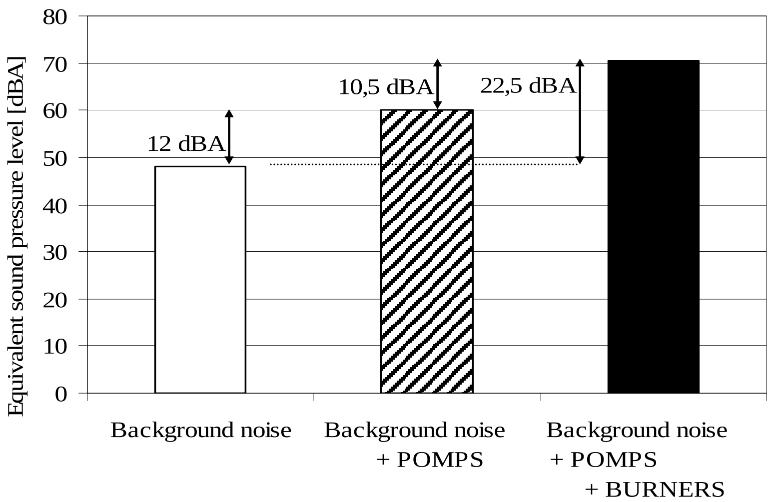


Figure 2. Variation of noise level in the thermal station according to the running status of the thermal station

We did not measure the noise level when only the burners are working because this situation is not often found in the regular running of the thermal station. However, we can calculate its value if we subtract the last two results, resulting that the corresponding noise level is 70 dBA.

Sound pressure level measurements inside the building

The simple visualization of the sound pressure level recorded inside the apartments (Figure 3) can underline some aspects relative to the: the noise sources and sound propagation ways. Theoretically, all the apartment owners wanted to record the noise level inside their apartments, however practically we were allowed inside only three apartments: 1,6 and 11.

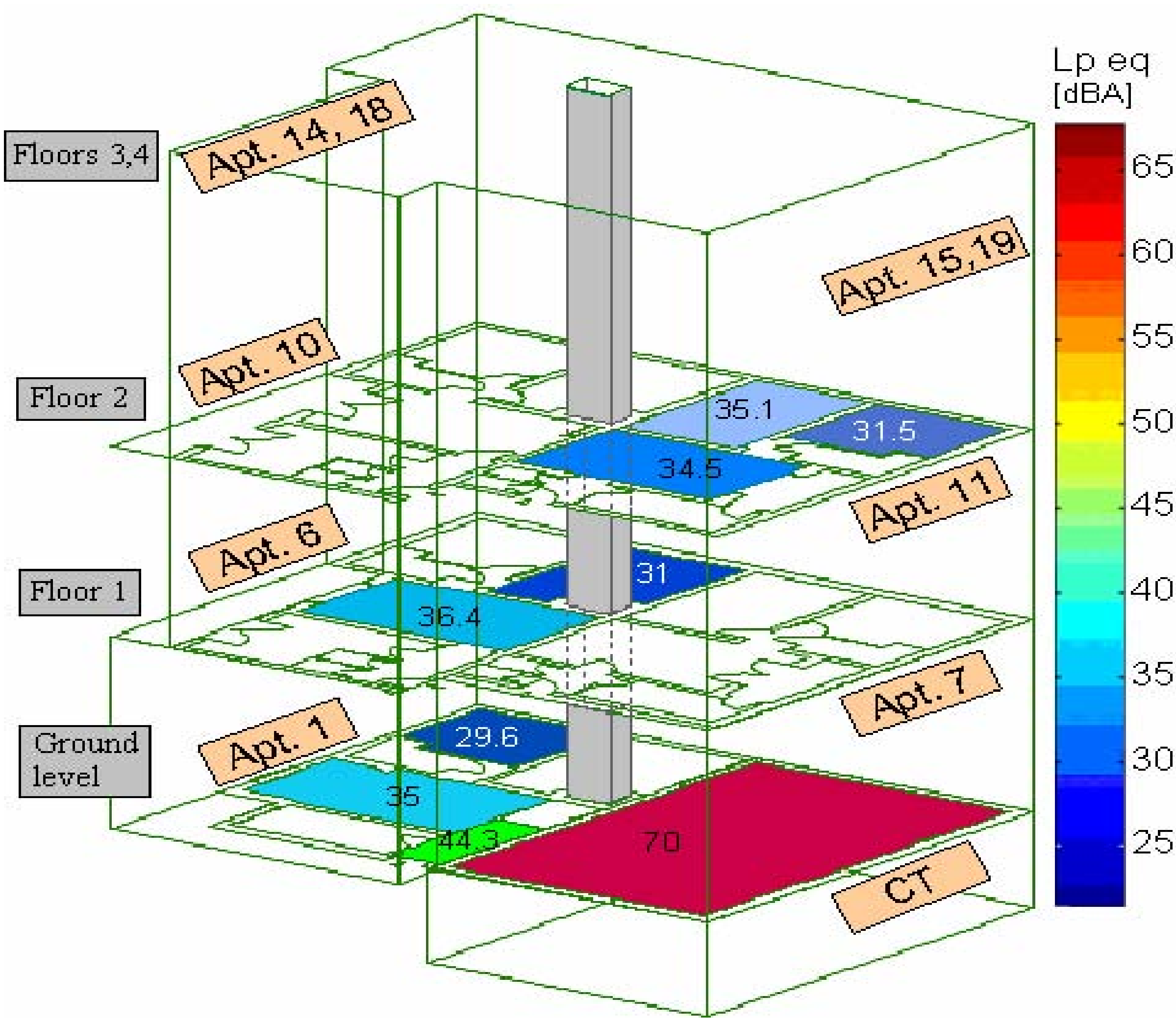


Figure 3. Variation of noise level inside the building

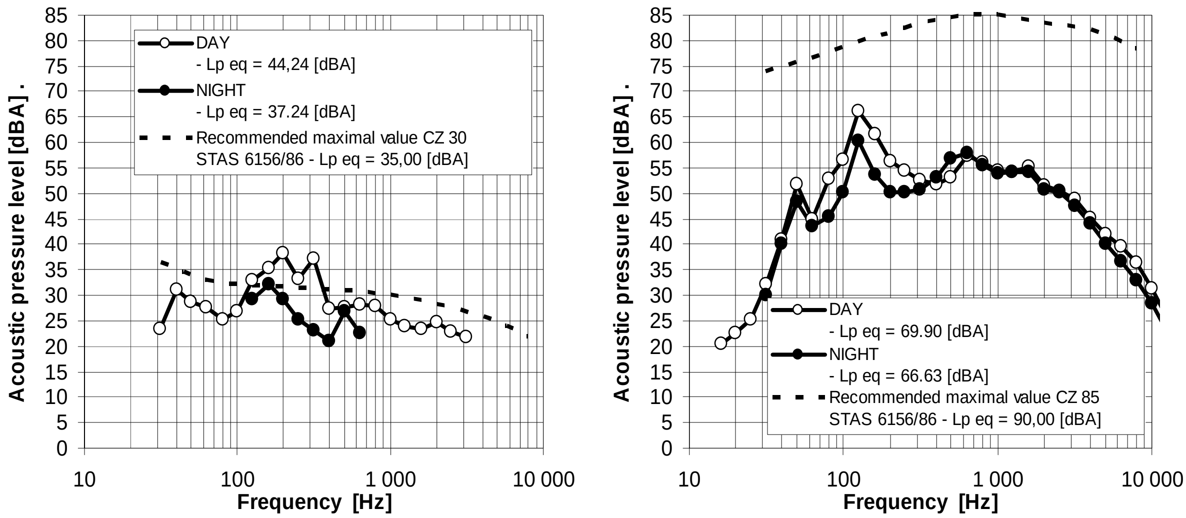
In Figure 3 we represented the architectural plan of the first three floors, and the value of the noise level according to each room. The hall of the apartment number 1 is the most affected (44 dBA) overtaking by 7 dBA the limit set by acoustic comfort standard [4]. In the apartment number 6, we recorded 36.4 dBA in the living room, more that the sound pressure level recorded in the living room of the apartment number 1 (35 dBA). Probably, the storage room of the apartment number 1 protects its living room from the airborne transmission of the noise. Finally, in the apartment number 11, the noise level was higher in the rooms adjacent to the chimney. We can draw out the conclusion that the most affected indoor spaces are those rooms adjacent to the thermal station and the rooms adjacent to the indoor chimney.

Night measurements showed sometimes higher noise level inside the apartments when the thermal station was not running, comparing to the periods when it was running. The conclusion is that the outdoor noise level contributes significantly to the resulting indoor noise level.

Airborne noise transmission through wall

In this paragraph we calculated the noise transmission [5] through the common wall between the thermal station and the apartment number 1, and finally compared its experimental value to a theoretical one.

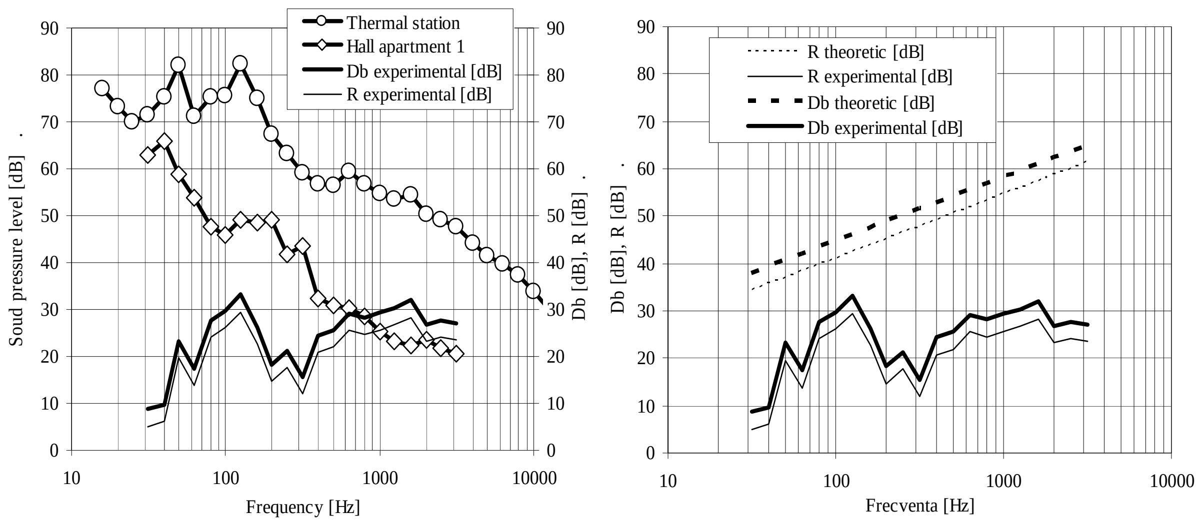
First, we note that sound level inside the apartment number 1 is higher (Figure 4a) than the maximal recommended value [4] for frequencies between 100 Hz and 400 Hz, during daytime. By the contrary inside the thermal station the noise level does not overtakes the maximal recommended value (Figure 4b).



a) b)

Figure 4. Sound levels in a) the hall of the apartment 1, b) the thermal station

The sound transmission loss, represented by means of the two common coefficients Db and R [1],[7], represents the difference of sound level between the thermal station and the hall of the apartment number 1 (Figure 5a). The transmission loss is under 30 dB for almost each frequency, while its theoretical value is much higher (40-65 dB) (Figure 5b). This variable error (15-35 dB) can by explained by a multiple phenomena of sound and vibration transmission.



a) b)

Figure 5. Noise transmission loss: a) experimental values b) theoretic values

Other laboratory experimental researches [6] of the sound transmission loss show a maximum 10 dB error due to airborne noise transmitted through lateral walls. The remaining 25 dB of the error between the experimental and theoretical sound transmission loss is explained by complementary transmission ways, among which we recall:

- the vibration of the boilers,
- airborne noise of the burners, or
- the vibration of the pumps.

DISCUSSION

The noise levels recorded inside the thermal station (1.6 MW) are: 48 dB while the thermal station is nor running, 60 dB when pumps were working and 70,5 dB when both burners and pumps were working. Another interesting result consists in the fact that even if inside the thermal station the sound level limit value was not overtaken, still inside the apartments its value overtakes the recommended maximal value of the acoustic comfort.

Concerning the noise propagation inside the building we would like to mention that some apartment owners were reluctant to "official" measurements inside their homes. Consequently, an acoustically proper installation of the thermal station equipments should be foreseen from the design stage of the installation. In our case, the most affected rooms are those adjacent to the thermal station and those adjacent to the chimney. The noise from the thermal station is transmitted to the apartments by many ways and therefore multiple solutions should be considered in order to reduce the noise transmission:

- to reduce the airborne sound transmission, one should add another small density wall layer (plaster and sound insulation) in order to interrupt the propagation of the sound wave to the wall,
- to reduce the vibration transmission generated by the pumps or boilers, one should avoid the contact between the pipes receiving the vibration from the devices and the walls and also foresee flexible support for the boiler [1],[7],
- to reduce the airborne sound level generated by the burners, one ought to foresee burner covers with sound insulation inside [8].

ACKNOWLEDGEMENT

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A Study of Ventilation and Indoor Air Quality in Hospitality Venues Where Smoking is Allowed

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SUMMARY

Ventilation is an effective method of introducing air into and moving air through buildings to achieve indoor air quality that is comparable to outdoor air quality. However, questions have recently been raised about the effect of ventilation on indoor air quality in hospitality venues where smoking is allowed. Dilution and removal of particles and gasses from various sources within a building and controlling temperature and humidity are the primary reasons for ventilation. Some of the first mechanical ventilation systems were designed to control particles and gasses from tobacco smoke [1]. The objectives of this study were to evaluate the performance of ventilation systems in a number of typical hospitality venues in the United Kingdom where smoking is allowed, and to compare the air quality in these venues with the air quality outside of these venues.

This paper presents measurements of ventilation and indoor air quality including environmental tobacco smoke (ETS) parameters sampled in several locations within each venue to assess the effectiveness of the ventilation systems. The indoor air quality measurements included carbon dioxide, nitrogen dioxide, carbon monoxide, temperature, relative humidity, respirable suspended particles, particulate matter 2.5 ($PM_{2.5}$) and particulate matter 10 (PM_{10}). Particle measurements specific to ETS included Solanesol. The investigators also present measurements taken in the outdoor air as a means of comparing outdoor air quality with the indoor air quality.

This study demonstrates that ventilation is effective in hospitality venues where smoking is permitted and that the indoor air quality of these venues is comparable to the outdoor air quality.

INTRODUCTION

In order to evaluate the effectiveness of ventilation in spaces where smoking is allowed, a case study measuring ventilation and Indoor Air Quality (IAQ) in three well-ventilated hospitality venues in the United Kingdom was conducted.

METHODS

The case study consisted of:

- Phase 1 – Selection of venues
- Phase 2 – Site testing/assessment

Phase 1 – Selection Of Venues

In November 2006, researchers conducted site inspections of a total of twelve hospitality venues located in London, England and Cardiff, Wales. The purpose of the site inspections was to select three well-ventilated hospitality venues where smoking is allowed to be included in the study. Researchers gathered data on each of the twelve venues using a checklist that included the following:

- Location (busy street, city, etc.)
- Type of establishment (pub, restaurant, etc.)
- Ceiling height
- Estimated floor area
- Carbon dioxide levels and ventilation rate
- Seating capacity
- Number of smokers present
- Type of ventilation system
- Tobacco smoke control measures in place

Three of the twelve venues were selected for the case study based on the review of the data collected during the site inspections. The other nine venues were considered inappropriate for inclusion in the case study for various reasons including, ineffective or limited ventilation, low occupancy levels, low smoking densities and limited or no appropriate locations for placement of sampling equipment. Of the three venues selected, one of the venues was located in London, England and two of the venues were located in Cardiff, Wales.

Cardiff

Venue #1 - Traditional Pub

A typical small one-room pub located on the Outskirts of Cardiff with seating for approximately 60 persons.

Venue #2 – Newly Renovated Pub/Restaurant

This venue is a purpose-built restaurant/pub and is considered an up-scale venue that has undergone extensive renovations and upgrades over the past few years. It is located in the heart of Cardiff on the main street, which is heavily traveled by automobiles. This venue has a seating capacity of 850.

London

Venue #3 – Wine Bar/Restaurant

The venue is an elegant restaurant/wine bar located on the basement level of a building located in central London. The street is quite heavily traveled by automobiles and the restaurant has a seating capacity of approximately 120.

Phase 2 – Site Testing/Assessment

The benchmarks used by a consensus of indoor air quality researchers to evaluate indoor air quality in commercial and hospitality environments are carbon dioxide (metabolic activity and ventilation), carbon monoxide and nitrogen dioxide (combustion byproducts), respirable suspended particles (air cleanliness), temperature and humidity (both comfort measurements) [2][3][4][5][6][7][8].

The majority of researchers evaluating indoor air quality in spaces where smoking occurs have used respirable suspended particles (RSP) as a marker for tobacco smoke. RSP is the most commonly used marker because it is easy to measure and the equipment to measure RSP is readily available. RSP is generally defined as a particle less than 4 micrometers in size [9]. Particles are generally measured and reported as RSP, $PM_{2.5}$ (particles less than 2.5 μm) and PM_{10} (particles less than 10 μm).

It is important to note that particulates found in both the indoor and outdoor air, including RSP, $PM_{2.5}$ and PM_{10} are not specific to tobacco smoke and are contributed by many sources including human metabolism, combustion sources (e.g., cooking, heating, dust, automobile exhaust) and other activities [10][11]. However, Solanesol is a particulate that is exclusive to tobacco smoke [12]. Solanesol is known to occur in a constant ratio to the total RSP contributed by tobacco smoke [13][14][15][16]. Thus, measuring Solanesol allows the researcher to determine the total amount of RSP contributed by tobacco smoke to the indoor air. The ratio of Solanesol is approximately 3% by weight of the RSP contributed by tobacco smoke.

Three sampling locations were selected inside each of the hospitality venues. These locations were selected to be representative of occupancy and ventilation. At each of these three locations, both instantaneous and integrating air sampling techniques were utilized. In addition, one outdoor location was selected as representative of ambient conditions.

Instantaneous sampling was conducted for the following parameters both indoors and outdoors:

- Carbon dioxide (CO_2)
- Carbon monoxide (CO)
- Temperature and relative humidity
- Nitrogen dioxide (NO_2)
- Respirable suspended particulate (RSP), Particulate Matter 10 (PM_{10}) and Particulate Matter 2.5 ($PM_{2.5}$)

Instantaneous air quality measurements for the IAQ parameters noted above were collected at one-hour intervals over the course of the 8-hour sampling day at each location.

Instantaneous measurements were collected with the following instrumentation.

Carbon dioxide (CO_2), temperature, relative humidity and carbon monoxide (CO) were measured with the TSI Q-Trak. The Q-Trak has the following specifications:

Sensor	Sensor type	Range	Resolution
Carbon dioxide	Non-dispersive infrared	0 to 5000 ppm	1 ppm
Temperature	Thermistor	0 to 50^{o}C	0.1^{o}C
Relative Humidity	Thin-film capacitive	5 to 95%	0.1%
Carbon monoxide	Electro-chemical	0 to 500 ppm	0.1 ppm

Nitrogen Dioxide was measured with the Biosystems Toxi Pro. The Toxi Pro has the following specifications:

Sensor	Sensor type	Range	Resolution
Nitrogen dioxide	Electro-chemical	0 to 20 ppm	0.1 ppm

Respirable Suspended Particulate (RSP), PM_{10} and $PM_{2.5}$ were measured using a TSI Inc. forward light scattering DustTrak™ aerosol monitor. The cyclone attachment was in place for the RSP measurements and different size selective adaptors were utilized to acquire the PM_{10} and $PM_{2.5}$ measurements. The sensor is a 90^o light scattering laser diode sensor that has a range of 1 to 100,000 ug/m3 and a resolution of +/- 0.1% of the reading or +/- 1 ug/m3, whichever is greater.

Indoor air quality research has shown that Dustrak measurements tend to overestimate indoor particulate levels when compared to the traditional and widely accepted gravimetric sampling and analytical methodology.

The Dustrak is factory calibrated to the standard ISO 12103-1, A1 test dust (formerly Arizona Test Dust). This standard test dust is used because of its wide particle size distribution (Model 8520 Dustrak Operation and Service Manual). In recognition of the potential for the instrument to overestimate dust differing in type and composition from the factory calibration standard a methodology is presented to determine an applicable correction factor for specific aerosols in the Operation and Service Manual.

Research suggesting the need for the application of a correction factor for the Dustrak includes Heal, et al, 2000, Edwards, et al. 2006 and Jenkins 2004 [17][18][19]. This research has determined that the Dustrak overestimates particulate levels when compared to gravimetric sampling techniques by factors ranging from 2 to 3.24.

A correction factor for this case study was calculated based on the method presented in the Operation and Service Manual for the Dustrak.

The correction factor was calculated using data from ten (10) different indoor air quality investigations including 101 comparable data points where both gravimetric RSP data and Dustrak RSP measurements were acquired side-by-side in smoking-permitted environments. From these data and comparison of side-by-side sampling results (Dustrak and gravimetric sampling), the appropriate correction factor was calculated to be 3.84.

In addition to the instantaneous measurements of CO_2, CO, NO_2, RSP, $PM_{2.5}$ and PM_{10}, integrated air sampling was conducted for both Solanesol and RSP (gravimetric) over an 8-hour period at each of the three monitoring locations at each of the three venues to determine time weighted average concentrations of particulates.

For determination of both Solanesol and gravimetric RSP the American Society of Testing and Materials (ASTM) Method D 6271-98 protocol was followed for both collection and analysis [13].

Within the three hospitality venues, investigators observed the following factors in addition to the air quality measurements:

- Number of cigarettes smoked per hour of testing.
- Overall occupancy load at 1-hour intervals.
- Floor layouts and space volume was determined.
- Air movement patterns using smoke pencil testing.

Measurement of Ventilation Rates

In the United Kingdom, the professional standards of practice for the design, installation, operation, maintenance and manufacturing of natural and mechanical ventilation systems are defined by the Chartered Institute of Building Services Engineers (CIBSE). CIBSE has provided recommendations for ventilation rates in smoking and non-smoking environments in "Ventilation and Air Conditioning CIBSE Guide B-2" (2001) [20]. A table of the CIBSE recommended outdoor air supply rates for smoking environments is presented below.

Table 1. Recommended Ventilation Rates – Smoking Environments

Level of Smoking	Proportion of occupants that smoke (%)	Outdoor air supply rate (litres per second per person)	Outdoor air supply rate (cfm per person)
No smoking	0	8	17
Some smoking	25	16	34
Heavy smoking	45	24	51
Very heavy smoking	75	36	76

cfm – cubic feet per minute

The performance of a ventilation system with respect to the amount of outside air provided is usually determined in indoor air quality research by measuring concentrations of CO_2 in the indoor environment [21]. For this study, ventilation rates were calculated using CO_2 as a surrogate marker. The following mass balance equation was used to determine ventilation rate in the spaces using the carbon dioxide levels measured inside and outside. This equation is presented in Appendix C of the ASHRAE Standard 62-2004 [22].

$$C_s = N/V_{oa} + C_{oa} \qquad \text{or} \qquad V_{oa} = N/(C_s - C_{oa})$$

Where,
C_s – carbon dioxide in the space
N – CO_2 generation rate per person based on activity level (0.0106 cfm for persons performing office work
V_{oa} – Ventilation rate in cfm
C_{oa} – CO_2 level outside using 350 ppm estimate (expressed as 0.00035)

RESULTS

Table 2 below shows data specific to each venue with regard to ventilation and occupancy.

Table 2. Information on Each Venue – Ventilation and Occupancy

	Venue #1	Venue #2	Venue #3
Ventilation Rate	80 (40)	64 (32)	51 (25.5)
Average hourly occupant count	13	65	40
Cigarettes smoked per hour	12	29	6.0
Smoking Rate*	1	0.5	0.2

Ventilation rate – in cubic feet per minute of air per person (litres per second per person)
* - Cigarettes smoked per person per hour

All three of the venues were observed to be well ventilated. The ventilation rates measured in the venues were within the ventilation rates recommended by CIBSE for the amount of smoking observed. The occupancy rates and smoking rates were typical for hospitality venues in the UK.

The following tables present indoor and outdoor air quality measured for each of the three venues.

Table 3. Indoor and Outdoor Measurements, Venue #3 – December 7, 2006

	Venue #3	Outdoors
CO (ppm)	2.9	3.0
NO_2 (ppm)	< 0.1	< 0.1
RSP – instantaneous ($\mu g/m^3$)	29	42
$PM_{2.5}$ ($\mu g/m^3$)	27.6	41.3
PM_{10} ($\mu g/m^3$)	34.0	48.7
RSP – gravimetric ($\mu g/m^3$)	30.0	Not measured
Solanesol ($\mu g/m^3$)	< 0.5	Not measured

Table 4. Indoor and Outdoor Measurements, Venue #2 – December 5, 2006

	Venue #2	Outdoors
CO (ppm)	2.9	2.5
NO_2 (ppm)	< 0.1	< 0.1
RSP – instantaneous ($\mu g/m^3$)	33.9	24.9
$PM_{2.5}$ ($\mu g/m^3$)	33.7	25.3
PM_{10} ($\mu g/m^3$)	34.9	29.4
RSP – gravimetric ($\mu g/m^3$)	56.7	Not measured
Solanesol ($\mu g/m^3$)	< 0.5	Not measured

Table 5. Indoor and Outdoor Measurements, Venue #1 – December 4, 2006

	Venue #1	Outdoors
CO (ppm)	3.1	2.9
NO_2 (ppm)	< 0.1	< 0.1
RSP – instantaneous ($\mu g/m^3$)	49.4	21.3
$PM_{2.5}$ ($\mu g/m^3$)	46.3	18
PM_{10} ($\mu g/m^3$)	49.4	24.3
RSP – gravimetric ($\mu g/m^3$)	43.0	Not measured
Solanesol ($\mu g/m^3$)	< 0.5	Not measured

Table 6. Average Results from All Venues

	Average	Outdoors
CO (ppm)	2.97	2.8
NO_2 (ppm)	< 0.1	< 0.1
RSP – instantaneous ($\mu g/m^3$)	37.4	29.4
$PM_{2.5}$ ($\mu g/m^3$)	35.9	28.2
PM_{10} ($\mu g/m^3$)	39.4	34.1
RSP – gravimetric ($\mu g/m^3$)	43.2	Not measured
Solanesol ($\mu g/m^3$)	< 0.5	Not measured

Indoor carbon monoxide levels averaged 2.97 ppm while outdoor levels average 2.8 ppm showing that indoor results were found to be consistent with outside.

Indoor and outdoor nitrogen dioxide levels were consistently found to be below levels of detection (0.1 ppm) for the methodology used.

Indoor RSP (instantaneous), $PM_{2.5}$ and PM_{10} averaged 37.4 $\mu g/m^3$, 35.9 $\mu g/m^3$ and 39.4 $\mu g/m^3$ respectively. Outdoor RSP (instantaneous), $PM_{2.5}$ and PM_{10} averaged 29.4 $\mu g/m^3$, 28.2 $\mu g/m^3$ and 34.1 $\mu g/m^3$ respectively. RSP (gravimetric) averaged 43.2 $\mu g/m^3$.

Based on the results of this case study measuring air quality and ventilation parameters, we conclude that the effective ventilation of the hospitality venues was able to achieve levels of particles and gases in an indoor environment, where smoking occurs, that are comparable to the levels of particles and gases present in the outdoor environment.

DISCUSSION

Ventilation is the intentional introduction into and movement of ventilation air through an enclosed space. The purpose of ventilation is to achieve levels of particles and gasses in the indoor environment that are comparable to levels of particles and gasses in the outdoor air and to control temperature and humidity. Neither air filtration (cleaning) or air conditioning is ventilation because neither process introduces air into or moves air through an enclosed space. The results of this study which includes three well ventilated hospitality venues which are typical of similar venues located throughout the UK demonstrates that ventilation systems when operated effectively can achieve levels of particles and gasses in an indoor environment where smoking occurs that are comparable to levels of particles and gasses present in the outdoor environment. It is interesting to note also that in the three hospitality venues studied where smoking occurred the levels of particles and gasses measured were also found to be comparable or lower than levels of particles and gasses measured in similar hospitality venues where smoking is not allowed and does not occur [23][24][25][26].

This is a case study limited to three hospitality venues however the results are important because they show that ventilation can be an effective method of providing air in enclosed spaces where smoking is allowed that is of comparable quality to the ambient outside air. Further research is required to statistically validate this finding.

ACKNOWLEDGEMENT

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Human responses to thermal environment

Individual Thermal Comfort and Energy Optimization

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SUMMARY

This paper describes a methodology for improving individual thermal comfort in an office building without increasing energy consumption. Our approach is based on the observation that an individual's preferred temperature is not a precise value, but a range around a preferred value. We take advantage of this to optimize the temperature settings of each occupant's personal space to minimize the overall energy use. We have employed a nonlinear programming approach to improve individual thermal comfort without increasing energy consumption, even possibly saving energy. This optimization approach has been applied with very encouraging results for many office population distributions in 15 USA cities, and over a wide range of interior wall thermal resistances. This approach has also been generalized and simplified to allow optimization with a reduced set of inputs.

INTRODUCTION

Problem Statement

Conditions for human thermal comfort in built environments have been defined [1] and adopted in the published standards, such as ASHRAE 55-2004 [2] and ISO 7730 [3]. These standards specify the range of thermal conditions (temperature, humidity, etc) under which a majority of the occupants would be satisfied, using the widely used PMV and PPD indexes of Fanger [1]. Because of individual physiological and psychological differences, it is impossible to satisfy everyone if the same indoor conditions are provided to all occupants. In fact, the standards are based on average criteria for population comfort, indicating that building environmental control systems be designed to meet the thermal comfort needs of about 80% of the occupants, leaving a significant (PPD ≈ 20%) of the occupants in various states of thermal discomfort. Such "*One-Size-Fits-All*" (OSFA) systems employ only one (or a few) thermostat(s) to control the temperature of relatively large building zones, each occupied by a relatively large number of people.

Numerous researchers have shown that allowing building occupants to adjust their local environments to their liking improves their satisfaction and productivity in the workplace [4-6]. This philosophy has been called "*Have-It-Your-Way*" (HIYW) in this study. Tham [4] studied the effects of the ambient temperature and outdoor air supply rates on the performance of call center operators. He observed that the operators were more productive at the lower temperature (22.5°C), and were able to tolerate, without a decrease in productivity, the more typical higher temperature of 24.5°C when the outdoor air supply rate was doubled. Wyon [5] estimated that by providing individual control equivalent to ±3°C, productivity would increase 2.7% for logical thinking, 7.0% for general office work, 3.4% for very skilled manual work, and 8.6% for very rapid manual work. In a later field study [6], Wyon also showed that poor air quality, resulting from low outdoor air supply rate and uncomfortable temperatures can have a negative effect on office productivity in the range of 6-9%. Kroner *et al* reported equally impressive gains in satisfaction and productivity when personal environmental control was enabled [7]. Fisk [8] estimated the potential annual gain of the improved worker performance is between $20 billion and $160 billion (in 1996 U.S. dollars) due to the changes in thermal environmental and lighting conditions.

While the studies cited above demonstrate the comfort and productivity value of personalized environmental control, the building-wide energy consumption implications of personalized control have not been adequately explored -- specifically, the concern that higher energy consumption would result from individualized environmental control in buildings where various, occupant-controlled microenvironments communicate through conducting walls, partitions and openings. In this paper, we present the results of an investigation of comfort and energy optimization in simulated office buildings employing the HIYW approach, and describe an optimization and control strategy that would provide either enhanced thermal comfort to *all* occupants at the same, or lower energy consumption than OSFA systems, or achieve lower energy consumption than OSFA systems at the same or lower PPD.

METHODS

For this study we employed a constrained optimization method to either: a) maximize the occupants' thermal comfort with no increase in energy consumption, or b) minimize energy consumption with no deterioration in thermal comfort. Between these two alternatives, there are many other feasible scenarios that achieve improvements in both objective functions. The essential technology enabling us to achieve these objectives is the provision of distributed environmental control systems that can both be adjusted by the individual occupants to suit their specific comfort conditions, and optimized by the system manager to achieve minimum overall energy consumption – what we designate as the optimized HIYW approach.

To achieve these objectives, it is necessary to develop two linked simulation models: 1) a building energy consumption model that accounts not only for internal and external thermal loads, but also for energy transfer between individual offices within the building, when each office is independently controlled by an occupant-adjusted thermostat, and 2) a model of individual (not population) thermal comfort. An added benefit of introducing an individual thermal comfort model is that it allows us to identify building environmental control scenarios under which some occupants might be very dissatisfied with their thermal environments over extended periods of time, even though the PPD is well under the 20% value accepted by the standards [2, 3]. These two basic models must be executed simultaneously in an iterative manner involving thousands of iterations within an optimization scheme to satisfy the specified objective function subject to the imposed constraints.

Building Energy Model

There are a number of software programs dedicated to the simulation of building energy usage, e.g., TRNSYS [9] and US DoE energy+ [10]. These models are designed to perform hourly (or sub-hourly) time-dependent simulations of complex buildings subject to diurnal and seasonal weather and internal load (e.g., occupancy) variations. As capable as these codes are, their execution time is too long to make them practical for the intensely iterative simulations necessary in the present optimization study. This is because HIYW systems require that the building be simulated as a large number of interacting zones (rooms), each with its own thermostat, not just a few well-mixed zones. Put differently, HIYW systems have a much higher number of degrees of freedom than the OSFA systems. For the optimization analyses explored in this study, a rapidly-executing building energy model was required (executing in ~1 second versus the several minutes needed for TRNSYS or e+ transient simulations).

In this study, we opted for a very fast thermal circuit model of the building, coupled with a temperature-bin weather model. This is a lumped parameter approach in which each room, as well as the outdoors, is represented by a node (a temperature) connected to other nodes by a network of thermal resistances [11]. Since the temperature differences between adjacent rooms are small, radiation was linearized and lumped with convection in the thermal circuit representation of the building. Equation (1) expresses the overall thermal conductance and heat flow between nodes *i* and *j*. A higher value of the conductance implies a stronger coupling between adjacent offices, which may make it more difficult to achieve HIYW.

$$U_{ij} = \dot{m}_{ij} c_p + \frac{1}{\frac{1}{A_{ij} h_i} + \frac{L}{k A_{ij}} + \frac{1}{A_{ij} h_j}} = C_{ij} + \frac{1}{R_{ij}}\,;\; \dot{Q}_{ij} = U_{ij}\left(T_j - T_i\right), \qquad (1)$$

The basic building used in the analysis consists of a single story office building, divided into *n* x *m* well-mixed zones (nodes), with a typical HVAC system consisting of a heat pump and a furnace. A node corresponds to an office with three internal heat sources (occupant, computer, and task lighting). Figure 1 depicts a typical node in the thermal circuit. Zone *i* at T_i is connected to 3 neighboring zones at T_{i-1}, T_{i+1}, and T_{i+n} by a resistance that is appropriate to internal walls (R_{in}), and it is also connected to the outside (at temperature T_o) by a resistance that is appropriate to the exterior walls (R_{out}). The model allows for prescribed advective flows, m_{ij}, between the rooms. The model ignores the thermal inertia of the walls and enclosure and does not account for time dependency, allowing its use in combination with the much simpler (and faster) temperature-bin method to incorporate NREL's TMY2 climatic conditions [12] in annual energy computations. In this study, we included only bin hours during business hours (6:00 AM to 7:00 PM), stipulating that a OSFA strategy is adequate during unoccupied hours. We later performed comparisons with time-dependent TRNSYS simulations for 15 cities representing a wide range of climatic conditions in the United States and for different building populations (individual thermal preferences). These calculations confirmed that the temperature-bin approach is adequate insofar as the computation of the annual energy consumption in our optimization studies is concerned [11].

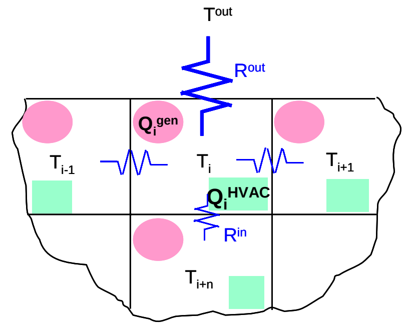


Figure 1: Typical Node

An important concern in implementing personalized control of thermally connected spaces is the potential increase of energy consumption resulting from internal energy transfer between adjacent warm and cool spaces, especially when the thermal resistance of interior walls is small or when the spaces are connected through large openings and corridors. We will show later in this paper that this concern can be mitigated by utilizing the optimization approach discussed here.

Individual Thermal Comfort

Indices of thermal comfort are well documented in the literature [1, 13-15]. The Fanger PPD-PMV curve [1] is widely used and is the basis of both the American and European standards [2,3]. These standards are designed to satisfy the thermal comfort needs of a majority of building occupants, not individual occupants. Little work has been published on individual thermal satisfaction. In this study, we have approached thermal comfort from the individual's point of view, which necessitated that we introduce a statistical model of individual thermal comfort that is inspired by Fanger's PPD, yet allows the preferred temperature and the temperature tolerance (comfort range) to vary from one individual to another. Each individual was assumed to have a preferred temperature, T_0, and a tolerance for departure from T_0, ΔT [16], which were related to the individual vote as given in Equation (2a) and Figure 2a. Both T_0 and ΔT were assumed to be normally distributed, with their means and standard deviations computed through a Monte-Carlo simulation such that the overall population PPD-PMV curve matches that in ASHRAE Standard 55-2004 [2]. Our analysis yielded mean values for T_0, and ΔT of 24°C, and 3.2°C, respectively, and standard deviations for T_0, and ΔT of 1.2°C, and 0.5°C, respectively [16].

A new measure – the degree of individual dissatisfaction (DID) – was introduced to describe individual dissatisfaction as a fuzzy concept rather than a binary one (satisfied or dissatisfied), because the binary approach implies an intuitively-implausible discontinuity in satisfaction. Our DID measure allows the concept of dissatisfaction to change gradually with the vote, ranging from 0 when the person is fully satisfied to 1 when the person is totally dissatisfied. Equation (2b) and expresses the relationship of the DID to individual vote, and Figure 2b depicts this relationship graphically. When Equations (2a) and (2b) are combined, a bucket-shaped curve centered on a different value of T_0 and with a different width (ΔT) can be generated for each member of a building population.

$$\mathrm{vote}(T) = \begin{cases} +3, & T > T_0 + 2\Delta T \\ -3, & T < T_0 - 2\Delta T\,, \\ 1.5\dfrac{T - T_0}{\Delta T}, & \textit{otherwise} \end{cases} \qquad (2a) \qquad \mathrm{DID}(vote) = \frac{1 + \tanh\left(2|vote| - 3\right)}{2}, \qquad (2b)$$

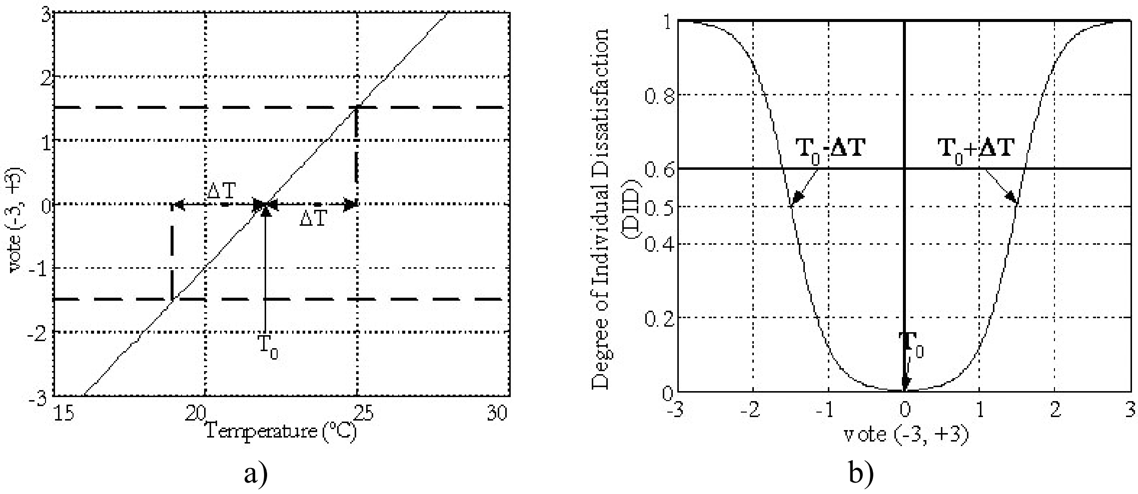


Figure 2. a) Individual Votes, b) Degree of Individual Dissatisfaction (DID).

Energy/Comfort Optimization

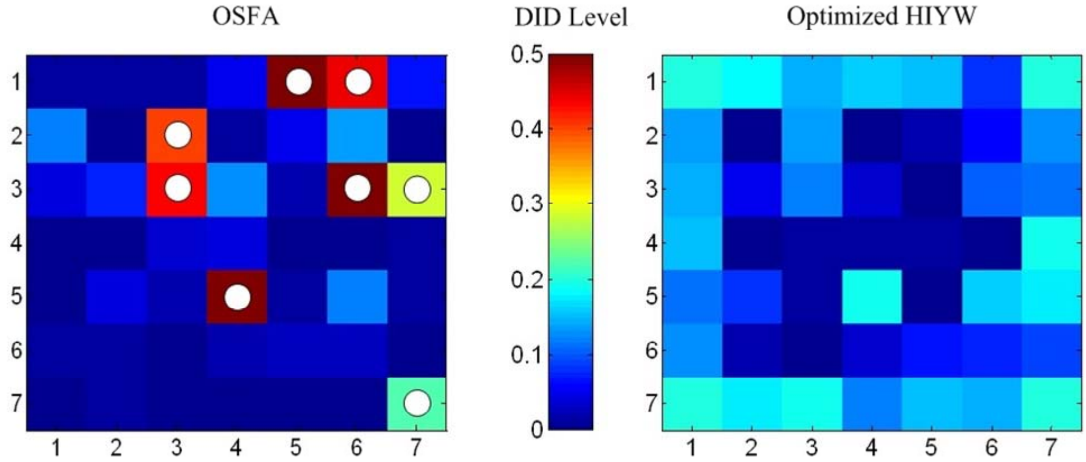
A variety of control strategies have been applied to building environmental control systems to lower energy consumption [18-22]. None has taken into account the effect on energy consumption of the thermal imbalances that would be created within the building under the HIYW strategy. In an earlier publication [16], we have shown that it is possible to achieve improved thermal satisfaction for individuals while saving energy through optimization and fuzzy approximation. In this study, a new approach has been established whereby a conventional optimization method such as gradient descent algorithm is used to find the optimum solution for a specific situation, and an intelligent system such as a fuzzy logic system or a neural network is used to generalize the solutions to a wider range of situations.

Our strategy is based on the observation that an individual's preferred temperature is not a precise value, but rather a range around a preferred neutral temperature. We have taken advantage of this fact to optimize the temperature settings of each occupant's personal space in a way that minimizes overall energy use. A gradient-descent algorithm has been utilized to improve the indoor environmental condition for each individual without increasing overall building energy consumption [16]. By varying the office temperatures of each individual in a given population within each individual's acceptable temperature tolerance range, energy consumption can been reduced, while the PPD of overall population building occupants has been kept at or below 10%. However, this overall PPD constraint doesn't guarantee that no individual is excessively dissatisfied with the environmental conditions. A hard constraint was imposed such that no individual had a DID greater than 20%. This constrained optimization approach is referred to here as "*Optimized HIYW*". As a baseline representing a well-managed but conventional approach, an optimized OSFA scheme was used whereby the temperature of the entire building was increased or decreased at each temperature bin within the population average temperature tolerance to achieve minimum energy consumption subject to a PPD constraint (10% in this study).

RESULTS

The computed performance of the optimized HIYW system has been compared with that of the optimized OSFA system for 15 USA cities and 10 populations. The energy consumption of a simulated 7x7 building has been minimized for each of 32 outside temperature bins, subject to the 10% PPD constraint for both OSFA and HIYW, and to the additional 20% DID constraint for HIYW. Sample results are displayed in Figure 3 for Chicago, IL, USA. The checkered plots are for the 1^{o}C temperature bin. It can be seen that while both the optimized OSFA and HIYW approaches provide a PPD of 10%, over 17% of the occupants in the energy-optimized OSFA case experience a DID in excess of 20% (higher than 50% for some occupants). On the other hand, the optimized HIYW approach eliminates any DID over 20% altogether, while reducing annual energy consumption by 8.5%.

We also assessed the robustness of the optimization approach when the thermal resistance of interior wall is significantly lowered, as it would be in the case of semi-open spaces (e.g., cubicles). The results are presented in Figure 4 (in box-and-whisker format) for 15 USA cities and one population distribution. The results are expressed as the ratio of annual energy consumption of HIYW systems and the OSFA systems for both the un-optimized and optimized cases. It can be seen that the higher inter-office energy fluxes (conductive and advective) associated with low interior thermal resistances lead to a marked increase in energy consumption ratio. It can also be seen that our optimization methodology leads to energy savings relative to OSFA over a wide range of thermal resistances from those for typical full interior walls (~1.0m^2K/W) to one-tenth that value. In all cases, the optimized HIYW system was able to reduce energy consumption relative to optimized OSFA system, while delivering greater satisfaction (no DID > 20%) than the optimized OSFA system. The main reason for this dramatic improvement is the reduction in the interoffice energy flow by slightly varying the individual preferred temperatures (T_0) within their temperature tolerances (ΔT).



↓ OSFA ↓	↑ The plot above is for the 1°C bin ↑	↓ HIYW ↓
0	← Annual Energy Savings (%) →	8.5%
17.4%	← Time Averaged DID over 20% →	0

Figure 3. Sample Results for a 7x7 1-Story Office Building in Chicago, IL, USA.

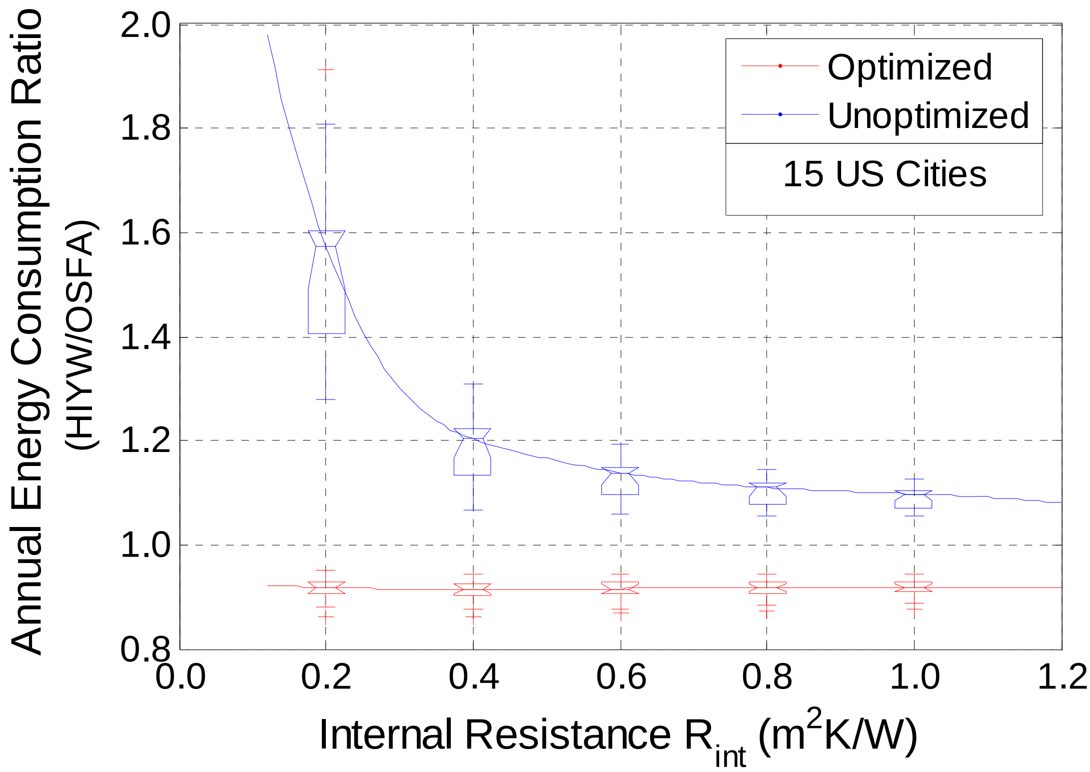


Figure 4. Effect of Internal Wall Thermal Resistance on Annual Energy Consumption.

DISCUSSION

The results of this investigation show that optimized HIYW distributed environmental control systems, which take advantage of the individual's temperature tolerances, can provide improved thermal comfort at lower energy consumption than optimized OSFA systems. By slightly varying individual temperature set-points within the individual's temperature tolerance using a nonlinear programming approach, energy consumption can be reduced while keeping each and every occupant within his/her thermal comfort range. This optimization method was also shown to be robust against higher interior wall conductance (thermal coupling between offices), which is characteristic of adjoining semi-open office cubicles.

The method assumes that both the individual's preferred temperature and temperature tolerances can be determined or inferred from his/her pattern of thermostat adjustment. While the existence of a tolerance range is well recognized for a population [23], more work needs to be done on the topic of individual thermal preferences and their persistence. Our ongoing work on optimal HIYW systems continues to explore intelligent control methodologies through which near-optimum solutions can be obtained with a considerably reduced set of inputs, and a higher potential for practical implementation [16]. Ongoing efforts are seeking to incorporate more realistic office building configurations, and improved energy and thermal comfort models that account for the advective coupling between adjacent, semi-open spaces [24]. Future efforts will assess the validity of the proposed HIYW control algorithms in realistic test-beds.

ACKNOWLEDGEMENT

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Occupants Have a False Idea of Comfortable Summer Season Temperatures

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SUMMARY

Thermal comfort studies and standards show that room temperatures should be higher in the warmer season than in the colder season. An interview survey with a sample size of 3,094 people was performed in Finland. The respondents were asked to state the Celsius values of room temperature they prefer in the winter and summer season in living room at home. The results show that people have a false idea of comfortable temperatures. 41% of the respondents think that room temperature should be lower in the summer season than in the winter season. 22% think that room temperature should be 19 °C or below in the summer season. Their principal idea behind this assumption is that the warm outdoor temperature should be compensated by cool room temperature. Only 15% of people have the correct idea: comfortable room temperature is higher in the summer season than in the winter season. The false idea may lead to unnecessary use of energy.

INTRODUCTION

When there is a lack of information, occupants develop own theories about how systems work and use these theories in controlling and adjusting their environments [1]. Kempton [2] presents a well-known example of that. He analysed folk theories for home heating control and found two common theories of how a thermostat works: a feedback theory and a valve theory. In the feedback theory thermostat senses room temperature, but in the valve theory, which is not understood, a thermostat dial is like a gas pedal and controls the amount of heat. The valve theory may lead users to adjust thermostats very often. In a study on use of room air conditioners [3], it was found that the operation of them was governed by multiple overlapping systems of belief and preferences concerning health, thermal comfort, folk theories about how air conditioners function, etc., in addition to economic factors. Diamond and Moezzi [4] have created a list of myths about people, energy and buildings.

Thermal comfort studies [5] and standards [6-7] show that room temperatures should be higher in the warmer season than in the colder season (line A in Fig. 1). There are two main reasons for that: the changes in clothing insulation related to the outdoor temperature [8] and adaptive relationship of comfortable room temperature with the mean monthly outdoor air temperature [8-11].

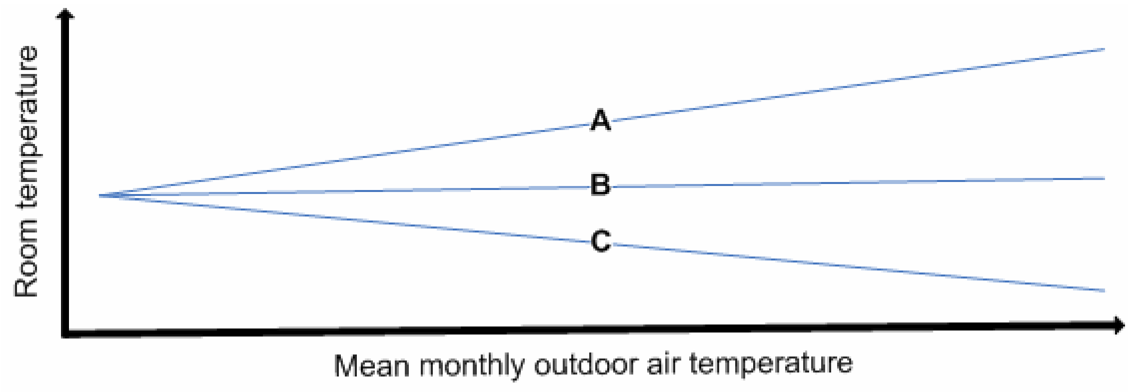

Figure 1. Three theories of how comfortable room temperature is related to mean monthly outdoor temperature, simplified.

Vastamäki et al. [12] found that office occupants have false mental models related to comfortable indoor temperature. About one-third of office occupants think that comfortable indoor temperature is lower in the summer season than in the winter season (line C in Fig. 1). More than one-third thinks that the comfortable temperature is not related to the season (line B). Neither of these views is supported by comfort criteria of the thermal comfort models. The study was realised in five European countries (Finland, Sweden, France, Italy and the Netherlands) with a total number of 735 respondents.

In the present study, ideas on comfortable indoor temperatures were studied among occupants in Finnish homes. In this paper, preferred room temperature refers to an occupant's idea of comfortable room temperature. Comfortable room temperature is what thermal comfort studies and standards see as comfortable.

METHODS

Inhabitants' ideas of comfortable indoor temperatures were studied using a quantitative interview survey with a nationally representative sample. The interview survey focused on thermal comfort and use of thermostats. In this paper, only the results concerning comfortable indoor temperatures are presented.

The interviews for the survey were carried out by telephone (computer assisted telephone interview, CATI). In the interviews, the respondents were asked to state the Celsius values of room temperature they prefer in their living room at home. The Celsius values were asked separately for the winter and summer season.

The target group of the study was the population of Finland. A random sample of the Finnish population aged between 15 and 74 was selected with quotas set according to gender, age and province. The total number of respondents was 3,094. A well-known Finnish data collection agency (Taloustutkimus Oy) was responsible for the practical realisation of the telephone interviews according to its quality system.

The climate in Finland is marked by cold winters and warm summers. The mean annual outdoor temperature varies between +6 °C in the southwest and –2 °C in the northernmost part of the country. The warmest month is typically July, with mean temperature between +14 and +18 °C in most parts of the country. Daily maximum temperatures can reach +30 °C in

July. The coldest months are January and February, with mean temperatures between –4 °C in the south and –15 °C in the north.

RESULTS

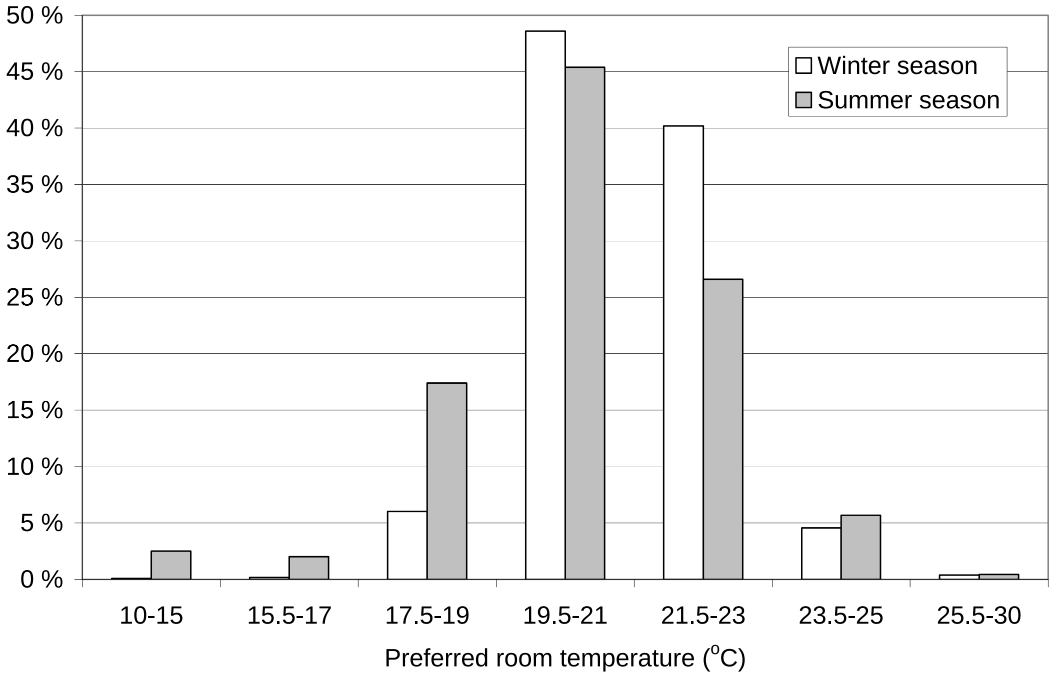

Figure 2. Preferred room temperatures in the winter and summer season in living room at home. N = 3064.

The respondents were asked to state the Celsius values of room temperature they prefer in the winter and summer season in living room. The results are shown in Fig. 2. 89% of the respondents say they prefer a temperature between 19.5 and 23 °C in the winter season. There is more deviation in the summer season temperatures: 22% think that room temperature should be 19 °C or below. A mean value for the preferred winter season temperature is 21.2 °C and for the preferred summer season temperature it is 20.5 °C.

41% of the respondents think that room temperature should be lower in the summer season than in the winter season (Fig. 3). Only 15% of people have the correct idea: comfortable room temperature is higher in the summer season than in the winter season.

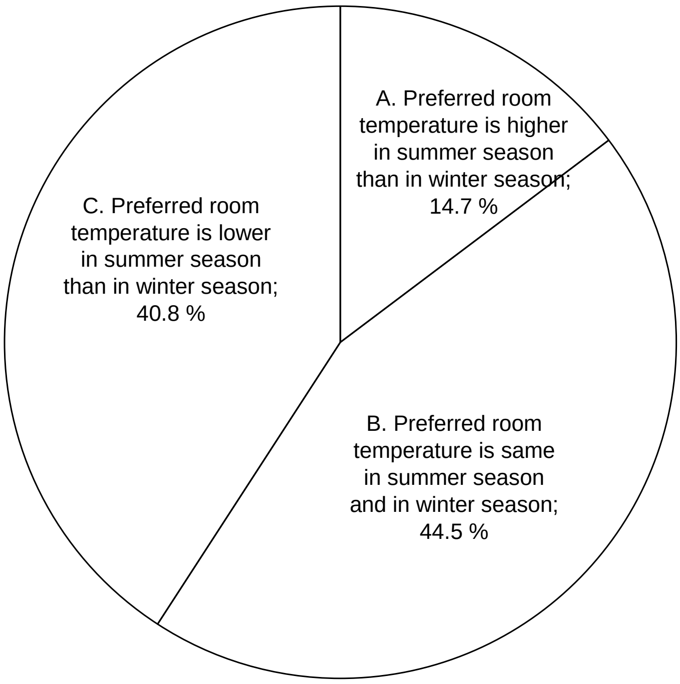

Figure 3. Influence of season on preferred room temperatures in living room at home.
N= 3000.

DISCUSSION

41% of the respondents think that room temperature should be lower in the summer season than in the winter season. Their idea behind this assumption is that the warm outdoor temperature should be compensated by cool room temperature: "If it is hot outside, I don't like hot inside". Occupants would feel very cold in the summer season, if room temperatures would be in the level most of them suggest.

The respondents of the present study were occupants in residential buildings in Finland. The results are consistent with the office occupant study [12] performed in five European countries. Occupants in offices do not typically have thermometers, but at homes they often do. However, they do not seem to have learned the Celsius values of comfortable temperatures.

The false idea regarding comfortable summer season temperatures may have energy implications. If occupants adjust temperature to, for example, 18 °C in a building with a mechanical cooling system, it leads to unnecessary use of energy. People should be educated about the values of comfortable temperatures in the summer season.

Occupants' ideas about comfortable room temperatures were studied as a part of usability research of temperature controls. Developers of heating and cooling controls should be aware of the misconceptions of occupants. Temperature controls should be designed to advise occupants on comfortable temperatures.

Future studies should investigate occupants' ideas about comfortable room temperatures in more detail. It would be valuable to know whether occupants think that the room temperature should vary according to different winter season outdoor temperatures (for example, –20 °C and 0 °C) since it has energy implications.

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Practical Investigation of Cool Chair in Warm Offices

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SUMMARY

This study investigated on a subject's thermal sensation and use of a "Cool Chair" in two actual warm offices. We developed a chair-mounted isothermal airflow generator called the Cool Chair that adjusts the local thermal environment by changing airflow velocity against the body surface. It was designed as a personal air-conditioning system. The Cool Chairs were used by workers in technical occupations in actual offices. The Cool Chair use rate, occupancy rate, and evaluations were recorded. The user's thermal sensation and comfort sensation were improved by utilization of the Cool Chair. Despite a uniform room temperature, the workers could select an airflow that provided individual comfort, which may result in a decrease in the worker complaints. Thus, the usefulness of the Cool Chair's thermal adjustability was confirmed.

INTRODUCTION

Individuals have different perceptions of thermal comfort. A comfortable temperature is dependent on factors such as a person's physical condition, clothes, and activity level. However, in a typical office environment, there are many workers in the same space, which maintains a uniform room temperature through air-conditioning. Therefore, allowing workers to adjust their local thermal environment is important. The chair-mounted isothermal airflow generator called the Cool Chair [1] is a personal air-conditioning system, which allows an individual to control the local thermal environment by adjusting the airflow (Figure 1). The conventional personal air-conditioning system uses a non-isothermal airflow. In contrast, the mechanism of the Cool Chair is simple, allowing the user to feel cool by using the isothermal airflow.

An adaptive model that requires a voluntary, passive action was proposed by R. J. de Dear *et al*. [2]. The adaptive model defines a person as a subject that transfers heat to the outside if uncomfortable. The adaptive model is different from a heat balance model. The adaptive model consists of three elements: behavioral adjustment, physiological adjustment, and psychological adjustment. Although a psychological adjustment is the most effective, it is difficult to evaluate quantitatively. A standard chair does not allow thermal adjustability that provides psychological adjustment. In contrast, the Cool Chair allows thermal adjustability. This simple device between the office environment and the worker contributes to thermal comfort, a feeling of control, and energy conservation [3].

METHODS

Cool Chair

The Cool Chair functions by drawing air into the seat and the backrest and blowing it out through the armrests (Figure 2) [4,5]. The amount and direction of airflow are easily adjusted. The maximum amount of airflow is 70 m^3/h. Airflow adjustment is accomplished using a controller on the right armrest. Power was supplied by a battery. The battery required several hours to charge, which is done by having the user plug the Cool Chair cord into an electrical outlet when out of the office.

Figure 1. External view of the Cool Chair.

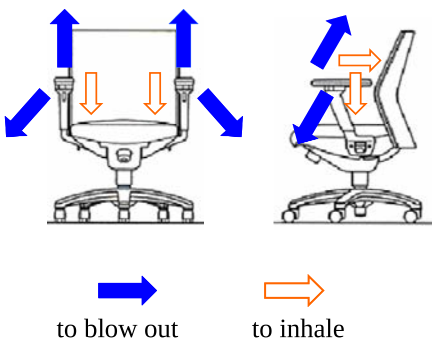

Figure 2. Function of the Cool Chair.

Summary of Practical Investigation

A summary of the study is shown in Table 1. Investigation was conducted at office K and office H in Japan (Figures 3 and 4). It was conducted in office K for 2 weeks from 31st August 2005 through 15th September. Eight subjects were divided into two groups (the A group was composed of 2 males and 2 females; the B group was composed of 1 male and 3 females). The workers in the A group used the Cool Chair during the first week and used a standard chair for the second week. The workers in the B group used the Cool Chair during the second week. All of the workers were employed in a technical field inside an office. From 27th September 2005, the investigation was conducted in the office H for one week. A male and a female, employed as a sales representative and an office worker, used the Cool Chair. Every morning and afternoon the workers supplied responses to a questionnaire regarding the type of clothing, thermal sensation, and comfort sensation.

Table 1. Summary of investigation.

Investigated Office	Office K		Office H
Place	Minato Ward, Tokyo		Osaka city, Osaka
Measurement Schedule	31st, Aug, - 7th, Sep	8th, Sep - 15th, Sep	27th, Sep - 4th, Oct
Cool Chair User	A,B,C,D	E,F,G,H	I,J
Standard Chair User	E,F,G,H	A,B,C,D	
Survey Item	Seat occupancy rate Thermal sensation vote, Comfort sensation vote Questionnaire Room temperature		

Figure 3. Office K involved in the study.

Figure 4. Office H involved in the study.

To determine airflow, the terminal voltage of the blower was recorded with a small voltage logger. In addition, the temperature of the weight-bearing surface was recorded with a small temperature logger to determine the times when the person was seated. Mean room temperature fluctuation during the investigation period is shown in Figure 5 (temperature was measured at a height of 1100 mm). Room temperature during working time was nearly 27°C.

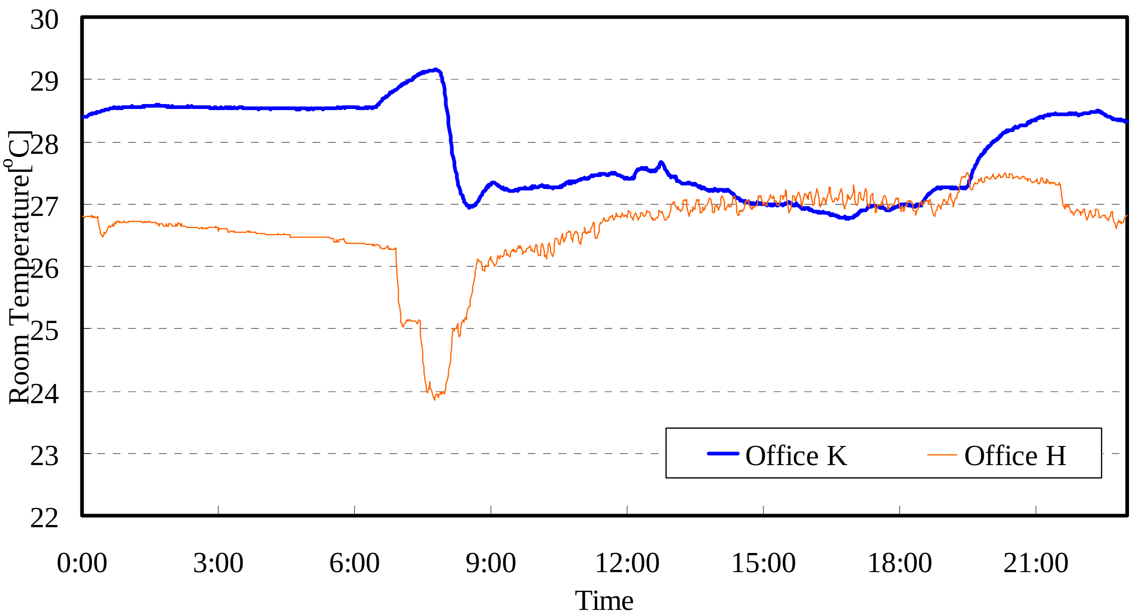

Figure 5. Ambient temperature.

RESULTS

Usage condition of Cool Chair

The airflow rate was not constant, because a worker repeatedly sits down and stands up during the day (Figure 6). Room temperature during the day was approximately 27°C and the amount of airflow selected by the user was 20 m^3/h. When the air-conditioning system stopped outside of office hours, the room temperature increased and the amount of airflow selected was 60 m^3/h.

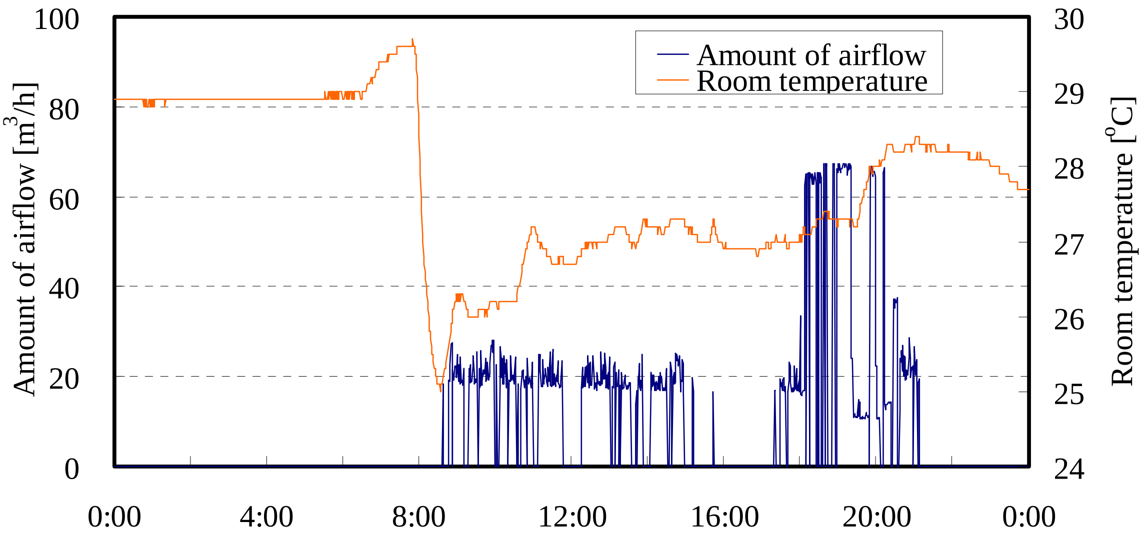

Figure 6. Usage of Cool Chair (an example).

Figure 7 shows the relation between Cool Chair use and amount of airflow. The respective Cool Chair users are shown on day 1. Females used a greater airflow for a longer time compared to the males. Figure 8 shows where the airflow was directed. Most of the time the workers directed the airflow toward the arms, followed by the side of the body, neck, and chest. The circumstances under which the workers used the Cool Chair are shown in Figure 9. The Cool Chair was used mainly when the activity level increased. In addition, the subject used the Cool Chair when he or she felt that the "air-conditioning system was not effective."

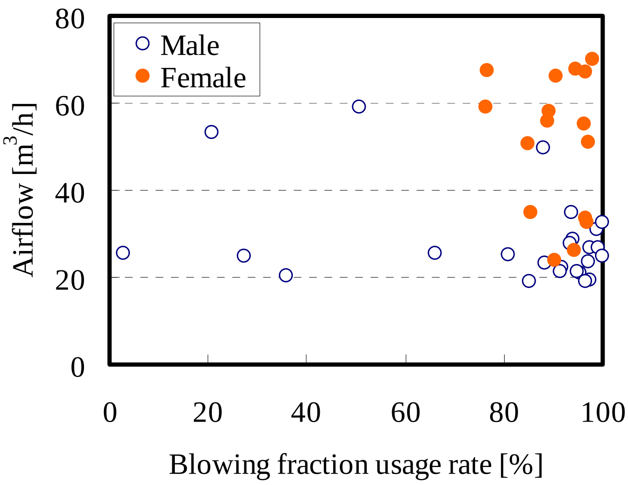

Figure 7. Usage/Airflow.

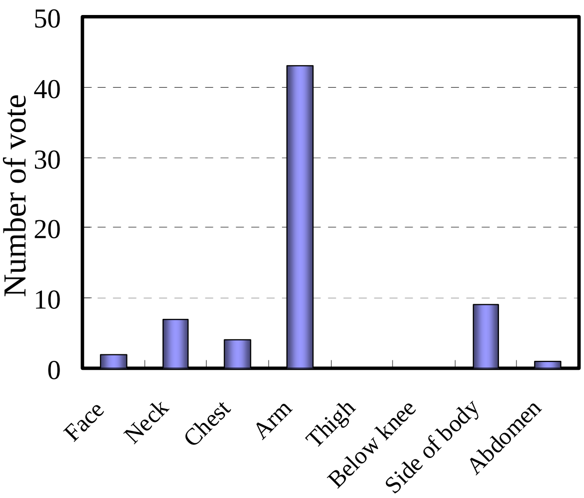

Figure 8. Applied airflow.

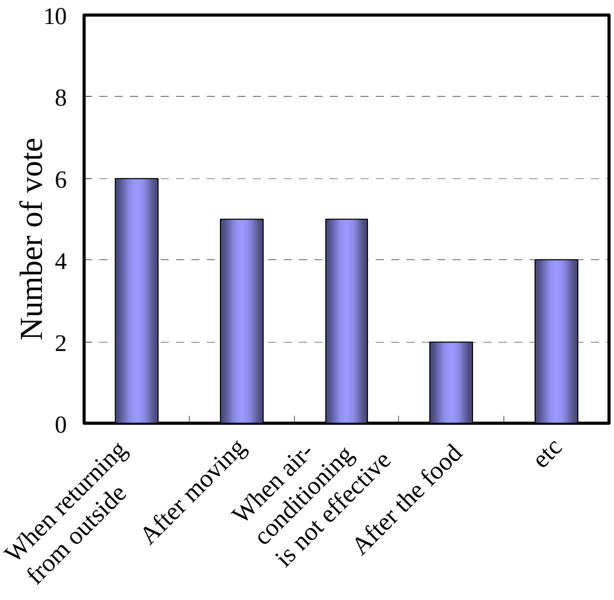

Figure 9. Circumstances when the Cool Chair was used.

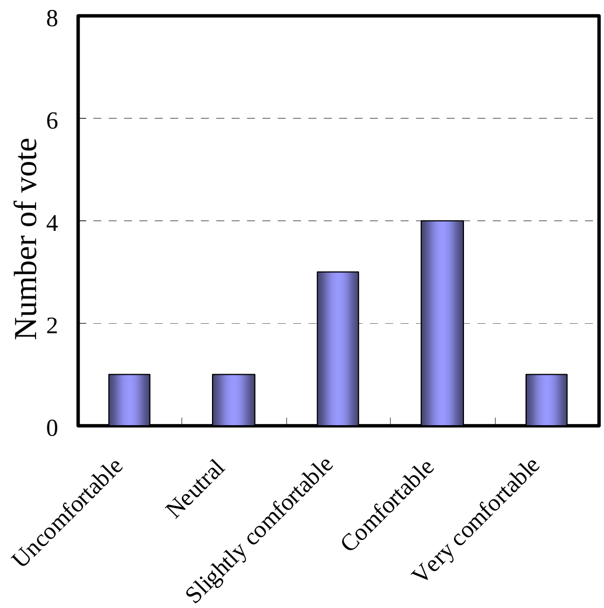

Figure 10. Impressions of Cool Chair.

Thermal Sensation Vote and Comfort Sensation Vote

The workers' impression of the Cool Chair is shown in Figure 10. Most workers reported that the Cool Chair was "Comfortable." The mean values for males and females for TSV and CSV in office K are shown in Figures 11 and 12. The TSV shifted from warm to neutral, and the CSV shifted from uncomfortable to comfortable through the use of the Cool Chair. The cooling effect of the isothermal airflow despite a relatively warm indoor temperature was confirmed by these results.

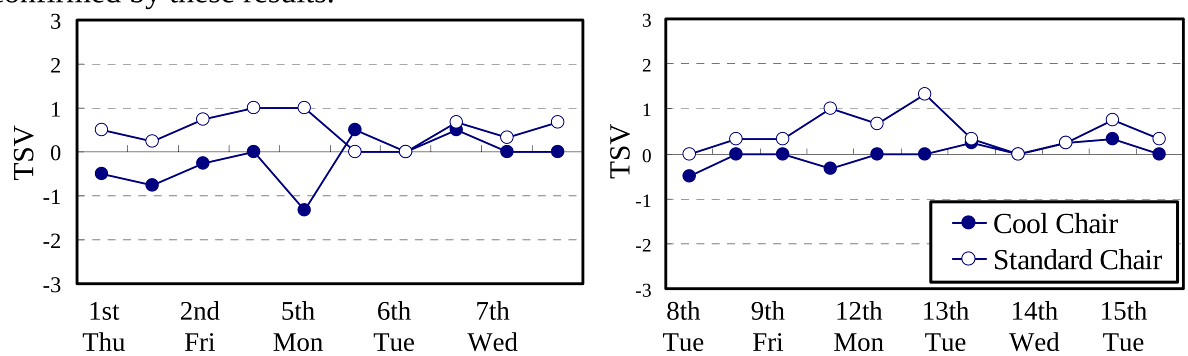

Figure 11. Comparison of TSV values.

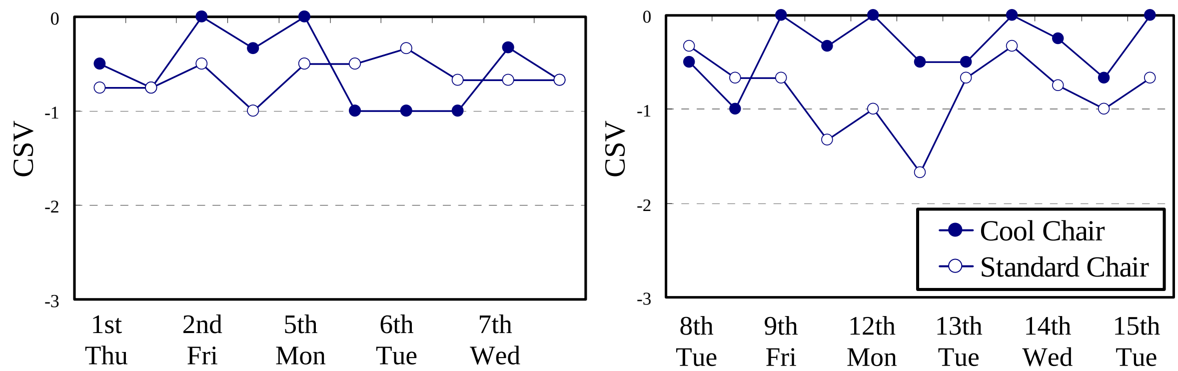

Figure 12. Comparison of CSV values.

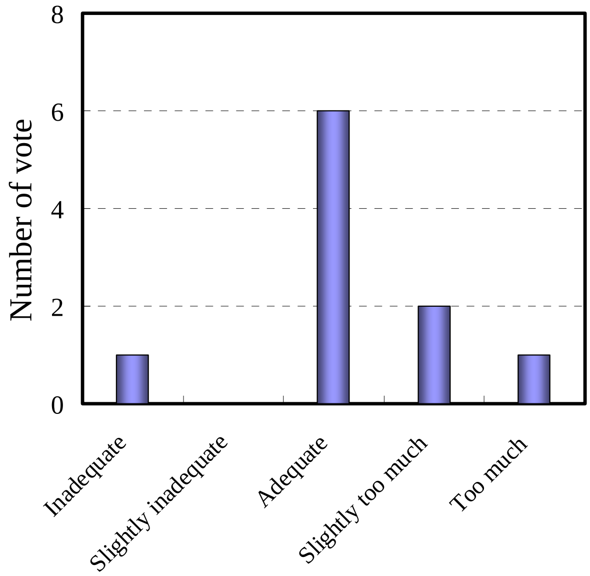

Figure 13. Impressions of airflow rate.

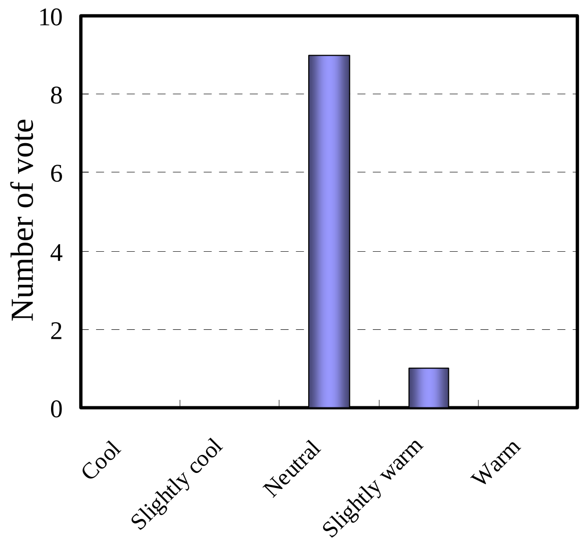

Figure 14. Impressions of airflow temperature.

Evaluation of airflow

Figure 13 shows the users' impressions of the Cool Chair's maximum airflow rate. Results indicated most users were satisfied with the airflow of the Cool Chair. Figure 14 shows user's impressions of the airflow temperature. Results indicated that the user was cooled sufficiently by the isothermal airflow.

DISCUSSION

The performance of the Cool Chairs was tested under actual office conditions. Airflow rate and temperature were sufficient in actual practice. The amount and direction of airflow were controlled by the user based on thermal comfort. Results indicated that the thermal adjustability of the Cool Chair increases user comfort. This thermal adjustable device contributes to worker thermal comfort and feeling of control, and energy conservation.

ACKNOWLEGEMENT

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Subjective Thermal Comfort in the Environment with Spot Cooling System

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SUMMARY

Subjective experiments were conducted in a climate chamber to evaluate subjective thermal comfort in the environment with spot cooling system, simulating a big factory. Five conditions combined supply air temperature and air volume were set for the first test. The condition without spot cooling system was also examined. Two types of diffusers were developed in the second test. Seven males were exposed for 90 minute under the conditions with and without spot cooling. Subjective metabolic rate were assumed to be 1.8 met. Subjects voted their thermal sensation, air velocity sensation, comfort sensation in the experiments every 30 minute.
Thermogram showed that spot cooling system without diffusers cooled down only upper body of subjects, where they felt the draght. On the other hand, it was possible to cool down their whole body using the diffuser. Subjects preferred the condition with the diffuser, where they felt “thermally neutral”.

INTRODUCTION

In large space such as a factory, it is not feasible to control the air in the entire space with HVAC system without huge energy consumption. The air-conditioning in large space usually targets only the occupied zone. It is important that the occupied zone air-conditioning satisfies a comfortable feeling of the human body, and decrease the air-conditioning load as much as possible. Spot cooling system used at the factory is one of the occupied zone air-conditioning, and have potential for energy conservation. The workers' physiological and subjective responses to such environments are different from those to typical uniform thermal environments, and are far less understood for Japanese.
In this research, subjective experiments were conducted to evaluate thermal comfort of Japanese subjects in the environment with spot cooling system, simulating a big factory. Different combinations of the jet air temperature and volume, and the shape of diffusers were examined.

METHODS

Experimental Design

Two kinds of subjective experiments were conducted in a climate chamber to evaluate subjective comfort in the environment with spot cooling system in 2005. Supply air temperature, air volume and shape of diffuser were set for the purpose of evaluating the human comfort under the spot cooling. In the first test, supply air temperature and volume from spot cooling system were controlled. The shape of diffusers was varied in the second test. Seven college-aged males who were in good health were exposed to the experiments. All subjects were volunteers who were paid for participating in the experiment. Considering their circadian rhythms, all subjects took part in the experiments at the same time of day.

Experimental Conditions

Table 1 lists experimental conditions.

In the first experiment, three supply air temperature, 19°C, 25°C and 31 °C, with 400m^3/h of supply air volume were set. The condition with small supply air of 200m^3/h at 19^{o}C were also examined. Additionally, the condition without spot cooling system was also analyzed.

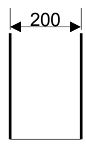
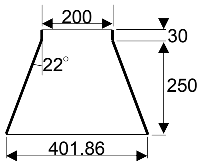
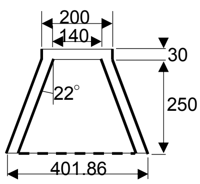
In the second experiment, two types of diffusers, type I and II, were developed. The condition without diffuser was compared with the condition with 2 types of diffuser. Supply air temperature and air volume was set at 19^{o}C and 400m^3/h for all condition. Table 2 shows 3 kinds of diffusers.

For all conditions in both the first and second test, outlet of spot cooling system equipped 3.0 m above the floor. The spot cooling system had a nozzle with an outlet diameter of 0.2m. The temperature where is not influenced by spot cooling system (ambient temperature) were kept at 31 ^{o}C simulating a factory [1]. In the experiments, subjects wore hat, work wear (long sleeve), shirt, shoes, socks.

Table 1. Experimental condition

	First experiment					Second experiment		
Condition	19_200	19_400	25_400	31_400	NONE	without diffuser	Type I	Type II
Supply air temperature [°C]	19.0	19.0	25.0	31.0	-	19.0		
Supply air volume [m^3/h]	200	400	400	400	0	400		
Ambient temperature [°C]	31.0							
Outlet altitude[m]	3m above floor							
Outlet diameter[mm]	200φ							
Diffuser	Nozzle						Type I	Type II
Clothing	hat, work wear (long sleeve), shirt, shoes, socks							

Table 2. Three kinds of diffusers

Without diffuser	Type I	Type II
 	 	 

Experimental Procedure

Figure 1 illustrates the experimental procedure.

Subjects were exposed for 90 minutes under the conditions with and without spot cooling. One test consisted of 3 sessions. During the 30-minute session, subjects performed walking up and down the steps [2] in the ambient area, light work which is putting machine screws in holes and clamping them with nuts in a standing position[3] with spot cooling for 20 minutes. Subjective metabolic rate was assumed to be 1.8 met [4]. Subjects rated at the beginning of the exposure time and after each task during the session. In the second experiment, the different type of diffusers was installed for each session.

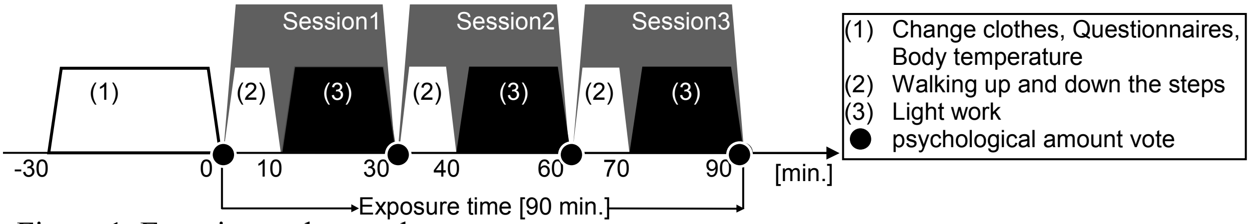


Figure 1. Experimental procedure

Measurement

The effect of supply air condition and air flow pattern on human comfort was evaluated by the result of environmental measurements (the vertical air temperature distribution, the velocity distribution, vector diagram of the supply air velocity analyzed by PIV), physiological response (the skin temperature, relative humidity and air temperature in clothing, the thermogram), and the subjective psychological measurements.

The thermogram is a image visualized the surface temperature of the object. In this experiment, the surface temperature distribution of the subject working under the spot cooling system was analyzed with thermogram.

PIV (Particle Image Velocimetry) is a kind of the speed measurement method of the fluid such as water and air. In second experiment, the vector analysis on the air flow with diffuser was done by using PIV analysis software (DIPP-Flow) from the image taken with high speed camera.

Subjects rated their thermal sensation, air velocity sensation, comfort sensation every 30 minute on the questionnaire as shown in Figure 2. The scales were given as visual analogue scales. Subjects were allowed to rate their sensation either just on the number or between the numbers on the scales. In the second experiment, subjects also evaluated effect of total comfort by difference of diffuser.

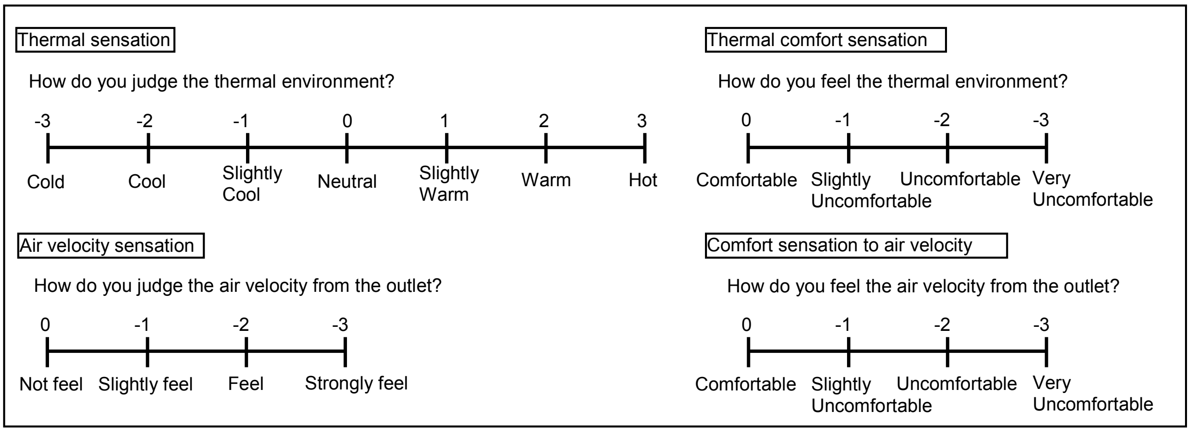


Figure 2. A part of rating scales

Statistical Analysis

Data obtained in the experiments were analysed with Non-parametric statistical analysis method [5]. The Wilcoxon Matched-Pairs Signed Ranks test was administered between each condition. In the first experiment, supply air temperature effect was discussed in the pair-wise comparison between 19_400 and 19_200. According to the comparison between 19_400, 25_400, and 31_400, supply air volume effect was reported. P-values presented in the figures indicate the level of significance.

RESULTS AND DISCUSSION
The First Experiment Result
Thermal Environment

Vertical air temperature distribution: Vertical air temperature distribution in occupied zone was less than 3 °C under the all conditions.

Surface temperature: The surface temperature distribution recorded with thermogram is presented in Figure3. It was found that spot cooling system without diffusers cooled down upper body of subjects such as the head, necks, and the backs. This tendency was remarkably observed under the condition at low supply air temperature or with large air volume.

Subjective vote: The general thermal sensation vote and general comfort sensation vote were displayed in Figure 4. In two conditions of 19°C in blow temperature, subjects reported significantly cooler at 19_400 than at 19_200 ($p<0.01$), and more comfortable at 19_400 ($p<0.01$). Among three conditions of 400m^3/h of supply air volume, people rated significantly cooler at 19°C and 25°C than at 31°C ($p<0.01$). Comfort sensation vote under the condition with 19°C of jet air was significantly greater than with 31°C ($p<0.04$). Thermal sensation vote was highest ($p<0.01$) and most uncomfortable ($p<0.04$) under the condition without diffuser. Subjective comfort sensation was the greatest at 19_400. It is found that subjective thermal sensation vote was scattered near "slightly cool" and subjects felt comfortable under the condition of low air temperature and large air volume.

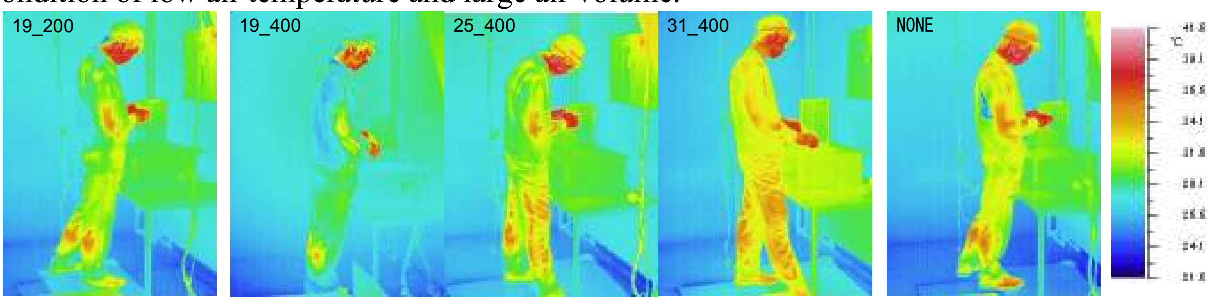
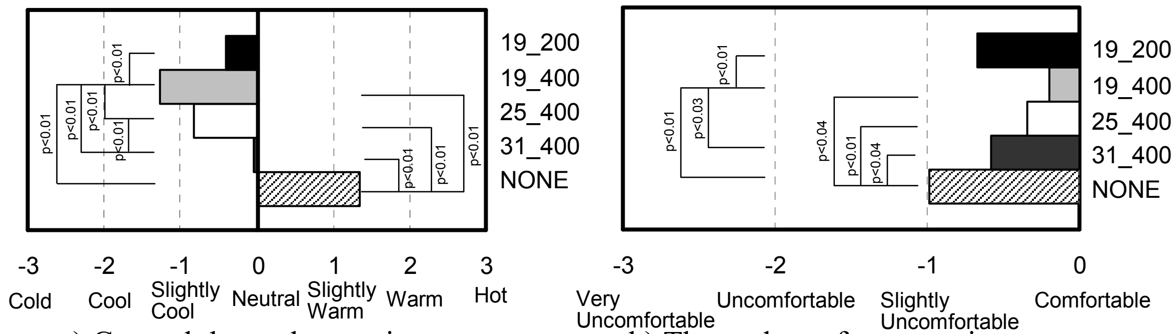


Figure3. Surface temperature of subject's body (First experiment)

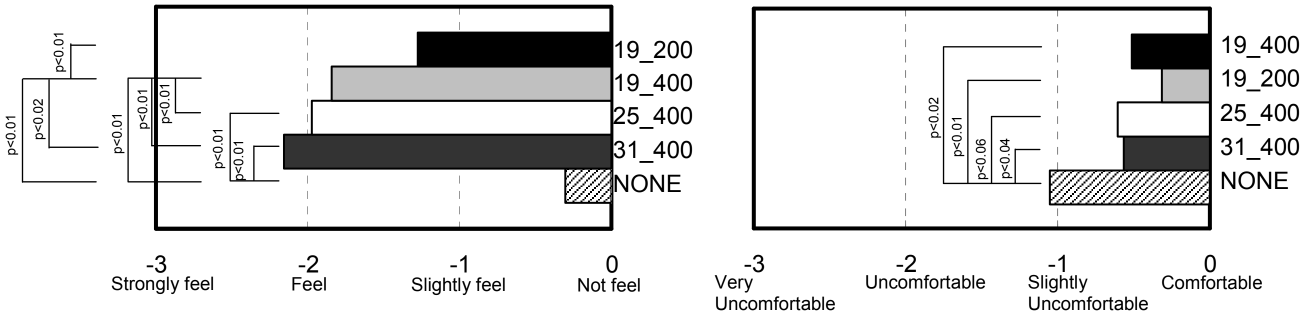


a) General thermal sensation vote b) Thermal comfort sensation vote

Figure 4. Subjective thermal comfort

Air Flow

Subjective vote: The general air velocity sensation vote and comfort sensation vote to air velocity were shown in Figure 5. Pair-wise comparison between 2 conditions with 19°C of supply air temperature, significantly greater air velocity sensation was observed in 19_400 air ($p<0.01$). Among three conditions of 400m^3/h of supply air volume, subjects reported significantly higher general air velocity sensation at 31_400 than at 19_400 ($p<0.02$). There was no significant difference in the comfort sensation to air velocity between any pair of conditions with spot cooling system by the Wilcoxon Matched-Pairs Signed Ranks test. Subjects rated significantly more uncomfortable under the condition without spot cooling system than at 19_400, 19_200 and 31_400 ($p<0.04$), and tend to feel more uncomfortable than at 25_400($p<0.06$). It is found that the comfort sensation to air velocity was improved by setting up the spot cooling system.



a) General air velocity sensation vote b) comfort sensation vote to air velocity

Figure 5. Subjective comfort sensation vote to air velocity

The Second Experiment Result

Thermal environment

Surface temperature: The surface temperature distribution recorded by thermogram is shown in Figure6. It showed that spot cooling system without diffusers cooled down upper body of subjects, while it was possible to cool down their whole body using the diffusers. It was remarkable in the condition using the diffuser type II.

Subjective vote: The general thermal sensation vote and thermal comfort sensation vote were presented in Figure 7. Thermal sensation votes were scattered near “slightly cool” and comfort sensation vote “comfort” for all conditions in the second test. Especially, subjects tended to feel thermally neutral and more comfortable under the condition using the diffuser type II compared with the condition without diffuser. It is found that subject felt comfortable when they perceived the thermal environment to be neutral. It is concluded that the thermal comfort sensation improved by cooling down their whole body using diffusers.

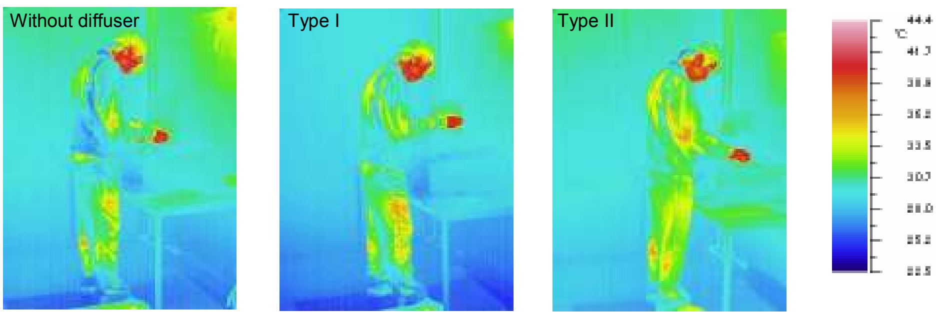
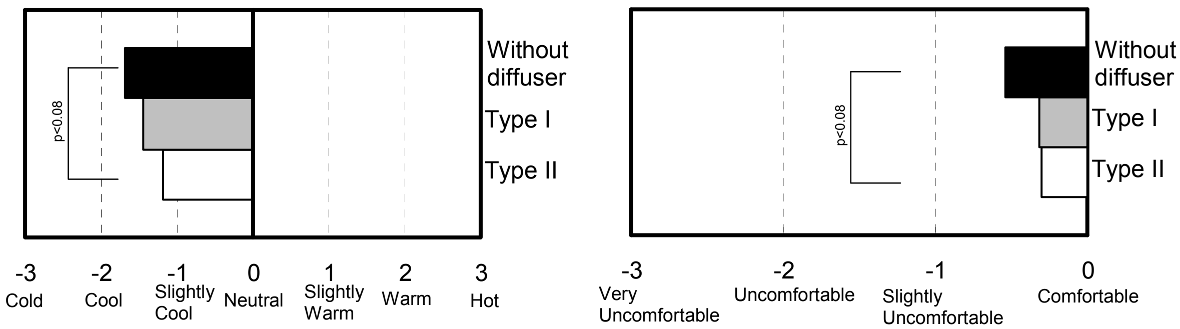


Figure 6. Surface Temperature(second experiment)



a) General thermal sensation vote b) Thermal comfort sensation vote

Figure 7. Subjective thermal comfort

Air Flow

Supply air velocity: The vector diagram of the supply air velocity measured with PIV from the diffuser is displays in Figure8. Under the condition without diffuser, little air flow diffusion was seen, and flowed toward the subjective position. Air velocity right after nozzle was about 4.1m/s. Supply air was diffused due to the diffuser, while the air velocity which reached directly subject got smaller and air flow to the subject was weakened. Especially, it was remarkable in the condition with diffuser typeII. Air velocity of the diffusion was about 3.5m/s, and that to the direction of the subject was 0.5m/s when the diffuser typeII was installed. As the result of the velocity distribution measurements, air velocity 1.7m above the floor indicated up to 1.5m/s under the condition without diffuser and with diffuser type I. In case of using the diffuser type II, air velocity on the same point was measures to be 0.24m/s.

Subjective vote: Figure9 shows the general air velocity sensation vote and comfort sensation vote to air velocity. Subjects reported significantly lower air velocity sensation under the condition with diffuser typeII than any other conditions ($p<0.03$). In the first experiment, there was no significant difference in the vote of comfort sensation to air velocity between any pair of conditions with spot cooling system. However, subjects reported significantly more comfortable in case of using the diffuser typeI and typeII than without diffuser ($p<0.05$) on the second experiment.

Subject reported the air velocity from the outlet was comfortable in case that the velocity sensation was small. It was found that people could feel great comfort due to cooling down their whole body even with very small velocity.

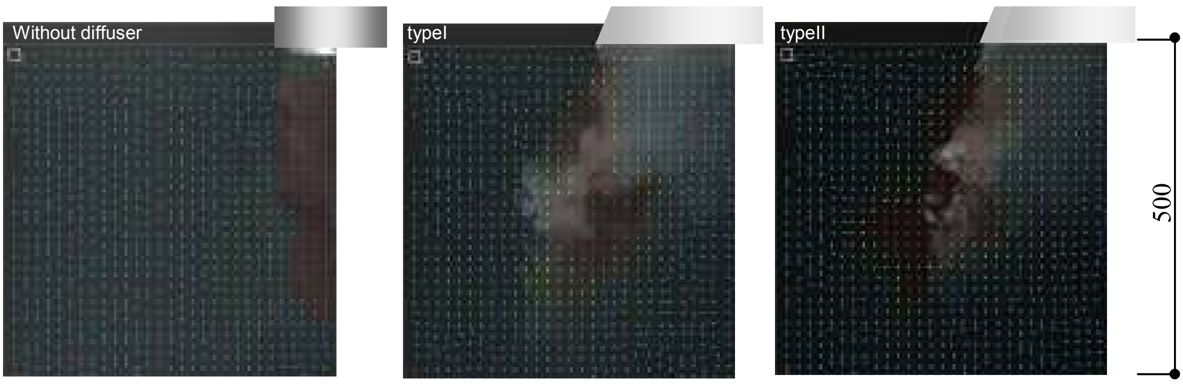
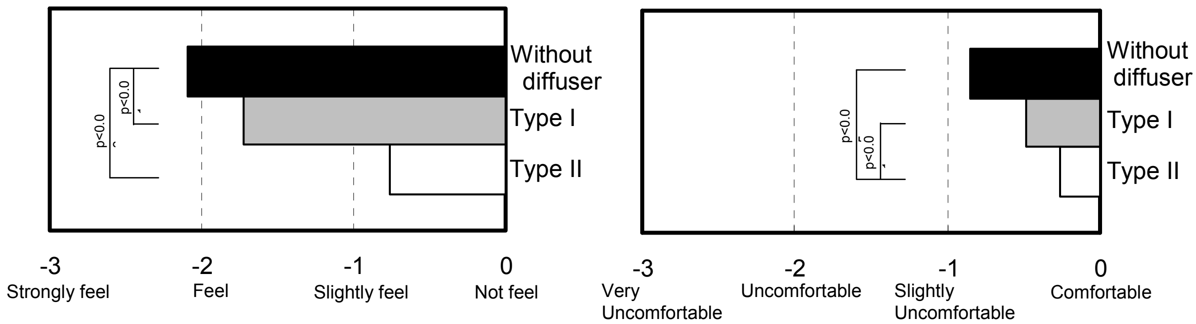


Figure 8. vector diagram of the supply air velocity



a) General air velocity sensation vote b) comfort sensation vote to air velocity

Figure 9. Subjective comfort sensation vote to air velocity

Evaluation of Diffuser

Subjects ranked three type of diffuser in the terms of air supply temperature and velocity. The evaluation of the diffuser typeI and typeII was higher than that of no diffuser as shown in Figure10. Especially, the majority people voted the diffuser typeII as 1st place. Though, there were people who rated the diffuser typeII as 3rd place, too. This is demonstrated that there was an individual variation in subjects' preference of diffuser.

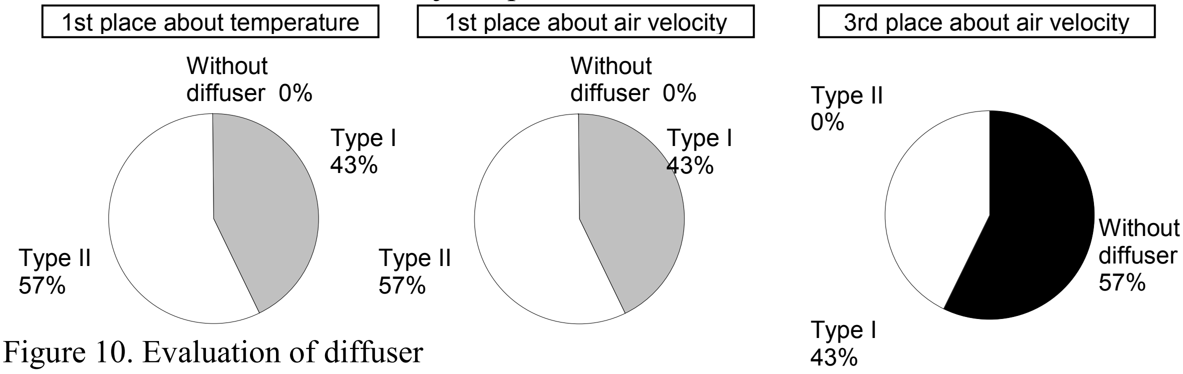


Figure 10. Evaluation of diffuser

CONCLUSIONS

In order to evaluate subjective comfort in the environment with spot cooling system, simulating a big factory, two subjective experiments were conducted in the climate chamber. The first experiments were carried out to evaluate the air supply conditions. Subjects were exposed to 5 conditions, 19_200, 19_400, 25_400, 31_400, none.
Spot cooling system without diffusers cooled down upper body of subjects. It was remarkable in the condition such as low supply air temperature and large air volume. It is found that subjective thermal sensation vote was scattered near “slightly cool” and subjects felt comfortable under the condition of low air temperature and large air volume. In this experimental condition, when supply air temperature rose and the volume increased, subjects rated “feel” on air velocity sensation vote. But there was no significant difference in the vote of comfort sensation to air velocity between any pair of conditions with spot cooling system

In the second experiment, subjects evaluate the air supply conditions using the diffusers. three experimental conditions were set, diffuser typeI, typeII and without diffuser.
It was possible to cool down their whole body using the diffusers. It was contemplated that the thermal comfort sensation improved by cooling down their whole body. Subject reported the air velocity from the outlet was comfortable in case that the velocity sensation was small. It was found that people could feel great comfort due to cooling down their whole body even with very small velocity. The majority people rated the diffuser type II as 1st place, though there was an individual variation in subjects' preference of diffuser.

ACKNOWLEDGEMENT

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The Use of Wireless Data Communication and Body Sensing Devices to Evaluate Occupants' Comfort in Buildings

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SUMMARY

Physiological parameters measured by an embedded body sensor system were demonstrated to respond to changes of the air temperature in an office environment. The thermal parameters were monitored with the use of a wireless sensor system that made possible to turn any existing room into a field laboratory. Two human subjects were monitored over daily activities and at various steady-state thermal conditions when the air temperature of the room was altered from 22-23°C to 25-28°C. The subjects indicated their thermal feeling on questionnaires. The measured skin temperature was distributed close to the calculated mean skin temperature corresponding to the given activity level. The variation of Galvanic Skin Response (GSR) reflected the evaporative heat loss through the body surfaces and indicated whether sweating occurred on the subjects. Further investigations are needed to fully evaluate the influence of thermal and other factors on the output given by the investigated body sensor system.

INTRODUCTION

The increasing miniaturization of radio frequency devices and micro electro-mechanical systems, as well as the advances in wireless technologies, has generated a great deal of research interest in the area of wireless sensor networks. These technologies have enabled the development of low-cost, low-power, multi-functional sensor nodes that are small in size and allow untethered communication over short distances. Such systems may be useful for close monitoring and adaptive control of building equipment, materials performance and environmental conditions including temperature, air flow, indoor air quality, lighting, sound and the stated well-being of the occupant in relation to their surroundings.
Wireless sensor networks will also enable a proper cohesive data management system to be organised for buildings. It will help to understand the strategies required for operating the building so that energy savings are achieved; and to obtain healthier environments and closer relationship between occupants and buildings. Wireless sensor networks will also contribute to realize a leaner and more effective design, construction, operation and facilities management regime. For example we should be able to understand more deeply the patterns of building use and also enable prediction of failure conditions thus informing maintenance regimes.

Apart from gathering information about the building environment and performance it is equally important for the facility management to collect direct feedback from the building occupants about their perception of the indoor environment and well being. Previously this was done occasionally by subjective pen and paper questionnaires that provided data in an awkward and expensive way for further processing. Today's technologies give us the opportunity to gather such information instantaneously and store them in databases [1] or even to use these data for direct control of the existing building services. Proposal of such device was already made in 1990 [2] and the practical development now becomes reality [3], [4].

Beside of subjective questioning more and more efforts are made to develop methods and devices that are able to measure physiological parameters in humans that are sensitive to environmental factors [5]. A recently developed wearable embedded body sensor system produced by Body Media enables monitoring not only personal energy expenditure and various body movements of people, but also skin and near body air temperature, GSR and convective heath loss [6], [7]. A number of these parameters given by this sensor can be used to detect physiological changes in the body or can be related to the thermal comfort of people.

The present investigation is focused to explore the wireless sensor technology to evaluate the thermal environment in an existing office and show physiological changes in the occupants exposed to thermal stress with the use of the embedded body sensor.

METHODS

One day long monitoring with the embedded body sensor

A male subject was monitored for a full day during his normal daily activities with the embedded body sensor system (see description below) to show variations in the output values of the sensor if the surrounding environment around the human body or the person's activity level changes. During this one day long monitoring no environmental parameters were recorded given the difficulties of carrying the environmental sensors. Instead, we can assume the temperatures in the middle of February when most buildings are heated (~23°C) and the daily outdoor temperature is about 2-7°C.

Output of the embedded body sensor in a controlled office environment

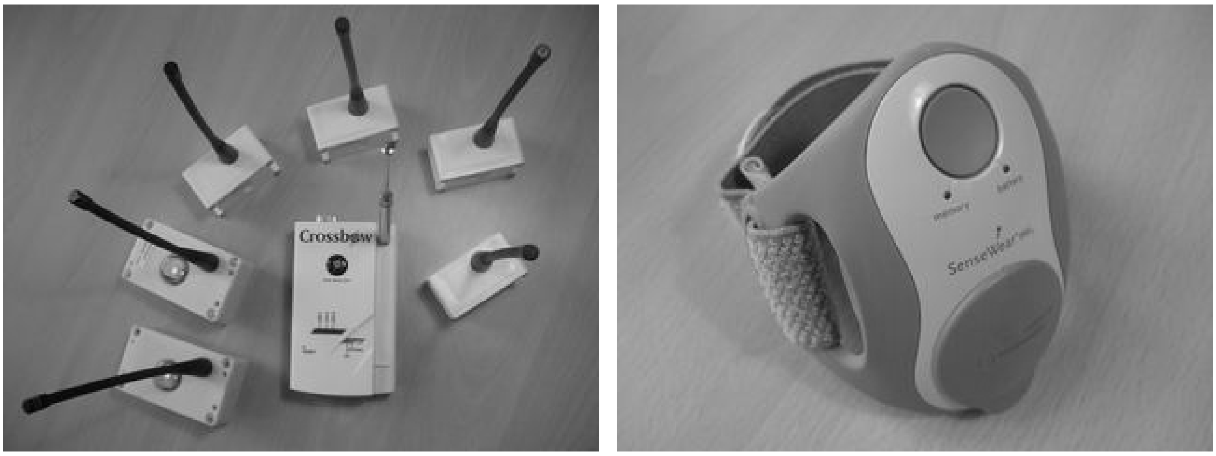
Three experiments (Table 1) were designed to demonstrate how the embedded body sensor system could be used in indoor environments for measuring body parameters that may affect building occupiers' well being. For this matter change in the value of the recorded factors were examined at different room temperatures. On four separate days three experiments were carried out in controlled office environments. In these experiments the same two – a male and a female – subjects were monitored during their office work. The experimental conditions are summarized in Table 1. The temperature was controlled using a small HVAC system installed to the rooms. The fresh air supply rate in all of the presented conditions was higher than 15 L/s per person that provided a good indoor air quality during exposures. The environmental parameters including air temperature, relative humidity, globe temperature and air velocity were recorded using a wireless sensor system (Figure 1.a) and with some additional sensors. The wireless sensors were distributed in the close proximity of the subjects to monitor the vertical temperature distribution at 0.1; 0.6 and 1.1m heights. The globe temperature sensor was placed at 0.6m from the floor level.

Table 1. Experimental conditions

	Experiment 1		Experiment 2			Experiment 3	
	Morning	Afternoon	Morning	Afternoon	Evening	Day 1	Day 2
Room temperature	High	Low	Low	High	High	Low	High
Room Volume [m^3]	150		30			30	

Instrumentation

The wireless sensor system (Figure 1.a) was used to measure the temperature and the relative humidity. The temperature sensor can measure in the range of –40 to +123°C with the resolution of 0.01°C and with ±0.5°C accuracy. While the relative humidity sensor measures in the range of 0 to 100% with the resolution of 0.03% and with ±3.5% accuracy. Some of the sensors in this system are also able to measure light and pressure but these were not used in the present investigation. The additional globe temperature sensor works with 0.01°C resolution and with ±0.3°C accuracy and the air velocity sensor has 0.05 ms^{-1} accuracy.



a) b)

Figure 1. a) The wireless sensor system; b) The wearable, compact, embedded body sensor system

The embedded body sensor system (Figure 1.b) is positioned on the rear side of the right upper arm and has several built in sensors including skin temperature sensor, near-body ambient temperature sensor, a two axis accelerometer, GSR sensor and heat flux sensor. For the evaluation of GSR the embedded body monitoring device uses two electrodes in contact with the skin that close an electric circuit and the current flow across the skin is measured. The obtained values represent conductivity between two points of the skin. The developer conducted studies comparing the GSR sensor in our body sensor system to the more traditional fingertip GSR sensor. The results demonstrated that the GSR sensor in our case provides a linear analogy to digital values for conductance but that it is significantly less sensitive than a GSR placed on the finger or palm [8].
The heat flux sensor picks up convective heat flow at the skin level through thermocouples and other thermally conductive elements. For further details of the embedded body sensor system see [8].

Subjective questionnaires

During Experiment 1-3 thermal questionnaires were filled in by both subjects several times to obtain subjective opinion about the indoor thermal environment. Therefore seven point thermal sensation scale [9] and sweating on a six point intensity scale were asked to be indicated.

Data analysis

To obtain representative measure of the thermal environment the operative temperature was calculated based on the air temperature and the mean radiant temperature provided by the globe temperature sensor [9]. Mean skin temperature, conductive heat transfer through the clothing (K/A_{Du}) and PMV (Predicted Mean Vote) was determined according to the thermal comfort model described by Fanger [10].
To see correlation between the various parameters regression analysis was carried out, where the resulting p value is used as an overall F-test of the relationship between the dependent variable and a set of independent variables [11].

RESULTS

One day long monitoring with the embedded body sensor

Figure 2 shows the monitoring results of GSR and the skin temperature of a male subject determined by the body sensors throughout a full day at various activity levels. As a comparison for a shorter time period the GSR of a female subject is also represented. She was at similar activity levels and exposed to the same environmental conditions as the male subject.

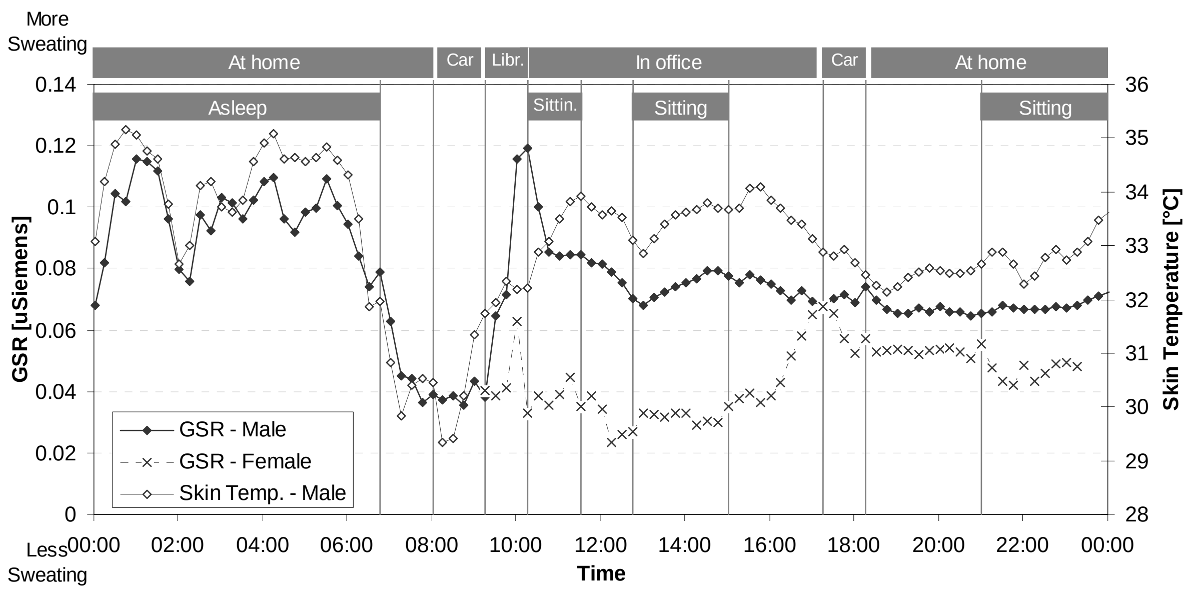


Figure 2. An example of parallel GSR and skin temperature monitoring results obtained by the embedded body sensor

The GSR level is the highest (~0.1uS) during the sleeping period when the subject is covered by a thick layer of blanket that has good thermal resistance. After getting out of the bed there is a steep drop in GSR when the body is exposed to a cooler environment (ending at ~0.04uS). Due to a higher activity level that includes walking and carrying books through different floors in a library GSR shows a sudden increase for both subjects. Maintaining sedentary activity in the office GSR stabilizes and shows minor changes only when the activity level is altered. The skin temperature follows a similar pattern as the GSR. The highest deviation of skin temperature path from the GSR sample occurs when the activity level is changing. Although the relative changes in the female and male subject's GSRs are similar there is a noticeable difference in the absolute values of GSR between the two subjects.

Output of the embedded body sensor in a controlled office environment

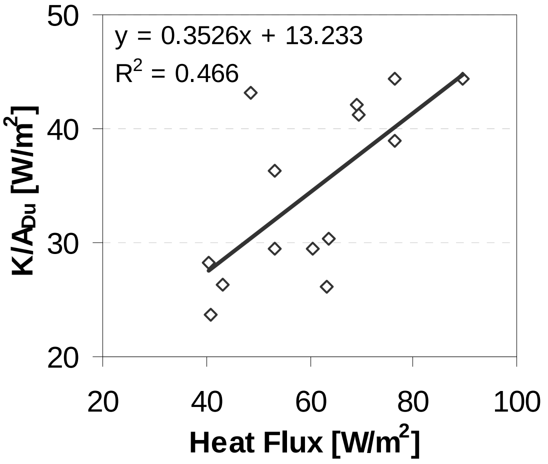
The following Table 2. summarizes the average levels of the physical and some physiological measures given by the wireless environmental sensor system and the embedded body sensor under steady-state experimental conditions. In addition, empirical values of the mean skin temperature, conductive heat loss through the clothing (K/A_{Du}) and PMV (Predicted Mean Vote) [10] are also indicated. Due to the small sample size the subjects' thermal and sweating intensity votes are shown informatively.

Table 2. Average physical and subjective results ±SD over one hour under steady-state conditions as function of the experimental conditions conducted on different days and at different times of the day

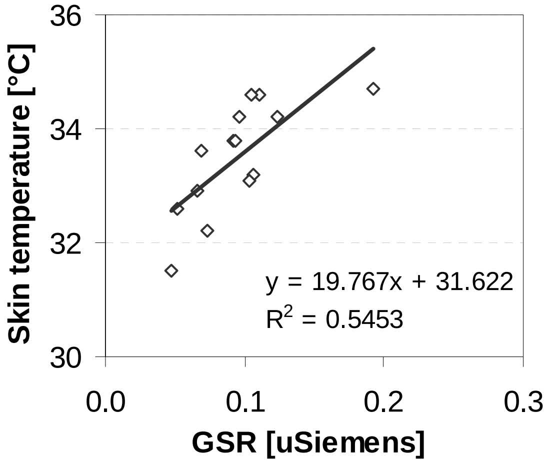
Activity level 1.2 ± 0.1 Met		Experiment 1		Experiment 2			Experiment 3	
		Morning 25°C	Afternoon 22°C	Morning 23°C	Afternoon 28°C	Evening 27°C	Day 1 23°C	Day 2 28°C
Air temperature [°C]		25.1 ± 0.3	21.8 ± 0.2	22.6 ± 0.0	26.6 ± 0.1	25.6 ± 0.2	23.4 ± 0.1	29.1 ± 0.3
Relative Humidity [%]		37.5 ± 0.6	36.2 ± 0.6	37.5 ± 0.4	36.8 ± 0.3	35.5 ± 0.3	39.9 ± 0.1	35.3 ± 0.3
Operative temp. [°C]		25.4 ± 0.5	22.3 ± 0.3	23.3 ± 0.3	28.2 ± 0.3	27.1 ± 0.3	23.1 ± 0.1	28.1 ± 0.3
Heat flux [W/m^2]	M[a]	53.1 ± 2.7	69.1 ± 1.5	89.6 ± 3.7	63.2 ± 2.5	60.4 ± 1.4	76.4 ± 2.6	40.6 ± 1.3
	F[b]	63.5 ± 3.1	76.4 ± 1.5	69.4 ± 2.1	43.1 ± 2.7	53.3 ± 4.3	48.5 ± 3.1	41.0 ± 1.2
Skin temp. [°C]	M	33.8 ± 0.1	32.2 ± 0.1	33.8 ± 0.1	34.6 ± 0.1	34.2 ± 0.1	33.2 ± 0.1	34.7 ± 0.2
	F	32.6 ± 0.2	31.5 ± 0.1	33.1 ± 0.1	34.6 ± 0.1	34.2 ± 0.1	32.9 ± 0.2	33.6 ± 0.3
GSR x 10^{-2} [uSiemens]	M	9.2 ± 0.6	7.3 ± 0.2	9.4 ± 0.2	11.0 ± 0.1	12.4 ± 0.6	10.6 ± 0.2	19.2 ± 2.4
	F	5.2 ± 0.1	4.7 ± 0.1	10.4 ± 0.2	10.5 ± 0.2	9.6 ± 0.1	6.6 ± 0.3	6.9 ± 0.2
Sweat[c] [-]	M	10.0 ± 0.0	3.5 ± 4.9	0.0 ± 0.0	21.7 ± 2.9	19.0 ± 10.8	0.7 ± 1.2	27.6 ± 7.7
	F	2.0 ± 2.8	0.0 ± 0.0	0.0 ± 0.0	11.0 ± 5.6	0.0 ± 0.0	0.0 ± 0.0	18.0 ± 2.8
Mean Skin temp.[°C]	M	33.7 ± 0.2	33.8 ± 0.0	33.7 ± 0.1	33.8 ± 0.0	33.8 ± 0.0	33.8 ± 0.0	33.8 ± 0.0
	F	33.8 ± 0.1	33.8 ± 0.1	33.9 ± 0.1	33.8 ± 0.1	33.8 ± 0.5	33.7 ± 0.3	33.8 ± 0.0
K/A_{Du} [W/m^2]	M	36.3 ± 1.9	42.1 ± 1.3	44.3 ± 1.1	26.2 ± 1.5	29.4 ± 1.6	44.4 ± 0.7	28.2 ± 0.5
	F	30.4 ± 2.3	39 ± 1.8	41.3 ± 1.3	26.3 ± 1.3	29.4 ± 1.1	43.2 ± 0.4	23.6 ± 0.2
PMV[d] (Fanger)		0.64 ± 0.13	-0.07 ±0.09	0.16 ± 0.07	1.25 ± 0.08	1.00 ± 0.08	0.08 ± 0.04	1.26 ± 0.95
Thermal Vote[d]		1.50 ± 0.20	0.05 ± 0.28	-0.10 ±0.00	2.92 ± 0.03	2.45 ± 0.09	0.31 ± 0.25	2.80 ± 0.23

a) Male subject
b) Female subject
c) On six point intensity scale: No sweating = 0, Slight sweating = 10, Moderate sweating = 20, Strong sweating = 30, Very strong sweating = 40, Overpowering sweating = 50.
d) On seven point thermal sensation scale [9]: Cold = -3, Cool = -2, Slightly cool = -1, Neutral = 0, Slightly warm = 1, Warm = 2, Hot = 3.

The changes in the room operative temperature are reflected in almost all of the measured parameters. The heat flux given by the body sensor is the highest at lower temperatures and it diminishes with temperature rise. This effect is shown in all three experiments. Although the absolute values are different, there is a correlation between the measured and calculated heat loss from the body (R^2=0.46; p<0.007) (Figure 3.a). The measured skin temperature is distributed close to the calculated mean skin temperature corresponding to the given activity level [10]. It is greatly influenced by the operative temperature (R^2=0.67; p<0.001) and shows correlation with the GSR values in steady-state conditions (R^2=0.55; p<0.003) (Figure 3.b). GSR appears to be higher in case of the male subject compared to that for the opposite gender and shows good agreement with the male's vote reporting more intense sweating. The absolute values of GSR at a given temperature show fluctuation even within the results of the same subject.



a)



b)

Figure 3. Correlation for male and female subjects according to all experimental conditions between a) the measured and the calculated heat flux leaving the human body; b) the GSR and the skin temperature

The calculated PMV values and thermal votes were close to neutral when temperatures fell within the comfortable range (22-23°C). However, at the higher temperature conditions the calculated PMV underestimated the thermal sensation of the subjects.

DISCUSSION

Using the advantages of the wireless data communication a quick and easy monitoring of existing and new building environments can be performed. Due to the small size and battery enabled power supply of the sensors they can be easily positioned within the personal area to gain reliable information about the exposure levels. The setup of the whole system can be made within minutes after choosing the right location of each sensor. Although the current measurements were carried out in a single office the wireless system can be extended for multiple spaces or for a whole building that can provide useful information about the building performance for the facility management. The real time data measured with the wireless sensors can be displayed only with the software provided with this system. However, the data are stored in a database that could be accessed externally for further processing e.g. control of building services systems.

From the results of the whole day long measurement with the body sensor we can observe that the human body is exposed to very different environmental conditions that modify skin temperature and GSR in a wide range within the measured area. It is the duty of the thermal regulatory system of the body to perceive these conditions within or outside of the comfortable range. Since the body measurements are limited to only a small area of the right upper arm it is interesting to see how representative these results are for the whole body. Calculation methods of the mean skin temperature are reported to be evaluated by using 10-15 different points of the body accounting each with different weighting factors. According to [12] the location of the upper arm has a weighting factor of 0.07 which is the fifth most weighted location from the measured fifteen following the weighting factor of the left chest and back and the anterior and posterior thighs.
In general, differences in the base level of GSR may be related to environmental stress affecting thermal comfort but may also be of psychological origin such as personal mood [13]. Looking at Figure 2 and assuming that the psychological state of the investigated subjects did not change over the experiment, the variation of GSR is a good reflection of the evaporative heat loss through the body surfaces. This evaporative heath is transmitted partly by diffusion through the skin and partly through evaporation of sweat from the surface of the skin [10]. At operative temperatures below 31°C the sweat rate is relatively low [14] and since the heat loss through diffusion is a function of the skin temperature [10] the variation of GSR on Figure 2 follows a similar pattern as the skin temperature. Consequently, when the GSR pattern deviates compared to the skin temperature pattern, it is a good indication that sweating occurred, as it may be expected during the time spent in the library when the subject was carrying some heavy books. The GSR pattern on Figure 2 also reflects that variations between genders may be found and it is lower for females. This is not surprising since it was previously shown [15] that GSR is generally lower for females than for males and that GSR varies between different ethnic groups and it irregularly decreases with age. Seasonal and time of the day effects were also found on GSR and sweating response [16], [17].
Provided that the heat flux measured by the body sensor reflects the convective heat transfer from the skin we expected to get similar values as the conductive heat transfer (K/A_{Du}) through the clothing [10]. Although correlation was found between these two parameters K/A_{Du} remained much lower than that measured by the body sensor system that may be caused by calibration error. The measured skin temperature matched the calculated mean skin temperature within ±1°C except in one condition with the lowest temperature. Therefore it may be considered in further investigations to use this measured value in PMV calculations. Previous investigations [14] showed that sweating starts in nude subjects when skin temperature reaches 34°C. Altough we had clothed subject in the present experiment, our data showed similar effect: whenever the measured skin temperature approached or exceeded 34°C the sweat vote of the subjects has also increased from no sweating to slight or moderate levels.
As stated earlier gender differences in GSR may appear, which is again confirmed by the results obtained in the controlled office exposure. Altough variations in subject's GSR were found at the same temperature it could be attributed to the time of the day effect or to the psichological state of the subjects. The humidity content of the air between the conditions varied between narrow limits, therefore we do not expect of having considerable effect on the mesured GSR [18].

The present investigation showed that the embedded body sensing system that measures physiological parameters can give valuable data that correlate with changes in the physical environment of building occupants. The current measurements were limited to investigate only the changes induced by various levels of the room temperature. Further measurements are needed to explore the effects of other parameters describing the thermal environment and other factors of psychological origin that affect the physiological output of the body sensing system.

ACKNOWLEDGEMENT

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Evaluation of Comfort and Fatigue of Japanese Subjects in Extremely Low Humidity Air

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SUMMARY

Subjective experiments were carried out in a climate chamber using 16 Japanese subjects of both genders, in order to evaluate human comfort at very low humidity. Two levels of absolute humidity were set, 2.0 g/kg(DA) and 10.0 g/kg(DA). Three air temperature conditions with absolute humidity of 2.0g/kg(DA), 20.0°C/13%RH, 25.0°C/10%RH and 30.0°C/8%RH, and 3 conditions with 10.0g/kg(DA), 20.0°C/68%RH, 25.0°C/50%RH, 30.0°C/38%RH, were examined. People were exposed in a chamber for 90 minutes with sedentary activity. Subjects rated their sensations and subjective fatigue on the questionnaires during the exposure time. Skin moisture was measured on subjective left forearm by means of measuring capacitance of the skin. Break up time (BUT) was measured by subjects with a stopwatch.

Absolute humidity has great impact on subject's skin moisture although air temperature effects on it were moderate. Subjective BUT got significantly shorter at extremely low absolute humidity. It was found that general dryness sensation at low absolute humidity was significantly higher than that at high absolute humidity. Subjects felt their eyes were dryer at low air temperature, when keeping the same moisture content in air. In case of the constant air temperature, low absolute humidity caused greater eye dryness sensation. Very low humidity air dry up the human mucous membrane and subjects perceived it as dryness sensation. Half of subjects reported their palm of hand as wetted segment even at 25.0°C/10%RH and 30.0°C/8%RH. Subjects felt more tired at very low absolute humidity under the cool and thermally neutral condition, while they complained more at high humidity in hot environment.

INTRODUCTION

In Japan, the "Law for Maintenance of Sanitation in Buildings [1]" is applied to offices whose total floor areas exceed 3,000 m^2. It states that the relative humidity in an office space should be kept between 40%RH and 70%RH. The ASHRAE Standard 55-92 [2] prescribed a lower boundary humidity of 4.5 g/kg which was equivalent to 30%RH at 20.5°C. This standard was revised as the ASHRAE Standard 55-2004 [3], which does not specify a minimum humidity level. The ASHRAE Standard 62-2001 [4] recommends the relative humidity of 30%RH - 60%RH. These lower boundaries of current humidity criteria are intended to limit the low humidity conditions in winter. However, improvement of recent HVAC technology has allowed engineers to create a thermal environment with low humidity even during summer using cold air distribution systems or desiccant dehumidifiers in many buildings. Further studies on the effects of low humidity on occupants' comfort in other seasons are needed, as well as in winter.

The previous subjective experiment [5] at 5%RH to 35%RH using Danish people reports that negative impacts on their tear film and blink rate were observed at below 15%RH, though the subjective discomfort was very mild even at 5%RH.

Tsutsumi et al. [6][7][8] have reported the effects of low humidity on subjective comfort and productivity in summer through the various subjective experiments using Japanese subjects. As the result, negative effects of humidity were not found in the thermally neutral air even at 30%RH, which is below the lower boundary of humidity mentioned in Japanese law under a steady state in summer. However, the conditions examined in this series of experiments mentioned above, were between 30%RH and 70%RH. The studies on Japanese comfort and fatigue in the very dry air, especially below 15%RH, has been required.

This paper reports the subjective experiment conducted in a climate chamber for the purpose of evaluating comfort and fatigue of Japanese subjects at extremely low humidity.

METHODS

Experimental design

Subjective experiments were carried out to evaluate the human comfort and fatigue in extremely low humidity air. A total of 16 Japanese adults of both gender, aged 20's-60's, were used as subjects. All subjects were volunteers, who were paid for participating in the experiments. Considering their circadian rhythm, all subjects took part in the experiments at the same time of the day.

The experimental conditions are listed in Table 1. Two levels of absolute humidity were set, 2.0g/kg(DA) and 10.0g/kg(DA). For each absolute humidity level, 3 air temperatures, 20.0, 25.0 and 30.0 °C, were examined. Mean radiant temperature (MRT) was estimated to be equivalent to air temperature. Air velocity was still for all conditions.

Table 1. Experimental conditions.

	Absolute Humidity [g/kg(DA)]	Air Temperature =MRT [°C]	Relative Humidity [%RH]	Air Velocity
Low_20	2.0	20.0	13	still
Low_25		25.0	10	
Low_30		30.0	8	
High_20	10.0	20.0	68	
High_25		25.0	50	
High_30		30.0	38	

As shown in Figure 1, subjects were exposed in a climate chamber for 90 minutes, where a desiccant dehumidifier was equipped so that extremely low humidity air could be created. During the exposure time, subjects sat on the sofa and watched TV programs simulating the daily life at home. Subjective sensations, symptoms related to fatigue and visual fatigue and their self-performance were reported on the questionnaire using the visual analogue scale 5 times while they stayed in a chamber.

Skin moisture on subject's left forearm was recorded with Moisture checker (Scalar). Moisture checker measured capacitance of the skin [9]. Subjects kept their left forearm exposed to the air in the chamber during the 90-minute experiment. Subjects also measured their interval time between each blink by themselves using a stopwatch as Break up time (BUT) [10].

People could control their clothing ensembles provided by the experimenter to keep their thermal sensation as neutral as possible. They were not allowed to eat or drink something and use eye drops during the exposure.

Air temperature, relative humidity, globe temperature during the experiments were logged every 10 second. Air velocity was measured before and after the exposure.

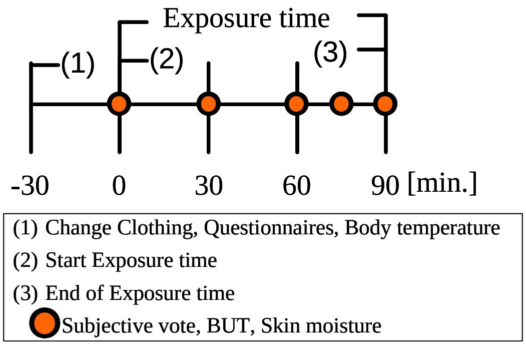

Figure 1. Experimental procedure.

Statistical Analysis

Data at the end of 90-minute exposure were analysed as that obtained under steady state with Non-parametric statistical analysis method [11]. Friedman nonparametric analysis was used for comparison among 3 conditions at the constant absolute humidity. The Wilcoxon Matched-Pairs Signed Ranks test was administered between each condition as a post-hoc test. This test was also used for the comparison between 2 absolute humidities at the same air temperature. The p-values mentioned in the next section represent the levels of significance.

RESULTS AND DISCUSSION

Physiological reactions

Skin moisture:

Skin moisture is one of the physiological responses that could be affected by humidity and would cause the subjective dryness sensation and discomfort. Figure 2 shows the skin moisture measured on the left forearm of subjects at the end of exposure time. No significant difference was observed among 3 conditions with 2.0g/kg(DA). Clear direction was not found among conditions at 10.0g/kg(DA), although significant difference was gotten ($p<0.04$). Skin moisture on left forearm at low absolute humidity was significantly lower than in high absolute humidity air at the same air temperature ($p<0.02$). It is concluded that absolute humidity has great impact on subject's skin moisture although air temperature effects on it was moderate.

Break up Time (BUT):

Break up Time (BUT) of precorneal film is one of the physiological reactions that might affect the subjective eye comfort. During the exposure time, the subjects measured their interval time between each blink by themselves using a stopwatch. Figure 3 presents BUT measured after 90-minute exposure. Although the Friedman nonparametric analysis revealed no significant difference among 3 conditions, BUT at 20.0 °C was shorter than that at other conditions for both absolute humidity levels. The Wilcoxon Matched-Pairs Signed Ranks test reported that BUT at 25.0°C and 30.0 °C with 2.0g/kg(DA) was significantly shorter than at 10.0g/kg(DA) at the same temperature ($p<0.04$). Shorter BUT was observed under Low_20 condition than High_20 condition, although no statistically significant difference was seen. It is found that air temperature did not affect their BUT in case of keeping the constant absolute humidity. On the other hand, subjective BUT got significantly shorter at extremely low absolute humidity.

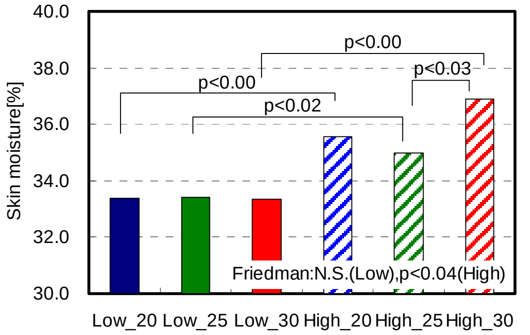

Figure 2. Skin moisture at the end of exposure

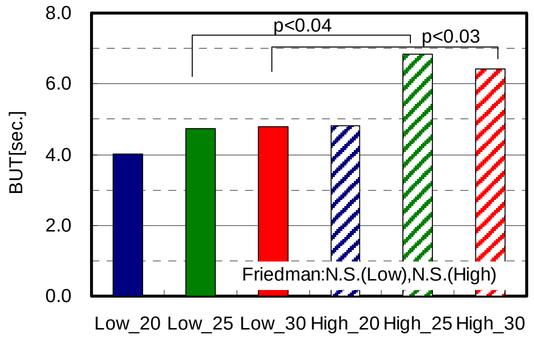

Figure 3. BUT.

Psychological Reactions

Subjects rated their general thermal sensation, comfort sensation, humidity sensation, eye dryness and so on during the exposure. A part of rating scales is illustrated in Figure 4. The scales were given as visual analogue scales. Subjects were allowed to rate their sensation either just on the number or between the numbers on the scales.

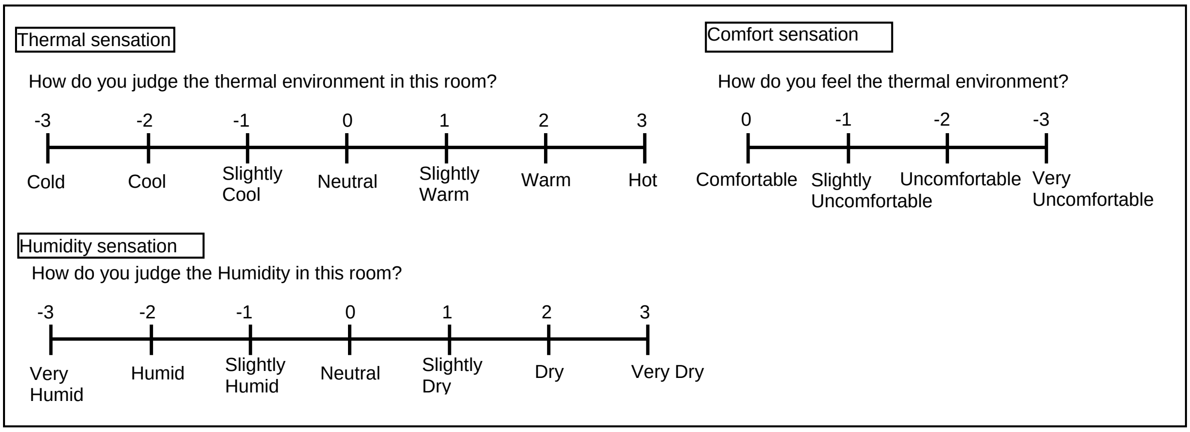

Figure 4.A part of rating scales

Thermal sensation:

Thermal sensation vote at the end of the exposure time is shown in Figure 5. In both absolute humidity air, the Friedman nonparametric analysis revealed significant difference among 3 conditions ($p<0.00$). On the other hand, pair-wise comparison between low absolute humidity and high absolute humidity condition at the same air temperature did not found any significant difference. Subjective thermal sensation was associated with air temperature, while the effect of humidity was moderate under the conditions set for this research.

General humidity sensation:

The last humidity sensation vote is presented in Figure 6. General dryness sensation tended to be higher at low air temperature among 3 conditions with 2.0g/kg(DA) of absolute humidity ($p<0.1$). Friedman nonparametric analysis revealed statistically significant difference among 3 conditions in high absolute humidity air ($p<0.005$). Subjects felt the air was significantly dryer at low air temperature than at high air temperature at constant moisture content in air. Air temperature had great impact on subjective general humidity sensation under the conditions set for this experiment. According to the pair-wise comparison between 2 levels of absolute humidity at the same air temperature, it was found that general dryness sensation at low absolute humidity was significantly higher than that at high absolute humidity ($p<0.03$).

Figure 7 displays subjective humidity comfort sensation rated using the same scale as comfort sensation. There was no significant difference among 3 conditions in low absolute humidity air. Significant difference was occurred in high absolute humidity air ($p<0.04$). According to the Wilcoxon Matched-Pairs Signed Ranks test, greater discomfort was observed at 30.0 °C than other conditions. Subjects reported greater discomfort in low absolute humidity air than high absolute humidity at 20.0 °C and 25.0 °C, while opposite result was found at 30.0 °C.

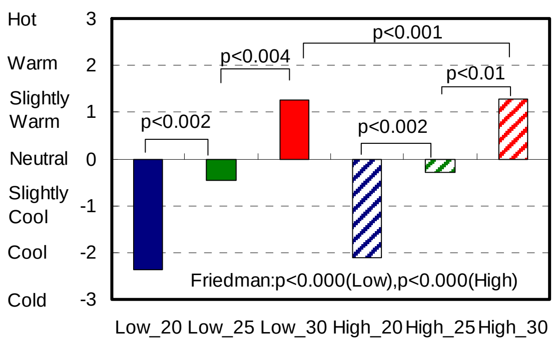

Figure 5. Thermal sensation vote.

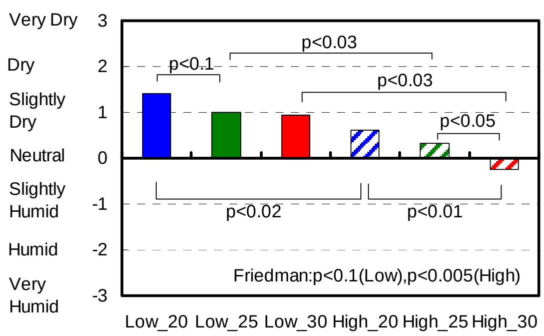

Figure 6. Humidity sensation vote.

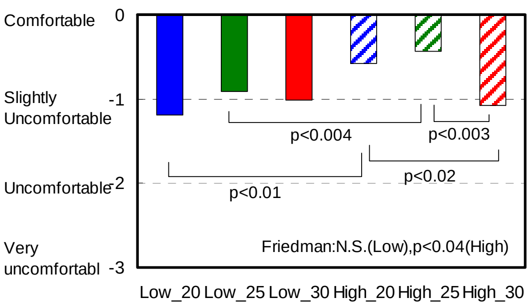

Figure 7. Humidity comfort sensation.

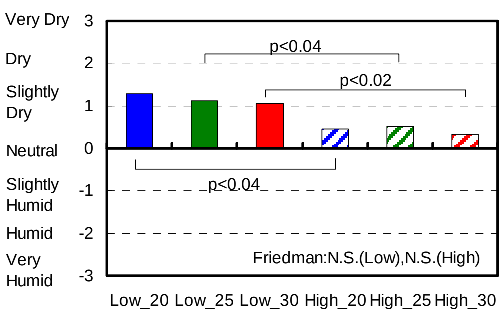

Figure 8.Sensation of eye dryness.

Eye dryness sensation:

Subjects rated their sensation of eye dryness using the same scale as general humidity sensation. Figure 8 presents subjective sensation of eye dryness at the end of exposure time. Greater dryness sensation of eyes was seen at low air temperature than at high air temperature, while no significant difference was found among 3 conditions for both absolute humidity levels. The Wilcoxon Matched-Pairs Signed Ranks test between 2 absolute humidity conditions reveals significantly greater eye dryness sensation under low absolute humidity condition than high humidity condition at the same air temperature at the range from 20.0°C to 30.0°C. It is concluded that subjects felt their eyes were dryer at low air temperature, when keeping the same moisture content in air. In case of the constant air temperature, low absolute humidity caused greater eye dryness sensation.

Body segments subjects feel dry/ wet:

Subjects were asked to report their body segments which they felt dry or wet. Figures 9 and 10 display the number of people who reported each body part to be humid after 90-minute exposure, and Figures 11 and 12 to be dry. As shown in Figures 9 and 10, more segments were felt to be dry under the low absolute condition than under high absolute condition. More than 6 subjects felt their eyes, nose and mouth were dry at low absolute humidity. Even in the air with 10.0g/kg(DA), eyes, nose and mouth were rated as the parts where they felt dry compared with other parts. This result clearly demonstrated that very low humidity air dry up the human mucous membrane and subjects perceived it as dryness sensation.

As for the segments subjects perceived to be wet, subjects rated their palm of hand more than other segments for both absolute humidity conditions. More subjects felt their palm of hand were wet at 25.0°C and 30.0°C than at 20.0°C for both absolute humidities. Half of subjects reported their palm of hand as wetted segment even at 25.0°C/10%RH and 30.0°C/8%RH.

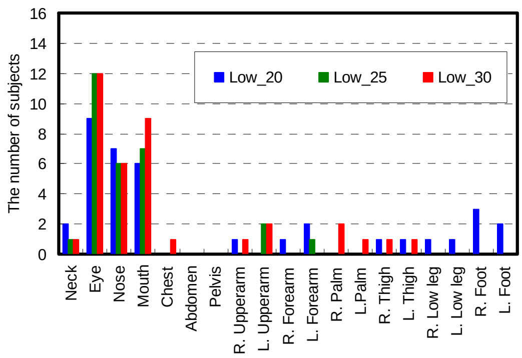

Figure 9. Body segment subjects felt "dry" at low absolute humidity.

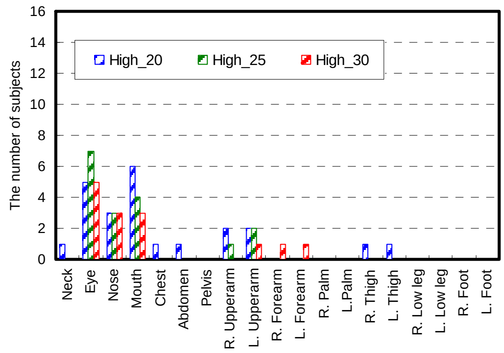

Figure 10. Body segment subjects felt "dry" at high absolute humidity.

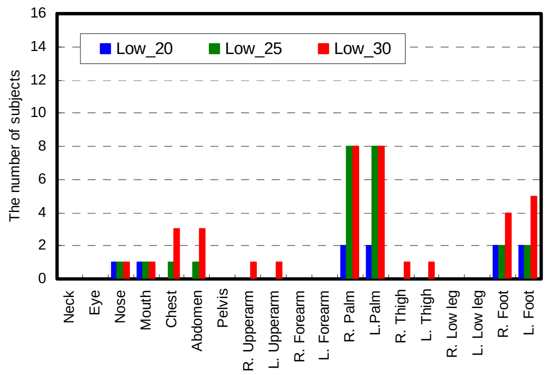

Figure 11. Body segment subjects felt "wet" at low absolute humidity.

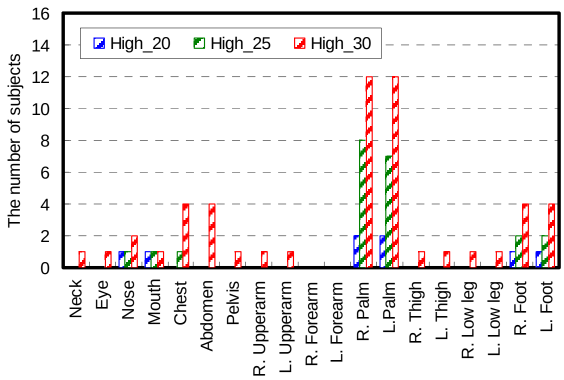

Figure 12. Body segment subjects felt "wet" at high absolute humidity.

Fatigue

Subjects were asked to assess their general fatigue. A questionnaire proposed by the "Working Group for Occupational Fatigue, Japan Society for Occupational Health" was used to evaluate their fatigue [12]. The questionnaire is composed of 3 groups, category I, category II and category III. "Category I" indicates drowsiness and dullness in subjects, "Category II" is about difficulty in concentration, and "Category III" is to do with physical discomfort. Each category has 10 symptoms related to subjective fatigue as listed in Table 2. Subjects marked "O" if they had the given symptoms, and marked "X" if they did not. Ratio of complaints was calculated for each category using the equation below:

$$Rate\ of\ complaints = \frac{Total\ number\ of\ complaints(=Total\ number\ of\ "O")}{(The\ number\ of\ symptoms)\times(Total\ number\ of\ subjects\ who\ used\ a\ questionnaire)}\times 100(\%)$$

Higher air temperature in 10.0g/kg(DA) of absolute humidity caused higher ratio of complaints. In pair-wise comparison between high absolute humidity conditions and low absolute humidity condition with the same temperature, it was observed that the ratio of complaints at 2.0g/kg(DA) was higher at 20.0 and 25.0°C. On the other hand, people felt more tired at 30.0°C with high absolute humidity than that with low absolute humidity. It is concluded that people could feel more tired at low humidity under the cool and thermally neutral condition, while they complained more at high humidity in hot environment.

Table 2. Symptoms related to the general fatigue.

	Category I Drowsiness and Dullness	**Category II** Difficulty in Concentrating	**Category III** Physical Discomfort
1	Feel heavy in the head	Feel difficult in thinking	Have a headache
2	Get tired through the whole body	Become weary of talking	Feel stiff in the shoulders
3	Get tired in the legs	Become nervous	Feel a pain in the back
4	Take a yawn	Unable to concentrate	Feel difficulty in breathing
5	Feel the brain hot or muddled	Unable to take interest in things	Feel thirsty
6	Become drowsy	Become apt to forget things	Have a husky voice
7	Feel eye strain	Lack in self-confidence	Have dizziness
8	Become rigid or clumsy in motion	Anxious about things	Have a spasm on the eyelids
9	Feel unsteady while standing	Unable to straighten up in a posture	Have a tremor in the limbs
10	Want to lie down	Lack patience	Feel ill

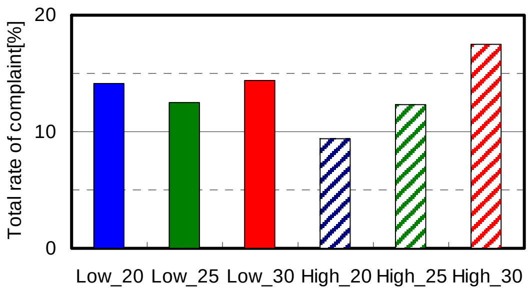

Figure 13. Total rate of complaint related to general fatigue.

CONCLUSION

Subjective experiments were carried out in a climate chamber using 16 Japanese subjects of both genders, in order to evaluate human comfort at very low humidity. Two levels of absolute humidity were set, 2.0 g/kg(DA) and 10.0 g/kg(DA). Three conditions with absolute humidity of 2.0g/kg(DA), 20.0°C/13%RH, 25.0°C/10%RH and 30.0°C/8%RH, and 3 air temperature conditions with 10.0g/kg(DA), 20.0°C/68%RH, 25.0°C/50%RH, 30.0°C/38%RH, were examined. People were exposed in a chamber for 90 minutes with sedentary activity.

Absolute humidity has great impact on subject's skin moisture although air temperature effects on it were moderate. Air temperature did not affect their BUT in case of keeping the constant absolute humidity. On the other hand, subjective BUT got significantly shorter at extremely low absolute humidity.
Air temperature had great impact on subjective general humidity sensation under the conditions set for this experiment. According to the pair-wise comparison between 2 levels of absolute humidity at the same air temperature, it was found that general dryness sensation at low absolute humidity was significantly higher than that at high absolute humidity.
It is concluded that subjects felt their eyes were dryer at low air temperature, when keeping the same moisture content in air. In case of the constant air temperature, low absolute humidity caused greater eye dryness sensation.
Very low humidity air dry up the human mucous membrane and subjects perceived it as dryness sensation. Half of subjects perceived their palm of hand as wetted segment even at 25.0°C/10%RH and 30.0°C/8%RH.
According to assessment of subjective fatigue, they could feel more tired at low humidity under the cool and thermally neutral condition, while they complained more at high humidity in hot environment.

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Surveyed Thermal Comfort in Iranian Offices

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SUMMARY

This paper presents the findings of a short-term monitoring exercise and questionnaire survey to assess the thermal comfort conditions actually being achieved in 6 Iranian Office buildings. The findings of the questionnaire and monitoring are compared to give confidence that the questionnaire is accurately reflecting the calculated comfort conditions obtained from the physically monitoring and site observations. The perceived thermal comfort conditions for the buildings are then compared with current thermal comfort standards to ascertain whether there is potential for improvement.

The results from this study indicate that there is a good agreement between the monitoring and the questionnaire for the majority of the offices. The results also show that there appears to be room for significant improvement in thermal comfort conditions in a number of the Case Studies. These findings will be used as part of further work to explore how we might improve the thermal conditions through improvements to the Offices fabric design.

Key words: thermal comfort, offices, thermal satisfaction, field study, thermal environment

INTRODUCTION

According to ASHRAE Standard 55 [1], 'thermal comfort is that condition of mind that expresses satisfaction with the thermal environment'. Thermal comfort not only affects on occupant's satisfaction but also affects on their productivity as well.

Thermal comfort receives the highest number of complaints in most surveys of office occupants. Only about half of North American office workers express the heating and air conditioning in their workspace is comfortable and 80% of them rate thermal comfort as being very important to them [2]. Clements-Croome, D and Baizhan, L [3] have proposed that thermal problems are one of the most significant factors affecting environmental satisfaction. Abdou and Lorsch [4] also noted a link between temperature and factory / industrial worker's productivity. The research also has shown that there is an increase in satisfaction level with thermal environment when the degree of personal control increases and there is a slight increase in overall job satisfaction when environmental satisfaction increases [5]. It has also been discussed by de Dear, R and Cena, K [6] in a field study carried out in Kalgoorlie-Boulder in Australia that there is a close correlation between expressions of thermal dissatisfaction and job dissatisfaction. They found that those who rated lower than mean in their job satisfaction were more likely to report dissatisfied with thermal conditions than those who scored above average job satisfaction.

There is a psychological aspect to individual control as well. Research has shown that people are more satisfied with their thermal environment when they believe it to be under their control, even if they were not in fact actually able to alter the temperature [5]. Ruck, N.C [7] indicated that thermal comfort is a complex psycho physiological condition of satisfaction that happens within a subtle range of human responses.

There are various environmental factors that are affecting thermal comfort. As Fanger, P.O [8] has addressed there are six important variable that influence the condition of thermal comfort; activity level (heat production in the body), clothing insulation (clo value), air temperature, mean radiant temperature, relative air velocity and water vapour pressure in ambient air. Thermal comfort can be measured when these six variables are measured or estimated. Fanger's comfort model is based on PMV (predicted mean vote) and PPD (predicted percentage of dissatisfied people). PPD can be calculated when PMV is estimated. On the other hand there are studies have shown that thermal comfort is an behavioural approach [9] and unlike the PMV method is not based on the clothing insulation and the metabolic rate to calculate required thermal comfort . They suggest that people tend to make themselves comfortable by changing their clothing, activity and posture in their thermal environment. Their adaptive model is based on a wider range of data derived from buildings, climates and cultures and outdoor temperature is the main factor to calculate the adaptive thermal comfort. Also, it has been argued that there is a discrepancy between predicted thermal comfort (PMV) and perceived thermal comfort of occupants in free-running buildings and naturally ventilated buildings [10, 11, 12]. It has been suggested that people of different cultures perceive themselves comfortable in a wide range of temperature about 17ºC to 31 ºC and they can adapt to extreme temperatures [12]. But researches have shown that in centrally controlled or air conditioned buildings the PMV model works well for calculating thermal comfort. de Dear and Brager [11] have shown that the PMV model is in a close agreement with the observed thermal comfort of occupant in centrally-controlled HVAC systems or air conditioned buildings. Fanger and Toftum [13] have shown that PMV model is in a good agreement with high quality field studies in HVAC buildings situated in cold, temperate and warm climates.

Overall, these studies emphasise the importance of thermal comfort for office occupants and highlight that achieving thermal comfort in offices not only delivers more satisfaction for the occupants, but also improves their performance as well.

To understand whether the published international standards for thermal comfort were applicable to the Iranian office context, a short-term questionnaire survey and a physical measurement and monitoring study have been undertaken in six office buildings in Tehran (Iran) in the hot season (July and August) in 2005.The aim of this study was:

- To establish via questionnaire how the occupants perceive the internal conditions with comfort.
- To establish the range of indoor and external conditions found in the offices.
- To compare measurement and physical monitoring with the perceived thermal sensation in the studied offices.
- To compare the results with international approaches such as ASHRAE standard 55 and ISO7730.

- To try to understand how the occupants achieve acceptable comfort conditions in extreme temperatures.
- To establish what routes may be available to improve comfort in these offices.

The environmental measurements and surveys were conducted in six office buildings in Tehran. These buildings were a mixture of types ranging from highly glazed, modern designs and up to 30 year old. The field experiments and an office occupant's questionnaire survey were carried out during the hot season in Tehran (July and August).
In all the offices studied, the air temperature, relative humidity and globe temperature have been measured as environmental factors for up to a two week period. The recording intervals were set to 10 minutes for the air temperature and relative humidity and 20 minutes for the globe temperature. Measurements of these parameters were taken in two office rooms in each building.

METHODS

For this research two data collection methods have been used; a questionnaire as a subjective measurement was used to obtain the occupants responses to their environment, and physical measurements were used to obtain quantitative data on the actual conditions in the offices.

Subjective measurement (questionnaire)

For the questionnaire the occupants' perceptions of the indoor environmental quality are obtained by a questionnaire survey. The scope and format of the questionnaire are based on a mixture of the Indoor Environmental Quality (IEQ) survey questionnaire from the Centre of the Built Environment (CBE) at the University of California, Berkeley [14] and a questionnaire for the study of Sick Building Syndrome (SBS) used by the Building Research Establishment (BRE) [15] in the UK. There were also some additional questions that were necessary for the research that were not covered by either of the two reference surveys, and were therefore added by the author. Satisfaction with the indoor environmental quality (IEQ) is evaluated by several criteria, including thermal comfort, air quality, lighting and noise. The occupants' perception of satisfaction with IEQ is assessed on a scale ranging from very satisfied (+3) to very dissatisfied (-3). The occupant perception of thermal comfort is evaluated by a scale based on the ASHRAE 55 [1] seven-point thermal sensation scales which ranges from hot (+3) to cold (-3). The perceived health of occupants is also being asked for and is based on the sick building syndrome questionnaire from BRE [15]. Occupants are asked whether they experienced one or more symptoms on a list of eight common SBS indicators. The symptoms are considered as being building related in office buildings. The approximate time required to complete the questionnaire was 5-10 minutes. Of the studied office buildings the response rate was 92% (195 out of 210).

Physical measurements

For the physical measurements the indoor air temperature, relative humidity and globe temperature were measured over a period of 2 weeks during the hot summer season in Tehran (July and August). The studied office buildings were all centrally cooled/ heated or air conditioned and none of them were naturally ventilated. Regarding to literature and studies stated above PMV model works well in buildings with air-conditioning or HVAC systems, therefore the data obtained from the physical measurements were used to calculate PMV (predicted mean vote) and PPD (predicted percentage of dissatisfied people) indices according to ISO 7730 [16] and ASHRAE standard 55 [1].

Analysis

the questionnaire survey and physical measurements were conducted over a series of two week periods in the hot season, during which the outdoor temperatures ranged between 23.7 ºC to 36.6 ºC with a range of 25% to 40% for the outdoor average relative humidity. This data is shown graphically in figure 1. The rain was unexpected in the measurement period as the Tehran climate is normally hot and dry in summer.

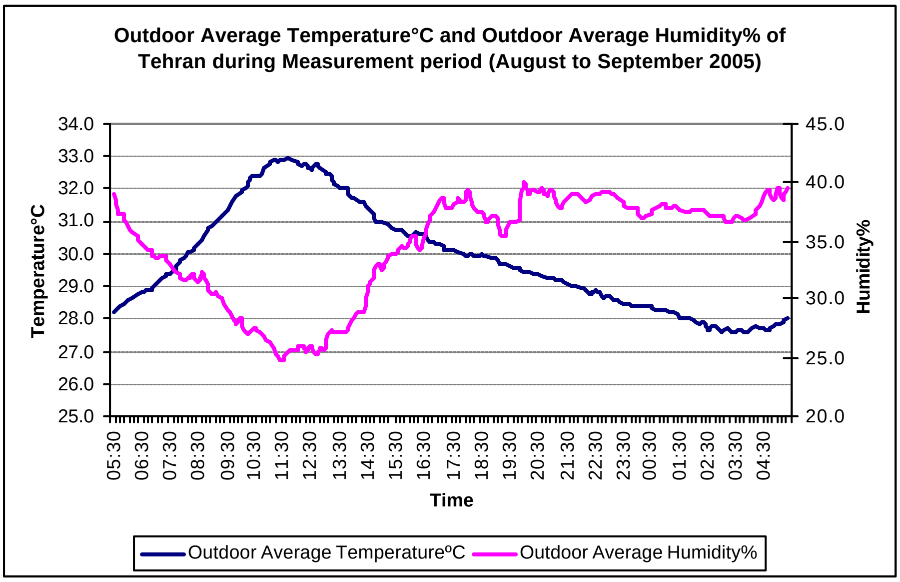

Figure 1: The measured outdoor average temperature ºC and outdoor average humidity of Tehran over measurement period (August to September 2005).

Results from the questionnaire

Figure 2 compares the results of the questionnaire over all the offices studied with the ASHRAE Standard 55 thermal comfort condition (80% or more of the occupants are satisfied with the temperature). It reveals that only 57% of the occupants reported they were satisfied with the temperature in general. Moreover, more than 23% of the occupants were positively dissatisfied with the temperature in their workspace which is more than the acceptable dissatisfied range (10%) specified by ASHRAE Standard 55 [1]. Therefore, it appears that the occupant's perceived satisfaction with the temperature is not in compliance with the acceptable thermal satisfaction rate within ASHRAE Standard 55 [1]. Summary of the occupant's specification are summarized in table 1.

Table 1: Summary of occupant's time spent at work, type of work, age, gender and clothing insulation,source from author [17]

Occupant's data		
	10 or less	9.80%
Time spent at workspace (hours)	11-30	24.20%
	More than 30	66.00%
	Managerial	6.30%
Type of work	Professional	73.80%
	Clerical/Secretarial	19.90%
	30 or under	36.90%
Age	31-50	51.80%
	Over 50	11.30%
Gender	Male	63.10%
	Female	36.90%
Clothing insulation (Clo)	Male	0.7
	Female	0.75

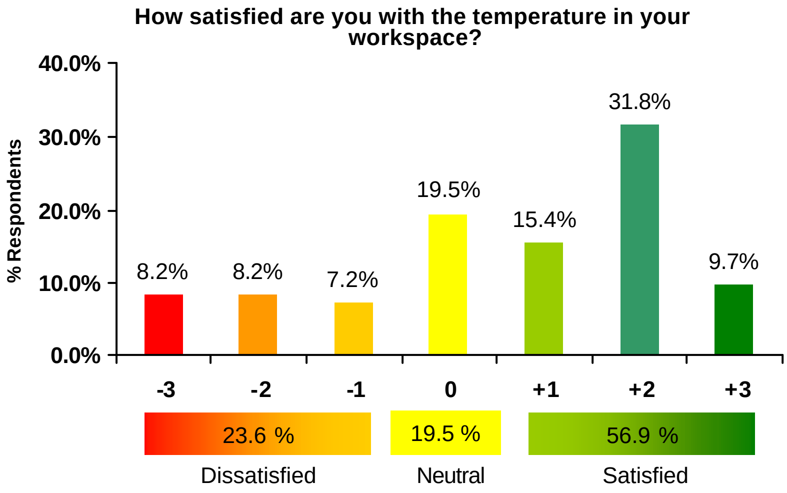

Figure 2: Occupants perceived general satisfaction with the temperature in all studied buildings.

Figure 3 shows the results of the questionnaire survey in more detail. It reveals that thermal neutrality was achieved in only around 22 - 25% of the occupants in winter or summer.

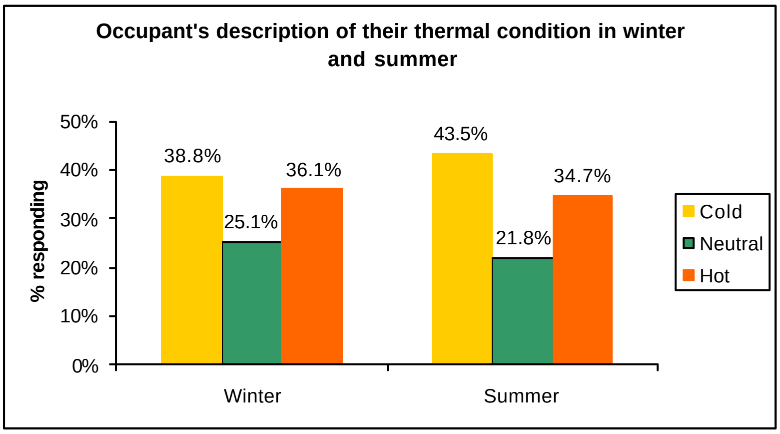

Figure 3: Respondent's description of thermal condition in their workspace in winter and summer in all studied buildings.

Figure 4 shows that, when considering the ambient conditions, the respondents reported that thermal comfort has the greatest effect on their perceived productivity. Taking this into account with their perceived low percentage of thermal comfort with temperature in winter and summer (figure 3) it can be reasonably suggested that improvement in the thermal conditions over the heating and cooling period would increase their perceived productivity.

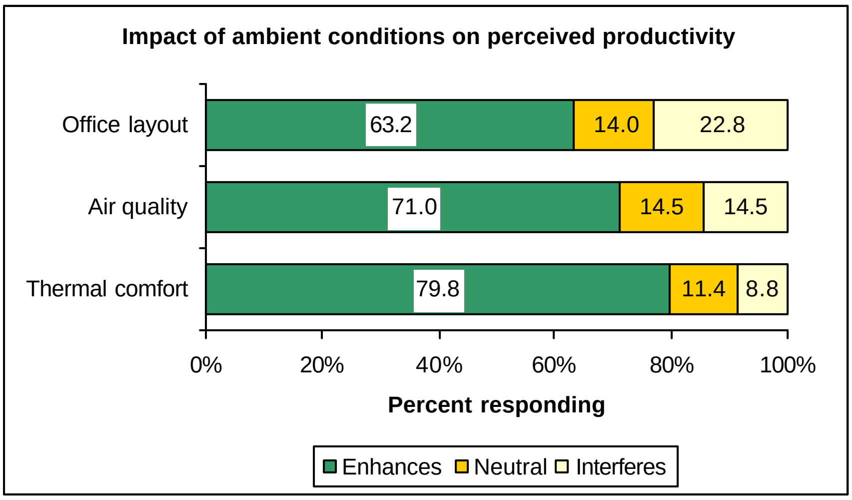

Figure 4: Respondent's perceived productivity with some of the main components of the ambient conditions (All buildings).

Findings from physical measurements and monitoring

The physical measurements, combined with observations about activity, clothing and air velocities in the spaces, allow calculation of the PMV according to ISO7730 [16] at each recording interval. Comparing the calculated range of PMV's for all the buildings monitored in figure 5 clearly indicates that a wide variety of PMV values are achieved. It seems that only 5 offices out of the 12 are controlled to within the specified acceptable thermal comfort range based on ISO 7730 [16] and ASHRAE Standard 55 [1] which is -0.5=PMV=+0.5 and PPD=10%. The PMV and PPD indices are based on the ASHRAE thermal sensation scale that defined as follows: +3 hot, +2 warm, +1 slightly warm, 0 neutral, -1 slightly cool, -2 cool, -3 cold.

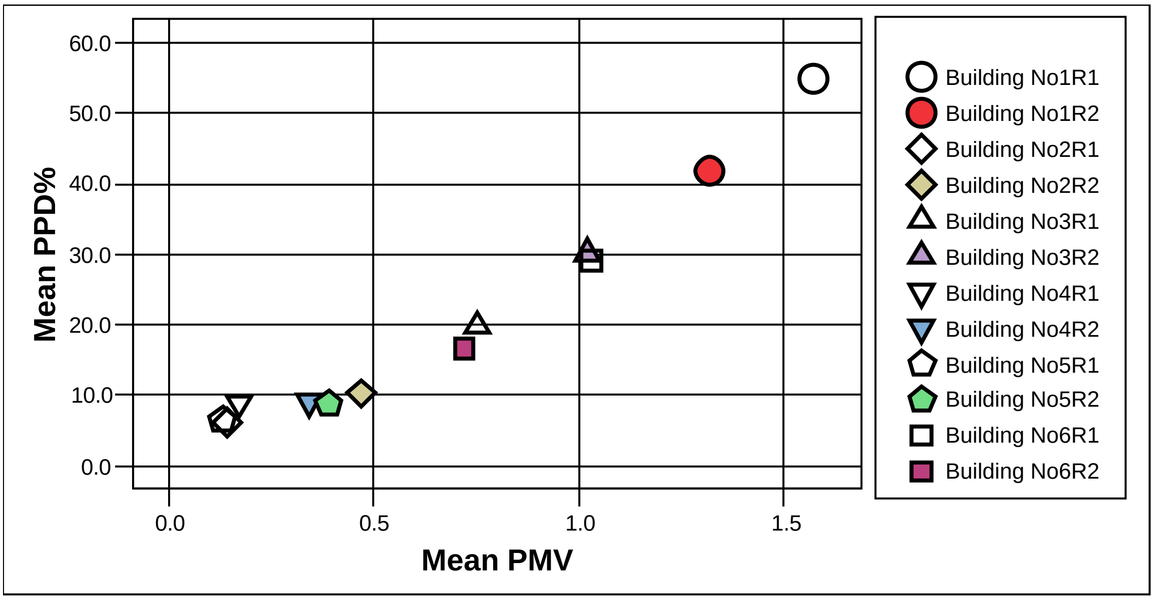

Figure 5: Mean PMV and Mean PPD of all studied offices.

Comparison between questionnaire survey and physical measurement

Figure 6 shows that generally there is a good agreement between the occupant's perceived mean thermal sensation from the questionnaire and the mean PMV obtained from the measurements. There is a slight difference between these values in building no6 and building no3. In building no6 the mean questionnaire value is neutral but the measured PMV shows slightly warm based on ASHRAE Standard 55. Also, in building no3 the occupants perceived their thermal condition as neutral but the calculated mean PMV indicates slightly warm. Overall the comparisons indicate no clear trend towards adaptation in the Iranian occupants of the buildings monitored. The main uncertainties in the findings come from the necessarily basic physical measurements undertaken, along with the assumptions about the air velocities in the spaces and the constancy of the clothing and activity levels.

Research has shown that for sedentary occupants humidity has relatively small effect on their thermal sensation and also perceived air quality, but in long term high humidity might cause microbial growth and very low humidity (<15-20%) might cause dryness and irritation of eyes and airways [18]. The sensitivity analysis shows that temperature is the most influence on the PMV value.

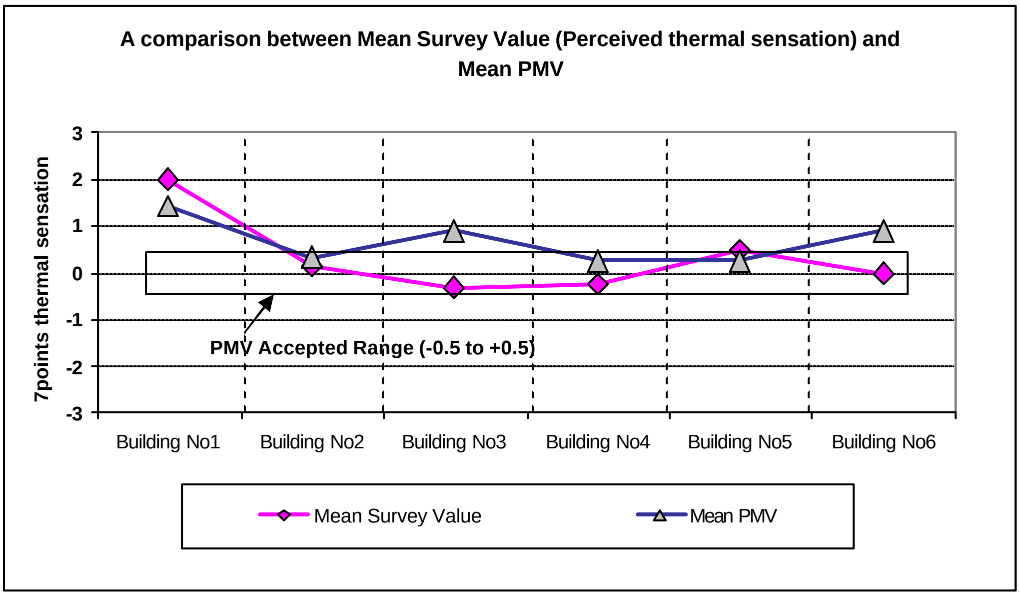

Figure 6: Comparison between occupants perceived mean thermal sensation with measured PMV from calculation.

Comparison between mean measured temperature and mean perceived thermal sensation

The data from table 2 clearly show that there is generally a good agreement between the mean measured temperature °C from physical measurement and perceived thermal sensation. In 4 out of 6 studied offices the mean measured temperature matched with the occupant's expression of feeling warm or neutral while there is a discrepancy between those in 2 studied offices.

Table 2: Statistical summary of mean measured temperature and mean perceived thermal sensation of all studied offices.

Buildings	**Mean measured temperature °C**	**Mean perceived thermal sensation**	
Building No1	29.3	2.00	warm
Building No2	25.1	0.13	neutral
Building No3	27.0	-0.33	neutral
Building No4	24.8	-0.24	neutral
Building No5	24.9	0.48	neutral
Building No6	27.0	0.00	neutral

Comparison to ASHRAE Standard 55 and ISO 7730 Standard

The research also has assessed the buildings compliance with the acceptable thermal environment for general comfort of ASHRAE Standard 55 and ISO 7730 standard. The acceptable range is -0.5<PMV<+0.5 and PPD<10%

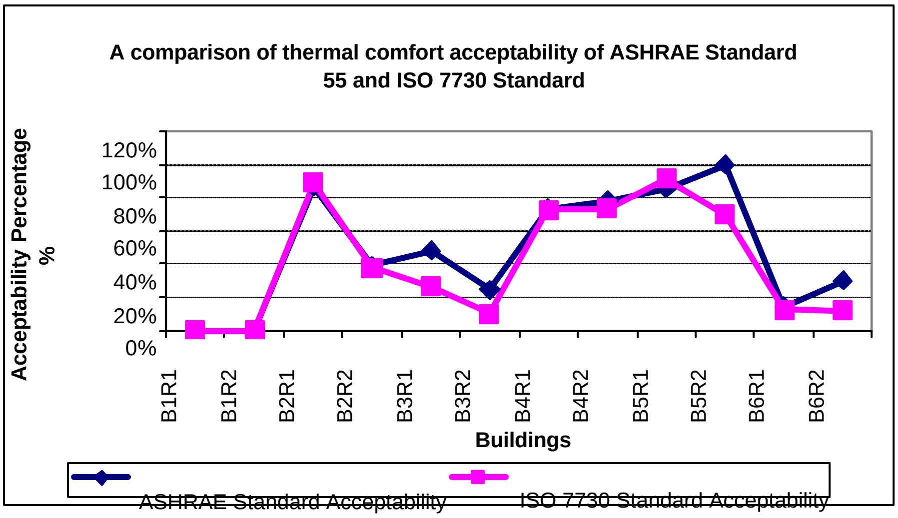

Figure 7: Compliance with ASHRAE Standard 55 and ISO 7730 Standard in all studied offices.

The results derived from figure no shows that 7 out of 12 cases are incompliance by 49% of monitored time and under, while only 5 of the cases are in compliance by 69% and over of monitored time during the working period.

CONCLUSIONS

- Only around 22-25% of the occupants report being thermally comfortable during the heating and cooling seasons in Iran.

- The research has found that there is a good agreement between mean measured temperature and mean perceived thermal sensation in two third of studied offices which indicates that in majority of the offices there was a direct link between the expression of thermal sensation as warm or neutral and the mean measured temperature ranges (table 2) (e.g. the occupants tend to express warm when the mean measured temperature was above the ASHRAE Standard 55 comfort range).
- It has also shown that there is good agreement between perceived mean thermal sensation and mean PMV in 4 out of the 6 office buildings studied. This underlines the legibility and reliability of the occupant perceived and predicted thermal sensation findings in the questionnaire survey and physical
- Due to non existence thermal comfort regulations in Iranian offices the research has assessed the offices within ASHRAE Standard 55 and ISO 7730 Standard defined comfort zone. Therefore, the comparison to ASHRAE Standard 55 and ISO 7730 Standard reveals that 7 out of 12 cases in studied offices are incompliance by only 49% and under of the monitored time during working period.

These overall findings suggest that it may prove beneficial to develop thermal comfort regulations for Iranian offices based on the ASHRAE and ISO standards, which would not only increase the occupant's satisfaction and their perceived performance but also potentially result in the reduction of energy consumption in the offices.
Further work is being conducted to assess how the thermal performance of these offices might be improved, thus achieving better thermal comfort and reducing energy consumption.

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Office workers' feedback on the control of office indoor environment

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SUMMARY

Energy efficient behaviour was studied by a questionnaire addressing office workers altogether in 34 office buildings in Finland, Sweden, the Netherlands, France and Italy. A Finnish occupancy study of the possibilities to adjust working environment and thus gain better working efficiency offered comparative reference results on similar themes.

The possibility to control personal working environment was most important for the office workers. The qualitative building automation had a positive influence on the feedback regarding workspaces. Control possibilities which take account of the dominant nature of the nature of work at workplaces demand for different solutions even on the same office floor. The gender differences dominate in the attitudes towards the environmental design, towards the control possibilities in particular. While the occupancy studies turned out to be a feasible asset for identifying the technical qualities, equalling the physical measurements, should the facilitation of workplaces follow the different needs of the female and male workers for satisfactory end-user feedback? The performance of building service and automation installations should be separated from other strongly influencing work organisational factors.

INTRODUCTION

To understand how to facilitate the indoor environment is not only a technical design task. It is constantly challenged, latest, by the knowledge workplace paradigms. The interplay between the workplace design, end-user satisfaction and the quality of structures and building service technology is a multidisciplinary design task of workplaces [1], [2], [3], [4]. A big challenge in the workplace design is to separate the organisational, physical and economical factors from each other in theory [5], and to learn how to measure them. In the international context, also national and cultural factors contribute to the research results [cf. 1].

The key focus in the EBOB project was on combining the human and social perspectives with advanced modern control and ICT (Information and Communication Technologies) solutions [6]. This was done in order to make energy efficient behaviour natural, easy and intuitively understandable for the end-users, and at the same time to achieve the most energy efficient solutions while improving the standards of indoor comfort in refurbished and new office buildings. According to this combining approach of human perspective and available new technology the results gained were called 'forgiving technology'. The EBOB project highlighted that misunderstanding of end-users about how the HVAC systems work, e.g. not seeing the relationship between indoor air temperature and energy used in different weather conditions explains the unsuccessful energy saving efforts.

The EBOB project studied the office environment as a whole and the forgiving technology in particular. The main concern was energy efficient behaviour and parameters influencing it.

The quality of the building properties and technical installations were expected to have influence on the evaluation of the workplace. This was studied by the EBOB interviews as well as on basis of previous research [2]. The best ranking obtained from the end-user evaluations was dependent on the gender and organisational factors, mainly on the organisational hierarchy which go-effected with gender factors [2], and also the workers possibility to influence the design of the workspace and its usage. Furthermore, best ranking was gained also by such building qualities as spaciousness and large building mass [2], of which the last mentioned is often considered by experts as an inhuman design scale. The latest experiences on the facilitation of qualitative office indoor air shows that the personal workspace area should not be smaller than 15 m^2, because otherwise the end-user might experience the indoor environment noisy or unpleasant due to too high velocities or imbalanced thermal conditions despite of the fact that the outcome of the correct normative indoor air design criteria has been successfully commissioned [8]. However, larger ducts and more sophisticated air-conditioning machinery than presently normally used, because of lean finance, could help in hitting the goals for end-user satisfaction. The post- or pre-occupancy studies often prove either the need for good indoor air or the lack of it even in the high class intelligent office buildings [8], [9]. A qualitative functional design of layout, interior and use of the ICT is as important as the proper functioning of the building service and automation facilities and maintenance as well as the use of control units and protocols of high standard.

In planning service systems and envelopes for buildings an important issue is the location of the building (country, latitude, availability and use of certain materials and technologies). It determinates the design criteria of indoor environment and interior design as well as how the building control technology should be operated although the building automation technology itself is universal.

METHODS

During the EU 5th FP project, EBOB, Energy Efficient Behaviour in Office Buildings (www.ebob-pro.com, NNE5-2001-0263) a web based questionnaire was addressed to office workers altogether in 34 office buildings in Finland, Sweden, the Netherlands, France and Italy (735 respondents). Another separate web-based questionnaire was aimed at the building managers, but finally the managers were interviewed face to face by the researchers in almost all participating countries. The two studies are hereafter called as the office workers' and the building managers' interviews. Information on the quality of the building properties and installations was retrieved from eleven of the studied office buildings and in all other countries except in France (Table 1).

Altogether 735 respondents, 291 men and 399 women (45 respondents did not answer the question of gender) filled out the office workers’ interview consisting of 135 questions [1]. All educational levels from primary school education to university graduates were represented. Over 62 per cent of the respondents were graduates or students either with university or with other high-level education. A share of 18.6 per cent of the respondents was managerial, 51.2 per cent professional or technical and 30.3 secretarial or clerical personnel. For the share of 5.6 per cent the working status remained unidentified.

At first, the end-user feedback on the workspace and indoor air quality was analysed [1]. The analysis of the results of the managers' interview studied both the quality of building properties and installations and the correlation of the building quality to the end-user feedback on the office workers' workspaces and indoor air quality [11]. Only after combining the results from both interviews it was possible to check whether the quality of the building properties and installations and the end-user feedback correlate, and how strongly. This was the methodological challenge of the study. The results from both interviews have been analysed using scientific statistical testing (by SPPS program).

The main concern of the EBOB project was the empowerment of the office workers' energy efficient behaviour. Such factors as needs, attitudes and possibilities of adjusting working environment and thus control energy efficiency and respondents' demography (working status, gender, age) were compared to the quality of the building installations and properties, such as the building age and the possible renovation as well as the quality of building services, the existence of building automation and personal control possibilities, type of maintenance service (in-house or out-sourced), occupancy, the size and type of personal workspaces. The final analysis provided knowledge of the specific characteristics of the buildings concerned, putting the focus on the use of control technology.

The Intelligent Buildings Survey (IBs Survey), carried out in twelve office buildings with 534 office workers in the Helsinki metropolitan area in Finland in 1994, used a questionnaire with 417 parameters and studied office workers' evaluations of similar properties of the office buildings than the EBOB interviews (in 2003) [2], [6]. During the IBs Survey the office workers were asked to evaluate the performance of their workplace to their working efficiency. The similarity of the themes opened up the possibility to use the results from the IBs Survey as references to this analysis of the EBOB interviews.

Table 1. The building properties and installations [11].

COUNTRY		**Sweden**		**the Netherlands**				**Italy**		**Finland**		
Building number		17	29	23	24	25	27	20	21	30	31	32
Construction year		1983	1984	1996	1970	1990	1996	1960	1978	1976	1984	1988
Gross floor area (1000m^2)		26.6	7.0	8.0	2.0	2.3	1.1	30.3	10.9	1.5	0.4	1.2
Layout (percentages of the gross floor area)	Single rooms	50		50	50	50	30	20	30	10	10	30
	Double room	0		40	50	50	70	60	40			
	Open floor	11						10	20	64	65	50
	Flexible workspace	0						10	10			
Number of floors (ground floor ≡ 1. floor)		6	5	4	3	3	4	6	9	1	1	2
OCCUPANCY												
Number occupants		680	200	350	75	59	40	192	500	81	18	62
Average personal workspace area (m^2)		22		23	27	**38**		14	**4**	14	17	15
MAINTENANCE[3]												
In-house		Yes	Yes	Yes	No	No	No	No	No	No	Yes	No
Out-sourced		Yes	No	Yes	Yes	Yes	Yes	Yes	Yes	Yes	No	Yes
BUILDING AUTOMATION[3]		1	2									
Regular energy consumption registration		Yes	Yes	Yes	No	No	Yes	Yes	Yes	No	No	No
Indoor air control		Yes	Yes	Yes	No	Yes	Yes	No	Yes	Yes	Yes	Yes
Personal room temperature control		Yes	No	Yes	Yes	Yes	Yes	No	No	Yes	No	Yes

[1] BMS-system?, [2] Uses Vitec ESS200 for monthly registration, [3] Yes ≡ existing, No ≡ not existing

The building properties and installations continued [11].

COUNTRY		Sweden		The Netherlands				Italy		Finland		
Building number		17	29	23	24	25	27	20	21	30	31	32
HEATING		[1]										
Construction year		1983[5]	1984	1996	1970	1990	1996	1984	1978	2001	1984	1988
Distribution type (percentages of the gross floor area)[7]	Air								5			
	Radiator; convector		100	100	100	100	100	90	5	100	100	100
	Fancoil; induction	[2]40						10	90			
COOLING												
Construction year		1983[5]	1984	1996	1970	1990	1996	1984	1978	2001	1984	1992
Distribution type (percentages of the gross floor area)[8]	Air							15	5			
	Fancoil; induction		[4]100						95	100	100	100
	Climate ceiling	[3]61										
VENTILATION												
Construction year		1983[5]		1996	1970	1990	1996					
Type: Balanced, mechanical ventilation, with heat recovery[6](per cent of gross floor area)		100		100			100			100	100	100

[1] District heating, [2] 100%/0 % Total 40 %, [3]Cooling beams: 0 %/100 % Total 61%, [4] Refrigerating compressor with heat recovery, [5] Renovated in 2001, [6] Other types asked were: natural, mechanical exhaustion, balanced mechanical without heat recovery, [7] Climate ceiling and floor heating also asked, [8] Floor cooling also asked

COUNTRY		Sweden		The Netherlands				Italy		Finland		
Building number		17	29	23	24	25	27	20	21	30	31	32
LIGHTING												
Type[3]	Fluorescent tube, conventional	[2]97	50					99	100	30	100	100
	Fluorescent tube, electronic ballasts	[1]1	[3]50	100	100		100			70		
	Halogen	[1]1						0.5				
	Light bulb	[1]1						0.5				
Light switching type[3]	Central	-	100						20			
	Building section	-								90	100	100
	Room	100	100					100	80	10		
	Presence detection	-	No	Yes	No		No					

[1] Some boardrooms, [2] All offices, [3]With HF-fitting, [3] Percentages of the gross floor area

THE EFFECT OF THE OFFICE LAYOUT

Conclusions regarding workplace design stated that the type of personal workspace matters on the experience of the workplace; a single room turned out to be a favourite for most of the office workers, a double room the worst of all types, and the open area office space a fairly neutral alternative. Conclusions from the IBs Survey were similar [3]. The interplay between workplace design and building automation turned out to be difficult in shared spaces because the indoor climate could not be adjusted according to personal needs, which meant poor personal control possibilities over indoor air and possible thermal discomfort, as well as even other inconveniences. This is the case of the double room offices in particular. In two cases out of three, the open area office space was found as a good solution regarding indoor air quality, which meant that the building automation systems were working also well enough.

AGE OF THE BUILDINGS AND THE EFFECT OF RENOVATION

The feasibility of the building automation could be presumed to be dependent on the age of the building as well as on the maintenance of the equipment. However, the age of the building did not necessarily play the main role, but the renovation turned out to be worth of trouble. The thermal comfort and indoor air quality in general were satisfactory in all renovated buildings. The end-user feedback on the work environment in general was poor regarding the old buildings and a little worse in the new ones than in those, which have been used for a while.

One could presume that after using a building for some years 'the things have settled' and the users were used to the quality of the building as well as had learned how to best benefit out of the physical environment. For confirmation, more accurate knowledge of the effect of the age of the building was gained from the evaluations. In addition, those buildings which were used for some years were occupied by a relative large portion of male professionals who tend to be rather satisfied than critical on the physical work environment. They have also good controlling possibilities over their working environment, which is an important advantage in making ones work environment comfortable. In those offices which were used for some years and had the majority of female workers − who tend to be critical on indoor environmental condition, and which still got a good end-user feedback, the women worked in the most favourable spatial arrangements of single rooms.

In addition to the age other obvious reasons for the low valuation of the old office buildings were found. In the oldest buildings with poor end-user feedback, the number of clerks was largest while in all other building age categories the modern diamond organisation structure[1] dominated. In addition, in them worked mostly women, who in general tend to be critical on indoor environmental conditions and they worked in rooms shared by several workers which has been evaluated as an unfavourable spatial arrangement.

THE BUILDING AUTOMATION OVER INDOOR AIR

The workers were more satisfied with the physical working environment in general and with the indoor air quality in their own workspace when the automation controlled the indoor air than if there was no control. The building automation over indoor air guaranteed that the air conditioning was not considered to run too efficiently during the summer time and workers did not feel cold indoors. Also, almost all those who worked in the offices without control technology (90%) were satisfied with the indoor air in the summer. However, only in the buildings with the control automation of the indoor air, the workers felt that the indoor air was too cool in the summer, in other words the air conditioning was running too efficiently.

PERSONAL TEMPERATURE CONTROL

In case of personal temperature control, [1)]the workers were more often some more satisfied with the temperature in their personal workspace than without the control possibility, and [2)] they did not feel too hot, not even in the winter time when the heating was evaluated too efficient if the building was without the personal temperature control technology. The personal workspaces were evaluated rather too hot than cool during the IBs Survey [2] as well, in both the intelligent and the other high quality offices. The feeling of hot was not as

[1] The majority of the current office workers are professionals while earlier they were clerks. Office organisation is called as a diamond organisation when there works as a small amount of clerks as executives and majority of professionals.

remarkable between these two control alternatives (personal control or not) in the personal workspace in general as it was when considering the winter situation alone.

However, [3)] those who had personal temperature control units felt either cool or not in their personal workspace. In the sophisticated office buildings with building automation and personal control units the indoor temperatures can be too cool, in the summer particularly.
According to the IBs Survey, women have fewer possibilities than men to influence their work environment although the quality of it seems to be more important to women than to men [2]. Women also spent more time in the office buildings than men did, in other words they are heavy users of their space, which might make them more demanding than men for the quality and performance of the building properties and installations. In parentheses, men work more often out of offices than women, but both genders work as long hours and for example work overtime as often and as much.

The male dominance at workplaces is also due to the fact, that men more often than women have a higher position in the hierarchy of the organisation. During the EBOB interviews it was found that he personal temperature control possibility was welcomed by all office workers, but it became statistically significantly remarkable when the gender aspect was taken into account. The EBOB interviews revealed the need for better control possibilities over working environment among the female office workers, now within the European context.

During the analysis of the EBOB office workers' interview Vastamäki et al [8] also found out that the gender played a statistically significant, even highly significant role over several aspects of the indoor climate. It was related to importance of control over ventilation, air humidity, air motion (air flow), indoor temperature (in all of these $p<0{,}001$), windows ($p=0{,}021$) and over energy saving aspects ($p=0{,}030$). In all of these aspects, the personal control was more important for women than men. However, the effect size was quite small. Only 3 % of the variance of the control over ventilation can be explained with gender. This was the item on which the gender effects were most relevant.

Interestingly though, the personal control possibility of the temperature in the own workspace was rather making the women more dissatisfied with the temperatures. As women more often than men work in the two-person rooms or rooms shared by even more workers, they might think that the existence of control possibilities does not result to a good indoor air quality anyway, which would explain the illogical result. Another explanation could be that despite the availability of the control units, they might not use them or not at least for their own benefit. This type of situation of unused possibility can cause extra frustration, which fact was revealed in a North American office environment study.

The IBs Survey proved [2] that the executives were most satisfied, professionals next and clerks least satisfied with their working environment. If in the analysis the gender is taken into account together with working status, the results are somewhat different. In general, women are more demanding in terms of spatial solutions and indoor air quality than men and the quality of technology concerned. However, variations appear in the end-user feedback. For example, of all studied office workers male clerks and female executives were the most satisfied groups with the office automation systems.

Female workers have been considered critical in terms of working environmental conditions for example because of their dominance in effective factors of sick building syndrome. Men turned out to be demanding in terms of the quality of technology, especially if the evaluation involved mobile technology [2]. If modern offices function more and more on technology, the men instead of women might become an important critical group in the end-user evaluation or the user requirements studies.

According to the EBOB studies on average the workers were more satisfied with the physical working environment in general if the personal control of the temperature was in use than if it was not. However, this was not the case in the IBs Survey [3, pp. 216-220], which found no influence of the personal control facility to the end-user satisfaction in the buildings. The selection of buildings caused the difference in the results. The EBOB interviews were comparing buildings rather with qualitatively very different than similar building service facilities and other building qualities. The selection criteria for the buildings compared during the IBs Survey was more selective. Best quality was the prime selection criterion. If only possible, pares of buildings were picked for comparison; the age, the size and the purpose of usage, etc. were as similar as possible in both categories: the intelligent and the other high quality offices.

Furthermore, the IBs Survey concluded a zero effect of the personal cooling unit to the end-user satisfaction of the indoor environment also in the intelligent office buildings, because the quality of the personal workspace in general was evaluated there poor and the possibility of the co-effective factors was most probable. Also, the location of the control unit was not always the best possible and the occupants did not know well enough of the control possibilities. The indoor air quality of all buildings within the IBs Survey was surprisingly poor for various reasons [3].

Nevertheless, the positive average result of the EBOB interview in general was not the whole truth. When looking the buildings separately as done during the IBs Survey, it can be concluded from the three buildings equipped exceptionally well in terms of the building services that in one out of the three the end-user feedback was good, in the two others not so good.

Finally, the results of the international EBOB interviews were exposed to the influence of the national and cultural norms, while the IBs Survey was a national case study. Seemingly relevant national differences were gained during the analysis of the EBOB workers' interview [10]. Regardless, no attention was paid, how, or if at all, the end-user evaluation was affected by the selection of the cases for the sample or if the user requirements could depend on behaviour and motivation as shown for example by the Malows' Hierarchy of Needs [3], [6]. Does indoor environment satisfy such basic human needs of shelter from weather, which are common to all of us without any cultural differences?

In both case studies it could have been paid more attention to the selection of the buildings also on basis of such factors as the shape of the lot, the orientation of the building, the dominating wind directions, or for example, the size of window area and the facing of windows to the cardinal points, the thermal weak points of the envelope.

Nevertheless, according to the EBOB interviews the personal control possibility kept the office workers more often satisfied with the temperature of their own workspace than if that possibility was not available. The control over the overheating in the winter was better with the personal control possibility than without it. The personal control possibility did not prevent from too low temperatures in the personal workspace. This result was based on the building specific comparison with minor influence of the selection of the sample.

INDOOR ENVIRONMENT OF KNOWLEDGE WORKPLACES

Interestingly, no strong correlation, it is, the non-existence of the statistically significant differences was found between the end-user feedback on the indoor environment and the feasibility of the building service systems or the standards of the existing building automation equipment excluding the gender aspect. A similar contradictory result yielded from the IBs Survey [2] as well. The research concluded that the building automation was not particularly sophisticated in the intelligent office buildings compared to the equipment in the other office buildings. However, the end-user evaluation of the technology was better in the buildings built using the intelligent building concept as design criteria than in those offices which were built according to other high design criteria parameters. Either way, the conclusions of these two studies were the same, no correlation between end-user opinions on the indoor environment quality − which is the outcome of the building services − and the standard of the building service and automation facility. In the EBOB study, the thermal comfort and the indoor air quality were not statistically significantly better despite the sophisticated building automation installation. However, the building automation made the buildings more acceptable for the office workers according to the feedback of the EBOB interviews. In general, the high technology building automation was worth of having. The paradigm of co-effecting factors did partly explain the results.

The nature of work is said to become ever more mobile. Mobile work means that working hours are spent out of the office, and also that the workers go out and come in several times a day. Such a situation is very challenging to the design of building service systems while the incoming person can feel the impurities of indoor air easily. On the other hand, the IBs Survey proved that people worked longer hours in the intelligence offices than in the other type of high quality offices. The situation is challenging regarding quality of indoor air climate, also when workers stay long hours indoors. A person cannot sense the bad air quality by smelling but might get head ace if the indoor air is not fresh enough or have allergic reactions due to the impurities in the air.

To make the workplaces suit better for the knowledge work than they do today, the separation of different forms of influencing factors could help. How the organisational and technical factors could be separated from each other when studying the modern offices and the current office automation three different cases were found:

- The case of positive co-effecting factors: A case of several positive organisational factors together with an efficient air conditioning and a recent renovation ended up to a good evaluation of both the thermal environment and the indoor air quality in general. There was no sophisticated building automation installation, but the dominant spatial arrangement of open area office suited well for the workers.
- The case of good technology destroyed by several negative co-effecting factors: A case with a sophisticated building automation installation together with the several negative organisational factors ended to a too efficient air conditioning. Furthermore, the dominance of double rooms caused the need to open the windows; and only once the thermal conditions and the air quality in general was evaluated as good. Neither any intelligent feature could save the building from bad feedback nor could it substitute and mitigate the unsatisfactory quality of the building properties and installations.
- The case of negative co-effecting factors: A case of several negative organisational factors in an old building with the dominance of double rooms and without any sophisticated building automation installation ended to a poor valuation of both the thermal environment and the air quality in general.

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Indoor Airflow and Human Thermal Comfort Research on One of the Chongqing Air-conditioned School Dormitory in Winter with CFD Simulation

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SUMMARY

Currently, numerical simulation research on indoor air and human thermal comfort usually focuses on the air-conditioning room in summer. This paper uses an Air-conditioned dormitory in winter as a model to study indoor airflow and human thermal comfort. This paper uses $\kappa-\varepsilon$ three-dimensional turbulence model and N-S equation, considers airflow with room shape and obstacles as one, calculates the indoor airflow and heat transfer problems with overall solution, studies indoor air form especially velocity field with numerical simulation. From the numerical simulation results, we can know that the layout of the school dormitory and air supply mode is appropriate, the air velocity and other parameters meet the relevant standards, and conducive to obtain comfortable indoor thermal environment. The results prove that $\kappa-\varepsilon$ three-dimensional turbulence model is suitable for calculating indoor airflow field, and can be used in other air-conditioned rooms with different structure in winter.

INTRODUCTION

Computational fluid dynamics, shortened as CFD, is a new subject which appears with the popularization of computer science. According to reasonable boundary conditions and parameters provided, it can apply the basic theories of hydrokinetics to build mathematical physical models to carry out analog computation of the air stream's velocity field, temperature field, pressure field, etc in the air conditioned area. And the room air's velocity field and temperature field is the basis for the design of the air-flow organization in the air conditioned room and the basis for the evaluation of the room pleasantness.

At present, domestic researchers mainly focus the room air research on the numerical simulation of the air conditioners' refrigeration in summer, but in this paper I will take the teachers and staff's dormitories in one university in Chongqing as a model to make a numerical simulation of the air flow in a room heated by air conditioner in winter. By using $\kappa-\varepsilon$ three-dimensional turbulent air stream model and N-S mathematical equation, this paper took overall approach to compute the problem of air flow and heat transfer in air conditioned rooms, it also made a numerical simulation of the air stream esp. the velocity field, considering the inlet air stream, room shape and obstacles as a whole.

At present, the approaches for the room air flow distribution fall into three groups: I. To make use of the jet flow theory to make analysis and prediction; II. To make use of similarity principle to make model experiment; III. To make use of the numerical prediction approach by solving room air control equations set with computer. The first approach is rarely used due to its limitations. As for the second approach, a great number of material resources, financial resources and labor were wasted. And in many cases, we have certain difficulties in measurement techniques, like the measurement of the direction of the low speed air stream and the measurement of overflow. With the development of computer and computing techniques, the numerical prediction approach grows rapidly.

This paper takes the room air flow in winter of the teachers and staff's dormitories in one university in Chongqing as the object of study. On considering the dormitory's outside temperature (in accordance with its location), its exterior protected construction and its heat engineering performance, the refrigeration of the air conditioning system, the heat dissipating capacity of the indoor personnel, etc., we make the numerical simulation of the room air flow.

MODEL STRUCTURE

On considering the practical situation of the dormitory, to build a dormitory model which is 5.2 meters long, 3.2 meters wide and 3.2 meters high. As the planar construction diagram in Diagram 1 shows, in the dormitory there are some obstacles like a desk (which is 1.2meters long, 0.5meter wide and 1 meter high) and a bed (which is 2meters long, 0.9meter wide and 0.5meter high), and personnel as well (only considering the influence of the personnel's heat dissipating capacity on the air conditioners' refrigeration duty.

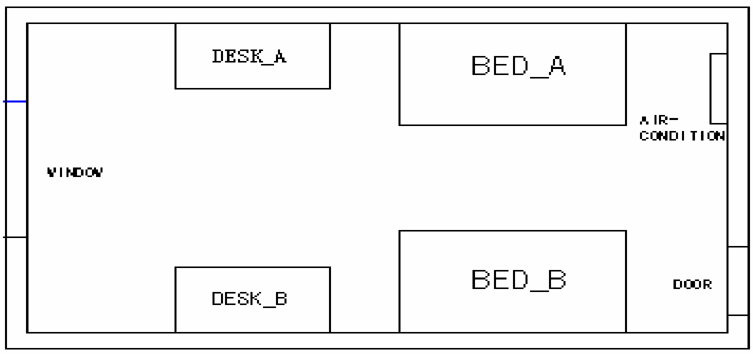

Figure 1. Plane chart of the dormitory

MATHEMATICAL MODEL IN THE TURBULENT REGION

At present, two turbulent analogue approaches are used in the computational fluid dynamics circle: one is Direct Numerical Simulation (DNS); the other is the approach of overflow flattening. Since the scale of the turbulent flow is 0.1μm and because of its important particulars, extremely thin grid spacing is needed for numerical solving, which is beyond the capacity of the present computer's memory and CPU. In regard to engineering problems, what we should concern is not the overflow's fine structure and its transient variation, but the related mean value of its extraneous variables, so overflow flattening and a series of overflow models are widely used in engineering. In the field of heating and ventilation, $\kappa-\varepsilon$ model is popular.

The turbulence partial differential equations describing the model room are as follows:

Continuous Equation

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho u_j)}{\partial x_j}=0\,, \qquad (1)$$

Momentum equation

$$\frac{\partial(\rho u_i)}{\partial t}+\frac{\partial(\rho u_j u_i)}{\partial x_j}=\frac{\partial}{\partial x_j}\left(\mu\frac{\partial u_i}{\partial x_j}-\rho\overline{u'_j u'_i}\right)-\frac{\partial p}{\partial x_i}+\frac{\partial}{\partial x_i}\left(\mu\frac{\partial u_j}{\partial x_i}\right)+\rho g_i\,, \qquad (2)$$

Energy equation

$$\frac{\partial(\rho h)}{\partial t}+\frac{\partial(\rho u_j h)}{\partial x_j}=\frac{\partial}{\partial x_j}\left(\frac{\mu_t}{\sigma_k}\frac{\partial h}{\partial x_j}-\rho\overline{u'_j h'}\right)_i, \qquad (3)$$

Component equation

$$\frac{\partial(\rho m_l)}{\partial t}+\frac{\partial(\rho u_j m_l)}{\partial x_j}=\frac{\partial}{\partial x_j}\left(\frac{\mu_t}{\sigma_l}\frac{\partial m_l}{\partial x_j}-\rho\overline{u'_j m'_l}\right)+S_i\,, \qquad (4)$$

There are three more transients in the turbulence models above than in the layer equations- Reynolds Stress $\rho\overline{u'_j u'_i}$, Reynolds substance flow $\rho\overline{u'_j h'}$ and Reynolds heat flow $\rho\overline{u'_j m'_l}$, which are all unknown items. Thus, different approaches to these transients lead to different models.

The modeling is that three transients above are presented by time averaged value. Then, according to Boussinesq's theory as well as with reference to the methods of laminar flow, the three transients are presented to time averaged grads function:

$$-\rho\overline{u'_j u'_i}=\mu_t\left(\frac{\partial u_j}{\partial x_i}+\frac{\partial u_i}{\partial x_j}\right)-\frac{2}{3}\rho k\delta\,, \qquad (5)$$

With the molecular transport process compared, we can know that the eddy impulse kinetic energy "k" and the length of turbulence "l" have an important impact on the turbulence transporting course. Afterward, by using algebraic or experiential formula and the way of linking k, l and the known quantity, we can make sure of μ_t At last, the turbulence viscosity coefficient models can be classified into such types as zero equation model, one equation model and double equation model in terms of the number of the differential equations which need to be solved.
The modeled equation k:

$$\frac{\partial(\rho k)}{\partial t}+\frac{\partial(\rho u_j k)}{\partial x_j}=\frac{\partial}{\partial x_j}\left(\frac{\mu_t}{\sigma_k}\frac{\partial k}{\partial x_j}\right)+\mu_t\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)\left(\frac{\partial u_i}{\partial x_j}\right)+\beta g_i\Gamma_T\frac{\partial T}{\partial x_i}-\rho\varepsilon\,, \qquad (6)$$

The modeled equation ε:

$$\frac{\partial(\rho\varepsilon)}{\partial t}+\frac{\partial(\rho u_j\varepsilon)}{\partial x_j}=\frac{\partial}{\partial x_j}\left(\frac{\mu_t}{\sigma_\varepsilon}\frac{\partial\varepsilon}{\partial x_j}\right)+\frac{\varepsilon}{k}\left(c_{\varepsilon 1}\mu_t\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)\left(\frac{\partial u_i}{\partial x_j}\right)-c_{\varepsilon 2}\rho\varepsilon\right), \qquad (7)$$

A set of close equations is composed of the current-sharing control equations, the modeled equations of k and ε, and the definition formula of the model equation (5) of the turbulence transporting flux.

As a result, in the turbulence zone simulation of this research, the velocity field of the turbulence zone has been worked out by iteratively computing every grid with an implicit differential method.

BOUNDARY CONDITIONS

Solid walls, windows and doors are boundary conditions for the dormitory flow field. The prototype of the dormitory is one of the central rooms in the fifth floor of a dormitory building. Its door towards the corridor is set to be closed while the windows is also closed, which do not obstruct the airflow though some leakage from the doors and windows. Due to the Chongqing's cold winter, the temperature of the walls is set up to 6 ℃ Otherwise, there's no fluid infiltration and friction on the wall surface, no heat transfer and airflow through the desk which would obstruct the indoor airflow. Suppose there are two people inside with an average heat release of 80W each (the indoor air computing temperature assumed to be 10 ℃.) Two boundary layers are considered in this simulation computation: One is outlet boundary layer, the other is wall boundary layer.

Outlet Model

The combination of the air supply and the indoor air is usually realized by outlet air jet for the purpose of air-condition and ventilation in the air-conditioned rooms.

Because of the complex actual geometry of the outlet and various types, we must carve it up into mm grids if we want to know the details of these parameters of outlet air supply, which is doomed to cause a huge number of grid nodes in the calculating region and slow computing, unable to be accepted by project applications.

Therefore, a way of model simplification is used to take the average speed. A 1.2m×0.6m inlet replaces the rectangular outlet, 2.3m above the ground, in the direction of 45° oblique. Then, compute the temperature of air supply according to the criteria for air-conditioning and the amount of air supply according to the indoor ventilation area and the number of the indoor people. It's computed that the temperature is 25 ℃ and the outlet airflow velocity is 0.5m/s (the air supply meeting the minimum requires computed).

Wall Boundary Layer

Generally speaking, there are two ways to deal with the wall boundary layer: one is applying the model of the turbulence core to the boundary layer; the other is to establish the wall boundary layer, which is dealt with separately from the core area. However, there are two preconditions for the former way- first, the model of the turbulence core must be applied to the descriptive boundary layer; second, the boundary layer should be divided into more detailed grids to ensure the accuracy of simulation-which will inevitably increase the number of nodes, the consumption of RAM memory and the CPU workload. Accordingly, the latter way is adopted here.

In terms of Launder's and Spalding's theories, the boundary layer is classified as viscous bottom and the logarithmic law layer and the mathematical model of the descriptive boundary layer is as follows:

$$U^* = \frac{1}{C}\ln\left(Ey^*\right) \qquad y^* > 11.225 \qquad U^* = y^* \qquad U^* > 11.225$$

$$U^* = \frac{U_P C_\mu^{1/4} K_\rho^{1/2}}{\tau_\varpi / \rho} \qquad y^* = \frac{\rho C_\mu^{1/4} K_\rho^{1/2} Y_\rho}{\mu}$$

C_k —Karman constant, value: 0.42;

E —empirical value, value: 9.81;

U_P —velocity of grid point;

K_P —turbulent kinetic energy of grid point;

Y_P —distant from grid point to wall;

μ —viscosity coefficient of turbulence flow

GRID DIVISION

The established mathematical models are divided into grids and the grids have been refined in the computation region with great velocity gradient airflow. In the computation, first use the sparse grids and go on refinements. Since the difference between the computed results when the flow field to be computed is divided into 66×57×20 grids (indicated as Figure 2) and the results of the formerly used grids may be ignored by careful comparison, the entire flow field of the dormitory for simulation computation should be divided into 75240 grids finally.

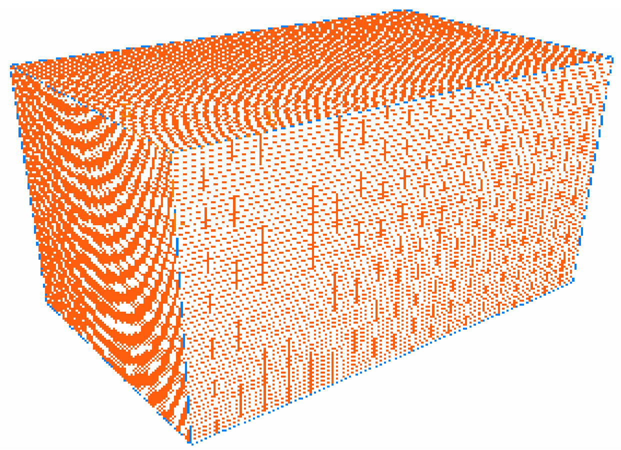

Figure 2. Gird division map of the dormitory model surface

NUMERICAL SIMULATION RESULTS

Making use of the previous mathematic and physical models, through the simulation computation of this dormitory, the organization form of the flow field and parameters such as airflow velocity inside the dormitory, etc. have been obtained(indicated as following figures). Figure 3 is the velocity vectorgraph in the XY direction 0.6 m above the ground-the body position above the ground when students sleep; Figure 4 is the velocity vectorgraph in the XY direction 2.4 m above the ground-the position of the air-conditioner;

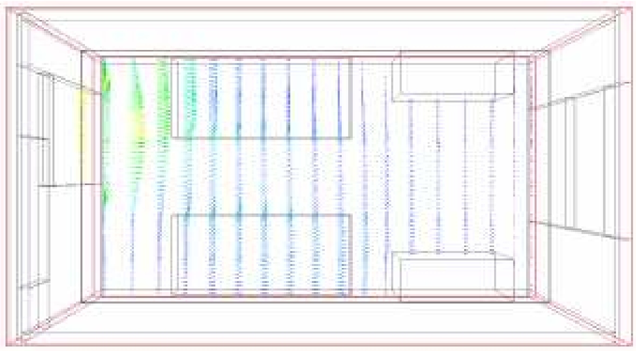

Figure 3. Velocity vectorgraph in the XY direction 0.6 m above the ground

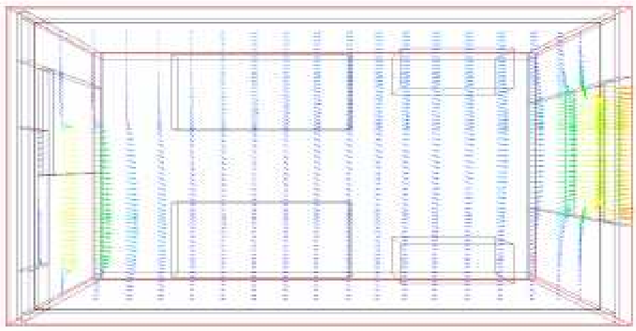

Figure 4. Velocity vectorgraph in the XY direction 2.4 m above the ground

Figure 5 is the velocity vector profile in the XZ direction above Bed A, Figure 6 is the velocity vector profile in the XZ direction above Bed B;

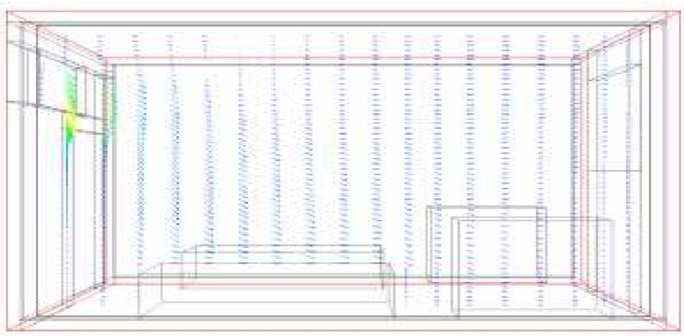

Figure 5. Velocity vector profile in the XZ direction above Bed A

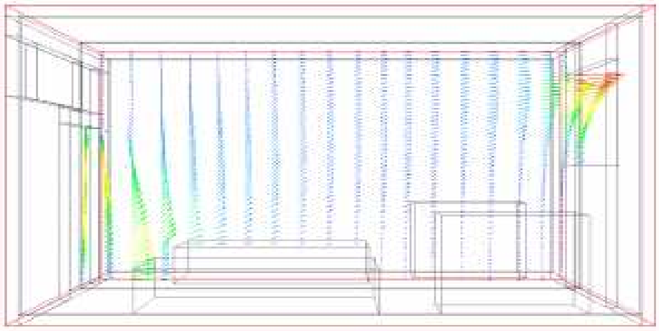

Figure 6. Velocity vector profile in the XZ direction above Bed B

Figure 7 is the flow field profile in the YZ direction between two desks-the centralized area for students' activities inside the dormitory.

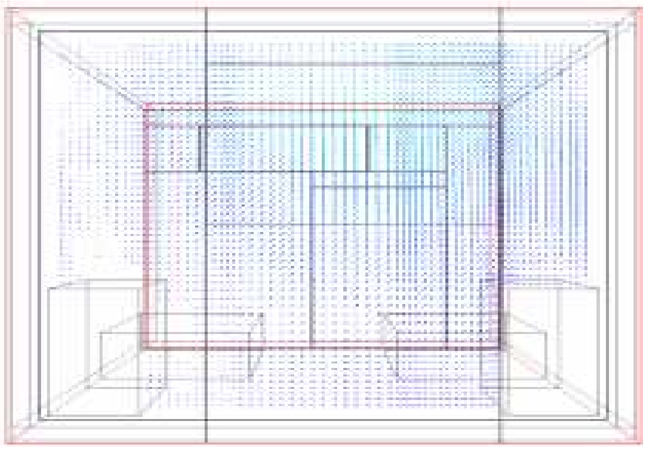

Figure 7. Flow field profile in the YZ direction between two desks

CONCLUSIONS AND ANALYSIS

Shown as above figures, the airflow velocity of the entire dormitory is below 0.175m/s(less than 0.3m/s), which meets the requirements of a comfortable air-conditioned room in winter. So the airflow organization form is reasonable for it's helpful to indoor air pollutant emissions. In addition, the way of air supply adopted inside the dorm is quite appropriate and the airflow velocity meets the relevant standards for ventilation and air-conditioning, in favor of obtaining a comfortable indoor thermal environment. Consequently, the current living conditions can satisfy the majority of college students in Chongqing area.

Besides, this numerical simulation results can be references to other indoor airflow studies of similar structures. Thus, I will keep studying those problems further such as the temperature field of the room, the pressure field and human body thermal amenity, etc. in the future.

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Climatic Adaptation: Impacts On The Thermal Comfort Of Offices Buildings At Curitiba - Brazil.

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SUMMARY

Concerns about energy efficiency, has brought new constructive trends researches. Aspects as environmental comfort is quite important, with regard to new constructive technologies, showing the need of standardizing Brazilian buildings to acquire thermal comfort. The necessity of new internal environment data in buildings has motivated surveys about it. The proposal is to present the constructive typologies of buildings and their indoor variables in Curitiba city, Parana state, South of Brazil, relating it to the Bioclimatic Chart, proposed by Givoni [1], for countries in development. This work analyzed the constructive characteristics of 39 commercial buildings and environmental variables measured in internal environment of each one. The data was separated in winter and summer being inserted in the Bioclimatic Chart equivalent to Curitiba climate. That way it has verified that constructive typologies were more suitable for the region.

INTRODUCTION

The use of new materials and technology as a way to identify enterprises of great value went beyond the good sense necessary for building new constructions. With the massive use of lighting and air conditioner equipments, the consumption of electric energy took a great deal of proportion aggravating the problem of energy supply.

Especially from the decade 1970 and with the population increase on the decade 1980 and more recently in 2001 with de problem called "a black out", the energetic situation in the country has been receiving especial attention. Due to such preoccupation, new constructive tendencies have arisen, and aspects as energetic efficiency and ambient comfort have become of extreme relevance concerning to the new constructive technologies, showing the need of bringing the construction of new buildings back to normal taking into consideration concepts as energetic efficiency hitched to the thermal comfort. This way, the need of new data and parameters about ambient conditions in constructions motivated the production of works all over Brazil.

In order to study the thermal comfort evaluation methodologies of constructions several works have been developed. According to Neto [2], the proposal norm structure is quite wide-ranging and seek not to shut the procedures for building national character regulations or yet specific regulations for States and/or regions. Janda, Katryn and Busch [3] carried out some information about sixty countries with the purpose of getting the experience obtained in normalization of many countries focusing areas where the results can be applied and developed in an effective form.

Seeking for solutions energetically efficient, it started for adopting passive measures in order to get human comfort, that is, measures which do not involve energetic wasting for it.

For this checking, it must take into consideration the local where the buildings are found, that is, its climatic region. According to Goulart [4] the climate in Curitiba, which is considered as sub-tropical, presents a unique characteristic in Brazil being the coldest capital of all, making people feel some discomfort in more than 80% of hours of the year, from the thermal point of view, mainly for cold weather.

A lot of field study was fulfilled, such as Nicol [5] and Humphreys [6], with the purpose of seeking an adaptable model of thermal comfort, based on the use of monthly average external temperature as a new parameter of comfort. The use of international parameters adapted to Brazilian regions, can reflect on a wrong way the real condition of thermal comfort.

Other studies were fulfilled taking into consideration thermal ambient conditions and bioclimatic analysis through predicted equations Xavier [7], Kruguer[8] and also the studies of energetic thermal simulation program by Ghisi and Lamberts [9].

The present study insert itself in a wider work developed in 8 Brazilian cities with the purpose of studying and carrying out some information about office buildings placed in different regions of Brazil. Some information were collected through field study regarding office building placed in Curitiba city to get thermal energetic parameters linked to the local climate, providing, consequently, alternative for a better climatic adequacy for such buildings. Through the obtaining information concerning to the constructive type predominant in office buildings in the city we have defined a typed classification for the entire constructions surveyed. It has been also studied the energetic thermal performance of the surveyed constructions, being in possession of information concerning to temperature and humidity and of each type, through measurement made with the help of equipments designed to this task.

METHODS

For the obtaining information relating to office buildings type present in Curitiba, 841 constructions with more than 5 floors were surveyed, condition imposed for the fulfillment of a survey in buildings with elevator. To carry out architectural characteristics, responsible for the type formation, some visiting were given in 39 buildings, which of them were grouped in 21 different types, considering 5 variable architectural. For experimental limitations in a second step, it was considered only 3 architectural characteristics to the bioclimatic study of types. As a result of the steps 6 typologies were considered, which ones were compounded throughout the combination among the characteristics relating to the building height, the external shading presents and the window area versus front area.

After separating the buildings in type groups, it was possible to carry out some information concerning to occupation standard of each office presents in each building representing one of the 6 typologies drew up. That way, it has been studied the activities performed in each building and it has got the relation of representative buildings in each one. For carrying out the thermal-ambient information, it was surveyed 6 representative buildings from the specific typologies. In each building it was installed equipments type HOBO H8 Family, to measure temperature and humidity in offices of a larger representative. Together with this equipment it has been used the software BoxCar Pro 4.0. The measurements were made considering the proximity with the winter and summer solstice. It has been applied a questionnaire to people who occupy the offices for them to give their opinion regard to the comfort condition.

After the measurement, it has been organized reports containing information relating to temperature changing and internal humidity of each office. Together with the external climatic information, it was made comparative tables of the internal thermal changing of the offices front to external thermal changing.

With the use of software AnalysisBio developed by Energetic Efficiency Lab in Constructions at Santa Catarina Federal University – LABEEE/UFSC, it has been made Psychometric charts, containing the bioclimatic zones proposed by Givoni [1]. Thus, such charts made possible the analysis relating to the thermal ambient performance of the buildings analyzed. This performance aims the thermal comfort with the minimum energetic consumption. Having such information made it possible to analyze the building performances in the summer and the winter months.

RESULTS AND DISCUSSIONS

The table 1 shows the representative architectural characteristics present in constructions analyzed. For the study of thermal performance it has been considered combination types of characteristics represented itself.

Table 1- Architectural Characteristics Representative

TYPES	Nº BUILDINGS	REPRESENTATIVITY
Tall building	20	51,3%
Low building	19	48,7%
With external shading	14	35,9%
No external shading	25	64,1%
PWF > 50%	25	64,1%
PWF < 50%	14	35,9%

The table 2 presents the 6 typologies considered on the energetic thermal study. The separation of typologies took into consideration the height of buildings, the presence of external shading and percentage of window for front (PWF).

Table 2 – Type Considered At Thermal Energetic Analysis

TYPOLOGY	HEIGHT	SHADING PRESENCE	PWF
1	Tall	With shading	> 50%
2	Tall	No shading	> 50%
3	Tall	No shading	< 50%
4	Low	With shading	< 50%
5	Low	No shading	> 50%
6	Low	No shading	< 50%

By the thermal ambient analysis in the summer the information reveal that, in all the typologies the results were inside the called comfort zone defined by Givoni [9]. The bioclimatic study based itself on the analysis of proposed alternative at the bioclimatic chart made through the help of software AnalysisBio.

The figure 1, presents the Bioclimatic Chart proposed by Givoni, it has already duly adequated for Curitiba City, where the rented points represent 8640 hours of the year allocated on itself. The climatic file of Curitiba used was TRY, determined for Curitiba by researchers of LabEEE [4], [10]. From these results it can be hoped that, part of the information placed itself inside the comfort zone (20,9%), some of them placed themselves into the hot region 2 and 11 (5,9%) and the rest of the points will fall down in 7,8 and 9 zones in need of some kind of heating.

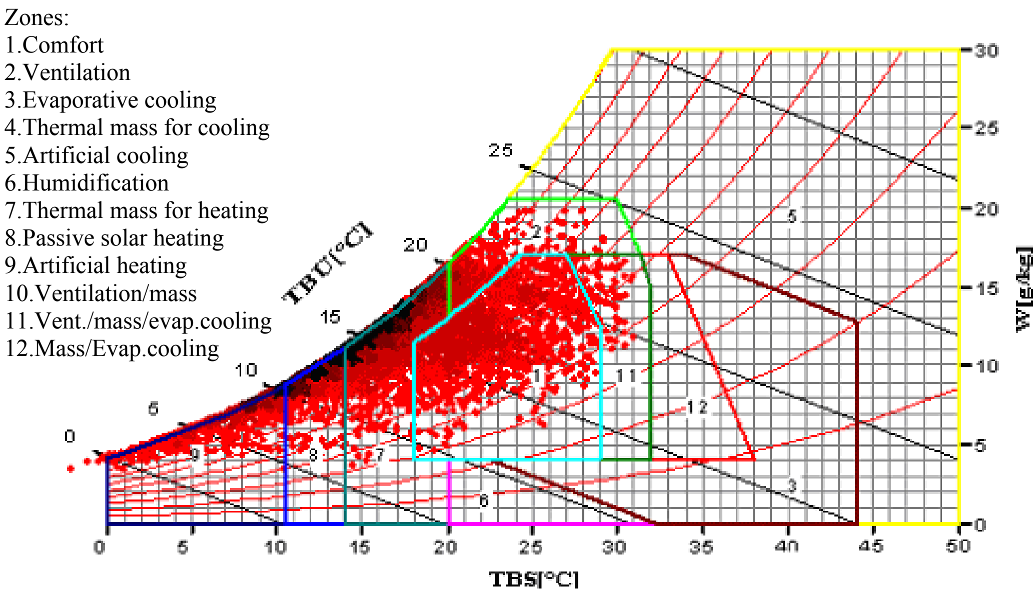

Figure 1 – Psychometric Chart for the reference year of Curitiba City.

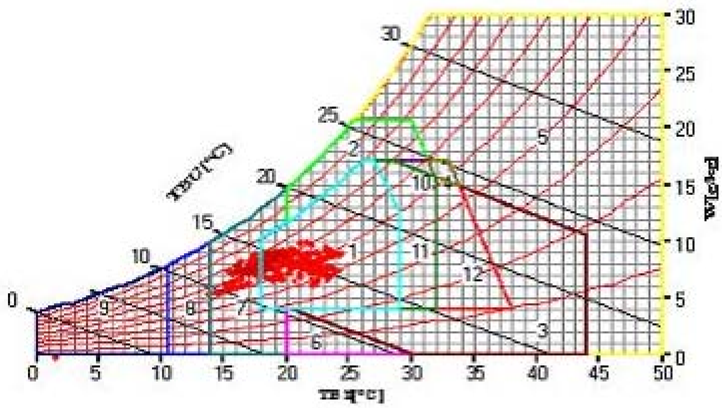

Figure 2a – Psychometric chart of type 3. Winter measurements.

The figures 2a, b e c show the bioclimatic results plotted on the psychrometric charts from 3 of the analyzed typologies. For making these charts it has been considered the temperature and humidity inside the whole offices places in this typology and it has been obtained the average of such measurements totaling 460 points. It has been plotted the information of typologies 3, 5 and 6 during the winter. For the summer almost all the interior points of the offices placed on the comfort zone.

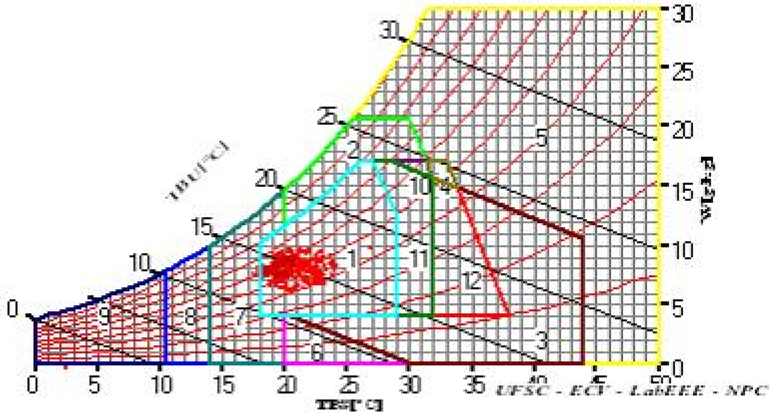

Figure 2b – Psychometric chart of typology 5. Winter measurements

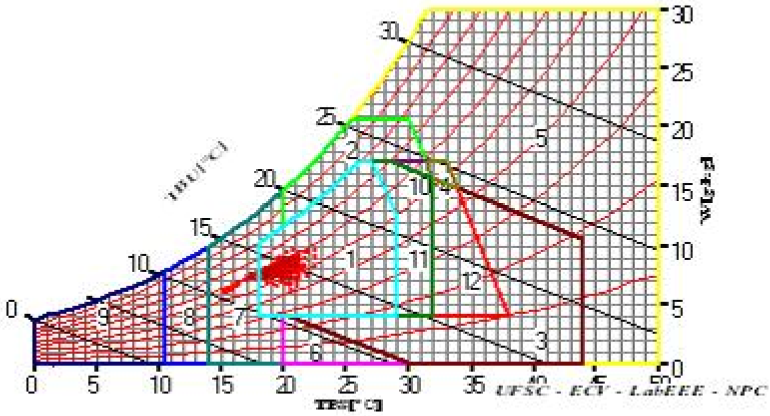

Figure 2c – Psychometric chart of type 6. Winter measurements.

Analyzing the office thermal-ambient (see table 3), is has been concluded that, the typology 3 which has the following characteristics, tall building, no shading and with less than 50% of PWF, revealed the worst thermal performance in the winter days. And the typology 6 (low building, no shading, with less than 50% of window area by front area),revealed have the worst thermal performance relating to the summer days, obtaining an average temperature of 25,9^{o}C. But the typology 5 (low building, no shading, with more than 50% of window area versus front area) which one it had been hoped to have the worst result during the summer, due to its architectural characteristics, it has obtained 100% of the information inside the comfort zone in the summer and 96,8% in the winter.

Table 3 – Typologies thermal Performance.

TYPOLOGY	SUMMER			WINTER		
	T_{med}	Humidity	Comfort	T_{med}	Humidity	Comfort
1	24,0	60,1 %	100%	19,0	59,8	84,9%
2	24,9	59,7 %	100%	19,3	57,8	91,0%
3	24,0	55,1 %	100%	18,9	57,2	60,9%
4	25,5	59,8 %	99,6%	19,8	58,6	91,6%
5	25,8	56,1 %	100%	20,4	55,3	96,8%
6	25,9	57,2 %	98,4%	19,3	55,2	82,4%

The figures 3a, b, c and d present the results of internal temperature variation comparing to external ambient temperatures, in the winter of 3 typologies during 15 days. Still analyzing the typology 3, it can be observed through graphs that its performance relating to great thermal variations showed itself relatively unwanted, because its thermal structure showed itself unable to absorb such variations in a satisfied way, showing a bad working of the envelope and its architectural characteristics. The typology 6 had a better performance and the 5 still better because even with colder external temperatures it has obtained 96,8% of the information inside the comfort zone. On the figure 3 it has been plotted together with the curve of the 3 typologies and the temperature of the external ambient in the winter. It has been seen that the typology 5 has obtained the major temperatures keeping the ambient always on the comfort zone; the temperatures of typology 3 has been the one that more accompanied the variation of external ambient.

It can be observed through the questionnaire that, relating to the climatic conditions in the offices analyzed, the thermal feelings of the people inside the offices it has driven to a less wide ranging comfort band, being considered discomfort condition, temperature variation less than the stipulated limits on the bioclimatic comfort zone proposed by Givoni [1].

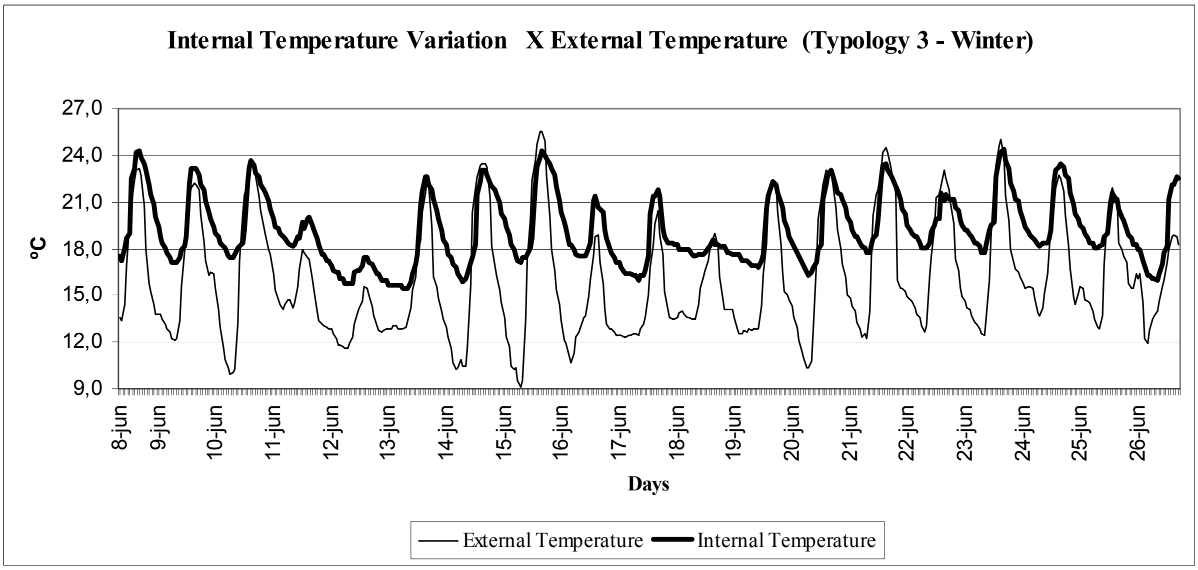

Figure 3a: Internal temperature comparison front of external temperature variation relating to typology 3 (Winter).

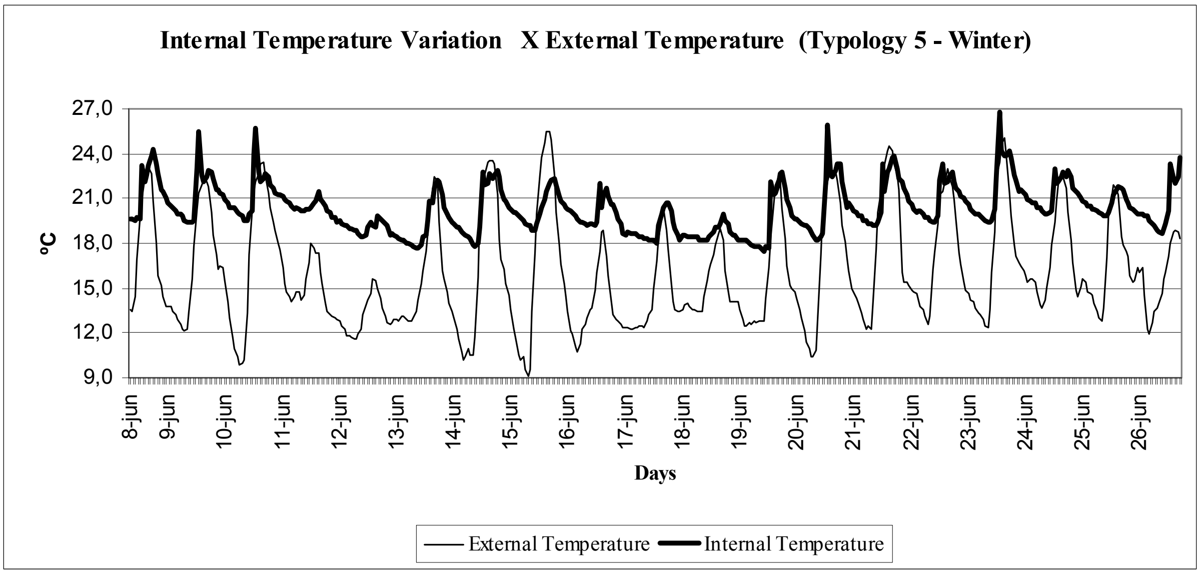

Figure 3b: Internal temperature comparison front of external temperature variation relating to typology 5 (Winter).

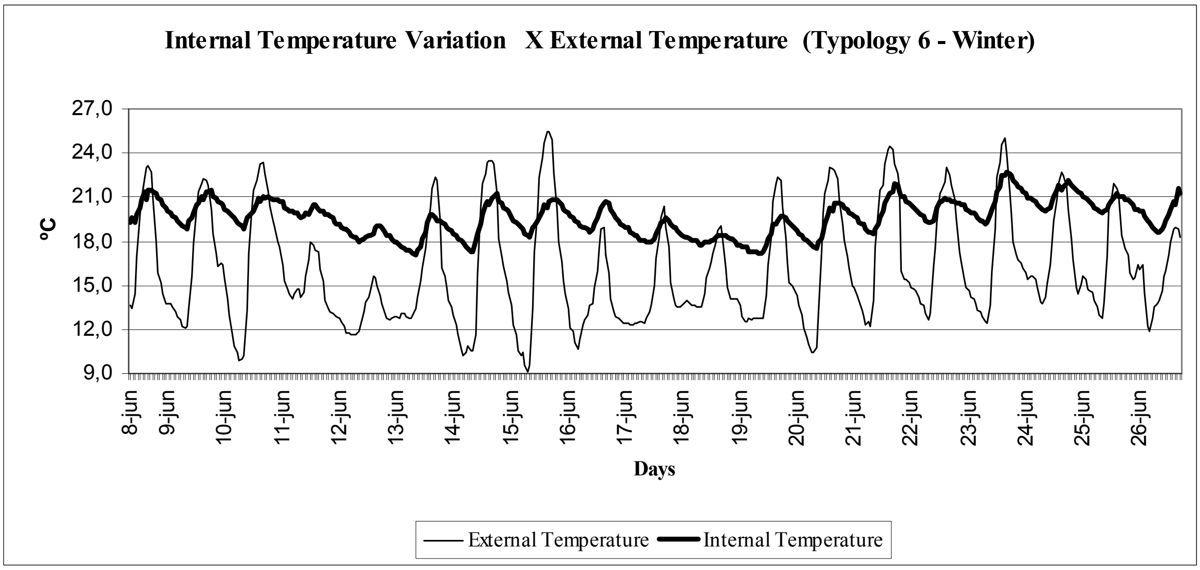

Figure 3c: Internal temperature comparison front of external temperature variation relating to typology 6 (Winter).

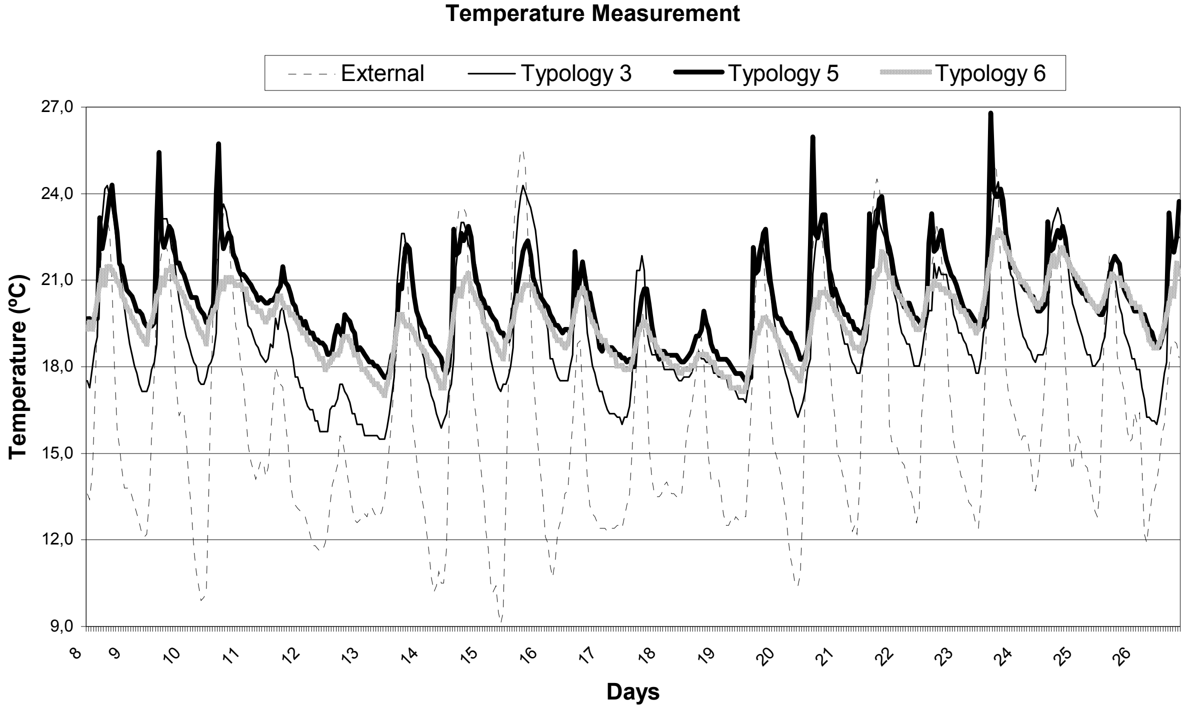

Figure 3d: Internal temperature comparison front of external temperature variation relating to typologies: 3, 5, 6 (Winter).

CONCLUSIONS

The present work has been consisted on carrying out 39 buildings representing a total of 841, placed in Curitiba city, thermal-ambient analysis measured in 6 buildings and to carry out the used standard of such ones.

It has been concluded that the discomfort climatic conditions, for Curitiba city, are predominantly caused for cold weather, as it was hoped. During the winter it was obtained better results for the typology 5 (low building, no external shading and with the percentage of window for front larger than 50%, having almost all the information placed inside the comfort zone. During the summer the comfort conditions of the various typologies were alike.

It must be still considered the difficulties found on the survey development. With the methodology used based on field analysis, the information collect was subjected to a several limitations. Such limitations have involved difficulties on the information collect and the lack of reasonable time for studying and analyzing of the software used in this survey.

This way, due to a temporal limitation to what the present work was subjected, we have suggested a large information collect to take into account some more typology considerations, such as kind of glass and window open, color and the front guidelines. It has been also recommended to make computerized study using a simulations analysis program.

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Thermal Comfort Requirements in Iranian Hospitals

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SUMMARY

This paper presents the results of a study to determine the thermal comfort requirements of the occupants in Iranian hospitals. It arrives at its conclusions through consideration of the wide range of metabolic rates and clothing levels experienced by the occupants. The study includes both patients and staff, and only considers patient recovery wards as these are where the greatest range of thermal comfort needs will normally be found. This study uses ISO 7730 as its basis, with reference also to ASHRAE 55 and CIBSE. The main conclusion from this study is that while the occupants of Iranian hospitals can require widely varying thermal conditions to achieve thermal comfort, it is possible to reconcile these different needs.

INTRODUCTION

The term comfort is associated with human health – defined not just as the absence of disease but in terms of a total sense of physical, mental and social well-being [1]. Comparing the comfort levels of sedentary people at home, at work and in a climate chamber, shows that being ‘at home’, in a familiar and under control environment, leads to comfort and makes people less sensitive to temperature [2].
There are two main approaches in thermal comfort standards: The laboratory based method, e.g. Fanger’s PMV-PPD model [3] and the field study method, e.g. the Adaptive Comfort Standard or ACS are the main approaches to thermal comfort researches.

In 1970 through his own equation, Fanger argued that knowledge of thermal comfort conditions was inadequate for environmental engineers involved in the specification of heating, cooling or ventilation technologies and indoor climates [3]. The Predicted Mean Vote (PMV) developed by Fanger provides a method by which the quality of indoor environments can be rated in practice and the degree of occupant discomfort assessed. The presented model by Fanger being used to define the combined conditions (i.e. air temperature, mean radiant temperature, relative air velocity and humidity level) in which the highest proportion of people are likely to be comfortable, for any specified level of activity and clothing [3]. Some international thermal standards such as ISO 7730 [4], still are using the Fanger’s model as the main framework their publication.

Against the laboratory model, other researches aim to account for variations in thermal comfort conditions and perceptions within buildings. Field studies challenge the validity of relatively fixed concepts of thermal comfort, such as those based on heat-balance equations. These results have led some researchers to develop more adaptive models that account for multivariate and dynamic human experiences in the real world [5]. One of the first field studies on the comfort of people in their natural habitats has been done on factory workers in the 1930s by Humphreys and Nicol [6].

Some of adaptively approaches field studies recorded significant differences between comfort values based on Fanger's predicted mean vote (PMV) and actual perceptions of comfort in office environments [7], [8]. Differences in the thermal sensation votes recorded of the same group of people -with the same clothing and activity levels [2]. Also Humphreys [9] argues that people are not inert recipients of the environment, but interact with it to optimize their own conditions. The objective for thermal comfort researchers is thus to observe the daily routines, practices and habits of occupants of building to see how they modify and adjust their environments to achieve comfort [9]. Both of laboratory and adaptive methods are represented in the latest edition of ASHRAE's thermal comfort standard [10].

Reviews on the adaptive models show that these models notice wider bands of temperature where the occupants may still sense the comfort zones, but the adaptive models of thermal comfort are more relative to free running systems buildings. Also being in the 'comfort zone' is represented as a threat to worker productivity and organizational efficiency [11], [12]. And, comparison between the static model (PMV) with the adaptive models in thermal comfort show that the PMV model is more accurate in building with HVAC systems and the adaptive models are more relative to buildings with natural ventilation [13]. On the other hand the Iranian regulations for healthcare buildings recommend the air conditioning systems for healthcare buildings, regarding to the hygiene problems and the type of activity and the occupants of these buildings. Based on this argument, this study selected the laboratory method (PMV), to predict the required thermal conditions by the occupants of these buildings.

For the standards used, ASHRAE, CIBSE and Iranian standards indicate the indoor air temperature and relative humidity ranges for thermal comfort achievement. ISO7730 indicates a range for Predicted Mean Vote (PMV) or Predicted Percentage Dissatisfied (PPD) to achieve thermal comfort based on the six variables which define PMV or PPD: air temperature, relative humidity, radiant temperature, clothing and activity levels, and the air velocity [14].

Due to different levels of clothing and activity in patients and hospital staff, these groups require different conditions to achieve thermal comfort [15]. The thermal conditions required to produce thermal comfort are further complicated by the fact that patients with different complaints may be located in the same wards, also a majority of the hospitals' staff always may stay for a while in the thermal conditions provided in patients' rooms, due to their activities; so to achieve the best balance of comfort it is necessary to first establish the conditions needed to achieve thermal comfort for patients and staff in hospital.

This paper presents the findings of an investigation undertaken to ascertain:

- those conditions required for thermal comfort by patients and staff of Iranian hospitals
- Potential solutions to improve thermal comfort achievement in these buildings

The study focuses on the Iranian hospital wards and mainly on the patients' rooms.

METHOD AND ANALYSES

Although generally, all the standards referred to for assessing thermal comfort require the measurement of relative humidity and air temperature but the calculation for ISO7730 also requires clothing and activity levels to be observed and radiant temperature and air velocity to be estimated [16]. Table 1 presents the relevant thermal comfort ranges from the standards.

Table 1: Recommended conditions to achieve thermal comfort

Standard	Recommended thermal condition to achieve thermal comfort
ASHRAE	23 ºC<Temp<26 ºC and 30%<Rh<60
ISO 7730	-0.5<PMV<+0.5, PPD<10% and 30%<Rh<60
CIBSE	22 ºC <Temp<24 ºC and 30%<Rh<60
Iranian Regulation	24 ºC <Temp<28 ºC and 30%<Rh<60

Buildings observation and site visiting have been done in this study to specify the types of occupants and their personal factors including the levels of activity and clothing. This study noticed low levels of activity for patients due to the physical conditions of their bodies. In terms of staff activities the study assumed an average of activities of those staff that are in direct relation with the indoor environment conditions of patients' rooms.

This study beside the environmental and job protocols, noticed high effects of cultural motives on both groups of occupants -particularly the staff- in terms of clothing in Iranian hospitals. This study assumes that the mixture of clothing types for patients without a blanket cover gives a 0.49 clo, for patients with cover 1.39 clo, and for staff 0.88 clo. This study assumes that the average activity rate for patients is 0.82 met, and for staff is 1.50 met.

The analysis in this study has been undertaken to determine the required thermal conditions in Iranian hospitals. From table 1, ASHRAE, CIBSE [17], and the Iranian regulation [18] indicate just the ranges of air temperature and relative humidity likely to achieve thermal comfort. However we know that the different user groups in hospitals are likely to have quite different temperature and RH requirements to achieve thermal comfort. These differences arise due to the variation in activity and clothing levels between more sedentary Patients who may use blankets to increase their insulation levels, Patients who are not able to use heavy clothing such as blankets because of their medical conditions, and Staff who generally have higher levels of activity.

The first issue to address for a study such as this is whether it is possible to find a range of thermal conditions that are suitable for these hospital users. This study uses the PPD calculation from ISO 7730 as the basis for assessing the different ranges of temperature and RH within which Patients and Staff in Iranian hospitals might achieve thermal comfort for the activity and clothing levels stated above. The study assumes 10 %PPD is an acceptable level [4].

The calculations show that, in theory, by varying the RH and air velocities within reasonably achievable values, that the Air Temperatures needed to achieve thermal comfort conditions for Iranian hospital Staff could be in a range between 19ºC to 26ºC; that the range for Patients with blanket could be in a range between 22.5ºC to 28ºC; and that for Patients without blanket the range could be between 27ºC to 31.5ºC.

Figure 1 represents the practical Air Temperature ranges that need to be achieved for each occupant type. We can see therefore that there is NO easily achievable overlap in acceptable Air Temperature ranges between Staff requirements and the requirements of Patients without blankets. The Patients with blankets are capable of being comfortable in a range of air Temperatures that straddle those for both Staff and Patients without blankets. Figure 1 presents, in graphical form, part of the results of the study where we assessed the variation of PPD with Air Temperature, when the air velocities were varied in a range between 0.1 m/s to 5m/s at a constant 40% Relative Humidity.

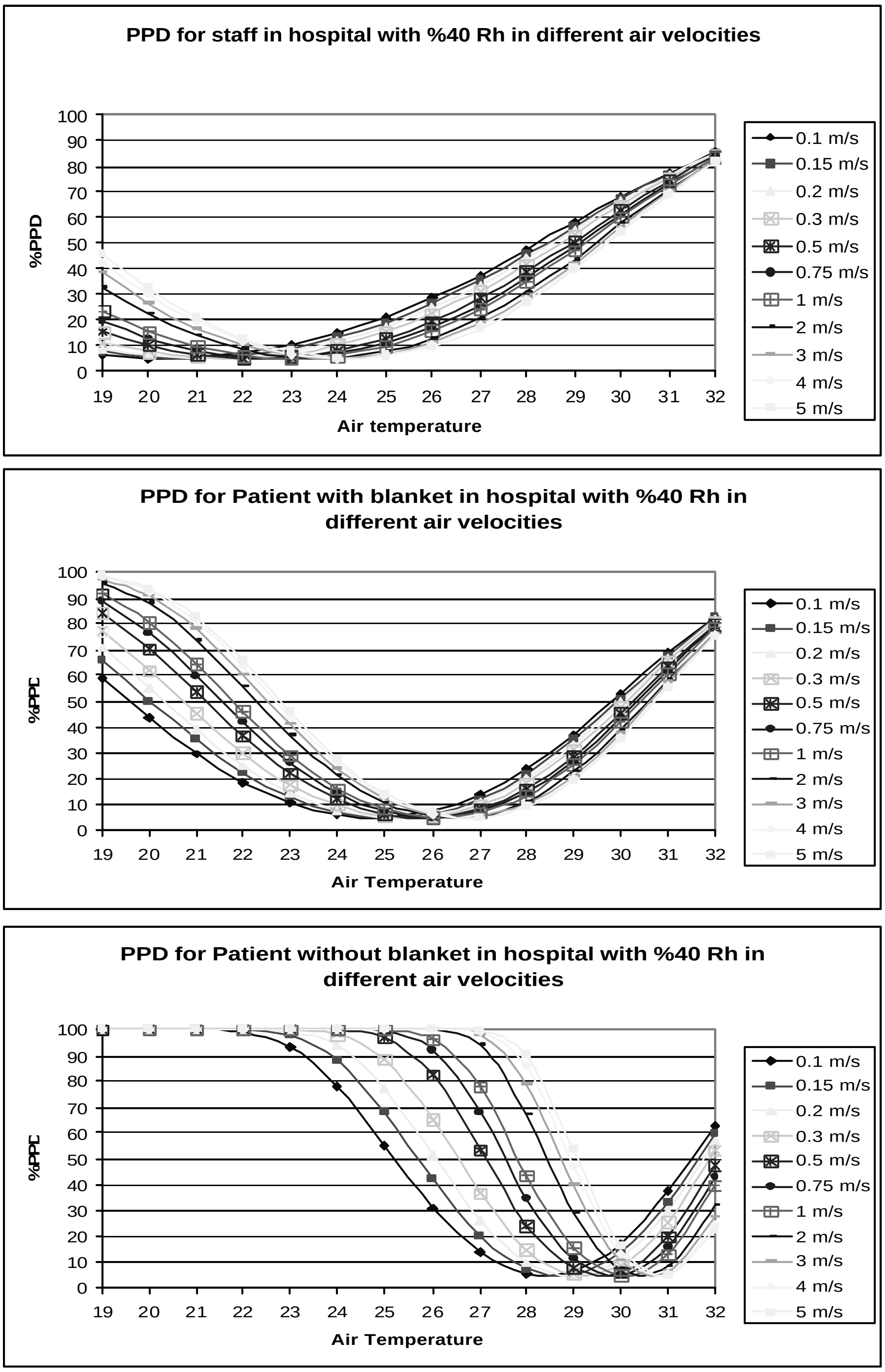

Figure 1: PPD achievements for all users in Iranian hospitals at 40% Relative Humidity with different air velocities.

The study shows that, as expected, higher air velocities extends the upper limit for the temperature range within which occupants can feel comfortable, while lower velocities extend the lower limit for comfortable temperatures. However there appears to be no combination of Air Temperature and Air Velocity which satisfies the comfort requirement of a PPD of less than 10% for both Staff and Patients without blankets. In fact, even if Staff experience a 5m/s air velocity at the same time as Patients with blankets experience an air velocity of <0.1 m/s we see that it is still not possible to achieve an acceptable common Air Temperature that will provide a PPD of less than 10%.

Assuming that Activity and Clothing Insulation levels are not able to be altered due to work and cultural reasons, the only other option left to try to achieve comfort is that of varying the mean radiant temperature experienced in the space by the different users. This could physically take the form of cooler or warmer floors, walls and ceilings in some areas depending on the predominant occupant of that area. Table 2 presents the results of a theoretical study where we assessed the variation of PPD with radiant temperature, when the air temperatures were varied in range between 20 ºC to 28 ºC at a constant 30% to 60% relative humidity and 0.1 m/s to 0.5m/s air velocity.

Table 2: Maximum radiant temperature ranges for hospital users in Iran
(All users assumed to be in conditions of: air temperature from 20ºC to 28ºC, air velocity from 0.1 m/s to 0.5 m/s, and relative humidity from 30% to 60%)

Users	**Maximum radiant temperature range**
Patients with blanket	12.3 ºC to 43.3 ºC
Patients without blanket	25.6 ºC to 50.5 ºC
Staff	-4.7 ºC to 35 ºC

The study shows, that to some extent, higher air temperature ranges can be offset by lower radiant temperature ranges within which all occupants can feel comfortable. It appears that theoretically a range of air temperatures between 20 ºC to 28 ºC can achieve the thermal comfort requirements of staff and patients (with and without blankets) if we were able to separately provide the appropriate radiant temperature for each category of occupant.

However, it is not practical to design spaces to achieve these large radiant temperature requirements. So the next part of the study explores what is more practically achievable. Referring to ISO 7730, the PMV and PPD express warm and cold discomfort for the body as a whole. But thermal dissatisfaction can also be caused by unwanted cooling or heating of one particular part of the body. This is known as local discomfort. The most common cause of local discomfort is draught. Local discomfort can also be caused by an abnormally high vertical temperature difference between the head and ankles, by too warm or too cool a floor, or by too high a radiant temperature asymmetry.

Figure 2 shows the percentage dissatisfied as a function of the floor temperature [4]. It is mainly people undertaking light sedentary activity who are sensitive to local discomfort. These will have a thermal sensation for the whole body close to neutral. At higher levels of activity, people are less thermally sensitive and consequently the risk of local discomfort is lower. This means therefore that the Patients will be more sensitive to radiant heating or cooling than the Staff.

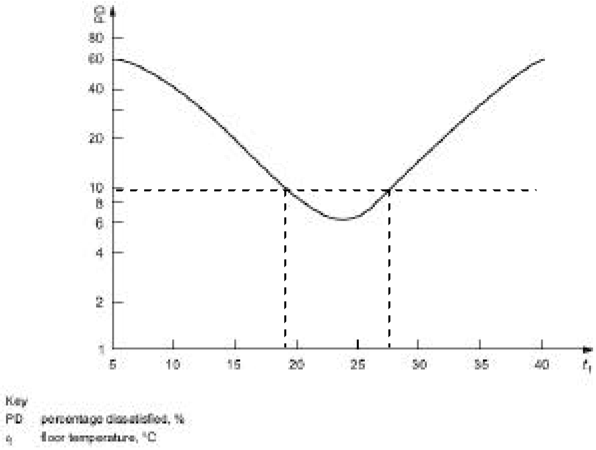

Figure 2: Local thermal discomfort caused by warm or cold floors

For horizontal radiant asymmetry, Figure 3 applies from side-to-side (left/right or right/left) asymmetry, the curves providing a conservative estimate of the discomfort: no other positions of body in relation to the surfaces cause higher asymmetry discomfort [4].

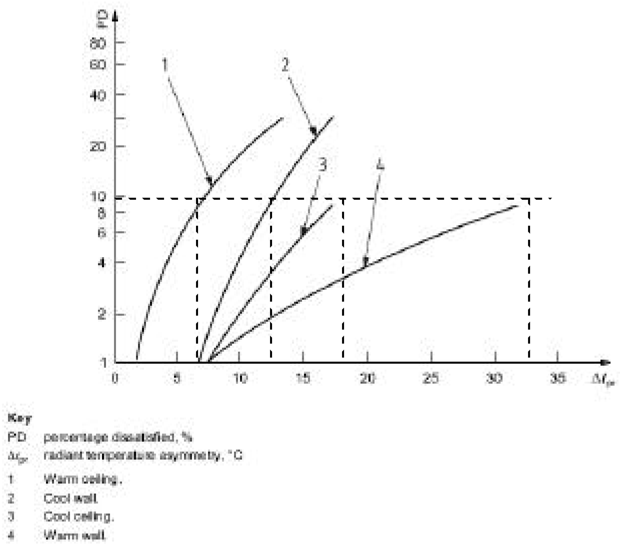

Figure 3: Local thermal discomfort caused by radiant temperature asymmetry

This figure shows that achieving the required radiant temperature via warm walls is likely to be the best way to attain thermal comfort when the air temperature is too cool. When the air temperature is too warm, thermal comfort is best achieved by using a cool ceiling.
When reviewing the foregoing work with regard to the activity, clothing levels and physical locations of staff and patients in hospitals, it can therefore be concluded that it might be possible to achieve thermal comfort for both staff and patients (both with and without blanket) by:

- Providing the additional cooling needed for the Staff via the floor (as this is the surface least visible to the patients)
- By providing the additional radiant heating needed by patients via, ideally warm vertical surfaces, but failing these warm ceilings.

Based on the study shown in Table 2 and the recommendations from ISO 7730 regarding local thermal discomfort (Figure 2 and Figure 3), in condition which the three referred groups of users are placed in separated rooms, the new thermal ranges within which comfort can be achieved for each of the three user groups in Iranian hospitals are as follows:

- Staff: air temperature from 20 ºC to 22 ºC and radiant temperature from 10.9 ºC to 29.3 ºC
- Patient with blanket: air temperature from 22 ºC to 24 ºC and radiant temperature from 20.6 ºC to 34.2 ºC
- Patient without blanket: air temperature from 26 ºC to 28 ºC and radiant temperature from 25.6 ºC to 35.3 ºC

Also by using different directional radiant temperatures (cooling via floor for staff and heating via ceiling for patients), there is now an overlap in the required air temperature to achieve comfort for staff and patients (with and without blanket) from 24 ºC to 26 ºC. By fixing the air temperature from 24 ºC to 26 ºC, and by using the air velocity from 0.1 m/s to 0.19 m/s for all users [4], the following radiant temperatures are required for thermal comfort achievement in Iranian hospitals.

- Staff: directional radiant temperature from the floor from 4.2 ºC to 23.7 ºC
- Patient with blanket: directional radiant temperature from the ceiling from 17.9 ºC to 31.6 ºC
- Patient without blanket: directional radiant temperature from the ceiling from 27.8 ºC to 37.8 ºC

But using directional radiant cooling will drop the temperature to or below dew point in the sources which is used for radiant cooling, then surface condensation will happen [19]. Also forming water in building will lead to mould growth. To minimize the surface condensation in directional radiant cooling, it is necessary to obtain low vapour pressure by ventilation and/or reduced moisture input to the buildings.

The dew point is associated with the relative humidity. A high relative humidity indicates that the dew point is closer to the current air temperature. As the relative humidity reaches 100%, the dew point will be equal to the current temperature. Given constant dew point, when the temperature increases the relative humidity will decrease. It is for this reason that equatorial climates can have low relative humidity yet still feel uncomfortable.

Following equation calculates the dew point in degrees Celsius to within ±0.4 °C. It is valid for:

$0\ °C < T < 100\ °C$

$0.01 < RH < 1.0$

$0\ °C < Td < 50\ °C$

Where:

T= Temperature in degrees Celsius [°C]

RH= Measured relative humidity as a fraction (not percent)

T_d = Calculated dew point temperature [°C]

The dew point temperature is:

$$Td = \frac{b \times \alpha(T, RH)}{a - \alpha(T, RH)} \qquad \{1\}$$

Where

$$\alpha(T, RH) = \frac{a \times T}{b + T} + \ln(RH)$$

With a= 17.27

And b= 237.7 [°C]

The uncertainty in the calculated dew point temperature is ±0.4 °C.

Based on the equation {1} and the highest risk of dew point in this research (26 °C for dry temperature and 60% for relative humidity), the dew point is 17.5°C.
So this research recommends the presented value in Table 3 for Iranian hospital users for thermal comfort achieve.

Table 3: Recommendations from this study- Acceptable parameter ranges for achieving thermal comfort in Iranian hospitals for different occupant types

Users	**Air Tem. °C**	**RH %**	**Air Velocity m/s**	**Radiant Tem. °C**
Staff	24 - 26	30 - 60	0.1 – 0.19	17.5 – 23.7
Patient with blanket	24 - 26	30 - 60	0.1 – 0.19	17.9 – 31.6
Patient without blanket	24 - 26	30 - 60	0.1 – 0.19	27.8 – 37.8

DISCUSSION

The different user groups of the hospital had thermal comfort requirements that were difficult to accommodate in one space. Therefore the best solution would appear to be to provide different thermal zones for different user groups in hospitals. In particular, it appears that staff should be provided with work areas in the wards that are controlled to very different conditions to the general wards.
In same indoor thermal condition, operating different radiance temperature for different users in Iranian hospitals, can reconcile their different thermal comfort achievement requirements.

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Measured Thermal Comfort Conditions in Iranian Hospitals for Patients and Staff

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SUMMARY

The occupants of hospitals like other buildings have widely differing thermal comfort requirements due to their different levels of clothing and metabolism. The study indicated three main groups of occupants in Iranian hospitals as: patients that are able to be covered, patients that are not able to be covered regarding their medical conditions, and staff. This study investigated the thermal comfort calculated to have been achieved for different occupants in Iranian hospitals. The 'thermal comfort achieved' findings have been derived from basic monitoring of those parameters which affect thermal comfort in 14 rooms in 4 separate Iranian hospitals. The thermal comfort results calculated for each room are compared with recommended Iranian and international standards for acceptable thermal comfort. The findings from this study showed a wide range of thermal comfort conditions were achieved in Iranian hospitals, and overall the thermal comfort conditions recorded during the measurement period were felt to be unacceptable.

INTRODUCTION

Thermal comfort can be expressed as "that condition of mind which expresses satisfaction with the thermal environment" [1]. A large volume of thermal comfort research over the last century has been distilled into international standards in terms of designing and maintaining comfortable thermal environments [2].

During this time two main approaches to thermal comfort research have been: the laboratory-based method, e.g. Fanger's PMV-PPD model [3] leading to EN ISO 7730 [1]; and the field-based method (e.g., the Adaptive Comfort Standard or ACS). Both of these methods are represented in the latest edition of ASHRAE's thermal comfort standard [4]. Air temperature, radiant temperature, humidity and air movement as the environmental variables combine with the metabolic heat generated by human activity and clothing worn by a person, to provide the six fundamental factors that define human thermal environments [5].

Hospitals occupants including Patients and staff require different thermal comfort conditions due to their different levels of clothing and activity. Comparing the comfort levels of sedentary people at home, at work and in a climate chamber, shows that being 'at home', in a familiar and under control environment, leads to comfort and makes people less sensitive to temperature [6].

This paper presents the findings of an investigation undertaken to ascertain:

- Current thermal comfort conditions being achieved in Iranian hospitals.
- Potential solutions to improve thermal comfort achievement in these buildings.

The thermal comfort conditions have been ascertained through a combination of monitoring those parameters which effect occupant thermal comfort along with occupant and site observations. The findings from the monitoring are compared with the Iranian 'Health Buildings Design' standard [7] and international thermal comfort standards from ASHRAE, ISO7730 and CIBSE [8].

For the standards used, ASHRAE, CIBSE and Iranian standards indicate the indoor air temperature and relative humidity ranges for thermal comfort achievement. ISO7730 indicates a range for Predicted Mean Vote (PMV) or Predicted Percentage Dissatisfied (PPD) to achieve thermal comfort based on the six factors which define PMV or PPD: air temperature, relative humidity, radiant temperature, clothing and activity levels, and the air velocity [9].

Table 1 presents the relevant thermal comfort ranges from the standards.
The thermal comfort of patients and hospital staff is affected by the thermal conditions provided including air temperature and relative humidity in hospital wards. [10] Due to different levels of clothing and activity in patients and hospital staff, these groups require different conditions to achieve thermal comfort.

Table 1: Recommended conditions to achieve thermal comfort

Standard	**Recommended thermal condition to achieve thermal comfort**
ASHRAE	23 ºC<Temp<26 ºC and 30%<Rh<60
ISO 7730	-0.5<PMV<+0.5, PPD<10% and 30%<Rh<60
CIBSE	22 ºC <Temp<24 ºC and 30%<Rh<60
Iranian Regulation	24 ºC <Temp<28 ºC and 30%<Rh<60

METHODS

All the standards referred to for assessing thermal comfort require the measurement of relative humidity and air temperature. The calculation for ISO7730 also requires clothing and activity levels to be observed and radiant temperature and air velocity to be estimated [5]. Building monitoring with occupant and site observations were used as the method for data collection in this study.

14 patient rooms in 4 hospitals were selected as case studies. To cover the majority of conditions likely to be found in Iranian hospitals the selected healthcare buildings included four different types of activity. The selected rooms in each hospital were also different in other ways, including the orientation, type of patients, dimensions etc.

The case studies consisted of: Two in-patient rooms in a small local clinic; three separate rooms in a private maternity hospital; four separate rooms in a new (prototype) hospital, and five separate rooms in a regional educational hospital. Table 2 presents the details of the monitored hospitals.

Table 2: Monitored hospitals

Case study	activity	No of wards	No of beds	No of active beds	Year of construction	No of monitored rooms
Building No1 TER1 TER2	General	1	7	7	1990	2
Building No2 KR1 KR2 KR3	Maternity	1	32	18	2003	3
Building No3 TR1 TR2 TR3 TR4	General	5	100	32	2002	4
Building No4 IKR1 IKR2 IKR3 IKR4 IKR5	General	11	200	135	1945	5

The temperature and relative humidity data was recorded by calibrated loggers at 10 minute intervals for around two weeks for each of the case studies. The globe temperature was also measured to allow calculation of mean radiant temperature. Site visits provided observations of clothing, activity and air velocity. No noticeable air movement was observed in any of the case studies, so a value of 0.1m/s was used to cover the air velocity in all monitored rooms. The clothing and activity levels observed during the study are recorded in Table 3 and Table4.

Table 3: Clothing types for patients and staff in hospital wards*

Clothing	**Clothing level** (clo)	**Patients without cover**	**Patients with cover**	**Staff**
Light-weight shirts with long sleeves	0.20	0.20	0.20	-
Normal shirts with long sleeves	0.25	-	-	0.25
Light-weight trousers	0.20	0.20	0.20	-
Normal trousers	0.25	-	-	0.25
Light summer jackets	0.25	-	-	0.25
Socks	0.02	0.02	0.02	0.02
Shoes (thin soled)	0.04	-	-	0.04
Panties-pants and bra	0.03	0.03	0.03	0.03
Singlet	0.04	0.04	0.04	0.04
Blanket **	0.90	-	0.90	-
Total clothing ***		**0.49 clo**	**1.39 clo**	**0.88 clo**

**- ISO 7730: Moderate thermal environments- Determination of the PMV and PPD indices and specification of the conditions for thermal comfort*
*** - the clo for blankets assumed to be same as boiler suit in ISO 7730*
**** - Total clothing = ? clothing*

Table 4: Activity rates for patients and staff in hospital's wards ****

activity	**Metabolic rate** (met)	**Patients** , % of time	**Staff** , % of time
Reclining	0.8	90%	-
Seated, relaxed	1.0	10 %	5 %
Sedentary activity	1.2	-	30 %
Standing, light activity	1.6	-	50 %
Standing, medium activity	2.0	-	10 %
Walking on the level	1.9	-	5 %
Total activity		**0.82** met	**1.50** met

*****- [1] - ISO 7730: Moderate thermal environments- Determination of the PMV and PPD indices and specification of the conditions for thermal comfort*

From Table 3, this study assumes that the mixture of clothing types for patients without a blanket cover gives a 0.49 clo, for patients with cover 1.39 clo, and for staff 0.88 clo. From Table 4, this study assumes that the average activity rate for patients is 0.82 met, and for staff is 1.50 met.

RESULTS

Comparisons of recorded data with ASHRAE, CIBSE, and Iranian standard recommended ranges of air temperature and relative humidity have been completed. PMV and PPD values have also been calculated for each case study and compared with recommended ranges in ISO7730. Table 5 summarizes the findings in terms of the percentage of the time that the buildings achieved the recommended values for the standards.

Table 5: Maximum radiant temperature ranges for hospital users in Iran
(All users assumed to be in conditions of: air temperature from 20ºC to 28ºC, air velocity from 0.1 m/s to 0.5 m/s, and relative humidity from 30% to 60%)

Users	Maximum radiant temperature range
Patients with blanket	12.3 ºC to 43.3 ºC
Patients without blanket	25.6 ºC to 50.5 ºC
Staff	-4.7 ºC to 35 ºC

Table 5 shows that overall the rooms monitored met ASHRAE recommended temperature and RH requirements 27% of the time; CIBSE recommended temperature and RH requirements 4% of the time; and Iranian regulations for recommended temperature and RH requirements 67 % of the time.

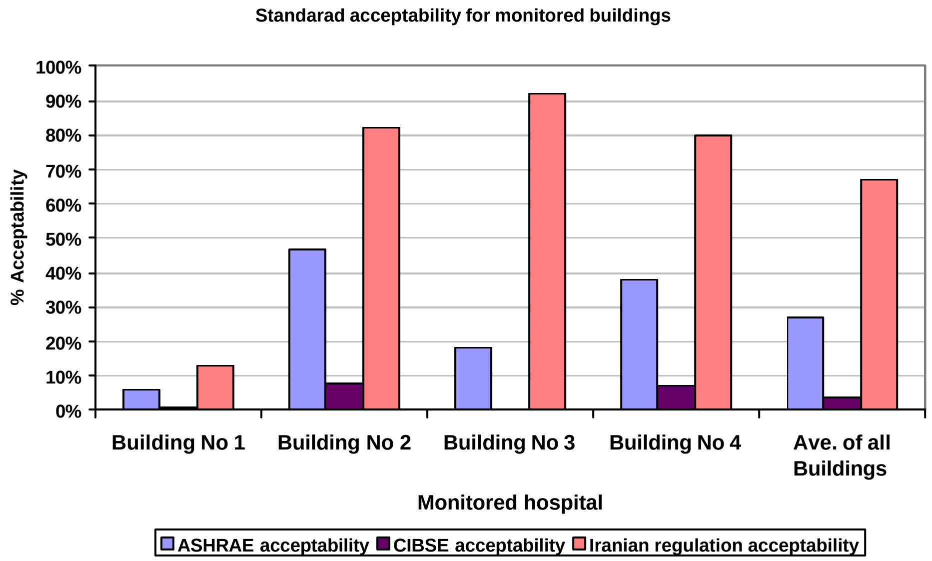

Figure 1: ASHRAE, CIBSE, Iranian regulation acceptability in monitored buildings.

Referring to the more detailed ISO7730 requirements we found that only 1 % of the monitored time were the staff estimated to be in the comfort zone, though these same conditions meant that thermal comfort conditions were met 39 % of time for the patients with blankets and 17 % of time for the patients without blankets. Figure 1 and Figure 2 present these results in graphical format.

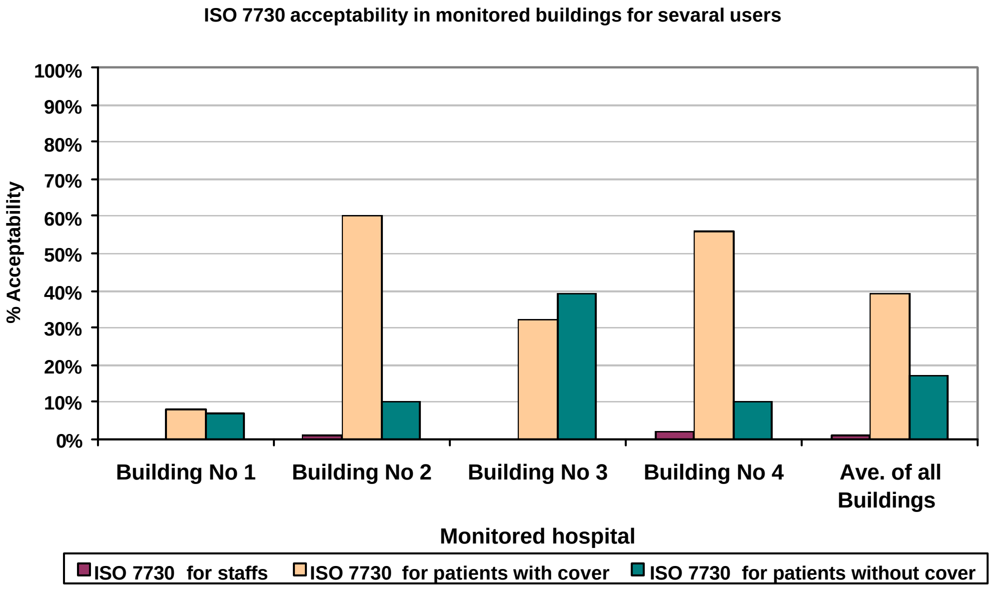

Figure 2: ISO 7730 acceptability in monitored buildings for Staff, Patient with blanket and Patient without blanket in Iranian hospitals.

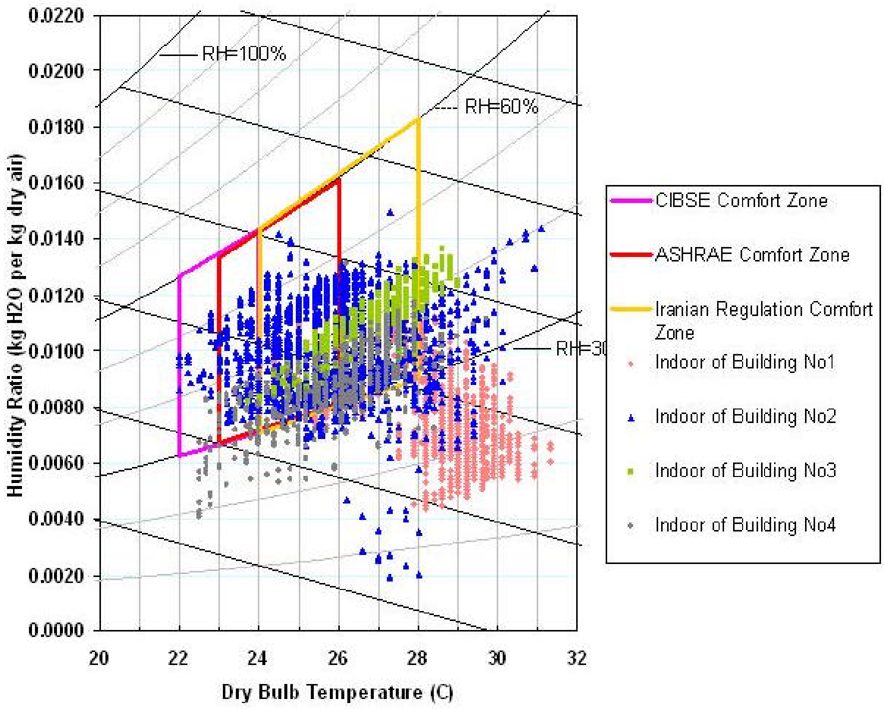

Figure 3: Measured indoor conditions in monitored hospital rooms compared to the zones of acceptable conditions for each standard.

Figure 3 presents graphically the spread of the monitored indoor thermal conditions across all the monitored buildings. This figure shows that nearly half of all the measured data fell inside the values recommended by the different CIBSE, ASHRAE and Iranian regulations. When the data fell outside these acceptable ranges it is seen that the rooms were then generally too hot and dry. This graph also shows building No1 had the most data outside the recommended ranges.

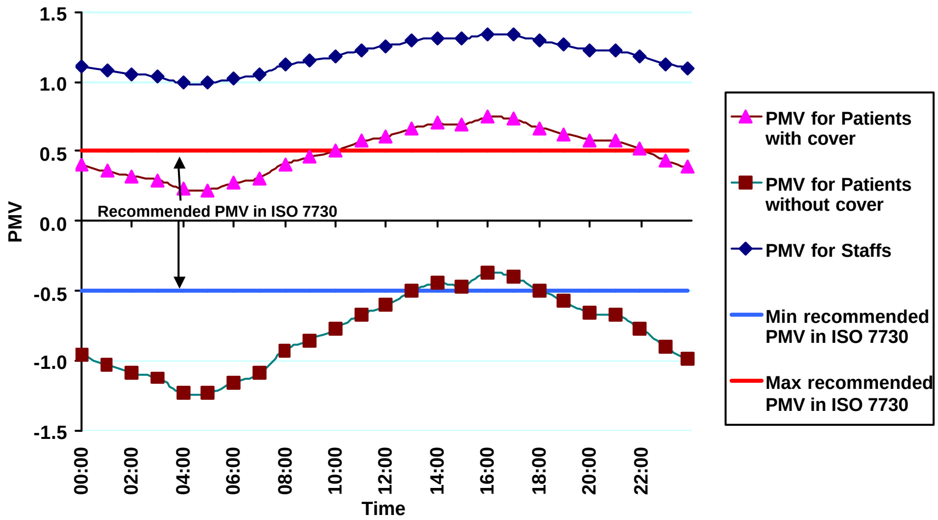

Figure 4: Average PMV achieved in Iranian hospitals for patients and staff

Figure 4 presents the comparison between the recorded data and the ISO7730 recommended ranges. On average, Figure 4 shows that while the hospital staff generally experience uncomfortably hot thermal conditions, patients with blankets are generally the most thermally comfortable group in the hospital, though with a tendency towards overly warm conditions in the afternoons. For patients without a blanket for any reason the thermal conditions will generally be too cool. Table 6 presents this data as the percentage of acceptable PMV and PPD for patients and staff by each of the monitored hospitals.

Table 6: PMV and PPD acceptability for three main user groups in case studies

Case study	***% PMV acceptability (-0.5<PMV<0.5) for***			**% PPD acceptability (PPD<10%) for**		
	Staff	**Patients** With cover	**Patients** without cover	**Staff**	**Patients** With cover	**Patients** without cover
Bu. No1	0 %	9 %	80 %	0 %	9 %	79 %
Bu. No2	1 %	68 %	12 %	1 %	67 %	12 %
Bu. No3	0 %	32 %	39 %	0 %	32 %	38 %
Bu. No4	3 %	65 %	10 %	3 %	64 %	10 %
Ave. All	**1 %**	**44 %**	**35 %**	**1 %**	**43 %**	**35 %**

On average the patients were more comfortable than the staff during the monitoring period. This table shows while 1 % of monitored time the hospital staffs were placed in comfort thermal conditions, 44 % of monitored time the patients with blanket and 35 % of monitored time the patients without blanket were placed in comfort thermal conditions. The table also shows that there were significant variations in the comfort conditions achieved in the different hospitals, though the staff were universally uncomfortable for all hospitals.

DISCUSSION

The different user groups of the hospital had thermal comfort requirements that were difficult to accommodate in one space. Therefore the best solution would appear to be to provide different thermal zones for different user groups in hospitals. In particular, it appears that staff should be provided with work areas in the wards that are controlled to very different conditions to the general wards.

Although, the monitored indoor thermal conditions in Iranian hospitals were generally within the recommended range of the Iranian thermal regulation, they were generally out of the thermal comfort zones recommended by ISO7730, ASHRAE and CIBSE.

The next stage of the study will try to answer some key questions including: is there any possibility for reconciling the thermal conditions required by different groups of occupants in Iranian hospitals? How much can modifying existing designs improve thermal comfort? How much more comfort could be achieved through a “new” vernacular for the design and operation of Iranian hospitals?

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Impacts of Indoor Temperature and Velocity on Human Physiology in Hot Summer and Cold Winter Climate in China

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ABSTRACT

The impacts of indoor thermal environment on body physiology have been carried on for four years (2003-2007) in laboratory in Chongqing, a typical city located in hot-summer and cold-winter region in China. Experimental objectives used are healthy university students. The range of indoor air temperature in summer is 25 °C - 37.5 °C.The objectives' physiological changes (Motor nerve Conduction Velocity, Sensory nerve Conduction Velocity, Skin Temperature etc.) under different temperatures and ventilations have been tested.

The results show that the effect of air temperature on skin temperature is significant; Skin temperature has a linear and positive correlation with MCV and SCV; Air temperature has a significant effect and a positive correlation with MCV and SCV when indoor air temperature are not too high (25 °C - 32 °C); In this temperature range, mechanical ventilation can significantly improve the thermal comfort sensation. However, increasing indoor air movement has a little effect on MCV and SCV when indoor air temperature is above skin temperature (34 °C - 37.5 °C).

Such experiments can provide basic data to establish the standard of acceptable indoor thermal environment for hot-summer and cold-winter region in China.

Keywords: thermal comfort, indoor thermal environment, human body physiology, Motor nerve Conduction Velocity, Sensory nerve Conduction Velocity

INTRODUCTION

Buildings are designed to suit the climate in which they are located and the functions for which they are intended. There is a unique relationship between an individual, the environment and the building he or she inhabits. Everyday experiences tell us that there are a host of factors which are relevant to this concept. Air and surface temperatures, humidity, air movement have an important role. A deficiency in one of the physical factors can spoil the balance of the environment. The physical environment can enhance one's work but an unsatisfactory environment can hinder work output. As building service engineering and building environmental control systems become more complex and expensive, cost justification becomes a greater issue for decision makers. Factors such as poor lighting, both natural and artificial, poorly maintained or designed air conditioning, and poor spatial layouts are all likely to affect performance at work. The variations in thermal comfort and well being were influenced by temperature changes.

The indoor thermal environment will directly influence human physical and psychological health, sense of comfort, as well as human's well being[1, 2]. Professor Fanger published the famous thermal equations based on experimental data[3]. He created the currently widely-used indexes: Predicted Mean Vote (PMV) and the Predicted Percentage of Dissatisfaction (PPD) to assess thermal comfort, based on the human body thermal equivalence equation[4]. The thermal comfort standard recommended by ASHRAE55-2004[5] and ISO7730[6] are widely used in the world to measure thermal comfort based on Fanger equations. However, more and more researches show that the PMV-PPD index obtained through experiment in Laboratory in the 1970s can not be suitably applied to all the countries and regions, especially for the free running buildings[7, 8,9,10,11].

The indoor environment of the hot-summer and cold-winter region is very much different from those experimental conditions in Professor Fanger's air-conditioned Laboratory. The suitability of applying Fanger's result as an international design standard into different local building conditions is now argued by experts. The impact of indoor thermal environment on human physiology has been carried on for four years (2003-2007) in Laboratory in Chongqing, a typical city located in hot-summer region in China. Experimental objectives used are healthy university students. The range of indoor air temperature in summer is 25 °C - 37.5 °C.The objectives' physiological changes (Motor nerve Conduction Velocity, Sensory nerve Conduction Velocity, Skin Temperature etc.) under different temperatures and ventilations have been tested[12,13,14,15]. The results show that the effect of air temperature on skin temperature is significant; Skin temperature has a linear and positive correlation with MCV and SCV; Air temperature has a significant effect and a positive correlation with MCV and SCV when indoor air temperature are not too high (25 °C - 32 °C); In this temperature range, mechanical ventilation can significantly improve the thermal comfort sensation. However, increasing indoor air movement has a little effect on MCV and SCV when indoor air temperature is above skin temperature (34 °C - 37.5 °C).

METHODOLOGY

The impacts of indoor thermal environment on body physiology have been carried on in Laboratory in the region of the hot-summer and cold-winter in China. The parameters of physiology including Motor nerve Conduction Velocity, Sensory nerve Conduction Velocity, Skin Temperature etc, have been selected. The impacts of indoor temperature and velocity on human physiology have been measured and analyzed.

The human nerve system consists of trillions of neurons, which have converted to functions to recognize, conduct and analyze information of other cells and environmental information. Nerve system is the quickest responding to heat among all parts of human body. For example, sensory cells receive the stimulus from the climate in a hot environment, and then conduct it to the human brain to cause a feeling of hot or cold. Consequently, nerve system is a main physiological system which has a directly influence on the reception and process of information of heat in the environment[16].

As far as the human nerve system is concerned, the most frequent and direct parameter is nerve conduction velocity, which is calculated along certain nerve phase, meaning the fastest (both sensory and motion) velocity of neuraxis[16]. When the electricity stimulates nerve, the impulse is conducted by motor, sensor or mixed nerve. In consideration to convenience and feasibility, the experiment chooses the right-hand median nerve as the stimulated nerve, as Fig.1 shows. The median nerve sensing fiber is mainly distributed in the hand palm side and hand outside skin. The experiment applies the surface stimulation by electrode to test. The stimulated parts and electrodes are located, as Fig.2 shows.

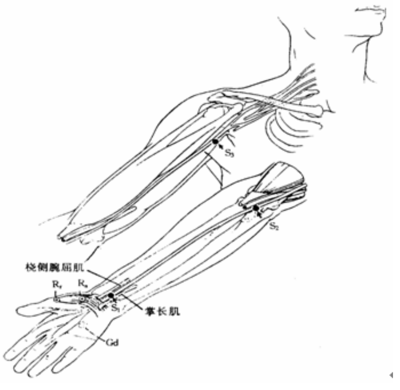	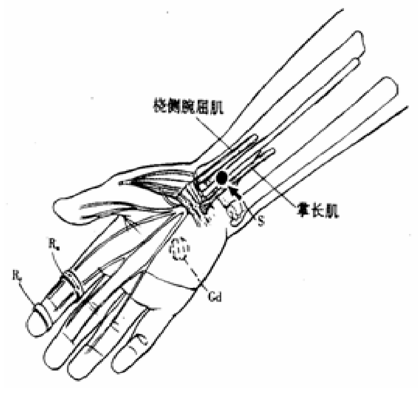
Fig.1 Test of median nerve motor conduction.	Fig.2 Stimulation of median nerve sensory conduction.

Skin temperature is an important physical indicator that reflects the effect on human body by surrounding climate condition and outside body condition (subjects' activities and clothes). In normal case, the skin temperature of human body varies between 15…42 °C. The skin temperature on any point of body surface depends on local balance between the heat streams that flowing from central to this point of skin and the heat radiation from there to outside surroundings. Since skin temperature is physical indicator that reflects the compact on human body by surrounding climate condition, it is easy to be affected by the surrounding conditions, which is a sensitive indicator.

Hanada's research found that different part on human body has different feeling on warmness[13]. The body parts as hest, the upper arm and curs are easy to feel cold, while the cold stimulation is most sensitive when the upper arm, forearm, curs and thigh are feeling together. Considering the practical condition, a normal experiment can measure skin temperature of 3 to 10 points to reflect the average skin temperature of the whole body. In the experiments, 6-point method has been used. Skin temperature also affects nerve conduction velocity, which is the most important physical factor that affects conduction velocity. When skin temperature varies between 25…35 °C, the decrease of nerve conduction velocity displays the linear relation, namely each 1 °C raise of skin temperature results in 2…3m/s increase of conduction velocity. Generally, when skin temperature is higher than 30 °C, temperature effect will be little. Therefore, the experiments focused on analysis of nerve conduction velocity, skin temperature and relationship between skin temperature and nerve conduction velocity in the different thermal environmental conditions.

Because the experiments relate to measurement of thermal environment physics parameters and the human physiological parameters, the measuring instruments are grouped into two terms: Environmental parameters and Physical parameters measuring instruments. There are Copper-Constantan Thermocouple to measure outdoor and indoor air temperatures, Multifunctional Intelligent Data Acquisition Instrument for checking and measurement, Indoor Air Quality Tester to measure indoor air relative humidity, Portable Hot Wire Anemometer to measure indoor air velocity; Twelve-Channel Electrocardiograph ECG-9103P and Multifunctional Intelligent Environmental Data Acquisition Instrument, MEB-9104 to measure nerve conduction velocity MCV and SCV, Digital Electroencephalogram System EEG-8200 Human Skin Impedance Tester is used to measure skin impedance, Infrared Thermometer to measure skin temperature.

All the testing apparatus have been debugged and adjusted. The testing apparatus and instruments have very high precision. Experts have invited to adjust instruments after installation. Inside the room, 30 thermocouples have been averagely distributed and settled at the height of 1.7m from the floor to monitor the distribution of the air temperature in the whole laboratory. The temperature of ceiling, floor, window, wall etc is measured directly by infrared temperature measuring instruments. The causal measuring points are located at 4 places, which are when the subject is away from the floor at 4 heights: 0.1m, 0.6m and 1.1m, which refer to the three body parts when the subject is sitting: feet, belly and head, and 0.5m, which refers to the lying gesture of the subject.

The experiments measure the subjects' physical parameters include EEG, MCV, SCV, ECG, skin temperature, skin resistance, heart beat rate, blood pressure, body temperature. These experiments include two extreme environments in winter and summer from July 2003 to Feb. 2007.

Table 1 Experiment parameters

Time	Experiment parameters	
Winter	Indoor air temperature: (13.2±1.5) °C Wind speed near human body: (0.17±0.2)m/s Comparative indoor air humidity: (80.9±4.2)%	Indoor air temperature: (13.2±1.5) °C Wind speed near human body: (0.35±0.2)m/s Comparative indoor air humidity: (80.9±4.2)%
Summer	Indoor air temperature: (32.4±0.7) °C Wind speed near human body: (0.10±0.02)m/s Comparative indoor air humidity: (73.6±7.4)%	Indoor air temperature: (24.2±0.7) °C Wind speed near human body: (0.35±0.2)m/s Comparative indoor air humidity: (44.2±5.3)%
Summer (High-temperature indoor environment)	Indoor air temperature: (37.3±0.6) °C Wind speed near human body: (0.10±0.02)m/s Comparative indoor air humidity: (65.4±7.2)%	Indoor air temperature: (37.3±0.8) °C Wind speed near human body: (0.47±0.02)m/s Comparative indoor air humidity: (63.8±6.7)%

RESULTS

Analysis of the winter measurements results

The experiments in the winter mainly measure the influence of air temperature and draught on nerve conduction velocity. From the Fig. 3, it can be observed that the longer the mechanical ventilation is, the more MCV is obviously decreasing. During the whole process, the MCV can be divided into 3 stages. The first stage is from no-draught to 5 mines time of mechanical ventilation, and the MCV has a sudden drop, for the human body is affected suddenly by draught under cold thermal environment, which leads to the bigger heat loss. In the second stage from 5 mines to 20 mines time of mechanical ventilation, the MCV is still decreasing, but with slower speed compared with the first stage. Although the human body heat loss is affected, the human body can automatically moderate its temperature under some environmental parameters. Thus the lowering of MCV caused by draught is controlled. In the third stage after 20 mines time of mechanical ventilation, The MCV change rates are increasing compared with the second stage, The longer the time of draught that subjects suffered, the more MCV decreases rapidly The thermal environments have a bigger influence on MCV.

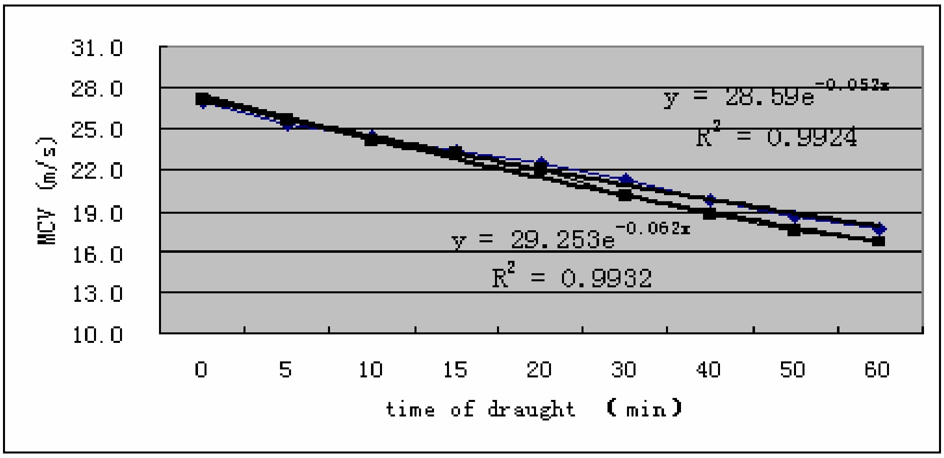

Fig.3 The impacts of draught on MCV in winter

The time of draught blow is negatively correlated to MCV and SCV. The regression equations are $y=29.253e^{-0.062x}$, y=47.968e-0.0621x respective for MCV and SCV, and correlation coefficients are R2=0.9932 and R2=0.9867 respective.

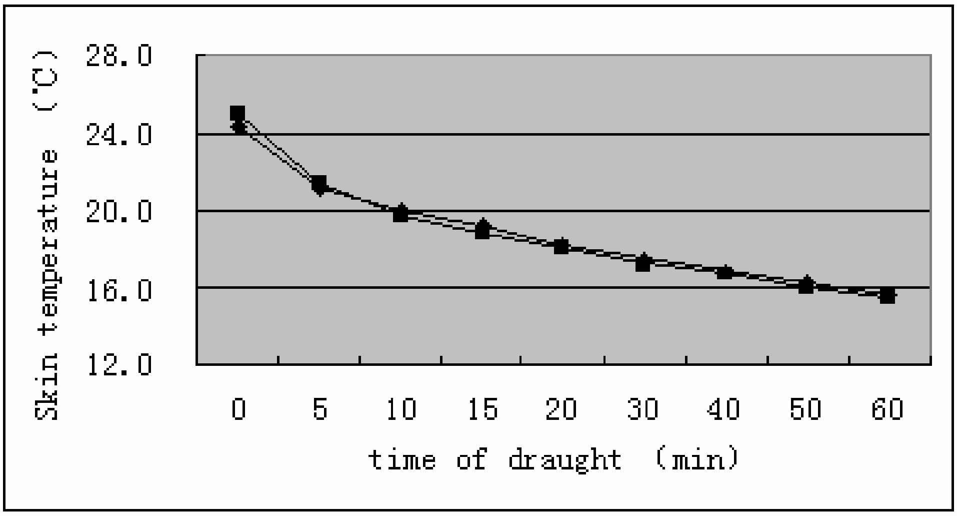

Fig.4 The impact of draught on Skin temperature in winter

Fig. 4 shows that there is an obvious decrease trend for skin temperature with the strong draught. And a sudden decline emerges from no-draught to draught. When draught duration is 5min, the decline of skin temperature decreases about 5 °C. With the whole testing period, the skin temperature reduces 10 °C with the final value of 15 °C. There is an obvious effect of draught in winter on skin temperature.

Analysis of the summer measurements results

The impacts of air temperature and velocity on the nerve conduction velocity and skin temperature in summer have be measured and assessed similar in summer and in winter. The MCV and SCV have downtrend with the time of ventilation. Fig. 5 shows there is a short suspension of SCV downtrend between 15min and 20min after ventilated. That means the subjects have had a corresponding physiological response to the stimulation made by ventilation during this period. For example vasoconstriction or increase metabolic heat production and reduce heat exchange with the outside to inhibit skin temperature falls, thus inhibiting nerve conduction velocity decline. Subsequently, the cooling capacity of the environment over the body heat production capacity, and the skin temperature further decreases and SCV has further reduced. Comparing to the experiment, the MCV and SCV downtrend in summer with ventilation are far little obvious than that in winter with draught. The subjects' skin temperatures with MCV and SCV have linear correlation.

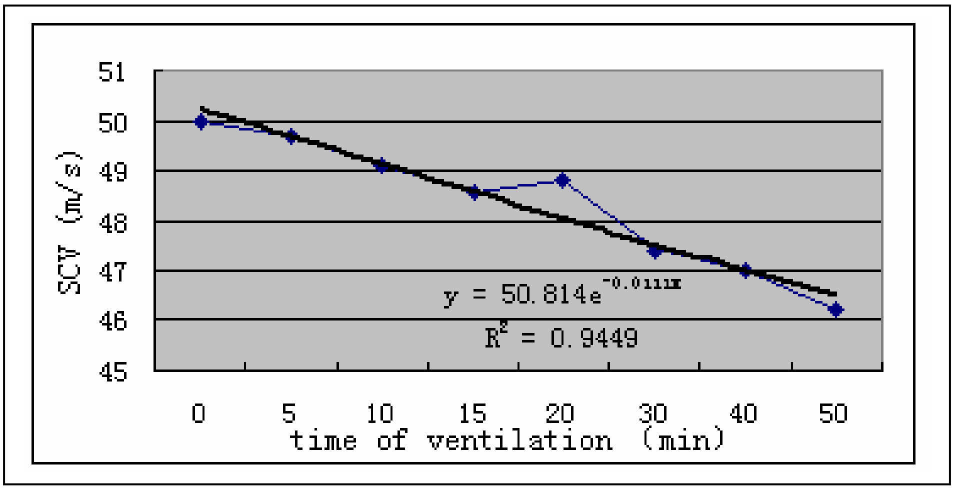

Fig.5 The impacts of ventilation on SCV in summer

Analysis of extremely high temperature (in summer) measurements results

The summer of 2006 in Chongqing has suffered the rare high-temperature and dry conditions for the last 100 years. Outdoor air temperature above 40 °C lasts for many ten days. The disastrous weather has provided a valuable opportunity for the experiment for us and gain a number of extremely high-temperature environments indoor physiological data.

The nerve conduction velocity has no significant difference for the environment condition. The ventilation could not affect the value of MCV and SCV in extremely high temperature environments. In the extremely high temperature environments, the outside air temperature is higher than the skin temperature; evaporation heat is the only way to dispel heat for the human being, so that the cooling effect of heat dispelled by convection would be ineffective when air flows. Comparing the average value of MCV and SCV in normal conditions with the average value of MCV and SCV in extremely high temperature in summer, the value of MCV and SCV under the extremely high temperature is higher. That means the higher the air temperature is, the faster the subjects' nerve conduction velocities are. The subjects would be prone to fatigue and be adverse to human health. The heat stress will increase the body's physiological load of the organizations and organs.

DISCUSSION

The impacts of indoor environments on human's physiology and thermal comfort in the regions of hot in summer and cold in winter is very representative and has important practical significance. In this paper, the human physiological parameters, which are affected by environmental factors change obviously (e.g. MCV, SCV, skin temperature,etc.) have been selected and measured in the region of hot-summer and cold-winter in China. The impacts of thermal environments, indoor temperature, draught and ventilation on Human Physiology have been assessed.

The results show that MCV and SCV decline obviously with the time of draught in the winter. The skin temperature corresponding to the MCV and SCV show a linear correlation. The higher the indoor temperature is, the higher the value of MCV and SCV are. The MCV and SCV in free-running room with high air temperature are higher than that in air condition environment. The impacts of draught on MCV and SCV in winter are much bigger than the impacts of ventilation in summer.

The effect of air temperature on skin temperature is significant; Skin temperature has a linear and positive correlation with MCV and SCV; Air temperature has a significant effect and a positive correlation with MCV and SCV when indoor air temperature are not too high (25 °C - 32 °C); In this temperature range, mechanical ventilation can significantly improve the thermal comfort sensation. However, increasing indoor air movement has a little effect on MCV and SCV when indoor air temperature is above skin temperature (34 °C - 37.5 °C). Draught has to be carefully avoided in the winter in order to achieve a comfort thermal environment. Such experiments can provide basic data to establish the standard of acceptable indoor thermal environment for hot-summer and cold-winter region in China.

ACKNOWLEDGEMENT

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Vertical Distribution of Air Temperatures in Heated Dwelling Rooms

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SUMMARY

The paper presents an experimental and theoretic research on one of factors influencing the indoor climate in dwelling rooms heated by heating systems – the vertical distribution of temperatures. The paper summarizes results from simulation of the room heated by a gas space heater and a plate radiator. Among main factors causing unfavorable distribution of temperatures in a room belong insufficient elimination of cold dropping airflows and high temperature of heating air.

INTRODUCTION

The subject of the paper belongs to the field of formation of indoor climate in residential buildings. One of factors influencing the quality of indoor climate in a dwelling room is the vertical distribution of temperatures. It is described by a vertical temperature gradient or by temperature difference between the position of a head and an ankle of a person in the given space. Indoor climate is created by various HVAC systems which have different influences on the observed vertical distribution of temperatures. We try to identify factors leading to unfavorable vertical distribution of temperatures in a room heated by a gas space heater and a plate radiator.

METHODS

The vertical distribution of temperatures of chosen spaces was investigated by experimental measurements and CFD simulations. The measurement was performed during a regular operation of chosen rooms. The room A is a dwelling room in an old brick house with one outer wall. The room is situated in the third floor under a cold sloping roof with leaking tiling. Surrounding spaces have the same or similar temperature regime. The walls are made of solid bricks; the ceiling is made of wooden girders. The leaking wooden windows and balcony doors in the room have a double glassing. The room is heated by a local gas space heater, see fig. 1a).
The room B is an office for two people in a new building with one outer wall made of medium thick façade, see fig. 1b). The office is surrounded by rooms with the same or similar temperature regime.

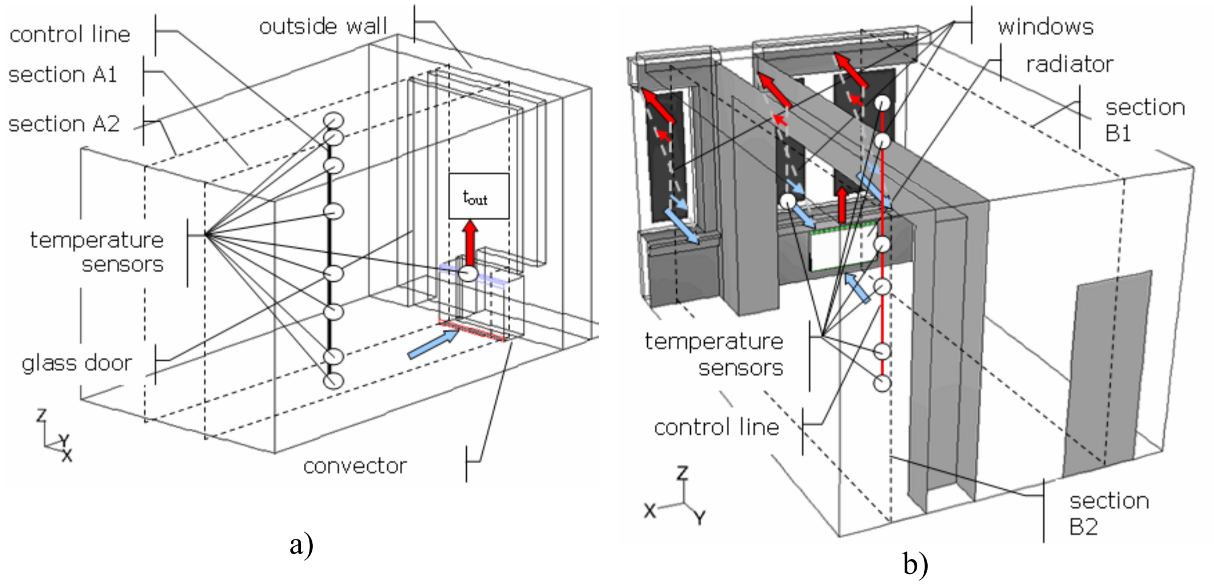


Figure 1. Geometry of rooms

In case A was measured and investigated the vertical distribution of temperatures while pre-heating.
In case B was measured and investigated the vertical distribution of temperatures while short-term ventilation by windows.

CFD simulations were performed on simplified geometric models of real rooms. Simulations included stationary and non-stationary solution of thermal transfer by convection, conduction and radiation. For airflow turbulence calculations the RNG k-ε model with standard wall functions was used. The heat transfer was solved by DO (Discrete ordinates) model [3]. Boundary conditions for CFD simulations were taken from the experimental measurement. The experimental measurement consists of measurement of air temperature and airflow velocity. We performed the measurement of the air temperature, the floor temperature, the ceiling temperature and the outer air temperature. We also compared thermal fields by using thermo-graphical images showing surface temperatures of internal constructions in the room. Airflow velocities were measured by anemometers. This paper presents only results from temperature measurements.
From the point of view of thermal comfort is recommended the maximal temperature difference between the position of a head and an ankle to be 2,0 K for standing person and 1,5 K for sitting person, see [2]. The legal regulation for working environment requires the maximal vertical temperature difference to be 3 K, see [1].

RESULTS

At fig. 2, 4 we can see the course of measured temperatures in chosen time intervals. Air temperatures are marked by a blue color and floor and ceiling temperatures by a red color. Vertical dashed lines indicate time moments in which were observed vertical distributions of temperatures in the room. In case A we can also see temperature of the air coming out of the gas space heater marked by black color at fig. 2.

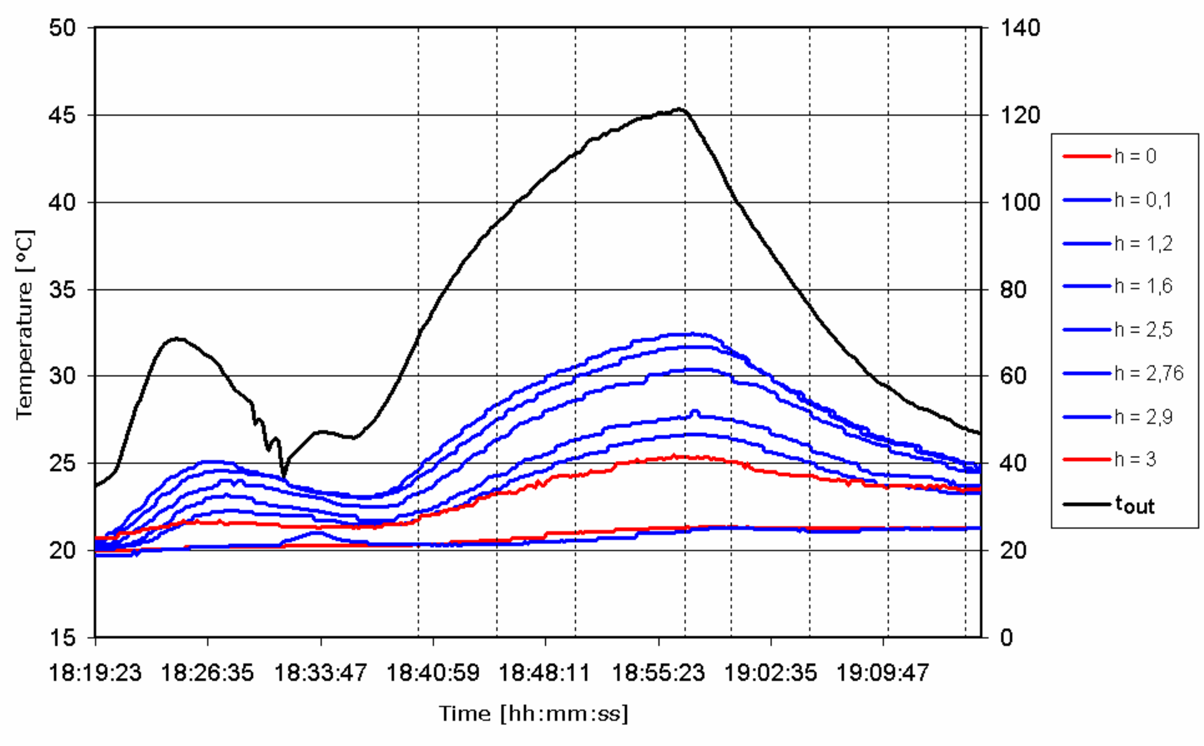


Figure 2. Temperature course in time, case A

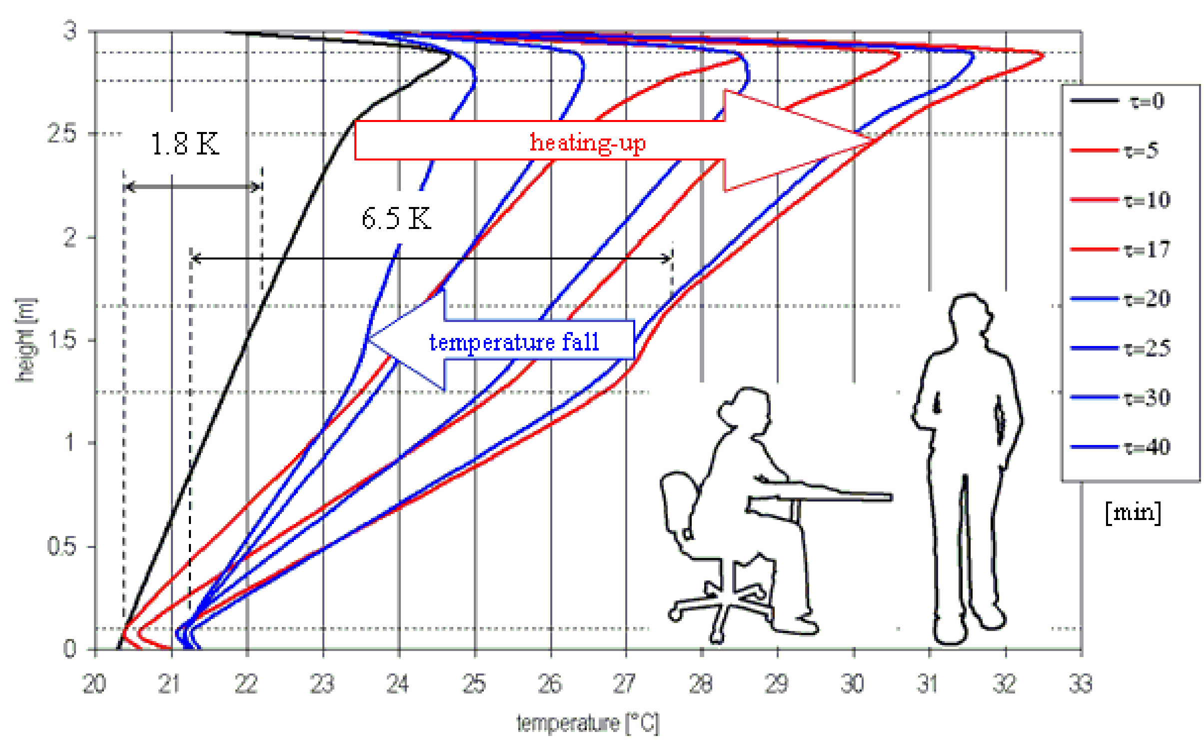
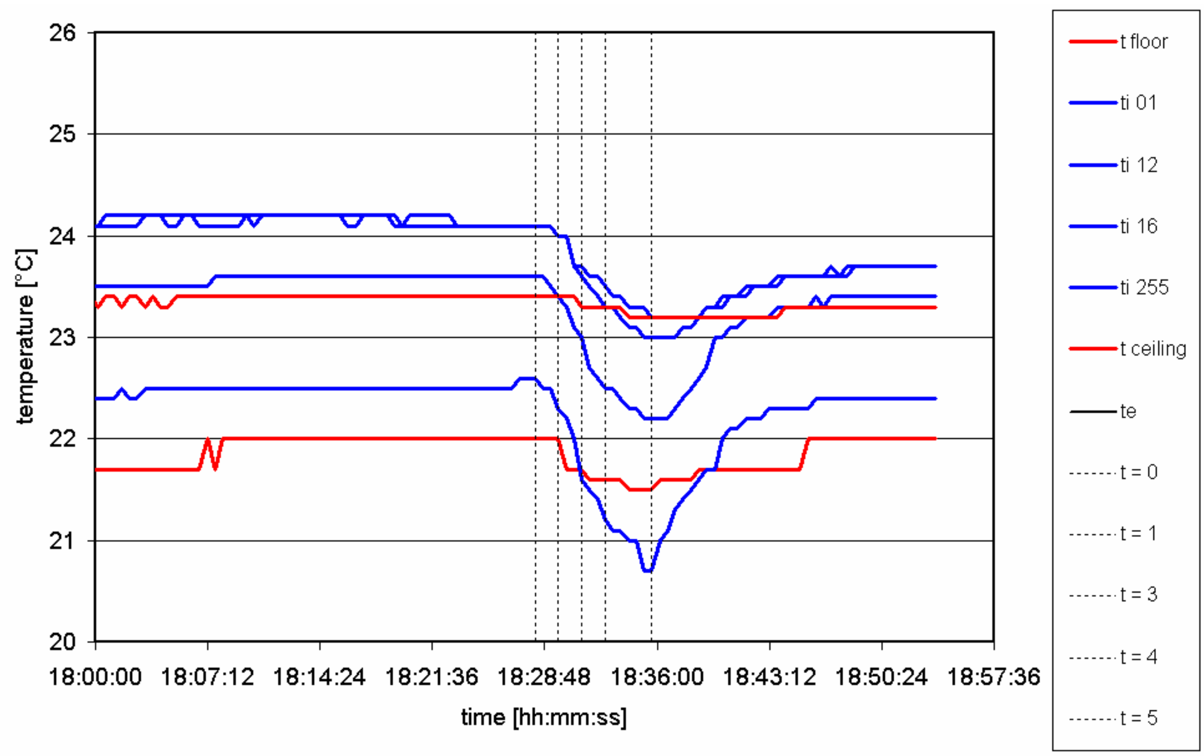


Figure 3. Vertical distribution of temperatures, case A



case B

Figure 4. Temperature course in time, case B

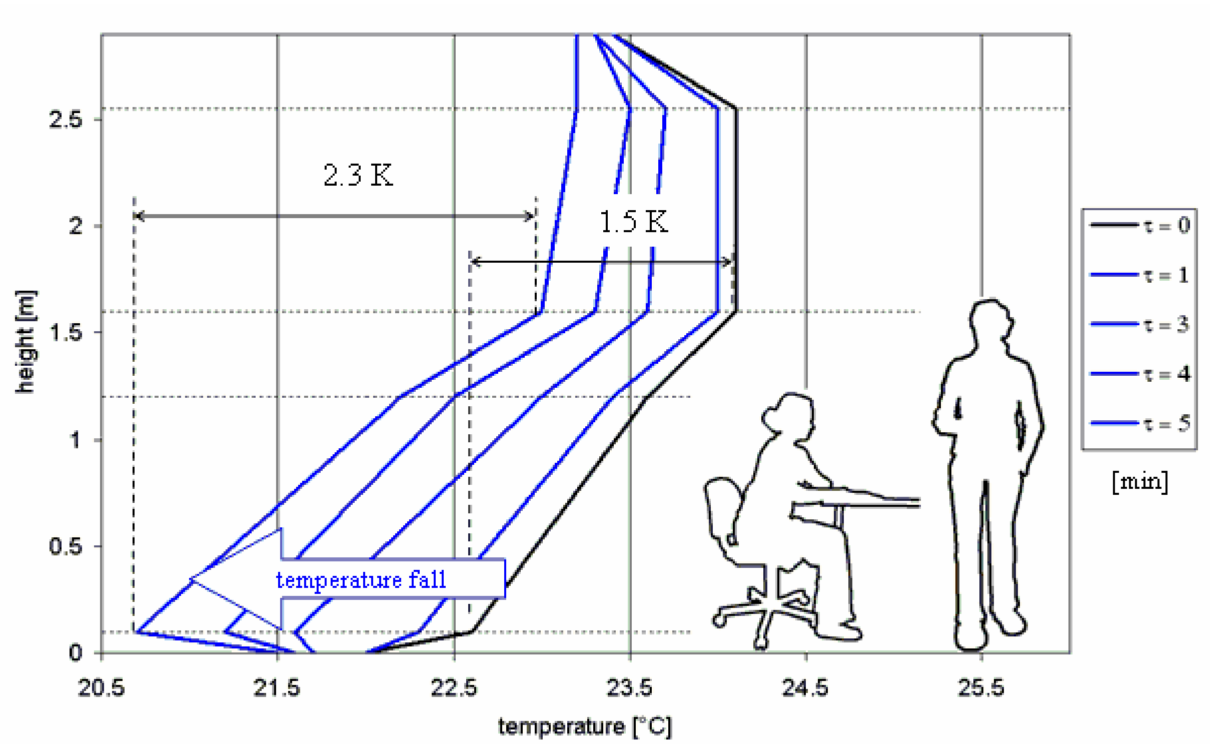
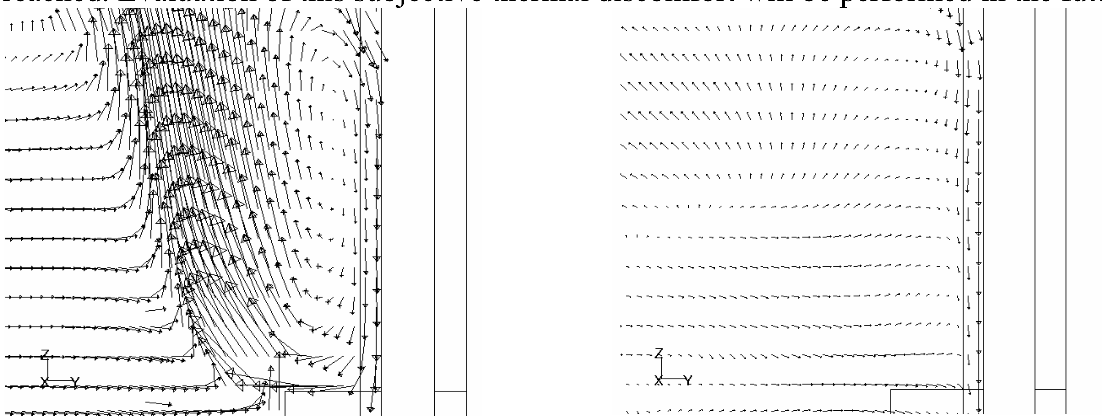


Figure 5. Vertical distribution of temperatures, case B

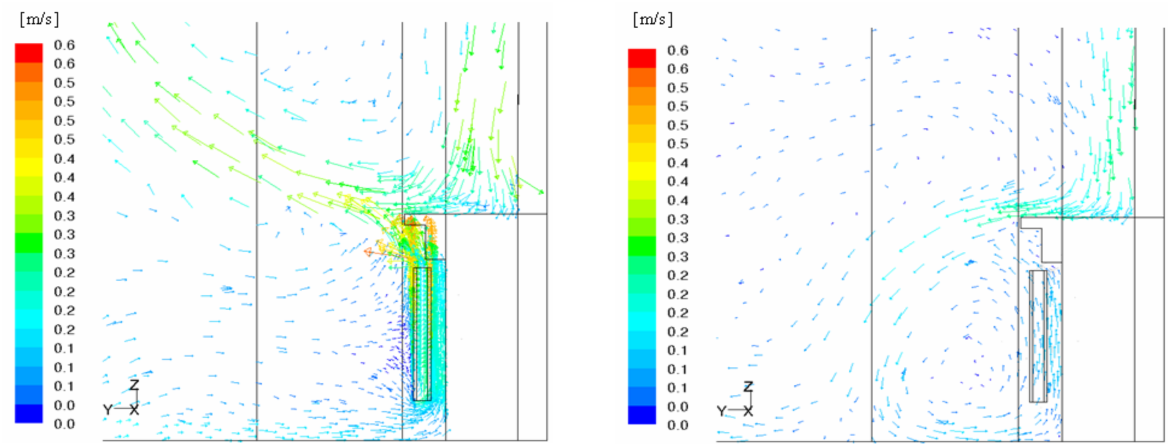
Obtained results show unfavorable vertical distribution of temperatures in the room for case A especially by pre-heating, see fig. 2. Inhabitants of this room are exposed to thermal discomfort because of the seriously overreached maximal temperature difference between the position of a head and an ankle required by [1] for time interval of approximately 30 minutes, see fig. 3. The gas space heater used in the room is equipped with jump regulation switch on-off. There is no possibility for step-by-step regulation or gradual control of heating, therefore the discomfort occurs several times per day, see fig. 9.

Case B offers favorable vertical distribution of temperatures in the room during a regular operation. While short-term ventilation by windows is formed a cold air layer close to the floor which cause a slight thermal discomfort felt by inhabitants. Nevertheless the maximal temperature difference between the position of a head and an ankle required by [1] is not overreached. Evaluation of this subjective thermal discomfort will be performed in the future.



Section A1 Section A2

Figure 6. Velocities, case A



Section B1 Section B2

Figure 7. Velocities, case B

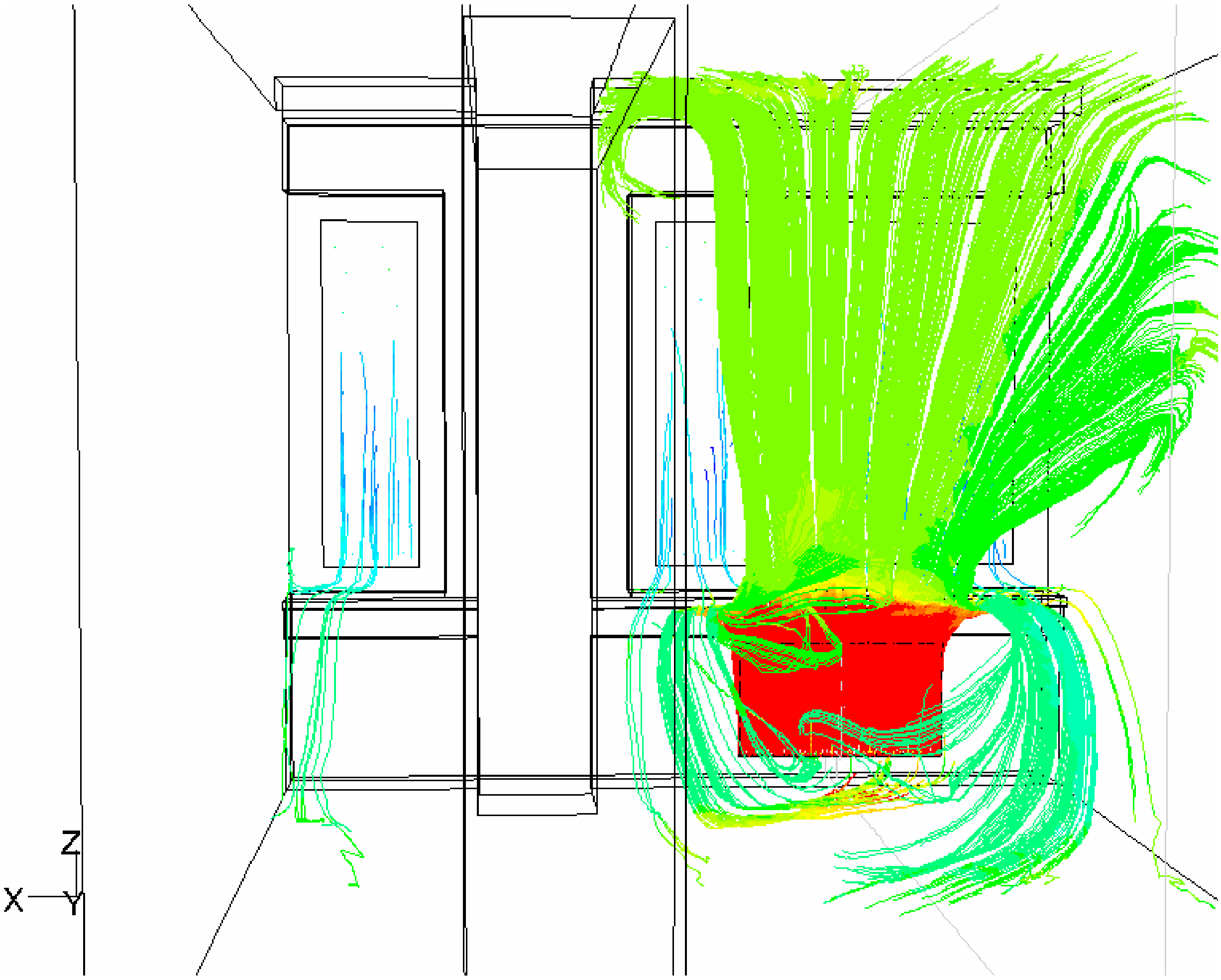


Figure 8. Path lines, case B

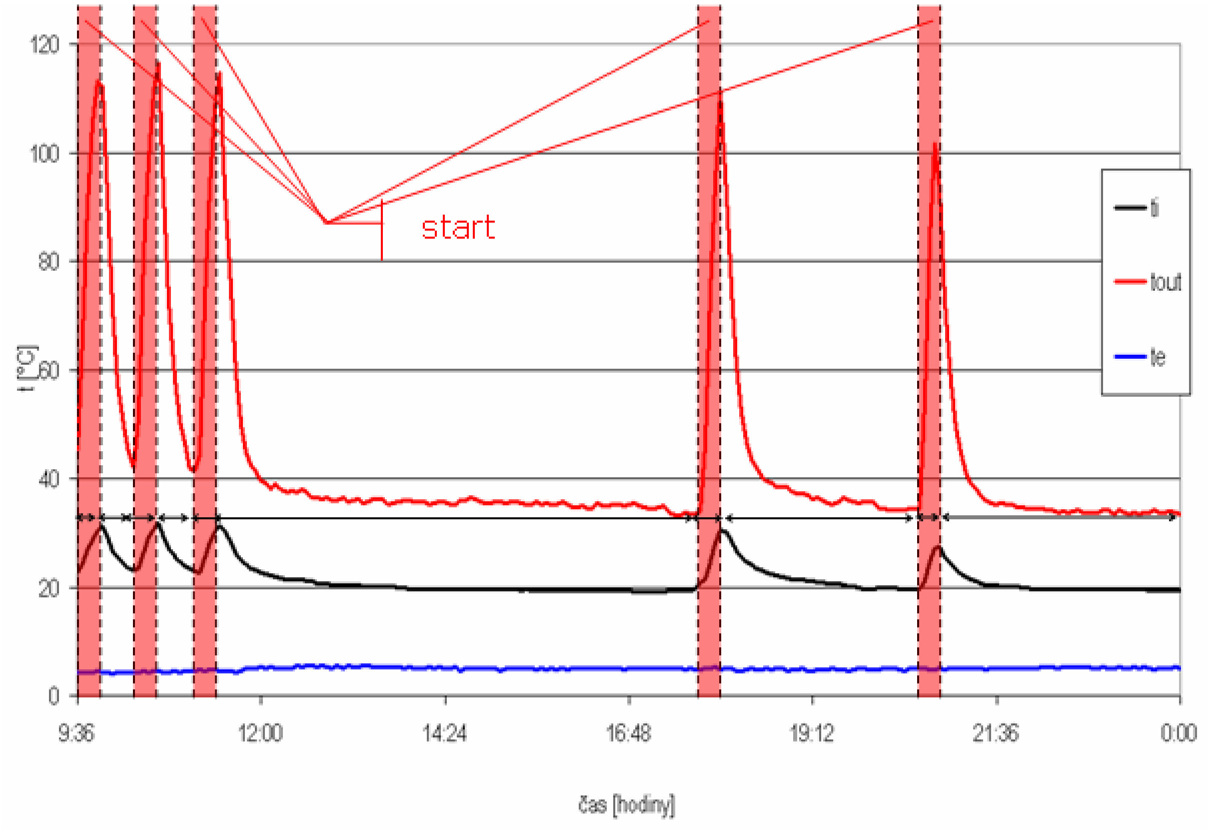


Figure 9. Temperature in the course of day, case A

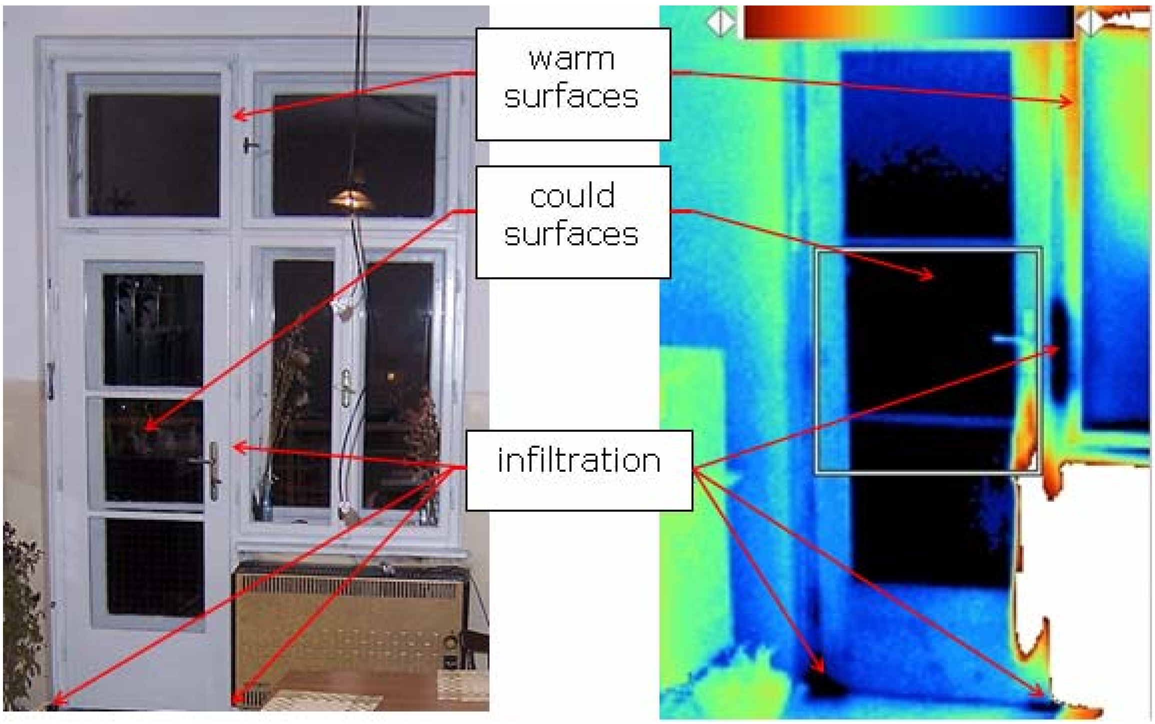


Figure 10. Surface temperatures, case A

DISCUSSION

Above mentioned and described results prove dependence of vertical distribution of temperatures in the heated room on time. In some dynamic states it can be considered as unfavorable, for example while pre-heating in case A, or while short-time ventilation by windows in case B. Following factors having unfavorable influence on vertical distribution of temperatures were found:

- ventilation by windows or high infiltration of the outer air
- the formation of dropping airflows at the vicinity of cool surfaces of outer walls and windows
- relatively high temperature of the heating air

Other factor causing unfavorable vertical distribution of temperatures is an insufficient shading out of dropping cool airflows by a heating air, see fig. 6, 7, 8. The reason for a relatively high temperature of a heating air in case A is above all the an unsuitable control system of a gas convector, see fig. 9.

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The use of thermal comfort approach in the energy conservation studies

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SUMMARY

There are numerous studies of energy conservation problems in buildings. In most of these studies the problem is handled as a study of heat loss or heat gain. This may be justified when the external conditions are too severe. However, when the summer or winter conditions are not very distinct, it is better to use the thermal comfort approach.

Thermal comfort approach for energy conservation problems can be applied to compare the same type of building elements with each other. There are a number of thermal comfort scales. In this study PMV (Predicted Mean Vote) which is proposed by FANGER and recommended by ASHRAE and ISO was used.

The thermal comfort evaluation can be done through a number of criteria. One of them can be "The total absolute PMV scale". The other criteria can be "The total time period in which the enclosure with a building element achieves thermal comfort". The tests can be carried out in air-conditioned, intermittently air-conditioned and none air-conditioned regimes.

Thermal comfort approach in energy conservation studies puts the human beings on the front row, rather than the energy cost. Another advantage of the method is to show which building elements use the solar heat gain effectively.

INTRODUCTION

There are numerous studies of energy conservation problems in buildings. In most of these studies the problem is handled as a study of heat loss or heat gain, either under stationary or variable conditions [1]. This may be justified when the external conditions are too severe. However, when the summer or winter conditions are not very distinct, in other words when we cannot tell whether the climate is hot or cold, it is better to use the thermal comfort approach.

There are a number of thermal comfort scales [2, 3]. They can be grouped into two categories as empirical and analytical indices. In this study an analytical index, PMV (Predicted Mean Vote), which is proposed by Fanger [4] and recommended by ASHRAE [5] and ISO [6] was used.

METHODS

Thermal comfort is defined as the state in which the body adapts itself to the environment by spending the least amount of energy ASHRAE (2001). It is possible to divide the thermal comfort factors into two groups as objective and subjective. Objective factors are air temperature, relative humidity, air velocity and radiation. Subjective factors are the activity level, metabolic rate, thermal insulation of clothing, dieting habits, sex, age, shape of the body and acclimatization.

The thermal comfort index proposed by Fanger (1970-1982) is an analytical one, and developed for the steady conditions of human environment. Later with the works of (Gagge, et al., 1986) the index was developed in order to measure thermal comfort in varying environments.

Fanger's index computes the heat exchange between human body and the environment and predicts the response. The inputs of the model are air temperature, relative humidity, air velocity, mean radiant temperature, thermal insulation of the clothing, activity level of the human body, and the instantaneous heat storage of the human body due to the activity level. The model gives Predicted Mean Vote (PMV) according to the following scale in Figure 1.

Hot	Warm	Slightly Warm	Comfort Scale Range			Slightly Cool	Cool	Cold
+ 3	+ 2	+ 1	+ 0.5	0	- 0.5	- 1	- 2	- 3

Figure 1. ASHRAE Thermal Sensation Scale

Both the ASHRAE Standard 55 (1989) and ISO Standard 7730 (1994) indicate that for the + 0.5 and – 0.5 thermal sensation scale interval, at least 80% of human beings will feel thermally comfortable while 20% might be dissatisfied. With this model it is possible to find Percentage of Persons Dissatisfied (PPD) for any PMV value.

PMV can be calculated with Equation 1 given below:

$$PMV = (0.303\, e^{-0.036M} + 0.028)\{(M-W) - 3.05 \times 10^{-3} \times [5.733 - 6.99\,(M-W)\, p_a] - 0.42 \times [(M-W) - 58.15] - 1.7 \times 10^{-5} M(5.867 - p_a) - 0.0014\, M(34 - t_a) - 3.96 \times 10^{-8} f_{cl} \times [(t_{cl} + 273)^4 - (t_r + 273)^4] - f_{cl} h_c\, (t_{cl} - t_a)\} \quad (1)$$

The parameters in Equation 1 can be found with the following equations.

$$t_{cl} = 35.7 - 0.028(M-W) - I_{cl}\{3.96 \times 10^{-8} f_{cl} \times [(t_{cl} + 273)^4 - (t_r + 273)^4] + f_{cl} h_c\, (t_{cl} - t_a)\}$$

hc =		$2.38(t_{cl} - t_a)^{0.25}$ for $2.38\,(t_{cl} - t_a)^{0.25} > 12.1\,(v_{ar})^{1/2}$ $12.1\,(v_{ar})^{1/2}$ for $2.38\,(t_{cl} - t_a)^{0.25} < 12.1\,(v_{ar})^{1/2}$
fcl =		$1.0 + 1.290\, I_{cl}$ for I_{cl} $0.078\ m^2\ {}^oC/W$ $1.05 + 0.645\, I_{cl}$ for $I_{cl} > 0.078\ m^2\ {}^oC/W$

where PMV is the predicted mean vote, M is the rate of metabolic heat production (W/ m^2), W is the rate of mechanical work accomplished (W/ m^2), I_{cl} is the thermal resistance of the clothing (m^2°C /W), f_{cl} is the external surface area of a dressed person (m^2), t_a is the air temperature °C, t_r is the mean radiant temperature °C, v_{ar} is the air velocity (m/s) , p_a is the partial water vapor pressure (Pa or N/m^2), h_c is the convective heat transfer coefficient (W/m^2°C), t_{cl} is the surface temperature of clothing (°C).

It is recommended that PMV scale thus calculated should be in the range of -2, + 2 and the six parameters used in the calculation of PMV should be in the following ranges:

M = 46 W/m^2 to 232 W/m^2 (0.8 met to 4 met)
I_{cl} = 0 m^2.^{o}C/W to 0.310 m^2.^{0}C/W (0 clo to 2 clo)
t_a = 10 ^{o}C to 30 ^{o}C
t_r = 10 ^{o}C to 40 ^{o}C
v_{ar} = 0 m/s to 1 m/s
p_a = 0 Pa to 2700 Pa

For any PMV value the percentage of dissatisfied people (PPD) can be found by equation 2.

$$PPD = 100 - 95 \ x \ e^{-A} \qquad (2)$$
$$A = 0.033\,53 \ x \ PMV^4 + 0.217\,9 \ x \ PMV^2$$

In the calculation of current thermal comfort indexes “Mean Radiant Temperature” is used as the input, instead of “Air Temperature” because it is more effective on thermal comfort. Mean Radiant Temperature is the uniform temperature of an imaginary enclosure in which radiant heat transfer from the human body equals the radiant heat transfer in the actual non-uniform enclosure.

Mean Radiant Temperature can be calculated from Globe Thermometer Temperature, Air Temperature and Air Velocity measurements according to the Equation 3. This equation has been written with the assumption that most of the materials enclosing a space have a very high emittance value.

$$T_r^4 = T_1^4 \ . \ F_{p-1} + T_2^4 \ . \ F_{p-2} + ... + T_n^4 \ . \ F_{p-n} \qquad (3)$$

where T_r is the Mean Radiant Temperature (K), T_n is the temperature of surface n (K), F_{p-n} is the angle factor between the (n)th surface and human body.

When the differences between the internal surface temperatures of the enclosure are not very high Equation 3 can be written as an equation of first degree as in Equation 4.

$$\mathrm{T_r = T_1 \ . \ F_{p-1} + T_2 \ . \ F_{p-2} + ... + T_n \ . \ F_{p-n}} \qquad (4)$$

The angle factors in these equations can be found by the aid of Figure 2.

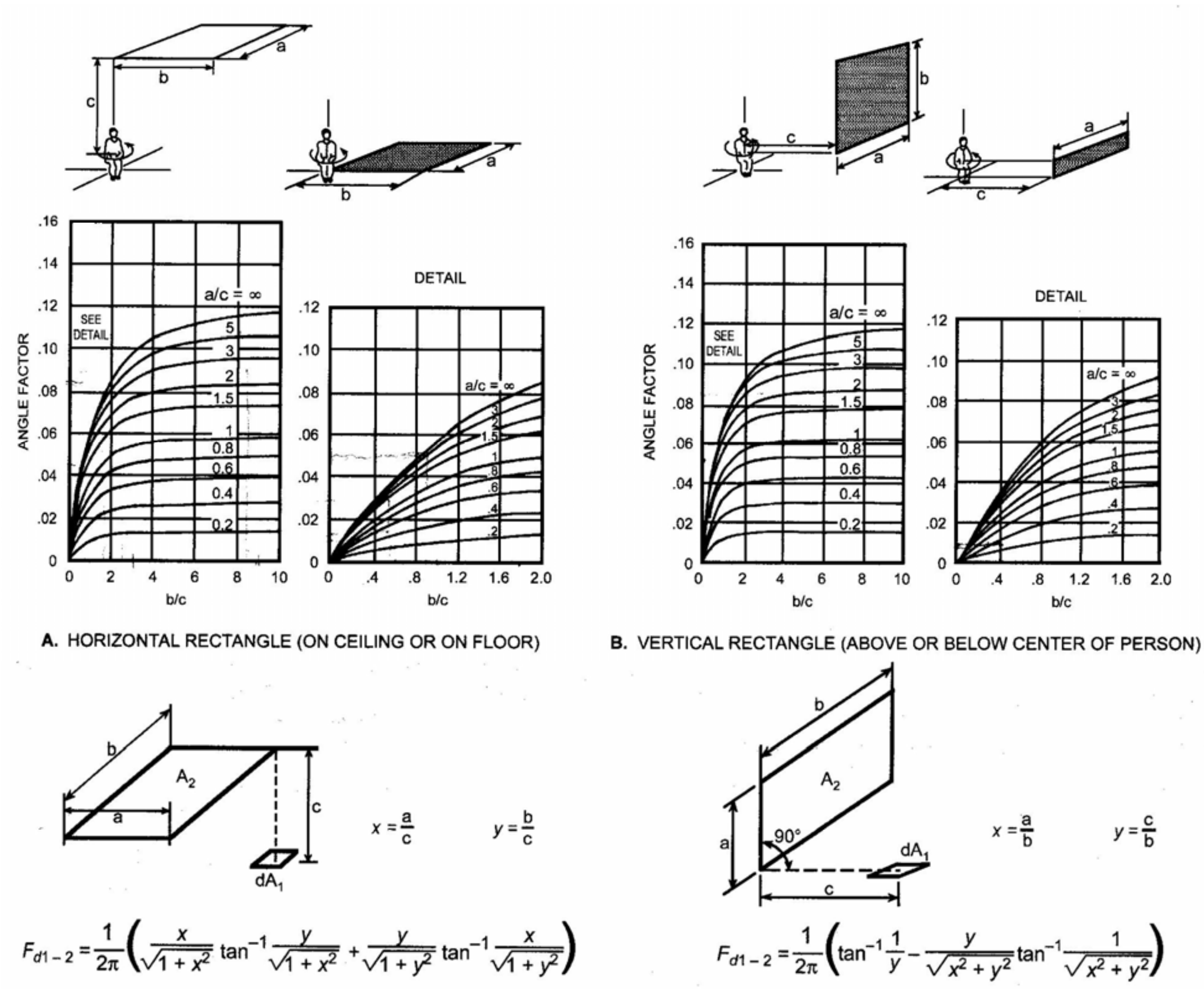

Figure 2. The graphs and the equations necessary to find the angle factor (ASHRAE, 2001) .

Plane Radiant Temperature (t_{pr}), defines the heat flow in one direction and its magnitude depends on that direction. Mean Radiant Temperature can be found from Plane Radiant Temperature as well.

(For a person standing)

$$T_r = \{0.08[t_{pr(up)}+t_{pr(down)}]+0.23[t_{pr(right)}+t_{pr(left)}]+0.35[t_{pr(front)}+t_{pr(back)}]\}\div[2(0.08+0.23+0.35)] \quad (5)$$

(For a sitting person)

$$T_r = \{0.18[t_{pr(up)}+t_{pr(down)}]+0.22[t_{pr(right)}+t_{pr(left)}]+0.30[t_{pr(front)}+t_{pr(back)}]\}\div[2(0.18+0.22+0.33)] \quad (6)$$

Radiant Temperature Asymmetry (Ä t_{pr}), is the difference between Plane Radiant Temperatures of the opposing surfaces. Sometimes the direction of the heat flow is given as from this surface to that surface, eg from the floor to the ceiling. If the direction has not been mentioned it is assumed as the direction which gives the maximum difference.

With this method it is possible to compare some building elements. In order to compare them air temperature, relative humidity, air velocity, mean radiant temperatures should be measured continuously at least for one year on test houses. The index also requires the values of thermal insulation of the clothing, activity level of the human body, and the instantaneous heat storage of the human body due to the activity level.

RESULTS AND DISCUSSION

A number of assumptions should be made to simplify the experiments.

1. As a person moves inside a building the angle factor will also change. Since in these experiments, we aim to compare either some buildings or some building elements, it can be reasonable to find the thermal sensation for a person sitting on a chair at the center of the space each time.
2. Average clothing values will normally change between summer and winter. Thus for November, December, January, February, March and April (Winter months) average clothing value can be assumed 0.8 clo and for the other months it can be assumed 0.5 clo.
3. Human activity level can be taken as 1.3 met which characterizes a person in rest or with light activities.
4. Air change number can be assumed 0.15 for summer and 0 for winter months or the actual measured values can be used.

With the data obtained at least for one year, a number of criteria can be tested. One of them is the yearly total of the absolute values of PMV. When the internal space is cool PMV will have a negative value and when it is warm PMV will be positive. Building or building element with the least total PMV absolute value has the best performance. These can be shown on a graph. If the graph prepared for this purpose shows the yearly totals together with the overheated and under heated period totals it will be very useful. Under heated period is the period when the climatic conditions are below the thermal comfort conditions and overheated period is the rest of the year.

Another criterion can be, the yearly total time when the thermal comfort is achieved (that is the total time when PMV is between -0.5 ~ +0.5). The maximum values of the yearly total comfort time indicate the best performance.

Yet another criterion can be to find the behavior of the building, on typical days of the year like 21st of July (summer), 21st of January (winter), 21st of April (spring) and 21st of October (autumn).

All of these criteria can be tested for non-air-conditioned, continuously air conditioned and intermittently air conditioned conditions. The method was applied to study the performance of fourteen roof constructions for a warm climate (OZDENIZ & HANCER, 2005). The best performed roof construction depends on how the building is used and air conditioned.

CONCLUSIONS

Thermal comfort approach in energy conservation studies puts the human beings on the front row, rather than the energy cost. We can reach to better conclusions to suit the wellbeing of the users with this approach. Certainly a number of assumptions can be made for the ease of the computations. More accurate results can be derived, if the assumptions are close to the actual conditions.

Another advantage of the method is to show which building elements use the solar heat gain effectively. Sometimes solar heat gain effects thermal comfort positively, as in the cool season. Sometimes its effect is negative, as in the warm season. Thermal comfort approach considers this phenomenon.

ACKNOWLEDGEMENT

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Evaluation on thermal environment installed ventilating fans in the rotunda at new national museum of Korea

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SUMMARY

In order to improve thermal comfort in the Rotunda, which is high and wide visiting space of the new national museum of Korea, eight ventilating fans were installed near the ceiling of Rotunda. It has been analyzed thermal comfort of Rotunda with/without ventilating fans by numerical simulation. To evaluate thermal comfort of the Rotunda, well-known indices, PMV and PPD were introduced. The results of present investigation show that temperature distribution of the case with fans is closer to target temperature than the case without fans at the breathing zone. In the case without fans, thermal stratification with 16℃ of temperature difference occurs along the height of the Rotunda which makes the thermal environment worse and the PPD values reach up to 50% in the 6th floor connection passage. In the case with fans, however, the vertical temperature difference was reduced to 9℃ and the PPD values were lower below 20%. Consequently, the ventilating fans adopted on this study are effectively used for improving the thermal comfort in large space structure with thermal stratification.

INTRODUCTION

Recently, as the living standard increases and the building technique develop, construction of large-space buildings has been increasing. Such structures accommodate gymnasiums, theaters, waiting rooms, aquariums, conference halls, and so forth, and most of them have the form of high ceilings and large open spaces. In these large buildings, glass is usually used as a roof or wall material to ensure a fine view and save energy. Such transparent outer wall materials as glass create environment-friendly outdoor atmospheres and bring in natural daylight, so they have advantages of not only producing cultural, economic and functional benefits from spatial characteristics, but also providing a visual sensation and freshness. But, it has a disadvantage of weakening the resistance to changes in the external environment, such as solar radiation or outdoor conditions rapidly worsen the indoor thermal environment. Also, it may have inefficiencies, such as thermal buoyancy has a great effect on indoor air flow, and the ratio of residing area to entire space reduces. In a large space, particularly in summertime, radiant heat from the roof or sidewall and thermal stratification in the interior may cause thermal discomfort to occupants. Thus, further studies of air conditioning and thermal environments are needed to predict the characteristics of air and temperature distribution in large spaces and present or solve such problems.

As a large-space structure, the Rotunda, the lobby of the National Museum of Korea currently under construction, serves as a traffic line to other spaces, a lounge, a meeting place, and the

passages to exhibition halls; in addition, most sections are constructed with glass to create environment-friendly outdoor atmospheres for visitors and introduce natural daylight. Particularly in summer, since the radiant load by sunshine comes in from glass walls, it is necessary to predict the indoor thermal environment of the Rotunda in advance. To this end, a study has been conducted to evaluate the thermal environment in case of introducing a ventilating system into its existing design and suggested that a plan should be prepared to solve the phenomenon of thermal stratification near the ceiling[1].
Therefore, this study introduces the technique of computational flow dynamics and analyzes the thermal environment of the Rotunda in summertime, with a view to investigating the adequacy of the ventilating fan installed as a plan to eliminate the thermal stratification.

THERMAL EVALUATION METHOD

Indoor thermal comfort or discomfort indicates a mental state that occupants feel in a given thermal environment, and some standardized methods for evaluating thermal comfort or discomfort have been suggested. This study uses the predicted mean vote (PMV) and predicted percentage of dissatisfaction (PPD) indices to evaluate thermal comfort. Detailed procedures for computing these indices are based on ASHARE Handbook [2]. Among the data necessary for PMV calculation in case of air conditioning in this study, other parameters than air current, temperature and mean radiant temperature are computed as shown in Table 1, which takes into account the viewing condition of the museum in summer.

Table1 Engineering data to estimate PMV

Parameter	Values
Metabolic rate	1.2met
Clothing value	0.5clo
Relative humidity	50%

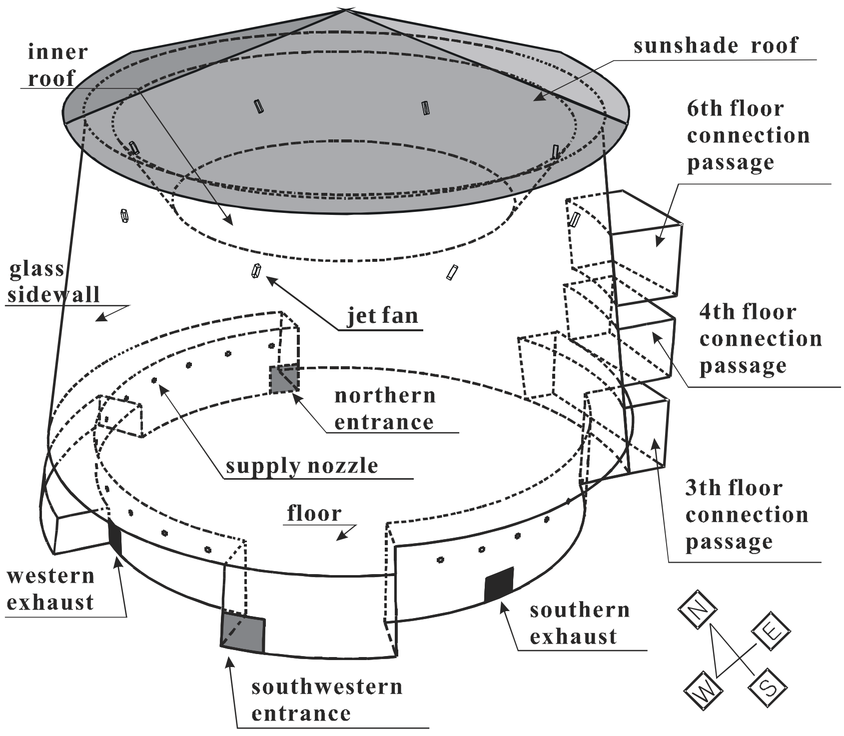


Fig. 1 Schematic diagram of the Rotunda

RESEARCH MODEL

Fig. 1 shows the schematic diagram of the Rotunda to be examined in this study. It has a shape of truncated cone with the greatest diameter of 48m and the greatest height of 38m. As shown in the figure, it consists of floor, glass sidewall, ceiling and sunshade roof and connected to the passages to adjacent exhibition halls.
To air-condition the interior in summer, the Rotunda has 21 supply nozzles along its edge and two exhausts in the south and west, respectively. The detailed characteristics of its interior structure, ventilation system and air conditioning load are the same as in the previous study[1]. Table 2 shows the details of summertime air conditioning load for this research model. The cooling load imposed to the glass sidewall by solar radiation is 91.14kW, about 54.5% of the total cooling load of 167.33 kW. We assume that among this, 30% is applied to the sidewall and the remaining 70% permeates the sidewall and is applied to the floor. Also, we assume that the load of 34.38kW caused by the lighting equipment installed around the glass roof of the Rotunda occurs entirely at the underside of the glass roof; the indoor sensible heat caused by visitors, a total of 8.62kW, is applied to the floor. Sine the latent load is the evaporation heat that occurs by evaporation of indoor moisture, it just increases indoor humidity but not indoor temperature. Thus, this study assumes that it is dealt with the air conditioner and thus does not include it in the cooling load.
To eliminate the phenomenon of thermal stratification, eight ventilating fans are installed on the ceiling at equal angles. Horizontally, they lean toward an adjacent fan clockwise; vertically, they are installed at 25° downward. The discharging air velocity and air volume of each fan are up to 17.5m/s and 3,000 CMH, respectively.

Table2 Cooling load in the rotunda (unit: kW)

Location	Cooling loads
Inner roof	7.07
Glass sidewall(by conduction)	17.33
Glass sidewall(by radiation)	91.14
Floor	8.79
Lighting equipments	34.38
Sensible heat gain by the gallery	8.62
Total	167.33

NUMERICAL ANALYSIS METHOD

The governing equations of a normal state that describes the characteristics of air current and temperature in the Rotunda chosen in this study are continuity equation, momentum equation, turbulent kinetic energy equation, dissipation rate equation of turbulent kinetic energy, and energy equation, whose details are the same as in references[1].
This study uses the commercial CFD code, FLUENT, to interpret the governing equations mentioned above[3]. While the non-staggered grid system is used for analysis of velocity and temperature field, the staggered grid system is introduced for proper prediction of the rapid

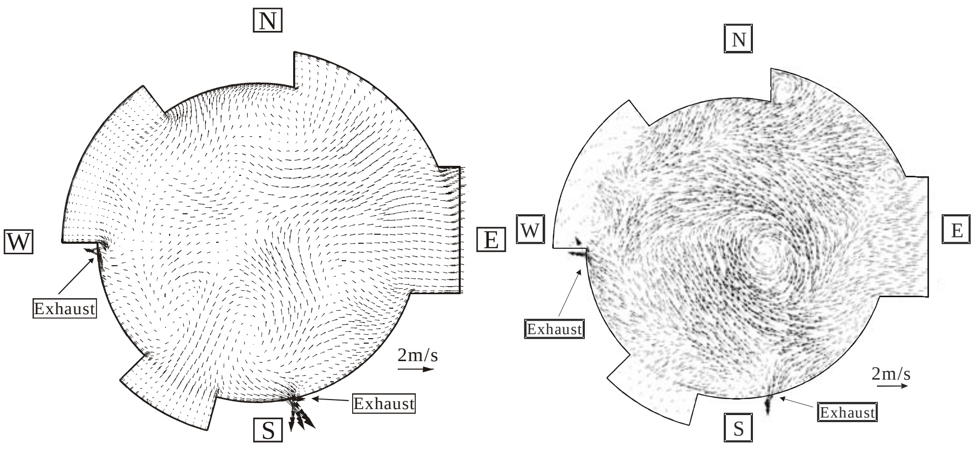
changes in pressure values occurring around the wall due to the load through the wall surface[4].
Table 3 shows the details of the boundary condition defined with the governing equations.

Table3 Boundary conditions for present study

Location	Conditions
Roof	$u = v = w = 0$, $q = 30.9\,\mathrm{W/m^2}$
Glass sidewall	$u = v = w = 0$, $q = 11.0\,\mathrm{W/m^2}$
Floor	$u = v = w = 0$, $q = 59.6\,\mathrm{W/m^2}$
Passages	$\frac{\partial u}{\partial n} = \frac{\partial v}{\partial n} = \frac{\partial w}{\partial n} = 0$, $\frac{\partial T}{\partial n} = 0$
Supply nozzle	$V_n = 6.6\,\mathrm{m/s}$, $T = 18$℃
Ventilating fan	$V_n = 17.5\,\mathrm{m/s}$(3000CMH)
Southern exhaust	$V_n = -1.62\,\mathrm{m/s}$, $T = 26$℃
Western exhaust	$V_n = -1.875\,\mathrm{m/s}$, $T = 26$℃
Entrance	$u = v = w = 0$, $q = 0.0\,\mathrm{W/m^2}$

RESULTS AND DISCUSSION

In a previous study of the indoor thermal environment of the Rotunda equipped with the initial ventilation system, we could observe that the dead air occurring in the upper part of the Rotunda severely worsens the thermal environment around the 4th and 6th floor passages. To mend this problem, this study explores the effects of the ventilating fans designed to mix the indoor air evenly across the entire room. To this end, we select as checking points the horizontal plane at the breathing zone (1.5m high from the floor) and the vertical plane at the center of the Rotunda. We examine the distributions of air, temperature and PMV/PPD at these checking planes.



(a) Without ventilating fans (b) With ventilating fans

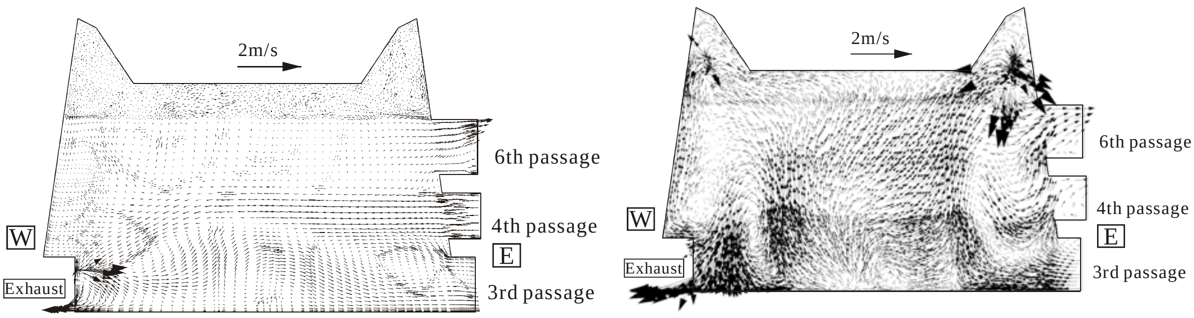
Fig. 2 Distributions of velocity vectors at breathing zone of the Rotunda

EVALUATION OF AIR ENVIRONMENT

To examine the effects of ventilating fans on the indoor thermal environment of the Rotunda during air conditioning in summer, we depict in Fig. 2(a) and (b) what distributions the air has at the breathing zone of the horizontal plane, whether or not ventilating fans are installed.

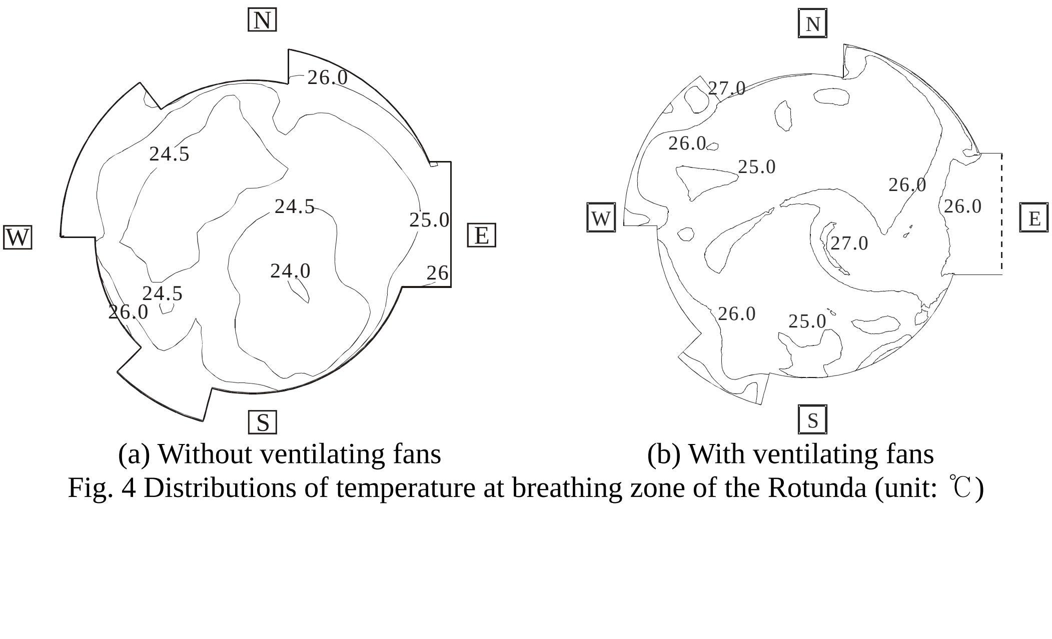
The results of air flow analysis reveal that the average wind velocity at the breathing zone increase by about 0.2m/s when the fans are installed, so the velocity at the breathing zone shows a range of 0.2 ~ 0.6m/s. As shown in Fig. 2(b), there is few area of dead air. But, because of the direction of the ventilating fans, the air flow is characterized by swirling clockwise centering the middle point of the Rotunda. This swirling flow is more active in the north and south, while somewhat weak around the 3rd connection passage and the exhausts. However, an area of dead air takes place in the center of swirling flow, so the cool air supplied form the supply nozzles cannot properly reach there, probably increasing the temperature a little. It is thought that the overall environment of air flow may provide satisfactory viewing conditions with visitors.

Fig. 3 shows the distributions of air currents at the vertical plane, with or without ventilating fans. Without ventilating fans, most of the air accumulates at the height of above the 4th connection passage; however, with ventilating fans installed, the indoor air rises or falls everywhere in the Rotunda, indicating that it circulates smoothly. This smooth rising and falling of the air is due to the effects of the ventilating fans installed on the ceiling. Thus, we can expect that these effects of ventilating fans will reduce the phenomenon of thermal stratification that results from the disproportionate air distributions between the upper and lower parts in a high-storied large space.



(a) Without ventilating fans (b) With ventilating fans

Fig. 3 Distributions of velocity vectors at vertical plane of the Rotunda



(a) Without ventilating fans (b) With ventilating fans

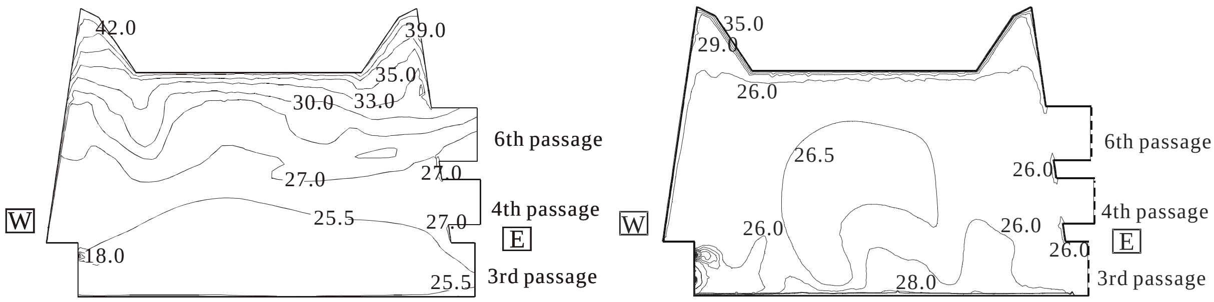
Fig. 4 Distributions of temperature at breathing zone of the Rotunda (unit: ℃)

EVALUATION OF THERMAL ENVIRONMENT

To examine the characteristics of temperature distribution in the Rotunda, we depict in Fig. 4(a) and (b) the temperature distribution in the same plane as we examined the distribution of air flow.

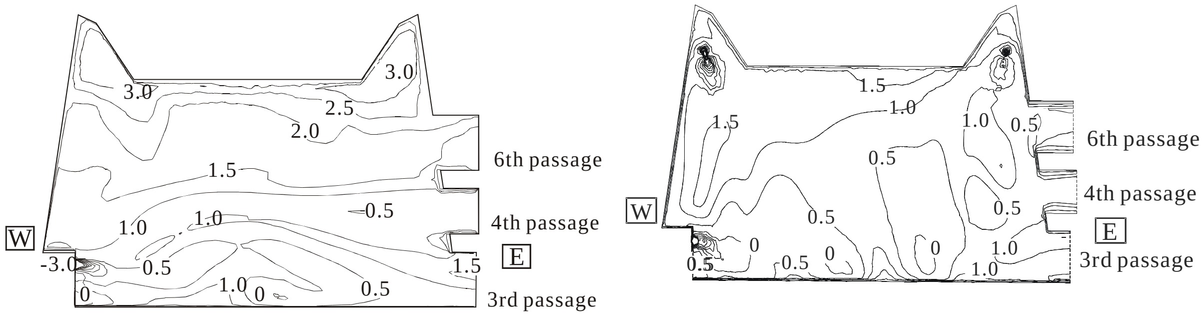
As shown in Fig. 4(a), without ventilating fans, the temperature distribution at the breathing zone shows 26℃ around the northeastern and northwestern walls, but the remaining areas have a range of 24 ~ 25℃, which is a little lower than the temperature of 26℃ set as an indoor temperature by air conditioning. This is because the cold air supplied by the supply nozzles for eliminating cooling load does not circulate all over the space but arrives only at the lower part (lower than the 4th connection passage).

In contrast, with the ventilating fans installed, as shown in Fig. 4(b), most areas have a temperature of about 26℃. Presumably, this is because ventilating fans circulate properly the air in the interior of the Rotunda. The reason that the areas of southern entrance and northwestern wall have a temperature of 27℃ is that most of the air diffused from the wall moves by forming a swirling air flow but recirculates there on a small scale. This phenomenon of air recirculation also takes place at the center of the Rotunda, thus showing a temperature distribution of 27℃.



(a) Without ventilating fans (b) With ventilating fans

Fig. 5 Distributions of temperature at vertical plane of the Rotunda (unit: ℃)



(a) Without ventilating fans (b) With ventilating fans

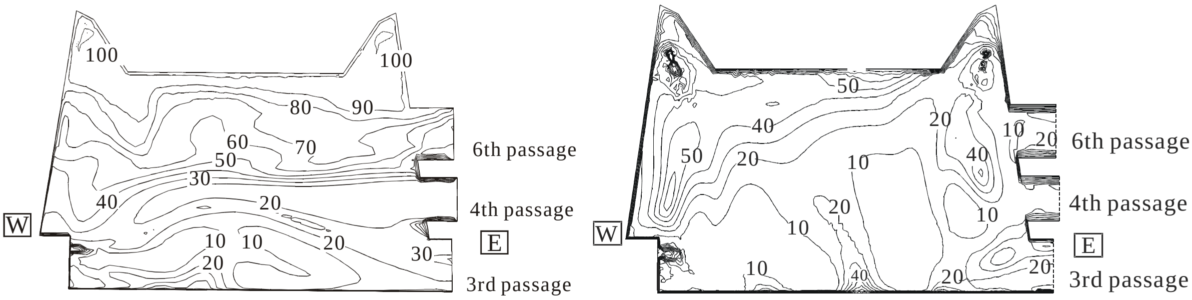
Fig.6 Distributions of PMV at vertical plane of the Rotunda

To examine how much the ventilating fans improve the phenomenon of thermal stratification at the upper region of the Rotunda, we depict in Fig. 5(a) and (b) the temperature distributions at the vertical plane of the Rotunda with ventilating fans installed and not installed.

As shown in Fig. 5(a), while a swirling air flow takes place, the lower region of the Rotunda has a temperature of about 25℃, which is thought to be an agreeable viewing environment in summer; however, the temperature near the roof where dead air exists reaches up to 42℃, thus showing a large difference of 16℃ in temperature between the upper and lower regions. This is probably because the radiant heat through the glass roof warms the indoor air of the

Rotunda. This temperature distribution is a common thermal characteristic in a high-storied large space where a ventilating system is not introduced.
In comparison, with ventilating fans installed, as shown in Fig. 5(b), because a swirling airflow takes place all over the space, the lower region as well as the upper region shows a temperature distribution of about 26C. Though some regions show a temperature of over 30℃, they are very small. This is because the ventilating fans mix properly the indoor air.

EVALUATION OF COMFORT INDEX
To evaluate the comfort sensation that the indoor environment of the Rotunda provides for visitors, we examine the distributions of PMV and PPD frequently used for comfort evaluation.
To examine how much ventilating fans improve the indoor thermal environment by solving the phenomenon of thermal stratification in the upper region of the Rotunda, we depict the distribution of PMV at the vertical plane of the Rotunda in Fig. 6(a) and (b), and the distribution of PPD at the same location in Fig. 7(a) and (b), with and without ventilating fans.



(a) Without ventilating fans (b) With ventilating fans
Fig.7 Distributions of PPD value at vertical plane of the Rotunda (unit: %)

As shown in Fig. 6(a), without ventilating fans, the PMV indices at the residing place, the third passage and the passage to the Gallery of History have a range of 0 ~ 0.5, indicating that these areas are regarded thermally comfortable. Also, the PMV at the 4th passage reaches up to 1, which is thought to be a relatively appropriate environment for visitors; however, that at the 6th passage shows more than 1.5, which is thought to be a disagreeable environment that visitors feel hot.
If ventilating fans are installed as shown in Fig. 6(b), the PMV indices at the residing place, the third passage and the passage to the Gallery of History have a range of 0 ~ 0.5, which is regarded thermally comfortable. Since the PMV at 4th and 6th passages shows up to 1, these areas are also thought to be a relatively suitable for visitors.
As shown in Fig. 7(a) describing the predicted percentage of dissatisfaction (PPD) in a given thermal environment, we can see that if ventilating fans are not installed, the PPD value at the residing area of the Rotunda is about 10%, indicating that visitors will be able to feel comfortable. The PPD at the 3rd and 4th passages is 20%; thus it can be said that considering these are part of the museum, they are a relatively suitable thermal environment. However, the 6th passage shows about 50% of PPD. This means that 50 out of 100 visitors express dissatisfaction with such a thermal environment. Probably, this results from the effects of the thermal stratification that takes place at the upper region of the Rotunda.
If ventilating fans are installed as shown in Fig. 7(b), the PPD at the residing place in the lower part of the Rotunda is about 10%, indicating that visitors will be able to feel comfortable. The 3rd, 4th and 6th passages show 20% of PPD. This means that considering

these are part of the museum for visitors, such a condition is a relatively agreeable thermal environment.
Thus, it can be concluded that if we install ventilating fans and generate the recirculation air through the Rotunda, we can easily solve the problem of the dead air that take place at the upper region of the Rotunda with no ventilating fan installed.

CONCLUSION

To prevent the worsening of thermal environment due to the dead air occurring at the upper region of the Rotunda in summer, this study has analyzed the distributions of air and temperature in the Rotunda with ventilating fans installed, using the commercial CFD code, FLUENT, and conducted evaluation of thermal environment with PMV and PPD. Thus, we have derived the following results.
(1) If ventilating fans are not installed, the temperature distributions at the breathing zone show a range of 24 ~ 25℃, which is a little lower than 26℃, the temperature set as an indoor air conditioning in summer; if ventilating fans are installed, the temperature distributions shows about 26℃.
(2) At the vertical plane on the center of the Rotunda, with no ventilating fan installed, the temperature difference between the upper and lower regions is large, about 16℃; with ventilating fans installed, the air circulates smoothly across the space so that the temperature distributions show about 26℃ not only in the lower region but also in the upper region.
(3) With no fans installed, since the dead air occurs in the upper region of the Rotunda, it is predicted that thee 6^{th} passage will have about 50% of PPD; however, with ventilating fans installed, the PPD has reduced to about 20%. Thus, we can conclude that producing a swirling air flow in the whole Rotunda will be able to solve the worsening of thermal comfort satisfactorily.

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Time Series Analysis of Cool Chair Operating Conditions

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SUMMARY

The operating conditions of a personal air-conditioning system in repeated subjective experiments were determined. When the right to adjust or control the environment was provided, psychological acceptability improved according to the adaptive model. Although a psychological adjustment is the most effective, it is difficult to evaluate quantitatively. A personal air-conditioning system, called the “Cool Chair”, was developed and evaluated. The study simulated a situation in which the subjects have entered the office from a hot day in summer in Japan. The experiments were conducted five times under identical conditions and the subjects surveyed. As the settings of the Cool Chair were varied, the responses of the subjects were recorded. Results indicated that the adaptive model was appropriate for use of the Cool Chair.

INTRODUCTION

A personal air-conditioning system differs from a standard air-conditioning system because it focuses at a point of thermal comfort. Thermal sensation differs among people and depends on clothing, physical condition, and activity level. Therefore, providing an environment that provides comfort for everyone is difficult. A personal air-conditioning system allows individual workers to adjust the thermal environment. When a person is given the right to adjust or control their immediate environment, their psychological attitude improves. In addition, the ability to adjust the conditions improves thermal comfort.

An adaptive model that requires a voluntary and passive action was proposed by R. J. de Dear et al. [1]. The adaptive model involves three adjustment elements: behavioral adjustment, physiological adjustment, and psychological adjustment. Although a psychological adjustment is the most effective, it is difficult to evaluate quantitatively. In this experiment, the psychological adjustment was examined after the subjects learn about the Cool Chair’s operating method, properties, and effects. In this paper, the cooling effect of the Cool Chair was examined, and the operating conditions during repeated subjective experiments were observed.

METHODS

Cool Chair Description

A personal air-conditioning system, called the “Cool Chair”, was developed [2]. The Cool Chair is a chair-mounted isothermal airflow generator. It contains additional functionality compared to a standard chair. It can adjust the thermal sensation of the user by changing air velocity near the surface of the body. Sitting in a standard chair with a back is equivalent to wearing another long-sleeved shirt. In contrast, the Cool Chair provides a cooling effect [3, 4]. Figure 1 shows an external view of Cool Chair with two fans on the back. The fans are battery-powered. Air is drawn into the seat and the backrest and blows out through the armrests. Maximum air flow is 70 m^3/h. The air flow rate and direction are easily adjusted. Although, the power also turned on by the user with a switch in the right armrest, it can be turned on automatically by applying weight to the seat of the chair.

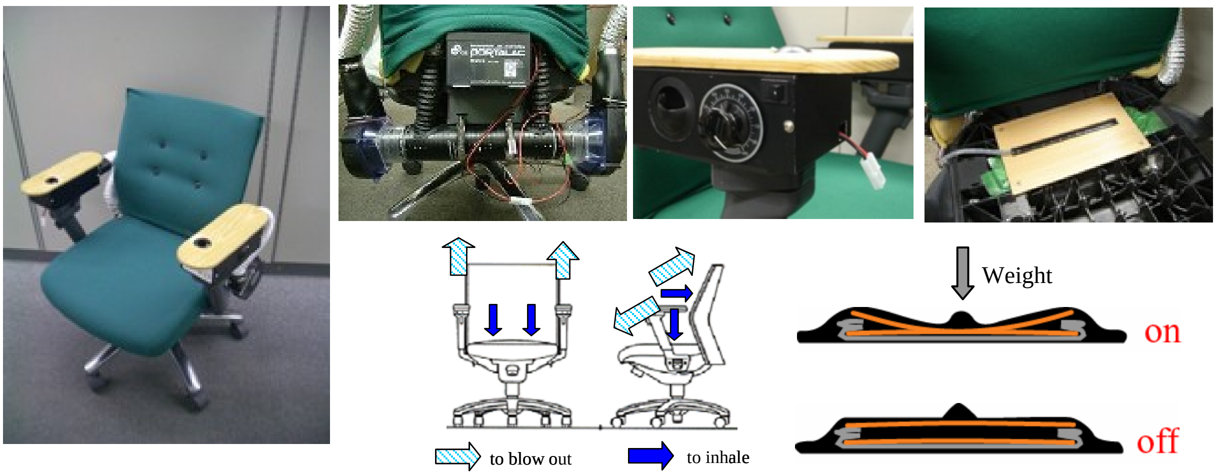


Figure 1. External view of Cool Chair.

Experimental Conditions

Subjective experiments were conducted from 11 December to 18 December in the climatic chamber at Kogakuin University in Japan [5, 6]. A schematic of the experimental room is shown in Figure 2. Eighteen male and nineteen female subjects of college age participated in the experiment. The thermal resistance of the male clothes was 0.64 clo and of the female clothes was 0.65 clo (Figure 3). The clothes of the male and female were chosen to simulate summer clothing in summer in Japan.

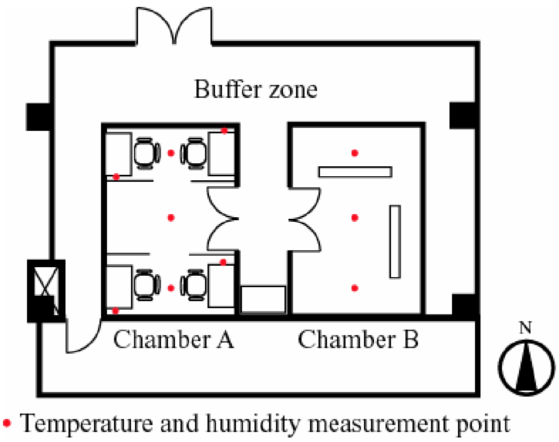


Figure 2. Plan of experimental room.

Figure 3. Subjects' clothes.

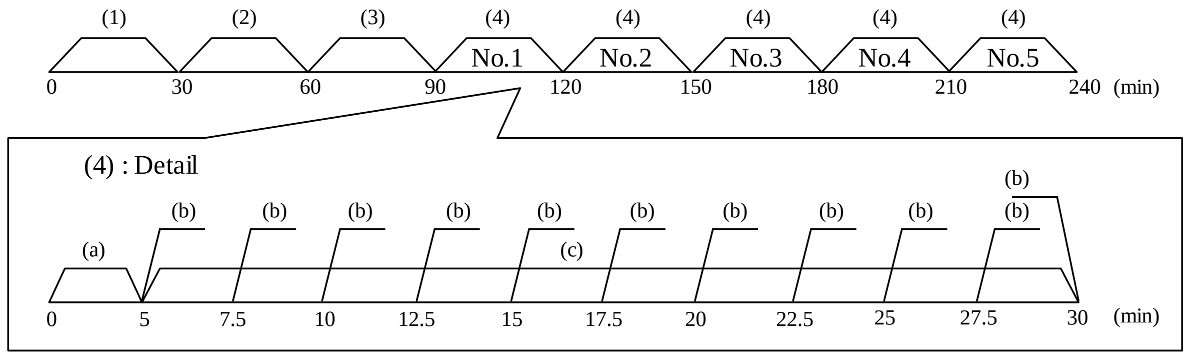
Outline of Experiments

Subjects were tested in the climatic chamber in groups of four. Experimental conditions simulated a situation in which the subjects enter the office during a hot day in summer in

Japan. Table 1 outlines the experimental conditions. The initial air temperature of chamber A was set at 28°C, 50% RH to simulate indoor conditions. All subjects were instructed to remain in the room for a certain period of time. Afterward, the subjects moved to chamber B, which had an ambient temperature of 33.4°C, 50% RH, which simulated outdoor conditions. The subjects were instructed to perform a step exercise with a metabolic value of 2.6 met. Then, the subjects moved back to chamber A and read a book. Figure 4 shows the experimental procedure. The subjects were tested under these two conditions using a standard chair and the Cool Chair. Each experiment lasted 30 min. A standard chair condition was executed once, the Cool Chair condition for the practice was once and the Cool Chair condition were fifth which are called from experiments No. 1 to No. 5. The subjects were allowed to use the chair freely during the experiment. The subjects provided information about thermal sensations (Thermal Sensation Vote, TSV) and comfort (Comfort Sensation Vote, CSV) every 2.5 min for 25 min (Table 2).

Table.1. Experimental conditions.

	Chamber A	Chamber B
Air-conditining method	Floor supply, return air flow through ceiling	Floor supply, return air flow through ceiling
Room airflow	Mild	Mild
Room air temperature	28.0 °C	33.4 °C
Mean radiant temperature	Equal to room air temperature	Equal to room air temperature
Relative humidity	50.0 %RH	50.0 %RH
Metabolic rate	1.0 met (addition test, reading)	2.6 met (step exercise)



(1) Entering into the Laboratory and Resting
(2) Using the Standard Chair
(3) Using the Cool Chair (Practice)
(4) Using the Cool Chair

(a) Step Exercise
(b) Addition Test
(c) Reading

Figure 4. Experimental procedure.

Table.2. Voting scales.

Declaration items	Voting scales
Thermal sensation Vote	-3 Cold, -2 Cool, -1 Slightly Cool, 0 Neutral, +1 Slightly Warm, +2 Warm, +3 Hot
Comfort sensation Vote	0 Comforta, -1 Slightly Uncomfortab, -2 Uncomforta, -3 Very Uncomfortab

RESULTS

Result of TSV and CSV

These mean values for males and females for TSV and CSV, while using the standard chair and the Cool Chair are shown in Figure 5 and 6. The TSV shifted toward neutral as male and female subjects compared the normal chair with the Cool Chair. CSV results indicated that subjects were more comfortable using the Cool Chair than using the standard chair. The results confirmed that TSV was improved using Cool Chair while CSV increased. TSV and CSV of experiments No. 1 and No. 5 were similar. Moreover, just after the step exercise ended (at the time of 0 minutes in the graph), TSV values were zero or less under two conditions. This experiment assumed that conditions accurately simulated an office environment. Thus, 28°C is considered a cool temperature for the subjects after moving from chamber B (33.4°C, 50% RH) to Chamber A (28°C, 50% RH).

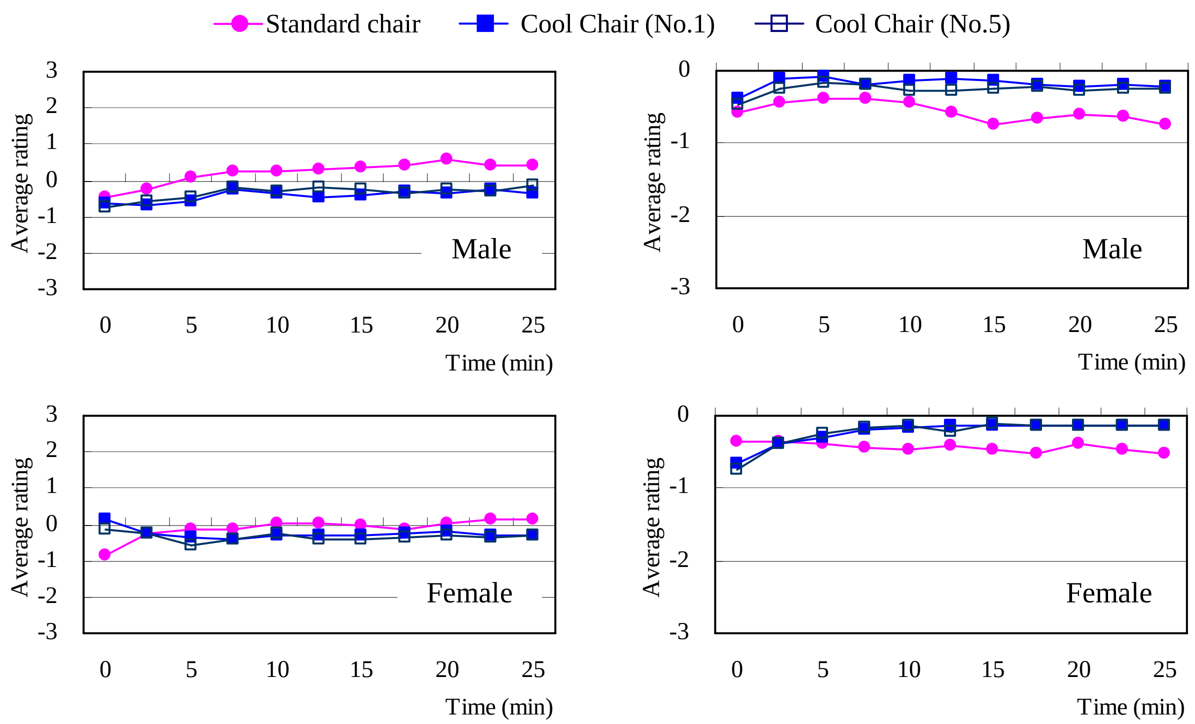


Figure 5. Thermal sensation vote.

Figure 6. Comfort sensation vote.

Air exposure

Figure 7 illustrates the parts where the subjects blowed the air flow. Both males and females preferred to blow the air toward the face and not toward the chest and stomach. The subjects also unexpectedly chose to blow the air toward the arms. Males and females reported facial dryness as shown in Figure 8. However, females adjusted the air flow rate much more often than did the males (Figure 9). Females mentioned clothes as the reason (although the clothes were similar for both males and females). Results indicated that the ability to adjust environmental conditions using the Cool Chair made the subjects feel comfortable. The negative aspect of facial dryness was offset by the comfort provided by the air flow.

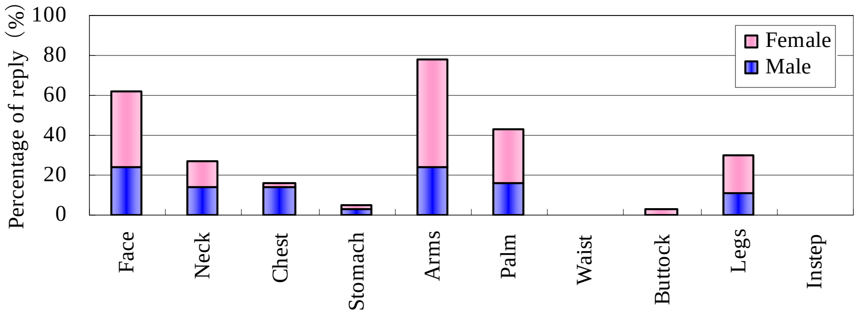


Figure 7. Air exposure parts.

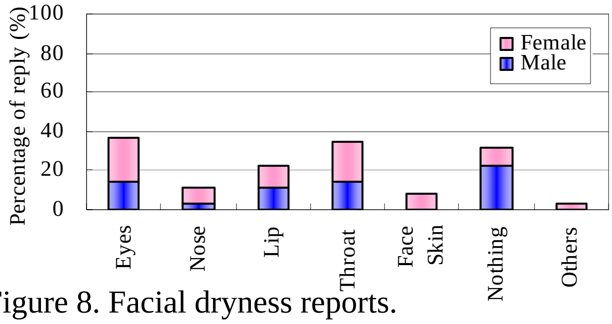


Figure 8. Facial dryness reports.

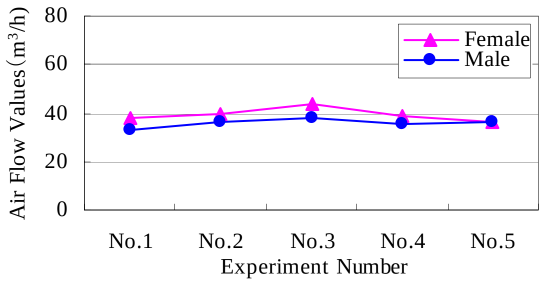


Figure 9. Air flow for each experiment.

Air flow rate

The adjustment of air flow rate was examined by calculating the interterminal voltage. The air flow rate for each subject was computed for each experiment based on values from experiment No.1. The correlation coefficient was calculated by flow rate and experiment number. A positive correlation coefficient indicated a tendency toward increasing air flow while a negative correlation indicated a tendency to decrease air flow. Figure 10 shows the distribution of every subject's correlation coefficient. They were not dependent on the gender of the subjects. Eighteen people decreased air flow from experiment No. 1 to No. 5 while f people increased it.

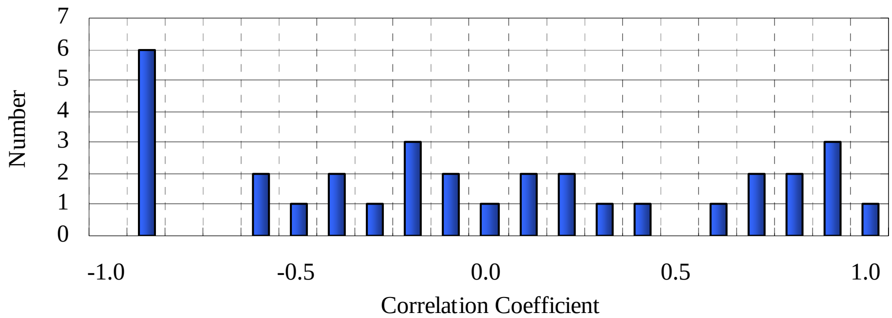


Figure 10. Distribution of correlation coefficients related to air flow rate change.

Adjustment frequency of Cool Chair

The relation between adjustment frequency of the Cool Chair and CSV was investigated. Figure 11 shows the dispersion from CSV value for each adjustment frequency. The dispersion shows as a standard deviation. The adjustment frequency of each subject was calculated. If the difference between two air flow values taken each 2.5 min was at 10 m^3/h, the subjects were considered to have adjusted the Cool Chair. A value of 10 m^3/h for air velocity corresponds to 2 m/s blown out. Both males and females showed the similar tendency. The most frequent subjects' correlation coefficient was higher than the few one.

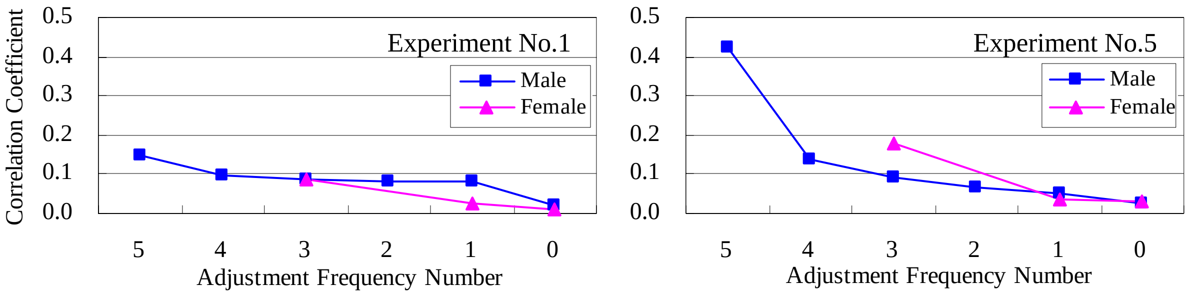


Figure 11. Relation between adjustment frequency and dispersion from CSV.

DISCUSSION

It was difficult to prove conclusively that the psychological adjustment occurred during the repeated experiments. However, subjects who were allowed to adjust the thermal environment produced the following results:

1) In a 28°C room, the TSV shifted from warm to neutral, and the CSV shifted from uncomfortable to comfortable by using the Cool Chair.
2) The subjects who felt comfortable at the first air flow value did not adjust the Cool Chair very often while the subjects who did not feel comfortable tended to adjust the Cool Chair several times to achieve the best thermal environment.
3) The subjects did not decrease air flow after achieving thermal comfort when they understood the properties of the Cool Chair in the repeated experiments.

For this experiment, psychological adjustment was assumed to occur after the subjects understood the Cool Chair's use, properties, and effects. Therefore, the air flow rate was decreased whenever the experiment was repeated. The experiment results indicate no difference in TSV and CSV between experiments No. 1 and No. 5; in addition, the tendency in adjustment frequency did not change from experiment No. 1 to No. 5. For these reasons, operating method and efficiency and subject understanding did not appear to affect results of experiments repeated only five times.

ACKNOWLEGEMENT

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Technical Solutions for Reduction of Heat Stress in Animal Houses

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SUMMARY

Today, especially in the developed industrialised nations, we have a high standard of food supply for the population. In these societies people are asking increasingly how and under which conditions their foods are produced. These questions relate above all to the complexes of animal health and animal management conditions, product quality, and influences on the environment caused by animal production. One problem in this connection is the high air temperatures together with high air humidity. There are not only in hot climates but also in Germany high temperatures during the summer period. These extreme climate conditions lead to reduction of animal performance. Technical solutions to reduce the heat stress are existing. For cattle, pigs and poultries different solutions will be presented and evaluated.

INTRODUCTION

The production of animal products serves on the one hand to supply the population with vital foodstuffs such as milk, eggs and meat. On the other hand, raw materials are also produced for various branches of industry, such as the pharmaceutical and leather industries. The prerequisites for this have been created in a development that has been going on for millennia. We left the age of "hunters and gatherers" some 15 000 years ago. Since then humans have pursued arable farming and cattle breeding and developed these steadily further. The starting point for developments in the field of animal management was the domestication of wild animals. The industrialisation in the last two centuries and the accompanying enormous further development in technical and scientific fields have also led to continuous further development in the field of animal management. Now it is not possible to say from the outset that animal welfare and environmental protection are necessarily neglected by the introduction of modern technology, new management systems, and the use of modern control and animal monitoring systems. There are certainly many positive examples in which these new techniques lead to improved animal welfare and lower environmental pollution. Despite this, the growing interest of the population in animal welfare and environmental protection is clearly evident. Therefore the questions of microclimate in animal houses, animal welfare, animal performance and economy play an important role in developing of new animal keeping systems including livestock building and ventilation systems.

Hot weather periods are one problem in this connection. High air temperatures together with high air humidity increase the heat stress for the animals. In Germany during summer there are high temperature values – 32 °C as maximum values of the day. For example during the summer 2006 we had in Germany a long period with maximum temperature values up to 40 °C outside. The high animal performance requires a reduction of heat stress. The demanded climate parameters inside animal houses are described in the German standard DIN 18 910 “Thermal insulation for closed livestock buildings”. This standard considers the thermo regulation system of the different kinds of animals. The scientific work has to develope technical solutions for a good thermal comfort especially for high yielding animals. Some examples are shown in this paper for different kinds of animals.

BACKGROUND

Influence factors on microclimate in animal houses

The main influencing factors on climate in animal houses are shown in the figure 1.

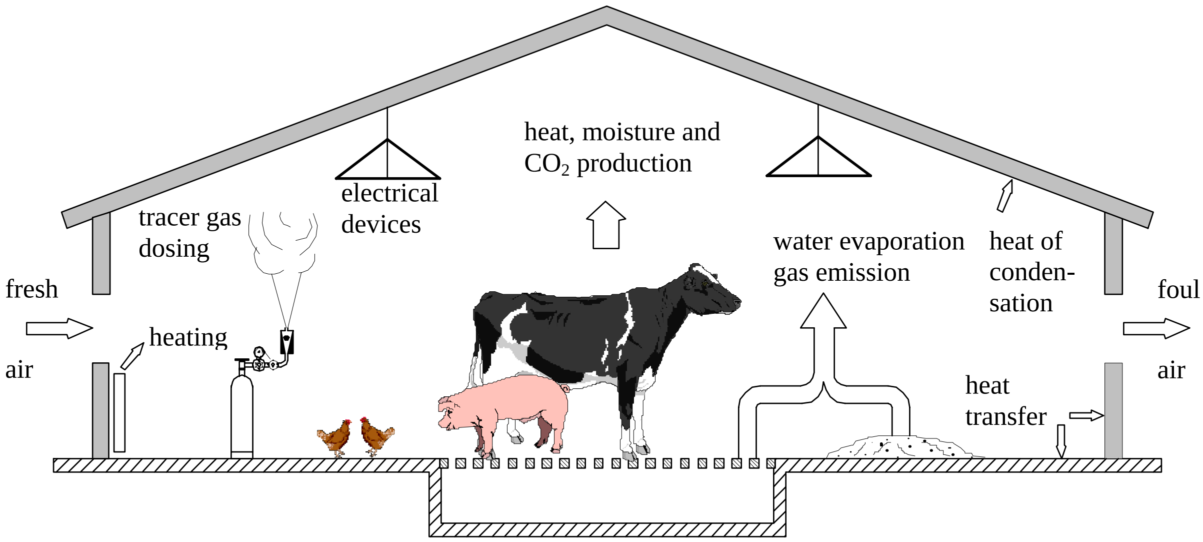

Figure 1. Schematic representation of the most important heat and substance flows for an animal house.

Additional parameters are:

Sun radiation in day time; Building orientation and shading; Outdoor temperature and outdoor humidity; Air velocity inside and outside; heat capacity of structure and floor.

More information regarding heat, moisture and CO_2 production can be found in [1]. Another factor is the diurnal variation in animal heat production. This variation results from the animal activities which are influenced by the light regime and feeding management and other management factors.

Thermal comfort index

The definition of thermal comfort for animals is much more difficult than for humans. In the literature are to find different models to evaluate the different influencing factors such as temperature, humidity and velocity, with a so-called "Thermal comfort index". A buildup of different approaches can be found in [1]. For broilers Tao and Xin [2] for example name the following equation:

$$THVI = (0.85xDBT + 0.15xWBT)xV^{-0.058} \qquad (1)$$

Where:
THVI = Temperature-Humidity-Velocity Index
DBT = Dry bulb temperature (°C)
WBT = Wet bulb temperature (°C)

V = Wind velocity

Evaluating the impact of specific climate parameters on animal performance is a difficult task combining the relationship between animal performance and thermal environment is needed. For that special investigation are necessary. The general analysis of impacts can be based on the animal response.

Possibilities for heat stress reduction

The main influencing factors on heat stress are temperature, humidity and velocity of the air and the direct solar radiation in the animal zone. If in the summer period the temperature increases during the course of day the air volume stream must be enhanced to dissipate the heat produced by the animals. Does exceed the temperature demanded critical values than the climate conditions for the animals can be improve by increasing the air velocity in the animal zone. Table 1 show, at which value the critical temperature values can be overstepped by increasing the air velocity.

Table 1. Cooling effect in K by increasing the air velocity in the animal zone [3] based on [4]

Temperature in °C	25		30		35	
Relative humidity in %	50	70	50	70	50	70
Air velocity in m/s	Cooling effect in K					
0.00	0.00	*-1.60*	0.00	*-2.20*	0.00	*-3.30*
0.50	1.10	*-0.50*	2.80	*-0.60*	2.80	*-0.50*
1.00	2.80	0.60	5.00	2.20	8.40	4.50
1.50	3.90	1.70	6.60	3.90	10.60	6.20
2.00	6.20	3.90	8.30	5.00	11.70	8.90
2.50	7.30	5.10	9.40	6.10	12.80	10.60

Depending on the room temperature the air velocity is recommended for cattle (Table 2).

Table 2. Recommended air velocity for cattle [5]

Temperature in °C	≥ 10	13	16	19	20	21	22
Air velocity in m/s	0.1	0.1	0.2	0.3	0.4	0.5	0.6
Temperature in °C	23	24	25	26	27	28	30
Air velocity in m/s	0.7	0.8	0.9	1.0	1.2	1.3	2.5

Technical possibilities to reduce the thermal load of the animals are the following:

- Reduction of solar radiation (shadow / insulation)
- Evaporative cooling systems
- Increasing of the air velocity in the animal zone
- Using heat store capacity of the ground
- Technical cooling systems

Selected technical solutions are described in the following text and the results of the investigations are shown. These investigations were carried out in cattle, pig and poultry houses.

METHODS

Especially the air temperature, the air humidity and the air velocity is measured inside the building. The air flow pattern are observed by smoke generator and recorded by video camera. The measured parameters are arranged in Table 3.

Table 3. Measured climate parameters inside the building

Parameter	Measurement technology
Temperature	Data logger (course); Portable measuring instrument (short term measuring)
Relative humidity	Data logger (course); Portable measuring instrument (short term measuring)
Gas concentration	Multi-Gas-Monitor (course measuring)
Air velocity	Portable measuring instrument (short term measuring)
Air flow pattern	Visualisation by smoke generator
Air volume stream	Portable measuring instrument (short term measuring); Measuring fan (long term measurement)
Air exchange rate	Tracergas method using tracer gases CO_2, SF_6 and Krypton 85

The air movement can be characterised by velocity measurements and observation by visualisation of the air flow pattern by smoke – these images can be recorded by video camera and analysed by Computer – Image- Analysis. For measuring the air exchange rates to calculate the emission streams the ATB has developed special tracer gas methods [6]. The advantage of using Krypton 85 (radioactive inert gas) is a high resolution in time and place, especially for complicated flow conditions.

RESULTS

Cattle houses

Nearly all cattle houses are naturally ventilated. In these livestock buildings the increasing of air velocity are used to reduce the heat stress for the animals. This increasing of air velocity can be achieved by additional fans. The different solutions are shown in Figure 2.

a) Cross ventilation: the fans are installed in one or both side walls and blow the fresh air from one side to the other side.
b) Vertical ventilation: ceiling fans blow the room air in direction of floor.
c) Step-by step ventilation: this system is a recirculation system. The air will be sucked by the fan and will be blown to the next fan. The fans are directed in this way that she are blowing slant in the animal zone. Figure 3 shows the reachable air velocities for different arrangements of the fans.
d) Tunnel ventilation: Gable fans suck the air trough the building. The air inlets are arranged in the opposite gable. The principle is used in the USA but the large cross sections of cattle barns in Germany lead to not enough high velocities in the animal zone.

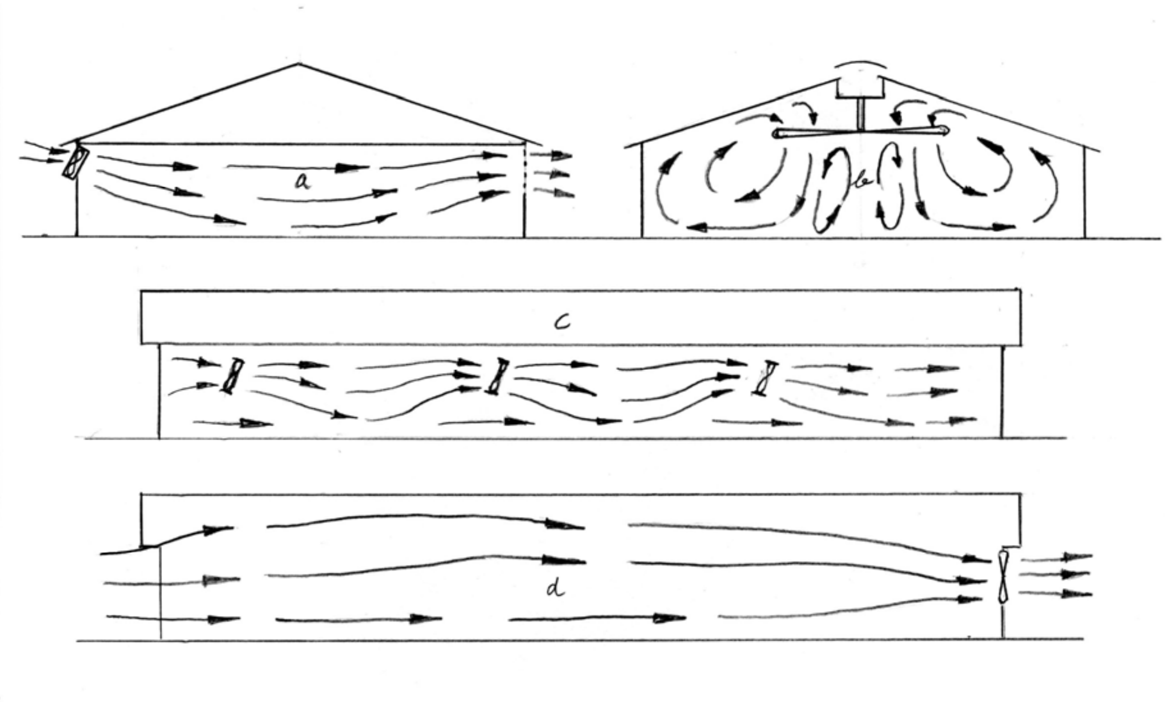

Figure 2. Different solutions for additional ventilation in cattle houses.

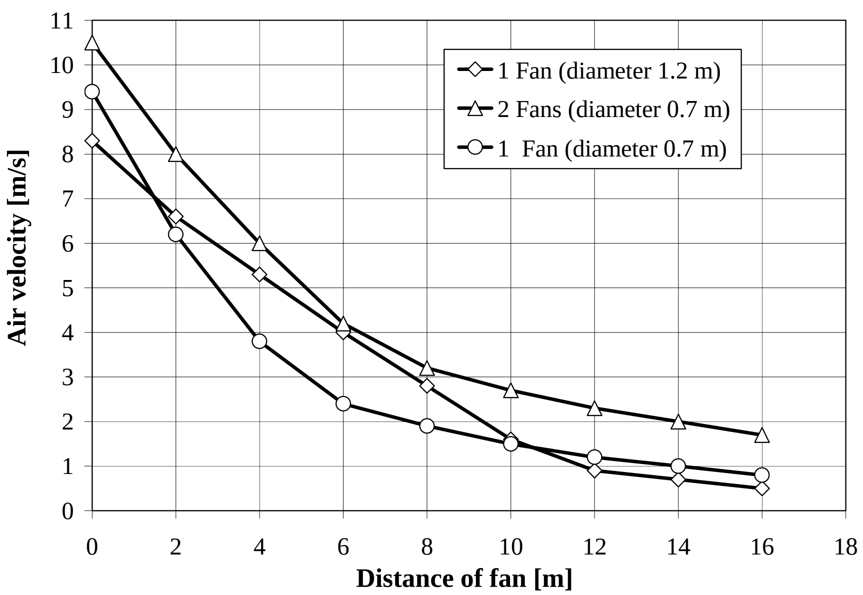

Figure 3. Air velocity on the air jet axis depending on the distance from the fan (different arrangements).

The own measurements in a naturally ventilated cow shed show, that the air velocity in the animal zone is strongly influenced by the outside wind situation. On 04 May 2006 the air velocity were measured in the animal zone with three different speed steps of the ceiling fan (speed step 1; 4.5; 9). Figure 3 shows the fan (left) and an example for speed step 4.5 (right). Directly under the fan the air comes up and the velocity is low (0.5 – 0.6 m/s). In a distance between 3 m and 6 m the air velocity increases up to 1.2 m/s. In a bigger distance the velocity decreases again and near the side wall there are the highest values caused by the outside wind near the big openings in the side wall. In the German standard DIN 18910-1 air velocity up to 0.6 m/s is permissible – but this standard concerns only for closed livestock buildings.

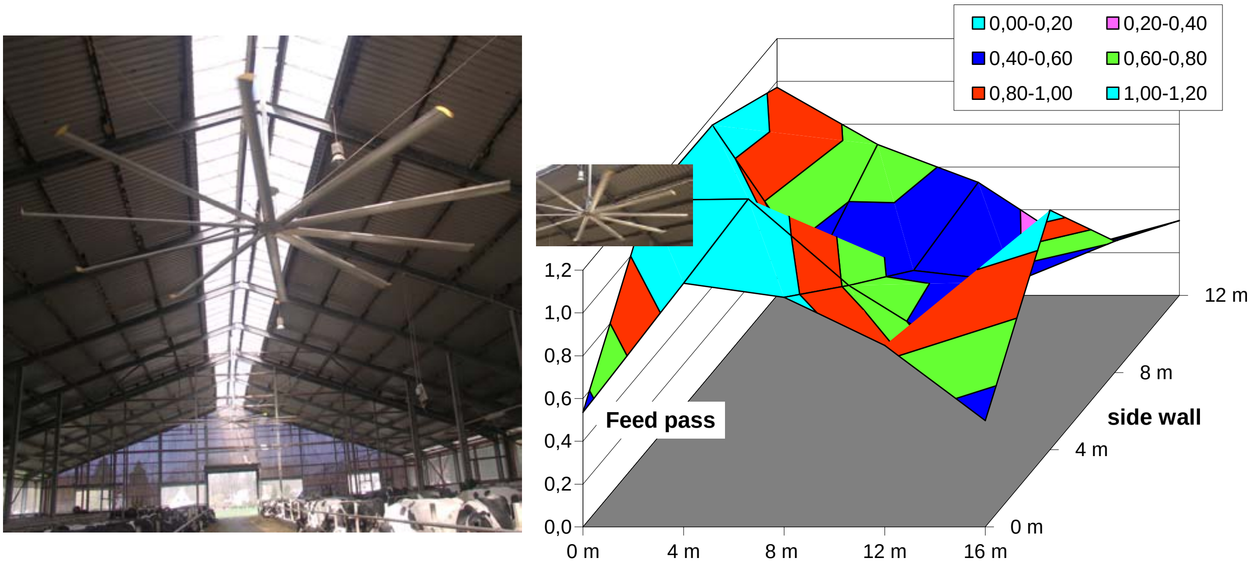

Figure 4. Ceiling fan (left) and air velocity field 1.2 m above the floor (04 May 2006). The velocity data are average values over 2 minutes

The air volume stream and concentration measurements show the strong influence from the outside wind velocity on the air volume stream trough the building and emission stream, for example for Ammonia, increases together with the out side air velocity.

Pig houses

In contrast to the cattle barns the pig houses mostly are forced ventilated. For this kind of animals cooling systems – like evaporative systems, geothermal heat exchanger using heat store capacity of the ground and technical cooling – are applied to reduce heat stress. An example is shown in Figure 5.

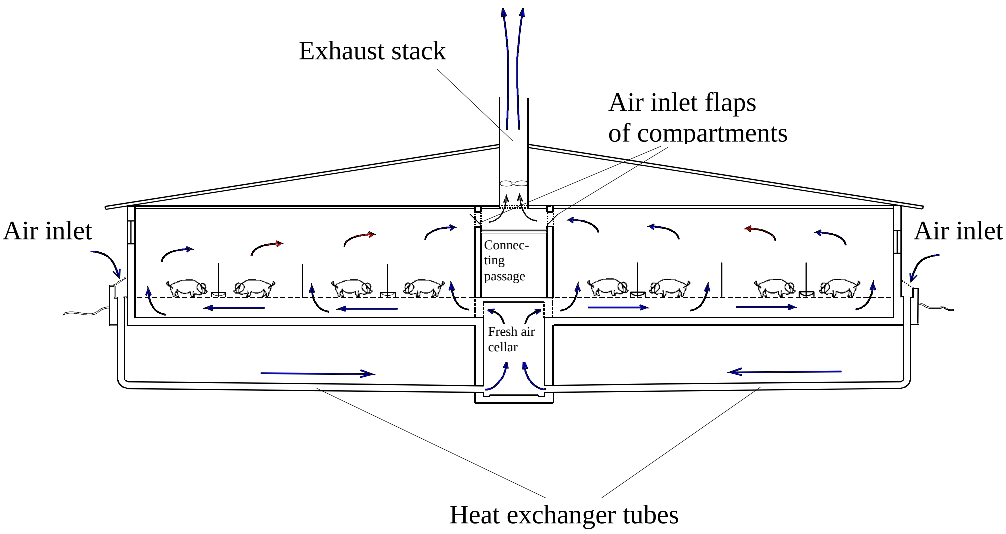

Figure 5. Cross section of a house for fattening pigs – forced ventilation connected with a geothermal heat exchanger

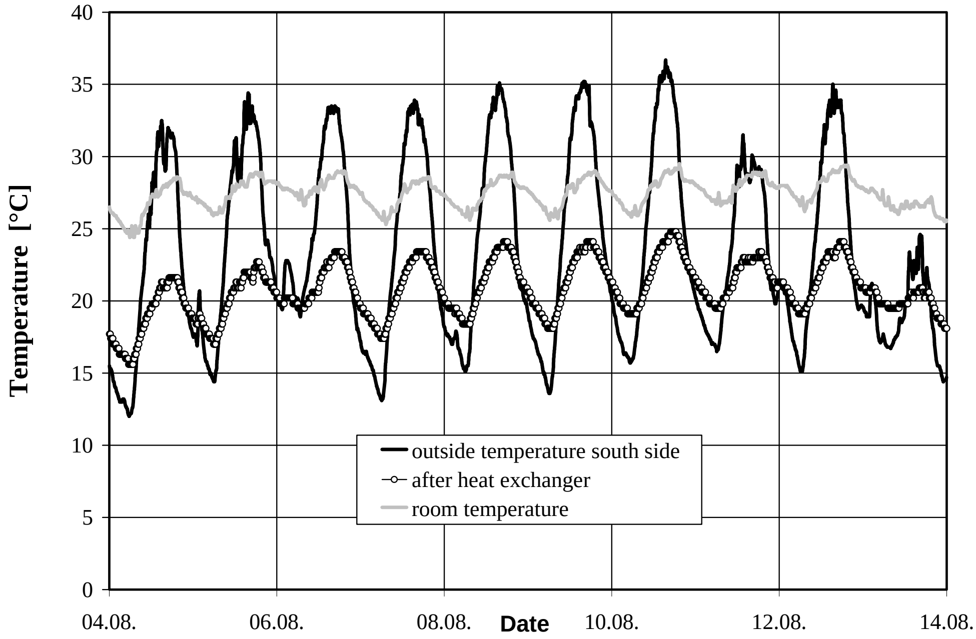

Figure 6. Course of the temperature outside, behind the heat exchanger and in the compartment in the summer.

In Figure 6 it is clear to see, that the outside temperature is damped behind the heat exchanger and the temperature peaks in the compartment is about 6 K lower than the outside temperature. This is a very good result and comfortable temperatures for the pigs during summer period. Beside the cooling effect in the summer the fresh air in the winter period will be pre-heated. Furthermore lower ammonia emission streams were measured and result from a lower air volume stream in summer. The economic examination shows that the higher costs for the heat exchanger will be balanced by the higher animal performance.

Poultry houses

In poultry houses mostly are used ceiling fans to improve the air movement in the animal zone and evaporative cooling systems. Measured velocity fields in broiler houses show a similar behavior like Figure 4. The highest values were measured under the fan and together with the distance a strong reduction of the air velocity in the animal zone was found.
An example for the effect of evaporative cooling systems is given in Figure 7. The inlet air is contaminated with mist. The exhaust air has a changing humidity and the temperature inside is lower than outside in the afternoon. As to see the cooling system in unit 13 started earlier than in unit 12, because of the sun rise. Further research has to look to the animal response to evaporative cooling system.

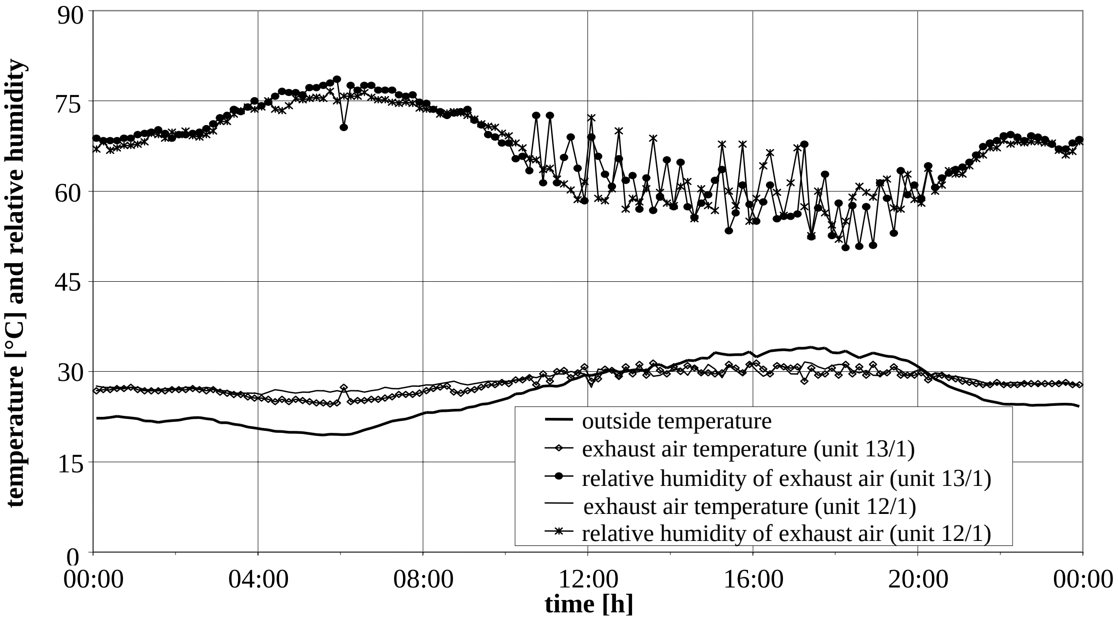

Figure 7. Effect of evaporative cooling under hot summer conditions

CONCLUSION

The animal production is an important factor to supply the population with vital foodstuffs and to deliver raw materials for various branches of industry. These products get out from livestock farming. Animal welfare and high performance require the compliance with good microclimate conditions inside the animal houses over the whole year. Especially in the summer period high temperatures lead to problems. In naturally ventilated cattle houses additional forced ventilation systems are used to improve the air movement. These systems are to advance regarding minimization of energy consumption and equalization of the velocity fields in the animal zone. In pig farming the geothermal heat exchanger stand the test of time. In connection with biogas production the application of technical cooling systems in pig production is under development in Germany. The evaporative cooling systems are successfully applied in poultry production.

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A12
Sources of indoor air pollution

Indoor Pollution from Building Materials and Ventilation Rates

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SUMMARY

Emissions and odors from different common indoor surface materials (waste wooden products, vinyl, linoleum, carpet, gypsum board, paint, wall coating materials etc.) were investigated within this study. The measurements were conducted in a test chamber under standardized conditions (23°C, 50%) and different ventilation rates. Chemical measurements (TVOC) and sensory assessments (odor intensity, perceived air quality) were done after the 3rd day of building material exposure to standardized conditions. The increased ventilation rate does not evoke adequate response in subjective sensory assessment as well in decreasing of odor indicator level. The source control appears as more effective then ineffective increasing of air change level. The impact of ventilation rate on perceived indoor air quality pollution from selected building materials will be discussed within the paper.

INTRODUCTION

In recent years, the range of new building material, products and construction systems significantly increased. The traditional way of material selection for building design has been primarily based on factors such as cost, aesthetic values, availability and durability. The environmental impact, both on the environment and occupants, has not been addressed till very recently. Selection of building and interior materials can positively or negatively long term affects user's health and comfort. Practically it is not possible to specify environmental friendly and unsafe building materials considering that each material in general view has certain advantages as well disadvantages. Many of the materials used in buildings, either as structural materials or as furnishings, are the main sources of indoor air pollution. From variety of building materials the significant emissions of volatile organic compounds (VOCs) were observed. Some of the emitted volatile organic compounds (VOCs) may except of serious health problems affect the perception of the indoor air quality (odor nuisance, eye and airway irritation).

The most important interactive factor of influence is air exchange. While ventilation is not the only determinant of IAQ, perceived air quality and health outcomes generally improve as ventilation rates increase. The review study [1] has shown that existing date do not indicate whether outside air supply per person or per unit floor area is more strongly associated with health and perceived IAQ. Finally, the reasons for improved health and perceived air quality with increased ventilation are uncertain. The impact of increased ventilation on the perceived air quality may vary from one building material to another [2]. The relationship between ventilation rate and the perceived air quality were summarized within the study [3]. The relationships are different for different building materials. The ventilation requirements to obtain a certain level of perceived air quality for emissions from different building materials are relatively large.

The improvements of perceived air quality after increasing the ventilation rate may be less than expected from simple dilution models, because more ventilation in a given space may increase the emissions from the inner surfaces [4].

METHODS

Coverings of surfaces (floors, walls ad ceilings) are considered to have an especially important role for the quality of indoor air. Chemical analysis and subjective sensory assessments of emissions from selected building finishing materials after their exposition to different air change conditions were carried out in a test chamber. The surface building materials used in this study were selected to present major groups of building products usually used indoors in Slovakia. The investigated building products are five floor coverings (PVC flooring, Linoleum, High Density Fiberboard laminated flooring, Oriented strand boards - OSB and PA carpet) and two walls and/or ceiling coverings (gypsum boards, wallpaper).

The air change levels were specified with respect to European legislation (prEN 15251:2005) for 3 different indoor air quality categories. Category A corresponds to a high level of expectation and leads to a highest percentage of satisfied occupants in respect of indoor environment (n = 1,2 1/h). Category B corresponds to a medium level of expectation (n = 2,0 1/h) and category C to a moderate level of expectation (n = 3,0 1/h). The air change levels were specified regarding that tested materials are non low polluting materials [5].

The principle of the test method is to assess the emission of pollutants from a test specimen, prepared from a sample of a tested material, by sensory assessments and VOCs concentration measurements of the air in a test chamber. The surface of the test specimen is exposed to the chamber air which is maintained at a temperature, humidity and velocity similar to that which can be expected in the indoor environment in which the material is usually used. In addition to these conditions the chamber VOCs concentration depends on the supply airflow rate in the chamber and the area of the test specimen. The test is performed with an area specific airflow rate similar to that which can be expected during the normal use of the material.

The samples of building materials were placed in test chamber made of glass (190 l). Before the tests, the chamber was cleaned. The background concentration of pollutants and sensory assessment of the chamber without the sample under the same conditions were estimated. The mean air velocity and air flow rate were measured using the thermal anemometer TESTO 425. The measurements were taken after 3 days of building material exposure to the standardized conditions (23°C, 50%) within the test chamber. The air velocity over the building product samples was adjusted to 0,1 – 0,2 m/s which corresponds to a usual range indoors. The size of the test specimens were determined so that the area specific airflow rate in the test chamber was corresponding to the area specific airflow rate in standard model room (3,2 x 2,2 x 2,4m). The specimens were placed vertically, so that the emission surfaces were parallel to the direction of the air flow.

In order to simplify the indoor volatile organic compound loading the TVOC value was estimated as the indicator of building material odors. Mølhave, the author of TVOC index proposed comfort level value 200 $\mu g/m^3$. The chemical analysis was based on detection of selected volatile organic compounds. Active sampling of VOCs was performed by using a pump (Aircheck 2000) with air flow rate of 400 ml/min on charcoal tubes ORBO 32S (Supelco) during 24 hours. The absorbed VOCs were analyzed by gas chromatography (GC Varian 3 300) after extraction into CS_2. The sum of VOCs was calculated based on individual

VOCs concentrations including unidentified peak areas converted to decane equivalents (equivalent to toluene).The exhaust air from each test chamber was led through a diffuser for sensory assessments.

An untrained sensory panel of 20 subjects assessed odor intensity and perceived air quality. Before the first assessment the panelists were instructed how to use the scale and the exposure equipment. The responsible person of the experiment assessed each subject's attitude and motivation concerning to experiment and subject's personal hygiene. There was no restriction on distribution of gender or smoking habits. The age ranged from 18 to 40 years with mean a 35 years, and 14 % of the subjects were smokers. The panel stayed in a well ventilated room without odors before the assessments. Then the subjects indicated their immediate evaluation on two continuous scales regarding odor intensity (from 0 no odor to 5 overwhelming odor) and acceptability of the air (from -1 clearly unacceptable to +1 clearly acceptable) from which the percentage of dissatisfied was estimated. The standard test method was used for calculation of acceptability and estimation of odor intensity. During the measurements, the test chambers were covered with aluminum sheets to hide the building products from the view of the sensory panelists.

RESULTS

The chemical analysis of emissions has shown that ventilation rates in non-industrial buildings should be based largely on sensory pollution sources and a desired level of perceived air quality. The air change level of category C guaranteed olfactory level of TVOC (Molhave 200 μg/m^3) however the subjects found the air as still unacceptable in almost all cases of tested interior materials (Fig. 1). Building products have affected the perceived air quality, even when the concentrations of almost all primary VOCs were well below to their odor detection thresholds.

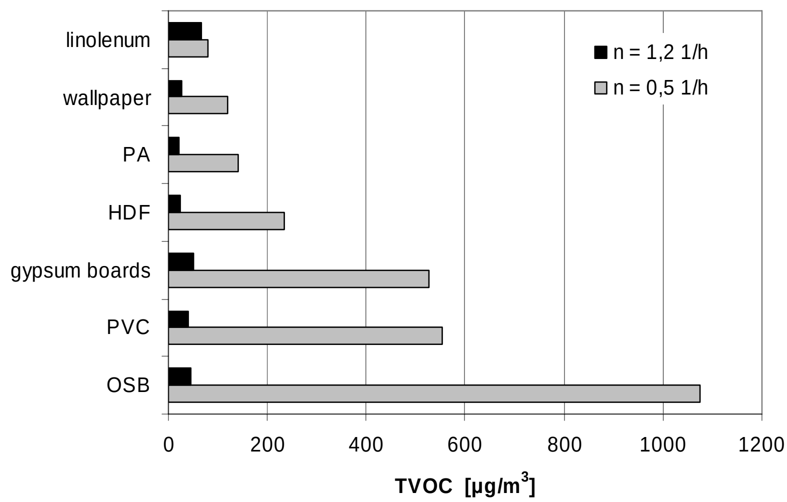

Fig. 1 The results of chemical analysis

Flooring finishing materials

The results of sensory assessments using the odor intensity scale in relation to air change levels are obvious from Fig. 2. The effect of air change level to percentage of dissatisfied is shown in Fig. 3. In case of PVC, HDF and PA finishing materials the significant decreasing in percentage of dissatisfied was reached by increasing the air change level from 0,5 1/h to air change level of category C (1,2 1/h). The decreasing presented approximately 50 %. Additional air change level increasing did not lead to important decreasing of percentage of dissatisfied (at average 15 %). Similar tendency were found for another indicator of perceived air quality – odor intensity. The highest decreasing in odor intensity was found for lower observed air change levels. Increasing of air change level from 0,5 1/h to 1,2 1/h results to decreasing within approximately 1 odor intensity grade. Additional air change level further decreased the odor intensity average by half a grade.

In case of OSB flooring system and linoleum the effect of air change levels to perceived air quality indicators has not been explicit. For OSB flooring system, substantial decreasing of both perceived air quality indicators was reached by increasing of air change level from 0,5 1/h to 1,2 1/h (category C) and from air change level from 2,0 1/h (category B) to 3,0 1/h (category A). The percentage of dissatisfied decreased approximately by 30 % and odor intensity decreased within almost 1 odor intensity grade. The air change level between the mentioned ranges leads to minimal decreasing in PN and odor intensity. For linoleum the increasing of air change level leads to almost linear degreasing of odor intensity. However the highest decreasing of percentage of dissatisfied was reached when air change level was increased from 2,0 1/h (category B) to 3,0 1/h (category A).

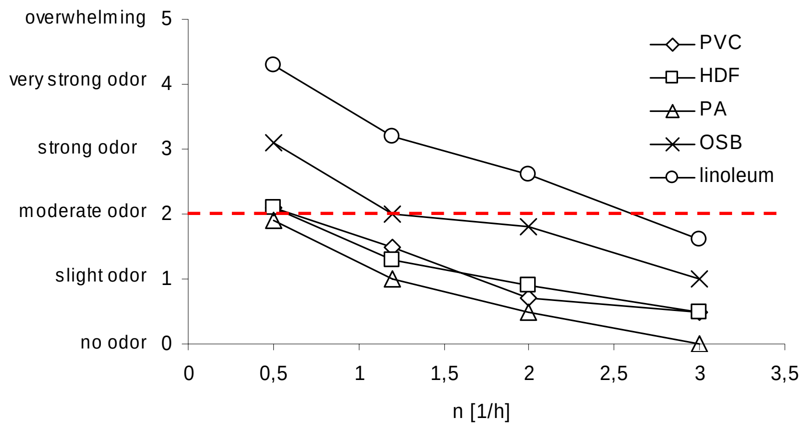

Fig. 2 Odor intensity and air change level

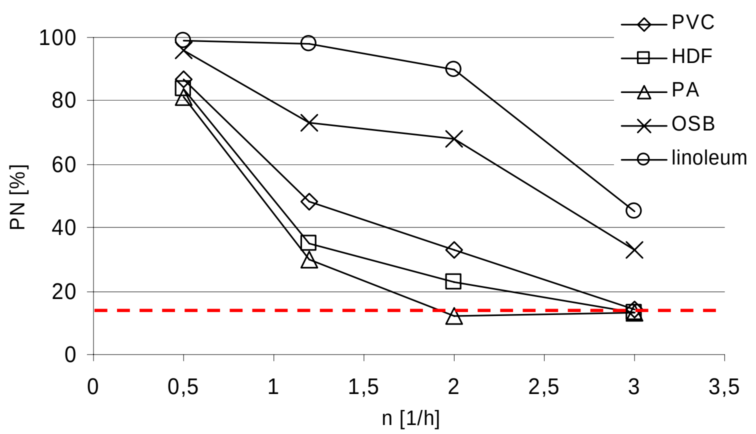

Fig. 3 PN and air change level

The indoor odor intensity is regarded as acceptable at value less than 2 - moderate odor. In case of PA carpeting the recommended value was reached by air change level n = 0,5 1/h. The air change level for category C seems to be satisfactory for indoor odor intensity in case of HDF laminated and PVC flooring. Acceptable odor intensity for OSB flooring was reached at air change level for category B (n = 2,0 1/h). The percentage of dissatisfied was acceptable for PA carpet with air change rates from 2,0 1/h (category B). The air change level for category A (3,0 1/h) must reach the level of 15 % for PVC and HDF laminated flooring. In the case of OSB flooring this level has not been reached even when the air change was on the highest indoor air quality level.

It is interesting that subject found linoleum as the most unacceptable materials from subjective sensory assessment point of view. However the TVOC values were the lowest from investigated indoor surface materials. Linoleum did not reached required percentage of dissatisfied even when the air change was on the level of highest indoor air quality category. The moderate odor intensity was achieved by air change level n = 3,0 1/h.

Walls, ceilings finishing materials

The results of sensory assessments using the odor intensity scale in relation to air change levels are obvious from Fig. 4. The effect of air change level to percentage of dissatisfied is shown in Fig. 5.

In case of wallpaper the effect of air change level to perceived air quality indicators has not been evident. Effective decreasing of PN was reached by increasing of air change level from 1,2 1/h (category C) to 2,0 1/h (category B). The percentage of dissatisfied decreased approximately by 52 % and odor intensity decreased within one and half of grade on the odor intensity scale. The air change level between ranges from 0,5 1/h to 1,2 1/h and 2,0 1/h to 3,0 1/h evocated minimal decreasing in PN as well in odor intensity.

Accordingly, in case of painted gypsum the effect of air change level to perceived air quality indicators has not been obvious; however with converse tendency. The evident decreasing of both perceived air quality indicators was reached by increasing of air change level from 0,5 1/h to 1,2 1/h (category C) and from air change level from 2,0 1/h (category B) to 3,0 1/h (category A), in same manner as in case of OSB floor system. The percentage of dissatisfied decreased approximately by 40 % and odor intensity decreased within half of the level of odor intensity scale. The air change level between mentioned ranges resulted to minimal decreasing in PN as well as in odor intensity.

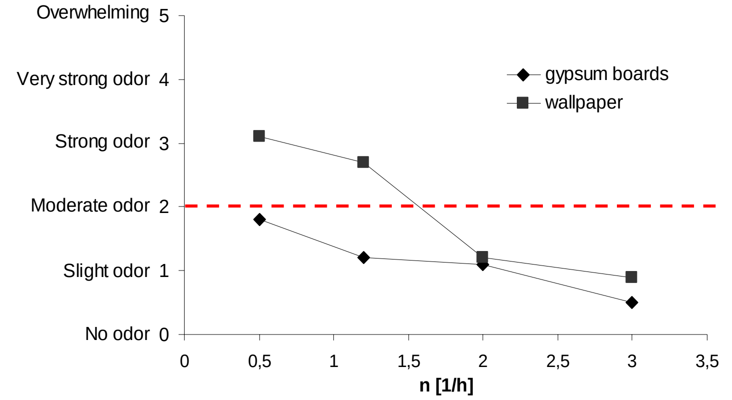

Fig. 4 Odor intensity and air change level

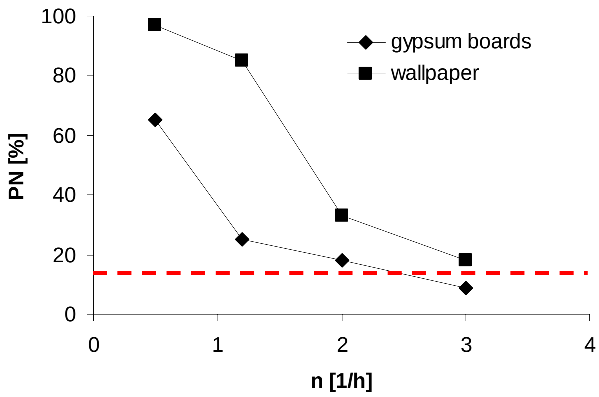

Fig. 5 PN and air change level

The indoor odor intensity is regarded as acceptable at a value of less than 2 - moderate odor. In the case of painted gypsum boards this recommended value was reached by air change level 0,5 1/h. The air change level for category B seems to be satisfactory for wallpaper from the indoor odor intensity point of view.

The air change level for category A (3,0 1/h) was necessary to reach the level of 15 % dissatisfied in case of painted gypsum boards. In case of wallpaper this level has not been reached even when the air change was on the level of highest indoor air quality category.

DISCUSSION

The presented study has shown that even when the TVOC concentrations were well below the olfactory comfort level the subjects found the air as still unacceptable in almost all cases of tested interior materials. Even when the VOCs concentrations are well below their odor detection thresholds the perceived air quality is affected. On the base of the results it is possible to assume that human nose is much more sensitive than chemical analysis.

Perceived air quality was improved and odor intensity was decreased when the air change level was increasing. However, the increased air change rate does not evoke adequate response in subjective sensory assessment. The significant decreasing of perceived air quality indicators with higher air change level was found in the cases of PVC, HDF and PA. In the cases of OSB flooring systems, linoleum, gypsum boards and wallpaper the effect of increased air change level has not been obvious.

Respecting other indoor climate parameters requirements (air temperature, humidity, and air movement) the increasing of air change level in order to achieve acceptable perceived air quality is not reasonable and makes no real sense It is very difficult to discharge air into a space at a high flow rate without creating draughts, increased energy consumption and other environmental problems. Consequently, the source control appears as more effective then ineffective increasing of air change level.

ACKNOWLEDGEMENT

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Variables affecting indoor air quality in newly finished buildings- a multivariate evaluation

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SUMMARY

This paper presents results from a principal component analysis (PCA) on variables affecting on indoor air quality in newly established buildings, in which low-emitting, classified building materials were used. The concentrations of TVOC, VOCs, formaldehyde and ammonia) were determined for the newly finished and 6-, and 12-month-old buildings. Temperature, relative humidity and air exchange rates were determined simultaneously. These values were included in the PCA models, which were used to reveal which variables affected indoor air quality (IAQ). The variables included floor covering, air exchange system and the time of its operation (in the newly finished building), floor structure / leveling agent, walls, time of construction and season and time point from last construction work.
The IAQ was by most affected by the air exchange system and ceiling product. Lowest concentration levels and classification of VOCs occurred for the buildings with mechanical supply and exhaust air system. IAQ was highly affected by indoor air humidity, especially in the 6 and 12-month- old building

INTRODUCTION

Indoor air quality (IAQ) is affected by both indoor and outdoor sources of pollutants. Indoors, building materials are important emission sources of pollutants, especially in new buildings [1]. Occupancy (activities of the inhabitants, furniture etc.) and chemical reactions may further increase concentration levels of harmful pollutants, which in some cases can lead sick building syndromes (SBS). Besides source control, ventilation is one effective way to control pollutant levels indoors.
The evaluation of the contribution from different sources and the effect of outer conditions (ventilation, temperature, humidity) on IAQ is difficult due to the complexity of indoor environments. This study investigates the potential of multivariate methods for the evaluation of variables affecting IAQ in newly finished buildings.

METHODS

Study buildings and measurement scheme

Fourteen apartments in eight residential buildings (seven apartment buildings and one two-family house), built according to the current Finnish Building Regulation Code, were investigated. Seven of the buildings were located in Helsinki and one in Turku. Low emitting, M1-classified materials were used in all the buildings. That is, the laboratory tests performed for 4 week- old samples have given TVOC-, ammonia- and formaldehyde emissions lower than 200, 30, and 50 µg/ m^2h, respectively [2]. The relative humidity (RH) of the structure was determined to be <85% before the floor covering was installed. In seven buildings, the walls were finished with screed and painted. Wallpaper was laid on the screed in the building 4. Ceilings were finished with screed (2 different products). The floors were finished with fine screed (dispersal 2-5 mm) in the site built buildings and with gross screed (dispersal 10-30 mm) in the manufactured buildings. Different types of PVC materials and parquets were used as floor covering materials. The installation of the floor covering material was the last construction work occurring 2-28 weeks prior to occupation, except in building 4, where painting was performed 12 weeks prior to occupation.

Indoor air measurements were performed first in the newly finished building, when the ventilation was operating, but before the occupants had moved in. The measurements were repeated after 6 and 12 months. In one apartment (Building 7, apartment 2), a water leak took place a few weeks before the apartment was finished. The apartment was heated and ventilated for about two weeks after which the indoor air was measured. These VOC results were not included in the statistical analysis.

Sampling and analysis

Indoor air samples were taken in a closed room (usually the bedroom) at the height of approximately 1.40 m before noon. The follow-up measurements were done every time in the same room and in the same place. The air exchange rates were determined simultaneously. The inhabitants were asked to avoid cleaning, smoking, and the use of fragrances in the morning prior to the measurements. Additional ventilation through doors or windows 24 h before the measurement was discouraged. No smoking or pets were observed during the measurements.
Air samples of VOCs were collected on Tenax TA adsorbent [3]. The sampling of ammonia and formaldehyde was performed into a 0.005 M sulphuric acid- solution at 2-4 l/min. The temperature and RH were registered using a Vaisala HMP41 moisture detector. The air exchange rate was determined simultaneously based on ventilation duct measurements with an Alnor AXD-530 thermo anemometer.
Tenax tubes were thermally desorbed and analysed with a HP 5890 series 2 gas chromatograph connected to a HP 5972 mass spectrometer and FID detector. MSD in SCAN mode was used to identify single VOCs and the FID response was used for quantification. TVOC was calculated as toluene equivalents from the total integrated FID signal between hexane and hexadecane. The ammonia concentration was determined with an ion-selective electrode and the analysis of formaldehyde was done with the spectrometric acetyl-acetone method.

Data handling

Statistical tests were performed with Simca-P 7.0 for Windows software. Principal component analysis (PCA) was performed to reveal which variables affected IAQ most.

The values for TVOC, formaldehyde, ammonia and single VOC concentration as well as the temperature, humidity, and air exchange rate were included in the models. Due to water damage in building 7, apartment 2, these results were excluded from the analysis. Models were calculated for the 0-, 6- and 12- month-old buildings. Automatic UV scaling was performed by the Simca software. Table 1 summarizes details for the models for the 0-, 6- and 12-month-old buildings. The variables investigated were as follows:

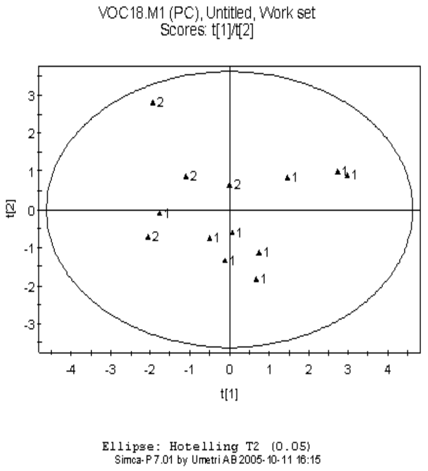
- Structure (on site built+ fine screed/ manufactured concrete cast slab+ gross screed)
- Floor covering (parquet/PVC)
- Wall material (paint/ wall paper)
- Ceiling material (screed, different manufacturer)
- Air exchange system (mechanical exhaust/ mechanical supply and exhaust air system)
- ACH system operation time (0.5-8 weeks, newly finished building)
- Time of construction (7-10 months, from the time point)
- Season (month finished, indoor air humidity)
- Time point from last construction work prior to use (4-28 weeks)

Table 1. PCA models details for the 0-, 6- and 12-month-old buildings.

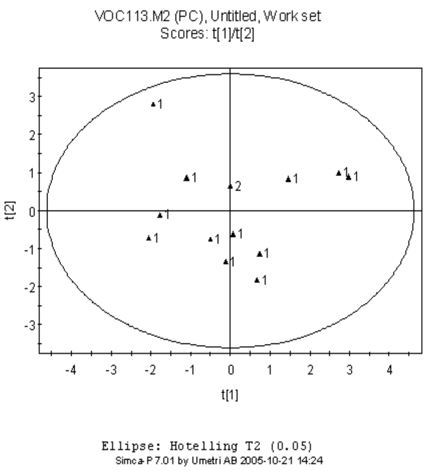
Model	**0 month-old building**	**6 month-old building**	**12 month-old building**
TVOC, ammonia, formaldehyde	13 measurements (apartments), R^2: 0.695 (2 components)	14 measurements (apartments) R^2: 0.812 (2 components)	14 measurements (apartments) R^2: 0.799 (2 components)
VOCs	VOCs occuring at n≥3 sites (76/ 238 compounds) R^2: 0.631 (3 components)	VOCs occuring at n≥4 sites (56/ 167 compounds) R^2: 0.576 (3 components)	VOCs occuring at n≥4 sites (61/189 compounds) R^2: 0.576 (3 components)

RESULTS

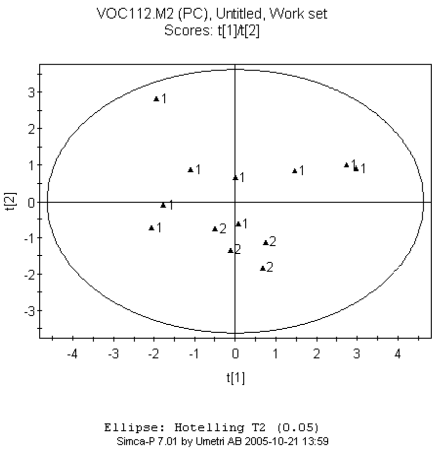
Selected examples of PCA results are presented in figures 1 and 2. Lower TVOC concentration was observed in the newly finished buildings with a mechanical supply and exhaust air system (Figure 1a and b). The same leveling agent (product 2) was also used on the ceiling structure in these buildings and the RH was low, less than 40%. The floor covering was parquet and walls were painted. The lowest TVOC, ammonia, and formaldehyde concentrations in the 6-month-old buildings were measured in the building where product 2 was used as ceiling surface product and when the RH was less than 30%. The clustering of VOCs also occurred at these apartments. The same was observed in the 12-month-old buildings, where the clustering of VOCs according to air exchange system (mechanical supply), ceiling surface (product 2), and indoor air humidity (<40%) was quite clear (Figure 2). Lowest pollutant levels/ VOC classification in the 0-, 6- and 12-month-old buildings are summarized in Table 2.



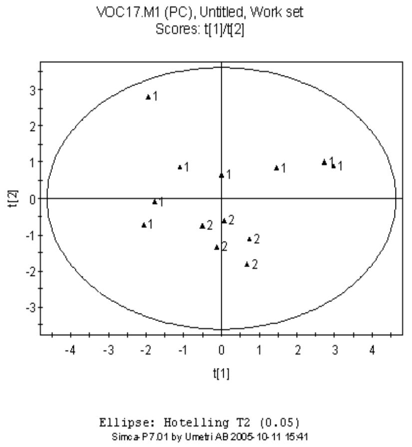
a) floor covering material: parquet =1 PVC=2



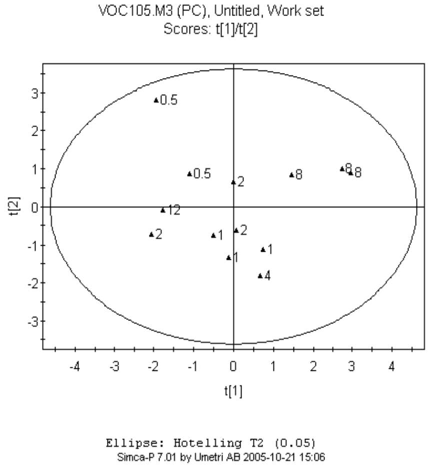
b) walls: paint=1, wall paper=2



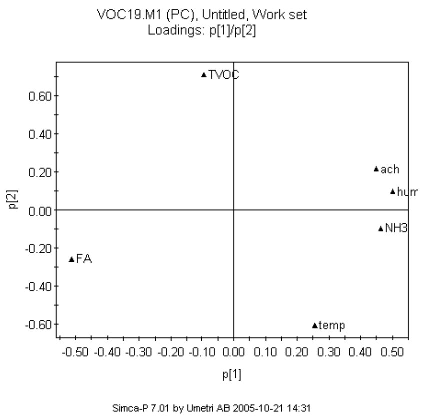
c) ceiling: product 1=1, product 2=2



d) air exchange system: mechanical exhaust=1, mechanical supply&exhaust=2

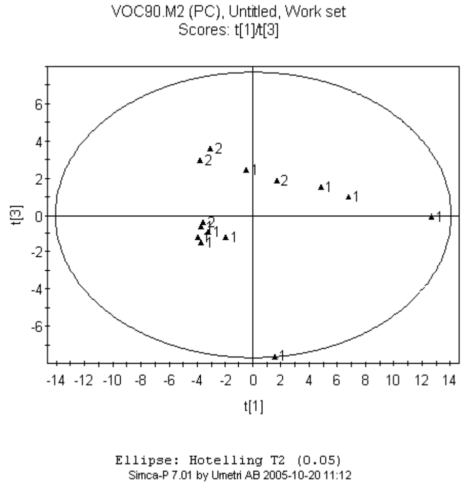


e) air exchange operation time 0.5-8 weeks

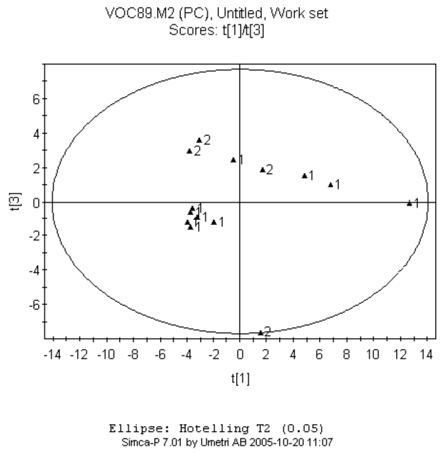


f) loadings plot

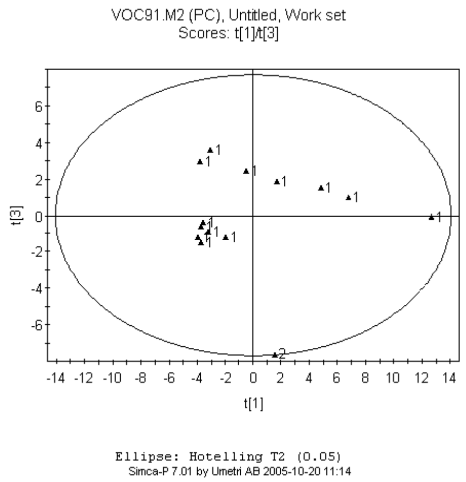
Figure 1 a-f : Example of PCA results: IAQ in the newly finished building. The score plots (a-e) shows how the 13 measurement sites and the model contribution of the measured parameters is shown in the corresponding loadings plot (f). The contribution of the measured parameter decreases as the distance of the parameter to the observation increases (TVOC=total volatile organic compounds, FA= formaldehyde, NH3= ammonia, temp=temperature, hum= relative humidity, ach= air exchange rate).



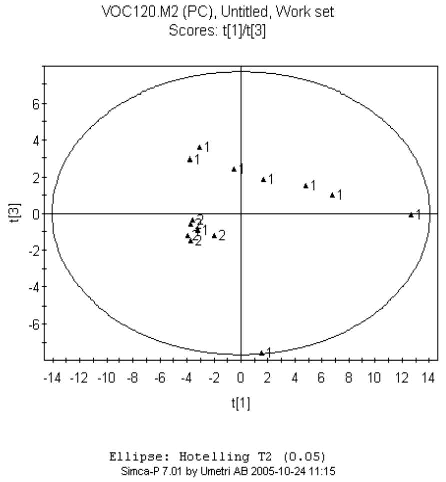
a) structure: on site built=1, manufactured=2



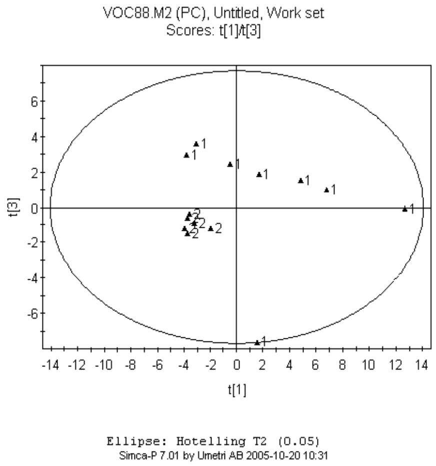
b) floor covering material: parquet=1, PVC=2



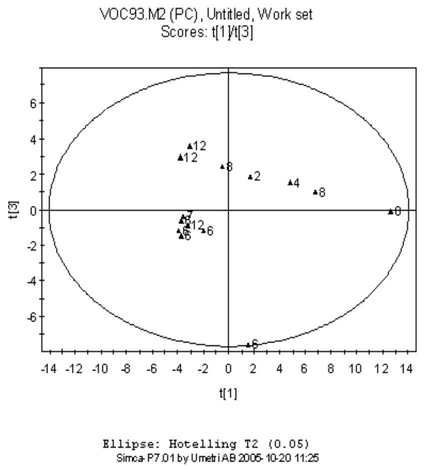
c) walls: paint=1, wall paper=2



d) ceiling: product 1=1, product 2=2



e) air exchange system: mechanical exhaust=1, mechanical supply&exhaust=2



f) season: month number

Figure 2. Classification of VOCs in the 12-month-old-buildings.

Table 2. Summary of PCA results in the 0-, 6- and 12-month-old buildings.

0-month-old building	**6-month-old building**	**12-month-old building**
Lowest TVOC concentration: air exchange system: mechanical supply and exhaust air system ceiling surface: product 2 walls: painted indoor air humidity: <40% VOC classification (clustering): Air exchange system: mechanical supply ceiling surface: product 2	Lowest TVOC, ammonia and formaldehyde concentration: ceiling surface: product 2 indoor air humidity: <30%! VOC classification (clustering): ceiling surface: product 2 indoor air humidity: <40%	Lowest TVOC, ammonia and formaldehyde concentration: air exchange system: mechanical supply air system ceiling surface: product 2 walls: painted indoor air humidity <40% VOC classification (clustering): air exchange system: mechanical supply air system ceiling surface: product 2 walls: painted indoor air humidity: <40%

DISCUSSION

The best IAQ was achieved in the 0-12-month-old buildings with mechanical supply and exhaust air system, screed product nr 2 as ceiling surface, and painted walls. These buildings had parquet as floor covering material. IAQ was highly affected by indoor air humidity, especially in the 6 and 12-month- old building. As the summary of (10) PCA, variables affecting the concentrations of indoor air gaseous pollutants in the buildings were season, relative humidity and temperature of indoor air, air exchange system, floor covering material, ceiling surface product, wall surface product, and occupancy. The multivariate analysis proved to be a very useful tool in revealing the most significant variables affecting IAQ.

ACKNOWLEDGEMENT

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Measurements of VOCs emission rate from building materials during bake-out with passive sampling methods

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SUMMARY

Bake-out of buildings is believed to have a potential to reduce indoor air pollution caused by VOCs and formaldehyde emitted from building materials although controversial discussions have been suggested. To clarify the effectiveness of bake-out, in this study, the variation of VOCs and formaldehyde emission rate from building material were investigated in residential housing units with passive sampling methods. For about a month, measurements of emission rate are carried out on various building materials such as wood based materials and paper based materials installed in real buildings at which bake-out was conducted. According to the results, the toluene emission rate from wood based materials clearly decreased with only ventilated conditions during bake-out. However, the toluene emission rate from wall paper decreased regardless to the ventilation condition. Compared to toluene emission rate, we couldn't observe a clear reduction of formaldehyde emission rate from most of building materials.

INTRODUCTION

Recently, due to synthetic building materials used in buildings, IAQ problem has become a major public concern in newly constructed apartment buildings. For resolving this problem, Indoor Air Quality Management Act[1], a regulation concerning the concentration limitations of formaldehyde and 5 VOCs in new apartment buildings, was enforced by the Ministry of Environment of Korea and came into effect from May 30, 2004. According to this act, construction companies, which construct apartment complexes, are required to measure the concentration levels of 6 toxic substances and give a public notification before the residents move in. To meet the requirements of the government regulations, source control strategies such as using low-emission building materials are widely used. However, Indoor concentration level of chemical compounds is partly high especially at pre-occupancy stage, due to high outdoor temperature in summer. With these backgrounds, construction companies are considering building bake-out as the urgent method which can be applied after the construction complete.

Based on the result of the former researches on building bake out [2],[3], it is believed that building bake-out has the potentials to improve indoor air quality in residential buildings, although contradictory results have been also reported [4],[5]. To clarify the effectiveness of bake-out, it is required to investigate variation of pollutants emission rate from building materials installed in buildings due to bake-out procedures.

The objective of this study is to investigate emission rate variations of VOCs emitted from building materials commonly used in residential housing units during building bake-out.

Using passive sampling methods, measurements of variation of VOCs emission rates from various building materials such as flooring materials, wall papers, and furniture were conducted during bake-out.

RESEARCH METHOD

Description of test house

Measurements were taken on residential housing units to evaluate variations of VOCs and formaldehyde emission rates due to building bake-out. Two identical residential housing units with an area of 85m^2, finished and furnished with the same materials were selected from the apartment buildings located in Seoul, Korea. A radiant floor heating system was adopted in test housing units. Mechanical ventilation system throughout the residential unit was not used, but rather an exhaust fan was installed in the kitchen and bathrooms. As shown in the unit floor plan shown in Figure.1, all of the room can be ventilated by natural means of opening windows and doors. In the units, the wall paper with water glue was finished on walls and ceilings. On the floors, a wood based material which is laminate flooring was finished. A different types of furniture was installed in bedroom2 and kitchen.

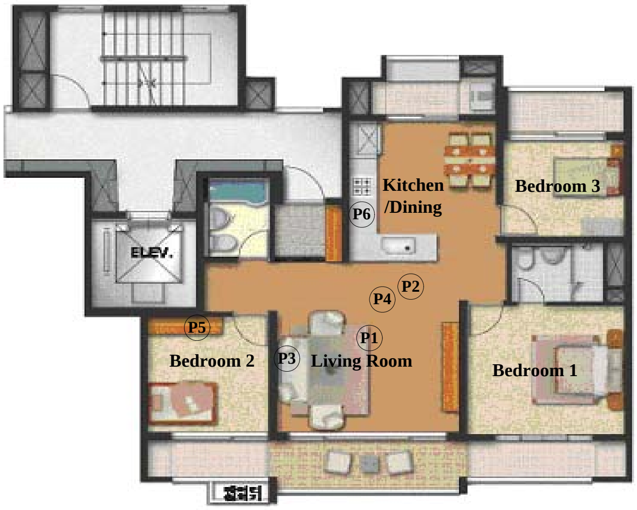

Figure 1. Floor plan of the unit &Measuring Point

Table 1. Investigated materials

Measuring point	Measured Location	Material composition
P1	indoor air	-
P2	Ceiling in living room	water glue + wall paper
P3	Wall in living room	water glue + wall paper
P4	Floor in living room	wood based material
P5	furniture in bedroom	wood based material
P6	furniture in kitchen	wood based material

Measurement Schedule

Measurements of VOCs emission rates from various building materials were taken three times in nearly same temperature and ventilation conditions. As shown in Figure 1 and Table 1, measurements were taken at ceiling, wall, floor and two types of furniture. Indoor concentration was also measured. During all of the measurements, the test units were shut tightly and the air temperature of the units was maintained around 30℃. After the initial measurement was carried out, the test units started to operate the heating system to raise the indoor temperature to reach 35℃ with different ventilation rates for five days. Then, all the units were naturally ventilated by opening windows for two days, and baked out for another five days. The second measurements were carried out after the test units were ventilated by opening windows for two days to be sufficiently cooled down. Measurement schedules and points are shown in Figure 2 and Table 1, respectively.

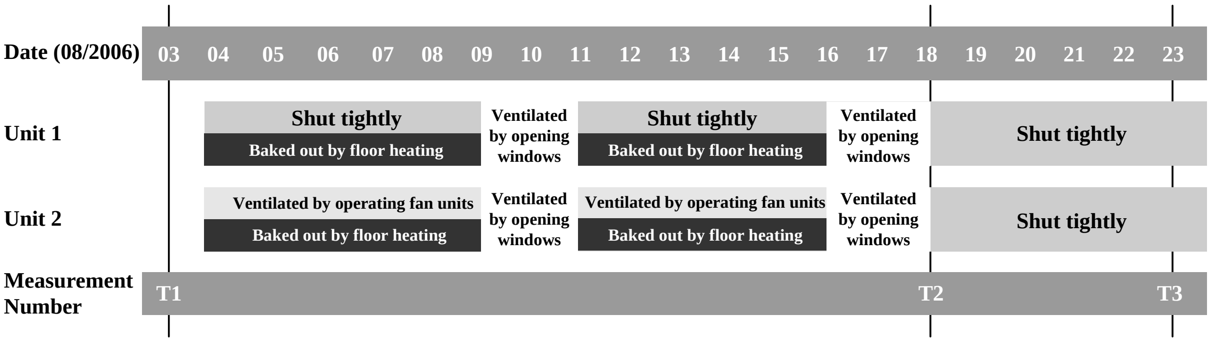

Figure 2. Measurements schedule according to bake-out conditions

Measurement and analysis methods

For measuring emission rates of building materials installed in test units, passive sampling methods were used [6]. VOC-SD and DSD-DNPH passive samplers were used for sampling VOCs and formaldehydes respectively. Each sampler was inserted into an instrument made of stainless steel and the sampling was carried out for 24hous. VOC-SD samplers are analyzed by GC/MS after solvent desorption using CS2 and DSD-DNPH samplers are extracted with acetonitrile and analyzed by HPLC. Among the various pollutants, toluene and formaldehyde levels were presented in this paper, and relative humidity and indoor temperature in the units were also measured.

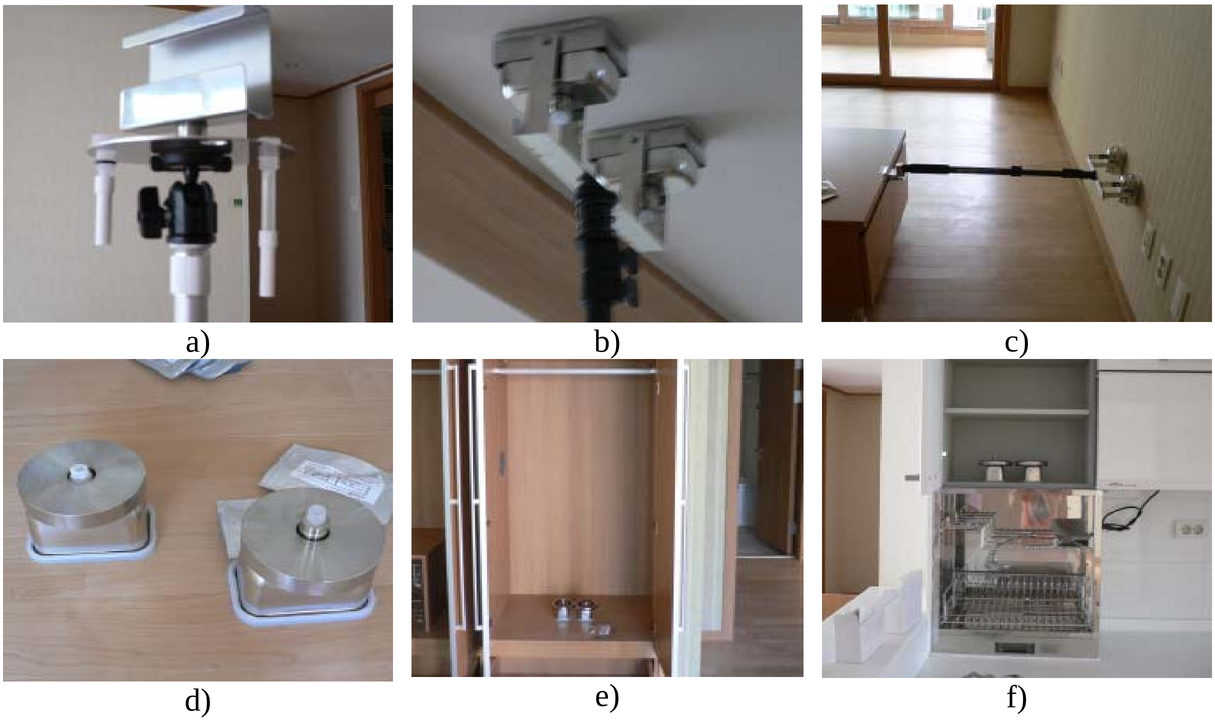

Figure 3. Measuring points a) indoor air(P1) , b) ceiling in living room(P2), c) wall in living room(P3), d) floor in living room(P4), e)furniture in bedroom(P5), f) furniture in kitchen(P6)

RESULTS AND DISCUSSION

Environmental conditions

The room temperature of each unit for whole test periods was plotted in Figure 4. The room temperature was maintained at around 30℃ before the bake-out. During the bake-out, the room temperature of both units reaches 35℃. Although, unit 2 was ventilated by operating fans and opening the windows, indoor temperature sufficiently reached about 35℃ due to fairly high outdoor temperature around 30℃ in summer. The figures show both of the units were under very similar temperature conditions in spite of the different ventilation condition.

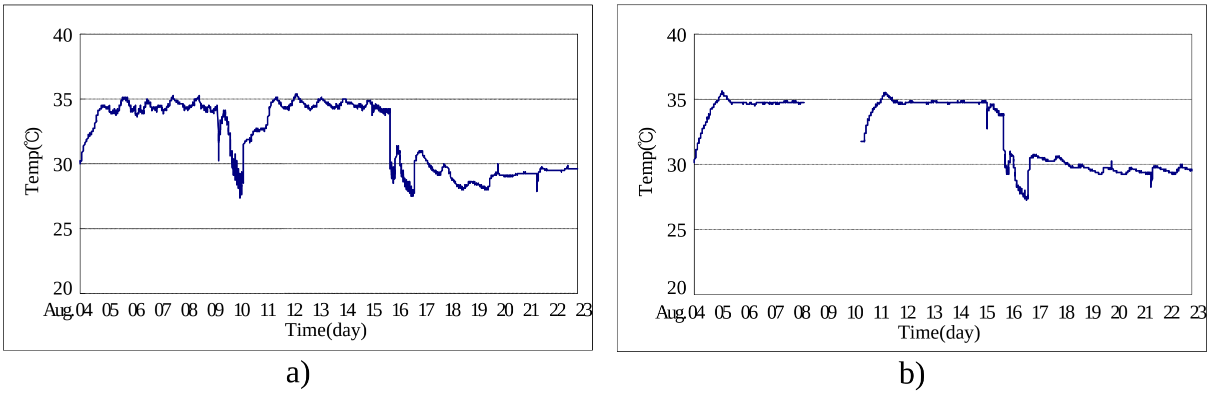

Figure 4. Indoor air temperature a) Unit 1, b) Unit 2

Variation of pollutants emission rates

Toluene emission rates of the tested materials and indoor concentration in units 1 and 2 are plotted in Figure 5. Regardless to the ventilation conditions during bake-out, after bake-out, the emission rate of toluene from all kinds of materials is clearly reduced compared to the initial values at measurement T2. At measurement T3, however, emission rate at T2 fairly rose up compared to the values at measurement T2, but most of them still has lower values than the initial values. These rises of the emission rate were observed in all kinds of materials. Although it is hard to interpret this result, one possible explanation is that rearrangement of pollutants distribution inside materials increases their emission rate. Based on the comparison of the values measured at the final measurement with the initial values, the emission pattern of wood based materials appears to be different from wall paper. Clear decrease of toluene emission rate from wood based materials such as floorings and furniture is observed, while the toluene emission rate in unit 1 slightly decreased or even rose up compared to the initial values. It is seemed that toluene emission from wood based materials is affected by ventilation conditions.

Formaldehyde emission rates of the materials and indoor concentration are plotted in Figure 6. The emission rate of formaldehyde showed the different patterns with that of toluene. In contrary to the result of toluene, we didn’t observe reduction of formaldehyde emission rate after bake-out. In unit 1, which was shut tightly during bake-out, formaldehyde emission rate rose or maintain at approximately same level compared to the initial level.

Based on the result, it is found that the variation of pollutants emission rates depends on types of materials and pollutants as well as ventilation rates. If materials showing a good reduction of pollutants emission rates by bake-out were mostly installed in a building, we could expect a better IAQ after conducting bake-out procedure. The effectiveness of conducting bake-out will vary with emission characteristics and installed area of each building material. Consequently, investigation of emission characteristics of building materials are required prior to conduct the building bake-out for confirming in the result of better IAQ by bake-out.

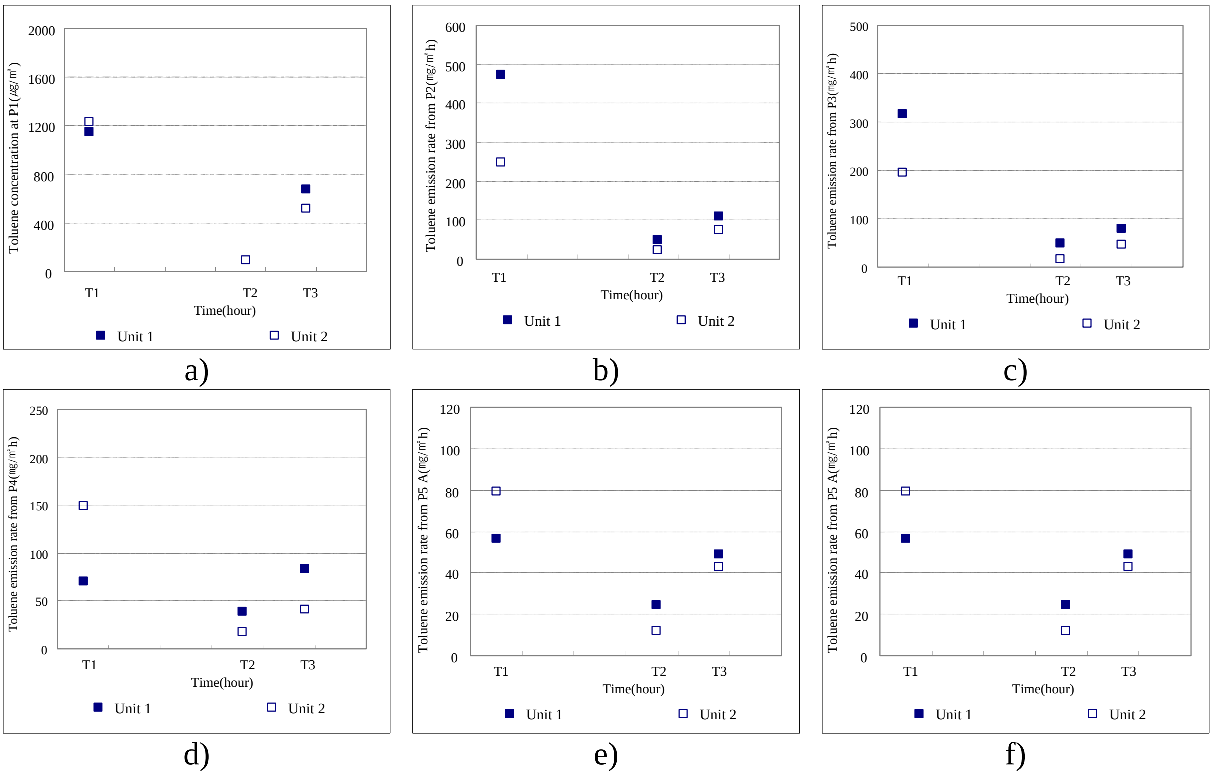

Figure 5. Emission rates of toluene a) indoor air(P1) , b) ceiling in living room(P2), c) wall in living room(P3), d) floor in living room(P4), e)furniture in bedroom(P5), f) furniture in kitchen(P6)

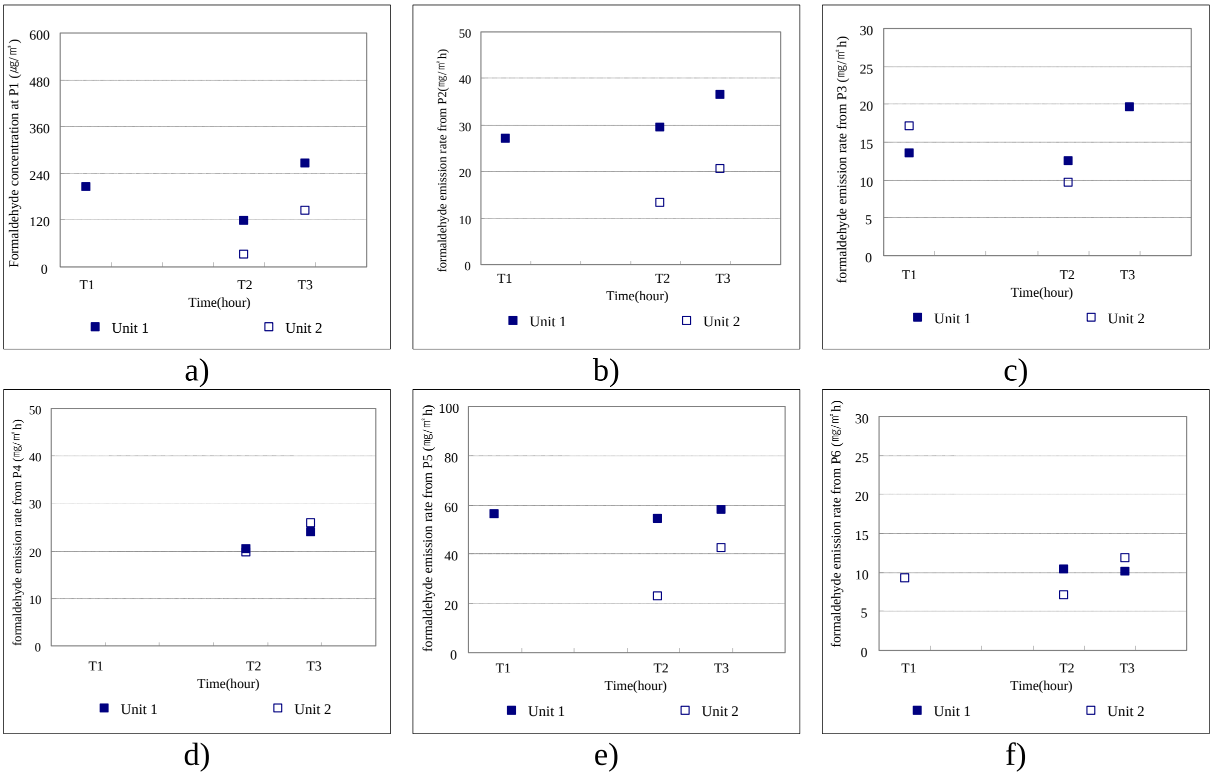

Figure 6. Emission rates of formaldehyde a) indoor air(P1) , b) ceiling in living room(P2), c) wall in living room(P3), d) floor in living room(P4), e)furniture in bedroom(P5), f) furniture in kitchen(P6)

CONCLUSION

In this study, we measured VOCs and formaldehyde emission rate from building materials during bake-out to clarify the effectiveness of bake-out conductions. Measurements of emission rates were carried out with passive sampling method for about one month in residential housing units.

The result shows bake-out easily drops down toluene emission rate compared to formaldehyde emission rate. In unit 1, which was shut tightly during bake-out, toluene emission rate from wall paper prominently decreased. In unit 2, which was ventilated during bake-out, toluene emission rate from wood base materials has relatively large reduction rate. However, formaldehyde emission rate is not affected by bake-out. Based on the result, we concluded that the effectiveness of conducting bake-out will vary with emission characteristics and installed area of each building material. Because indoor air quality can be better or worse according to the emission characteristics and installed area of building materials, investigation of emission characteristics of building materials are required prior to conduct the building bake-out for confirming in the result of better IAQ by bake-out.

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Estimation Method of Emission Rate and Effective Diffusion Coefficient using Micro Cell

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SUMMARY

In this report, we proposed the estimation method of building material properties by using both numerical analysis and measuring the time history of concentration in micro cell, and showed that the effective diffusion coefficient, emission rate and initial concentration in the building material is provided with a single measurement. The effective diffusion coefficient of testing building materials were estimated to be from about 4.0×10^{-10} to 2.5×10^{-8} [m^2/s] in this proposed method. Although this proposed method is necessary to improve to obtain more reliable result, it is able to apply to relative comparison about VOCs emission characteristic in building materials as a screening test.

INTRODUCTION

In order to prevent indoor air pollution by VOCs emitted from building materials, it is necessary to select suitable building materials with lower VOCs emissions, and to confirm a VOCs concentration range in rooms in advance. Therefore, the information of building material properties, which is VOCs emission rate, effective diffusion coefficient and chemical content (initial concentration in building materials), are important. Generally, VOCs emission rate is measured by the chamber method [1][2], FLEC [3][4], ADSEC [5] etc., and the effective diffusion coefficient is measured by twin chamber method [6], cup method [6][7], mercury intrusion porosimetry (MIP) method [8]. However, it is the actual fact that the information of building material properties has not been got ready adequately from the viewpoint of effort and cost taken in testing.
The overarching goal of this study is to develop the simultaneous measurement method that is able to estimate the emission rate and effective diffusion coefficient by single measurement. In this repot, we propose a estimation method by using micro cell and convenient gas monitor (in this research, we adopt the multi-gas monitor produced in INNOVA), and compare the effective diffusion coefficient by proposed method with conventional methods.

OUTLINE OF PROPOSED METHOD

In this study, we focused a time history of VOCs concentration in the air (gas) phase that occurred when target building material is covered with an airtight container (in this research, we call as Micro Cell). Generally, the VOCs concentration in micro cell gradually increases, and finally reaches an equilibrium concentration. In other words, VOCs emission rate form building materials gradually decreases toward zero. The profile of the time history of VOCs concentration in micro cell is determined by effective diffusion coefficient and initial concentration (level and distribution) in target building material. Further, when the initial concentration in the building material is uniform, the profile of the time history of VOCs concentration that normalized to the initial concentration in the building material is determined by the effective diffusion coefficient alone.

This proposed method estimates VOCs emission rate and effective diffusion coefficient by using both numerical analysis and continuous concentration measurement in micro cell. Before the measurement, we make the chart of time histories of VOCs concentration by numerical analysis that assumed various effective diffusion coefficients. And we measure the concentration in micro cell until it becomes equilibrium. Subsequently, the initial concentration in the building material is obtained from the equilibrium concentration, and the effective diffusion coefficient is estimated by overlapping the measurement result with chart. After estimating the effective diffusion coefficient, the emission rate when reference concentration is guideline value (e.g. 100 ug/m^3 for Formaldehyde in WHO) is estimated as a representative value. In this way, this proposed method is able to estimate the emission rate, effective diffusion coefficient and initial concentration in the building material with a single measurement.

Outline of numerical analysis

The micro cell is shown in Figure 1. The numerical analysis is carried out in one dimensional, since the micro cell is nearly rectangular (100mm× 550mm× 600mm^h) and free slip can be assumed for the internal wall.

For the inside of the building material, the diffusion equation shown by Equation (1) is solved.

$$\frac{\partial C}{\partial t} = \frac{\partial}{\partial x}\left(D_c \frac{\partial C}{\partial x} \right) \tag{1}$$

Here, C is the equivalent vapor-phase concentration of the objective chemical compound in building material, and Dc is the effective diffusion coefficient of the objective chemical compound.

The flux conservation shown in Equation (2) exists on the interface between the surface of the building material and the air phase.

$$-D_c \left.\frac{\partial C}{\partial x}\right|_{ws+} = -D_a \left.\frac{\partial C}{\partial x}\right|_{ws-} \tag{2}$$

Here, Da is the molecular diffusion coefficient of the objective chemical compound in the air phase, ws is the interface between the building material and the air, plus (+) indicates the building material side and minus (–) indicates the air phase side.

For the air phase in the micro cell, the diffusion equation shown by Equation (3) is solved. The object of this analysis is only the diffusion phenomenon attributable to molecular diffusion assuming that the convection phenomenon and turbulent diffusion in the micro cell are negligible, and assuming isothermal distribution.

$$\frac{\partial C}{\partial t} = \frac{\partial}{\partial x}\left(D_a \frac{\partial C}{\partial x} \right) \tag{3}$$

Equations (1) through (3) are analyzed by the finite differential method. The objective region of numerical analysis is shown in Figure 2.

Concentrations are analyzed using a fixed value for the normalized initial toluene concentration in the building material and a fixed value for the molecular diffusion coefficient of toluene in the air phase, by changing the effective diffusion coefficient in the building material in steps. Calculation conditions and boundary conditions for the numerical analysis are shown in Table 1.

Figure 1 External appearance of Micro Cell

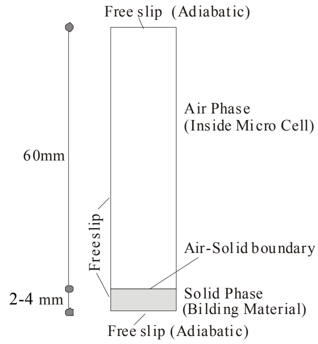


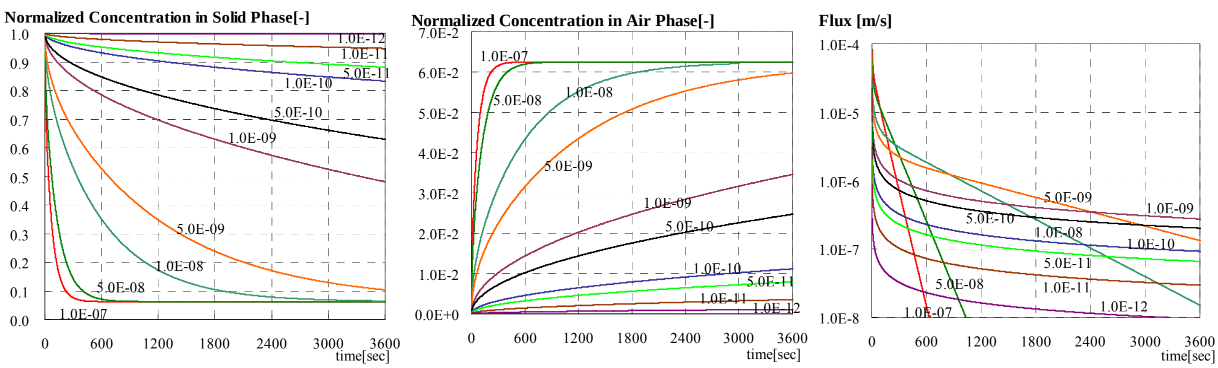
Figure 2 Analysis model

Table 1 Calculation conditions and boundary conditions

Mesh division	Air : equally spaced mesh with a 1.0×10^{-4} [m] interval Solid : equally spaced mesh with a 1.0×10^{-6} [m] interval
Initial concentration	Air : 0 [-] Solid : 1.0 [-] (Normalized concentration)
Chemical substance (Toluene)	Air : $Da=7.8\times10^{-6}$ [m^2/s] Solid : $Dc=1.0\times10^{-7}\sim1.0\times10^{-12}$ [m^2/s]
Air-Solid boundary	Equation (2)
Time	0～3600 [sec] unsteady analysis

Effective diffusion coefficient prediction chart

Figure 3 shows the prediction results of toluene-equivalent TVOC concentration histories in the building material and in the micro cell, changing the effective diffusion coefficient (*Dc*) in the building material stepwise from 1.0×10^{-7} [m^2/s] to 1.0×10^{-12} [m^2/s]. All the values in the figure are normalized to the initial concentration in the building material. Figure 3(1) shows the relationship between the time history of the average concentration in the building material and *Dc*, Figure 3(2) shows the relationship between the time history of the average concentration in the micro cell and *Dc*, and Figure 3(3) shows the relationship between the emission rate (emission flux) and *Dc*. In this report, uniform distributions of initial concentration in the building materials are assumed as initial conditions.



(1) Solid phase (2) Air phase (3) Emission flux

Figure 3 Time history of toluene concentration for various *Dc* (4mm). Similarly, the charts when building materials thickness is different were made by numerical analysis

When *Dc* of the target building material is 1.0×10^{-7} [m^2/s], the concentration in the building material and in the micro cell become almost equilibrium at 500 [sec] from the beginning of analysis. When *Dc* is 5.0×10^{-9} [m^2/s], concentrations approach equilibrium values in 3600 [sec] from the beginning of analysis. As *Dc* becomes smaller, the change in concentration per unit time also becomes smaller.

OUTLINE OF EXPERIMENT

The building materials used in the experiments are sandwiched between stainless steel (SUS) boards and controlled at a constant temperature (301 [K]) for at least a month to rid the concentration gradient in testing building material in incubator [9]. A SUS board is placed lower surface of the testing building material and the upper surface is left open to allow one-sided diffusion (emission).
The time history of concentration in the micro cell is continuously measured with a photo-acoustic spectrometry method (multi gas monitor, INOVA). In order to ensure a closed experimental system, the air sampled from the micro cell is returned to the micro cell after the multi-gas monitor measurement of the concentration. The atmospheric temperature of the experiments is controlled at 301 [K] by placing the test building materials to which micro cells are attached in a incubator. Humidity is not controlled. Concentration measurements are continued until an equilibrium concentration in the micro cell is reached.
The measurement system is shown in Figure 4, and the experimental conditions are shown in Table 2.

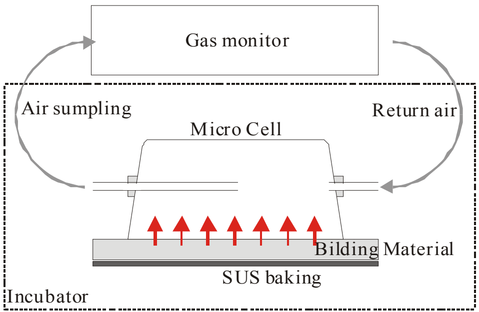


Figure 4 Measurement system using Micro Cell

Table 2 Experimental conditions

Temperature	301 [K] (28 [℃]) (Humidity is not controlled)
Concentration measurement	Multi-gas monitor (INOVA) Filter: Total Organic Carbon ref. Toluene Water compensation: ON Sampling interval: 60 sec Sampling rate: 140cc/sample
Testing building materials	(1) Plywood; 4mm (2) Medium Density Fiberboard (MDF); 4mm (3) Cushion Floor (CF); 2.5mm (4) Poly Vinyl Chloride (PVC) sheet; 2mm (5) Synthetic rubber; 2mm
Aging conditions	Testing building materials sandwiched between SUS boards were placed in a incubator at 301 [K] and 50 [%RH] for more than a month

Outline of effective diffusion coefficient measurement by Cup method

In order to validate the measurement value by micro cell method proposed in this research, cup method and MIP method to measure effective diffusion coefficient are carried out. The effective diffusion coefficient by cup method is estimated from the measurement result for the flux value in the condition that made the concentration difference on the top and bottom of building material. In this experiment, liquid toluene was placed in the stainless steel cup is shown in Figure 5 and the opening of the cup was covered with the building material. The cup was placed in a incubator at 301[K], and the toluene concentration on the upper surface of the building material was kept at zero by sufficient ventilation. The diffusion coefficient is calculated from Equation (4).

$$Dc = \frac{m}{A_b}\frac{d}{C_{sat}} \tag{4}$$

Here, m is the weight change per unit time of toluene in the cup [mg/h], A_b is the surface area of the building material [m^2], d is the thickness of the building material [m], and C_{sat} is the saturated vapor phase concentration at the ambient temperature [mg/m^3].

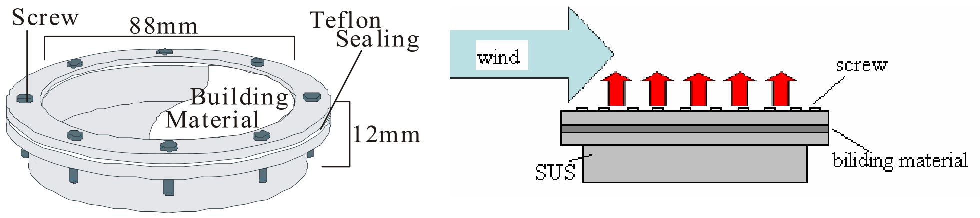


Figure 5 Detail of the Cup

Outline of effective diffusion coefficient measurement by MIP method

The effective diffusion coefficient by MIP method is estimated using the porosity (ε) and tortuosity factor (τ) are estimated from the measurement results for the amount of mercury that has entered the fine pores of the building material under pressurization. The effective diffusion coefficient is calculated from Equation (5) based on the parallel pore model [10].

$$D_c = \frac{\varepsilon}{\tau}\frac{1}{\frac{1}{D_{KA}}+\frac{1}{D_a}} \tag{5}$$

Here, Da is the molecular diffusion coefficient [m^2/s] of chemical substances in the air, and D_{KA} is the Knudsen diffusion coefficient [m^2/s] calculated from Equation (6).

$$D_{KA} = 3.067r\sqrt{\frac{T}{M}} \tag{6}$$

Here, r is the mean fine pore diameter [m], T is the absolute temperature [K], and M is the molecular weight [kg/mol].
Testing building materials were vacuum-dried for a day as pretreatment, and an AutoPore III 9420 (Micromeritics) was used for the measurement. The range of the fine pore distribution measurement was from a radius of about 0.0018 to 100 [μm].

EXPERIMENTAL RESULTS

The time histories of concentration in micro cell are shown in Figure 6. Although the concentrations gradually increase with time, the profile and range of concentrations are different with each building materials.

Time histories of Normalized concentration in micro cell are shown in Figure 7. Concentration values in Figure 7 were normalized to the initial concentration in the building materials. The mean concentration histories in micro cell by numerical analysis are superposed on Figure 7. As for the early times, experimental values tend to rise slightly up than calculation values. However the profile of the concentration history almost fitted. In this experiment, the effective diffusion coefficients (toluene-equivalent TVOC) of the building materials are estimated to be from about 4.0×10^{-10} to 2.5×10^{-8} [m^2/s].

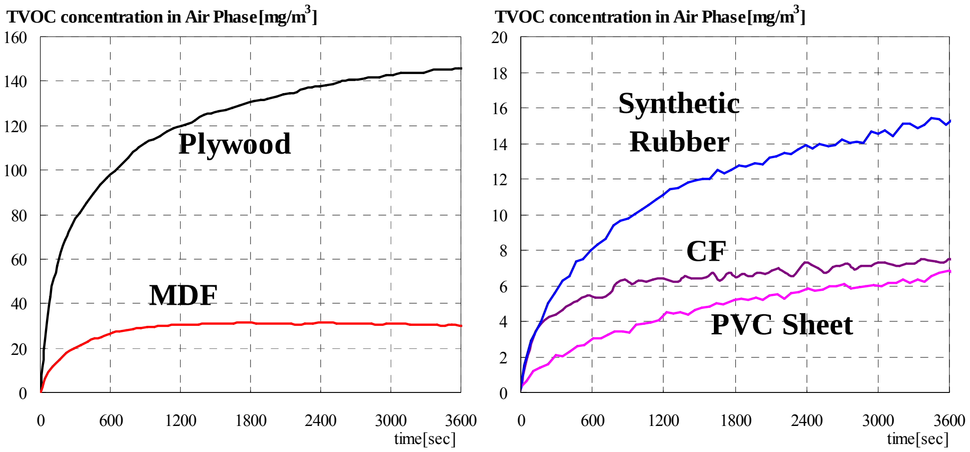
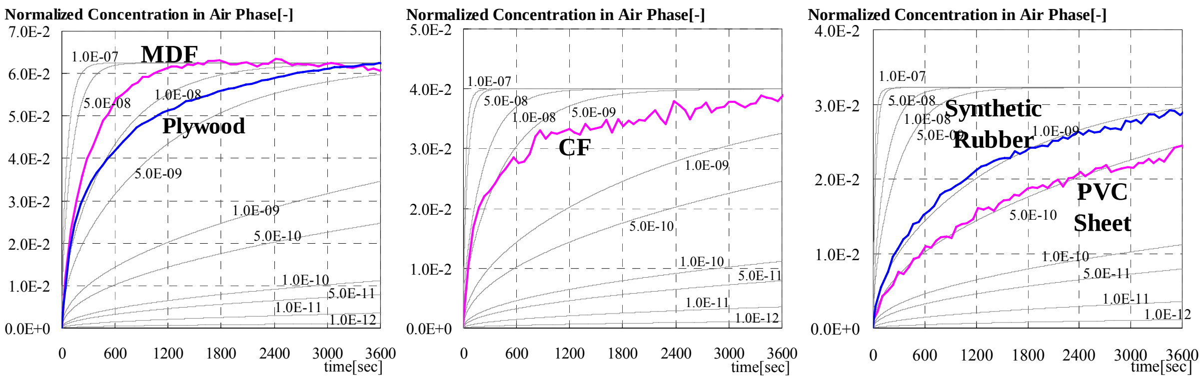


Figure 6 Time history of TVOC concentration in Micro Cell



(1)Plywood, MDF (4mm) (2)CF (2.5mm) (3) PVC sheet, Synthetic rubber (2mm)

Figure 7 Time history of normalized concentration in air phase

The emission flux is estimated by reading the emission flux chart when the concentration in micro cell reaches the guideline value. In this study, since the dynamic range of the filter for hydrocarbons (Total Organic Carbon ref. Toluene) in the multi-gas monitor is wide, measured concentration values are overestimated in comparison with ordinary TVOC values. Therefore, the emission rate when the reference concentration in micro cell reached about 10 times as high as the TVOC guideline value (4 [mg/m^3] in this case) is estimated in this report. The emission rates when the air concentration is 4 [mg/m^3] are estimated to be from about 0.3 to 460 [$mg/m^2/h$].

Results of the effective diffusion coefficient, emission rate and initial concentration in the building materials are shown in Table 3. This proposed method is able to obtain three values of the building material property with a single measurement.

Table 3 Results of the effective diffusion coefficient, emission rate and initial concentration

Testing Building Materials	Effective Diffusion Coefficient (Dc) [m^2/s]	Emission Rate (E) [$mg/m^2/h$]	Initial concentration (C_0) [mg/m^3]
Plywood	7.0×10^{-9} - 1.1×10^{-8}	2.9×10^{2} - 4.6×10^{2}	2.3×10^{3}
MDF	1.5×10^{-8} - 2.5×10^{-8}	4.0×10^{1} - 6.7×10^{1}	5.0×10^{2}
CF	2.0×10^{-9} - 6.0×10^{-9}	7.6×10^{-1} - 2.3×10^{0}	1.9×10^{2}
PVC Sheet	4.0×10^{-10} - 5.5×10^{-10}	2.9×10^{-1} - 3.9×10^{-1}	2.8×10^{2}
Synthetic Rubber	8.5×10^{-10} - 1.2×10^{-9}	2.2×10^{0} - 3.1×10^{0}	5.3×10^{2}

Figure 8 shows the effective diffusion coefficient by each measurement method. In this experiment, the effective diffusion coefficient value by MIP method was the largest, followed by cup method and then micro cell method. The effective diffusion coefficient of PVC sheet and synthetic rubber were small in comparison with other building materials. That difference is order of 10^{-2} in MIP method, and is order of 10^{-1} to 10^{-2} in micro cell method.

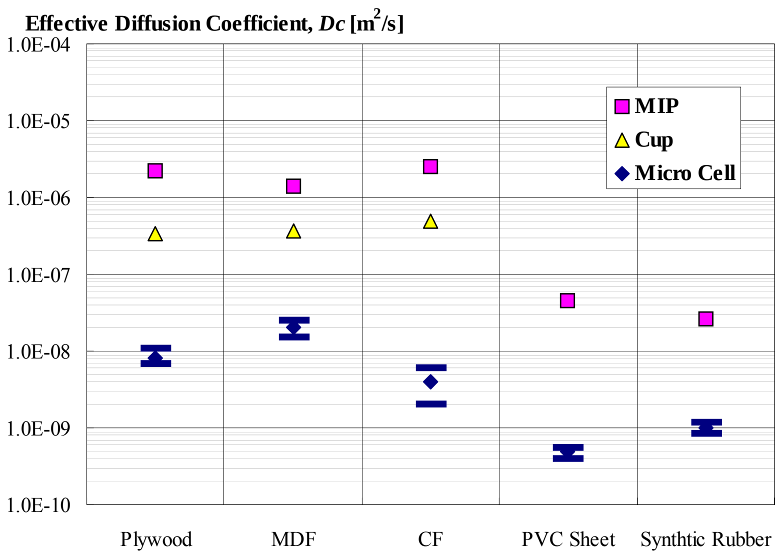


Figure 8 Effective diffusion coefficients by each measurement method

DISCUSSION

The effective diffusion coefficient values obtained by this proposed method are evaluated smaller than those obtained by the other measurement methods. Previous studies reported that effective diffusion coefficient values from MIP method tend to be larger, since the effective diffusion coefficient values are calculated from the physical properties of the building materials alone without considering the influence of interactions with absorption and desorption in the building materials and influence of the humidity [8]. In case of experiments by cup method, the concentration in the cup is far higher than ordinary indoor concentration. Consequently the effective diffusion coefficient may be overestimated when the concentration of the pore in the building material is saturated or when the concentration gradient of the cavity in cup is not neglected. From this, the result that the effective diffusion coefficient values by micro cell method tend to be smaller than those by MIP method and cup method is considered

to be valid. Also, the effective diffusion coefficient of PVC sheet and synthetic rubber tends to be smaller in comparison with other building materials, in MIP method and both of micro cell method. This proposed method is able to apply to relative comparison about VOCs emission characteristic in building materials as a screening test.
In this report, although the building material properties values by micro cell method are toluene-equivalent TVOC value by multi-gas monitor, those by MIP method and Cup method are toluene alone. The precision of estimated value greatly depends on the precision of the concentration history measurement. In order to improve the precision of estimated value in this proposed method, it is necessary to validate using measurement apparatus capable of high-precision analysis of single components.

CONCLUSIONS

(1) In this report, we proposed the estimation method of building material properties by using both numerical analysis and measuring the time history of concentration in micro cell, and showed that the effective diffusion coefficient, emission rate and initial concentration in the building material is provided with a single measurement.
(2) The effective diffusion coefficient of testing building materials were estimated to be from about 4.0×10^{-10} to 2.5×10^{-8} [m^2/s] in this proposed method, and were smaller than the effective diffusion coefficient values by cup method and MIP method.
(3) Although this proposed method is necessary to improve to obtain more reliable result, it is able to apply to relative comparison about VOCs emission characteristic in building materials as a screening test.

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A European Project SysPAQ

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SUMMARY

The European research project Innovative Sensor **Sys**tem for Measuring **P**erceived **A**ir **Q**uality and Brand Specific Odours (SysPAQ) is started under the VIth framework programme under the work programme “New and Emerging Science and Technology” (NEST PATHFINDER "Measuring the Impossible"). The Kick-off of the project was on the first of September 2006. Ten partners (3 Companies, 4 Universities, 3 research Institutes) from 5 countries are involved.

The main goal of this project is to develop an innovative system to measure indoor air quality as it is perceived by humans to be used as an indicator and a control device for the indoor air quality. The system will also be able to detect brand specific odours and it will serve as a novel interior odour design tool for the vehicle industry.

INTRODUCTION

This innovative sensor system is highly demanded by the European society considering that humans spend about 90% of their time indoors, either at work, at home or when commuting between work and home. Recent data shows that improved indoor air quality will result in fewer complaints, increased comfort, less health problems and higher productivity. Consequently, quality of life will be improved.

Up to now, the indoor air quality has been quantified applying three different measurement methods separately. The three methods are based on the human perception of indoor air quality, chemical measurements and sensors for specific odours. Considering measuring perceived air quality human assessments are still being superior to chemical measurements because of the unmatched sensitivity to many odorous indoor air pollutants. One of the reasons is that in most cases the chemical measurements or signals from chemical sensors designed to detect special odours could not be correlated with the assessments made by humans. They do obviously not measure the relevant indoor air pollutants triggering human sensory response. This project will build upon current knowledge on the perceptual effects of indoor air pollutants and on the experience gained in using chemical measurements and sensors for specific odours. The approach of the project will be to enhance the present state of the art of sensor systems, the perceptual methods and the software tools for modelling human response, and integrate them into one innovative system for measuring indoor air quality as it is perceived by humans. A bridge will consequently be created between the previous works in this area and progress will be achieved by integrating measurements, sensors and modelling by a holistic approach.

The main challenges are the finding of a perceptual space by using different reference odours, the measuring procedure for perceived air quality, and a characterisation method of brand specific odours. This work requires advanced knowledge in perception psychology and technical excellence in sensor system design and indoor environment research, knowledge and experience which the project partners contribute. It is still an open question how to mimic the human perception of odours and air quality. A certain risk of failure is involved in this project’s unsolved problem of how to define an adequate system of reference odours for human perception of odours. In any case the project will provide new insight to the human perception of odours and indoor air quality, and will deliver a new approach to assess the perceived air quality and brand specific odours by an advanced sensor system.

PROJECT OBJECTIVES

The main goal of this project is to develop an innovative system to measure indoor air quality as it is perceived by humans based on perception modelling combining measurements of sensors and assessments of perceived air quality by sensory panels. The innovative sensor system can be used as an indicator, monitor and control device for the indoor air quality in buildings and vehicles. Furthermore, the system will be able to detect brand specific odours and it will serve as a novel interior odour design tool for the vehicle industry. The main objectives of the project are:

1. Definition of a method for measuring the perceived air quality and perceived odour intensity in buildings and vehicles. This method will be used by all different labs using sensory panels.
2. To find an advanced perception model for indoor air assessments. The model will be the major input to the software design for the innovative sensor system and it provides new insight to the human reaction to odours.
3. Development of an innovative sensor system for measuring, correspondingly, the perceived air quality and brand specific odours quality.
4. Calibration and test of the innovative sensor system for measuring, correspondingly, the perceived air quality and brand specific odours. The final version of the system is intended for the following applications :
    - Monitoring of the ambient air within buildings and vehicles.
    - Monitoring of the quality of the inlet air to buildings to ensure health and comfort for occupants.
    - Labelling of materials based on emissions from buildings and vehicles.
    - Control of the production process of building and vehicles materials.

Along with human activities, emissions from building materials, furnishing and equipment are main contributors to air pollution indoors. To reduce indoor air pollution loads it has been recommended to use low-polluting materials and to increase outdoor air supply rates. The new EU Energy Directive requires substantial reduction of energy use which may lead to reduction of ventilation rates and increased indoor air pollution enhancing the need of low-emission materials. In addition to measurements of indoor air quality as perceived by humans the system developed in the project can be used to control the emission rates from building materials already at the production stage. At present, manufacturers generally reduce the emissions from building materials by monitoring the emission rates of a few compounds in practice, but not necessarily the most relevant odour active compounds for perceived air quality. The pollution mixture affects the perceived air quality indoors. It is not taken into account so far. The proposed system can be used by the producers of building materials, furnishing and equipment materials to ensure that the emissions from their products would not negatively affect the perceived air quality indoors. In many countries labelling systems of building materials exist, so that the end-users can select the materials with reduced emissions. The suggested system for measuring the perceived air quality can also be used to quantify whether a material can get a label.

The selection of interior materials is a very important factor for the vehicle and transportation industry (train, car, boat, airplane etc.). The goal of the selection process is to create a high-standard perceived air quality in vehicles combined with a brand specific odour impression. To meet this goal the system measuring the perceived air quality seems indispensable.

The interdisciplinary structure of the project consortium will enable innovative research and it will provide new insights to the human perception of air quality and brand specific odours. The project management will ensure a strong interaction between new perception models, hardware development as well as software design for the innovative sensor system.

APPROACH AND METHODOLOGY

The overall approach of the project and the priorities of the different work-packages are shown in figure 1. The figure illustrates the parallelism of the human perception of air quality and brand specific odours on the left hand side and the sensor system development on the right hand side.

The two arrows in the centre of figure 1 indicate the coupling of sensory panel experiments and the sensor system hardware and software development. The innovative sensor system has to detect all relevant odour active substances. Based on the knowledge of the project partners and experiments of the project a list of relevant substances is submitted to the two sensor specialists. The second arrow indicates the input of the new perception model of the perception specialist to the software development. The mathematical model of the new software for the sensor system will apply a reference odour system to reproduce an odour space that covers perceived air quality and brand specific odours in buildings and vehicles.

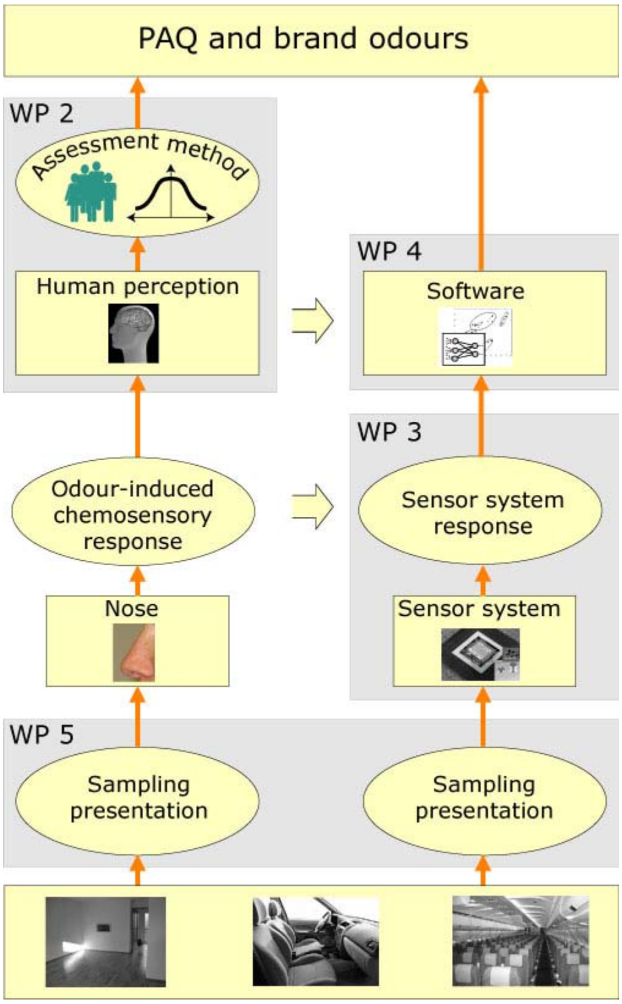

Figure 1: Parallelism of the human perception of air quality and brand specific odours on the left hand side and the sensor system development on the right hand side

WORKPLAN OF THE PROJECT

Introduction - general description

The work of the project is divided into two main tasks. The first task is focused on human perception of odours. This part includes the definition of a method to measure perceived air quality and odour intensity by sensory panels. The used assessment methods can have a major influence on the assessment results itself. Besides the assessment method the main challenge of the human perception task is the definition of an odour space using reference odours. The multidimensional odour space has to account for all interior odours in buildings and vehicles.

This new odour space is the major input to the second main task of this project. This part is focused on the hard- and software development for the innovative sensor system. The contribution of two sensor specialists enables the application of the most advanced sensor technology. The sensitivity and selectivity of the sensors is adapted to odour active substances and the new odour space. The combination of different sensor types leads to novel sensor system that covers considerable more substances than any present stand alone system. The data processing for this sensor systems combines advanced statistical pattern recognition methods with the novel reference odour systematic. The link between the sensor signals and the perceived air quality consists of a classification process in terms of the reference odours followed by a regression method calibrated by the experimental data of the project.

This two main tasks **human perception** and **system development** break down to 4 work-packages (human perception and odour space, sensor system, data processing and pattern recognition, system calibration). The management process and the dissemination activities control and link these four work-packages, see figure 2.

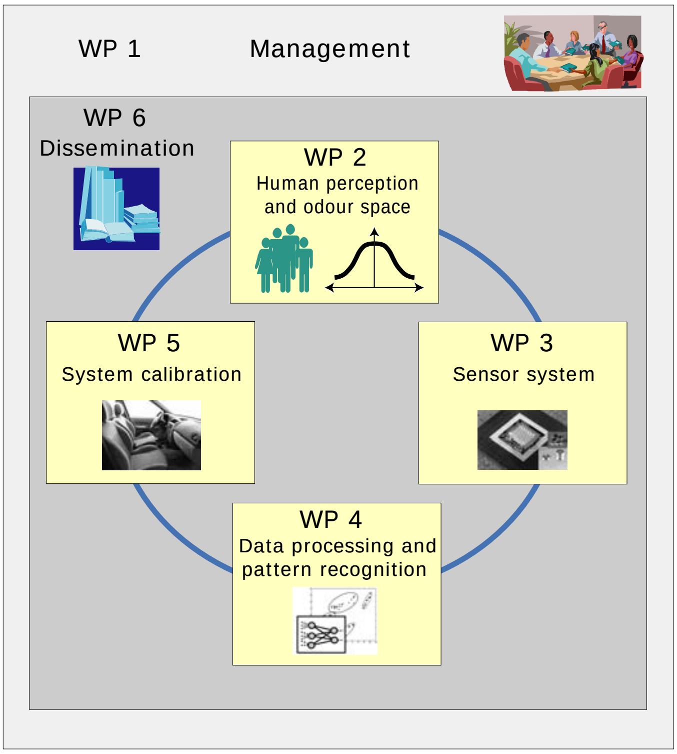

Figure 2: Work-package structure

Work planning

All overall **management** activities are included in WP1. The duration of the management activities covers the whole project life time. The main management activities are directed to the organisation of the project meetings and the knowledge transfer between the different work-packages. The management of the project insures a punctual delivery of all reports to the European Commission and it checks milestones and deliverables of all work-packages. The management also handles the global risk management of the project. The risk management includes the call for extra expert meetings in case of unsolved project tasks and the adjustment associated work-packages in the case of missing milestones or deliverables.

The **human perception** task of the project is mainly handled in WP2. Starting point of WP2 is the definition of a method for all sensory panel assessments of perceived air quality and odour intensity. The method is based on the knowledge of the experimentally experienced project partner and the input of new multidimensional psychophysical perception models. This common standard for the assessment method guarantees comparable measurements at all labs for the innovative sensor system calibration data. The second and the most important part of WP2 is the definition of a new odour space based on reference odours. The new odour space has to cover all relevant odours of indoor air related environments. All project partners of WP2 have been working on the characterisation of odours in the past. The work-package leader Karolinska Institute (KI) will provide a human perception model as a theoretical base for the new odour space. First experiments of KI will examine the useability of the new odour space based on reference odours to reproduce indoor air odours from building and vehicle environments. The quality of the odour space is the critical factor for the innovative sensor system and it will influence the software design process. A continuous knowledge transfer between WP2 and WP4 will insure a fast translation of new findings into the software system.

WP3 covers all activities related to hardware improvements for the **innovative sensor system**. This work-package includes the improvement and adaptation of single multi-gas sensors as well as the combination of different multi-gas sensor technology. The adaptation process and the selection of sensors are based on literature review and some analysis of indoor air environment of buildings and vehicles in terms of odour active compounds. The sensor development and sensor combination is handled by two experienced sensor specialist Alpha MOS and Forschungszentrum Karlsruhe (AM, FZK). The two partners can provide gas chromatography/mass spectrometry combined with sniffer experiments to prove the response of the multi gas sensor system to odour relevant substances. The check and minimisation of cross sensitivity (to other chemical compounds, relative humidity and temperature of sample air) of the sensors provides reproducible measurements. The sensor specialists (AM, FZK) manufacture three similar measurement devices for the calibration and validation measurements. The three systems will be jointly used by all partners. Additionally, the partner AM und FZK will provide all necessary hardware information and prototypes to the partner Centre Scientifique et Technique du Bâtiment (CSTB) and Technical University of Berlin (TUB) to insure a parallel development of the data processing and pattern recognition software (WP4).

The set-up of the general software layout for the **data processing** will be handled in WP4. The work package leader Technical University of Berlin (TUB) contributes many years of experience in pattern recognition methods and software development to calculate the perceived odour intensity based on multi gas sensor systems. The separation of software and hardware (WP3 and WP4) enables a company independent and extendable mathematical modelling of low-concentration odour mixtures as regards sets of critical odours for building and vehicle materials and products and their combination. The task of the computational data processing is to model the human odour perception of air samples. The method shall consider existing theories of perception and it shall link the measurements with psychophysical aspects of odour sensing. The response values of the sensor device will act as the stimuli and the software shall mimic the human perception and evaluation process. The core of the data processing method will be the “memory”, the calibration database. This database will have a major influence on the performance of the method, especially on the classification of the investigated odour sample. Therefore the data shall span the whole odour space which is developed in WP2. The principle components of this odour space will be the basic odours. The data of the database consists on a combination of sensor response values and air quality assessments according to the methods of WP2 for a specific odour sample and concentration. The database shall contain for each odour class data for different stimuli concentrations. The system shall be adaptive which requires the database to be expandable and open for new data of odour investigations. The calculation models will always refer to the current data and therefore the algorithms will be adjusted to the extended data set.

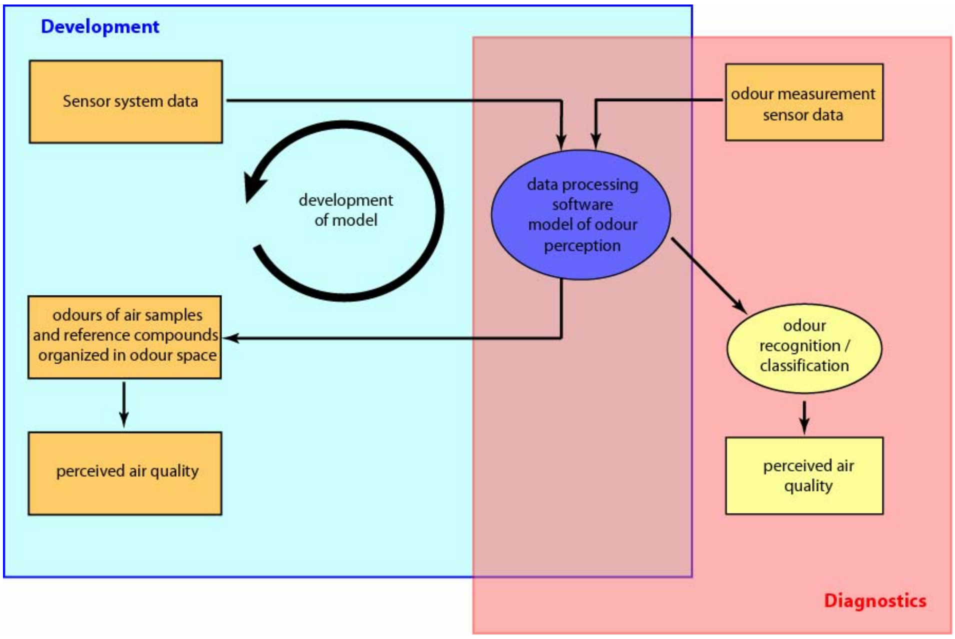

Figure 3: The software will handle the development/calibration process and it provide the final version for diagnostics of unknown odours

The unknown investigated odour shall be positioned inside the odour space which is a classification process for "odour recognition". Unknown odours shall be expressed by a combination of the basic odours. Once the sample is classified the quantitative and qualitative parameters of the perceived air quality shall be estimated. This step of the data processing uses the calibration data of the basic odours, the regression and calculation algorithms for this odours as well as weighing algorithms which consider the position of the unknown odour sample in the odour space. Data of odour samples other than the basic odours is needed to develop, test and optimize the weighing algorithms and to estimate the perceived air quality of different odours. Most of the odour space and combinations of basic odours should be covered. During the validation process of the software the new collected data will be included in the database which may further improve the performance of the data processing method.

The innovative sensor system has to identify all reference odours and their combination in indoor environments. Additionally, the system shall link the pattern recognition method for the identification process to the sensory panel results. Due to the multidimensional character of the problem a large effort is needed to calibrate the innovative sensor system. WP 5 handles the **collection of the calibration data**. It consists of simultaneous measurements of perceived air quality as well as the perceived odour intensity by a sensory panel and measurements with the sensor systems using the most relevant sensory methods selected in WP2 and the most promising innovative system/technical device selected in WP3. All data of WP5 iterates back to the software enhancement of WP4. The odours for calibration measurements of WP5 are produced in emission chambers. The air flow through the chambers is polluted by a series of building and cabin materials. The concentration of air pollutants will be varied within an indoor realistic range to ensure and test if the system works appropriate. The variation in concentration will be achieved by varying the material loading and the ventilation rate and by selecting high and low emitting materials. Measurements will be performed for individual materials in a laboratory setting, for combinations of materials in a full-scale setting, and in real buildings and vehicles. All data of WP5 is stored in a database. A final test run of the innovative sensor system after calibration will show the ability to predict perceived air quality and to characterise brand specific odours. During the sensor development WP5 will provide preliminary test set-ups of building materials in order to secure reasonable sensitivity and discrimination power of the novel sensor system.

The **dissemination** task of WP6 provides the publication of all project results. The main communication medium of the project is an advanced internet portal that offers a public and a non public information area (www.syspaq.eu). The public area publishes all non confidential findings of the project partner during the project life time. The non public area handles the data exchange of all project partners. All project related findings will be open for the public after the completion of the project. The project will provide a final workshop in order to discuss and disseminate all findings of the project. The consortium will prepare a brief project presentation in English which is written in a style which is accessible to non-specialists, avoiding technical language, mathematical formulae and acronyms as much as possible. Publication will be done via the NEST www page.

A more detailed information on the major work package interdependencies is given in figure 4.

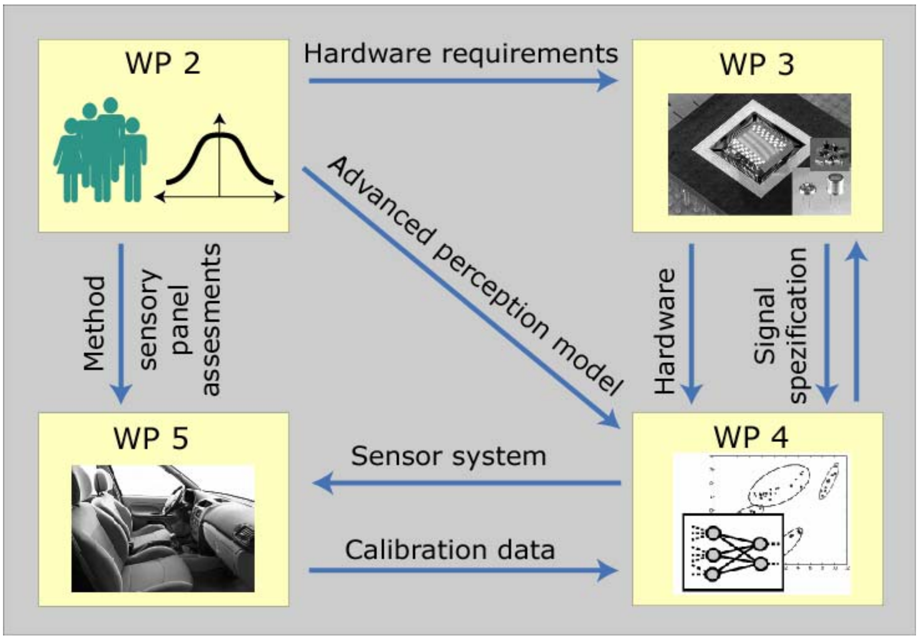

Figure 4: Major work-package interdependencies

ACKNOWLEDGEMENT

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Prediction of Volatile Organic Compound Concentrations Emitted from the Plywood Floor of the Ondol System

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SUMMARY

Recently in Korea, plywood has been widely used as a floor material for Korean floor heating systems (Ondol) instead of the conventional oiled floor paper. Volatile organic compounds (hereafter VOCs) are easily emitted from the plywood and adhesives that constitute the floor into the indoor environment due to the warm temperature of the floor. In this study, emission characteristics of VOCs from adhesive and paints within an Ondol floor are assessed using the Small Chamber Method in residences with plywood floors. From the experimental results, the concentration of VOCs is predicted as an exponential function, dependent upon various loading factor and air exchange rate conditions.

INTRODUCTION

The traditional Ondol heating system has been used in Korea since 400 B.C.[1] Before the 1950s, the conventional Ondol system was used, which heats the air by burning wood and straw in the furnace, sending the heated air below the floor, and exhausting the air through the chimney. [2] After the 1950s there was a shift to anthracite coal Ondol systems that had improved fuel and furnace facilities; afterwards, a pipe-laid system using a hot-water boiler was widely used to prevent CO gas emission and to improve heating efficiency. Despite the CO gas poisoning problem and lower burning efficiency, briquette was widely used until the late 1970s as a heat source in the Ondol system. In order to enhance combustion efficiency and constrain gas emission, the structure of the combustion chamber was improved, and the pipe was laid inside the floor structure, causing a fundamental change in the principal and structure of the conventional Ondol system. [3] During the 1980s, a briquette-fueled Ondol system was again adopted instead of the oil-fueled boiler due to the increasing petroleum price during the energy crisis period. After 1986, use of the oil-fueled boiler surpassed the briquette-fueled boiler due to economic growth and a decrease in oil price. Since 1987, the gas boiler has been used owing to the spread of Liquefied Natural Gas as a fuel for residences. Beginning in the 1990s, a thermal storage system which uses the relatively cheap nighttime surplus heat has received increasing attention due to the active utilization of nighttime electricity. [4]

Currently, the conventional hydraulic Ondol system utilizes the heat of hot water circulating through a pipe inside the floor structure to heat the floor and the home. Thus, the entire floor functions as the heating surface. [5]

Vegetable oil floor paper was used as the finishing material of the Ondol system until the 1970s, but since the 1980s, plywood began to be used as the floor material. [6] Currently, the Ondol floor is widely used in residential buildings instead of plywood. Since the Ondol floor consists of individual pieces of wood, adhesives are used to fixate the separate parts together and a variety of petrochemical paints are used as a surface treatment. These plywood adhesives and petrochemical paints emit significant amounts of hazardous volatile organic compounds and formaldehyde. [7] Although plywood is an important wood-based panel with many advantages, formaldehyde emission from plywood, especially from urea-formaldehyde bonded plywood, is a serious air pollution problem that causes adverse effects on human health, such as irritating the eyes, nose, and throat, and even causing cancer. [8]

By the early 2000s, research related to indoor air quality (IAQ) mainly focused on particulate matter, CO, CO_2, and natural and mechanical ventilation. Recently, provisions and experimental standards for IAQ were established, and measurement methods were actively developed for improving IAQ. In order to implement these new standards, construction companies replaced conventional finishing materials with environmentally sound materials. Sung et al. [9] carried out research on the emission rate of VOCs from individual products such as adhesives and paints, and Kim et al. [10] assessed the emission factor of VOCs using the small chamber method.

However, research by Kim et al. [11] did not include field measurement of the Ondol floor installed in real residences. They measured the concentration of formaldehyde using the decicator method in an artificial experimental chamber.

In this study, the emission factor of VOCs emitted from paints and adhesives from the Ondol floor was assessed using the small chamber method. In addition, a plywood Ondol floor was installed in a real residence constructed ten years ago, and concentrations of VOCs were measured. From the experimental results, prediction equations of VOC concentration decay were developed. Furthermore, prediction methods under various loading factor and air exchange rate conditions are also provided in this paper.

MATERIALS AND METHODS

Currently, plywood is widely used in Korea as the floor material of living and bed rooms in apartments and residential buildings.

In the case of a plywood floor, 0.5-mm thick patterned lumber is adhered to plywood. Typically, oak, beech, cherry, and maple are used as the patterned lumber. This finish wood is adhered to 7.2mm thick plywood of after being heated to 160 C. Figure 1 illustrates an overview of a plywood floor. The finished floor has a thickness of 7.7 mm and the of 75 x 900 mm. Table 1 presents the paints and adhesives used in the plywood floor and patterned lumber finish.

Painting consists of approximately three composite layers: primer, waterproofing, and the coating layer. Primer is a kind of finishing material at the innermost part of the painting process, while the waterproofing layer prevents water penetration. The coating layer determines the overall color of the floor.

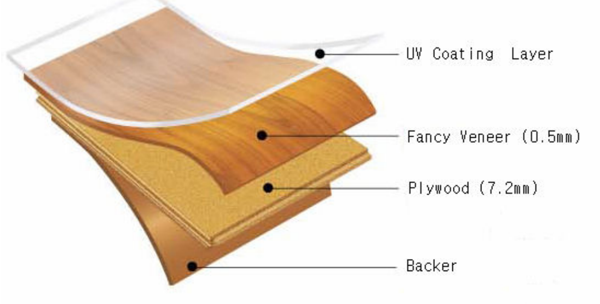

Figure 1. Composition of Ondol floor.

Table 1. Measured materials

	Floor materials
Measured materials	Primer
	Waterproofing layer
	Coating layer
	Primer(adhesive)
	Hardener(adhesive)
	Primer + hardener(adhesive)

Usually, two liquid-type adhesives are used, which are applied in two liquid layers. The first layer contains an adhesive component and the other controls the hardening rate of the liquid. Those two layers should be attached to each other for proper operation. The two-liquid type adhesive consists of primer and hardener in a one to one mixture ratio.
Figure 2 presents the experimental chamber, sample box, and fixing device inside the small chamber. Table 2 shows the experimental conditions such as chamber material, interior volume, sample area, air exchange rate, air flow rate, loading factor, temperature, and relative humidity.

Table 2. Experimental conditions of small chamber.

Specifications	Conditions
Chamber Material(CM)	Stainless Steel
Chamber Volume (CV)	0.02[m^3]
Sample Area(SA)	0.063×0.063 = 0.0039[m^2]
Air Exchange Rate(AER)	0.5 ACH
Air Flow	167[mℓ/min]
Loading Factor(LF)	0.4[m^2/m^3]
Temperature(T)	25±1.0[℃]
Humidity(H)	50±5[%]

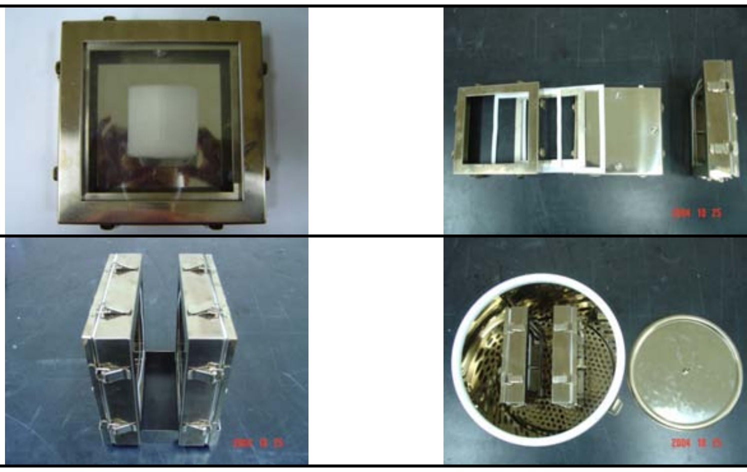

Figure 3. Experimental chamber, sample box and fixing device.

In order to determine the emission characteristics of VOCs emitted from the plywood of an Ondol floor, an Ondol floor was constructed inside the experimental room. Figure 3 (a) and figure 3(b) illustrate the plan (4.5m x 3m x 2.35m) and the construction process, respectively. The experimental room was installed on the rooftop of a residence constructed ten years ago. This residential building was selected since VOCs are not emitted into the indoor environment when a building is more than ten years old. Figure 3(c) illustrates a section of the Ondol system including the concrete slab of the living and bed rooms.
Measurement and analysis of VOCs was carried out based on NIOSH 1501[14] and 1500[15] using a GC/FID device.

(a)Plan (b)Construction process (c)Section

Figure 3. Experimental room.

RESULTS

Table 3 shows the time-dependent factors of VOC emission from the primer, waterproofing layer, coating layer, and adhesives used in the Ondol floor. Floor paints demonstrated an emission factor of 150～250 [㎍/㎡·h], except for benzene. However, benzene showed an emission factor five times greater than other substances ranging from 900～1300 [㎍/㎡·h], which did not decrease as time elapsed.

Table 3. VOCs emission factor from floor materials [㎍/㎡·h]

Elapsed time[h]		benzene	toluene	ethylbenzene	m,p-xylene	o-xylene	TVOC
Coating layer	1	1228	182	159	211	195	1977
	2	1123	167	206	197	190	1886
	6	1235	156	192	185	179	1950
Waterproofing layer	1	1103	181	236	224	206	1952
	2	1012	169	222	212	198	1815
	6	1297	161	215	205	197	2077
Primer	1	906	180	245	236	224	1793
	2	1295	175	242	230	223	2167
	6	1201	156	205	198	196	1958
Primer	1	2426	526	365	421	35405	211536
	2	2263	375	1860	2057	1318	11813
	6	2123	410	19615	325	8990	223798
Hardener	1	2130	459	351	391	371	153167
	2	2165	367	1345	1592	1116	12677
	6	2457	378	14621	324	6798	55636
Primer +Hardener	1	1401	248	185	216	11515	90721
	2	1619	253	772	1206	834	10626
	6	1236	181	4330	158	1909	16591

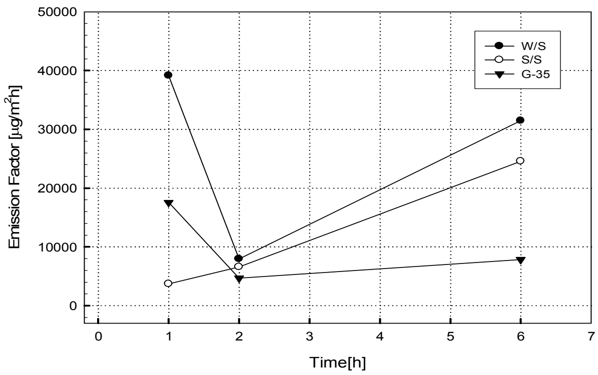

Figure 4. BTEX emission factor of adhesive.

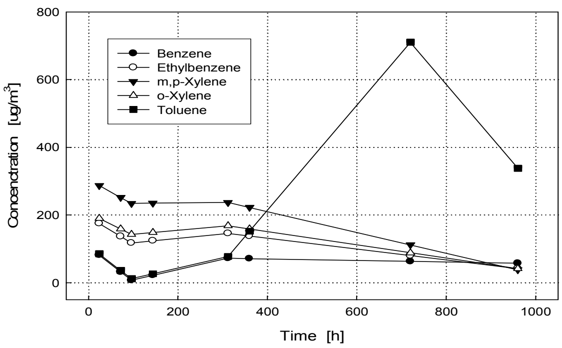

Figure 5. VOCs concentration after floor installation in experimental room.

Figure 4 illustrates the total BTEX emission factor. As shown in this figure, adhesives have a higher emission factor than paints. In addition, VOCs are continuously emitted with time, suggesting that adverse indoor air pollution will be caused more by adhesives than by paints. W/S stands for wood-silla used for the filling of wood mesh and s/s is a paint used for smoothing surfaces. G-35 is a finishing material in the upper most part of the surface.
Figure 5 illustrates the time-dependent variation of indoor VOC concentrations after the installation of an Ondol floor in a real residence. BTEX concentration decreased dramatically during the initial 100 hours after installation, and then decreased gradually.
Benzene showed a relatively constant concentration throughout the measuring period. Ethylbenzene and m,p,o-xylene showed different concentrations, but their decrease patterns were similar.
Although Toluene showed results similar to other substances at first, the concentration suddenly increased and then decreased. This could be due to VOCs being emitted from the floor material at a relatively early time, while petrochemical adhesives dominate the indoor concentration variation as time progresses.
After the floor installation, VOCs are gradually emitted into indoor environment as the petrochemical adhesives harden. This pattern can also be confirmed by the VOC emission characteristics assessment using the small chamber method, which is presented in figure 4. As can be seen in figure 4, the VOC emission factor of adhesives decreases temporarily but increases again as time elapses.

VOCs concentration prediction of the experimental room

A time-dependent concentration decrease of ethylbenzene after the installation of a plywood Ondol floor in a real residence is illustrated in figure 6. The ordinate used is a logarithmic scale and the profile can be expressed as a straight line in the form of an exponential function. Ethylbenzene concentration as a function of time can be expressed by the following equation.

$$C_{(ti)} = 159.26 \times e^{(-0.0010 \times t)} \tag{1}$$

In the expression above, R^2 showed a relatively high value of 0.72 and t is the elapsed time. Figure 7 shows the m,p-xylene concentration prediction. Indoor m,p-xylene concentration can be predicted using an expression similar to that for ethylbenzene.

$$C_{(ti)} = 292.7 \times e^{(-0.0013 \times t)} \tag{2}$$

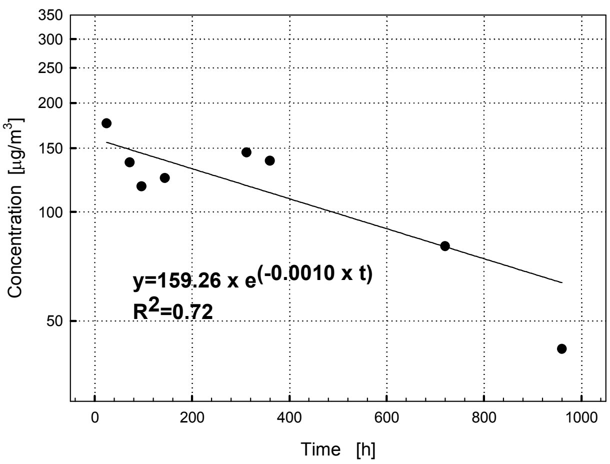

Figure 6. Concentraon predicton of Ethylbenzene.

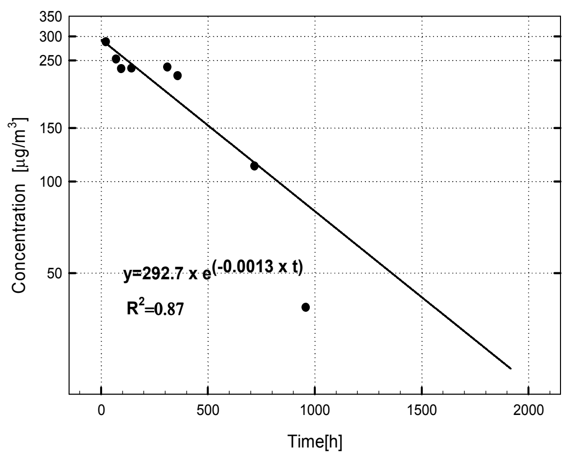

Figure 7. Concentration prediction of m,p-xylene.

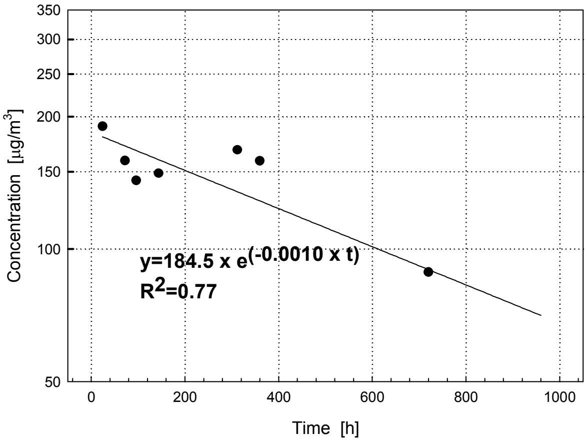

Figure 8. Concentation prediction of o-xylene.

Table 4. Result of regression analysis for volatile organic compounds.

substances	Prediction model	R^2
ethylbenzene	$C_{(ti)} = 159.26 \times e^{(-0.0010}$	0.72
m,p-xylene	$C_{(ti)} = 292.7 \times e^{(-0.0013\times t}$	0.87
o-xylene	$C_{(ti)} = 184.5 \times e^{(-0.0010\times t)}$	0.77
xylene	$C_{(ti)} = 476.9 \times e^{(-0.0012\times t}$	0.98

The R^2 value for the above equation showed the high value of 0.87, indicating that indoor m,p-xylene concentration can be properly predicted when applied in a real residence. Figure 8 shows the o-xylene concentration profile with an R^2 value of 0.77, indicating that it can also be used in real residences.

$$C_{(ti)} = 184.5 \times e^{(-0.0010\times t)} \tag{3}$$

Indoor VOC concentrations of each substance can be predicted as described in Table 4. Figure 9 shows indoor ethylbenzene concentration decay under the loading factor of 0.465 [㎡/㎥] and air exchange rate of 0.5-1.0 ACH. As the air exchange rate increases, the ethylbenzene concentration decreases more rapidly. Figure 10 shows the m,p-xylene concentration decay under different air exchange rate conditions, and figure 11 illustrates the time-dependent concentration decay of o-xylene. Figure 12 shows the time-dependent concentration decay under the loading factor of 0.25-0.60 and the air exchange rate of 0.7 ACH. As the loading factor increases, the indoor ethylbenzene concentration decreases more slowly. Figure 13 shows the m,p-xylene concentration decay as a function of loading factor, and figure 14 shows the time-dependent concentration decay of o-xylene under different loading factor conditions.

Figure 15 shows the time-dependent concentration decay of VOCs under different air exchange rates and loading factors. The concentrations decay rates of ethylbenzene and o-xylene were equal, while m,p-xylene showed a higher concentration decay rate than the other substances.

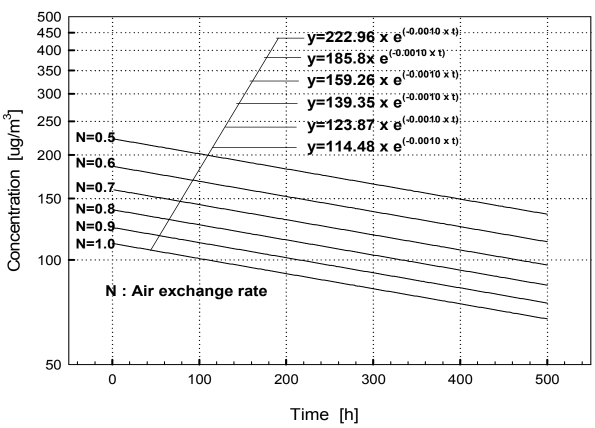

Figure 9. Ethylbenzene concentraion decay prediction under different air exchange rate.

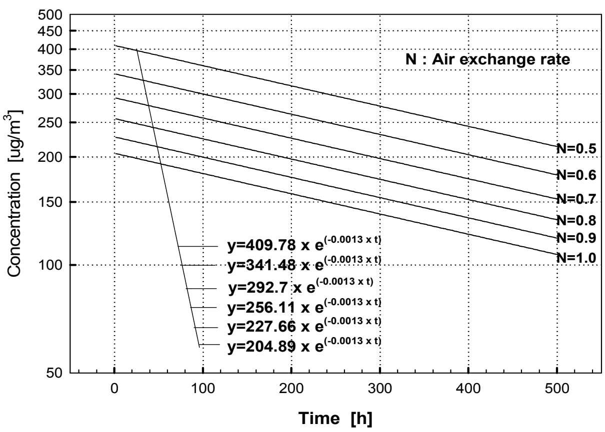

Figure 10. m,p-xylene concentration decay prediction under different air exchange rate.

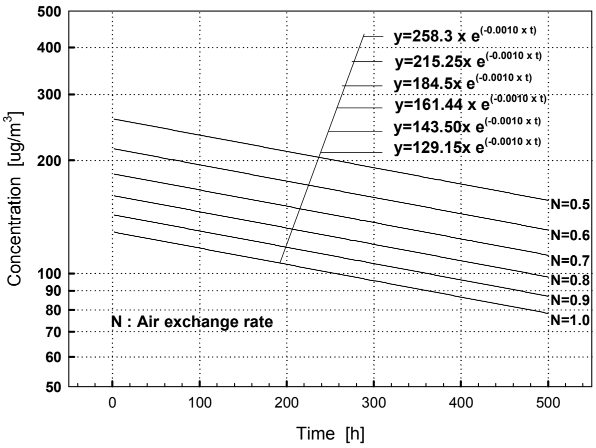

Figure 11. o-xylene Concentraion decay Prediction under different air exchange rate.

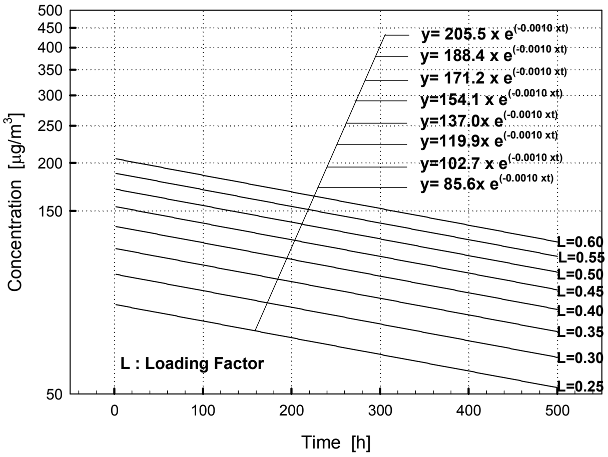

Figure 12. Ethylbenzene concentration decay prediction under different loading factors.

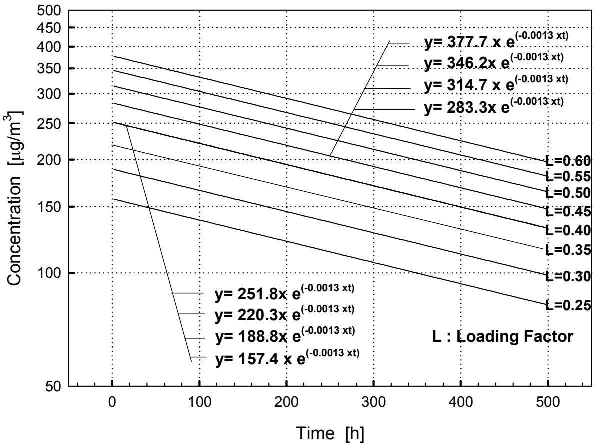

Figure 13. m.p-xylene concentration decay prediction under different loading factors.

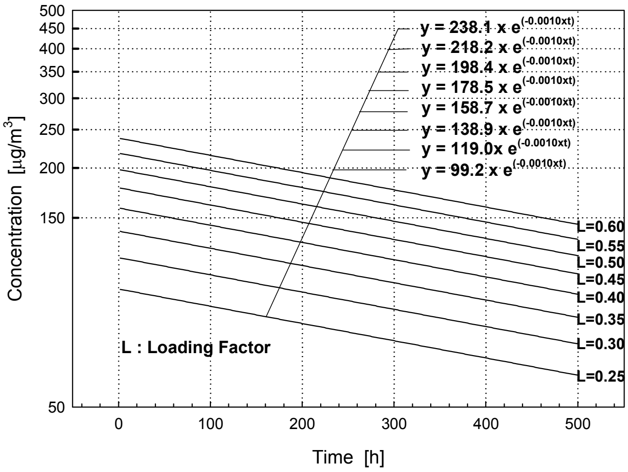

Figure 14. o-xylene concentration decay prediction under different loading factors.

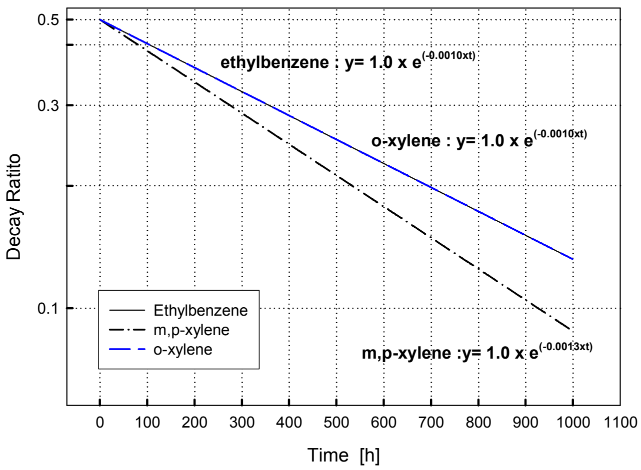

Figure 15. Concentration decay under different air exchange rate and loading factors.

CONCLUSIONS

In this study, indoor VOC concentrations emitted from an Ondol floor and individual floor materials were measured after the installation of an Ondol floor in a real residence. Based on the experimental results, prediction equations for indoor VOC concentrations were developed under various loading factor and air exchange rate conditions.
The following conclusions were made.
1. VOC emission from the Ondol floor made of plywood was dominated by adhesives used in the construction process rather than by paints.
2. Indoor VOC concentrations in the experimental room can be expressed linearly using an exponential function when the ordinate is converted into logarithmic scale.
3. When the Ondol floor made of plywood was installed, indoor VOC concentrations increased as the loading factor increased.
4. As the air exchange rate increased, the indoor VOC concentration decay rate increased more rapidly.

ACKNOWLEDGEMENT
This work was supported by the POSCO Engineering & Construction Co.,Ltd.

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A measurement on chemicals emitted from computers and printers using test chamber method

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SUMMARY

The printers emit the chemical compound when its temperature rises during normal operations. And it primarily influenced the chemical concentrations such as formaldehyde and VOCs (Volatile Organic Compounds) in a living space. The measurements on the various types of prints from different manufacturers such as laser jet type, inkjet type were performed in emission test chambers of 1m3 and 5m3 with ventilation rate of 1.0 ACH(air change rate) supplying low-polluting clean air. During the test, chemicals emitted per PCs or Printers were analyzed on the two modes of stand-by and operation at 25°C and 50% RH. It is found that PCs and printers are a significant source of chemical emissions in small indoor environments, even with adequate ventilation. The chemicals may affect a potential health hazard for computer users.

INTRODUCTION

Poor indoor air quality increasing chemicals has been focused as related in sick building symptoms such as respiratory illness, itchy, eye irritation, and headaches. Exposure to pollutants in the air may cause health risk. Depending on the health effect of particular chemicals, guidelines of concentration level were established by Korean Government for avoiding the significant health effect generally associated with given concentration levels. There are many pollution sources in building and they emit hundreds of different chemicals. In indoor environment the chemicals are emitted from various kinds of products including building materials and furniture. Over the several years, emission test chamber method has been developed and several databases have been established for the information of chemicals emitted from building material only.

Recently it is common for the design strategies to reduce the chemicals by selecting low emission materials and products. It would be practical data to measure the pollutants emitted from electrical equipments. Computer equipment and printers are focused on as one of the serious pollutant sources indoor. Some approaches have been performed to figure out the potentials of chemicals emitted from office equipment including computer, monitor and printer. Computers are a significant source of allergenic emissions in small indoor space like offices and individual rooms. Even with adequate ventilation, the compound may be a potential health hazard for computer users. The Purpose of this study is to clarify the emission characteristics and to determine the emission rates of chemicals on the computer set and printer.

METHODS

It is known that the perceived indoor air quality caused by VOCs emitted from new building materials generally decrease over time in concerns. Computer set including a monitor and a printer can emit the VOCs during operation continuously. Some questions often arise that how much will the VOCs emit and how the emission rate change with different mode of operation.

The measurements of VOCs and HCHO emission from the computer and printer were conducted using the test chamber method. The chemicals were measured the variation of the emission rates with two different stages of usages. One is for the standby mode, which is not operating the PC but emitted the chemicals continuously, and the other is for the condition of operating mode, of which temperature is goes up with chemicals emitted intensively during operation. Several cases were tested how the profiles of emissions are varied with products from different manufacturers.

Test Chamber and Equipments

Two systems of emission test chamber were built for measuring emission rates from the computers and printers. One is prepared $1m^3$(W 1m x L 1m x H 1m) in scale of air volume for measuring relatively small products like computer sets, which is combined with double chamber system as outer and inner chamber. The buffer zone is realized between two chambers for precisely controlling environmental parameters of temperature, humidity and pressure differences. The other is a large scaled test chamber with volume of 5 m^3 for large products, which is constructed as single chamber type and using emission test for printers. Both of two inside chambers are constructed with stainless steel for confirming not emitting any chemicals themselves.

Environmental conditions of the chamber, such as temperature, relative humidity, air change rate and air velocity could be precisely controlled at setting points. In this measurements, they were controlled to the temperature of 25 ^oC + 0.5 ^oC, relative humidity of 50+5% RH, ventilation rate of 1.0 + 0.1 air exchange per hour and air velocity of 0.2 – 0.3 m/sec. The tests were performed for measurement of formaldehyde and VOCs from the products under designed conditions to simulate the situations of products use indoor.

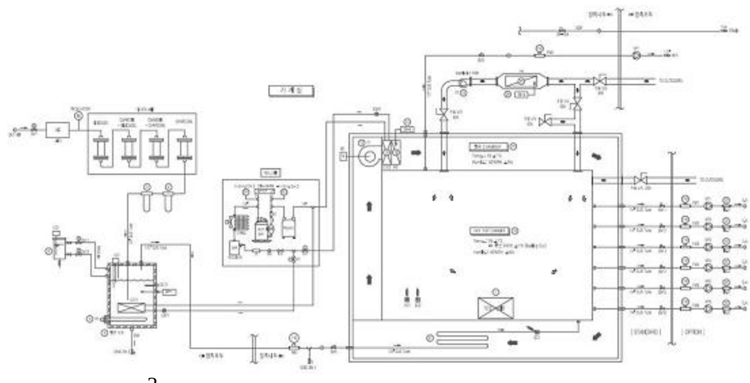


Fig. 1 Diagram of the 1 m^3 test chamber

Test Products

Desktop computer sets with keyboard and monitor were selected for the test of formaldehyde and VOCs emitted. Three laser printer sets from different manufacturers and one ink jet printer were selected for the measurements. They are all new products which were ordered through a retail distributor directly to ensure the brand new and ordinary shipping procedures. The outlines of tested samples are shown in table 1.

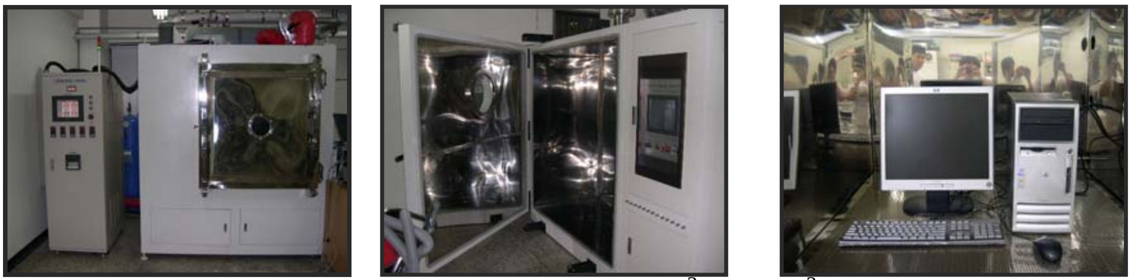


Figure 2 figures of test chamber facilities both of 1 m^3 and 5 m^3

Table 1. Test samples of measurement for determining emission rates

Test Sample	Composite of the Samples	Remarks
Computer Set A	Computer + Keyboard + TFT Monitor	Desktop Manufacturer 1
Computer Set B	Computer + Keyboard + TFT Monitor	Desktop Manufacturer 2
Computer Set C	Lab-top computer + TFT Monitor	Lab-top Manufacturer 1
Printer A	Laser / Black and white	Manufacturer 2
Printer B	Laser / Color	Manufacturer 2
Printer C	Laser / Color	Manufacturer 1
Printer D	Ink jet / Color	Manufacturer 1

Protocol for Chamber Test

Preparation of the test chamber Before placing the test product in the chamber, background air samples were collected from the chamber to be assured the test condition. And adjustment was performed for the air flow, temperature, and relative humidity to assure the standard test condition. And then background samples were collected for analysis of formaldehyde and VOCs. After every emission test, the camber were cleaned with distilled water and baked out at 100 ^{o}C with purified air. After confirming the background level should not exceed the standard background levels, the test product was to set placing and test was started with test procedure.

Sampling and analysis method Sample products are tested for emissions of pollutants using stringent environmental chamber test protocols. Emission concentrations were determined by placing the test objects in the chamber under specified test condition, then measuring chamber concentration of HCHO and VOCs with selected time intervals. Product emission factors were calculated from the chamber air samplings. A typical test period is 8 hour with measurement time at 2, 4, 6, and 8 hour. Some products like printers may require a specified procedure during test period to accurately define the emission profile.

Sampling of Volatile Organic Compounds Sampling VOCs were collected on sorbent cartridges (TA tubes). VOCs trapped on the cartridges were thermally desorbed then analyzed by GC/MS. Testing results of these analyses were used to estimate both of TVOCs and individual VOC concentration. Detailed procedures for sampling and analyzing the chemicals with chamber test method and TA tube cartridges described in ISO 16000 – 6, 7, 9

Sampling of Formaldehyde Air samples for formaldehyde were collected on 2,4-dinitrophenylhydrazine(DNPH) coated silica gel cartridges with ozone scrubbers. The DNPH derivatives on the cartridges were eluted with acetonitrile and analyzed by high performance liquid chromatography(HPLC) with ultraviolet(UV) detection.

Air Samples from chamber Calibrated air pumps located outside the chamber were used to pull chamber air through sampling cartridges mounted in the chamber ports (couples of two sampling ports) installed both as one is for the center of the chamber and the other is for the outlet of the exhaust duct.

Calculation of emission factor An emission factor is the amount of chemical that is emitted from a product at selected point in time. It is normalized to a particular size or unit of the product. For most building material, the emission factor is usually measured for exposed unit area of the product as square meter. However in this study, a product test reports the emission factor explain test results relative to a complete unit (µg/unit*hr). If the product is not constantly emitting pollutants, the emissions are decreasing gradually over time, the emission factor is a qualitative estimate of emissions release at a particular point in time. The emission factors of chemicals were calculated by the following equation.

$$EFa[mg/h \cdot unit] = \frac{(C_t - C_{tb,t}) \times Q}{1\,unit} = \frac{(C_t - C_{tb,t}) \times nV}{1\,unit}$$

C t : Chemical concentration at time t (㎎/㎥)
Ctb,t : blank concentration at time t (㎎/㎥)
n : ventilation rate (ACH/h)
Q: Airflow rate in chamber (㎥/h)
V : Air volume of test chamber (㎥)

RESULTS

Ventilation Rate in test chamber

In order to evaluate the performance of ventilation in test chamber, air change rate, ventilation effectiveness and age of air were measured at different location using concentration decay method with SF6 as tracer gas. Sampling tracer gases at the center point, and upper zone and lower zone in the chamber were performed with controlled airflow rate at 0.5 ACH/h, 1.0 ACH/h and 2.0 ACH/h. Table 2 shows the profiles of ventilation rate and age of air at the measuring points in 5 m^3 chamber. Air change rates were pretty much similar to the setting values at the center point for all cases. It shows the air change rates were controlled within 10% at every location in the chamber

Computer set and Chemical emissions

Some chemicals composed electronic equipment that have recently received attention with chemical component of the plastic case and printed circuit board. In order to fully understand how a product is going to contribute to indoor concentrations of pollutants, it is necessary to know a product's emission rate profile, or how the product's emission rates change over time. During the test period, multiple pollutant measurements are performed which are then used to calculate the product's emission profile over time. Fig. 4 shows TVOC profiles of emission from the computer set over time measured at center and outlet in the chamber with its operation. TVOC concentration at the center point measured much higher than the outlet point. TVOC concentration indicated the peak after 4 hr from the start of measurement, and decreased gradually over time. Some of individual VOCs were detected. Emissions of toluene, ethylbenzene and xylene in VOCs as caused for health risk compound were emitted with its operation as shown in table 4.

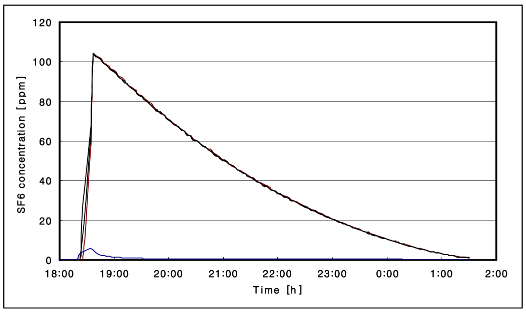


Figure 3. Profiles of SF6 concentration

Table 2. Ventilation rates in 5㎥ chamber

Air Flow	Sampling points	ACH/h	Ventilation Efficiency[%]	Age of Air [min]
Air Change Rate 0.5 ACH/h 41.67 [ℓ/min]	Upper Zone	0.48	94.0	127.6
	Center	0.49	99.0	122.4
	Lower Zone	0.46	92.0	130.0
Air Change Rate 1.0 ACH/h 83.33 [ℓ/min]	Upper Zone	0.98	98.0	61.2
	Center	0.99	99.0	60.6
	Lower Zone	0.97	97.1	61.8
Air Change Rate 2.0 ACH/h 166.7 [ℓ/min]	Upper Zone	1.93	96.5	31.1
	Center	1.95	97.7	30.7
	Lower Zone	1.89	94.5	31.7

Fig. 5 and Fig. 6 shows the profiles of TVOC and formaldehyde concentration emitted from tested computer sets. Emission rates of chemicals from desktop computer indicated much more higher than those of lab-top computer set. It may be possible to assume that the emission rates could be higher from the computer due to the bigger size and more consumption of electricity emitted much VOCs. Emissions of HCHO concentration ranged from 2 $\mu g/m^3$ to 44 $\mu g/m^3$ were not much higher relatively.

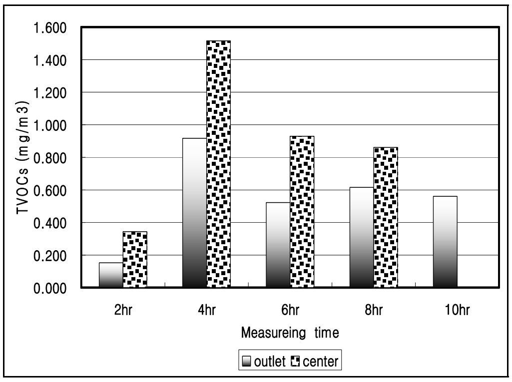


Figure 4. TVOC profiles of chemicals

Table 3. Individual VOC emitted from computer set

Compound Name	2 hour		4 hour		6 hour		8 hour		10 hour
	Outlet	Center	Outlet	Center	Outlet	Center	Outlet	Center	Outlet
Toluene	0.012	0.031	0.035	0.059	0.022	0.040	0.031	0.046	0.032
Ethylbenzene	0.006	0.014	0.030	0.050	0.017	0.030	0.022	0.032	0.031
p-Xylene	0.010	0.022	0.044	0.075	0.022	0.042	0.031	0.044	0.031
o-Xylene	0.003	0.008	0.017	0.029	0.008	0.016	0.010	0.015	0.011
Benzene,1,3,5-trimethyl-	0.001	0.001	0.004	0.006	0.001	0.003	0.002	0.003	0.002
Benzene,1,2,4-trimethyl-	0.001	0.003	0.008	0.014	0.003	0.007	0.005	0.000	0.004
Unidentified	0.118	0.265	0.778	1.280	0.450	0.790	0.515	0.720	0.449
TVOC[㎎/㎥]	0.152	0.344	0.916	1.514	0.522	0.930	0.615	0.861	0.560

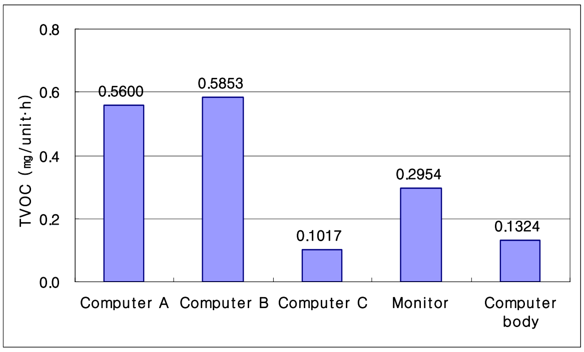


Figure 5. TVOC profiles from PC sets

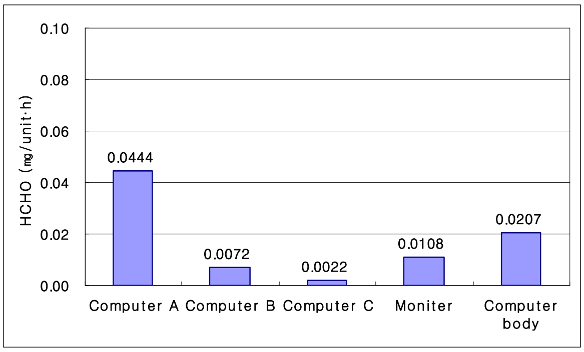


Figure 6. Formaldehyde profiles from PC sets

Printers and Chemical emissions

The chemical emissions printers can result from plastic construction materials, circuit boards, the inks and toners, papers, coatings on transparencies. Many of these chemical emission profiles show decreasing with usage of the product. But emissions from the printers as the harmful pollutants is not decreased much during the operation, because the VOCs emissions from the laser printer caused by heating up the drum and toner at 160°C for compress toner on paper with chemical components. However, the result in the mechanical operation of printing procedure emitted harmful chemicals to indoor environment. During the measurement of printers, chamber was controlled at 25 °C, 50% RH and 1.0 ACH/h. Temperature and humidity of the chamber goes up to 48 °C and 57% RH due to generate heat and moisture from the printer procedure.

Fig. 7 shows the profiles of chemical emission from the printers. The emission rates of the TVOC concentration ranged from 0.31 mg/m^3 to 8.00 mg/m^3 at the standby mode. Some printers emit still high concentration of chemicals during standby mode and ready mode without any operation. On the operation, the emission rate of chemicals goes up to the 10.12 mg/m^3. Even though a similar printing speed and quality, Printer B emits 5 times more chemicals comparing to those of printer A. Laser printer C of B&W emits TVOC to 3.12 mg/m^3 during operation and still highly emits the chemicals on the standby mode. Inkjet printer also emits relatively high during operation. Table 8 indicates the individual VOCs from the printers. Individual organic compound of benzene, toluene, ethyl-benzene, xylene, styrene were dominant. It could be possible that laser printer has been associated with building-related symptoms and health effects due to the release of chemical gases during operation.

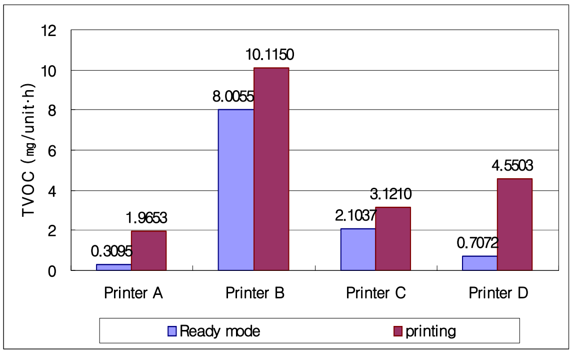


Figure 7. TVOC profiles from printers

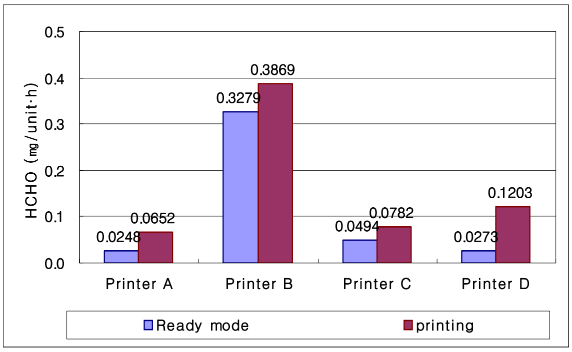
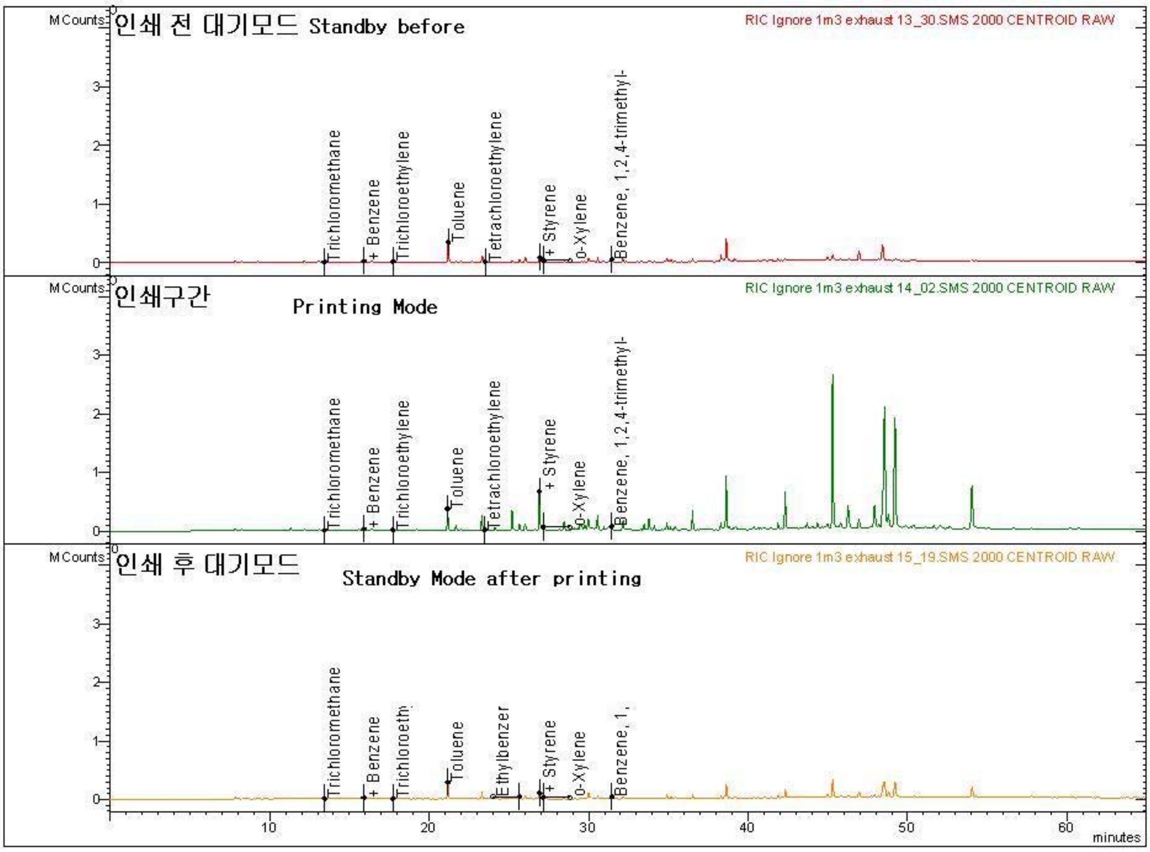


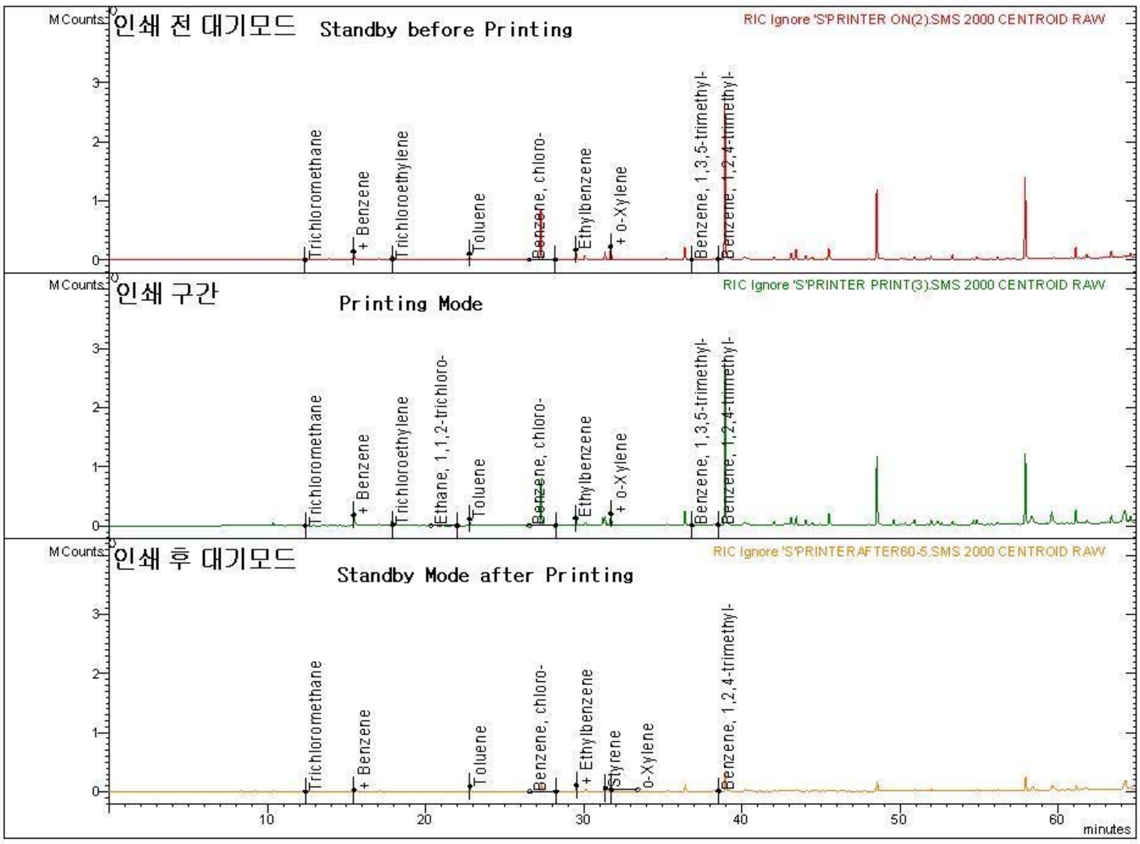
Figure 8. Formaldehyde profiles from printers

Table 4. Individual VOCs profiles emitted from different types of printers

	Laser Printer A(color)		Laser Printer B(color)		Laser Printer C(B & W)		Inkjet Printer	
	Standby(1hr)	On printing	Standby(1hr)	On printing	Standby(1hr)	On printing	Standby(1hr)	On printing
Benzene	0.0002	0.2210	0.0260	0.0460	0.0010	0.0015	0.0004	0.0041
Toluene	0.0147	0.0442	0.3348	0.4228	0.0519	0.0494	0.0256	0.1854
Ethyl-benzene	0.0022	0.0056	0.0143	0.0163	0.0033	0.0056	0.0034	0.0271
p-Xylene	0.0030	0.0087	0.0230	0.0253	0.0059	0.0069	0.0043	0.0297
Styrene	0.0017	0.0076	0.0932	0.0997	0.0030	0.0255	0.0045	0.0624
o-Xylene	0.0010	0.0028	0.0055	0.0068	0.0013	0.0025	0.0013	0.0094
Unknown	0.2911	1.5676	7.4829	9.4700	2.0343	3.0263	8.4960	4.4414



a) Chemicals from Printer A



b) Chemicals from Printer B

Figure 9. Examples of Gas Chromatogram for the different printers.

DISCUSSION

Though none of these chemicals from computers and printers is likely to be measured in quantities that would exceed any established workplace health and safety limits, the unknown factor is how some of these chemicals may interact with each other to cause problems many years in the future. In addition, there are no established health and safety limits for exposure to chemicals by electronic devices exactly right now.

Computer with laser printer has been associated with building-related symptoms and health effects due to the release of chemicals during operation. These emissions can result from the inks and toners, papers, plastic construction materials, circuit boards. Many of these chemicals will decrease with usage of the product such as computers. Laser printers in particular have been associated with headaches, mucous irritation and eyes and nose irritation As a result, computers should be placed in other areas of the home in order to minimize health risk in the bedroom.

ACKNOWLEDGEMENT

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Measurement of Perceived Odor Intensity Using Gas-Sensor Systems

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SUMMARY

In the present study Multi gas-sensor systems (MGSS) are used to evaluate the odor intensity as it is perceived by humans. These systems measure volatile compounds in a holistic manner and present a signal pattern of the air sample. The model for the determination of the odor intensity consists of two major parts: discrimination into odor classes and subsequent quantification of the intensity. The discrimination is achieved by multivariate statistical methods or neural networks. The correlation to the intensity is done by odor class specific regression methods. The development of the data processing is based on calibration data consisting of simultaneously obtained MGSS responses and assessments of the perceived intensity by a trained human panel using a comparative scale. Different building materials were investigated. The study shows that MGSS have the potential to be used as a measurement device for perceived intensity using a data processing method on basis of an odor classification.

INTRODUCTION

Air quality in the indoor environment impacts the health and the wellbeing of the occupants. Studies showed that bad air quality has also a negative effect on the productivity of office workers [1]. Therefore there is a growing demand for technical devices to measure, maintain and control the indoor air quality. The air quality is determined by the volatile compounds in the air and mainly the provoked odor impression is the cause of discomfort. But so far no technical measurement technique for volatile compounds is able to measure the impact of the substances on the odor. Analytical chemical methods are able to detect the compounds in the air and their concentrations but the transfer to the human perception cannot be done and is still unknown. The concentrations of the interesting compounds are often below the detection thresholds of the measurement devices.
By now, the odor can only be assessed by human panels. Depending on the question formulation different aspects of the odor perception are evaluated, such as the acceptance or intensity of the odor. These methods are time consuming and cost-intensive due to the expenses for the panelists and in addition human panels cannot be used for continuous measurements for monitoring and control.

Since the analytical methods aren't able to evaluate the perception of odors another approach is to measure gaseous substances in a holistic rather than analytical way. In recent years multi gas-sensor systems (MGSS) have been developed to evaluate mixtures of volatile compounds and odors. The fundamental idea behind these systems is to mimic the human sense of smell. Analogues to the human odor perception the systems evaluate the sample air as a whole. The sensor systems were often called electronic or artificial noses. Though the systems apply the principle of odor perception the performance (sensitivity and ability to recognize and

categorize odors) falls far behind the human sense of smell. The comparison with the human nose raises expectations which these systems cannot fulfill. Nevertheless these systems may be able to measure the odor intensity as it is perceived by a human panel for typical room air odors with adequate accuracy.
The strength of MGSS is the recognition of differences in a mixture of gaseous compounds and deviations from a defined composition. The systems respond within seconds to a few minutes to changes in the mixture and can be used for online monitoring. Therefore the main areas of interest are the control and monitoring of product processes and quality control. They are widely used for applications and research in the food and nutrition industry as well as in the perfume and cosmetics industry [2]. In the last years in more and more areas of application the systems gain on interest and are discovered as useful instrument. In the field of indoor environment only few investigations have been done to assess the air quality [3]. The challenge for measuring the indoor air quality is the low concentrations of the volatile compounds and compared to the concentration the high intensity impression they invoke.

In the current study assessments of the perceived intensity of a trained panel were compared with measurements of a MGSS. On basis of these measurements it is tried to build up a model to correlate the measurements to the human perception of odor intensity.

MEASUREMENT APPROACH

Technical measurement devices are inherently not capable to measure the odor. To evaluate the odor intensity with multi gas-sensor systems the measured sensor responses must be correlated to human odor perception. A data processing model has to simulate the perception process in a simplified and abstract manner. In this study the focus is set on the odor intensity, which is a dimension of odor perception which is closely related to concentration and composition of the compounds and hence the measured sensor signals. As experience of odors made during the whole life is a main component of the odor perception and evaluation, a knowledge database consisting on measurements and associated odor assessments of a human panel will be the backbone of the data processing model.

Due to some preliminary measurements it was obvious, that it is very unlikely that one single mathematical algorithm is able to establish a relationship to the perceived intensity. Instead a theoretical data processing model was formulated which consists of two steps: a classification of the odors followed by a quantification using an odor class specific transfer function. The theoretical structure of the two-step data processing is illustrated in figure 1.

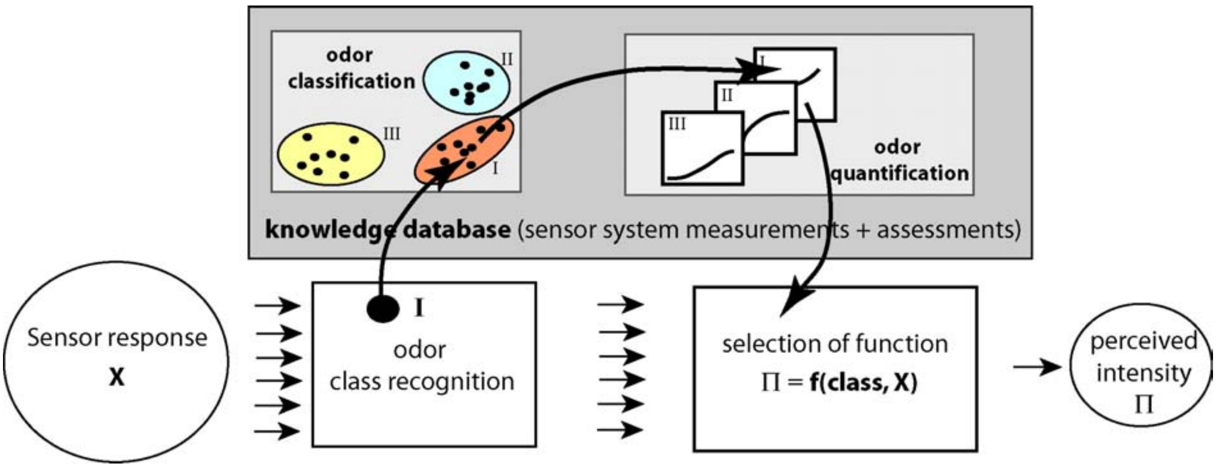


Figure 1. Schematic representation of the data processing model

The crucial part of the model is the knowledge database. It contains measurements of different mixtures and single compounds of emissions found indoors in several varying odor concentrations covering an intensity range. For each measurement a couple of sensor responses and panel assessments data were stored in the database. In the first step the measurements are combined and classified into classes with similar odor and sensor response data. For each determined odor class in a second step a class specific transfer algorithm to quantify the odor intensity is calculated. When measuring an unknown sample with the sensor system, the measurement is allocated to an odor class. The associated transfer algorithm for this odor class is selected and used to calculate the perceived intensity.

METHODS

Measurements of emissions from building materials and furnishing were used to construct the database. The objective was to investigate if the data processing model is principally applicable for the estimation of the perceived odor intensity with MGSS. Emissions from building materials and furnishing are main contributors to the indoor air contamination. They were chosen because the emissions can be easily and reliable produced and controlled in a laboratory environment.

Figure 2 shows the experimental setup to measure the odor emissions of building materials. The samples of building materials are placed in CLIMPAQ emission chambers [4]. These chambers were slightly advanced to optimize the air flow pattern and to connect them to an existing air ventilation system.

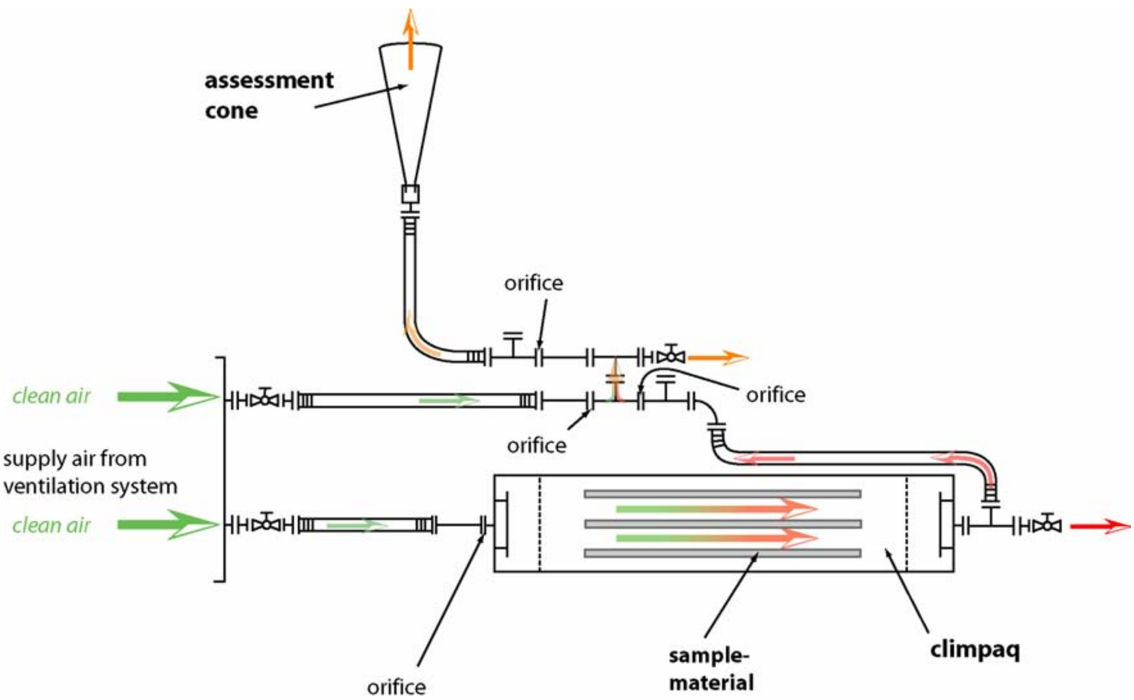


Figure 1. Test facility to measure the odor emissions of building materials

Clean air from the supply system flows through the CLIMPAQ and is loaded with the emissions of the building material sample. The sample air can be diluted by adding clean supply air before it is presented at a glass cone for the assessments. The dilution ratios can be adjusted continuously from pure air to 100% sample air from the CLIMPAQ. The tubes were built of stainless steel to keep the odor pollution emitting of the system small. The air flows can be adjusted by ball valves and can be measured using orifices of teflon. During the measurements the airflow through the emission chamber was kept constant at 1.0 l/s. After dilution with clean air, the polluted air can be extracted to set the air flow at the cone to a constant value of 0.9 l/s to ensure that variations of the air flow during the evaluation has no influence on the results.

The perceived intensity was assessed by a trained panel of 8 to 12 subjects. The odor intensity of the sample was ranked in a reference scale made up by acetone samples of 5 different odor intensities. The perceived intensity was assessed in the unit "pi" (perceived intensity) which was introduced by Müller in 2004 [5]. A pi of 0 is defined to an acetone concentration of 20 mg/m³ which corresponds approximately to the odor threshold of acetone. The scale was constructed linear in respect of the concentration. A step of 1 pi is equivalent to a concentration step of 20 mg/m³ of the reference substance acetone.

The assessments of the panel for a measurement series with varying dilution rate, the so called exposure response curves, can be approximated by a logarithmic function which was shown by Böttcher in investigations on the perception of odors [6]. Since the concentrations of the emitted substances of the building materials in the current investigation are not known the concentration dependency can be expressed by the airflow specific area load [7]. Figure 3 shows the dilution characteristics of two different building materials, a carpet and a painted wallpaper. The approximation was done using the mean values of the assessments of the panel members. In the model for data processing the calculated perceived intensities by the dilution characteristic were used as the intensity reference values instead of the assessment means in order to minimize the uncertainties which are inherent within the subjective sensory assessments.

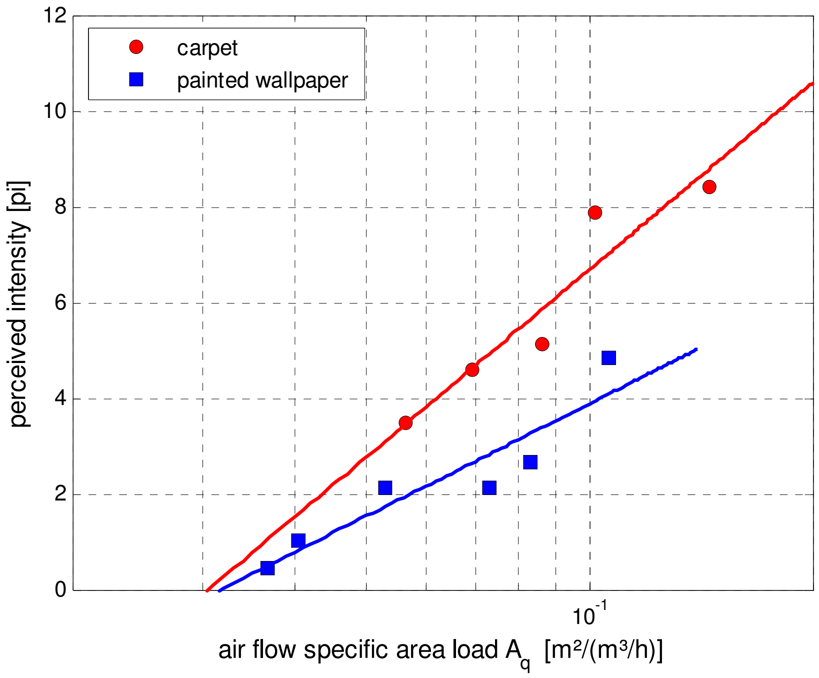


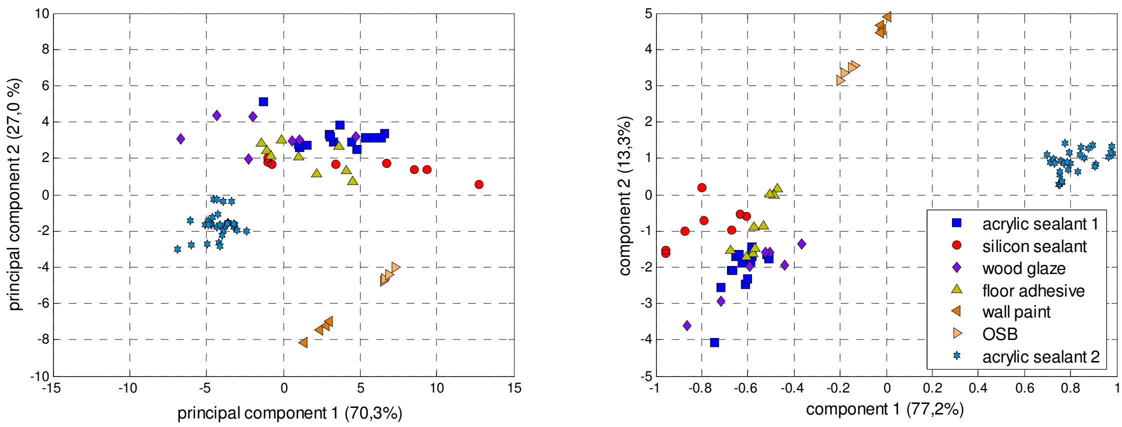
Figure 3. Panel assessments of the perceived intensity of 2 building materials

As a multi gas-sensor system the commercially available system KAMINA from the Forschungszentrum Karlsruhe is used in this study. It operates with 38 tin oxide sensors, which are placed on one single chip. The metal oxide surface is coated with a sensitive layer of SiO_2. The different sensitivity and selectivity of the sensor elements are achieved by a gradient of the sensitive layer and a temperature gradient from 250-300°C across the sensor-chip. The transport of the sample air to the sensors is done by a fan in the housing. The sensor sensitivity and selectivity and combination of sensors have a major impact on the performance of the intensity estimation. But it is not part of this study to investigate and optimize the sensor device.

For the measurement of the emissions of building material the KAMINA was placed on top of the glass cone according to location where also the panel assessments took place. The measurements were conducted directly after the assessments of the panel. The duration of the measurement was about 5 minutes until the stationary sensor signals were achieved. The stationary sensor responses further were used for the data processing.

RESULTS

For the classification of the measurements of seven building materials statistical multivariate methods can be applied. One method is the principal component analysis (PCA). This method reduces the relevant parameters and maximizes the variance between the measurements. The PCA describes a data transformation where the eigenvectors of the covariance matrix are the new coordinates. The advantage of this transformation is that the main difference of the sensor signals will be presented with a few new coordinates only. The other coordinates can be dropt with little loss of the information on the variance between the different measurements. The transformed results can be shown in a 2 or 3-dimenional diagram, see figure 4a. The diagram shows the result of a principal component analysis with measurements of seven building materials. Three of the materials form independent material groups. The other four materials cannot be separated properly and build a wide spread group. The measurements of the silicon sealant form a chain and don’t show a grouping effect. This is caused by a change in the emissions as the sealant was investigated during the first days after preparation and was still in the process of hardening.



(a) (b)

Figure 4. Classification of measurements with a MGSS of seven different building materials using (a) Principal Component Analysis (PCA), and (b) Linear Discriminant Analysis (LDA). The percentage behind the axis title describes the fraction of the components on the total variance.

To improve the classification a linear discriminant analysis (LDA) was used. This method uses classifiers which were assigned to the measurement samples by the experimenter to build the discriminant functions. The discriminant functions are optimized to maximize the separation between the groups and to minimize the within group variance during a learning process. Figure 4b shows the results of the LDA. The four material classes which were inseparable with PCA are still near together but at least the silicon sealant and the floor adhesive build separate clusters.

The third classification approach was done using a self organizing map (SOM). SOM is a special type of neural networks which was developed by Kohonen [8]. As the PCA the SOM is an unsupervised method which uses only the sensor response data to classify the data. For each sample the SOM determines one single “wining neuron”, which fits best to the input data. During the learning phase the network weights of the wining neuron and the near by neurons were adjusted towards an optimized result. The Matlab neural network toolbox was used and a map of 7x9 neurons was used to classify the building materials, see figure 5. Each symbol characterizes a neuron. In the illustration the neurons are displayed equally distributed. The shading demonstrates the distance of the neurons (dark: near, white: far). With the selected SOM except the floor adhesive the building materials can be separated into groups. Some materials show some higher within group differences as can be seen for the acrylic sealant 1. Here the measurements are distributed on several neurons of light shading (bigger distances between neurons). The acrylic sealant 2, however, is spread over several neurons but these are located closer to each other.

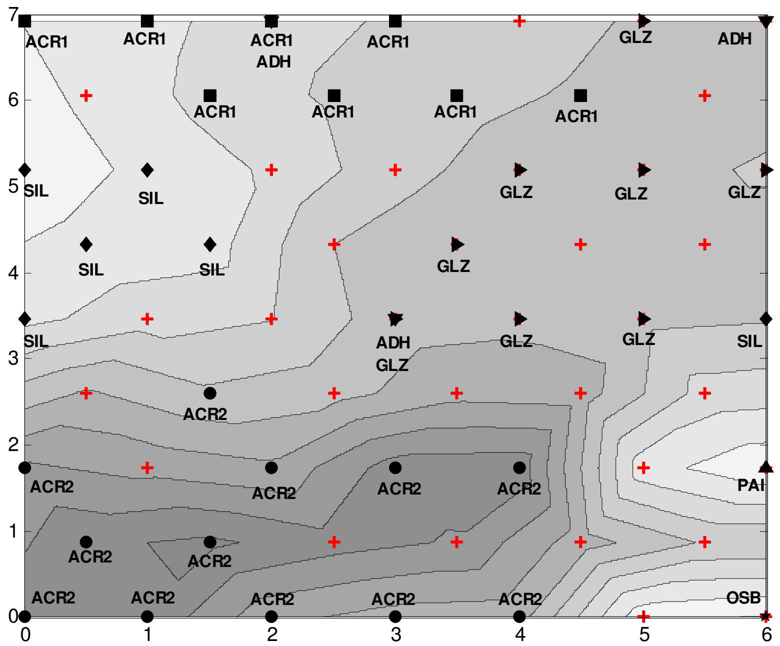


Figure 5. Self organizing map to classify building materials (ACR1: acrylic sealant 1, SIL: silicon sealant, ADH: floor adhesive, GLZ: wood glaze, OSB: oriented strained board, PAI: wall paint, ACR2: acrylic sealant 2)

A separation of the measurements of different building materials into classes seems to be possible, but some of the materials can not be separated perfectly. More sophisticated classification methods and an enhanced and modified sensor system with higher sensitivities might solve this problem. The classification was done in respect of building materials, which means every material builds its own class. That might not be the best classification approach since this can lead to many different classes. Finding different classification criteria could optimize the separation of classes.

The second step of the data processing model is to find a correlation to the odor intensity. The data for a specific class is used to find a transformation algorithm to estimate the perceived intensity. For the correlation a principal component regression (PCR) was applied. A PCR is a combination of a principal component analysis and a multiple linear regression. The PCA reduces the dimension of data set and maximizes the variance for the first components. Since the later components contain only noise signals only the first three components were used for the multiple linear regression. The following diagram shows the deviation from the measured intensities from the assessments of the panel for all of the seven investigated materials, figure 6. It is assumed that an odor classification method is used which properly separates each material in its own class and the regression was done using measurements for one single material.

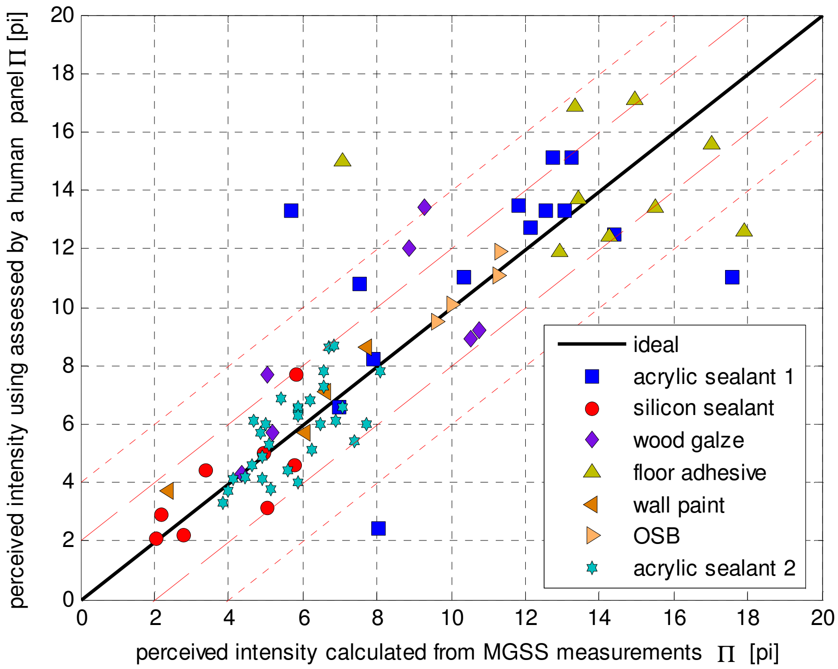


Figure 6. Comparison of the measured intensities (calculated by PCR for each material separately using the “leave-one-out” method) with the intensities assessed by the human panel.

The shown values were calculated with the “leave-one-out” method. In this method all except one (the tested) sample were used to estimate the regression parameters. The regression algorithm was then used to calculate the intensity for the left out sample. This is done n-times for all n samples of the odor class.
The diagram shows two ranges with 2 pi and 4 pi deviation from the ideal estimation. It should be mentioned here, that the reference value of the assessed perceived intensity is not an exact value itself and has also some uncertainties. Most of the estimations of the perceived intensity using the MGSS lie within the 2 pi range. Some calculated intensities for the acrylic sealant 1 and the floor adhesive are totally different from the assessed values. At a closer look it can be seen, that these values mark the edges of the range of intensities of this odor class. Near the edges of the intensity range the regression function is inaccurate.

DISCUSSION

MGSS have the potential to estimate the perceived odor intensity of building materials. It depends strongly from the sensitivity of the sensors and on the data processing model. A two-step-method for data processing is appropriate since a direct calculation method for a determination of the intensity cannot be found. A crucial point in the model is still the development of an odor space for the odor classification. It won't be practical to have for each building material a different odor class with corresponding transfer algorithm. It is the aim to find a few primary classes which cover the relevant odors which occur in indoor environment.

In this study basic algorithms for classification and regression were used. More sophisticated methods may improve the results. An improvement of the sensors itself in optimizing the sensor combination and in increasing the sensitivity of the sensor elements would be a further step towards a measurement system for the intensity of the odor in the indoor environment - towards systems, that can be used for measuring, monitoring and controlling the indoor air quality. To reach this aim, the European project SysPAQ (Innovative Sensor System for Measuring Perceived Air Quality and Brand Specific Odors) has been started to develop an innovative system to measure the indoor air quality as it is perceived by humans.

The first investigations were done in a laboratory environment using single materials only. In real indoor environment the emissions of several materials were present at the same time. The additive behavior of the odorants must be investigated in further research work to transfer the laboratory results to real indoor environment.

ACKNOWLEDGEMENT

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Sensory evaluation of building products - Results of a German round robin test

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SUMMARY

In the german research project a method had been developed and tested, which enables the simple and safe integration of sensory tests into current test procedures under the AgBB scheme (see below). The method can provide a proper amount of sample air to the panellists for assessment, regardless of the size of the emission test chamber.

A round-robin test is conducted to evaluate the combined emission/odour testing scheme. A standardized building product is evaluated by eight laboratories concerning chemical and odorous emissions. At each laboratory between six and fourteen members of an odour panel have scored the perceived intensity of the building product using a comparative scale based on acetone.

INTRODUCTION

The indoor climate and indoor air pollutants influence the health and comfort of the occupants of indoor spaces. Indoor air pollutants are emitted by a various sources. Among these sources, building products are of particular importance hence their selection is often not within the occupants' discretion and they cover large surface areas in a room.
Since VOC emissions are frequently able to be recognized, and can also lead to health impairment, sensory testing was included as an important aspect in the assessment scheme of the Committee for Health-related Evaluation of Building Products (AgBB scheme) [1]. Despite improving analysis possibilities and the development of artificial noses, replacing the human nose in the determination of perceived air quality has not been successful until today. Odours develop from a number of chemical substances but not all materials generate the perception of smell in humans. Many thousands of different substances can be detected in the room air, but even a quantitative determination of each single material would not enable a statement about the smell effect of a combination to be made.
Within the research project "Determination of emissions from building products using test chamber measurements and development of product-specific test conditions for low-emission building products" a new method to investigate the odour impression of the emissions of construction products was developed [2]. The odor measurements use the same chamber conditions like the VOC emission tests in accordance with EN 13419-1 [3]. Typically, chamber tests following this standard have exhaust air streams of much less than 0.7 liters per second, the minimum volume flow rate for odor measurements [4]. Therefore, the exhaust air of the chambers is collected in 300 l gas sampling bags.
A human panel assesses the air collected in the air sampling bags using a comparative scale of acetone/air mixtures, which help to determinate its intensity Π. The introduced perceived

intensity Π can only be determined with panels using a comparative scale. Unlike the acceptability method with large panels, the intensity of odorous substances in the air is determined by comparing different specified intensities of the reference material acetone. The smelling capability varies from human to human. Training and use of comparative sources ensure that the influence of subjective perception of the test result is reduced since all panel members evaluate air quality based on the same scale.
50 building products were tested in this project in emission chamber tests according to the provisions of the AgBB scheme. VOC emission tests were performed in each case on the first, third, tenth and 28th day and, in addition, the odour emission tests were conducted on selected products.

METHODS

In the project a laboratory comparison was planned to validate the odor measurement method. However, none of the other institutes possessed the same odor measurement instruments as the Herrmann Rietschel Institute, therefore a suitable procedure had to be found. Thus a timely graduated process was planned. A transportable version of the comparative scale was designed. The objective of this was to achieve a constantly adjustable acetone concentration in the sample air independent of the ambient conditions. The design scheme of the comparative scale is illustrated in Figure 1.
The comparative scale is in essence composed of three parts: sample air circulation, source of acetone and dosing device. The units in contact with air are almost wholly manufactured from stainless steel, glass or PTFE (polytetrafluoroethylene), which are practically odorless.

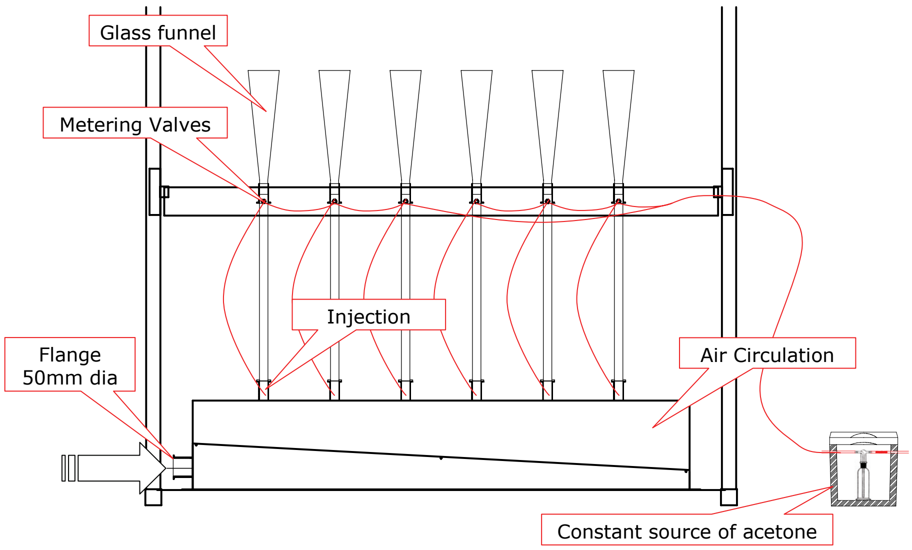

Figure 1: Scheme of the transportable comparative scale

Sample air circulation is connected via a flange to a suitable smell-neutral air supply. The sample air circulation provides constant flow rates between 0.9 and 1.0 l/s per marker (5.4 to 6.0 l/s for six markers) which ensures an undisturbed operation. The constant source of acetone consists of a pressure-resistant wash bottle and a cooling device. The acetone filled wash bottle is supplied with compressed air which is pumped through and then enriched. Cooling prevents an over saturation of the compressed air and a consecutive condensation in the pipes. The acetone fog is effectively separated by a stainless steel filter from the enriched air.

The six funnels are supplied with the constant air/acetone mixture via a distribution hose. A metering valve per funnel regulates the amount of the acetone/air mixture added to the sample air within the range of 0 to 1150 mg m^{-3}.
Six different mixes of acetone concentrations in the range between 20 mg m^{-3} (0 pi) and 300 mg m^{-3} (15 pi) help the panellists gain their orientation in determining the perceived intensity of an unknown sample. In addition, the comparative scale enables uniform assessment criteria on different testing days and facilitates the standardisation of the method. The comparative scale was equipped with its own fan unit for an independent mobile operation. An electronic fan control unit enables the exact adjustment of the total flow to provide a supply for the comparative scale.
In view of the need of mobility, the AirProbe [5] was further developed for the laboratory comparison. To achieve a better and more constant provision of sample air, the operational principle was fundamentally changed (Figure 2).

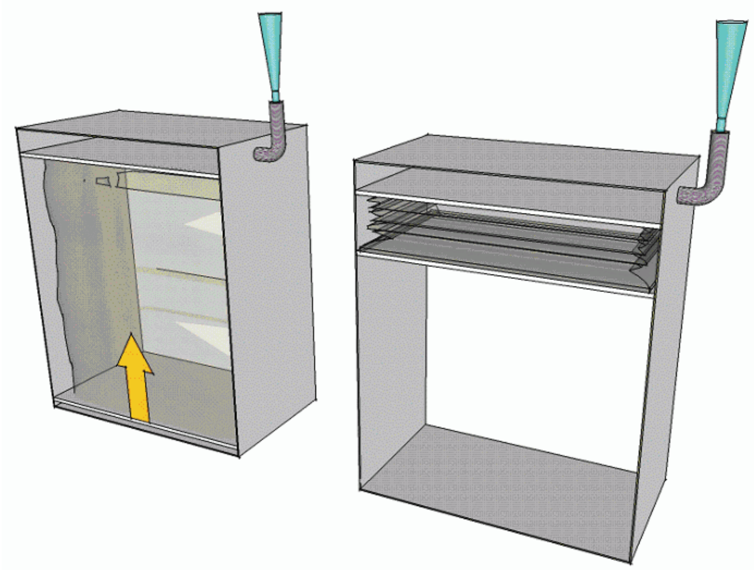

Figure 2: Operational principle of sample provision in AirProbe II

The sample provision in AirProbe II is based on the principle of a press. The sample container in AirProbe II is between an upper fixed plate and a mobile lower plate or piston. When the lower plate is moved at a constant speed, a constant flow rate is produced from the sample container, which is attached to a high-grade steel pipe leading to the glass assessment funnel. Based on a pressure difference measurement at an orifice, the flow rate can be calculated and indicated on a display. The piston velocity can be changed with a potentiometer over a wide range. The panellists have a longer time for assessment, since only the operation of a switch is needed during the smelling procedure to provide the full flow rate. In the period between two smelling procedures the flow rate is reduced to a minimum to prevent a back flow. Due to a smaller average flow rate up to 12 panellists can perform an assessment at the glass funnel.
The body of AirProbe II consists of a light aluminium transportation box with the external dimensions 1200 x 800 x 510 mm.
The equipment was sent from participant to participant. The experimental setup only remained with each participant for three weeks. In this time an eight-day chamber test combined with chemical analysis and sensory testing with a minimum of six panelists of sample air was performed. Because of the new sensory testing the participants were instructed in a one-day introductory course in the handling of devices and the measurement technique. In addition to the air quality manual, extensive documentation and detailed guidance on the test methodology were available to the participants. Brief device-related descriptions were compiled during the tests for quick orientation.

In addition to the innovative odour tests, an emission chamber test was also performed in accordance with DIN EN ISO 16000-9 to -11 and/or -6 [6] within the interlaboratory comparison. Planning the emission chamber tests was a challenge, since all participants had to carry out the tests under the same conditions over a longer period. Thus a roll or stacked commodity, which would age in the course of time, had to be rejected. Instead, a sample had to be selected which remains stable during a period of at least six months. Furthermore the sample had to exhibit good measurable emissions and at least a clear, well identifiable odour. The choice finally fell on an acrylic sealing compound, which can be prepared very reproducibly for the tests. The acrylic sealing compound was placed in an aluminium standard channel, levelled with a trowel and placed into the chamber after a short waiting period. Emission and odour tests on air from the chamber were carried out on the first, third and eighth days. The odour transport containers were filled at the chamber exit. When this was done in chambers smaller than 1 m³, the container had to be filled dynamically over night. At the 1 m³ chamber it was easy to exchange the tank contents three times within one hour, so that the odour sample could be taken directly before the assessment. For the third measurement day additional tubes were sent, which the participants loaded simultaneously to their sampling. They were then analysed by the Federal Institute for Materials Research and Testing (BAM) to decide whether the chamber influences or analysis caused deviations between the results of the institutes. The good comparability of the VOC data of the chamber test shows that the different participants largely had the same test conditions in the chambers. Each participant additionally received another standard solution, which could be used for the quantification of the emissions.

RESULTS

After completing the tests in the institutes, the data sets were conveyed to the Hermann-Rietschel-Institute for assessment. All laboratories involved managed to complete the sensory assessments despite the short time. Six to 18 panellists per laboratory participated in the tests. Between 69 and 76 individual values were produced each test day. Figure 3 illustrates the results from the laboratories.

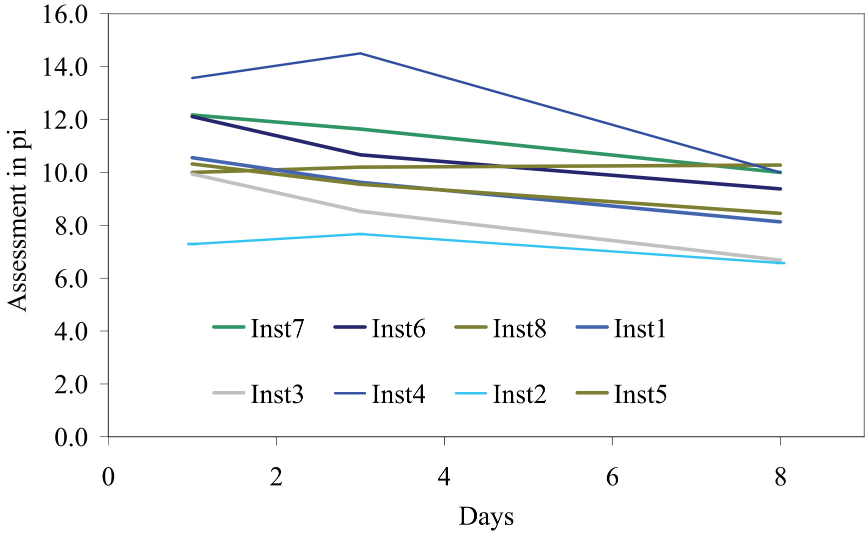

Figure 3: Intensity of all laboratories

The test days are plotted on the abscissa and the perceived intensity Π on the ordinate. The results of each institute were calculated as an arithmetic average of the individual answers for each test day.

The results of Institutes 2 and 4 were excluded from further consideration in the assessment. Technical problems in the test procedure in one of the institutes and deviation from the procedural regulations in the second institute provided results under non-comparable test conditions. Technical problems in AirProbe II forced Institute 8 to do a short-term modification during the test run, which may be an explanation for the horizontal profile of intensity assessments. Since all other boundary conditions were adhered to in this Institute and AirProbe1 is in principle suitable for the execution of the tests, the results were included in the overall assessment. Figure 4 shows the cleaned laboratory results and the average value of all laboratory results.

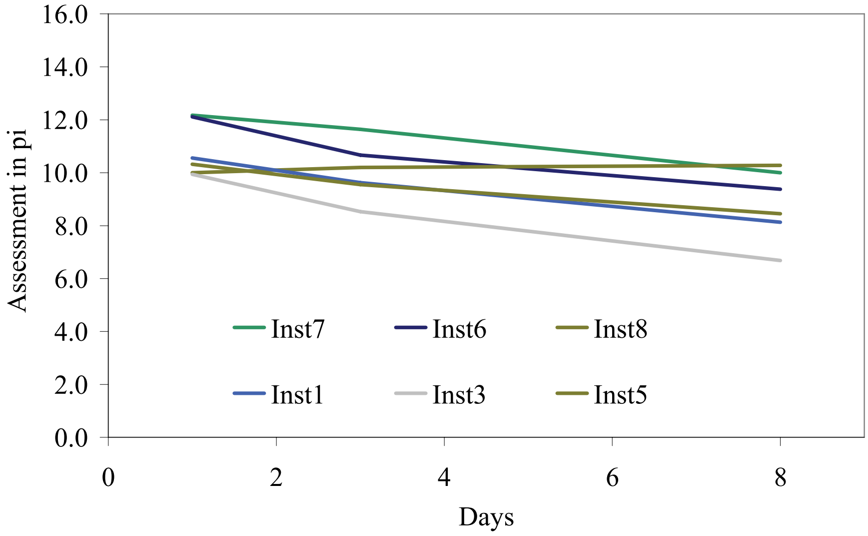

Figure 4: Cleaned intensity, average of all laboratory results

In addition to the medium standard deviation, Figure 5 also shows the maximum deviation of the laboratories from the average value of all laboratory results for a better assessment of the test results.

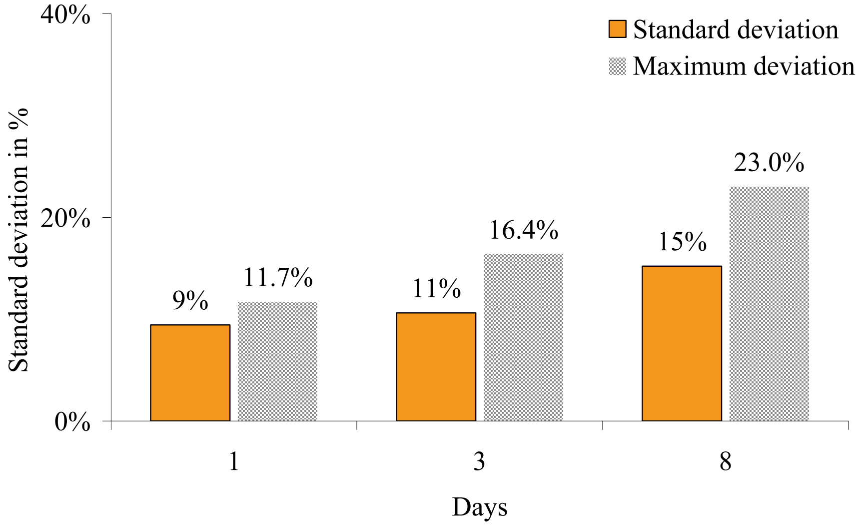

Figure 5: Medium standard deviation and maximum deviation of the laboratories from the average of all laboratory results

The medium standard deviation is between 9% on the first test day and 15% on the last test day. The maximum deviation is between 11.7% and 23%. These reasonably good values from a first co-operative test could be improved by a focused selection of the panellists.
Since the laboratories involved exclusively used 1-m³ chambers for test execution, the Herrmann Rietschel Institute carried out additional tests in 1-m³ and 20-litre chambers.

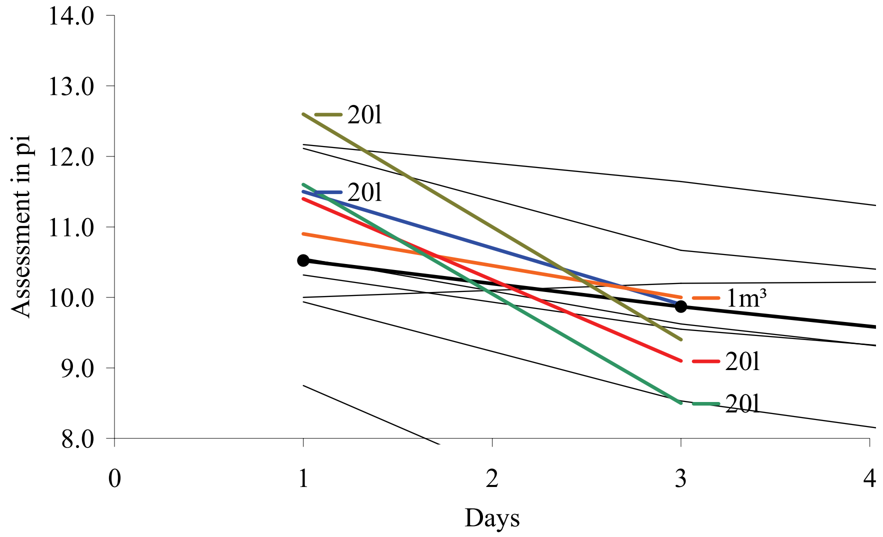

Figure 6: Intensity assessment of samples from the 20-litre and 1-m³ chambers

The investigations are limited to the first and third test day. Figure 6 shows an enlarged excerpt of the two test days. The curves show that the chamber size obviously has no influence on the sensory assessment.

DISCUSSION

The results of odour assessment show that newly developed devices, guidelines and test descriptions allow comparable intensity assessments to be obtained between the laboratories for the selected building material, although the laboratories involved had little or no experience with sensory assessment of building materials.

The good overall result of the co-operative test is illustrated in Figure 7. In addition to the standard deviation of the laboratories from the overall mean, the maximum deviation of the individual laboratories over the entire test period is also shown for a better assessment. With a standard deviation of 9% (first test day) to 15% (eighth test day) very good results were obtained under the conditions specified above. The Hermann Rietschel Institute achieved a very constant deviation of only 4 % from the general mean with trained panellists under optimised laboratory conditions over the entire test period.

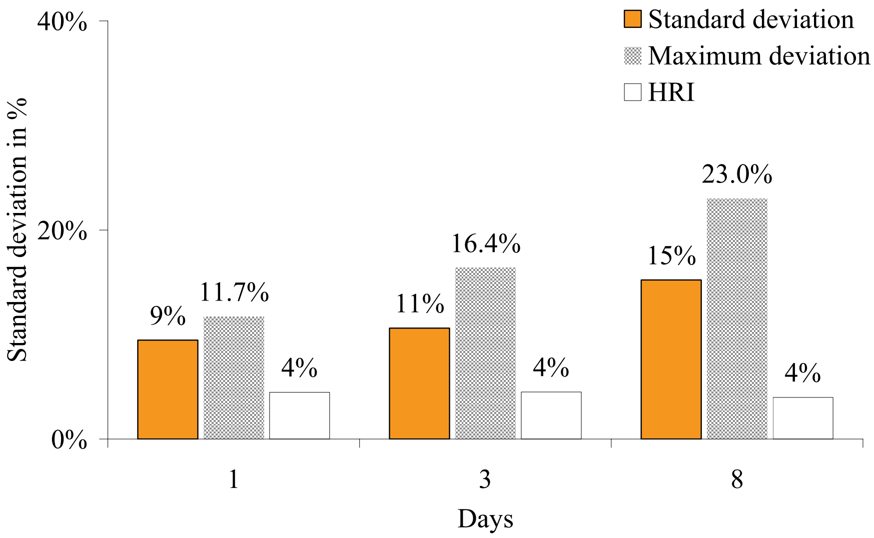

Figure 7: Standard deviations of the laboratory results

Figure 8 shows the medium standard deviation of the individual answers from a laboratory and the maximum deviation over the test days. It provides information as to how large the dispersion of the individual answers of the panellists of a laboratory is on a particular test day. The maximum standard deviation here shows the enhancement potential of the results if a focused selection of the panellists was undertaken.

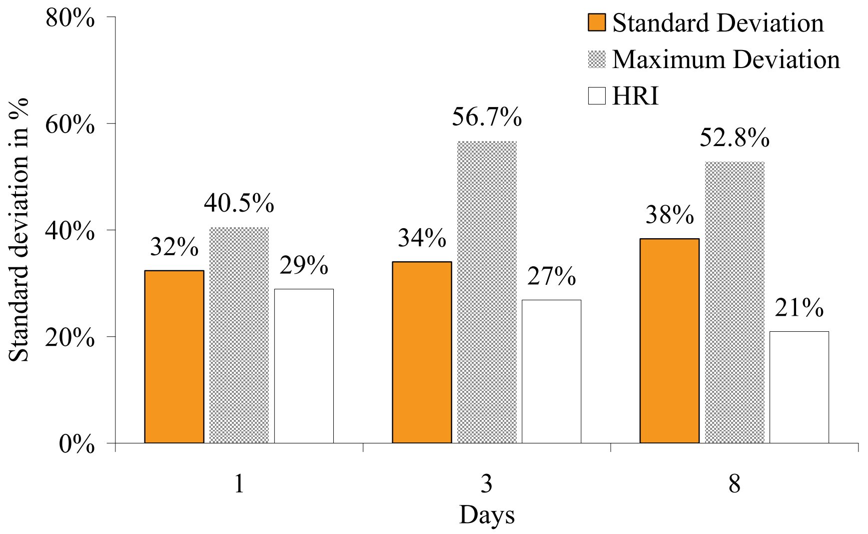

Figure 8: Medium standard deviation and maximum deviation of the individual answers of the individual laboratories, standard deviation of the Herrmann-Rietschel-Institute

The standard deviations of the Herrmann-Rietschel-Institute, which works with a focused selection of trained panellists under optimised ambient conditions and whose staff members have been familiar with the method for years, are accordingly well below the medium standard deviation.
Since the co-operative test was only performed on a single building material sample due to time constraints, other co-operative tests would be necessary for a possible validation of the method where several building materials were tested by panelists.

ACKNOWLEDGEMENT

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Basis Odor Model for Perceived Odor Intensity and Air Quality Assessments

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SUMMARY

This paper shows first results of an ongoing research project, which aims to develop a transfer model to link the odor intensity with the perceived air quality assessments. This model is based on basis odors, which were in this research project selected according to the primary odors defined by Amoore. Each basis odor measurement of the perceived intensity and air quality establish a transfer function between the intensity and the acceptability values. The basis odor samples are generated by using the saturation method and a dilution process with clean air. The experiments include the examination of single basis odors as well as mixtures of basis odors. Additionally, the influence of the relative humidity of the air on the perceived intensity and acceptability is part of the investigations. First results of the experiments show different correlations between perceived intensity and acceptability for all basis odors.

INTRODUCTION

A well known method developed by Fanger determines the perceived air quality with a sensory panel judging the acceptability of the air quality [1]. A category scale is used for the assessments with no absolute reference. Inherent to the acceptability assessments is a high standard deviation between the individual assessments of the panelists and therefore a large panel of subjects is required for statistically significant results. Bluyssen applied a group of trained panel members for air quality assessments using a comparative scale [2]. The training and the comparison with a reference reduces the standard deviation and a smaller panel size can be used.

Up to now it is very difficult to compare the results of both assessment methods. The influence of the thermal state of the air on the assessments is different for the acceptability method and for the method using a comparative scale. Studies [3, 4] show that the acceptability of air is decreased with increasing specific enthalpy. Results from a panel using a comparative scale indicate that the specific enthalpy is not important as long as the relative humidity stays constant. This discrepancy reveals that both methods appear to detect different measures of the odor perception. The use of a comparative scale forces all panel members to concentrate on the perceived intensity. Thus, a comparative scale cannot be used to measure the perceived air quality in terms of the acceptability and may be used for perceived intensity assessments only.

Subjects who are asked for the acceptability of the air quality do not differentiate between the hedonic tone and the intensity level of the odor impression and assess a combined impression instead. On the one hand, the perceived air quality in terms of the acceptability is an appropriate measure concerning the comfort in indoor environment, but on the other hand the perceived intensity judgments are more reliable because of the common reference of the comparative scale. A transfer model which relates the intensity to the acceptability assessments enable to use the smaller panel size and the more precise intensity assessments to evaluate the perceived air quality.

In this study the correlation between acceptability and perceived intensity assessments for defined basis odors is investigated. The basis odor transfer model should close the gap between intensity and acceptability measures. The dependence of the hedonic impression on perceived intensity and acceptability can be extracted from the measured data which gives more insight in the structure of the interrelationship. Seven odors, which are defined as primary odors in the stereochemical theory of Amoore [5], are used for the investigations on the transfer model. Each odor is characterized by reference substances some of which were substituted by other substances with similar odor characteristics due to safety reasons. Later on, the investigation of the influence of the relative humidity on this relationship will be part of the research project as well as the investigation of the combinations of the basis odors which shed light into the interaction of the transfer functions of the basis odors.

SENSORY ASSESSMENTS

Samples of the basis odors are presented in the air quality laboratory of the Hermann-Rietschel-Institute. The laboratory is constructed mainly of glass and stainless steel and ventilated with a high air change rate which provides an odorless measurement environment.

The subjects for the assessments of the acceptability (assessments without the use of the comparative scale) are selected without any specific criteria except the absence of any respiratory diseases or known anosmia. The panel consists of 30-40 persons, mainly students. 45% of the subjects are women and 77% declared themselves as no smokers. The mean age of the panelists is 24. The subjects are asked for the acceptability using a scale divided into 20 steps ranging from -10 (clearly not acceptable) up to 10 (clearly acceptable) [1] and for the hedonic impression a scale from -4 (extremely unpleasant) to +4 (extremely pleasant) [6].

The panel using the comparative scale consists of 9-12 persons and is selected from subjects passing the training test. This test evaluates the capability to compare the odor intensities of the unknown samples with the comparative scale. In addition to the determination of the perceived intensity of the air samples the subjects are also asked to judge the hedonic impression using the same scale as the other sensory panel. At the Hermann-Rietschel-Institute acetone is used as a reference substance for the comparative scale. The perceived intensity is assessed in the unit pi, which was introduced by Müller [7]. Currently, the scale is constructed linearly in respect to the acetone concentration and an intensity step of one pi corresponds to a concentration step of 20 mg/m³. At an acetone concentration of 20 mg/m³ which is approximately the odor threshold the perceived intensity is defined to be zero. An odor perceived intensity value above zero should be detectable for most of the panel members.

For every air sample the arithmetic mean of the acceptability, the perceived intensity and the hedonic impression of both groups are calculated. The first set of experiments use seven reference substances, listed in Table 1. The reproducible and constant air samples are generated by newly developed dosing equipment (see Figure1) operating with a saturation method. Synthetic air is enriched with the substance in a wash bottle at room temperature. The air is cooled down to a defined temperature to reach a saturated substance-air mixture with constant concentration according to the vapor pressure. The saturated air is dosed in an air flow of clean air and is presented to the sensory panel through glass cone. The presented concentrations of reference substances in air are selected considering safety rules and health considerations. The stability of the odor samples has a major impact on the quality of the results of the investigations since the assessments have to be done rather in sequence over a longer period than it can be done parallel. The developed dosing equipment fulfills this requirement adequately.

Table 1. Reference Substances

Odor classification	**Chemical substance**
Camphoraceous	Eucalyptol
Musky	15-pentadecenolide
Floral	Geraniol
Pepperminty	(-)-menthone
Ethereal	Diethyl ether
Pungent	Acetic acid
Putrid / fishy	Pyridine

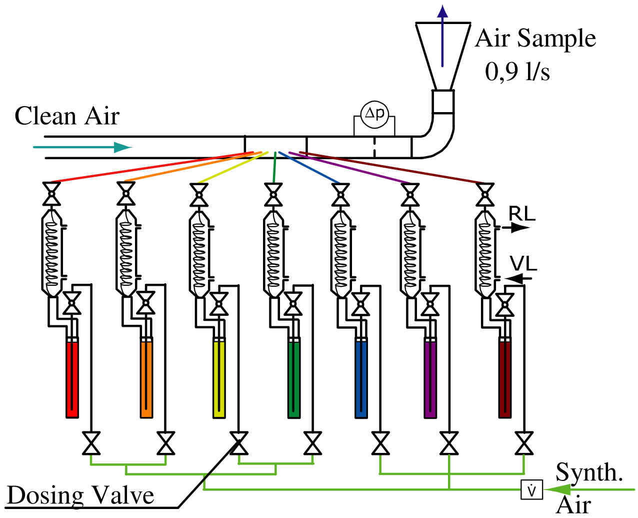

Figure 1. Schema of the dosing equipment

During the first set of experiments the relative humidity is 45% (± 5%) and the temperature of the air is 21.5 °C (± 1°C). The second set of experiments will be focused on the combination effects of different basis odors and the influence of temperature and relative humidity. The influence of the relative humidity will be determined for only three of the basis odors and for the relative humidity of 30% and 70%.

RESULTS

Comparison of acceptability and perceived intensity

Figure 2 shows the interrelationship of the assessments of the acceptability and the perceived intensity for the first set of experiments using the basis odors. The figure indicates that the acceptability values are generally decreased by increasing the perceived intensity. But the transfer functions are not completed yet. More measurements have to be done to cover a wider range of the odor intensity. By now it can be seen, that the seven odors can be separated in two groups, one of acceptable odors in the positive acceptability range and another one of unacceptable odors in the negative acceptability range.

Peppermint, floral and musky odors are almost for all considered concentration levels in the acceptable area. The gradients of floral and peppermint odor are very similar. The camphoraceous odor seems to change over from positive to negative acceptability values at higher intensity values. Ethereal, pungent and putrid odors do not show any positive values on the acceptability scale and are not accepted even for low odor intensities.

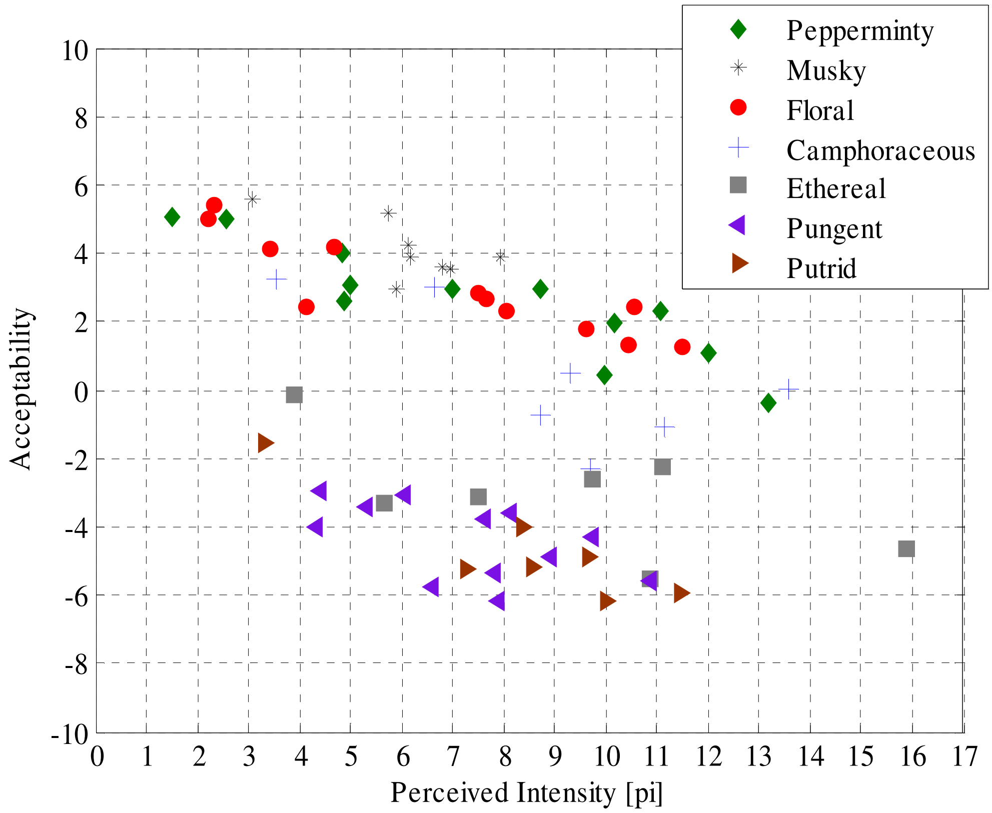

Figure 2. Relationship between the perceived intensity and the acceptability of basis odors

The hedonic impression

The hedonic tone is assessed by both the panel using the comparative scale and the panel judging the acceptability. Thus the relationship of the hedonic tone to the intensity as well to the acceptability can be derived from the measured experimental data. The experiments indicate that the hedonic tone is not or only weakly depending on the perceived intensity (see Figure 3). Using a comparative scale leads to perceived intensity assessments.

The acceptability on the other hand depends mainly on the hedonic impression. With decreasing acceptability the hedonic impression decreases linearly (see Figure 4). This correlation seems to be independent from the kind of odor. The measurement points are grouped in the more acceptable positive odors in the upper right quadrant and the not acceptable negative odors in the bottom left quadrant of the diagram.

The assessments of the acceptability and the intensity are done by two different sensory groups. The comparison of the results of the intensity and the acceptability assessments require that both panels represent a similar sample of society and have a similar perception of odors. The hedonic impression is therefore assessed by both groups. This enables to evaluate the differences in the odor perception. Figure 5 shows the assessments of the hedonic impression of the seven basis odors for the panel using a comparative scale in relation to the hedonic impression assessed by the panel for acceptability judgments. The voting of the hedonic impressions shows only a small deviation from the one-to-one linear correlation. The assessments of the panel using the comparative scale tend to be less positive for the pleasant odors. The sizes of the panels are not identical and hence the uncertainties and the standard deviation of the hedonic assessments are equal for both groups the data for the larger group should be considered more accurate.

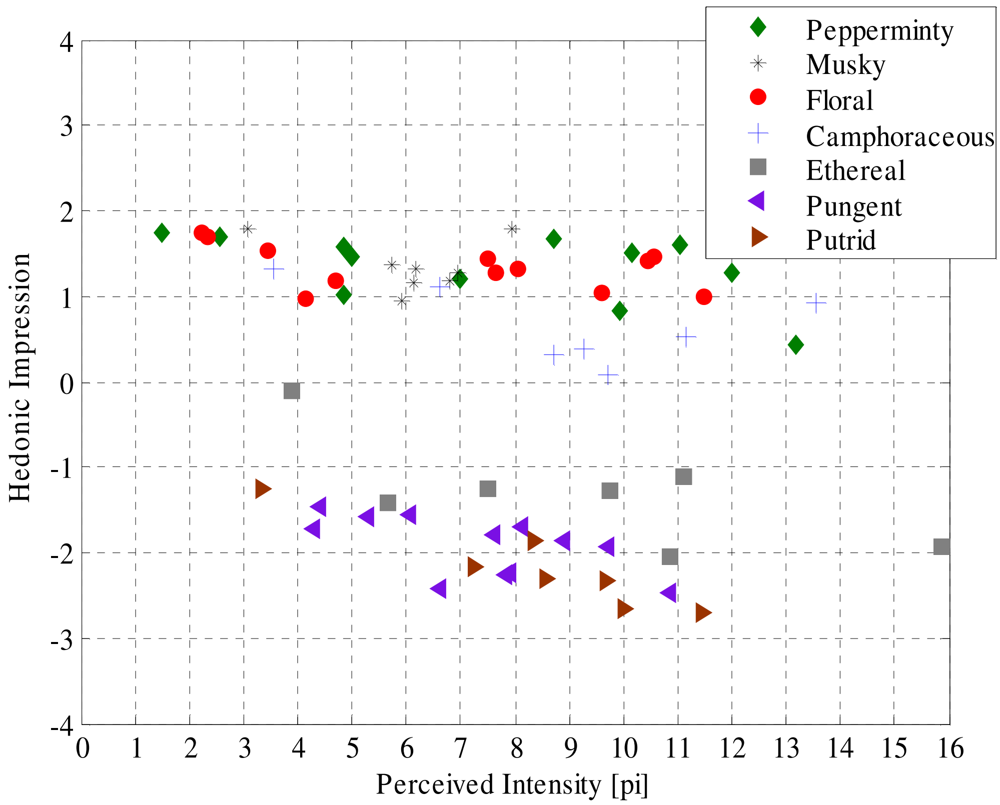

Figure 3. Comparison of the hedonic impression and perceived intensity

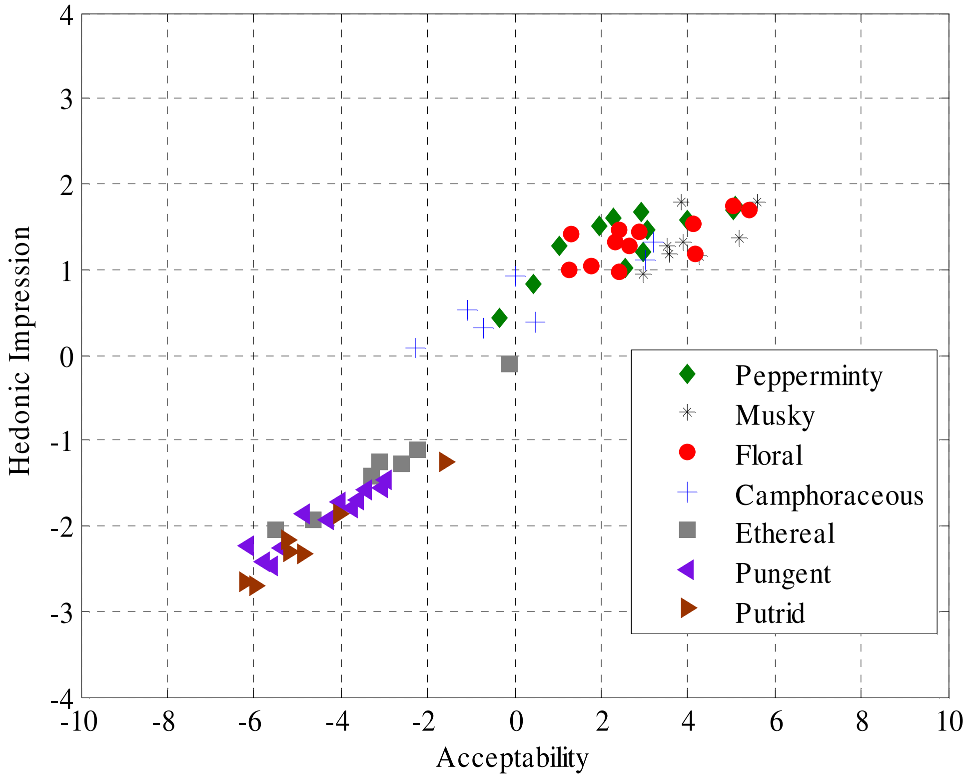

Figure 4. Comparison of the hedonic impression and acceptability

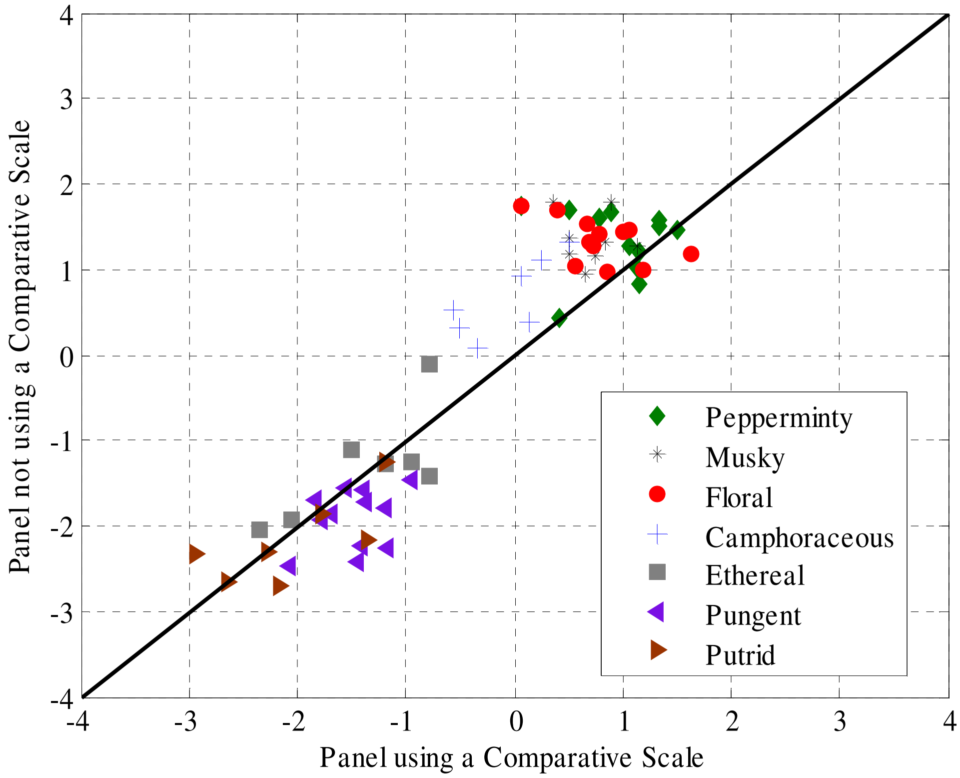

Figure 5. Comparison of the hedonic impression of the two different panels

DISCUSSION

The experiments clearly show that the acceptability votes of a panel which uses no comparative scale is a combined impression of the perceived intensity and the hedonic tone. The intensity assessed by a panel using the comparative scale assesses the perceived intensity almost not influenced by the hedonic tone and thus without a mental valuing process of the perceived odors. The acceptability votes mainly decrease when the ratings of the intensity are increasing. The assessments can be grouped in the pleasant, acceptable odors and the unpleasant, not acceptable odors. This, however, indicates that there exists no universal transfer function between the acceptability and intensity votes. It always depends on the pleasantness of the odors. A combination of assessments of the intensity and the hedonic tone to estimate the acceptability is a promising approach.

So far, the investigation has been done at constant relative humidity and temperature. As mentioned before, the relative humidity has an influence on the perception of the intensity whereas the specific enthalpy is influencing the acceptability of the odors. In the course of this project the influence of the relative humidity on the acceptability-intensity relationship will be investigated. Combination of basis odors, which will also be measured later on, will give insight in the perception of mixtures especially when combining a pleasant with an unpleasant odor.

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Effect of Mechanical Ventilation System(s) to Indoor Chemistry Products in Air Conditioned Buildings: Quest for Tropical Research

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SUMMARY

The potential for reactions among indoor pollutants to generate reactive and highly irritating products is a reason to maintain adequate ventilation rates and clean ventilation filters. Terpenoid (from recirculated air, a scenario common in the tropics) captured by ventilation filters can react relatively quickly with ozone which may lead to downstream air supply that contain oxidized terpenoid and this may be perceived to be less acceptable than outdoor air. The chemical composition (water solubility and chemical reactivity) of these particles strongly governs their toxicity. The composition determines either how the respiratory tract reacts or how the body responds. There are few studies examining the impact of ventilation rate and ventilation filters on indoor chemistry. Studies documenting the effect of filters, ventilation and recirculation rate on ozone initiated chemistry in buildings utilizing recirculation of conditioned air are lacking. Thus the need to stimulate research in this area to better understand the effect, recirculation of conditioned air phenomenon could have on indoor air quality serves as the motive for the quest for tropical research. The paper demonstrates that reaction between reactive gases to generate highly irritating products may even be more important in mechanical ventilation system that utilizes recirculation of conditioned air.

Key words –: Filters; ventilation rate; recirculation rate; ozone initiated chemistry; oxidized terpenoid; secondary organic aerosols (SOAs).

INTRODUCTION

In various parts of the world, it has been documented that morbidity and mortality rates have continued to be on the larger scale due to the poor indoor air human beings are constantly being exposed to [1][2][3]. This gives indication that air we breathe in the indoor environment must be given due and consistent attention. In order to avoid outdoor environmental hazards and discomfort, human beings, whether consciously or unconsciously, do spend 90% of their time in this indoor environment (e.g. offices, homes, factories, buses, aircrafts etc) where chemical reactions of indoor gases do take place whether at the gas phase or on the interior surfaces (surface chemistry). Indoor chemistry is being recently recognized as a phenomenon important to IAQ perception, irritation, and health [4]. To ensure clean indoor air within buildings mechanical system(s) must expel stale air and replenish it with clean, “fresh” outdoor air. However, simple as this may seem, it is not always the case. If they are poorly designed, operated, or maintained, however, ventilation systems can contribute to highly irritating products resulting from indoor chemistry. In the tropics many buildings utilize high recirculation of conditioned air to minimize the energy required to remove the high humidity of outdoor air and media filters are the most commonly used filtration method for the removal of particles from outdoors and recirculated air. If these filters are not properly maintain they may be an important to issue to indoor air quality perception, irritation and health as a result of everyday occupant exposure. The presence of a used filter in a ventilation system can have an adverse impact on perceived air quality, SBS symptoms, and performance of office work [5]. In addition to removing particulate contaminants, they accumulate a particle layer that can react with ozone. Ozone particle cake reactions serve as a sink for ozone and a source of secondary carbonyls [6]. Of special note is oxidized terpenoid that could be formed due to reaction between recirculated air containing terpenoids and ozone captured on the filters to form oxidized terpenoids.

MOTIVATION OF THE STUDY

Building occupants may be chronically exposed to ozone-initiated chemistry oxidation products if appropriate measures are not taken. The toxicity potential of indoor air and particularly those where ozone-terpenes reaction have occurred could lead to several adverse health effects associated with both respiratory and cardiovascular health effect including mortality and morbidity [4][7][8]. It appears that from limiting ozone concentration and certain reactive volatile organic compounds (VOCs), maintaining clean filters, adequate ventilation rate, and recirculation air change rate under recirculation of conditioned air scenario are very important. The role of recirculation has been under estimated as compared to non recirculation scenario (where all studies have focused on), recirculated air may contain terpenoids which can sorb onto the surface of particles captured on ventilation filters. It can react relatively quickly with ozone to produce oxidize terpenoids which have great influence on perception and acceptance of indoor air [9]. Ventilation rate and recirculation air change rate could also influence the production of SOAs from ozone initiated chemistry and also its particle size distribution. The particle size distribution determines where deposition occurs in respiratory tract. The chemical composition of these particles (water solubility and chemical reactivity) strongly governs its toxicity. The composition determines either how the respiratory tracts react and of how the body respond. Surprisingly, despite the importance of recirculation of conditioned air scenario (commonly use in air- conditioned buildings in tropics) to indoor air which building occupants are exposed to everyday, there are little or on effect of filters, ventilation, recirculation on ozone initiated chemistry in air conditioned buildings. This review demonstrates that reaction between reactive gases to generate highly irritating products may even be more important in mechanical ventilation system that utilizes recirculation of conditioned air.

PARAMETERS AFFECTING OZONE INITIATED CHEMISTRY

The review focus only on ventilation filters and air change rate (ACH) while others may include ventilation ducts, building materials, relative humidity and temperature etc.

Ventilation Filters

Collected particles in clean filters if not removed from air stream, are the source of indoor air pollution [10][11][12][5]. Apart from ventilation filters negative impact on perceived air quality, the indoor air pollution from used filters could lead to other perception and even SBS symptoms after longer exposure [13]. In a field study by Wyon et al., (2000), significant improvement in self- estimated productivity of workers was observed when new filter were used as compared to used filter. Wargocki et al., (2004) did a well controlled field intervention experiment in a call-centre providing a telephone directly service. Their results agreed with earlier studied by Wyon et al., (2000) (uncontrolled field experiment). They reported significant improvement in some SBS symptoms and environmental perceptions with use of new filter. Used ventilation filters could also contribute to indoor chemistry to produce noxious compound [5].

Beko et al, 2006 studied contribution of sensory pollutant by oxidation process of ozone initiated chemistry from used filters. They observed that organic compounds that react with O_3 in the surface of filter cake are transform to more highly oxidized species which degrade the quality of perceived air quality as judged in the subjective evaluation.

Mysen et al. (2006) studied emission from different types of used ventilation bag filter and their impact on perceived air quality. New filter banks of cityflo filter and F7 filters were put into two similar ventilation systems with continous air flow (24hours per day). They found that the used F7- filter deteriorate the air quality significantly compared to new/no filters. The used F-7 filter deteriorated the air quality significantly compared to the similarly used cityflo-filters.

Hyttinen et al., (2006) did a study on removal of ozone by clean, dusty, and sooty supply air filters. The removal of ozone (O_3) on supply air filters was studied. Especially, the effects of dust load, diesel soot, relative humidity (RH), and exposure time on the removal of O_3 were investigated. Some loss of O_3 was observed in all the filters, except in an unused G3 pre-filter made of polyester. Dust load and quality influenced the reduction of O_3; especially, diesel soot removed O_3 effectively. Increasing the RH resulted in a larger O_3 removal. The removal of O_3 was highest in the beginning of the test, but it declined within 2 h reaching almost a steady state as the exposure continued. However, the sooty filters continued to remove as much as 25–30% of O3. Up to 11% of O_3 removed participated in the production of formaldehyde. Small amounts of other oxidation products were also detected.

Beko et al. (2005) measured ultrafine particle concentration upstream and downstream of filter sample when air passing through the filters either did or did not contain ozone. They conducted the experiment using small scale test equipment in low polluting office with very low background ozone level. They reported that the downstream/upstream ratio was higher when O_3 was present. This indicates the creation or growth of ultrafine particles after the air has passed through the filter.

Hytttinen et al (2003) conducted laboratory and field experiment (from nine different Finnish office buildings) to examine potential reaction of ozone on used filters. The filters had removal efficiencies (F5 to F8 ratings) and had been in service for approximately one year. Their study indicates that used filter removed a part of the ozone from supply air. This removal could lead to more chemical reaction on used filter surface.

Implications of these studies

Ventilation filters are employed with the primary belief that they improve indoor air quality and present building occupants cleaner air. Adequate filtration of ventilation air to indoor environment could enhance occupant health significantly. An example of this is using the filter to reduce the outdoor ozone concentration. However improper maintenance of these filters may produce adverse effects because it could be source for ozone-initiated chemistry. Ventilation filters serve as an absorption bed which in the presence of air passing over it, and may lead to production of other gaseous chemical products that might result in sensory irritation. The above studies provide evidence of oxidative transformation on used filter due to the presence of ozone and organic compounds. These oxidation products do result in supplied ventilation air that is less acceptable and highly irritating.

Ventilation rate and recirculation rate

Ventilation rate

A measure of how quickly the air in an interior space is replaced by outside air by ventilation and infiltration. Air change rate is measured in appropriate units such as cubic meters per hour divided by the volume of air in the room, or by the number of times the home's air changes over with outside air. For example, if the amount of air that enters and exits in one hour equals the total volume of the heated part of the house, the house is said to undergo one air change per hour. The rates at which used air is exchange with outdoor air do have influence on secondary organic aerosols generated by ozone chemistry.

Recirculation rate

A measure of how quickly the air in an interior space is replaced with recirculated air. Recirculated air is defined as air that is returned from a heated/cooled space, reconditioned and/or cleaned, and returned to the space. Technically the recirculated air into the space is supposed to be clean but that is not usually the case if the ventilation filter is badly maintained. In the tropical context, recirculation of conditioned air is usually adopted to minimize the energy required to remove the high humidity of outdoor air. In Singaporean office buildings typically 90% of the air is recirculated while 10% is from outdoor air. To lower indoor space temperature the recirculated air can be moved at a faster rate or simply by passing over chilled water. It is important for the reader to note recirculation air change rate/ recirculation rate (ACH) is a different term from proportion (%) of air recirculated.

Relationship between recirculation rate and ventilation rate

If ventilation rate (outdoor air change rate) is kept constant and recirculation rate is varied (from low to high) in an experimental scenario, it means the residence time[4] in the system would basically be the same for all the experiments. The higher recirculation rate would simply be moving the air through a closed system at a faster rate. Thus, as recirculation rate increases, the boundary layer adjacent to the surface of the indoor space and recirculation loop becomes thinner. This presumably will affect deposition of particles and therefore resulting into larger surface removal of particles (reactants and reaction products). This is totally different from a situation when ventilation rate alone is used and varied, the residence time will be different for such set of experiments.

Effects on indoor chemistry

Weschler et al. (2000) did a study on the influence of ventilation on reactions among indoor pollutants. They use a one compartment mass balance model to simulate unimolecular and bimolecular reactions occurring indoors. The initial modeling assumes steady-state conditions. They also modeled a non-steady state scenario because at low air change rates, there may be insufficient time to achieve steady state. In the cases examined, the results demonstrate that the concentrations of products generated from reactions among indoor pollutants increase as the ventilation rate decreases. They also supplement the modeling studies with a series of experiments conducted in typical commercial offices. The reaction examined was that between ozone and limonene. The ozone was present as a consequence of outdoor-to-indoor transport while limonene originated indoors. Results were obtained for low and high ventilation rates. They observed a consistent result with the modeling studies. The concentrations of monitored products were higher at the lower ventilation rates even though the ozone concentrations were lower.

Weschler et al (2003) examine the influence of air change rates on the concentrations of secondary organic aerosols as well as the evolution of their particle size distributions. The experiments were performed in a manipulated office setting containing a constant source of d-limonene and an ozone generator but was remotely turned “on” or “off” at 6h intervals. The air change rates during the experiments were either high (working hours) or low (non-working hours) and ranged from 1.6 to >12h^{-1}, with immediate exchange rate. They observed evidence for coagulation among particles in the smallest size range at low air change rates (high particle concentrations) but no evidence of coagulation was apparent at higher air change rates (low particle concentrations). They also observed that at higher air change rates the particle count or size distributions were shifted towards smaller particle diameters and less time was required to achieve the maximum concentration in each of the size ranges where discernable particle growth occurred.

Implications of these studies

Above studies demonstrate that low ventilation rate, in addition to allowing accumulation of indoor pollutants leads to larger concentration of SOAs derived from reaction among such pollutants as well as for particles to accrete organic materials and undergo coagulation process. Relationship between ventilation rate and recirculation air change rate could also affect ozone initiated chemistry and health. Despite recirculation of conditioned to air indoor chemistry and health there are only very few studies examining the influence of ventilation rate on indoor chemistry (e.g.[21][22]), and with little or no study on the effect of ventilation rate and recirculation rate on indoor chemistry under recirculation of conditioned air scenario.

Knowledge gap

There are little or no studies documenting the *effect of filters, ventilation and recirculation rates on ozone initiated chemistry in buildings utilizing recirculation of conditioned air*

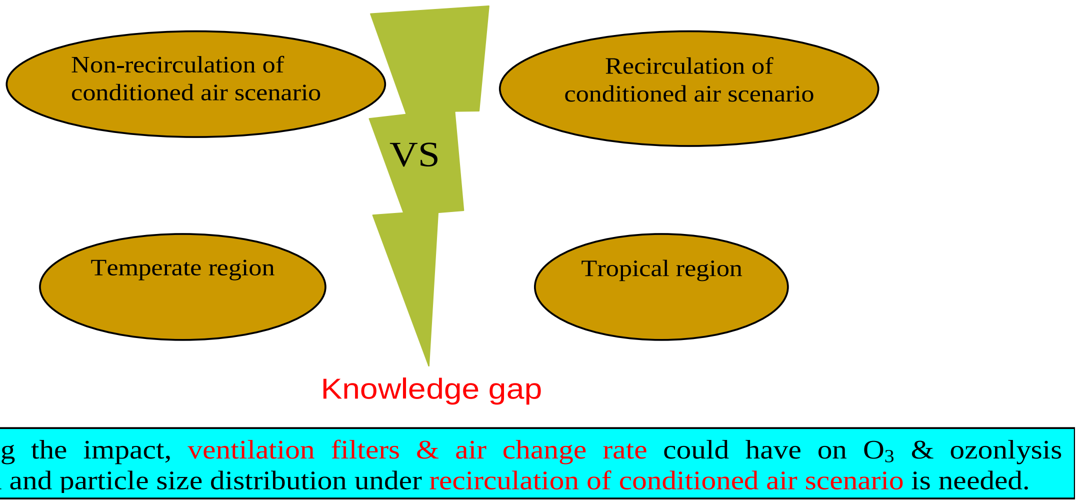

Figure 1: Knowledge gap

INDOOR CHEMISTRY IN TROPICS:

Research from IAQ unit of Centre for Total Building Performance, NUS

Given the above review, the role of filters, ventilation and recirculation rates could have on ozone and ozone initiated chemistry products' concentration and its particle size distribution (mass and number concentration) under recirculation of conditioned air (scenario common to tropical regions) clearly warrants further research. This understanding would help prioritize efforts to improve indoor air quality of air-conditioned buildings in the tropics. To this effect, The Indoor Air Quality research unit of Centre for Total Building Performance, National University of Singapore is interested in understanding the effect, filter, ventilation and recirculation rates could have on ozone initiated oxidation products (especially formation of oxidized terpenoids) under recirculation of conditioned air scenario. This section summarizes preliminary findings from 2 separate studies conducted in a large field environmental chamber (FEC) with its own Air Handling Unit (AHU). But, it is important to note that the research group has planned series of experiments focusing on AHU to be able to achieve controlled experiments as compared to using FEC as a primary experimental setup, which by its nature, there are lot of confounders that might be affecting the results. However, as a supplement to AHU experiments, FEC experiments would be conducted to give big picture of what could happen in indoor environment. The first study examines the effect of recirculation rate on the concentrations of SOA resulting from reactions between indoor limonene and ozone. In the second study, 3 experiments (without filters, with clean filters and with used filters in the AHU room) were conducted at constant recirculation rate and fresh air supply. The First study has shown that the rate at which air is recirculated through a room can have a significant effect on the Particle count, Size distribution and concentration of SOAs and oxidation products of ozone-initiated indoor chemistry [23][24][25]. For second study, the fact that the used filter has very little effect on the particle levels in the FEC compared to "no filter" at the initiate stage of the experiment suggests that two separate processes are occurring – 1) the used filter is removing SOA from the air stream and 2) chemistry is occurring on the surface of the used filter that contributes SOA to the air stream downstream of the filter. The chemistry that is occurring on the surface of the used filter includes ozone oxidation of organics associated with the captured particles/SOA, as well as ozone oxidation of limonene and limonene oxidation products such as 4-AMC (4-Acetyl-1-methylcyclohexene) and IPOH (3-isopropenyl-6-oxoheptanal) sorbed to the used filter. In terms of differences between a new filter and a used filter, one would expect greater sorption of limonene and its oxidation products on a used filter than on a new filter. This, in turn, would lead to more SOA being produced as a consequence of ozone passing through a used filter compared with a new filter [25][26].

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Building Material Interactions and Perceived Air Quality

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SUMMARY

The impact of common building materials interaction on perceived air quality related to air change level was studied. The methods of chemical analysis (TVOC) and subjective sensory assessment was used. The measurements were conducted in test chamber under standardized conditions (23±1°C, 50±5%) and different ventilation rates. The positive impact of sorption "phenomenon" on perceived air quality and user's wellbeing was found for most of the tested combinations. The more significant sorption effect was found for materials with larger specific surface. The results have shown that sorptive material such as gypsum board can affect the ventilation rate needed to achieve the satisfactory perceived air quality and wellbeing.

INTRODUCTION

The variety of studies have shown that common indoor surface materials are significant sources of indoor pollution and have essential impact on perceived air quality. In addition these materials can decrease the level of pollution through sorptive processes. Study [1] shown that sorption of common indoor air pollutants (volatile organic compounds) onto surfaces such as gypsum boards and Semia can markedly improve the perceived air quality, decreasing the ventilation rate required to achieve a given degree of acceptability. On the other hand, the re-emission of VOCs can prolong the presence of indoor air pollutants. This study demonstrate the significant and prolonged effect that sorption/desorption phenomena can have on perceived air quality under realistic indoor conditions. The VOCs sorption onto building materials depends not only on the material properties but also on the VOCs type and indoor environmental condition such as temperature, relative humidity and air velocity. These parameters in relation to VOCs sorption onto different building materials were studied in [2]. According to [3] the VOCs adsorption increases with decreasing vapor pressure of VOCs.

The indoor pollution sources impact on perceived air quality is studied within the scientific research project The theory of Indoor Making with reference to Building Materials, Constructions and HVAC Systems supported by Slovak Scientific Grant Agency. The first results of this project which is aimed to investigate the effect of various building materials and their combination on perceived air quality is presented within the paper as continual research to previous project. The aim of the research is to obtain the wider understanding of the indoor surface materials impact on perceived air quality as well as recommendations for proper interior materials selection in order to guarantee acceptable perceived indoor air quality.

METHODS

In this study the impact of common indoor surface materials interactions on perceived air quality in relation to different air change levels was studied. The increased ventilation rate caused different conditions from VOC concentration and air velocity point of view. The effect of odorants sorption on perceived air quality was studied in a small-scale test chamber. Flooring coverings are considered to be important for the indoor air quality. Regarding this fact the flooring coverings were considered as a potential source of air pollution. The investigated flooring materials were OSB flooring system (waste wooden products) and PVC flooring. The gypsum boards finished with water based painting were considered as a sorption material. To compare the obtained results the gypsum board were substituted by impregnated wallpaper which has smaller specific surface then gypsum boards.

Chemical analysis and subjective sensory assessments of emissions from selected surface materials and their combinations were carried out in test chamber after their exposition to different air change conditions. The air change levels were specified for standard model room (3,2 x 2,2 x 2,4m) with respect to prepared European legislation (prEN 15251:2005) for 3 different indoor air quality categories. Category A corresponds to a high level of expectation and leads to a highest percentage of satisfied occupants in respect to indoor environment (n = 3,0 1/h). Category B corresponds to a medium level of expectation (n = 2,0 1/h) and category C to a moderate level of expectation (n = 1,2 1/h). The air change levels were specified regarding that the tested materials are non low polluting materials [4]. The surface of the test specimen was exposed to the chamber air which was maintained at a temperature, humidity and air movement similar to that which can be expected in the indoor environment. In addition to these conditions the chamber VOCs concentration depends on the supply airflow rate in the chamber and the area of the test specimen. The tests were performed with an area specific airflow rate similar to that which can be expected during the common use of the material.

The chemical analysis was based on detection of selected volatile organic compounds. Active sampling of VOCs was performed by using a pump (Aircheck 2000) with air flow rate of 400 ml/min on charcoal tubes ORBO 32S (Supelco) during 24 hours. The absorbed VOCs were analyzed by gas chromatography (GC Varian 3 300) after extraction into CS_2. In order to simplify the indoor volatile organic compound loading the TVOC value was estimated as the indicator of building material odors. Mølhave, the author of TVOC index proposed a value of comfort level 200 $\mu g/m^3$. The sum of VOCs was calculated based on individual VOCs concentrations including unidentified peak areas converted to decane equivalents (equivalent to toluene).

The exhaust air from each test chamber was led through a diffuser for sensory assessments. An untrained sensory panel of 20 subjects assessed odor intensity and perceived air quality. Before the first assessment the panelists were instructed how to use the scales and the exposure equipment. The responsible person of the experiment assessed each subject's attitude and motivation concerning to experiment and subject's personal hygiene. There was no restriction on distribution of gender or smoking habits. The age ranged from 18 to 40 years with the mean of 35 years, and 14 % of the subjects were smokers. The panel stayed in a well ventilated room without odors before the assessments. Then the subjects indicated their immediate evaluation on two continuous scales regarding odor intensity (from 0 no odor to 5 overwhelming odor) and acceptability of the air (from -1 clearly unacceptable to +1 clearly acceptable) from which the percentage of dissatisfied was estimated. The standard test method was used for calculation of acceptability and estimation of odor intensity. During the sensory assessments, the test chambers were covered with aluminum sheets to hide the building products from the view of the sensory panelists.

RESULTS

OSB boards and gypsum boards / wallpaper interactions

The perceived air quality were regarded as acceptable when odor intensity is less than 2 - moderate odor and air acceptability more then 0 – just acceptable/just unacceptable. In case of individual painted gypsum boards considered as a sorption material were both mentioned criteria fulfilled at air change level of category C. The air change level n = 0,5 1/h assured only required level of odor intensity, the air acceptability was not reached. The air change level for category B seems to be satisfactory for wallpaper as a comparing material from both subjective perceived air quality indicators point of view. In the case of individual OSB flooring system the required level of odor intensity has been reached at air change level for category B (n = 2,0 1/h) however the air was still unacceptable. Finally, the acceptability of assessed air was reached at air change level for category A (n = 3,0 1/h).

The positive effect of odorants sorption onto gypsum boards on perceived air quality is obvious from Fig. 1. The tested air was perceived more acceptable for OSB boards with addition to painted gypsum boards than for flooring material individually. The subjective assessment was not so high than assessment of individual painted gypsum boards. The criteria of subjective indicators were fulfilled at air change level for category B (n = 2,0 1/h) when both materials were in the interaction although the individual OSB flooring system these criteria did not fulfilled. The impact of increased air change level on VOCs sorption assessed on both subjective scales has not been evident. The most significant VOCs sorption observed on scale of odor intensity was in case of the highest air change level. The observation on scale of air acceptability showed the most significant sorption effect in case of air change level n = 2,0 1/h where individual OSB board was assessed as unacceptable however in combination with gypsum boards the panel assessed the air as acceptable.

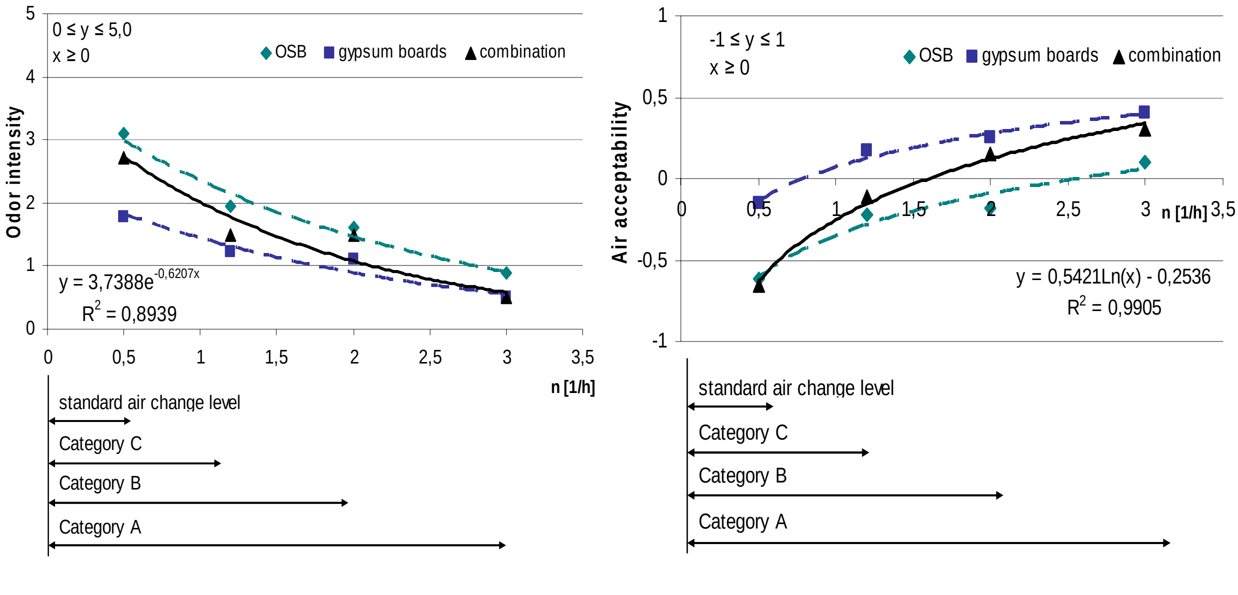

Fig. 1 Perceived air quality for OSB flooring – gypsum boards combination

The effect of odorants sorption onto impregnated wallpaper on perceived air quality is obvious from Fig. 2. No positive impact of materials combinations on perceived air quality was noticed for the OSB boards with addition to PVC wallpaper. The tested air was perceived less acceptable in all cases of studied air change levels for flooring material with addition to PVC wallpaper as for those ones individually. The required levels of both subjective indicators in the case of investigated indoor surface materials combination were reached by air change level for category A (n = 3,0 1/h).

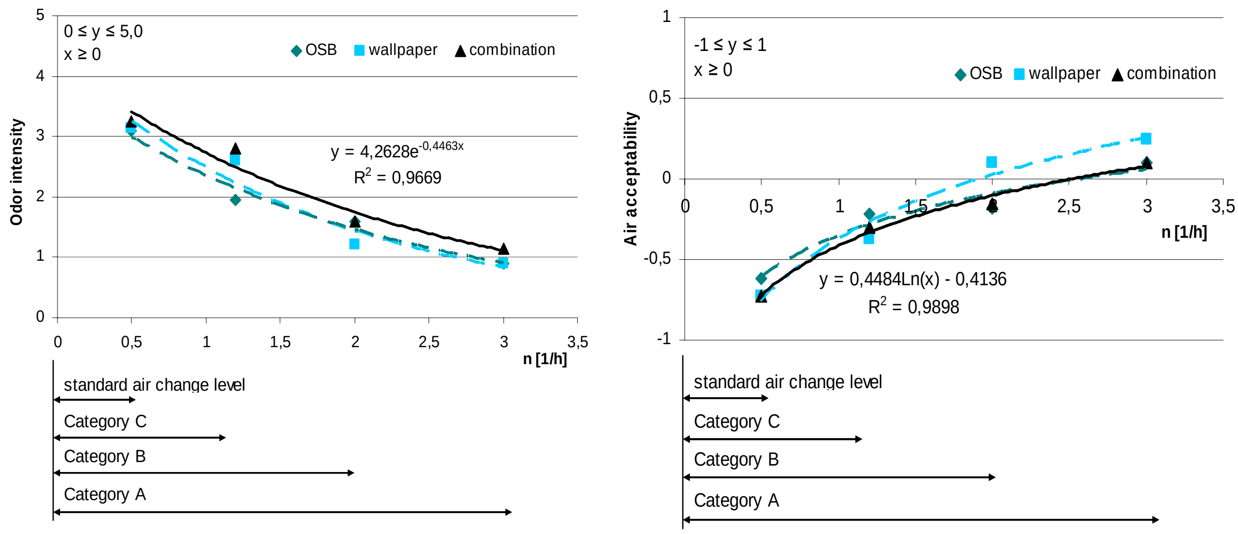

Fig. 2 Perceived air quality for OSB flooring – wallpaper combination

The air change level of category C (n = 1,2 1/h) guaranteed olfactory comfort level of TVOC (Molhave 200 μg/m^3) however the subjects found the air as still unacceptable in all cases of tested indoor surface materials (Fig. 3). The chemical analysis of emissions shown that ventilation rates in public buildings should be based on required level of perceived air quality. Unexpectedly the results of TVOC analysis did not clearly confirmed VOCs sorption when OSB flooring system was combined with painted gypsum boards. The negative effect was found with regard to both individual tested materials. The positive sorption effect for OSB flooring and wallpaper was found with regard to individual tested OSB flooring system.

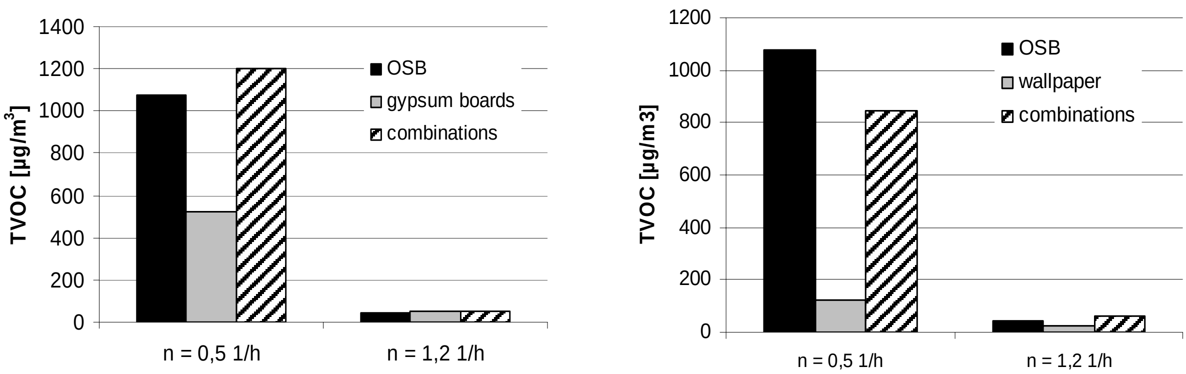

Fig. 3 TVOC analysis for OSb flooring system and its combinations

PVC flooring and gypsum boards, wallpaper interactions

In the case of individual PVC flooring the required level of odor intensity has been reached at air change level for category C (n = 1,2 1/h) however the air was slightly unacceptable. Both criteria of subjective indicators were reached at air change level for category B (n = 2,0 1/h).

The positive effect of odorants sorption onto gypsum boards on perceived air quality is obvious from Fig. 4. The tested air was perceived more acceptable for PVC flooring with addition to painted gypsum boards than flooring materials individually. The air acceptability was almost as high as for individual painted gypsum board. When both materials were in the interaction the criteria of subjective indicators were fulfilled at air change level for category C (n = 1,2 1/h). At this air change level these criteria were not fulfilled for individual PVC flooring. The sorption effect observed on scale of odor intensity decreased with increasing air change level. For air change level up to 2,0 1/h the sorption effect was not obvious. The assessment on scale of air acceptability almost copied the results from subjective assessments of gypsum board. The impact of increased air change level on VOCs sorption assessed on both subjective scales has not been evident.

The effect of odorants sorption onto impregnated wallpaper on perceived air quality is obvious from Fig. 5. The tested air was perceived less acceptable for flooring material with addition to PVC wallpaper as for those ones individually for almost all tested air change levels. The perceived odor intensity was lower than for the individual PVC wallpaper but higher than for the PVC flooring. The required levels of both subjective indicators in case PVC flooring and wallpaper combination were reached at air change level for category B (n = 2,0 1/h). The sorption effect observed on scale of air acceptability decreased with increasing air change level. For air change level u to 1,2 1/h the sorption effect observed by air acceptability scale was not obvious. Nevertheless the impact of increased air change level on VOCs sorption assessed on both subjective scales has not been evident.

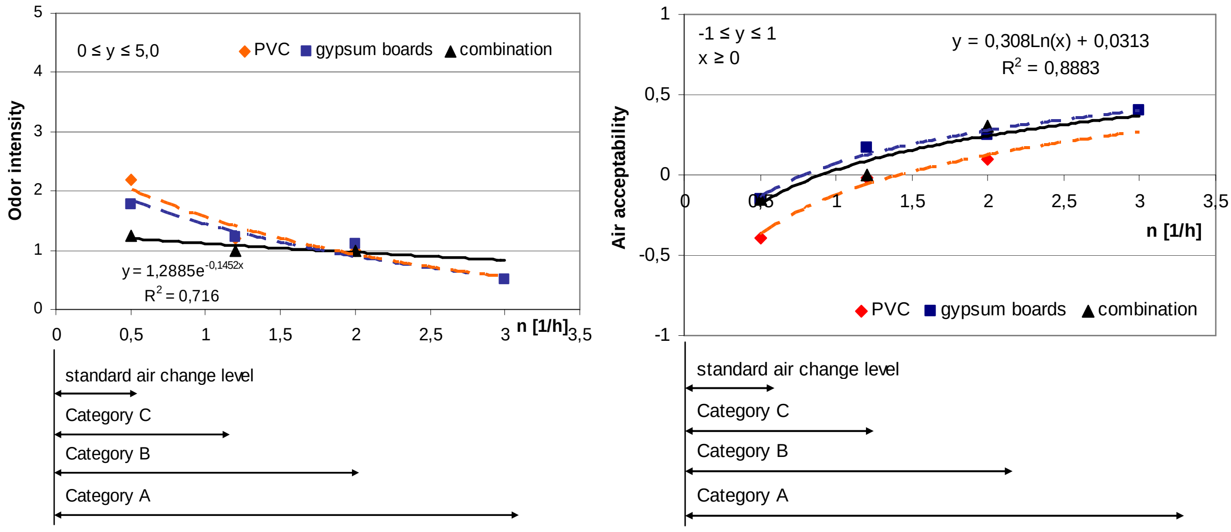

Fig. 4 Perceived air quality for PVC flooring – gypsum boards combination

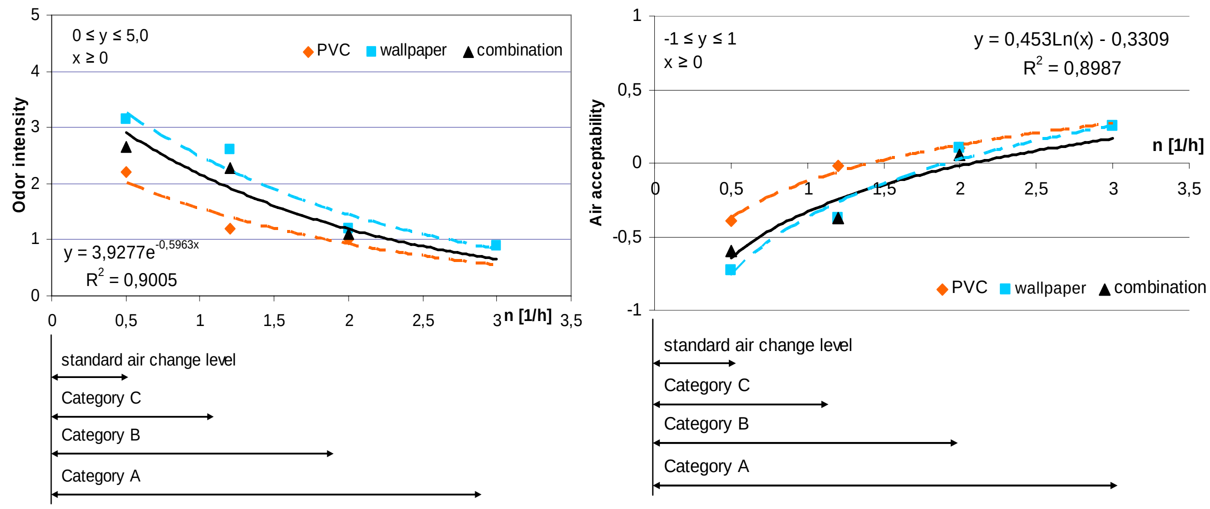

Fig. 5 Perceived air quality for PVC flooring – wallpaper combination

The air change level of category C guaranteed olfactory comfort level of TVOC (Molhave 200 µg/m^3) however the subjects found the air as still unacceptable in almost all cases of tested surface materials (Fig. 6). The positive effect of VOCs sorption expected from subjective assessment was confirmed by chemical analysis. The effect was markedly positive for PVC flooring and painted gypsum board with regard to both individual tested materials as well as for PVC flooring and wallpaper with regard to individual PVC flooring.

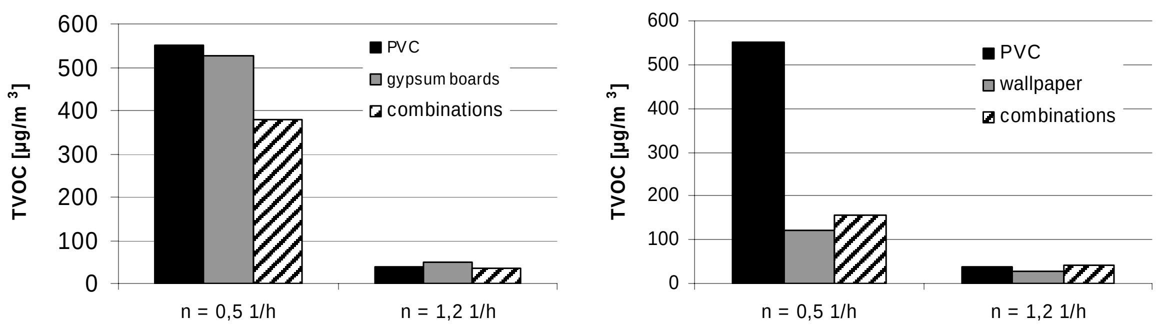

Fig. 6 TVOC analysis

DISCUSSION

The increased air change rate did not cause expected increasing of perceived air quality assessed by panel of subjects. Respecting other indoor climate parameters requirements (air temperature, humidity, and air movement) the increasing of air change level in order to achieve acceptable perceived air quality is not reasonable and makes no real sense [4].

The presented study has shown that there exist the real possibilities to apply sorption effect of odorants in order to achieve acceptable perceived air quality. More significant sorption effect was found for material with larger specific surface. Purposive using of material sorption effect creates opportunities to decrease the ventilation rate needed to achieve satisfactory perceived air quality and wellbeing however with respect to another possible indoor pollution unrelated to perceived air quality and building materials. The sorption of odorants onto gypsum boards studied within this study reduced the ventilation rate needed to achieve specified criteria of subjective indicators approximately by 1 Vol/h. The impact of increased air change level on VOCs sorption assessed on both subjective scales has not been evident.

ACKNOWLEDGEMENT

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CFD Modelling of Indoor Environment Quality Affected by Gas Stoves

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SUMMARY

Gaseous emission - CO_2, NO, and NO_x of gas stove has significant and harmful effect on indoor air quality (IAQ) in residential kitchens. To avoid increasing health risk it is essential to use mechanical ventilation such as kitchen hood. However, according former laboratory studies, extract air flow generated by typical hoods is not adequate to achieve required IAQ. Thus there is an increasing need to develop a new kitchen ventilation system and to estimate the required exhaust ventilation rate.
In order to describe required ventilation of kitchens, emission of gas stoves has been investigated for years at Department of Building Service Engineering, BUTE. Laboratory and field studies were carried out, afterwards, based on the results, pollutant distribution was described by CFD simulation (FLOVENT).
In this paper an overview of method and results of CFD modelling is published.

INTRODUCTION

Gas stoves as pollutant sources can be described in accordance the power of burners. However occupants' behaviour (the way of use) takes significant part as well. If gas stoves are used in order to reduce required heat load of boiler, burner will run free. In this case pollutant concentration could be increase by 200-300%. From health point if view it should be taken into consideration that occupants normally stand near to the source during the cooking. Thus air in breathing zone contains higher level of pollutants. Therefore occupants are primary affected by combustion products. Application of hoods decreased the concentration levels but it has not appeared to be satisfactory in IAQ point of view.
In case of gas stoves Hungarian standards has a main issue providing fresh air to ensure burning process and to defend human health. However indoor air quality has an increasing impact on design process.

In order to help design, implementation and maintenance to develop more effective kitchen ventilation system, effect of gas stoves has been studied at laboratory and field studies. Based on results it is proved that trade hoods ($V_{ex,max}$ =180 – 250 m^3/h) can not ensure the required ventilation from IAQ point of view.

To estimate required exhaust air flow CFD modelling has been made by FLOVENT. Number of cases has been investigated as regards amount of air removed by kitchen hood. Supply air has been provided using air terminal devices installed above the windows. Validity of CFD model has been investigated based on results of former laboratory and filed studies.

METHODS

Using CFD modelling it can be predicted air moving in the enclosure. Calculations are based on fundamental flow and energy equations such as:

- Navier – Stokes equations
- Energy conservation
- Mass conservation
- Concentration equations
- k – ε turbulence model

To solve these it is necessary to make modelling assumptions in order to describe the investigated phenomenon in an accurate way. This also helps to decrease number of parameters, required computer capacity and solution time. Main modelling assumptions are follows:

- outdoor air does not contain any of pollutants;
- supply air temperature is equal to outdoor air temperature;
- invariant outdoor climatic circumstances (air temperature, air velocity, pressure);
- continuous ventilation by given amount of ventilated air.

Geometrical model has developed according architectural parameters of enclosure investigated at field studies. Gas stove model has been made following proportions of a device with average technical parameters, therefore commonly used one.

Pollution loads have been estimated by laboratory measurements. Considered operating time: 60 minutes. Initial indoor air temperature $t_i(\tau=0) = 20°C$.

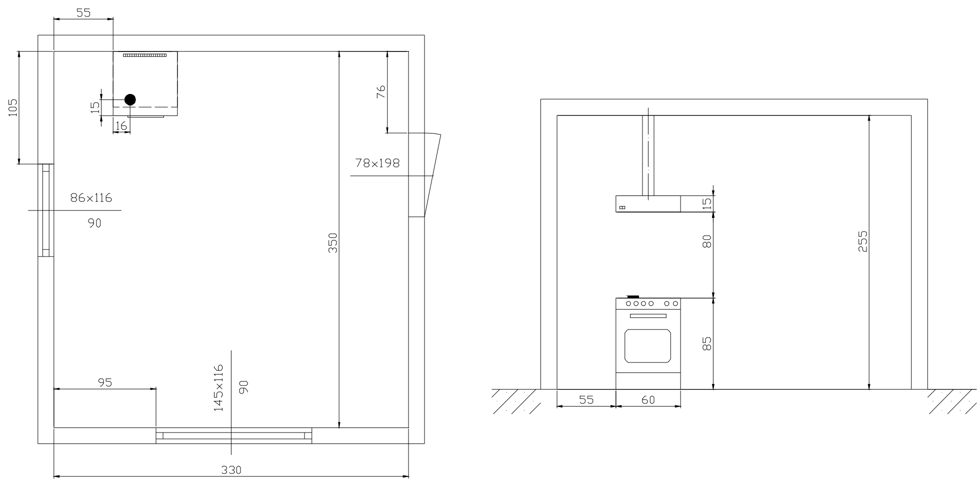

Figure 1. Plan of model kitchen (V_h = 29,5 m^3)

Table 1. Investigated kitchen ventilation systems

Burner power P=5kW		
Air flow extracted by hood (h =80 cm)	Supply air terminal device	Pollution load
0 m^3/h	-	
60 m^3/h	1 x 60 m^3/h	G_{CO2}= 1,80*10^{-4} kg/s
180 m^3/h	2 x 90 m^3/h	G_{NO2}= 2,48*10^{-8} kg/s
250 m^3/h	2 x 125 m^3/h	G_{NO} = 7,80*10^{-8} kg/s
500 m^3/h	4 x 125 m^3/h	

Thermal properties of stove have been studied in laboratory measurements as well, by measuring air temperature changing around the working stove at 10 points. At the same time surface temperature at different parts of stove envelope has been measured too.
To describe effect of increasing surface temperature approximate functions have been developed based on measurements' results.

Burner has been model by a heat source element with same heat exchanger area as the real one. Pollutant source have been placed above to the "burner". Three sources have been used for main gaseous products of burning: carbon-dioxide (CO_2), nitrogen-oxide (NO) and nitrogen-dioxide (NO_2).
For any sort of contaminant initial concentration is zero ($k_b(\tau=0)=0ppm$). Outdoor air has been taken into consideration as clean air.

Modelling grid is a main issue as regards accuracy of model. It is highly recommended to avoid any over dimensioning or using too poor grid system. In order to provide a proper solution at lower need of computer capacity, different regions have been worked up in the CFD model. In regions with high expected concentration, temperature or velocity gradients fine grid has been applied.
Seven regions have been defined with different grid:

1. Burning area (above top plate of stove)
2. Extract area (beside under plate of hood; extracting zone)
3. Front area (beside front plate of stove; primary occupied zone)
4. Left area (left side of stove; secondary occupied zone)
5. Right area (right side of stove; secondary occupied zone)
6. Envelope area (around the stove, along the surface of envelope)
7. Wall area (along inside surface of walls)

Total grid system of the model are contain 72 528 cells.

To describe time depending parameters time-step(s) has to be estimated. Total transient period (60min) has been divided into following steps: 2min, 5min, 10min, 15min, 20min, 30min, 45min and 60min. Data saved at 10th, 30th and 60th minutes.

Changing of calculated parameters (concentration, temperature and air velocity) has been followed at 10 monitor points placed in front of the stave, at 1.5m high along the centreline of the burner, in every 10cm.

RESULTS

Results are presented in table form and by visualisation as well, at above mentioned transient time step. At visualisation investigated areas have been selected according monitor points.
In following Figures results at 30min can be seen, in case of no ventilation and 250 m^3/h exhaust air flow.

- CO_2 concentration: Figure 2 a. and b.
- NO concentration: Figure 3 a. and b.
- NO_2 concentration: Figure 4 a. and b.

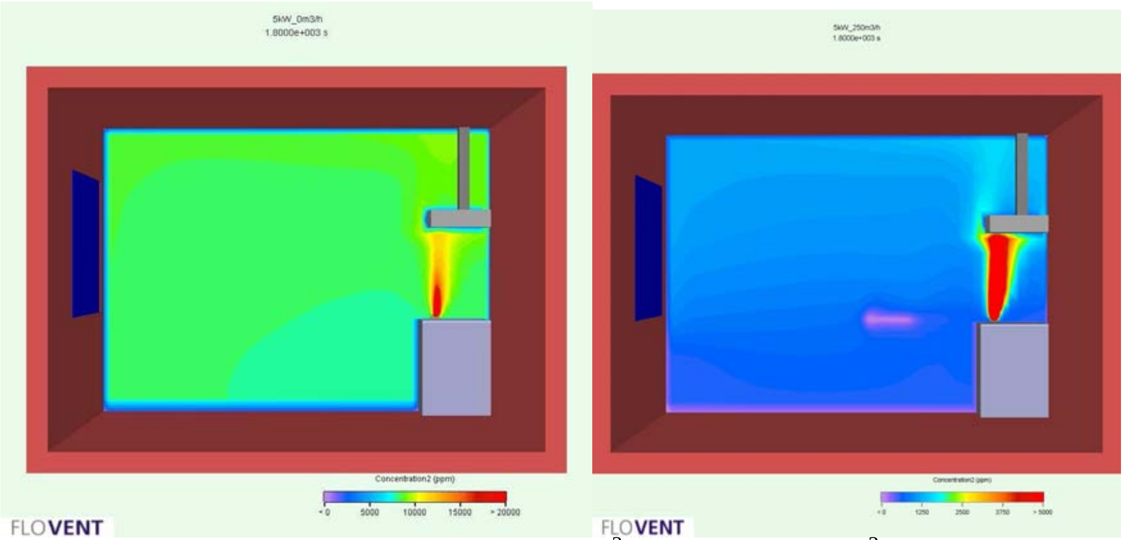

Figure 3. CO_2 concentration (30min) a.) V=0 m^3/h b.) V=250 m^3/h

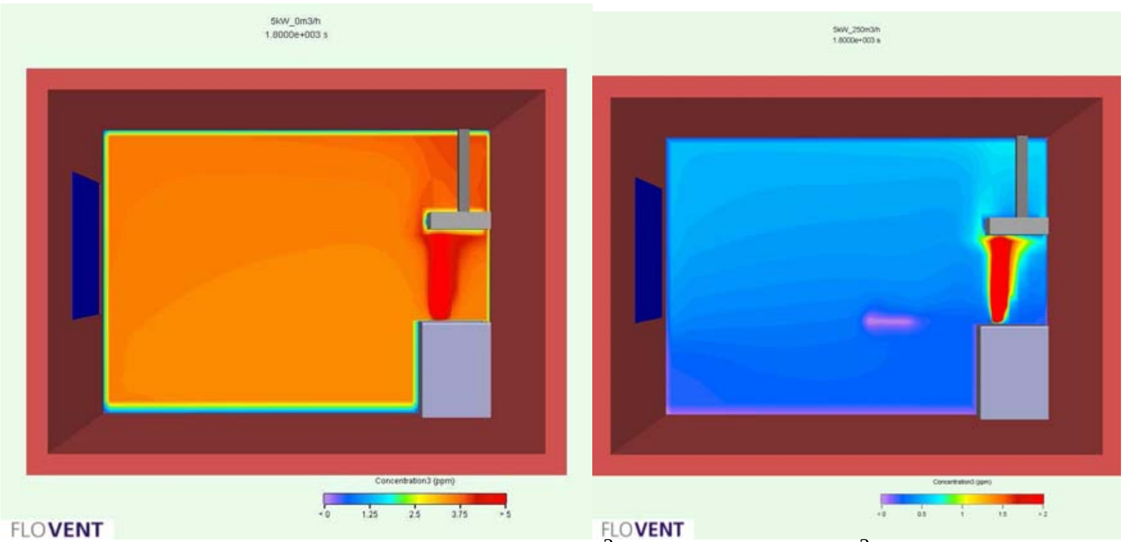

Figure 4. NO concentration (30min) a.) V=0 m^3/h b.) V=250 m^3/h

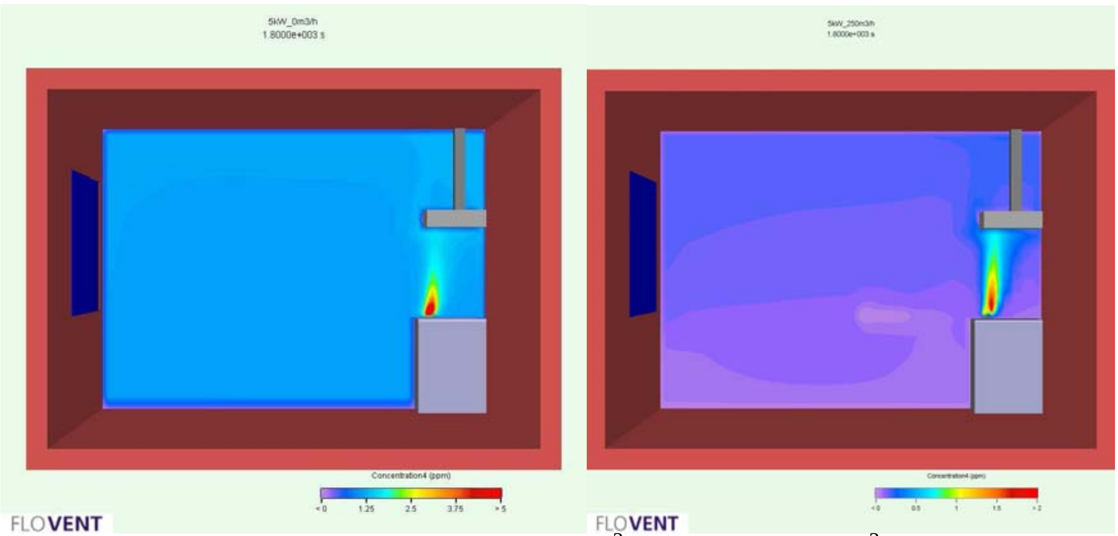

Figure 5. NO_2 concentration (30min) a.) V=0 m^3/h b.) V=250 m^3/h

CONCLUSIONS

- In kitchens equipped gas stoves it is strongly recommended to install mechanical ventilation system to provide required concentration level of CO_2 and NO_x.
- Using kitchen stoves as heating equipment results significantly decreasing indoor air quality level.
- Kitchen hoods with typical technical parameters ($V_{ex,max}$ =180–250 m^3/h) can not provide Hungarian requirements for NO_x concentration (200 μg/m^3)
- As regards CO_2 concentration required air flow is given in Table 2.

Table 2. Required exhaust air flow

Working period [min]	V_{ex} [m^3/h]	
	c_{CO2} < 2 500 ppm	c_{CO2} < 5 000 ppm
15	65,0	0,0
30	140,0	66,0
60	160,0	92,0

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Determination of phthalates – effects of extraction parameters on recovery

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SUMMARY

Phthalates are ubiquitous contaminants in indoor environments, and are reported to cause adverse health effects. Due to their semi-volatile nature, phthalate samples require a multi-step preparation before quantification with GC-MS. Many different sampling media, extraction procedures, and solvents are utilized in phthalate analytics, while comparisons and justifications for each choice are lacking. In this study, phthalate recoveries were compared from glass fibre filter (GFF), XAD resin + GFF, and polyurethane foam (PUF) after ultrasonic treatment (US), and Soxhlet extraction (SX) with different solvents. The recoveries varied between phthalates, sampling media, solvents, and extraction methods. The overall best performance was obtained using US and toluene (TOL) for GFF; SX and TOL for XAD+GFF; and dichloromethane and US for PUF. The recoveries ranged between 76 and 131 %. Each of these combinations is suitable in terms of recovery, whereas selecting the optimum sampling medium to be used in the air samplings requires further research.

INTRODUCTION

There is a growing concern over a wide variety of chemicals in our living environment and their possible adverse health effects. Among these compounds are phthalates, the worldwide consumption of which is in the range of millions of metric tonnes per year. The phthalates are used as plasticizers in various building materials, furnishings, and household products, such as PVC floor coverings, wallpapers, cosmetics, toys, rain coats, and footwear. Recently, environmental organizations have raised these compounds into people's awareness by detecting these compounds from blood samples of various study groups (politicians, and females in different age groups). Also in the scientific literature, these compounds / their metabolites have been reported in the environment, in indoor air and dust samples, and in humans. This is a consequence of escape from the materials the phthalates were added to by volatilization, leaching or abrasion. The major concern related to phthalates is their suspected toxicity to reproduction [1-2]. Also, asthma and allergies have been suggested to be associated with two phthalates; di(2-ethylhexyl)phthalate (DEHP) and butylbenzylphthalate (BBP), respectively [3].

Research groups around the world have made great efforts in setting up analytical methods for the determination of phthalates from air and dust samples. Due to their semi-volatile nature (Table 1), phthalate samples require a multi-step preparation before quantification with gas chromatography-mass spectrometry (GC-MS). Many different sampling media, extraction procedures, and solvents are utilized in phthalate analytics, while comparisons and justifications for each choice are lacking. Only after large databases on phthalate emissions and concentration levels in the (indoor) environment have been gathered, it is possible to estimate the risks related to exposure via respiratory tract and skin. Eventually, this will also increase possibilities for material and product development.

The aim of this study was to address the problematic issues of phthalate analysis by comparing phthalate recoveries from glass fibre filter (GFF), XAD resin + GFF, and polyurethane foam (PUF) after ultrasonic treatment (US) and soxhlet extraction (SX) with different solvents.

METHODS

Reagent and material preparation

Clean glassware were either rinsed with ethanol, hexane (HX), and toluene (TOL) or baked at 450 °C for 4 h prior to use and wrapped in aluminium foil. Caps and metal equipment were rinsed with acetone or ethanol. Glass fiber filters (GFF, ø 25 mm, pore size 1 µm, Gelman Sciences, Ann Arbor, MI, USA) and cotton wool for filtering liquids were sonicated with dichloromethane (2×20 min). Polyurethane foam (PUF, ø 18 mm×5 cm, Klaus Ziemer GmbH, Langerwehe, Germany) was cleaned using Soxhlet extractor with hexane (16 h). XAD-2 resin (Amberlite XAD-2, Supelco, Bellefonte, PA,USA) was rinsed with hexane and the solvent residue was removed using purified nitrogen. The filter cassettes (ø 25 mm, Millipore) were washed with purified water and dried in nitrogen flow. The XAD-GFF sampler was prepared right before the experiments by packing 1.7 g of XAD-2 resin on filter cassettes covered with a nylon net filter (ø 25 mm, pore size 180 µm, Millipore, Billerica, MA, USA), and a GFF was placed on the top to cover the XAD-2 resin.

The solvents (HX, TOL, dichloromethane (DCM), acetone, ethanol) used were of p.a. or HPLC grade. Phthalates and the internal standard of analytical grade (Table 1) were supplied by Sigma-Aldrich and Fluka (Steinheim, Germany), and the recovery standard (deuterated phenanthrene) was obtained from Sigma-Aldrich. Stock solutions of the chemicals were prepared in TOL at concentrations from 0.6 to 1.9 mg/ml, and analytical standard solutions were made from appropriate dilutions of these stock solutions (target concentrations were 103-191 µg/ml).

Table 1. Properties of phthalates.

Phthalate (abbr.); Cas RN	Formula; Molecular weight (g/mol)	Boiling point (C°); vapour pressure (mmHg) @ 20 °C	Phthalate (abbr.); Cas RN	Formula; Molecular weight (g/mol)	Boiling point (C°); vapour pressure (mmHg) @ 20 °C
dimethyl phthalate (**DMP**); 131-11-3	$C_{10}H_{10}O_4$; 194,19	284; 0.01 (@ 25 °C)	butyl benzyl phthalate (**BBP**); 85-68-7	$C_{19}H_{20}O_4$; 312,39	370; 6*10-7
diethyl phthalate (**DEP**); 84-66-2	$C_{12}H_{14}O_4$; 222,24	298; 7,5*10-3	dicyclohexyl phthalate (**DCHP**) **; 84-61-7	$C_{20}H_{26}O_4$; 330,42	n.a.; 0,1 (@ 150 °C)
*dipropyl phthalate (**DPP**); 131-16-8	$C_{14}H_{18}O_4$; 250,29	305; n.a.	n-dioctyl phthalate (**DNOP**); 117-84-0	$C_{24}H_{38}O_4$; 390,6	220 (390-420); 0.2 (@ 150 °C)
diisobutyl phthalate (**DIBP**); 84-69-5	$C_{16}H_{22}O_4$; 278,35	320; 7,5*10-5	bis(2-ethylhexyl) phthalate (**DEHP**); 117-81-7	$C_{24}H_{38}O_4$; 390,6	385; $7,5*10^{-6}$
dibutyl phthalate (**DBP**); 84-74-2	$C_{16}H_{22}O_4$; 278,35	340; <7,5*10-5			

* = internal standard; ** = solid; n.a. = not available.

In preliminary tests of 3 replicate injections to the GC-MS system, the repeatability (relative standard deviation, RSD) of the response of 5 phthalates dissolved in TOL at 8 concentrations ranging between 0.01 and 10 µg/ml was better than that obtained when HX was used as a solvent. The RSDs were <12 % with TOL whereas the RSDs ranged between 17 and 46 % with HX. However, the instrumental detection limits were lower (0.01 µg/ml) with HX for all the 5 phthalates compared to the detection limits obtained with TOL (0.1 µg/ml for 4 phthalates and 0.5 µg/ml for BBP). Each compound showed linearity in the range between 0.1 and 5 µg/ml only with TOL. Thus, TOL was chosen as the solvent for GC-MS analysis in the following experiments.

Recovery tests

For the recovery tests, GFF, XAD-GFF samplers, and PUF plugs were spiked with 2.1-3.8 µg of each phthalate and 2.9 µg of internal standard (DPP), and immediately extracted with one of the tested solvents (TOL, HX, or DCM) using ultrasonic (US) or soxhlet (SX) extraction. However, the PUF plugs were not extracted with TOL because the plugs were found to get deformed and loose their composition in the preliminary tests.

After ultrasonic extraction (2×5 ml×20 min), the samples were filtered through a clean cotton wool wad plugged in a Pasteur pipet. After filtration, samples were evaporated at room temperature (RT) using a gentle stream of nitrogen. In one experiment, when GFF was extracted with US-TOL, nonane was added to the sample to prevent sample losses during solvent evaporation with nitrogen. To the 1 ml volume of each sample in TOL, 0.57 µg of phenanthrene-d10 was added as a recovery standard before GC-MS analysis (Table 2) to measure the recovery of the internal standard.

For comparison, the spiked samples were extracted with 150 ml of one of the tested solvents for 16 h with Soxhlet extractor. Nonane was added onto the samples after Soxhlet extraction, and the volume of the extracts was reduced to <1 ml using rotary evaporator at RT under reduced pressure. The samples were then transferred to test tubes with TOL and the excess solvent was evaporated at RT with nitrogen. The samples (V=1 ml) were transferred to GC vials, and recovery standard added as with the US extraction. Quantification was carried out using GC-MS (Table 2).

The measured phthalate concentrations were compared to the spiked quantities to determine the recoveries (ARR, relative analyte recovery rate). For DPP, the recovery was calculated by comparing the ratio of the peak areas of internal and recovery standards in a sample with the corresponding ratio obtained in the analysis of the analyte standard solution. The criteria for the best extraction method and solution for each sampling material included the uniform distribution of all the determined phthalates around the 100 % optimum recovery.

Table 2. GC-MS conditions for the determination of phthalates.

Gas chromatograph - mass spectrometer	HP 5890 GC + HP 5970 MSD
Column	HP-5MS (28 m × 0.25 mm i.d., 0.25 µm film thickness; Agilent Technologies, J.W Scientific, USA)
Injection volume	2 µL
Injector temperature program	275 °C (constant flow)
Carrier gas	Helium, 1 ml/min
Purge gas	Helium, 40 ml/min
Oven temperature program	70 °C (2 min), 25 °C/min to 200 °C, 5 °C/min to 280 °C (held 10 min)
Ionization source	Electron impact ionization (70 eV)
Interface temperature	280 °C
Selected ions (at *m/z*)	DMP:163,194, DEP:149,177,178,222, DPP:149,191,209:, DIBP:149,167,205, DBP:149,205,223, BBP:149,178,206, DCHP:149,167,191,249, DNOP:149, 161,167,279, DEHP:149,167,191, FEN*d10*:188,180

RESULTS

The extraction recoveries varied with respect to the phthalate, sampling material, solvent, and extraction method (Table 3). The loss of internal standard (ARR_{DPP} 12-45 %) was pronounced when samples were extracted with HX, and also in the experiment with PUF-SX-DCM. Thus, the quantitation of the analytes could not be reliably performed in those experiments.

Table 3. The extraction recoveries of phthalates from diverse sampling materials following different extraction methods and extraction solvents.

Sampling material	**Extraction**		**Extraction recovery, %**								
	method	**solvent**	**DMP**	**DEP**	**DIBP**	**DBP**	**BBP**	**DCHP**	**DEHP**	**DNOP**	**DPP***
Added amount (ng) on each sample			*3,41*	*2,84*	*2,26*	*2,71*	*3,82*	*2,06*	*2,80*	*2,29*	*2,99*
GFF	SX	DCM	55	84	113	132	105	104	368	107	72
	SX	TOL	157	187	127	199	221	208	201	261	57
	US	TOL+n	84	120	101	98	93	106	136	74	79
	US	TOL	57	22	82	104	97	114	156	142	67
	US	DCM	25	63	115	126	151	162	317	225	80
XAD+GFF	SX	TOL	92	97	102	90	107	94	4860	108	69
	SX	DCM	46	75	124	214	97	92	9370	194	72
	US	TOL	72	90	111	119	131	129	187	136	55
	US	DCM	61	82	109	121	132	139	513	179	59
PUF	US	HX	25	63	115	126	151	162	317	225	60
	US	DCM	84	88	94	108	76	131	744	198	67

*= internal standard; GFF=glass fiber filter; XAD=XAD-resin; PUF=polyurethane foam; US=ultrasonic extraction; SX=Soxhlet extraction; TOL=toluene; DCM=dichloromethane; HX=hexane; n=nonane.

Another problem arose with the obvious contamination caused by DNOP and DEHP indicated as extremely high recoveries in several samples (Table 3). Therefore, the results on DNOP and DEHP were excluded when the best extraction method for each sampler was determined.

The best average recoveries of 6 phthalates and DPP using US were obtained with TOL+nonane for GFF samples (ARR 84-120 %) and with DCM for PUF samples (76-131 %), whereas SX was the best method for XAD+GFF when extracted with TOL (ARR 90-107 %). In these samples, recovery of the internal standard ranged from 67 to 79 % (Table 3). Generally, the recoveries DPP and DMP were the lowest regardless of the extraction parameters. The loss of DPP was controlled, and the recoveries of DMP, DEP, and DIBP were improved when nonane was added to the US sample compared to the extraction without nonane (Table 3). The glass fiber filters should not be extracted with US+DCM, and the SX+DCM combination seems to be the worst choice for the extraction of all the tested phthalates on XAD+GFF. The HX+US was not the optimum extraction combination for PUF samples. However, each of the three last mentioned less-optimal extraction combinations may work satisfactory when only few phthalates are targeted, but with respect to detection of all 6 phthalates at the same time, the recoveries from the samplers can be optimised by choosing other extraction parameters.

DISCUSSION

In the construction and furnishing products, such as floor and wall coverings, wire and cable insulation, and electrical conduits, vinyl plastics are widely used because of their durability, easy installation, and cost-effectiveness. The phthalates make possible the flexible, colour-fast, durable, and low-maintenance qualities of vinyl. Unfortunately phthalates migrate from the products, and have been recognized as ubiquitous contaminants in the environment. Due to their potential impact on human health, standardized procedures are warranted to monitor the exposure levels to these compounds in dwellings and workplaces. The large variety of sampling adsorbents and extraction procedures utilized in the indoor air phthalate analytics emphasises the need for their reciprocal comparison. This study was attempted to evaluate the capacity of two common extraction methods to recover phthalates from three frequently used sampling media.

In this study, three solvents, HX, DCM, and TOL, were compared. The loss of the internal standard in HX experiments indicated that HX may not be the best choice for an extraction solvent. This can also be deduced from the literature, as HX has been used as an extraction solvent only as a mixture with diethyl ether [4-6]. On the other hand, acetone that was not evaluated in the present study might be another suitable solvent and worth comparing [7,8]. The selection of DPP as the internal standard may be justified with the fact that it has not been extensively found in the environmental samples [6,9]. The recoveries of DPP ranging from 67 to 79 % in the best extraction combinations for each sampling material indicate some losses during sample handling, and this may also be the case for DMP and DEP having lower boiling points than DPP (Table 1). Thus, due to similarities in volatility, DPP is a suitable internal standard for at least DMP and DEP. Other researchers have described losses of DBP during evaporation of samples [10] but this was not evident in these experiments. However, it might be advisable to use another internal standard, e.g., a deuterated one, to detect possible losses during sample handling of those phthalates with boiling points in the range of 320-390 °C.

The contamination caused by DNOP and especially DEHP in this study was partly traced to plastic sample holders and a contaminated nonane reservoir. Xie et al. [8] suggest that all solvents be distilled for purification and stored thereafter in full glass bottles. These authors also used active carbon cartridges to filtrate the air entering the Soxhlet apparatus and rotary evaporator, and XAD-2 resin in connection with nitrogen evaporation to catch any impurities from the nitrogen and the bleeding of tubings. In addition, in the field sampling, the adsorbent holders should be made of glass instead of plastic to avoid contamination thereof. These precautions and collection and analysis of laboratory and field blanks should be routinely performed in the phthalate analytics to control and identify potential contamination sources and to verify the authenticity of the results.

With respect to the recoveries of DMP, DEP, DIBP, DBP, BBP, and DCHP, the overall best performance was obtained using US and TOL for GFF; SX and TOL for XAD+GFF; and DCM and US for PUF. The recoveries ranged between 76 and 131 %. These conclusions are based on single experiments. The recoveries of DNOP and DEHP with the presented extraction procedures should be verified as well. Even though the sampling media and/or the extraction methods and solvents have varied in other studies, the reported recoveries for these compounds have ranged between 78 and 138 % (Table 4) suggesting that these are probably the reasonably obtainable and acceptable recoveries in the phthalate analytics. The recoveries exceeding 100 % may be due to matrix effects: residues of the sampling media that remain in the extract protect the analytes from adsorbing, and thus, allow the injection of a higher concentration into the GC column compared to the concentration injected from a “clean” analyte standard solution prepared by dissolving analytes in a solvent. It might therefore be advisable to prepare quantification standards following the same procedure as with the sample preparation.

Table 4. Phthalate recoveries (%) reported in international studies.

Sampling material [ref]	**Extraction method: solvent**	**Recovery (%) for**							
		DMP	**DEP**	**DIBP**	**DBP**	**BBP**	**DCHP**	**DEHP**	**DNOP**
XAD [7]	elu:DCM	84	91	-	101	90	-	135	92
PUF+XAD [8]	SX: HX+DE (9:1)	138	131	123	117	78	-	99	108
GFF+Empore C18 filter [9]	sta+US: TOL+AC (3:7)	91	100	97	92	90	100	90	-
Quartz fiber filter+ Empore disk C18 filter [11]	US+shake:AC	97	98	98	101	105	102	106	-
Charcoal [12]	US:TOL	-	110	-	120	94	110	98	-
Silica gel [13]	elu:AC	88	91	95	-	99	94	106	-

- = not determined; AC=acetone; DCM=dichloromethane; DE=diethyl ether; ELU=elution; sta=static elution; SX=Soxhlet; TOL=toluene; US=ultrasonic extraction.

In conclusion, the occurrence of phthalates everywhere including laboratory environments causes high risk of contamination during the analysis. This and the lack of standardized methods for sampling and analysis of phthalates in indoor environments pose extra challenges in the phthalate analytics. However, this study showed that various sampling media can be utilized in conjunction with optimum extraction technique and solvent. The analytics enables quantification of phthalates in indoor environments and eventually also offers possibilities to develop products and materials with lower phthalate emissions. The suitability of the sampling media and analytics tested in this study for field sampling warrants further study.

ACKNOWLEDGEMENT

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External Environment and Indoor Microclimate

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SUMMARY

The article deals with air parameters in building exterior and interior. External air temperature is the most important air parameter in the exterior. The temperature is influenced by local area parameters. The annual external air temperature measurement in city centre and periphery in Central Europe is presented. The article also contains multizone modeling and simulation of indoor air quality during different boundary conditions (external environment), various occupations of interior and different types of ventilation and ventilation air amount.

INTRODUCTION

Indoor environmental quality depends on technical equipment ability adjust external variable parameters of entering factors to parameters not such much variable. External parameters correspond to climate region and local area quality. The urban environment quality is different in the city centre, suburb and surrounding intact areas. The differences are in the air quality, air temperature as well air humidity.

Indoor air quality and pollutant concentration is significant for human health because people spend most of their time in buildings. The aim for heat losses lowing directed to limiting natural ventilation by windows. Tight windows have insufficient infiltration; they are unsuitable from the hygienic point of view. It leads to pollutant concentration and relative humidity enhancement, mould reproduction and environment not corresponding to human organism. So it is necessary to ensure mechanical ventilation during people presence.

Indoor air quality depends on many factors, especially on: outdoor air quality, air amount per person or ventilation rate, ventilation plant, amount of air pollutants. Carbon dioxide is the most important indoor pollutant whether the pollutant source is presented only by people.

The study is focused on some external and internal air parameters.

METHODS

The external air temperature is the basic parameter with direct influence to the building energy demand. The external air temperature is fluctuated and local temperature change especially depends especially on the surface properties.

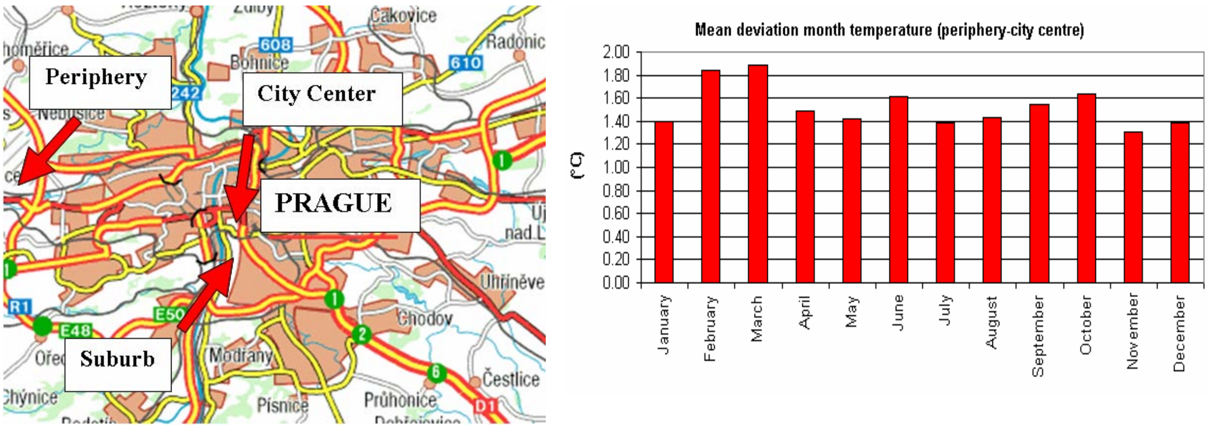
The solar reflection and absorption is markedly depended on a surface colour. Simply we can say that the darker surface increases energy absorption and, in dependency on a thermal capacity, the energy is accumulated. The solar radiation is a thermal energy source, especially direct part of the global solar radiation.

This effect causes warming up of surfaces with solar radiation absorption and thermal conduction to the deeper layers and the surfaces radiate thermal energy during a period with lower solar intensity with a time delay. The asphalt roads in zone with people or pedestrian zones with usually black colour are the biggest accumulators of thermal energy. The building

operation is the next energy source in the city centres and the building heat losses are becoming energy gains for local environment.
Next technology equipment operation in a relationship with buildings and transport are not negligible. Technical equipments provide stable parameters of the internal environment and warm up or cool down environment. The cars produce the thermal energy in generally too. Differences between climate in cities and countryside are evident from mentioned reasons.

Measurement

The external air temperature increasing is significant especially during summer period in the city centres in Central Europe conditions.
While the temperature increasing during winter is not so significant trouble-it decrease energy consumption for heating, warmer summer period is negative and the active cooling systems must be installed. The air temperature change is simple, but higher temperature difference increase energy consumption. Negative effect can be supported by unsuitable building design with high glass ratio and without passive shading or low energy cooling systems.



a) b)

Figure 1. External air temperature measurement: a) City plan b) Mean deviation-month temperature city centre-periphery

Urban heat islands measurement was already done in countries with hot summer (temperature increases 30°C) and with mild winter where summer effect is significant.
Long time measurement of the external air temperature was done in Prague during year 2005 and partly in year 2006. The temperatures were measured in the city centre, suburb and periphery.

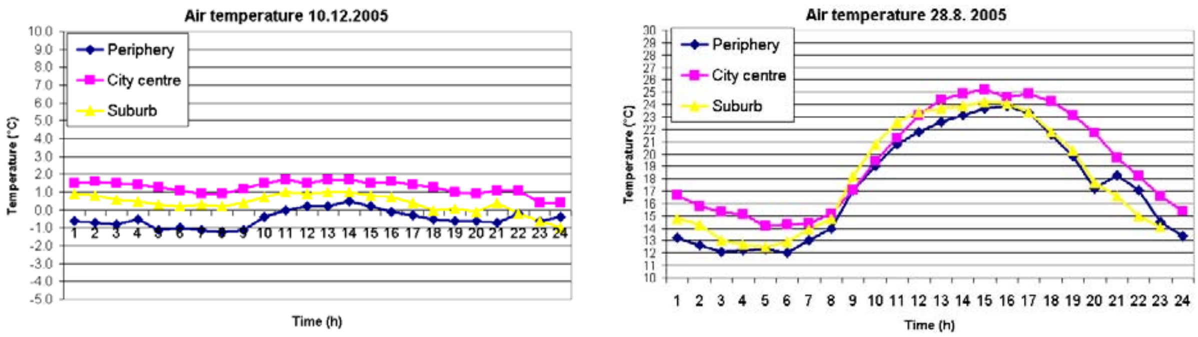


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Figure 2. External air temperature: a) Winter b) Summer (influence of direct solar radiation)

Outdoor air quality influence on ventilation air amount

Outdoor air quality has an influence on ventilation air amount for exhausting pollutants. The worse outdoor air quality means the more ventilation air amount. Carbon dioxide is the most important pollutant in interior. Ventilation air amount dependence on outdoor carbon dioxide concentration shows Figure 3. The relation is exponential. The difference between the country (330 ppm) and large city (450 ppm) for one person is small, but for whole building the difference will be large.

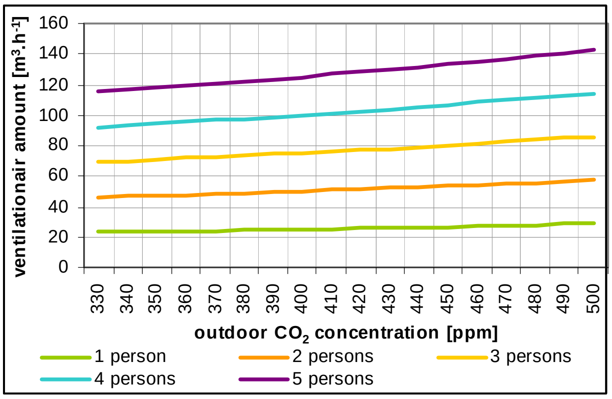


Figure 3. Ventilation air amount dependence

Model of the exterior microclimate

Many models were developed for time acceptable parametric solving city microclimate in history. But many of them were applicable only in the area of their origin. Individual variables was not purify from constants derived from the measurements. CTTC (Cluster thermal time constant) model is the most hopeful from the unit of parametrical models. Model is based on the Hoffman model.

The principle of the model is split the urban area to separate clusters and parameters calculate separately in the cluster with influence of surface properties, solar radiation, and convection to the climate.

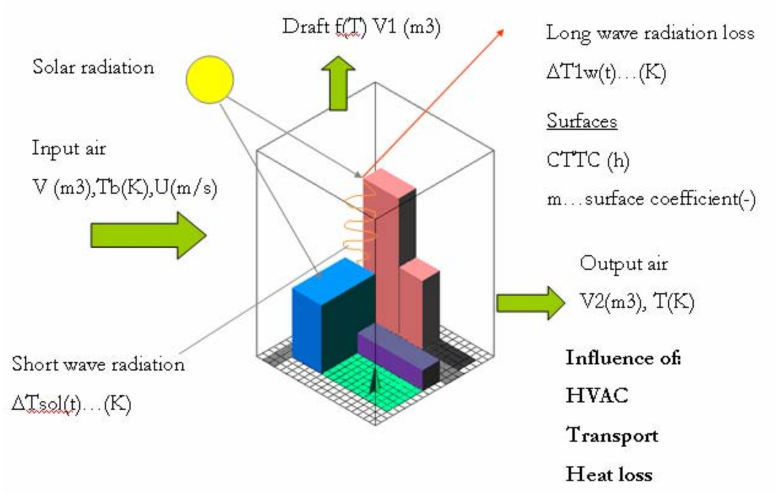


Fig. 4 CTTC model (urban climate simulations)

Model of carbon dioxide concentration

This part is about modeling and simulation of five different alternations of ventilation of two-rooms flat in Contam 2.4. This program helps to specify airflow – infiltration, exfiltration, flowing among particular rooms during natural ventilation and mechanical ventilation, wind pressure on building facade. It is possible to count pollutant concentration on the base of knowledge of the quantity airflow and pollutant production. Optimal air distribution is considered.

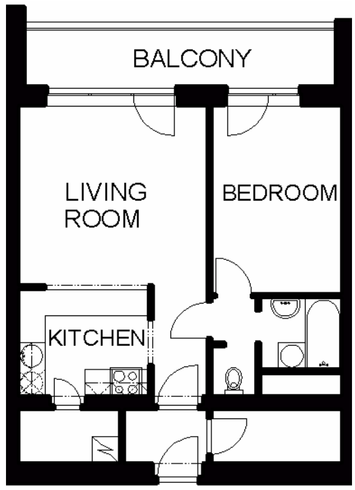
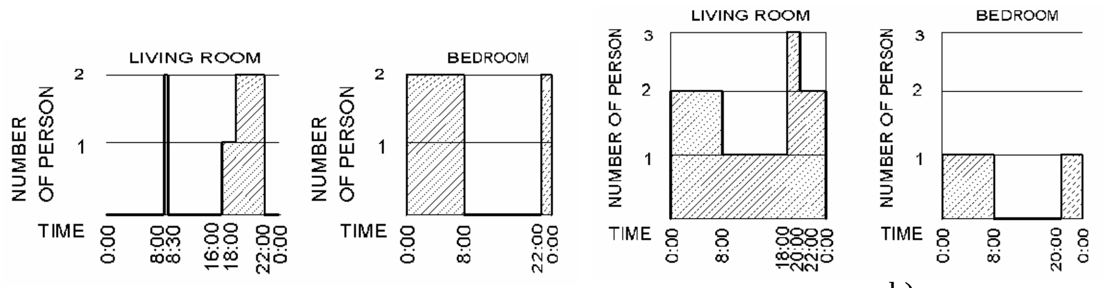


Figure 5. The plan of the flat

The plan of the observed flat shows Figure 5. The detected indoor pollutant is carbon dioxide. It was counted with outdoor carbon dioxide concentration 350 ppm and respiration production 20 $l.h^{-1}$ per person. Living room volume is $67m^3$ and bedroom volume is $31,2m^3$. The flat was studied in two alternations: "A" - which is occupied by two persons and "B" - which is occupied by three persons due to figure 6. Carbon dioxide concentration was observed in living spaces (a living room and a bedroom), where occupants spent most of their time.



a) b)

Figure 6. Space occupation a) for flat "A", b) for flat "B"

Five ventilation alternations were selected for ventilation assessment. Alternations characterize not only hygienic rules but also ventilation used in current buildings.

Alternation no. 1 – ventilation according to ventilation air amount per person due to Pettenkofer`s rule 25 $m^3.h^{-1}$ per person

Alternation no. 2 – hot-air heating and ventilation with circulation (75 % circulation air)

Alternation no. 3 – exhausting sanitary equipment according to DIN 1946-6 with 0,45 h^{-1} ventilation rate

Alternation no. 4 – infiltration by "untight" windows, flow coefficient i_L=1,4*$10^{-4}m^2$ $s^{-1}Pa^{-0,67}$ and exhausting sanitary equipment according to ČSN 74 7110

Alternation no. 5 – infiltration by "tight" windows, flow coefficient i_L=0,1*$10^{-4}m^2$ $s^{-1}Pa^{-0,67}$ and exhausting sanitary equipment according to ČSN 74 7110

It is considered: the air flows into interior in spite of tight windows, none windows opening during people presence (difficult behavior estimating).

RESULTS

External environment

The solution of oversized warming is decreasing of insolate surfaces absorption. Green fields could be use for shading or moisture production (evaporation). The wind has the high influence to the thermal situation in the city, flow change temperature in given area. Generally, climate changes will increase energy consumption because energy for cooling is more expensive then energy for heating. Give a definition of maximal impact of building to neighbourhood could be set up. This parameter creates restriction for building with high glazing ratio and with active cooling systems. Environment friendly buildings are result of these parameters.

Indoor microclimate

Figures 7-10 show carbon dioxide concentration runs.

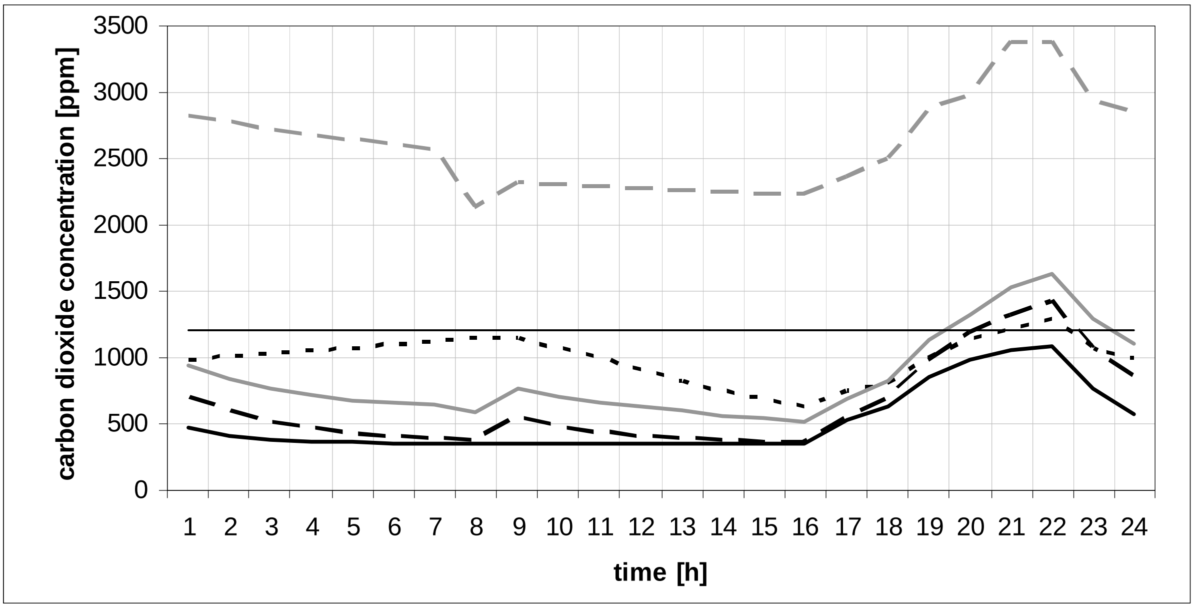


Figure 7. CO_2 concentration runs for living-room "A"

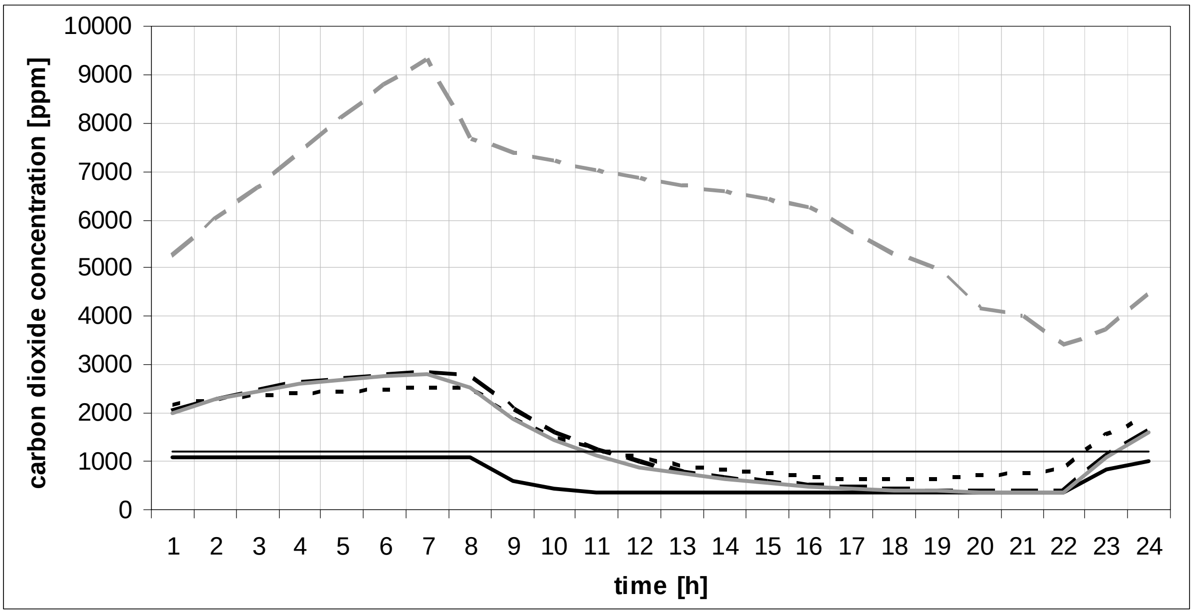


Figure 8. CO_2 concentration runs for bedroom "A"

Legend for figure 7 and 8:

Alternation no. 1 –Pettenkofer`s rule 25 $m^3.h^{-1}$ per person

Alternation no. 2 – hot-air heating and ventilation with circulation (75 % circulation air)

Alternation no. 3 – exhausting sanitary equipment according to DIN 1946-6 s 0,45 h^{-1} ventilation rate

Alternation no. 4 – infiltration by "untight" windows and exhausting sanitary equipment

Alternation no. 5 – – infiltration by "tight" windows and exhausting sanitary equipment

Requirement

The results show, that CO_2 concentration run is similar to interior occupation, zone volume and ventilation air amount. Increasing time of concentration to constant value and decreasing time of concentration to ventilation air concentration depend on room volume, pollutant production and ventilation air amount.

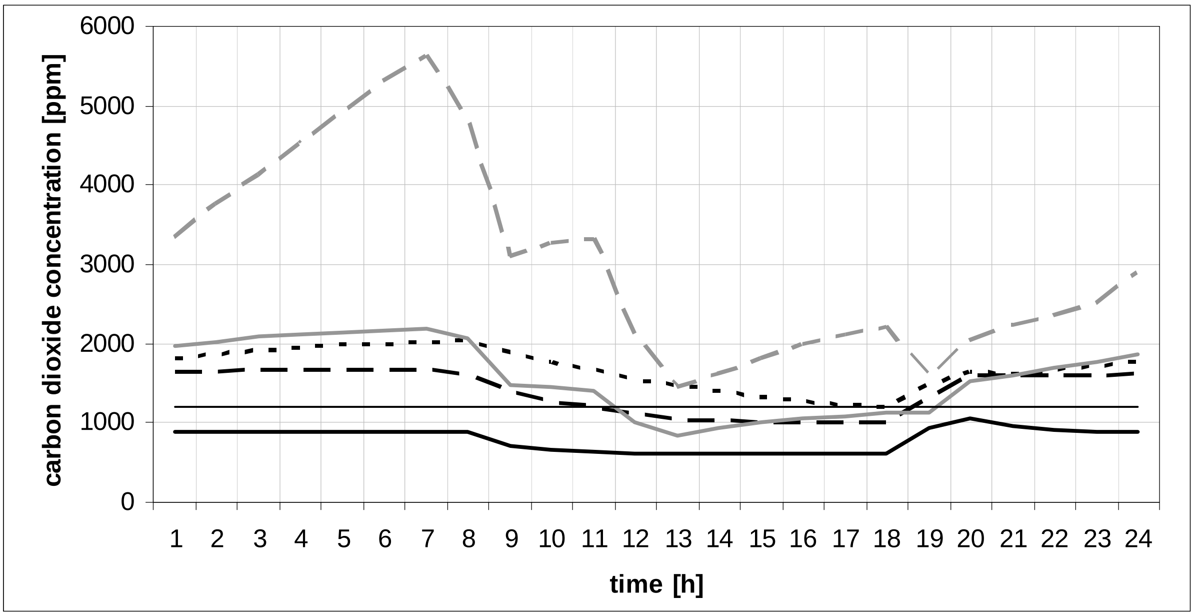


Figure 9. CO_2 concentration runs for living-room "B"

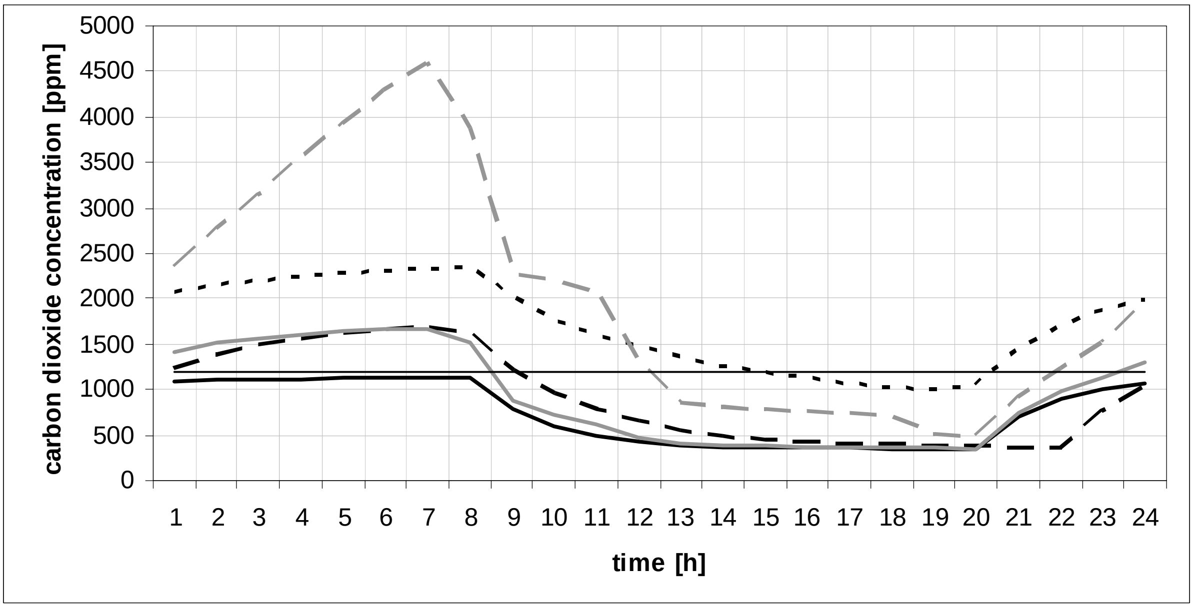


Figure 10. CO_2 concentration runs for bedroom "B"

Legend for figure 9 and 10:

Alternation no. 1 –Pettenkofer`s rule 25 $m^3.h^{-1}$ per person

Alternation no. 2 – hot-air heating and ventilation with circulation (75 % circulation air)

Alternation no. 3 – exhausting sanitary equipment according to DIN 1946-6 s 0,45 h^{-1} ventilation rate

Alternation no. 4 – infiltration by "untight" windows and exhausting sanitary equipment

Alternation no. 5 – – infiltration by "tight" windows and exhausting sanitary equipment

Requirement

No rule for indoor air quality is in the Czech Republic. Maximal concentration 1200 ppm was required in this study, this value is according to EN CR 1752 CEN for class „C“.
The worst alternation is ventilation by infiltration with tight windows (no. 5). Also alternation no. 4 is very bad. Maximal required carbon dioxide concentration is overstepped more than 90 % time of people presence. The best alternation is the first alternation. Alternation no. 2 is suitable only for living room in flat A. Maximal required carbon dioxide concentration is overstepped more than 90 % time of people presence in bedrooms, maximum is 2500 ppm. Required concentration is also overstepped in alternation no. 3, but it is better than ventilation by infiltration.

DISCUSSION

The weather forecast is possible with limited precision only because the border conditions are too much complicated. Speaking about cities scale, areas with potential unsuitable temperature change can be calculated from the city surface parameters.
Internal microclimate quality must be arranged without influence on the local exterior conditions.
For the reduction of unsuitable building design and building situating tools must be created. The tool can show energy requests to buildings situated in the city centre or in the periphery.
The results show, that CO_2 concentration runs is similar to interior occupation and ventilation air amount. Simulation in program Contam 2.4 showed, that necessary air amount per person for ensuring maximal CO_2 concentration 1200 ppm is 25 $m^3.h^{-1}$ per person. It was established that, Pettenkofer`s rule is valid.
For good indoor air quality, we should design ventilation air amount due to this rule 25 $m^3.h^{-1}$ per person.

ACKNOWLEDGEMENT

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11 June 2007 at 13:30 - 15:00

A02
Performance of ventilation systems

Effect of using low-polluting building materials and increasing ventilation on perceived indoor air quality

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SUMMARY

The potential of improving perceived air quality indoors was quantified when low-polluting materials are used and when building ventilation is increased. This was done by studying the relationships between ventilation rate and the perceived indoor air quality. A sensory panel assessed the air quality in test rooms ventilated with realistic outdoor air supply rates, where combinations of high- and low-polluting wall, floor and ceiling materials were set up. These materials were ranked as high- and low-polluting using sensory assessments of air quality in small-scale glass chambers, where they were tested individually. Substituting materials ranked as high-polluting with materials ranked as lower-polluting improved the perceived air quality in the test rooms. This improvement was greater than what was achieved by a realistic increase of the ventilation rate in the test rooms. Thus reducing pollution emitted from building materials that affects the perceived air quality has a considerable potential of limiting the energy for ventilation without compromising indoor air quality.

INTRODUCTION

There is a need for reducing the energy consumption worldwide. One initiative to reach this goal is the EU Directive 2002/91/EC Energy Performance of Buildings [1] that makes it obligatory to reduce energy consumption in buildings while taking into account the indoor environment. For most buildings this can only be achieved if the energy used for ventilation is also reduced, as it constitutes about 20-30 % of the total energy consumed in buildings today. This however may lead to reduced ventilation rates and increased levels of air pollution from buildings, people and their activities, and thus to poorer indoor air quality, which contradicts the requirements of the EU Directive. The obvious solution for these apparently opposing requirements would be to reduce the pollution sources indoors.

Several studies have previously investigated the effects of pollution emitted by building materials on the indoor air quality and related these effects to ventilation requirements [2]. However, there is a lack of systematic experiments, in which building materials are first ranked according to their pollution strength, e.g. by using methods applied in labelling schemes [3], and then the effect on the indoor air quality of using these materials in real rooms is examined. Such studies would make it possible to quantify to what extent using low-polluting building materials would reduce the energy used for ventilation of buildings without compromising indoor air quality. The main objective of an ongoing research project is to quantify this energy saving potential based on the effects on the perceived air quality. The specific objective of the experiments described in this paper is to systematically study, how the selection of low-polluting building materials affects the perceived air quality and to compare this effect with the effects of increased outdoor air supply rate on the perceived air quality.

METHODS

A sensory panel assessed the air quality in full-scale test rooms ventilated with three different outdoor air supply rates and polluted by typical building materials including wall, floor and ceiling materials; the materials ranged from high- to low-polluting. The relationships between the perceived air quality and ventilation rate were examined for different combinations of materials to assess the impact of using low-polluting materials and/or increasing ventilation rate on the perceived air quality.

The assessments took place in three similar test rooms with a floor area of 18 m^2 and a volume of 57.6 m^3, each constituting an independent unit. The test rooms are served by a HVAC system supplying outdoor air to each room through a duct system and ceiling diffusers; the air is exhausted through wall-mounted grills. There is no recirculation. Outdoor air is supplied to the test rooms by an air handling unit with a fan and is conditioned by an electric pre-heater; no filter is installed. The temperature of the supplied air is independently controlled for each test room by electric heaters mounted upstream of the ceiling diffusers. The rate at which outdoor air is supplied to the rooms is controlled by IRIS dampers with motorized shut-off dampers independently for each test room. The air is exhausted directly outdoors by duct fans. The rate at which the air is exhausted is determined by a pre-defined overpressure in the test rooms (relative to adjacent spaces) which is controlled by motorized dampers. The test rooms were fully refurbished a few months prior to the experiments: the ceiling and flooring materials were changed, walls were painted and sliding doors of laminated wood were installed, so that one larger space could be turned into three separate test rooms. The HVAC system was completed in the week prior to the experiments.

A special cabinet is mounted in each test room. In this cabinet pollution sources can be hidden, so that the exposure conditions are not revealed. Room air enters the cabinet through a slot close to the floor and is pulled through the cabinet by an axial fan mounted at the top, where the air is exhausted to the room air. The cabinet also contains ultrasonic humidifiers that are mounted on the rails above the space for the pollution sources, immediately upstream of the axial fan; the humidifiers are used to control relative humidity in the test rooms. The air circulation through cabinets ensures that the air in tests rooms is well mixed. The cabinets were completed in the week prior to the experiments.

Nine different building materials were used (Table 1 and Figure 1). They were selected using the results of preliminary experiments in which 20 building materials were screened individually in small-scale glass chambers, following the principles of the Nordtest methods [4,5]. The 20 materials were selected based on a review of studies reporting the relationships between ventilation rate and the perceived quality of air polluted by building materials [2]. The aim was to select wall, floor and ceiling materials that can be ranked in a range from high- to low-polluting. The nine materials were tested individually in small-scale glass chambers, CLIMPAQs [4], following the procedures used in preliminary tests outlined above. For that purpose eight glass chambers were placed in a 26.8 m^3 stainless steel chamber [6] ventilated with an outdoor air change rate of 57 h^{-1}, and the sensory panel assessed the air quality in the glass chambers. The temperature in the chamber was kept constant at 22±0.1°C; the relative humidity was not controlled and averaged 31±6%. Each material was tested at 3 different area-specific ventilation rates, i.e. the ratio between outdoor air supply rate to the area of material. Different area-specific ventilation rates were obtained by changing the surface area of materials and keeping the ventilation rate through glass chamber constant at 0.9 L/s. The area of materials tested in glass chambers were selected according to the principles of the Nordtest methods [4,5] - the area-specific ventilation rates in glass chambers were kept the same as in the test rooms ventilated with 1, 3 and 9 h^{-1} for floor and ceiling materials and 1.3, 4 and 12 h^{-1} for wall materials (Table 1); the higher air change rates for wall materials were due to limitations on the material loading that could be placed in a glass chamber. The sensory panel also assessed the air quality in empty glass chambers.

Nine different combinations of the selected 9 materials were tested in full-scale test rooms (Figures 2 and 3). In order to resemble typical indoor setting, each combination consisted of a ceiling, floor and wall material, and included both high- and low-polluting materials. The amount of materials set up in the test rooms corresponded to the actual area of ceiling, walls and floor in the test room (Table 1). A sensory panel assessed the air quality in the test rooms polluted by the combinations of materials and ventilated with three different outdoor air supply rates corresponding to the outdoor air change rate of 1.3±0.1, 2.8±0.1 and 6.4±0.2 h^{-1}. The sensory panel also assessed the air quality in empty test rooms, i.e. without materials set up in test rooms. During all assessments, the temperature in test rooms was kept constant at 22.2±0.3^{o}C; the relative humidity was not controlled and averaged 36±5%.

Table 1. Building materials used in the full-scale and small-scale experiments

Material		Area of material (m^2)			
Acronym	Description	Full-scale test room	Small-scale glass chamber, CLIMPAQ		
Ceiling 2	10 mm plain gypsum board covered with plastic coated material	18	0.113	0.338	1.013
Wood	14 mm beech wood parquets, untreated	18	0.126	0.372	1.106
Carpet 1	6.4 mm tufted loop polyamide carpet with supporting layer of polypropylene web and polypropylene backing				
Linoleum 2	2.5 mm linseed-oil-based flooring, 52 % wood meal	18	0.113	0.338	1.013
PVC	2.0 mm homogenous single layered vinyl flooring, reinforced with polyurethane				
Polyolefine	2.0 mm homogenous polyolefine-based resilient flooring, reinforced with polyurethane				
Gypsum board	13 mm plain gypsum board lined with cardboard				
Paint 1	Gypsum board painted with one coat (0.14 l/m^2) of water-based acrylic wall paint	52	0.243	0.730	2.187
Paint 2	Gypsum board painted with one coat (0.14 l/m^2) of water-based wall paint with linseed oil				

The materials were purchased about two months prior to the start of sensory assessments. The specimens of materials to be used in test rooms and glass chambers were prepared (cut and/or painted) 4-6 weeks prior to the beginning of experiments; the specimens were stored in a ventilated hall. During sensory assessments, the specimens of materials used in test rooms were hung on trolleys, which were placed in the cabinets, while the specimens of materials were placed in the glass chambers using a special supporting system; the backside of materials was not exposed. The materials were set up in test rooms and glass chambers about 21 hours prior to the sensory assessments. The sensory panel could not see the materials hidden in the cabinets in tests rooms or the materials in glass chambers, which were covered with aluminium screens.

The sensory panel of 38 subjects was recruited from among 50 applicants. The subjects were students at the Technical University of Denmark, aged on average 24 years; 42 % were females and 21 % were smokers. Ten subjects had previously participated in similar experiments, where sensory assessments of air quality were made. These person-related data were collected via a questionnaire filled in by applicants upon recruitment. Subjects received written and oral instructions concerning sensory assessments. The first day of experiments was a practice session, but this was not communicated to the subjects; the assessments collected during this day were discarded. The subjects assessed air quality, using continuous acceptability scale [7] printed on paper. To test whether the sensory assessments were drifting/stable, on each day of experiments the subjects assessed the air quality in two glass chambers, where the loading of Carpet 1 and Ceiling 2 was the highest (Table 1) and kept unchanged during the whole course of the experiment. No drift was observed.

Sensory measurements were made on 15 days in three consecutive weeks during November/December 2006, each day both in full-scale test rooms and small-scale glass chambers. Exposures were randomly assigned to subjects in a design balanced for order of presentation. Each substitution with low-polluting material was always tested at three different ventilation rates in the same room to minimize the impact of the possible differences in air quality between empty test rooms. The three different area-specific ventilation rates of the same material established in glass chambers were always tested on the same day of the experiment. The assessments in tests rooms were made immediately upon reaching a marked spot on the floor, about 2 m from the door. This procedure was used to standardise the position in the middle of the room and the approximate time spend in the test room prior to assessment of the air quality. The doors to the test rooms were closed during assessments. The subjects entered tests rooms one at a time. The assessments in glass chambers were made upon taking one inhalation of polluted air exhausted from the chamber through a diffuser. A break of at least two minutes was made between assessments in glass chambers and test rooms. The break was taken in a well-ventilated hall adjacent to test rooms and the stainless steel chamber with the glass chambers.

The ratings made on paper questionnaires were digitized; 10 % of the ratings were digitized twice for quality control. The acceptability scale was coded as follows: Clearly not acceptable = -1; Just not acceptable/Just acceptable = 0; Clearly acceptable = 1. Using individual ratings made by the subjects means were calculated separately for each condition assessed in the tests rooms and in the glass chambers. The mean votes of acceptability were plotted against the logarithm of air change rate (for the ratings performed in test rooms) or the logarithm of area-specific ventilation rate (for the ratings performed in glass chambers) [8]. This was done to examine the relationship between the perceived air quality and ventilation rates [2,9,10], which was then used to assess the effect of changing ventilation on perceived air quality and compare it with the effect of using low-polluting materials.

RESULTS

Figure 1 shows the mean assessments of acceptability of air quality in glass-chambers at different area-specific ventilation rates when the building materials were placed individually in glass chambers. The results show that the selection of the materials used in the combinations examined in the test rooms turned out well, because both high- and low-polluting materials were included. These materials can be ranked in the following order, starting with the highest-polluting material: Paint 2, Wood, Carpet 1, Linoleum 2, Paint 1, Gypsum board, Ceiling 2, PVC and Polyolefine. The least polluting was the empty glass chamber, for which the acceptability of air quality was assessed to be the highest.

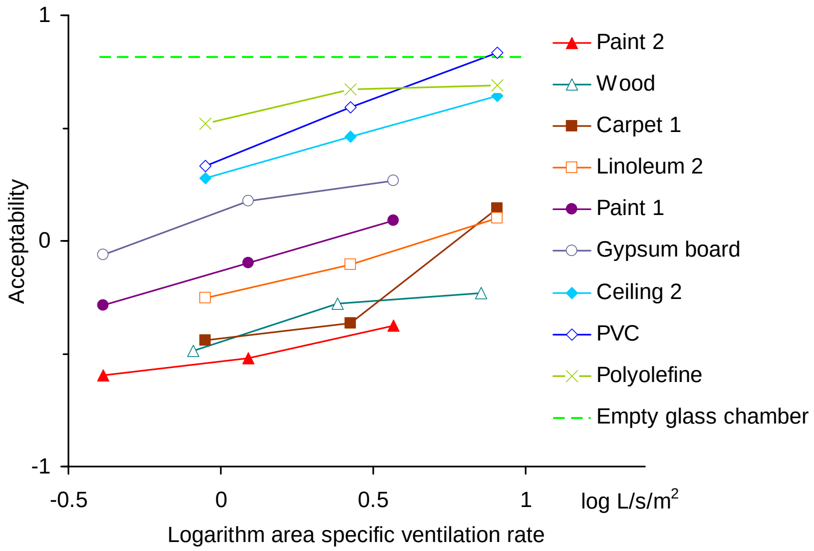
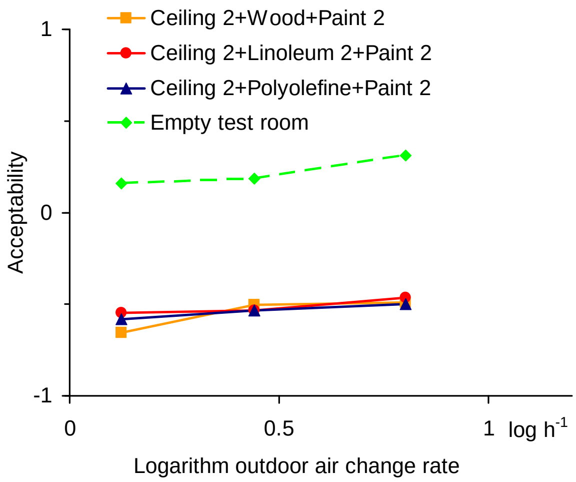


Figure 1. Acceptability of air quality as a function of the area-specific ventilation rate in small glass chambers containing the individual building materials that were examined in combinations in test rooms.

Figures 2 and 3 show the mean assessments of acceptability of air quality in test rooms at three different outdoor air change rates when the test rooms were empty and in the test rooms with different combinations of building materials including both high- and lower-polluting wall and floor materials, ranked by means of sensory assessments made in small-scale glass chambers; ceiling material was unchanged and always the same in all combinations examined in test rooms. The acceptability of air quality in empty test rooms was lower than in empty glass chambers, probably because the test rooms had undergone a renovation only a few month prior to the experiments and primary emissions were still affecting the perceived air quality.



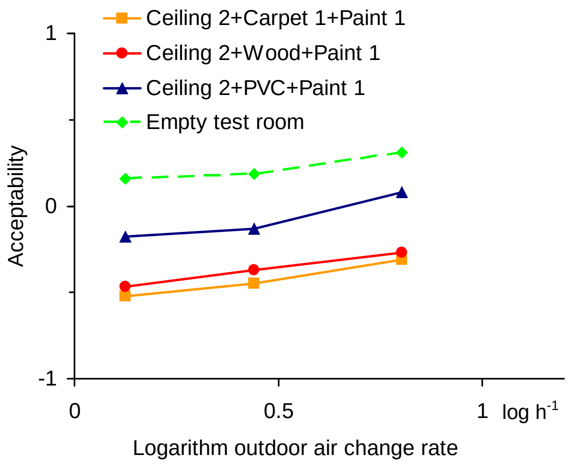


Figure 2. The effect of substituting high-polluting floor materials with lower-polluting materials on the acceptability of air quality in the tests rooms ventilated with different outdoor air change rates, when the combinations of materials included high-polluting (left) or lower-polluting (right) wall materials; ceiling material was unchanged.

Figure 2 left shows that substituting high polluting Wood with lower-polluting Linoleum 2 or Polyolefine did not improve the assessments of acceptability of air quality when combinations of materials included high-polluting Paint 2. When on the other hand the combinations included lower-polluting Paint 1 (Figure 2 right), substituting high-polluting Carpet 1 or Wood with lower-polluting PVC improved the assessments of acceptability of air quality.

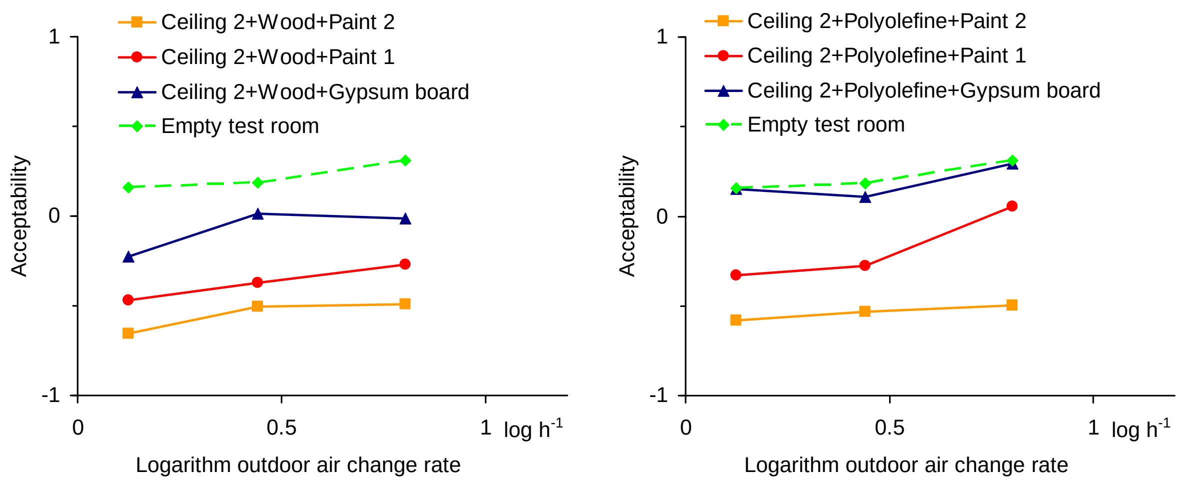


Figure 3. The effect of substituting high-polluting wall materials with lower-polluting materials on the acceptability of air quality in the tests rooms ventilated with different outdoor air change rates, when the combinations of materials included high-polluting (left) or lower-polluting (right) floor material; ceiling material was unchanged.

Figure 3 shows that the assessments of acceptability of air quality improved when the high polluting Paint 2 was substituted with lower-polluting Paint 1 or unpainted Gypsum board, independent of whether the combinations of materials included high-polluting Wood (left) or lower-polluting Polyolefine (right). However, the improvement seems to be somewhat greater in the latter case (Figure 3, right).

Figures 2 and 3 also show that increasing the ventilation rate improved the assessments of acceptability of air quality in the test rooms with different combinations of building materials. This effect was, however, small for the combinations of materials including high-polluting Paint 2.

DISCUSSION

The study shows that reducing pollution sources by selecting lower-polluting building materials, ranked by means of sensory assessments made in small-scale glass chambers, improves the perceived air quality in full-scale rooms where these materials are used. The improvement is greater than what was achieved by increasing the outdoor air supply rate in a realistic range. For example, a sevenfold increase of the outdoor air supply rate improved acceptability of quality of air polluted by a combination of materials including Ceiling 2, Polyolefine and Paint 1 less than substituting Paint 1 in this combination with lower-polluting Gypsum board (Figure 3, left). Such results can be provided for nearly all substitutions with lower-polluting building materials examined. The only exception was the substitution of high-polluting floor material with lower-polluting materials when the room air was still polluted by the high-polluting wall paint, Paint 2 (Figure 2, left). In this case the perceived air quality did not improve, probably because the strongest pollution source, in this case Paint 2, determines the perceived air quality in a room polluted by different materials. This phenomenon has also been observed for mixtures of odorous compounds [11] and is further illustrated in Figure 4, which combines the results of sensory assessments of air quality in test rooms and small-scale glass chambers in one chart. Combining both data was possible because the area of materials tested individually in glass chambers was calculated using the area-specific ventilation rates estimated using the actual dimensions and outdoor air change rates in the test rooms. Figure 4 shows that the assessments of acceptability of air quality in rooms polluted by different combinations of building materials were similar to the assessments for the material that was the strongest pollution source in the combination according to the ranking made by means of sensory assessments made in small-scale glass chambers: Paint 2 in the combination including Ceiling 2, Wood and Paint 2, and Wood in the combination including Ceiling 2, Wood and Paint 1. The exception was the combination of Ceiling 2, Wood and Gypsum board. In this case the air quality was assessed to be more acceptable than the assessments of air quality polluted individually by Wood, the strongest pollution source in this combination, suggesting an air cleaning effect, probably due to adsorption on Gypsum board which is known to be a strong sink [12]. A similar air cleaning effect is also observed in Figure 3, right, where the acceptability of quality of air polluted by Ceiling 2, Polyolefine and Gypsum board is similar to the acceptability of air in the empty test room. The results show that the decision, which materials to substitute with low-polluting alternatives, should be based on ranking of materials depending on their pollution strength; the highest polluting materials should be substituted first. Such ranking can be made, e.g. using sensory assessments of air quality in small-scale glass chambers, which as shown in the present experiments, is suitable for estimating the relative effects on perceived air quality in real rooms.

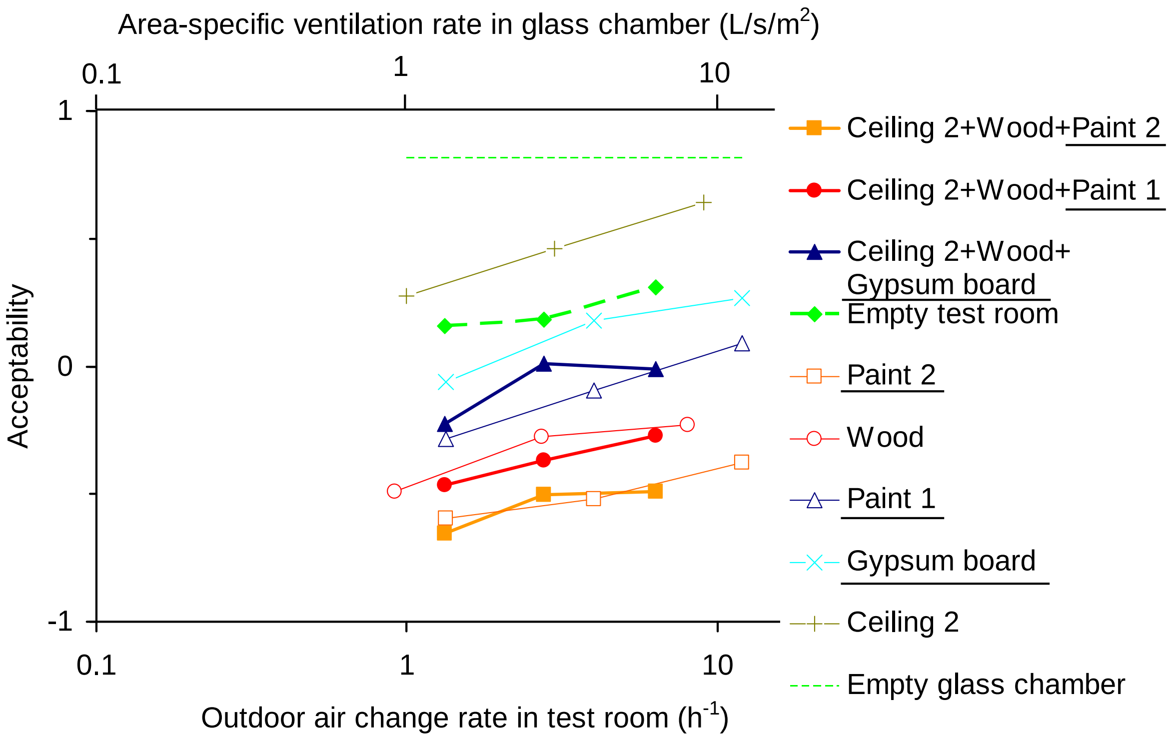


Figure 4. Assessments of acceptability of air quality as a function of the outdoor air change rate for the combinations of materials examined in the test rooms (thick lines) or the area-specific ventilation rate for individual materials tested in glass chambers (thin lines). The materials that were substituted are underlined. Note, the logarithmic scale on the abscissa.

The effect of using low-polluting building materials and increasing outdoor air supply rates on the perceived air quality were studied by examining the relationships between acceptability of air quality and ventilation rate, Figures 1-4. Such an approach has also been shown to be very useful in previous studies [2,9,10] when these effects have been quantified. These relationships can be used to assess to what extent a reduction in ventilation requirements in buildings will contribute to comply with the EU Directive 2002/91/EU [1] that requires reduction of energy use without negative consequences for the indoor environmental quality. They can also be used for specifying the requirements for emissions from building materials, e.g. by labelling schemes for building materials or ventilation standards.

The effects on perceived air quality were examined in rooms polluted only by building materials. Indoor air is however normally polluted also by human bioeffluents. The relationship between perceived air quality and ventilation rate for human bioeffluents was established in the 1980's [13]. However, the assessments were made by a sensory panel on a dichotomous (yes/no) acceptability scale that can not be directly compared with the relationships for building materials examined in this study. Future studies should investigate how the presence of bioeffluents in rooms with otherwise low-polluting building materials will influence the perceived air quality, and consequently ventilation requirements and energy use.

CONCLUSIONS

Substituting building materials with materials shown in small-scale chamber tests to be lower-polluting improved the perceived air quality in full-scale test rooms. The improvement of the perceived air quality was greater than the improvement obtained by increasing the outdoor air supply rate within a range that was realistic for indoor settings.

- The perceived air quality in a room polluted by building materials was improved when the highest-polluting material was substituted with a lower-polluting material.
- The greatest improvement of the perceived air quality is obtained when all high-polluting materials are substituted with low-polluting materials.

ACKNOWLEDGMENTS

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Ventilation strategies for low-energy buildings involving air purification and energy recovery – Climate, energy, comfort and indoor air quality aspects

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SUMMARY

In many climate zones, low energy buildings will not be able to utilize natural ventilation at all times. Hence, low-energy mechanical comfort systems will still be required. For green buildings there are design objectives well above minimum standards for comfort and indoor air quality (IAQ) that are meant to improve occupant productivity. The latter can make meeting low-energy requirements even more challenging. To meet these sometimes-conflicting requirements, the principles of concurrent engineering may be applied to support design charettes. Candidate solutions should include: (1) reducing the amount of outside air while producing an equivalent clean air delivery rate (CADR) using purification technologies, and/or (2) using an energy recovery ventilator (ERV) on the remaining outside air coupled with a smart economizer operating strategy. Simulation results are presented that support these strategies by addressing the cost of conditioning ventilation air and dependence of contaminant levels on the amount of ventilation.

INTRODUCTION

In climates that are not appropriate for complete reliance on natural ventilation, buildings are equipped with mechanical conditioning systems. Traditional systems, that are used when the outdoor air is not within the comfort range, can be energy intensive; however, they usually provide reasonable comfort and adequate indoor air quality. Most such systems use prescriptive minimum ventilation rates specified by the ASHRAE Standard 62.1 Ventilation Rate Procedure [1]. However, conditioning this outside air may account for up to 30% of the total HVAC energy consumed. Proposed standards for new green buildings and green building rating systems such as LEED-NC [2] give credits for ventilation rates well above the minimum standard, exacerbating the energy usage associated with conditioning ventilation air. The interrelationship between reducing exposure to indoor contaminants and the associated cost is not always clearly understood. Clearer procedures for evaluation of tradeoffs as well as better strategies for simultaneously meeting IAQ and energy requirements are needed. Since urban areas sometimes have poorer outdoor air quality, the local conditions must be explicitly considered. The following specific strategies were considered in this study.

1. Use an air-to-air heat exchanger operating between the outside air inlet and building exhaust air streams to recover sensible heat and moisture from ventilation air. This device acts as a preconditioner that permits downsizing of the associated mechanical conditioning equipment peak capacity. It saves energy and reduces utility demand charges as well. This device is called a heat recovery ventilator (HRV) if it recovers

only sensible heat and an ERV if it recovers both sensible and latent heat. This strategy can be used with all the strategies that follow.

2. Use duct-mounted air purification devices to remove particles and volatile organic compounds (VOCs) from the supply air stream, removing contaminants from both the outside air and the recirculated air. Use the ASHRAE Standard 62.1 IAQ Procedure [1] (Alternate Ventilation Rate Procedure) to produce the air quality at least as good as that which would be obtained with either the prescriptive minimum ventilation standard or the augmented ventilation rate recommended for green buildings. CADR from the purifier substitutes for clean outside air.
3. Use duct-mounted air purification devices to remove particles and VOCs from the recirculated air stream alone, assuming excellent outdoor air quality.
4. Use standalone air cleaning equipment to remove contaminants in zones with sources.

METHODS

To help create designs that will meet multiple objectives during the building design process UTRC has been developing a method called Integrated Building Energy and Control Systems (IBECS) [3]. This project seeks to apply the techniques of concurrent engineering to the design process. It works best in a “charette” environment in which all the principal stakeholders are present during design sessions. To permit the design team to investigate as wide a possible set of solutions, a computing infrastructure has been created that is operated by domain specialists. Following this process, the combined design team is able to evaluate a design space that is richer than that which ordinarily would have been investigated with the same amount of resources. Key elements of the IBECS process are:

1. A tool such as the Building Investment Decision Support (BIDS™) [4] framework is used to enrich the list of design options that the team may consider for good indoor environmental quality (IEQ). The potential payoff of considering each option is evaluated to be sure that economically-justifiable solutions are considered.
2. Energy, lighting, airflow, and other simulations are performed early and often during the sessions, so the sensitivities become known. For computationally-intensive situations, pre-work can be done to generate approximate correlations that can be evaluated quickly in a concurrent session.
3. As design changes are made, a common repository of data is used to drive all the different types of simulations, reducing the possibility of inconsistency. A version control and checkpointing system permits the results for each variant to be archived.
4. Key to the process is having a set of at least semi-quantitative metrics for each IEQ category of performance. For energy, the amount of energy and peak demand is used. Cost and/or primary carbon are sometimes used as an alternative.

In general, the value of a configuration may be represented by an objective function to be minimized such as:

$$U = fF + eE + cC + qQ + vV. \qquad (1)$$

Each metric is defined so that a smaller value is better. Lifecycle values are integrated over the remaining lifetime of the building. F represents equipment first and lifecycle operation/maintenance costs, E represents energy use and/or carbon production, C represents discomfort (based on T, RH, and air movement), Q represents air quality as contaminant levels, and V represents ventilation ineffectiveness. The leading f, e, c, q, and v coefficients are used to normalize the individual attributes of performance and express their relative importance to the owner. If demand charges are important, utility charges can be substituted for energy. C is evaluated over the likely occupant positions of a comfort function such as in ASHRAE Standard 55 [5]. Some aspects of a useful air quality metric are still a matter of research, but the integrated mean age of air at the occupant positions would be one choice. Another would be to use a numerical score based on contaminant hazard and irritancy levels. For VOCs, Hollick and Sangiovanni [6] developed a preliminary IAQ index that is scaled to have a value of 1.0 or less for acceptable indoor air. From the contaminant threshold or limit value a contaminant tolerance index is defined as the ratio between the concentration of contaminant, C_i, found in an observed IAQ situation, relative to the allowable limit, $C_{i,max}$, according to the simple expression $T_i = C_i/C_{i,max}$. For a mixture of contaminants, the total tolerance index is given by $\Sigma\, T_i = \Sigma\, (C_i/C_{i,max})$. A total tolerance index greater than 1.0 would represent a case where further ventilation or air purification is warranted. Research is underway to augment this for particulates.

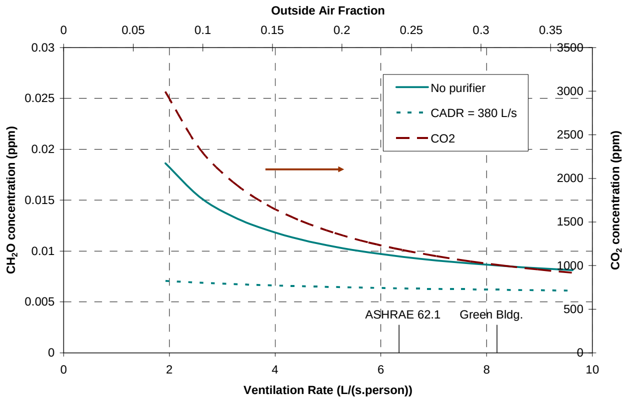
To begin to add increased IAQ functionality to the IBECS process, a spreadsheet model was built that includes as part of its functionality the differential equations displayed in ASHRAE Standard 62.1-2004. Routine IBECS energy simulation was also performed.

To understand the effect of ventilation and purifier efficiency on contaminants concentration, an analysis of the effect of ventilation on the CO_2 and contaminant concentrations in different types of buildings was carried out. At the same time, the effect of a purifier on the quality of the air was studied. It was assumed that the purifier is based on a catalyst, so that the efficiency does not change with time, as long as the catalyst maintains its activity. In the case of reactive solids or adsorbents, it is well known that the efficiency will vary with time. An adsorbent can show 100% efficiency when it is fresh or a certain value lower than 100% according to its actual design specifications. As the adsorbent becomes saturated, and the adsorption front moves towards the exit of the purifier, the efficiency decreases as breakthrough occurs. In the case of a reactive solid, it may show high initial reactivity towards oxidizing air contaminants, but according to the resulting reduction product, a layer can be formed which will introduce an additional resistance that will slow down the access of the contaminants to the reactive surface, and the rate of reaction will decrease.

The calculations were carried out for a 1000 m^2 office and a 334 m^2 school building. The occupancy density was 5 and 36 persons per 100 m^2, respectively. The CO_2 generation rate was assumed to be 18 L/(person·h). Formaldehyde, toxic and ubiquitous in buildings, was chosen as a typical contaminant generated by surfaces. The production rate was varied between 40 and 70.4 $\mu g/(m^2 \cdot h)$ [7,8]. The outdoor concentrations were 400 ppm for carbon dioxide and between 0 and 55 ppb for formaldehyde [9]. ACGIH set the TLV (ceiling limit) at 0.3 ppm. We use 0.1×TLV as an acceptable value for a gaseous contaminant, so in this case the acceptable limit is 30 ppb [6]. A catalyst-based purifier, when considered, was assumed to be in the recirculating air. The supply air rate needed for thermal comfort was 4720 L/s for the office and 3165 L/s for the school.

RESULTS

The variations of formaldehyde (CH_2O) and CO_2 concentrations as a function of the ventilation rate for the two types of buildings are shown in Fig. 1.



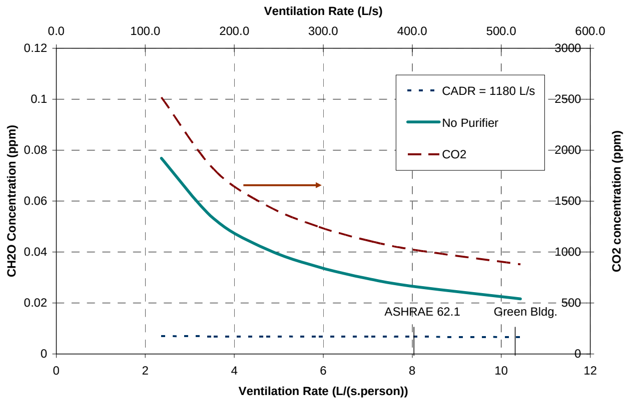


Figure 1. Effect of ventilation rate on contaminant concentrations at a CH_2O generation rate of 40 µg/(m^2·h) and a CO_2 production rate of 18 L/person with outdoor concentrations of 5.5 ppb for CH_2O, and 400 ppm for CO_2. a) 334 m^2 School, b) 1,000 m^2 Office.

It can be seen that, when the data is presented as a function of the ventilation rate per person, the CO_2 curve is coincident for both cases. The steady-state value is only a function of the outdoor concentration value and the per-person generation- and ventilation- rates. However, when we consider the actual ventilation rates, we can see that these are much higher in the school building. The recommended ASHRAE 62.1 values were used for both cases, namely, 2.5 L/(s·person) and 0.3 L/(s·m^2) for the office, and 5 L/(s·person) and 0.6 L/(s·m^2) for the school (classroom age 9 plus). When the overall ventilation rates recommended by ASHRAE Standard 62.1 in terms of people and area are recalculated in per person terms, the respective values are 8.0 and 6.4 L/(s·person). The 30%-higher operating value for Green Buildings [2] is indicated in Fig. 1.

The variation of the CO_2 and CH_2O concentrations with the ventilation rate, at a constant Clean Air Delivery Rate (CADR), are also shown in Figs. 1. It can be seen that to maintain the CO_2 level below 1000 ppm the overall ventilation rate required is 8.1 L/(s·person), independent of the type of building. To maintain the CO_2 level below 2000 ppm, the ventilation rate can be as low as 3.2 L/(s·person).

The effect of the formaldehyde generation rate for the school is shown in Figure 2.

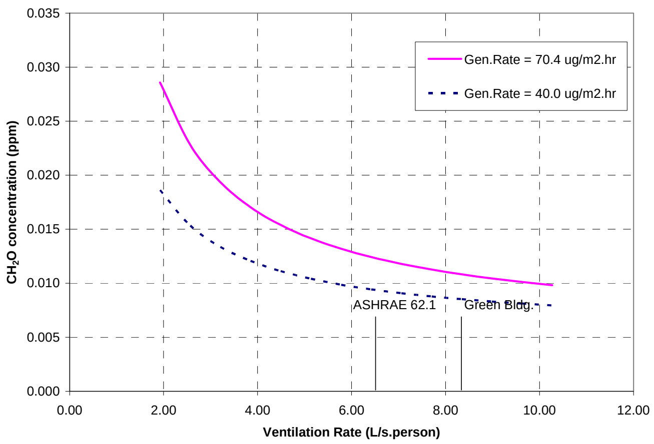


Figure 2. Effect of CH_2O generation rate on contaminant concentrations with no purifier on a 334 m^2 school with an outdoor concentration of 5.5 ppb for CH_2O.

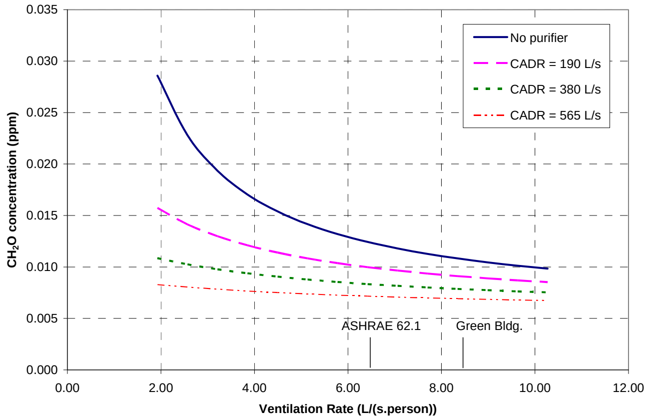
At high ventilation rates, as the generation increases from 40 to 70.4 µg/(m^2·h), the contaminant concentration increases by about 20%, but at low ventilation rates the increase is much higher. At 2.0 L/(s·person), the increase is about 50% and the Tolerance Index is close to one. It is obvious that to achieve an acceptable level of formaldehyde, a purification system needs to be implemented.

The effect of different levels of purification on the contaminants concentration is shown in Fig. 3. In both types of building, the increase in the purification level has a significant effect on the formaldehyde steady state concentration. Considering equivalent levels of purification (same CADR:supply air ratio), the effect is more noticeable in the case of an office. This is due to the relatively lower ventilation rates that are recommended for these types of buildings.

To simulate the design size of the HVAC equipment required to handle the building loads as well as condition outside air, an industry-standard manufacturer-developed HVAC design code was used. This code can consider and size unitary equipment with options for either an economizer or an ERV. Figure 4a shows for the school how the size and first cost of the equipment depends on the amount of outside air used during heating and cooling seasons. Figure 4b shows the annual energy usage dependence. For simplicity, energy was selected for reporting since it is related to global warming and avoids assumptions regarding local utility rates and demand charges involved in discussions of operating cost. The buildings have typical thermal characteristics and usage schedules, but many detailed assumptions are involved in such simulations.

DISCUSSION

The variation of formaldehyde concentration with ventilation rate shows a different relationship for the office and school, since the generation rate is proportional to the space area, while the ventilation is a function of both area and occupancy. When no purification system is used, and the ventilation rate is cut in half, the formaldehyde level increases between 40 and 80% (Fig. 1). The lowest increase is observed for schools where ventilation rates are significantly higher than in offices. On the other hand, when a purification system is added to the return flow, the increase in CH_2O concentration as the ventilation rate decreases is much lower. Moreover for a CADR of about 25% of the supply air rate the sensitivity of the CH_2O concentration to the ventilation rate is quite low. In the case of the office, the relative amount of CH_2O entering the space with the fresh air is much lower than the generation rate, and the steady state concentration is approximately given by C_{ss} = Source Rate/(CADR + OA). On the other hand, for the school building, the ventilation rate is quite high and the amount of contaminant generated in the building is relatively small. Therefore, the steady sate concentration is close to the value in the outdoor air.



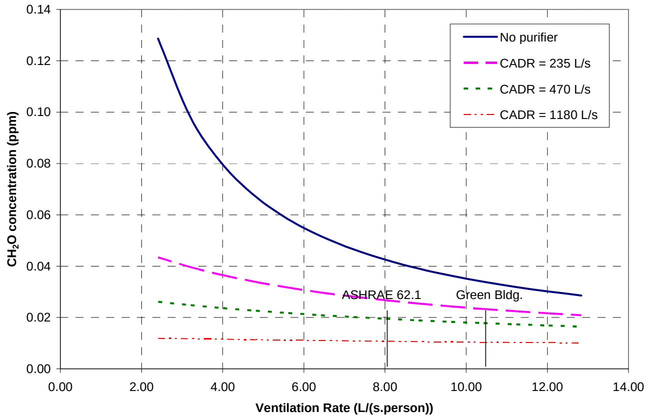


Figure 3. Effect of purifier efficiency on contaminant concentration at a CH_2O generation rate of 70.4 μg/(m^2·h) and an outdoor concentration of 5.5 ppb CH_2O. a) 334 m^2 School; b) 1000 m^2 Office.

Budaiwi and Al-Homoud [10] studied the effect of ventilation on the concentration of air contaminants and concluded that although a ventilation rate of 7.5 L/s·person was sufficient to keep the CO_2 concentration below the recommended level, it was not enough to reduce the formaldehyde concentration below the allowed level of 120 μg/m^3 [11]. It should be taken into account that these authors assumed a formaldehyde generation rate of 150 μg/s. For the space considered here, it corresponds to a value of 675 μg/(m^2·h), which is much higher than the typical values found in buildings and used in the current analysis. The actual lower limit for ventilation when using the IAQ Procedure may be set by (1) a local regulation on CO_2 concentrations, (2) calculated contaminant levels that create a Tolerance index greater than one, or (3) determining that the air quality with purification is equivalent to that which would have been obtained using standard minimum or enhanced prescriptive ventilation rates.

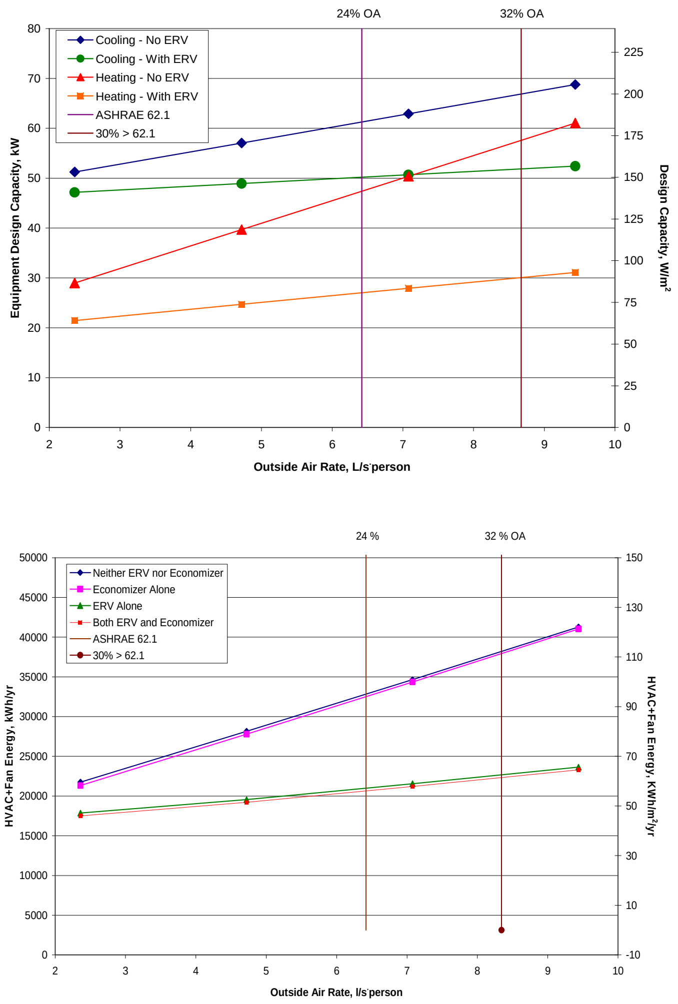


Figure 4. New York school HVAC characteristics as a function of outdoor air flow rate. a) Heating and cooling equipment design capacities with and without and ERV, b) annual energy consumption with and without economizer and ERV options.

Studies of the effect of ventilation quantity on cost and air quality for additional building types and locations are planned. Some systematic studies of the ERV configurations have been done in the past [12]. The result shown in Fig. 4 should be considered as an example of the relevant trends. Clearly there is much variability based on the many factors. A method similar to the one used herein should result in a good understanding of the tradeoff of energy and exposure. It is clear from this case that equipment first cost increases with the outside air fraction used; introducing an ERV can result in a substantial downsizing of the heating and cooling plant.

For the case of using an ERV with the large outside air rate used with green buildings, the plant capacity is still smaller than for the case of using no ERV with an outside air rate just meeting the Standard 62.1 minimum ventilation rates. Also, the size of the equipment is significantly reduced if the IAQ procedure is used with a local air purifier. Downsizing of such equipment can help defray the first cost of either ERVs or air purifiers.

Fig. 4b shows that in this particular case of a high-ventilation flow and a school operating schedule, the effect of the economizer is minimal. Elaborate strategies such as night cooling were not considered. Use of the ERV makes a considerable difference, especially in the heating season when the difference between the indoor and outdoor temperatures is larger. Even at the high ventilation rates proposed for green buildings, the annual energy cost is lower with the ERV than it is for the minimum standard value and no ERV. Hence, the ERV makes bringing in the additional air affordable. If one chooses to operate according to the IAQ procedure, there are also substantial energy benefits during the lifecycle of the equipment. The end result is to provide the designer with the methodology and tools to permit wider consideration and adoption of these configurations.

ACKNOWLEDGEMENT

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Experimental Study of Thermal Environment and Comfort in an Office Room with a Variable Air Volume (VAV) System under Low Supply Air Temperature Conditions

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SUMMARY

The air distribution components in a variable air volume (VAV) flow system influences the overall function of the system. A low supply air temperature and a wide working range in terms of airflow rate, facilitate high energy efficiency. However, in order to achieve these properties, high demands must be set on the function of the supply-air diffusers.

This paper analyzes the thermal environment and comfort in an office room where the supply air diffusers have been chosen in accordance with such high demands. The study involves determining thermal comfort properties under different flow conditions and with a low supply air temperature, about +15^{o}C, in a full scale test room. It also includes measured velocity profiles and calculated draught ratings. The results indicate that the risk of draught was quite low despite high airflow rates and a low supply air temperature with the studied type of a supply-air device.

INTRODUCTION

The selection of air distribution components in the design of a variable air volume (VAV) flow rate system has a crucial importance for the overall function of the system. Varying airflow conditions need to be managed in a way that the requirements for the indoor climate parameters are always fulfilled. Here the airflow control devices and supply air outlets are critical components that need to be selected carefully, since improper selection of these devices is a common cause of excessive noise and draught in occupied spaces [1,2].

The main demand for the supply air diffuser in a VAV system application is to have high induction properties with varying airflow rates. It is essential that the diffuser supplies cold air to the occupied space evenly under any airflow condition without causing uncomfortable drafts by air “dumping” or by excessive room air motion. This requirement means that the air should be introduced to the room at a sufficient velocity to ensure good mixing with the room air.

If the discharge area of the diffuser remains constant, the velocity of the supply air stream falls in direct proportion to the reduced airflow rate. Due to the naturally denser cold air a “dumping”, defined as a dropping of a horizontal supply air jet into the occupied zone, can occur and result in a sensation of draught. Thus, using conventional fixed supply air outlets in a variable air volume flow application needs to be carefully considered. The fixed devices are commonly used when VAV boxes are installed for the room airflow control. As a result the supply air temperatures and minimum airflow rates must be kept relatively high in order to avoid problems with thermal comfort. Moreover, over-cooling the premises can occur at low internal heat loads and high minimum airflow rates if the supply air temperature is kept constant.

Relatively constant supply air velocity irrespective of the airflow rate is maintained with a diffuser with a variable discharge area, commonly referred to as a VAV diffuser. This is a device, which changes its outlet configuration automatically when controlling the supplied airflow rate to the room. In commonly installed devices the airflow rate is determined by the diffuser's opening (between 0 to 100%) and the constant static pressure at the inlet side. Therefore, a stable pressure should be maintained upstream of the diffuser. This demand is achieved by keeping constant pressure in the branch ducts by active control dampers, which can make the system more complicated and costly. This can especially be the case when adopting VAV systems in existing buildings, where extensive refurbishment is not always possible and the active control dampers are not easy to install in the duct system.

This study aims to look for uncomplicated VAV system configurations that can assure a good indoor climate, while at the same time minimize the energy use of the system. A possibility of building up a VAV system with VAV diffusers for airflow control and without active control dampers in the duct system have been considered. For applying this kind of a technical solution it is essential for the VAV diffusers to be able to absorb the pressure unbalance in the system during varying airflow rates, while maintaining good airflow control properties without causing problems for the indoor climate, such as excessive noise. Moreover, from an energy use point of view it is beneficial for the VAV diffusers to be able to control the airflow within a wide range and manage low supply air temperatures without any risk for the thermal comfort in the room. With lower supply air temperatures the cooling capacity of the supply air will be improved and better control of the room temperature can be achieved. Low supply air temperatures are also advantage in systems that work with 100% of outside air and where a possibility for free-cooling can be applied.

This paper analyzes the thermal environment and comfort in an office room where the supply air diffusers have been chosen in accordance with the demands discussed above. The paper accounts for laboratory tests, which aimed to study how the requirements from the thermal comfort point of view are met under different supply airflow conditions and with low supply air temperature. The results indicate that is possible to fulfill the demands that must be set on the VAV diffusers in order to apply them in a wide airflow range and at low temperature conditions.

Although the tests have been carried out with a specific diffuser, the results are general in the sense that they show that the high demands on supply air diffusers can result in products which fulfil them.

METHODS

The function properties of the selected VAV supply air diffuser were tested in a simulated indoor environment: in a full size cellular office cube built inside the laboratory hall. The test room was constructed with plaster boards on a wooden framework. The internal dimensions of the test room were 3,9(L) x 2,8(W) x 2,7(H) m. To imitate a common office environment the room was filled with usual office equipment: a table, a chair, a computer and lighting. The internal heat loads were simulated with a PC-model (150 W), a dummy (80 W) and lighting (total 220 W). The artificial lighting consisted of two luminaries with two fluorescent tubes each (85 W per luminaire), plus a desk lamp (50 W).

The test set-up included a supply air fan with frequency inverter and with a pressure control (controlled pressure level was approx. 50 Pa.), a sound attenuator, a supply air heater with an air temperature control, an airflow meter, a VAV diffuser and temperature sensors for monitoring temperatures in the duct as well as inside and outside the test room. All sensors were connected via a logger to a personal computer for monitoring purpose.

The technical properties of the tested VAV diffuser enable to measure the incoming supply airflow rate and adjust the discharge area respectively. Therefore strict pressure control at the inlet of the device is not needed. The discharge area of the diffuser varies according to the needed airflow rate and is internally controlled by a traversing motor, which gets impulses from the controlling sensor. The control and regulating equipment as well as the sensors are built into the supply air device and the simultaneous values can be read with the computer.

Two different VAV diffuser mounting arrangements were tested: one with the diffuser free from the ceiling and one with the diffuser in the suspended ceiling. Without suspended ceiling the height from the ceiling to the discharge area of the device was 30 cm and from the device to the floor 2,4 m. With suspended ceiling the latter height was increased to 2,7 m. For the exhaust air a transfer air device was installed on the wall close to the ceiling.

Thermal comfort measurements were carried out with constant supply air temperature +15°C, but with different airflow rates under steady-state conditions. For all the tests the operative temperature in the occupied zone and temperature outside the office cube was kept 22 ± 1°C, which corresponds to the winter conditions and *A* level comfort class [3].

The heat loads in the test room were adapted to the airflow in order to obtain the correct room temperature. Table 1 presents the conducted test cases with different combined heat loads and cooling capacities. There is a small difference between the heat load and cooling power values in the table. This gap is due to the heat transmission through the envelope of the test room.

Table 1. The test cases completed in thermal comfort measurements

Case	Mounting condition	Airflow rate, l/s	Supply air temperature, °C	Cooling capacity, W	Balancing heat load, W
1	no suspended ceiling	10	15	85	122
2	no suspended ceiling	25	15	210	250
3	no suspended ceiling	50	15	420	450
4	suspended ceiling	10	15	85	122
5	suspended ceiling	25	15	210	250
6	suspended ceiling	50	15	420	450

Air temperature and air velocities were measured in a number of room points as shown in Figure 1. At each position the measurements were taken at 3 heights - 0,1 m, 0,6 m, 1,1 m, which is based on the position of a sitting person. All together the results were obtained from 27 room points.

The sampling period for each measurement was 3 minutes. Every measurement case described in table 1 was done in three replicates and the results are presented as an average over these three measurements. The risk of draught in the test room was evaluated by using draught rating (*DR*) model and the calculations for every measured point were done according to ISO 7730 [4]. Draught rating expresses the percentage of people predicted to be dissatisfied by draught.

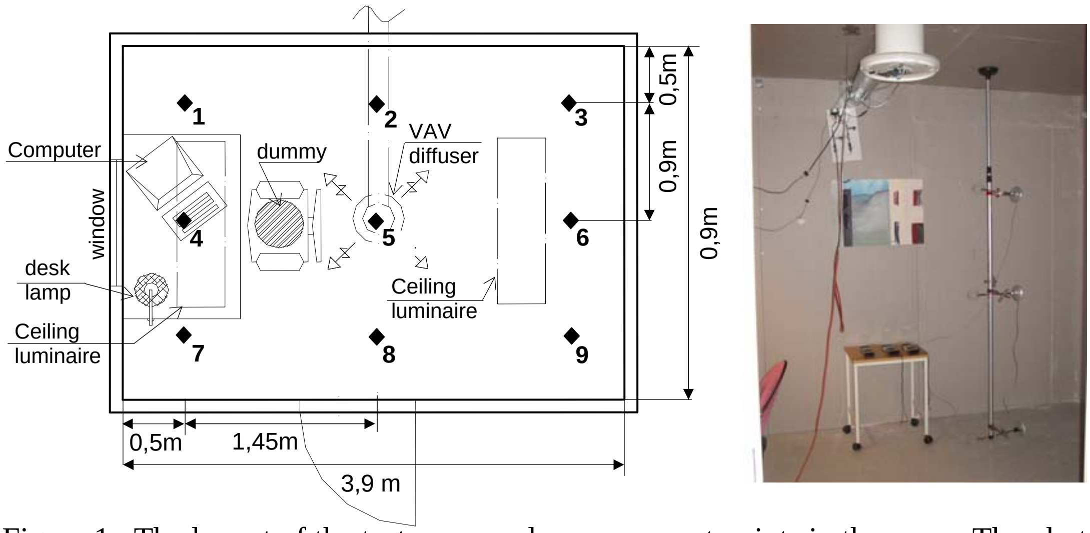

Figure 1. The layout of the test room and measurement points in the room. The photo presents the measurement set up.

Additionally, the measurement results were statistically analyzed with an analysis of variance (ANOVA) test, in order to see if the measured values of air velocities in the occupied space depend on different parameters that were varied during the experiment. The statistical significance of an effect of different parameters such as the room point, the level of a measured point, the ceiling and the airflow rate was studied. The main and combined effects of the described variables were first found by analyzing all the airflow rates together and then by each airflow case separately. The chosen confidence level in the analysis accounted here is 95% (p = 0,05).

RESULTS

The mean air speed and draught rating distributions at different supply airflow rates in two mounting cases, with and without suspended ceiling, are presented in Figures 2 and 3. The figures show the percentage of the measured points being in the specified range of air velocity and draught rating values. For example, it can be seen from the figures that the average air velocity in majority of measured points was less than 0,15 m/s with maximum airflow conditions 50 l/s and less than 0,10 m/s at lower airflow rates. According to thermal comfort guidelines, the air velocity in the occupied space should not exceed 0,15 m/s [3] and the draught rating should be below 15% [4]. These limits were exceeded only in few measured points in the room and that occurred mainly at maximum airflow rates (see Fig. 2 and 3).

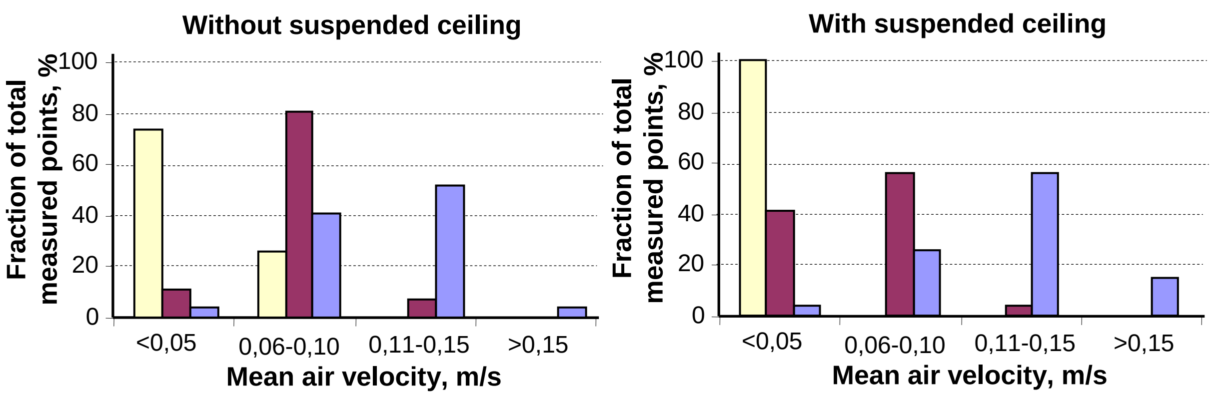

Figure 2. Air speed distributions in the test room with different supply airflow rates and with different mounting cases. The supply air temperature was +15°C.

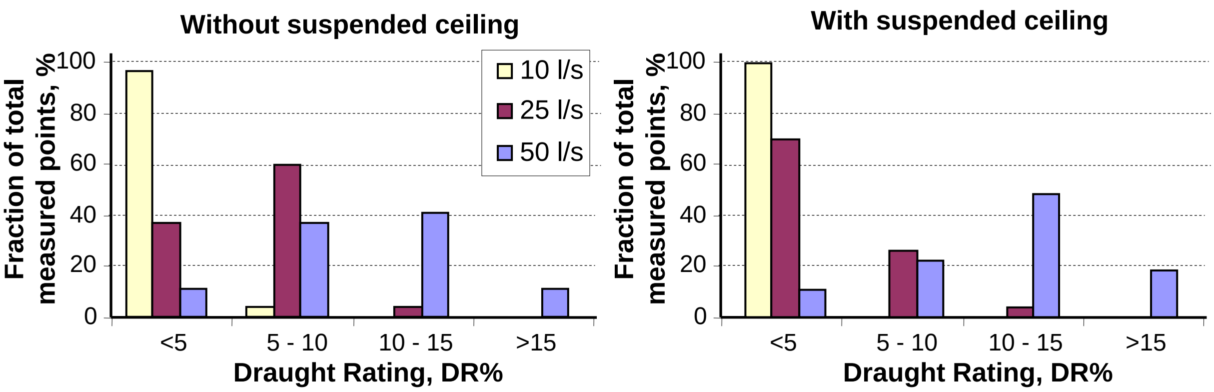

Figure 3. Draught rating distributions in the test room with different supply airflow rates and mounting cases. The supply air temperature was +15°C.

In addition, there seem to be no substantial differences between the results with and without suspended ceiling mounting cases. However, some diversity can be seen at average airflow conditions (25 l/s), where the mean air velocities and draught ratings were somewhat lower with the suspended ceiling. The results from statistical analysis of variance test, summarized in Table 2, also indicate that the ceiling has an effect at average and minimum airflow conditions, but no effect at maximum airflow condition in the specified confidence level (p=0,05). The table shows if an effect of each parameter that varied during the experiment is a statistically significant or not. The probability values of calculated F value compared to the value given for the *F-distribution* in the *F-table* are also presented for the variables which have an effect. The table accounts for the main effects only, meaning that if the parameter alone and not in interaction with other parameters influences the room air speed. The interaction effects of different parameters were also tested, but no higher order effects were revealed from the results.

Nevertheless, even though different parameters such as the room point, the room level and the ceiling revealed to affect mean air velocities in the occupied space, there seem to be no regularities between the main effects. With maximum airflow rate 50 l/s the room point and the room level showed a significant influence, yet in the minimum airflow rate 10 l/s the effect revealed to be only from the room level and the ceiling. However, it was preliminary assumed that the airflow has an effect and as it can be seen from the Table 2, the probability that the variability of the mean air velocity values with different airflow rates can be attributed to experimental error is very low.

Table 2. Statistically significant effects of different variables on the mean air velocity in the occupied space. The chosen confidence level is 95% (p = 0,05)

Main effect	**Combined all airflows**	**10 l/s**	**25 l/s**	**50 l/s**
Room point	NO	NO	YES $P(F>5,05) = 0,025$	YES $P(F>4,79) = 0,030$
Room level	YES $P(F>15,29)=0,0001$	YES $P(F>60,47)=9,005 \cdot 10^{-13}$	NO	YES $Pr(F>6,45)=0,012$
Ceiling	YES $P(F>15,29)=0,0026$	YES $P(F>63,05)=3,52 \cdot 10^{-13}$	YES $P(F>17,65)=4,42 \cdot 10^{-5}$	NO $P(F>3,80)=0,053$
Airflow rate	YES $P(F>577,85) = 2,2 \cdot 10^{-16}$	-	-	-

In general, it should be noted that the results of an ANOVA test do not show the size of a single effect, e.g. if the room level has a higher effect compared to the ceiling. Moreover, the direction of the variation is not known, e.g. in which case the highest results appear. The test shows statistically if different parameters affect the results or if the variation is mainly due to experimental error.

Figures 4 and 5 illustrate the velocity profiles evaluated from the results from the measurement cases 3 and 6 (see Table 1). These were the cases where the required draught rating and air velocity values were exceeded in some room points. The critical points given in the figure, where the air velocities were exceeding 0,15 m/s are also the points where the draught rating was exceeding 15 %.

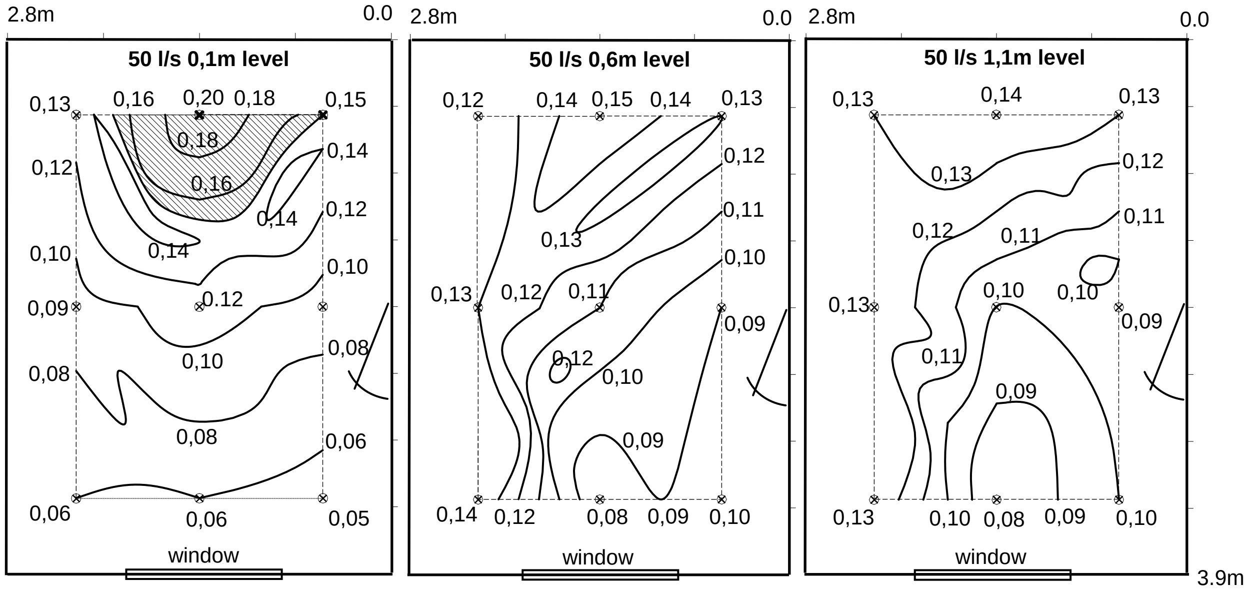

Figure 4. Iso-velocity profiles in measurement case 3. Crossed dots mark the measuring points.

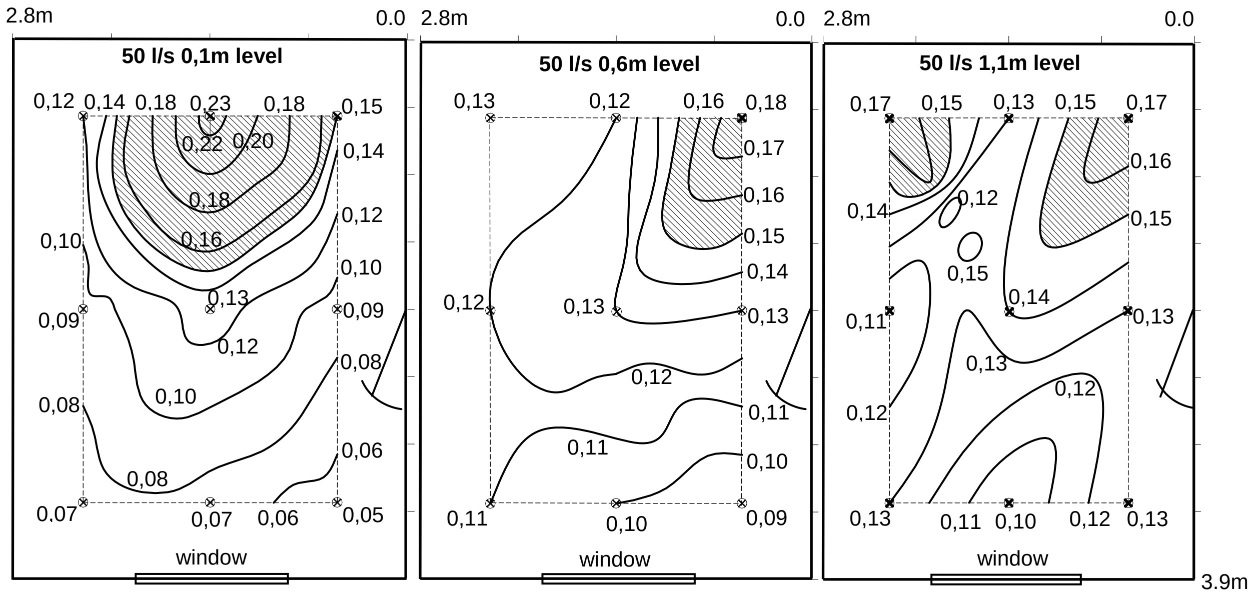

Figure 5. Iso-velocity profiles in measurement case 6. Crossed dots mark the measuring points.

As shown in Fig.4 and Fig.5, all of the critical room points were locating on one side of the room (measuring points 3, 6, 9). This was the empty side of the test room. Moreover, the most critical point, room point nr 6, was locating on the level of 0,1 m above the floor. The upper levels of the same measuring point did not have any higher velocities.

Since the turbulence of airflow has an important influence to the perception of draught in the occupied space [5], the fluctuation of the air velocities in the test room has been further analysed. Figure 6 presents the standard deviation as a function of the mean air velocity with all different airflow conditions at three measurement levels for no suspended ceiling installing case. The results from the suspended ceiling mounting case were similar. It can be seen that the fluctuation of the air velocity was increasing when the average air velocity in measured points increased. In addition, a small decrease in the gradient of the regression lines, as the measuring level decreased from 1,1 m to 0,1 m, shows that the velocity fluctuations were more significant at the ankle level. Similar data has also been found in previous studies about airflow characteristics [6].

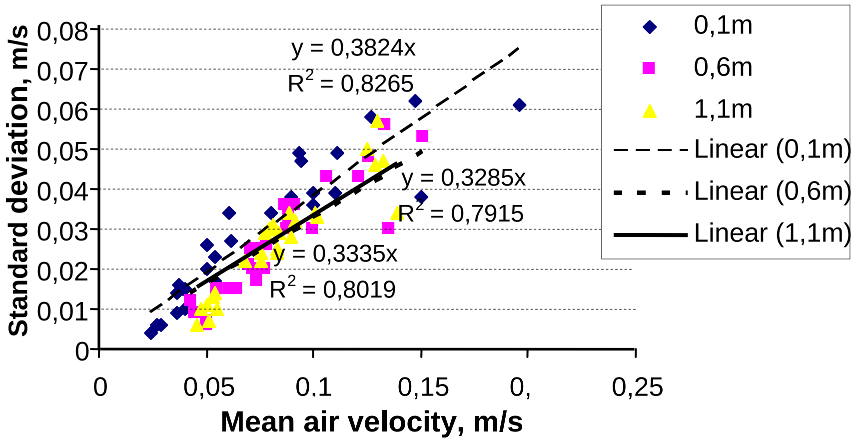

Figure 6. Standard deviation of air velocity with three different airflow rates and at different levels in the test room. The diagram corresponds to no suspended ceiling mounting case.

DISCUSSION

Thermal comfort guidelines address minimal draft levels by placing limits on the allowable mean air velocity as a function of air temperature and turbulence of airflow. The measurements in the test room revealed that the air movements and the draught levels at medium airflow (25 l/s) and minimum airflow (10 l/s) conditions did not exceed the required levels stated by comfort standards and regulations [3,4]. Even at the lowest airflow rate, no risk of “air-dumping” was indicated. Marginal draught risk was registered in a few measuring points at maximum airflow conditions 50 l/s. In addition, the results showed no substantial differences between the two diffuser mounting cases depending on the ceiling. However, the ceiling had an effect at lower airflow conditions (see Fig.1 and Fig. 2).

The statistical analysis test revealed that the measurement point and the measurement level have a statistically significant effect to the results at maximum airflow conditions. As was illustrated in Fig.4 and Fig.5, all critical points were situated on the empty side of the test room, opposite the workplace. No draught risk was observed in the normal working zone. One possible explanation for this could be the specific distribution of heat sources in the room. In general, all of the heat sources in the room have an influence to the air motion in the room by giving rise to buoyancy induced velocity scale which can match the velocities generated by a jet in the occupied zone [7]. The only influencing heat source on the empty side of the test room was the ceiling luminaire, while the other heat sources where distributed to the other side of the room. Nevertheless, since the air motion in the room is complex, it is hard to make any definite conclusions for the causes of the draught risk in these specified room points and further studies should be conducted in order to identify the reason.

A direct relation between air velocity and turbulence intensity was indicated from the measurement results. Moreover, the points with higher turbulence intensities were situated close to the floor, measured at ankle level. Coincidentally, this was also the level where the most critical points were located (see Fig. 4 and Fig.5). However in the present case, higher air turbulence on the ankle level may partly have been caused by the floor temperature, which was some degrees lower than the room temperature.

This laboratory experiment was part of a study which aimed to look for uncomplicated VAV system configurations that can assure good indoor climate, while at the same time minimize the energy use of the system. A possibility of building up a VAV system with VAV diffusers, which can manage pressure variations (at least 100 Pa) in the system and work with a wide airflow range and at low supply air temperatures, has been considered. However, in order to achieve these properties, high demands must be set on the function of the supply-air diffuser. One type of a device that seemed to have the technical properties to fulfil the demands was tested in a full scale test room with the aim to study the thermal environment and comfort under different supply airflow conditions and with low supply air temperature. The results indicate that it is possible to fulfil the demands that must set on the VAV diffuser in order to apply it in the specified conditions.

The VAV system configuration with this type of a VAV diffuser has also been tested in buildings in operation, focusing on indoor climate and the need of energy. The results yielded that this technical configuration provides an adequately functioning system that ensures good indoor climate and work energy efficiently [8].

ACKNOWLEDGEMENT

This work described has been carried out as a part of a pilot project CAVA (From Constant Air Volume to Variable Air Volume), which has been conducted under the EUFORI program funded by Swedish Energy Agency (*Statens Energimyndighet*).

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Conceptual analysis of Intensive Care Room with Computational Fluid Dynamics

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SUMMARY

Since 2005, the Leiden University Medical Center started the building of the new department of children Intensive Care (IC). Several types of geometry have been planned in the department, In order to minimize the cost and get the most applicable result, an IC room with two beds in serial has been chosen for the detailed aerodynamic study. The CFD model of the IC room with two beds in serial is studied. The CFD model will split the whole room as many small mesh elements to fulfil the conservations of mass, momentum and energy. The IC room has a high internal heat load from the devices (approx. 60W/m^2) and the hospital owner has a strong expectation on the indoor comfort of the air speed and temperature. Therefore, several difference ventilation concepts have been modeled and the best choice has been made for the final design.

INTRODUCTION

Intensive Care (IC) Medicine is a branch of medicine concerned with the provision of life support or organ support systems in patients who are critically ill who usually also require intensive monitoring.

Patients requiring intensive care usually require support for hemodynamic instability, airway or respiratory compromise (such as ventilator support), acute renal failure, potentially lethal cardiac dysrhythmias, and frequently the cumulative affects of multiple organ system failure. Patients admitted to the intensive care unit not requiring support for the above are usually admitted for intensive/invasive monitoring, such as the crucial hours after major surgery when deemed too unstable to transfer to a less intensively monitored unit.

Medical studies suggest a relation between intensive care unit (ICU) volume and quality of care for mechanically ventilated patients. [1] After adjustment for severity of illness, demographic variables, and characteristics of the ICUs, higher ICU volume was significantly associated with lower ICU and hospital mortality rates. For example, adjusted ICU mortality (for a patient at average predicted risk for ICU death) was 21.2% in hospitals with 87 to 150 mechanically ventilated patients annually, and 14.5% in hospitals with 401 to 617 mechanically ventilated patients annually. Hospitals with intermediate numbers of patients had outcomes between these extremes.

It is generally the most expensive, high technology and resource intensive area of medical care.

Since 2005, the Leiden University Medical Centrum started the building of the new department of Children Intensive Care. The new department will have 40 IC rooms and the gross floor area is about 3250 m^2.

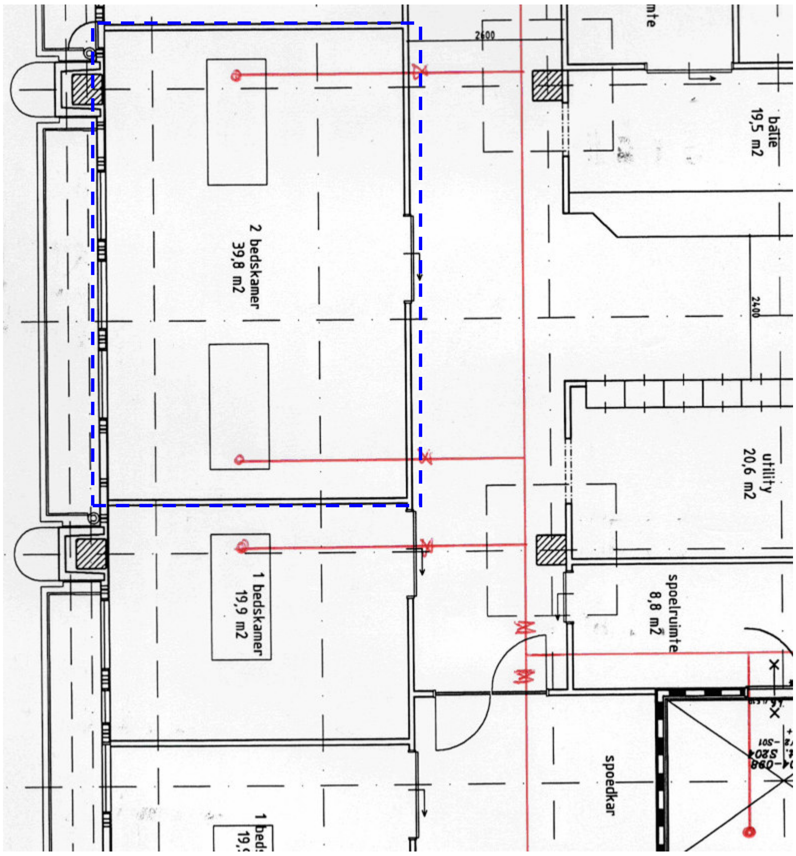

Figure 1. Floor plan of one part of IC department

Figure 1 shows an example floor plan of the IC department. Several types of geometry have been planned in the department, for example, one bed room; two bed room; two beds in parallel in one room or two beds in serial in one room, etc.

In order to minimize the cost and get the most general result, an IC room with two beds in serial has been chosen for the detailed aerodynamic study which is indicated with a blue dot line in Figure 1.

APPROACH

In order to study and evaluate the performance of the new concept, Computer Fluid Dynamics (CFD) analysis has been carried out.

Computational Fluid Dynamics (CFD) has grown with the progress of computer technology and numerical analysis [2]. It comes from a mathematical curiosity to become an essential tool in almost every branch of fluid dynamics, from aerospace propulsion to indoor climate analysis. CFD is commonly accepted as referring to the broad topic encompassing the numerical solution, by computational methods, of the governing equations which describe fluid flow, the set of the Navier-Stokes equations, continuity and any additional conservation equations, for example energy or species concentrations.

As a developing science, Computational Fluid Dynamics has received extensive attention throughout the international community since the advent of the digital computer. The attraction of the subject is twofold. Firstly, the desire to be able to model physical fluid phenomena that cannot be easily simulated or measured with a physical experiment, for example fire and smoke development. Secondly, the desire to be able to investigate physical fluid systems more cost effectively and more rapidly than with experimental procedures.

There has been considerable growth in the development and application of Computational Fluid Dynamics to all aspects of fluid dynamics. In design and development, CFD programs are now considered to be standard numerical tools, widely utilised within industry [3].

The CFD model of the IC room with two beds in serial is shown in Figure 2. The size of the room is 7.875x5.055x2.7m (LxBxH).

The CFD model will split the whole room as many small mesh elements to fulfil the conservations of mass, momentum and energy. For this IC room, the amount of mesh elements is ca. 500,000 (Figure 3).

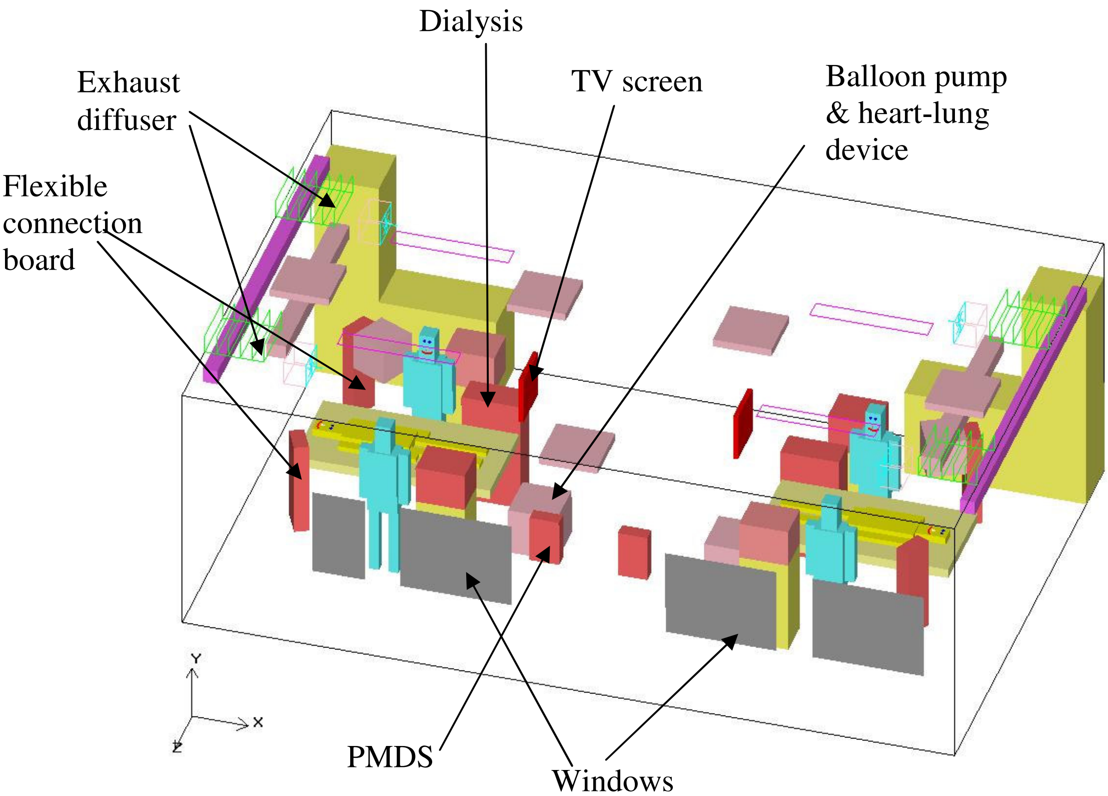

Figure 2. CFD model of the IC room with two beds in serial

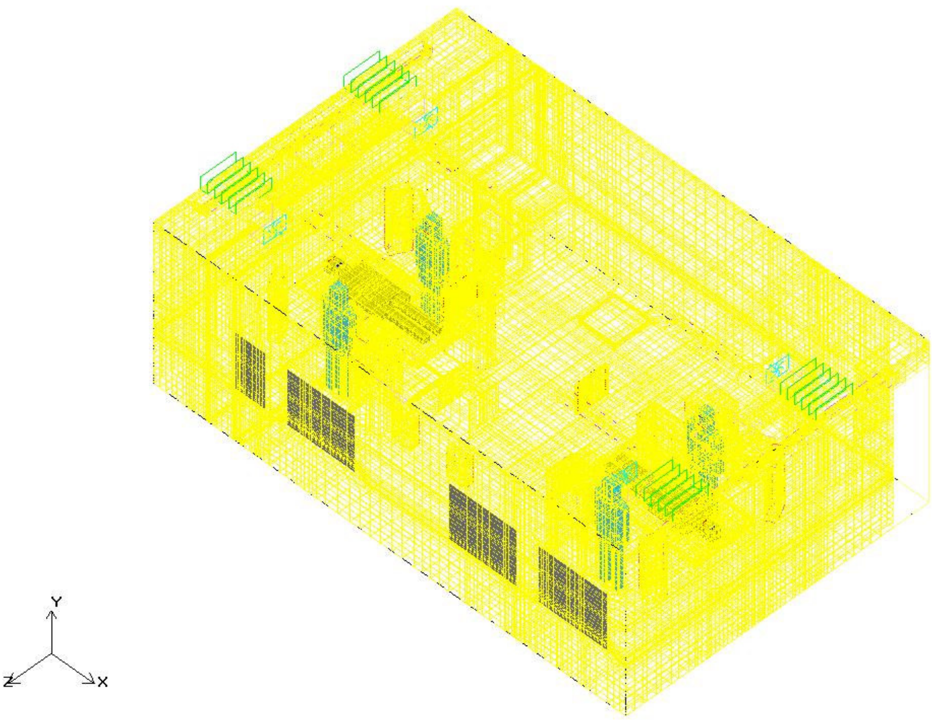

Figure 3. Mesh element of the CFD model of the IC room

SIMULATION CONDITIONS & REQUIREMENTS

The simulation condition is based on the GTR (General Technical Requirement, M/E) [4] of LUMC. Several conditions related to CFD simulation are:

- The IC room is 7.875x5.055x2.7m (LxBxH). This is 39.8m^2 and 1007.48m^3;
- Climate ceiling will be applied; condensation must be avoid
- In order to avoid condensation problem on the surface of climate ceiling, the surface temperature is set as 18^{o}C;
- Air speed keeps lower than 0.12m/s in the patient zone;
- Air temperature approximately 22^{o}C when Tout<=26^{o}C;
- Internal heat load approximately 112W/m^2 for thorax situation.

BASIC CASE STUDIES

In this research, four concepts of ventilation supply are analyzed:

a) air supply diffusers from the side walls of the middle block (Figure 4a)
b) downflow air supply diffusers from middle ceiling (laminar flow) (Figure 4b)
c) downflow air supply diffusers from ceiling of sides of bed (laminar flow) (Figure 4c)
d) ceiling diffusers (one side supply) (Figure 4d)

Models of four concepts

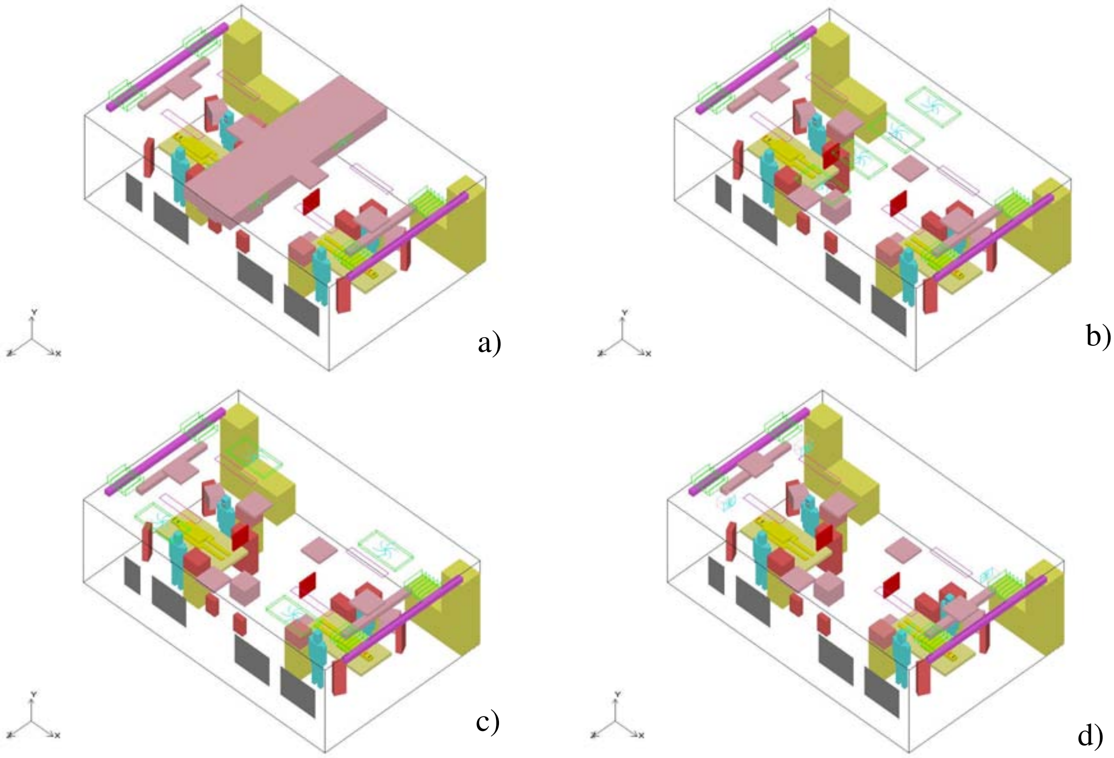

Figure 4. CFD model with a) air supply diffusers from the side walls of the middle block; b) downflow air supply diffusers from middle ceiling (laminar flow); c) downflow air supply diffusers from ceiling of sides of bed (laminar flow); d) ceiling diffusers (one side supply)

Temperature comparison of four concepts

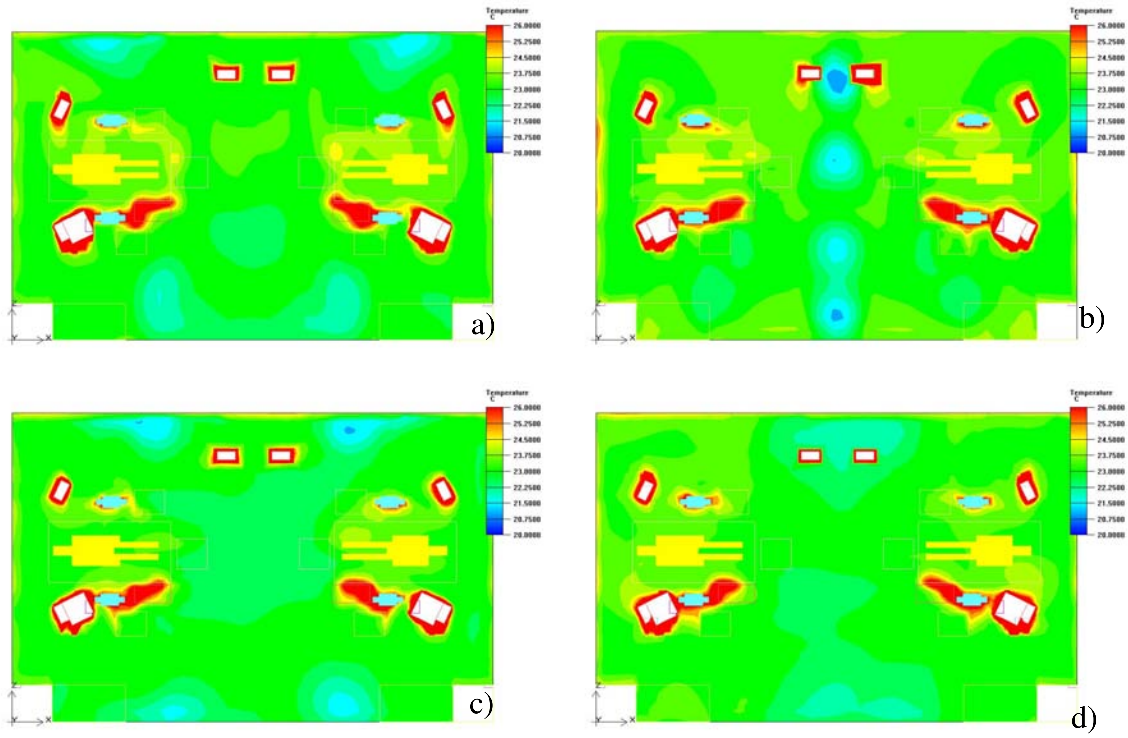

Figure 5. Temperature distribution with air supply diffusers for case a) to d)

Air speed comparison of four concepts

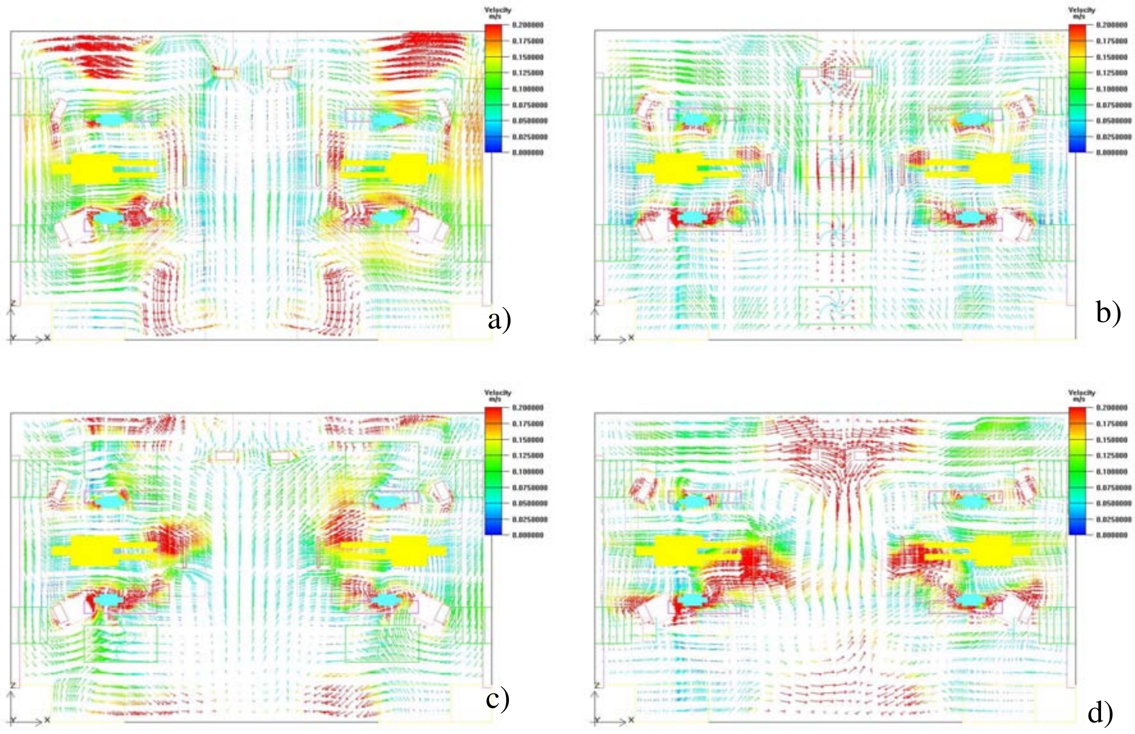

Figure 6. Air speed distribution with air supply diffusers from case a) to case d)

Air stream comparison of four concepts

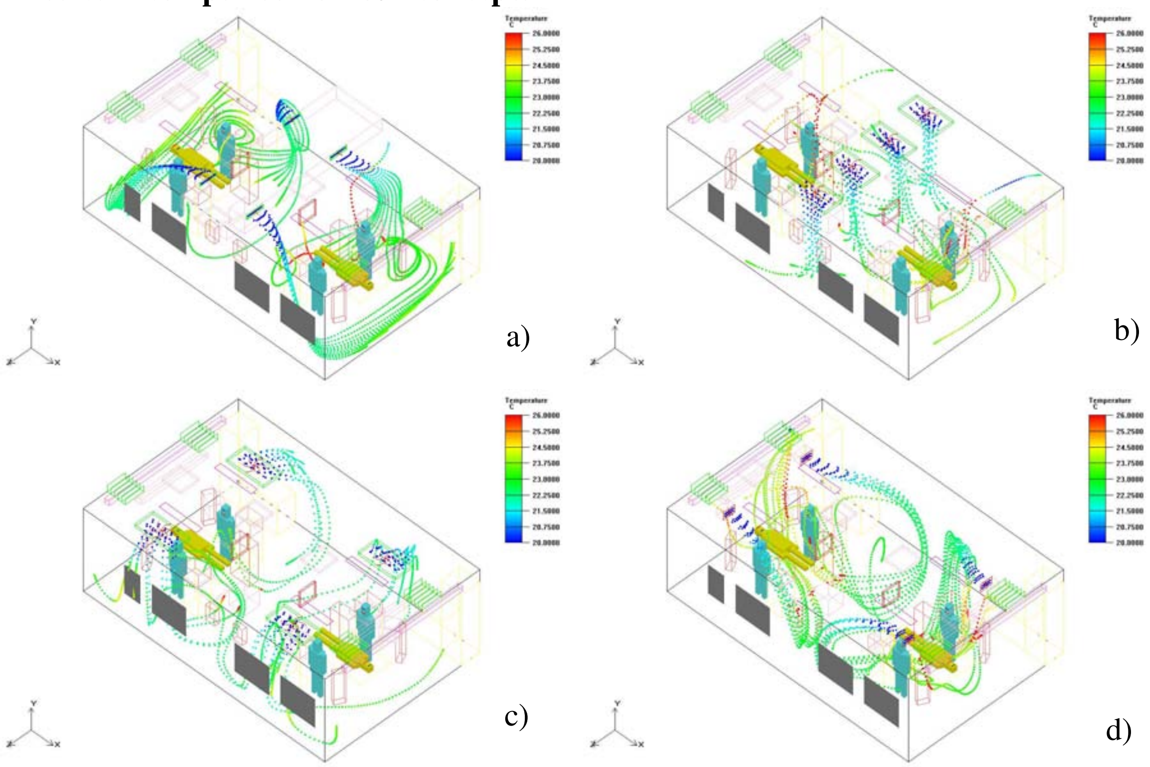

Figure 7. Air stream distribution with air supply diffusers from case a) to d)

SPECIAL CASE STUDY: ASYMMETRIC SITUATION

The purpose of the analysis on this asymmetric situation is to check what will be indoor climate condition in case one bed is fully occupied for the IC and another bed is only with one patient laying on the bed (the most asymmetric condition).

The following figures (8a ~ 8c) is made in such a way that the unoccupied side will only have the TV screen on and all the other apparatus will be switched off. Several control strategies have been tried. And the following figure is with the conditions of:

1. The ceiling of the room is split into two parts. Each takes care of one bed. The unoccupied side will have surface temperature as 22^{o}C and the occupied side as 18^{o}C.
2. The air supply temperature will be kept as the same as 19^{o}C.

By these, the air temperature (at 1.5 meter above the floor) will be ca. 22^{o}C. The air speed in the patient zone is lower than 0.12m/s.

With a smart control strategy, a good indoor climate will be approached.

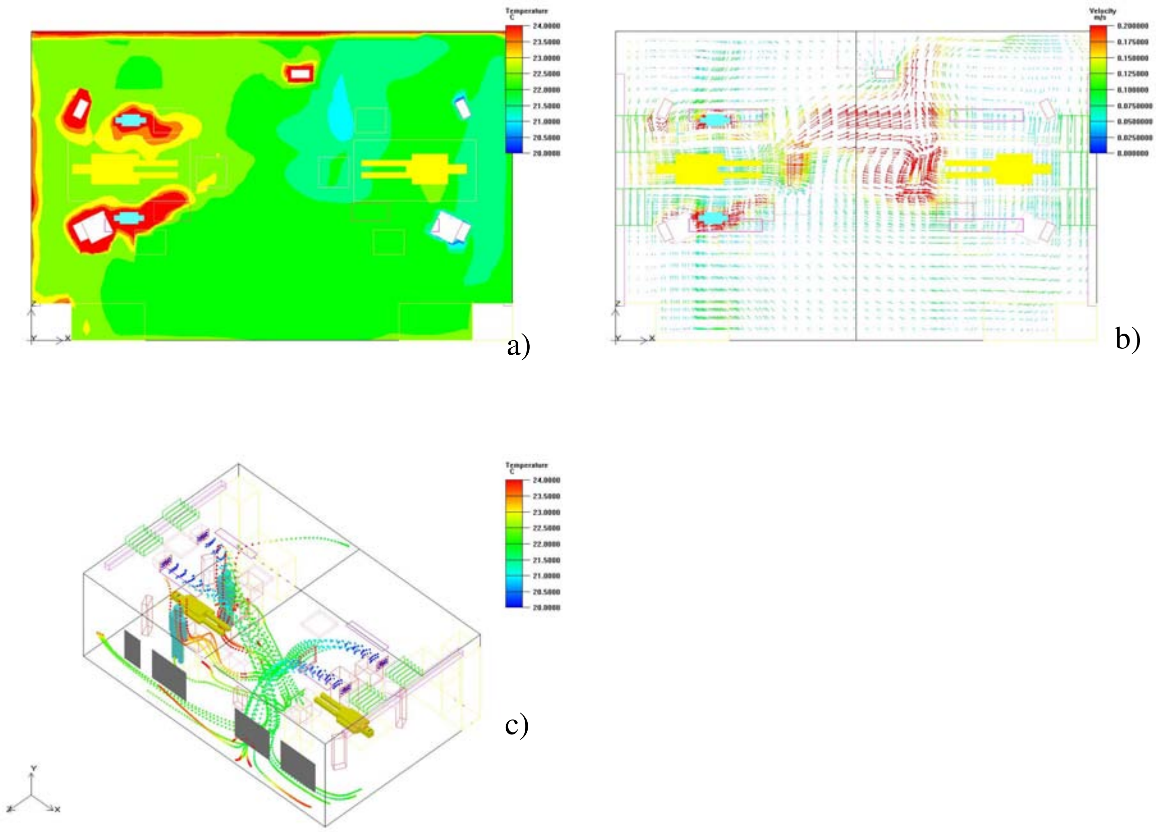

Figure 8. Ceiling diffusers (one side supply) at asymmetric internal heat load a)Temperature distribution; b) Air speed distribution; c) Air stream distribution

CONCLUSIONS

By the CFD simulation, some conclusions can be made:

The air speed in the patient zone is limited and lower than 0.12m/s for all concepts. However, if looking at the temperature distribution in the room, the fourth concept (ceiling diffuser to one side) has the most homogenous temperature distribution. Therefore, it will give the least temperature variation for the nurses who will walk around in the IC room.

Above results are all based on the thorax situation, i.e. the highest internal heat load condition. Under this condition, the average room temperatures are ca. 23.5 ~ 24^{o}C. This means, the room temperature could not approach the requirement of 22^{o}C. This will only occur with high internal heat load. In the reality, two big apparatus, dialysis apparatus and balloon pump/heart & lung device, which are 850 W, will not be used for the most situation. And this will lead to a condition under the requirement and has been studied in another report.

The general conclusions in this study are:

- The highest internal heat load condition (ca. 112 W/m^2) can not lead to a required indoor environment, i.e. the room temperature will be approximate 23.5^{o}C instead of 22^{o}C;
- Different concept has different advantage and disadvantage. The relatively best concept has been chosen with the ceiling diffusers of one direction supply (Waterloo DE/DF 111). This is the case (d).
- The asymmetric situation will not lead to any problem if a smart control strategy is applied.

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Efficiency of different ventilation concepts in meeting rooms

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SUMMARY

A typical virtual meeting room with occupants displays different situations of airflow pattern for different locations of the air supply even though the ventilation system has the same location of the air exhaust in the room. The pollutant is defined here as CO_2 emitted by the occupants in the room. - The investigation is realised with full-scale numerical modelling under Computational Fluid Dynamics (CFD) method

The results of these numerical experiments indicate the following:
Even if the same ventilation system can provide enough outside air for a room in respect of a defined air change rate, there will be different ventilation efficiencies for different locations of the air supply and exhaust. The indoor air quality in a room is strongly affected by the location of air supply and air exhaust of a ventilation system

INTRODUCTION

The task of a ventilation system for non-industrial applications is providing the occupants comfort in room and removing the contaminants as CO_2, VOC and particulate matter etc., as well as unpleasant odour. Indoor air quality and comfort depend on the organization of supply and exhaust openings of the ventilation system for the room. How can ventilation concepts affect comfort and indoor air quality for occupants in it, in particular for places where typically many occupants stay, like meeting rooms or classrooms? We here study different ventilation concepts in the view of ventilation efficiency to show comfort and IAQ in a meeting room. Numerical simulation (CFD) experiments are carried out for this purpose.

It is compared here the conventional airflow pattern e.g. mixing ventilation with the recently favoured ventilation system, displacement ventilation that has been studied in the literature [1]. In respect of real environment in this meeting room, human bodies instead of cylindrical dummies are used for the simulations. The contaminant is here limited just to CO_2 from the occupants.

Three cases are presented. The basic case shows airflow pattern in the room with mixing ventilation at 4 K under-temperature. The other two variants are a mixing ventilation system at 8 K under-temperature, which has the same organization of supply as in the basic case and a displacement ventilation system at 4 K.

The results demonstrate that all concepts here satisfy the demands of comfort and indoor air quality in the room. A displacement ventilation system can provide better ventilation efficiency and keep better IAQ for the occupants concerning contaminants in the room.

MODEL AND BOUNDARY CONDITONS

The meeting room with the mixing ventilation system and the displacement ventilation is defined as shown in figure 1a and 1b. The size of the room is: 5.8 m length, 4 m width and 2.9 m height. Four occupants are sitting on chairs around a rectangular table (2.4 m x1.6 m). For the mixing ventilation, supply openings are four swirl inlets located in the ceiling, the exhaust (0.75 m x 0.045 m) is on the top of the door. For the displacement ventilation, the supply opening (4 m x 0.25 m) is defined on the bottom of the right wall. The exhausts (d = 0.204 m) are the supply openings of the mixing ventilation. The grey zone in figure 1 is defined as occupied zone [2]. The distance from walls is 0.5 m and from ceiling 0.9 m. There is no distance from the floor. The correspondent names of component are demonstrated in figure 1, too.

The commercial simulation software FLUENT serves as a tool for the experiments . RNG $k-\varepsilon$ turbulence model is defined for airflow in room. The simulation of CO_2 and humidity is done by species model in Fluent [3]. The grids have about 750,000 cells (hexahedron and tetrahedron). The simulations run under steady-state conditions.

The simulations are for the transition periods of the year. There are no heating loads for the simulations. The cooling loads come from occupants and lamps. The lamps emit 6 x 58 W. Every occupant emits 75 W of sensible heat (e.g. 1.2 met per occupant) and gives to the environment 60 g/h of dampness and 0.02 m^3/h CO_2 from breathing. The conditions of comfort are according to the standard [4]. The room temperature is defined at 24 °C. The surfaces as walls, table and chairs are adiabatic. The supply air has 6 g/kg moisture and 16 °C respectively 20 °C for mixing ventilation and 22 °C and 6g/kg for displacement ventilation.

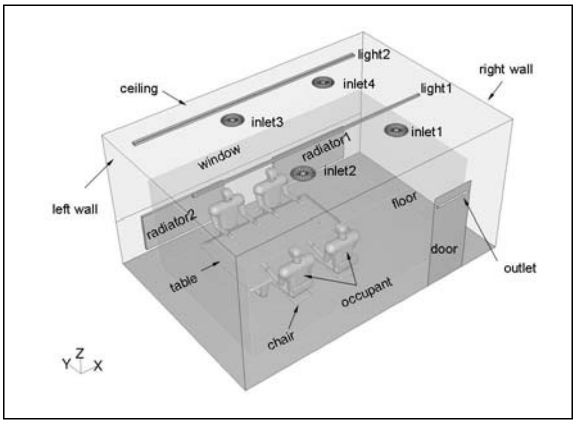

Figure 1a): Meeting room model for mixing ventilation

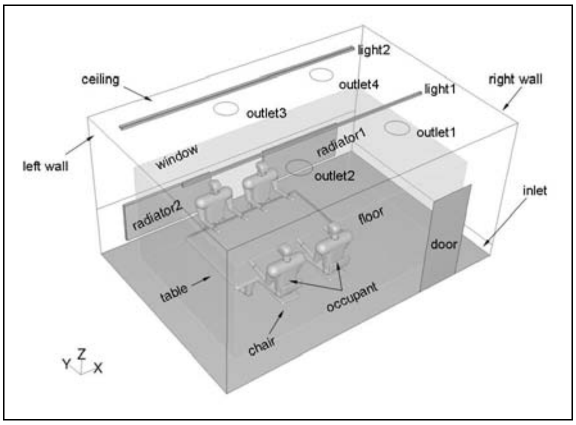

Figure 1b): Meeting room model for displacement ventilation

RESULTS

Airflow pattern and comfort estimation

Figure 2 displays path lines of velocity for mixing and displacement ventilation at 4 K undertemperature. Asymmetric allocation of thermal loads by the occupants in the room cause clearly an air roll clockwise in the room for the mixing ventilation, see figure 2a. The displacement ventilation shows other phenomena. Supply air enters into the room through the supply opening in the right wall, flows towards the opposite wall and then returns to occupants. From there most of the air flows in vertical direction to the exhaust because of the thermal loads of occupants and lamps. The airflow forms a mixing zone above the upper limit of the occupied zone (at 2 m height above floor), see figure 2b. The gradient of temperature near the floor is greater than beneath the ceiling. A comparison of velocity between the three concepts is displayed in figure 6, it is discussed later. Figure 3 and 4 show the temperature distribution for mixing and displacement ventilation each at two x- and y-planes.

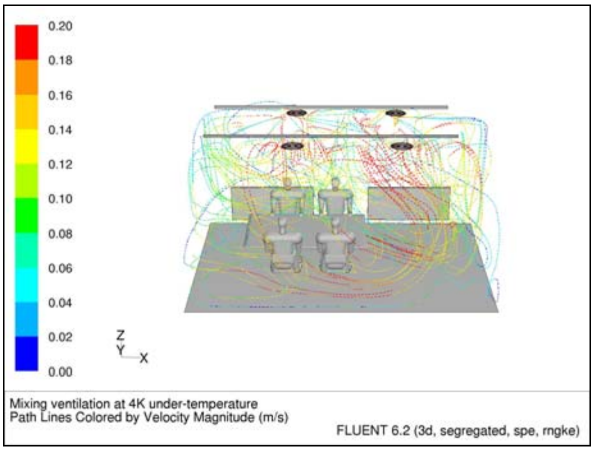

Figure 2a): Pathlines in the mixing ventilation at 4 K under-temperature

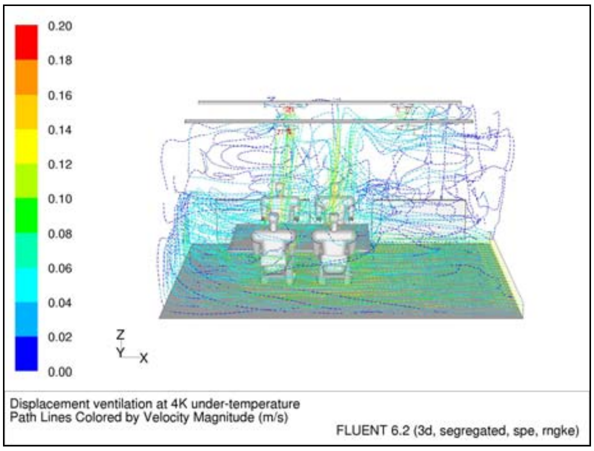

Figure 2b): Pathlines in the displacement ventilation at 4 K under-temperature

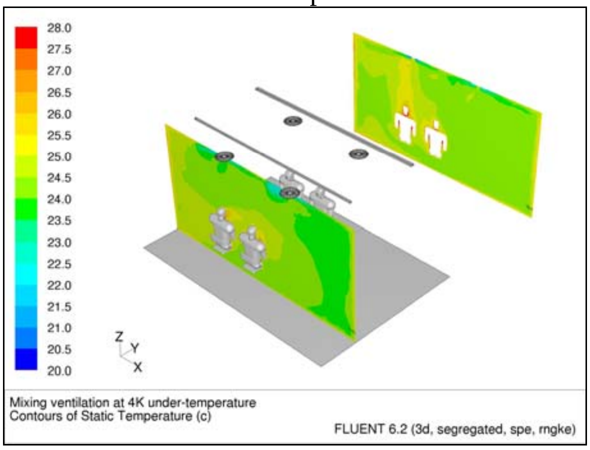

Figure 3a): Temperature distribution at the y – planes for mixing ventilation at 4 K under-temperature

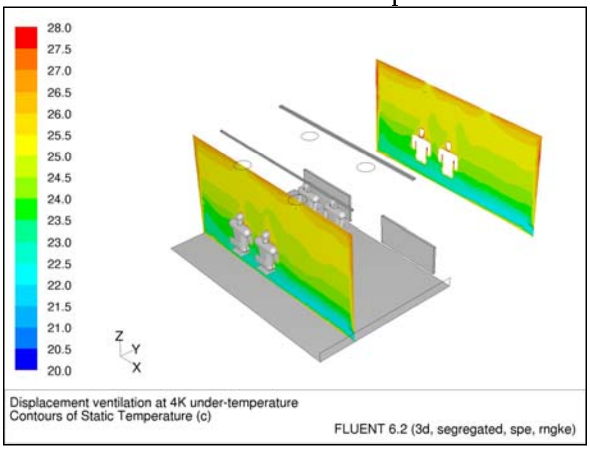

Figure 3b): Temperature distribution at the y – planes for displacement ventilation at 4 K under-temperature

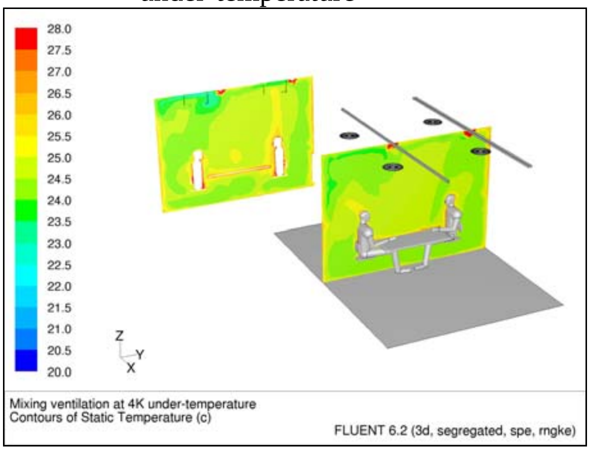

Figure 4a): Temperature distribution at the x – planes for mixing ventilation at 4 K under-temperature

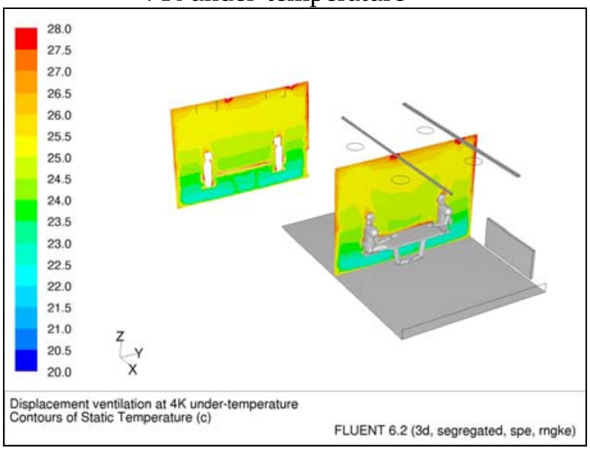

Figure 4b): Temperature distribution at the x – planes for displacement ventilation at 4 K under-temperature

They display clearly an increased temperature level on the top of the room for the displacement ventilation system. In order to achieve a good displaying, the rear y-plane is transferred far from the original location.

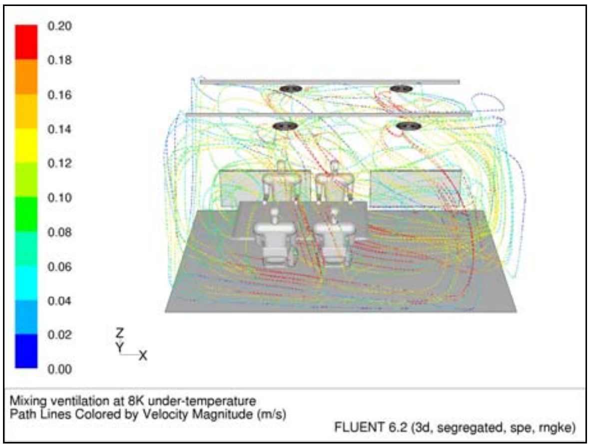

Figure 5a): Pathlines for the mixing ventilation at 8 K under-temperature

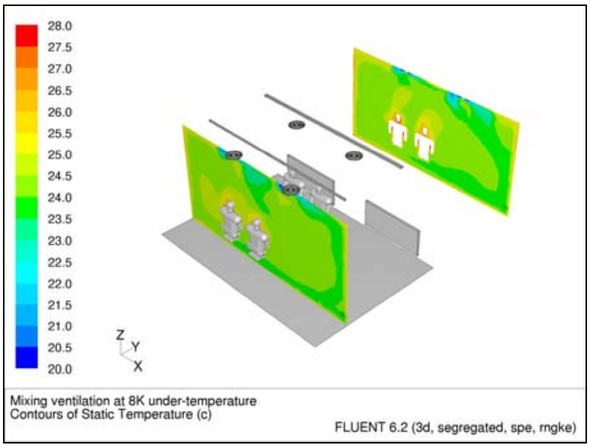

Figure 5b): Temperature distribution at two y – planes for mixing ventilation at 8 K under-temperature

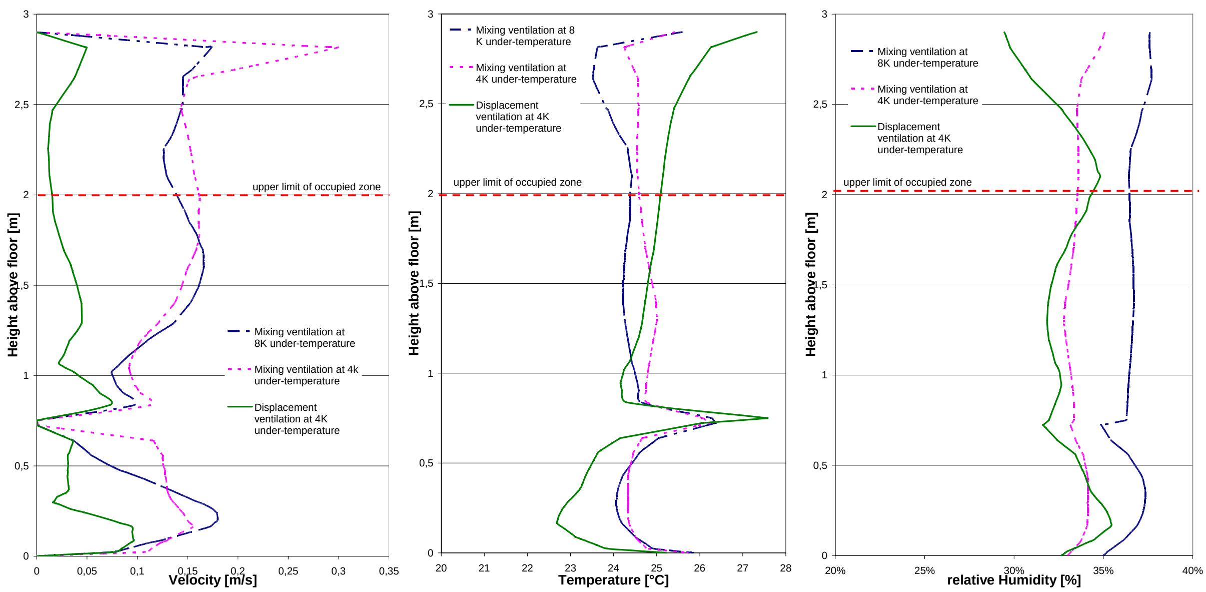

Figure 6: Parameters at the vertical centerline: Velocity, Temperature and relative Humidity

Figure 5 displays mixing ventilation with lower air change rate compared with basic case. Here the clockwise air roll is more distinctive than at 4 K under-temperature. The flow mixing is weaker than for the higher air change rate at 4 K. A vertical centerline from the top to the bottom of the room in the middle of the table is selected; this line is supposed to be representative for comfort and IAQ of the persons, ventilation effectiveness and ventilation efficiency. The left diagram in figure 6 displays the velocity. The displacement ventilation shows very low velocities. All three concepts satisfy the requirement of velocity in the room according to standard [2]. The middle diagram shows very little temperature gradient for all concepts except for the area of the table. The right diagram in figure 6 displays the relative humidity in this centerline. The gradients are very low as well. For the mixing ventilation at 8 K under-temperature the humidity is a little higher than for the two alternative concepts.

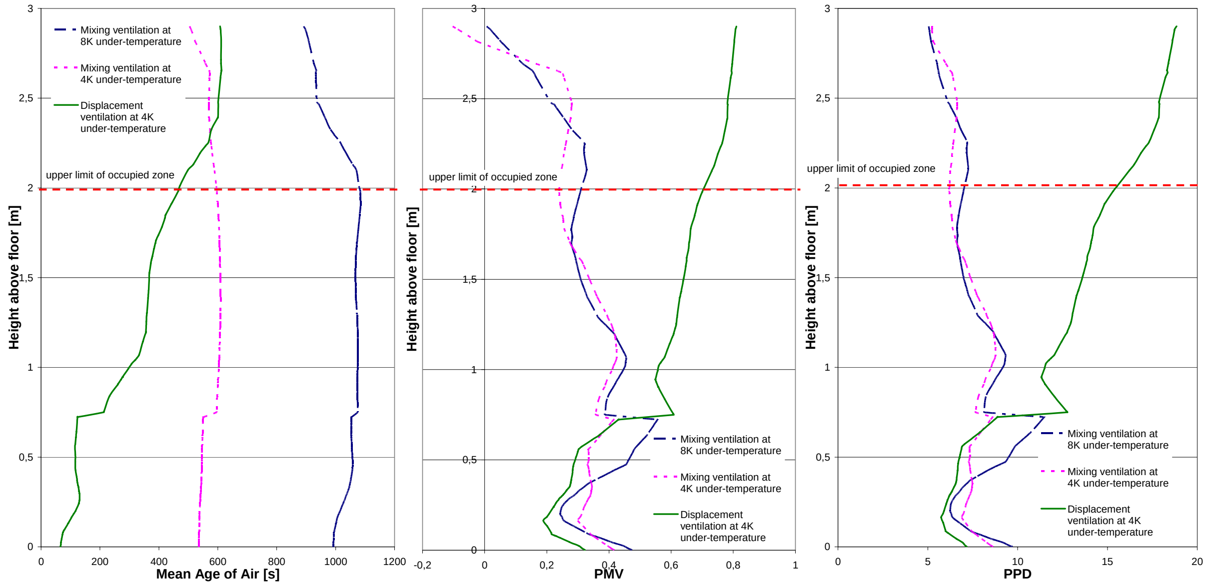

Figure 7: Parameters at the vertical centerline: MAA, PMV and PPD

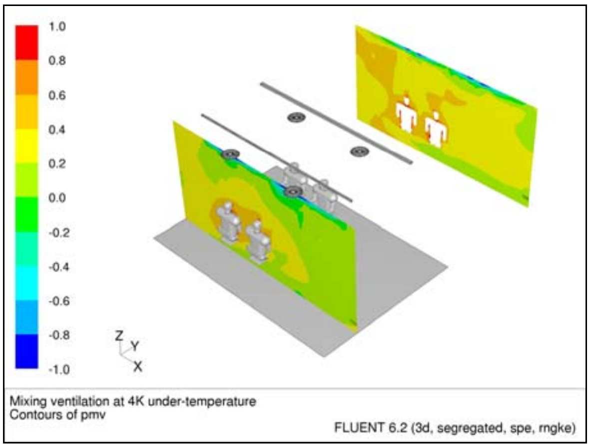

Figure 8a): PMV at y-planes for mixing ventilation at 4 K under-temperature

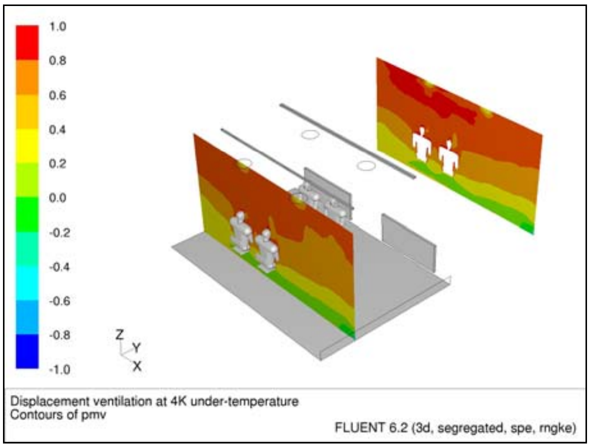

Figure 8b): PMV at y-planes for displacement ventilation at 4 K under-temperature

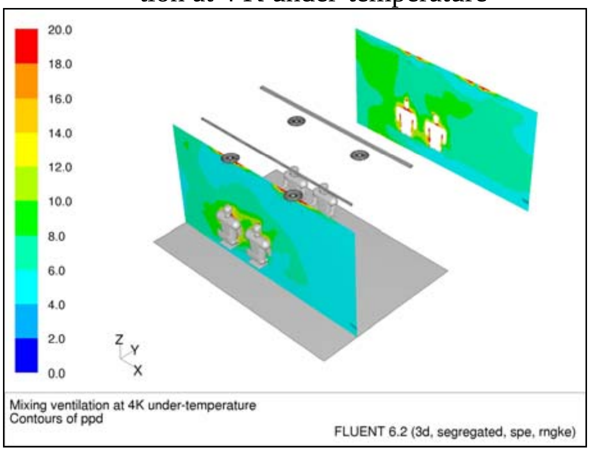

Figure 9a): PPD at y-planes for mixing ventilation at 4 K under-temperature

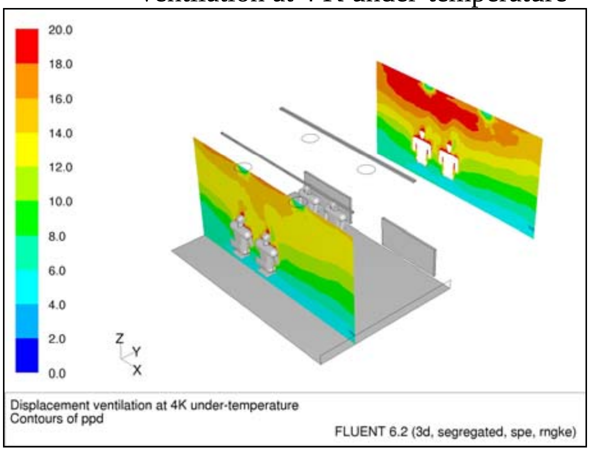

Figure 9b): PPD at y-planes for displacement ventilation at 4 K under-temperature

Accepted criteria for the estimation of comfort are MAA, PMV and PPD [4]. The comparison for the displacement ventilation shows worse quality in respect of PMV and PPD, but better quality in respect of MAA for the occupants, see figure 7. Figure 8 and 9 show the PMV and PPD distributions at two y-planes of the room.

CO_2 distribution and estimation of ventilation effectiveness respectively efficieny

An important criterion for indoor air quality is the CO_2 concentration. It should be less than 1000 ppm according to Pettenkofer [5]. The terms for estimation of ventilation systems are defined as following (in this study, the concentration of CO_2 in the supply air is set to zero):

Air change efficiency $$\eta_a = \frac{\tau_n}{\tau_{r,v}}$$

Contaminant load $$\mu_R = \frac{C_m - C_s}{C_e - C_s}$$

Ventilation effectiveness $$\varepsilon_v = \frac{C_e - C_s}{C_{m-} C_s}$$

Ventilation efficiency $$\eta_v = \frac{\varepsilon_v}{1 + \varepsilon_v}$$

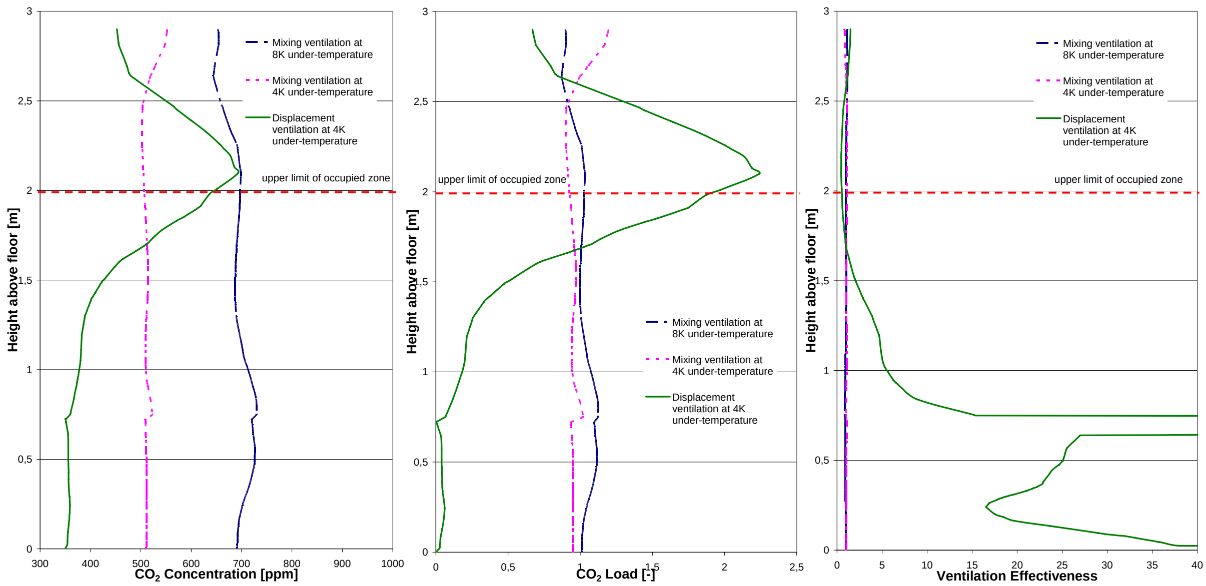

Figure 10: Parameters for CO_2 at the centerline

Concept	Air change rate [1/h]	Air change efficiency	Ventilation effectiveness	Ventilation efficiency
Mixing ventilation at 4K under-temperature	7.19	0.45	1.1	0.52
Mixing ventilation at 8K under-temperature	3.61	0.48	0.97	0.49
Displacement ventilation at 4K under-temperature	7.19	0.67	82	0.988

Table 1: Mean value of ventilation efficiency

The simulations show that the three concepts can keep hold of CO_2 concentration under 1000 ppm, see left diagram in figure10.

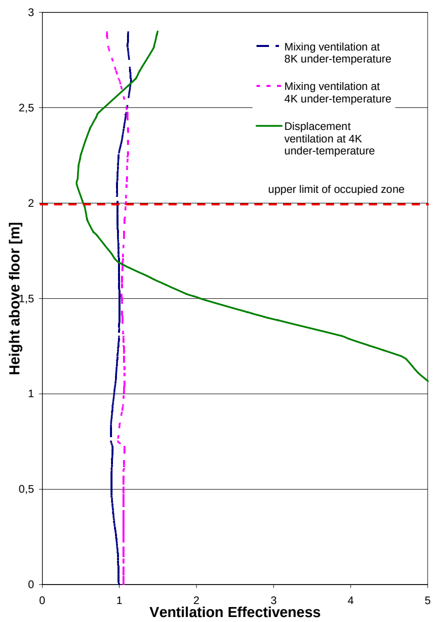

Figure 11: Ventilation Effectiveness (larger scale)

The displacement ventilation system can reduce the CO_2 concentration in the occupied zone distinctively, the level of concentration remains very low, even though it has the same air change rate as the mixing ventilation system at 4 K under-temperature. Naturally, the CO_2 concentration is higher for the mixing ventilation system at 8 K under-temperature because of half of airflow quantity. In the view of CO_2 load (middle diagram in figure 10), the displacement ventilation system has also advantages comparing to mixing ventilation. There is no significant difference between both mixing ventilation systems. Figure 11 shows the content of the right diagram of figure 10 drawn to a larger scale. It shows that the displacement ventilation has a very high effectiveness compared to mixing ventilation.
Figure 12 to 14 show the contours of CO_2 concentration and load as well as the ventilation effectiveness at two y-planes of the room.

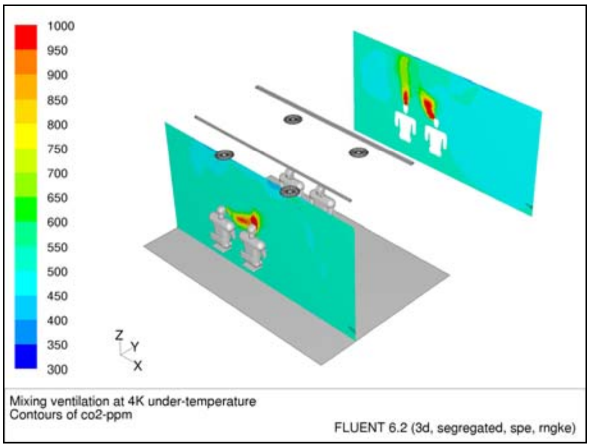

Figure 12a): CO_2 concentration distribution at two y-planes for the mixing ventilation at 4 K under-temperature

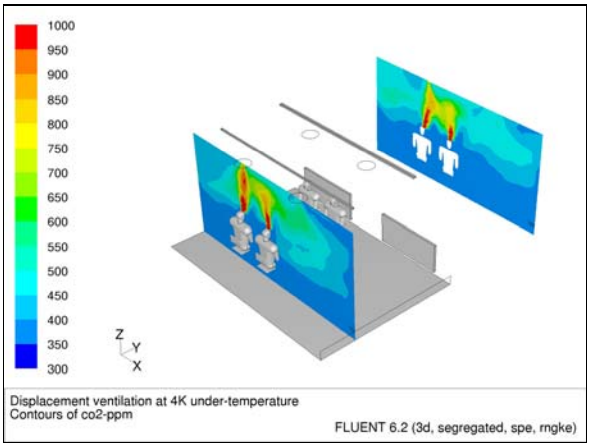

Figure 12b): CO_2 concentration distribution at two y-planes for the displacement ventilation at 4 K under-temperature

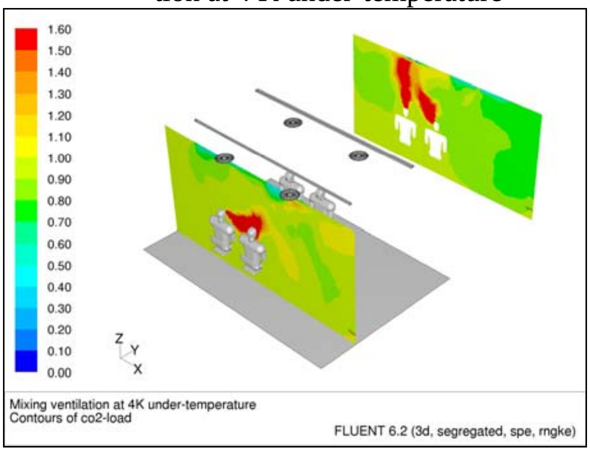

Figure 13a): CO_2 load distribution at two y-planes for the mixing ventilation at 4 K under-temperature

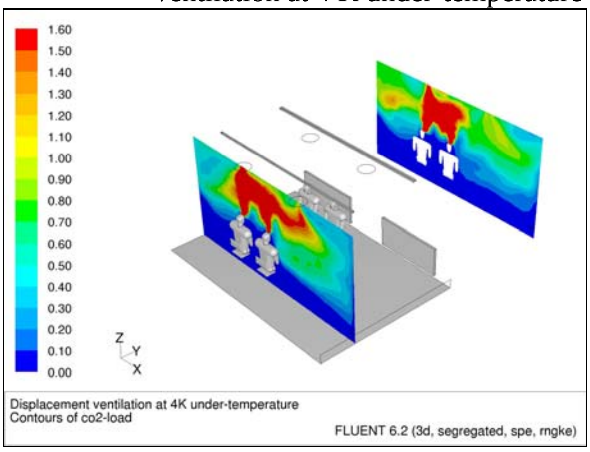

Figure 13b): CO_2 load distribution at two y-planes for the displacement ventilation at 4 K under-temperature

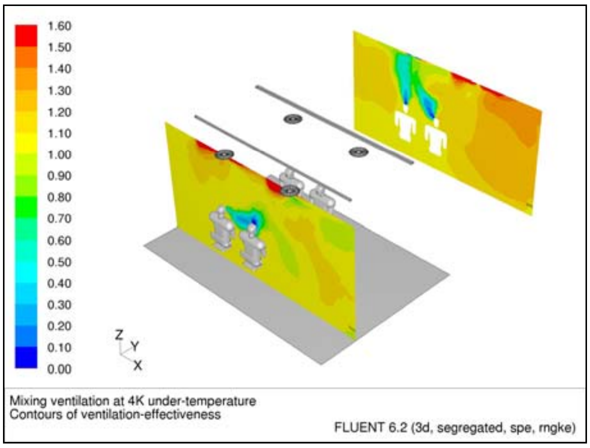

Figure 14a): Ventilation effectiveness at two y-planes for the mixing ventilation at 4 K under-temperature

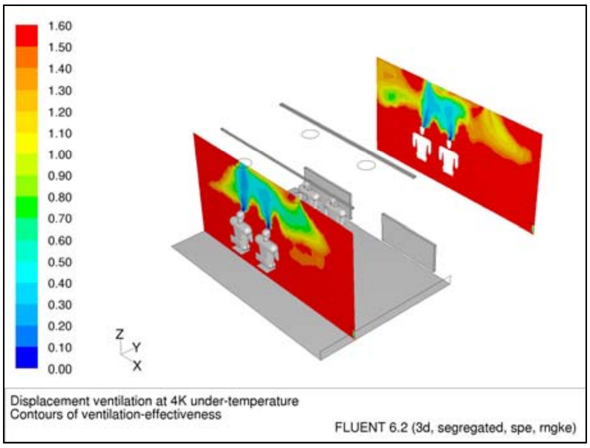

Figure 14b): Ventilation effectiveness at two y-planes for the displacement ventilation at 4 K under-temperature

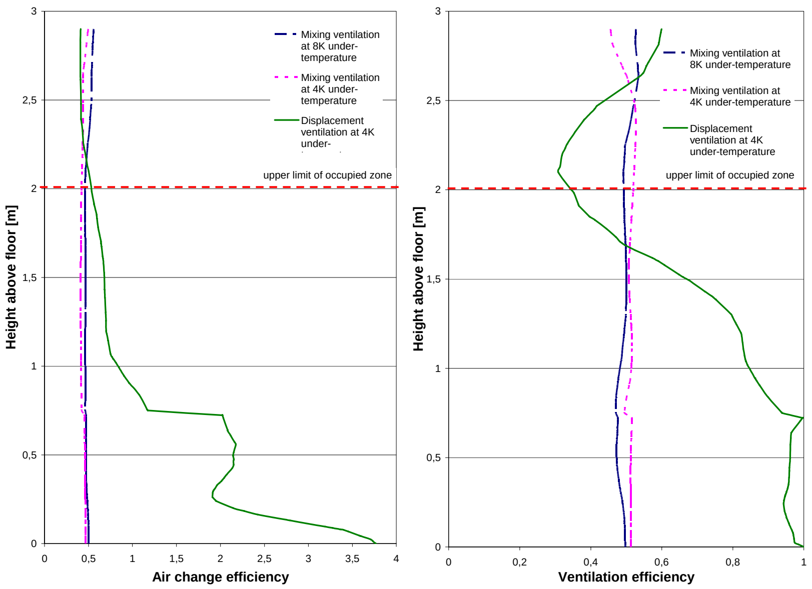

Figure 15: Parameters at the vertical centerline: Air change efficiency and ventilation efficiency

A ventilation system is estimated among others by air change efficiency and ventilation efficiency; see figure 15 and table 1. The air change efficiency for mixing ventilation is at about 0.5. The displacement ventilation has higher air change efficiency than mixing ventilation. The same results are obtained for ventilation efficiency. Figure 16 gives a quantity display for CO_2 concentration as well as CO_2 load and ventilation effectiveness in occupied zone.

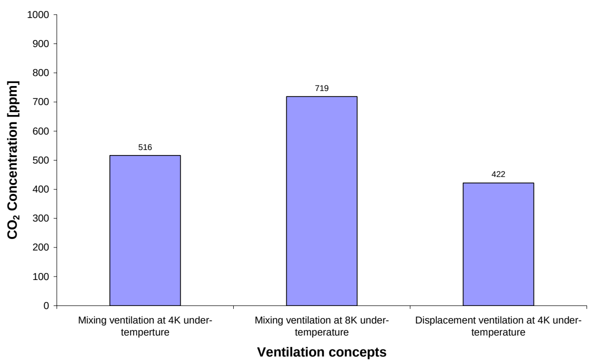

Figure 16a): Diagram of CO_2 concentration in occupied zone for different ventilation concepts

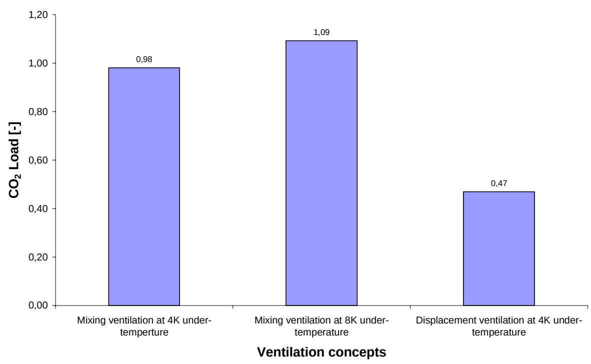

Figure 16b): Diagram of CO_2 load in occupied zone for different ventilation concepts

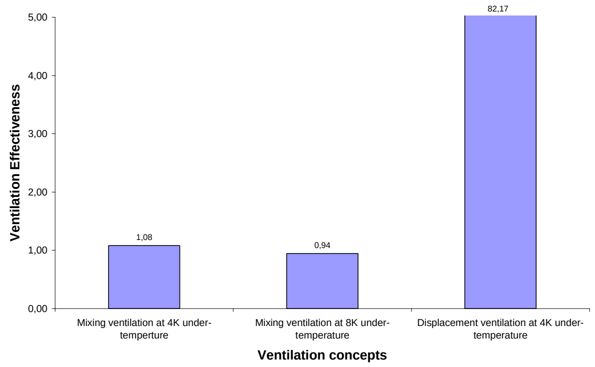

Figure 16c): Diagram of ventilation effectiveness in occupied zone for different ventilation concepts

Names and units

C_e	Contaminant concentration in exhaust [kg/kg]
C_m	Mean contaminant concentration in room or in occupied zone [kg/kg]
C_s	Contaminant concentration in supply air [kg/kg]
PMV	Predicted Mean Vote
PPD	Predicted Percentage Dissatisfied
ε_v	Ventilation effectiveness [-]
η_a	Air change efficiency [-]
η_v	Ventilation efficiency [-]
μ	Contaminant load [-]
τ_n	Time constant [h]
$\tau_{r.v}$	Mean age of air [s]

DISCUSSION

All compared ventilation concepts can provide good comfort in room. The detailed results show that displacement ventilation cannot provide better comfort than mixing ventilation in terms of temperatures and air velocity. The particular advantage of displacement ventilation is the providing of higher ventilation effectiveness that is to say to achieve rather low concentration of contaminant in the room, particularly in the occupied zone. The satisfaction of thermal environment for occupants cannot guarantee the satisfaction of indoor air quality. A new criterion for the estimation of a comfortable and clean environment should be considered in the future.

Further studies also should take other contaminants like VOC or odour into consideration. Heat and contaminant sources in other places in the room or parts of the human body can also be investigated with this simulation model.

A detailed model of the human body is very time-consuming for grid creating and simulation running. Therefore, it is not advisable for situations where many occupants stay in a room (i.e. classrooms or other bigger locations). The affect of the modeling of the human body (i.e. cylindrical dummies instead of detailed body modeling) to the results of such simulations should be investigated.

ACKNOWLEDGEMENT

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Perceived Control of Stratum Ventilation

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SUMMARY

Current control strategies for central air conditioning typically adopt a machine-centered, energy-consuming approach that focuses on creating constant, uniform neutrality-conditions which might actually be perceived by some occupants as thermal monotony or sensory deprivation. There are needs for more sophisticated and responsive environmental control strategies, enhanced levels of thermal comfort and acceptability among occupants. Stratum ventilation, a new ventilation mode proposed recently by the authors, has the potential to strike a balance between fully automated controls at the system side and manual controls that the users are able to alter. Stratum ventilation should be integrated with specific control strategies for various applications and varieties of control option available to occupants. A working example using CFD simulations is given to illustrate the performance of the proposed new system. Fresher air is directly introduced to breathing zone hence air age of the air layer should be younger. Air temperature gradient is low in the occupied zone. Because less fresh air bypasses occupants, fresh airflow rate may be reduced. The air supply temperature is usually high for stratum ventilation. The evaporating temperature of the refrigerating plant can also be lifted, which results in higher coefficient of performance.

INTRODUCTION

Conventional ventilation modes applied in an air-conditioned non-industrial indoor space are broadly categorised into two groups, namely: mixing ventilation and displacement ventilation as task ventilation is less widely applied.

Mixing ventilation is a traditional mode of ventilation and is still widely used. It could take care of both heating and cooling with highly varying load patterns. The mixing is achieved by supplying the ventilation air as a high velocity jet to entrain the air already in the room. Mixing ventilation could provide comparably uniform air temperature distribution in the occupied zone. However, it also leads to problems, such as poor IAQ, air draft in the occupied zone and some other thermal discomfort. In addition, it normally consumes more energy than displacement ventilation.

Displacement ventilation has been used quite commonly in Scandinavia during the past twenty years as a means of ventilation in industrial facilities to provide better IAQ and to save energy. More recently, its use has been extended to ventilation of offices, classrooms, commercial buildings and other non-industrial premises. In contrast to mixing ventilation, buoyancy forces (induced by heat sources) govern the airflow in displacement ventilation. Because the airflow is thermally driven, this mode of ventilation functions satisfactorily only when excess heat is to be removed. In a room ventilated by displacement, the air quality in the breathing zone is usually better than in a mixing system operated with the same airflow rate. And ventilation efficiency of a displacement-ventilated room is also significantly better than that of the mixing ventilated room [1]. It has been found that an increase in ventilation efficiency is often achieved at the breathing level [2]. But there are more severe restrictions due to thermal discomfort in a displacement system than in a mixing system. ISO 7730 recommends a maximum air temperature gradient of 3K between 1.1m and 0.1m above the floor [3], which corresponds to a percentage dissatisfied of 5%. However, the ASHRAE Standard recommends the same temperature gradient between 1.7m and 0.1m above the floor [4], which related to a standing person. Otherwise it will cause thermal discomfort. So supply air temperature and airflow rate must be carefully studied to assure proper temperature distribution and velocity distribution. Often a displacement ventilation system excessively cools the lower part of a room, where usually heat sources are absent, resulting in discomfort and energy waste.

As far as the IAQ of breathing zone and energy efficiency are concerned, the most effective way is task ventilation. It may be used to remove excessive cooling load and maintain a comfortable indoor environment. Task ventilation systems supply air through nozzles located near occupants (e.g., at an edge of a desk). Potential draft exists because of the short distance between supply gears and the occupants. A field study found that workers in a task ventilated office satisfied with the thermal conditions because they could individually control local environment [5]. The occupants can control the temperature, flow rate, and direction of air from the nozzles. The measurements conducted by Faulkner *et al*. [6,7] showed that the age of air at the breathing level with task ventilation was approximately 30% less than that with mixing ventilation. However, application of task ventilation depends much on indoor furnishings. On one hand, sometimes it is difficult to equip nozzles and connect duct in various indoor spaces. On the other hand, some occupants in certain spaces do not normally stay in a fixed location, e.g., the situation in a retail shop. These limit the use of task ventilation.

DEVELOPMENT OF THE CONCEPT OF STRATUM VENTILATION

Mixing ventilation, displacement ventilation plus task ventilation all have their limitations. To overcome these problems, a new ventilation mode is proposed [8]. The new mode should be able to provide good IAQ in breathing zone, to have minimum temperature difference between head and ankle levels to obtain thermal comfort in occupied zone, to equip duct and terminals conveniently, and to have high energy efficiency.

The underlying principle of displacement ventilation implies that in an air-conditioned room, the conditions of IAQ and thermal comfort beyond the occupied zone (say $H > 2$ m) are of little interest. By the same token, the IAQ beneath the breathing zone (say $H < 0.8$ m) is less important unless the occupants are expected to be very young children. For conventional ventilation modes, the air for breathing is transported by the boundary layer around the body of a human occupant and the air quality is a weighted average of the air quality in the room from the breathing level to floor level. Ventilation efficiency would be maximized if air is supplied directly into the breathing zone, and the air forms a well controlled “fresher air layer” to fill the breathing zone. The thickness of the fresher air layer depends on the nature of occupancy. At the same time, a quasi-stagnant zone is also form between the breathing zone and the floor (say $0 < H < 0.8$ m). The temperature in the quasi-stagnant zone should not be as low as that of displacement ventilation. The problem of “cold ankles” could be solved. Energy can also be saved by avoiding over-cooling of the lower part of a room.

The stratum ventilation works by creating a layer of fresher air in the occupants’ breathing zone. This is done by placing large supply inlets at the side-wall of the room just above the height of the occupants, sedentary or standing depending on modes of occupancy. The fresh air enters the room and then gradually loses momentum, further away from the supply. The fresh air supply is sufficiently strong enough to provide adequate fresh air of young age. The range of face velocity and the locations of air supply and exhaust should be carefully optimized to break the boundary layer around the body of a human occupant, to minimize risks of draft and cross contamination. The thickness of the fresher air layer required depends on the nature of occupancy. At the same time, a quasi-stagnant zone is also form between the breathing zone and the floor (say $0 < H < 0.8$ m). The temperature within the quasi-stagnant zone should be reasonably controlled. In addition the supply of air at this level increases the convection effect of the heat and helps to displace the contaminants into the unoccupied zone. It brings fresher air to breathing zone than that of conventional modes of ventilation. Indoor air should mix well and air temperature gradient should be low hence not to cause thermal discomfort. Because air is supplied directly to breathing zone, less fresh air bypasses occupants. Thus there are also possibilities to reduce fresh airflow rate for energy saving. Also because air is supplied directly to breathing zone, the air supply temperature should usually above 20°C. This implies that the evaporating temperature of the chiller(s) could also be higher, which could result in higher coefficient of performance (COP), an indicator of refrigerating plant energy performance.

STRATUM VENTILATION FOR ENHANCING PERCEIVED CONTROL

Non-isothermal airflows enter a room via the air supply outlets positioned in a convenient way as long as it is slightly above the breathing height. A horizontal air layer should be formed with no reflux and minimal entrainment. Air age of the air layer should be young. Compared with more conventional modes of ventilation, it would bring a greater amount of fresh air into the breathing zone. The range of face velocity and the locations of air supply and exhaust should be carefully optimized to break the boundary around the body of a human occupant layer. Indoor air should mix reasonably well and air temperature gradient should be low so as not to cause thermal discomfort. Because air is supplied directly to breathing zone, less fresh air bypasses the occupants. There are also possibilities to reduce fresh airflow rate resulting in energy savings. The range of air velocity and supply types should be carefully matched to form the fresher air layer at the breathing zone. The positioning of the supplies should be carefully studied to avoid any reflux flows in the breathing zone of the room.

The occupants may opt for a control mode — manual or automatic:

- For the manual mode, temperature and air velocity (air recirculation) should be adjustable in wider ranges by the occupants; and

- For the automatic mode, multiple control options based on combinations of different temperatures and air velocities should be available to the occupants. To avoid the problem of draft, each air outlet will be equipped with an occupancy sensor to determine the proximity of occupants. The occupancy sensor and volume control damper of each and every air outlet are linked via a central control system. The control system will hence be able to control the supply air accordingly. As a starting point, a simple control strategy is adopted. If an occupant is less than 0.5 m to the face of the outlet then the air supplied will be stopped. If the occupant is greater than 0.5 m but less than 1 m to the outlet then the volume flow rate is reduced by up to ¾, but the face velocity is not less than 0.1 m/s. The basic principle is illustrated in Figure 1.

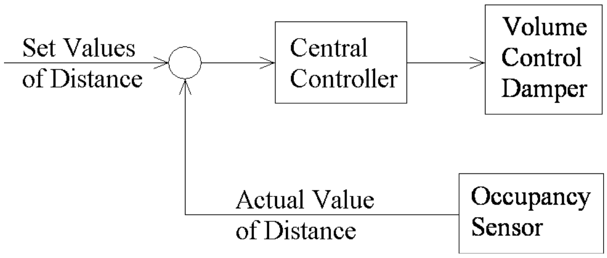


Figure 1. Automatic control of stratum ventilation

To maintain the required air supply volume to the room, the control system increases the fresh air supplied to the other parts of the room. The increase in supply will be optimized so that the increase will be for those outlets not far away from the closed outlet (the occupants). A fuzzy logic system consisting of a database relating the drop distance of an air jet based on the Archimedes number could be developed.

By directly or indirectly opting for a higher temperature setpoint, the occupants actually lower the cooling load of the room and save energy. This should be made known to the occupants. The research also attempts to increase the air recirculation applied in the design in order to remove a cooling load up to 200 W/m^2 of floor area. This is an inevitable problem because more and more equipment that generate a considerable amount of heat is being introduced indoors. The study assesses the stratum ventilation for enhancing perceived control in terms of IAQ and thermal comfort.

METHODOLOGY

The new concept of stratum ventilation needs to be tested. Two main approaches are available for the study of airflow and pollutant transport in buildings: experimental investigation and computer simulation. Direct measurements give the most realistic information but are expensive and time consuming. The purpose of the research at the first stage was to find out whether the new ventilation mode is feasible. Computational Fluid Dynamics (CFD) technique was applied. Due to limited computer power and capacity available at present, turbulence models have to be used in the CFD technique in order to solve flow motion. The use of turbulence models leads to uncertainties in the computational results because the models are not universal. Therefore, it is essential to validate a CFD program by experimental data [9].

A CFD model based on the Re-Normalised Group k-ε model and wall function was used for simulation. Validated for both mixing ventilation and displacement ventilation, the model has been tested for hundreds of cases, including offices, classrooms, shops and workshops with air supplied upwards and downwards [10]. The flow characteristics between displacement ventilation and mixing ventilation are similar – both have strong pressure and buoyancy driven flows [11]. The condition of stratum ventilation is found to be between those of mixing ventilation and displacement ventilation based on identical internal layout and similar boundary conditions. The very model was also validated recently for stratum ventilation with air supplied horizontally. A commercial CFD code CFX was used for the computations. By default, the code uses the finite-volume method and the upwind-difference-scheme for the convection term [12]. Simulation results agree well with experimental results [13].

Table 1. Major parameters for the typical classroom

No. of person	West wall [W]	Window [W]	Projector [W]	Lamp [W]	Air circulation [m^3/s]	Fresh air intake [l/person/s]
41	900	3168	350	1170	0.732	10

TEST CASE

The air velocity, the percentage of dissatisfied people due to draft (PD), and the predicted percentage of dissatisfied for thermal comfort (PPD) are widely used as criteria to evaluate thermal comfort [14,15]. All these performance parameters are determined by the thermal and flow boundary conditions, such as the size and geometry of the space, rates and temperatures of airflows and heat sources.

The case of a typical classroom is used to illustrate the effectiveness of the stratum ventilation. The classroom layout is shown in Figures 2. The major parameters are shown in Table 1. The classroom is supplied by 6 supply inlets located at the east wall (x = 8.4 m), and also by another 6 supply inlets at the west wall (x = 0.0 m). There are a total of 40 sedentary occupants and 1 standing occupant in the room. There is a long window at the west side of classroom.

DISCUSSION OF RESULTS

Airflow Pattern

The supply velocity is set low at 0.3 m/s to minimize any draft effect to the occupants. The air initially enters the room and gradually loses momentum and falls towards the floor under gravity. The air falls across the tables. This forms a stratum layer of fresher air in the breathing zone. This can be clearly seen in the cross-section y = 6.0 m and x = 0.5 m in Figure 3. The convection effects of the air can also be seen to rise near the external wall and window as it is heated. This effect is not observed on the other three internal partitions, as the adjacent rooms are at the same temperature.

The air velocity can be seen to be low in the order of less than 0.3 m/s in the occupied zone. Higher velocities however are observed near the external wall. Velocities tend to be in the range of 0.3 to 0.5 m/s. At the external wall location the airflow rises rapidly into the ceiling area. The highest velocities are at the window location with velocities averaging between 0.4 and 0.5 m/s. This area is the location of the most intensive heat source.

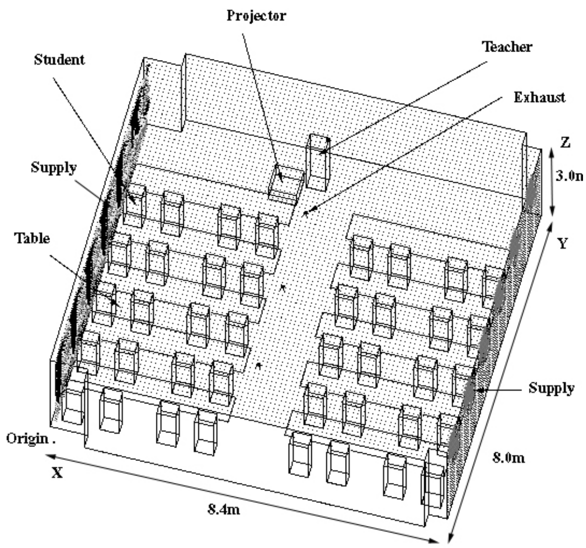
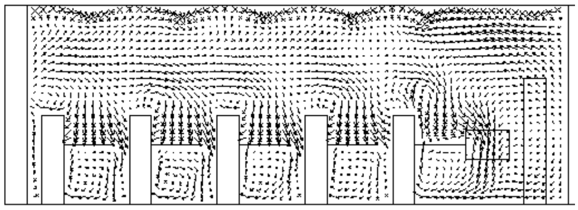
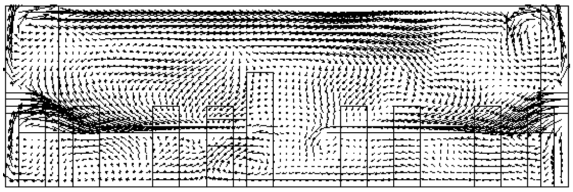


Figure 2. Layout of the typical classroom

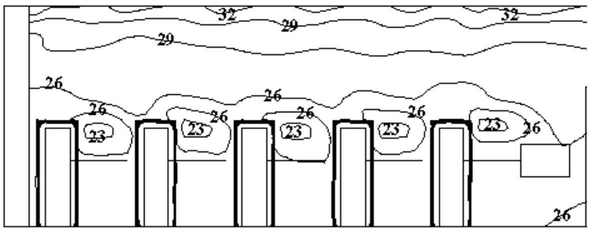


(a) x = 0.5 m

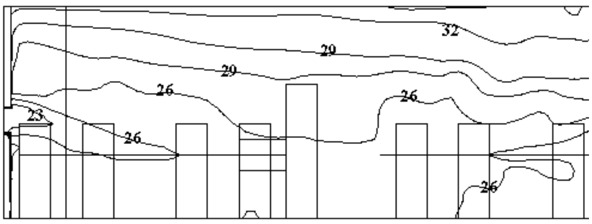


(b) y = 6.0 m

Figure 3. Airflow pattern

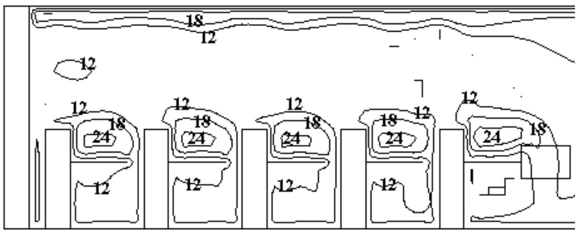


(a) x = 0.5 m

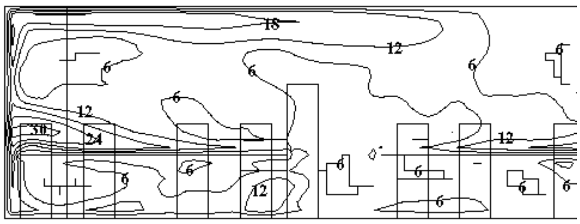


(b) y = 6.0 m

Figure 4. Temperature distribution



(a) x = 0.5 m



(b) y = 6.0 m

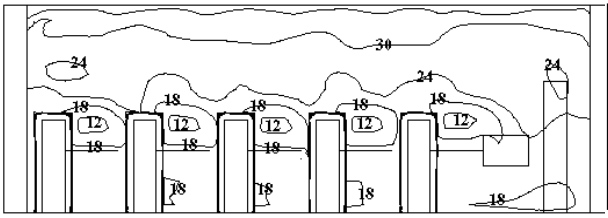
Figure 5. PD index (%)

Temperature Distribution

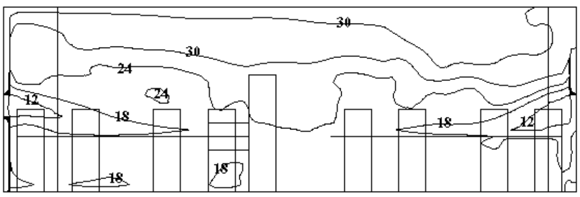
The temperature is nearly uniform in the horizontal direction, except close to the supply inlets as shown in Figure 4, which reduces any draft effects due to temperature differences. The temperature is 23°C at the floor level and at the vicinity of the supply inlets, and is 26°C at the centre of the room. The supply temperature is 21°C. The room temperature at a distance away from the air supplies increases due to the various heat sources, such as external heat transmission, equipment, occupants, etc. As a result the temperature at the floor is higher than that of the supplied air. The temperature is even higher at the ceiling due to the loads including the heavy lighting. However the ceiling is within the unoccupied zone so the high temperature (slightly above 30°C) has little effect on thermal comfort. The effect of radiation due to temperature stratification of this kind is also insignificant because of the small temperature difference. The highest temperature in the occupied zone is near the projector in front of the teacher, which is 3°C higher than the average room temperature. Temperature gradient is maintained within the range of 3°C as stipulated in ISO 1984 and ASHRAE 1992.

Predicted Dissatisfied (PD) and Predicted percentage of dissatisfied (PPD)

The PD levels are higher at the supply inlets, at around 16% to 20%, Figure 5. This is at a distance of approximately one meter from the supply. The rest of the room is around 6% and rises slightly to 12% close to heat sources such as occupants and the projector. The PPD levels are mostly around 8% in the occupied zone and 10% in the unoccupied zone, as shown in Figure 6. In the vicinity of a supply outlet (< 1 m), the PPD levels increase gradually from 6% at the face of the supply outlet. The recommended level is 15% for both PD and PPD (ISO 7730).

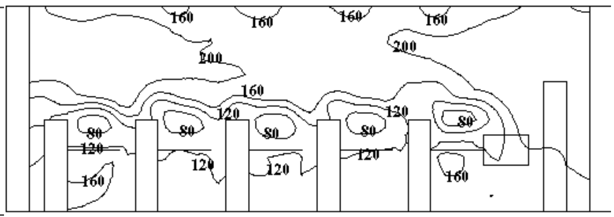


(a) x = 0.5 m

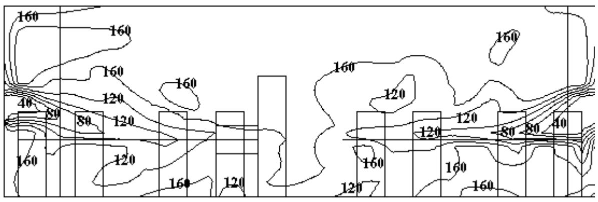


(b) y = 6.0 m

Figure 6. PDD index (%)



(a) x = 0.5 m



(b) y = 6.0 m

Figure 7. Mean age of air (s)

Mean age of air

The mean age of air is a good indicator for the effectiveness of the stratum ventilation. As shown in Figure 7, at the centre of the room the air age approaches 160 s. Approaching the side of the room the air age decreases progressively from 120s to 80s. The location of the inlets at the side of the room means that air is supplied into the room unimpeded by furniture, resulting in fresher air throughout the classroom.

CONCLUSION AND IMPLICATIONS

For the illustrative example, the stratum ventilation with perceived control has shown the potential to be effective in providing satisfactory indoor thermal comfort and air quality. In general the calculated PPD and PD levels are satisfactory in most parts of the classroom. The system is able to provide air of young age to the breathing zone without the penalty of over-cooling lower part of a room.

The results indicate that there is potential application for this system. Further research needs to be conducted to investigate the effects of various parameters such as air velocity, temperature and locations of supply to improve the performances the new type of ventilation system under perceived control.

ACKNOWLEDGEMENTS

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Ventilation of ETA 3 rooms in buildings

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SUMMARY

The European standard prEN 13779 – Ventilation for non-residential buildings [1] distinguishes four categories of decreasing air quality, ETA 1 to 4, for rooms in buildings. We tested the separation effectiveness between a special designed smoking porch, which is a typical ETA 3 room, and the adjoining lobby at Delft University of Technology. The ventilation effectiveness in the smoking porch, which was heated and ventilated by a pseudo-displacement ventilation system, was measured as well.

For the ETS measurements we used an innovative Ultra Fine Particle sensor, newly developed by Philips Research. The measurements were carried out at four different ventilation rates, ranging from 4.8 to 19.2 $dm^3/s.m^2$, with a difference of -1.9 $dm^3/s.m^2$ between supply and exhaust. In each test approximately 40 cigarettes were burned. The measured separation effectiveness was 0.97 in all tests, which is higher than is found in literature on this subject. The ventilation effectiveness, expressed in the Contaminant Removal Effectiveness, is low, which can be attributed to the cold downdraught along the outer facade, which mixes with the supply air. In order to realize a high separation effectiveness, a ventilation rate of 4.8 $dm^3/s.m^2$ with a difference of -1.9 $dm^3/s.m^2$ between supply and exhaust is sufficient. In order to obtain a passable air quality in the smoking porch the ventilation rate should be 5-6 times higher

INTRODUCTION

The European standard prEN 13779 distinguishes four categories of decreasing air quality, ETA 1 to 4, for rooms in buildings. Class ETA 3 represents rooms with a high degree of air pollution due, for example, to vapour emission, processes or chemical pollution which substantially reduce the air quality. Rooms cited as examples are toilets and washrooms, kitchens, some chemical laboratories, photocopying rooms and specially designed smoking rooms. The ventilation of such rooms serves two purposes. Firstly, as much as possible the air from these rooms must be prevented from escaping into the cleaner ETA 1 and ETA 2 rooms. As a parameter for this we have introduced the term separation effectiveness. This is affected by the underpressure to be created in the smoking room, whereby wind loading and infiltration, geometry and position, and the separation between ETA 3/4 and ETA 1/2 rooms play a role. Secondly, the best possible air quality must be provided for the people who visit or work in these rooms. This is affected by the ventilation rate and ventilation effectiveness.

We have conducted measurements in a special smoking porch in the Faculty of Architecture at Delft University of Technology, which can be regarded as ETA 3, and at peak usage even as ETA 4. However, the approach and results of the study can also be applied to other situations.

THE SMOKING PORCH

The smoking porch was initially an outside space in which smokers could shelter from rain and wind, and it is therefore constructed entirely of glass; in this form it was naturally ventilated. The staff then asked if the room could be heated to some degree in the winter months. However, it was important that the smoking room formally retained the character of an unheated room, because it would otherwise have to meet the requirements of the Dutch Building Regulations in terms of thermal construction, facilities, and energy performance. The solution to this was found by ventilating the smoking porch with air extracted from a substation of the central heating system, which in winter has a temperature of approximately 25^0C. By making use of unused residual heat, the smoking porch has no effect as such on the faculty building's energy performance. The overall room temperatures to be achieved are shown in figure 1.

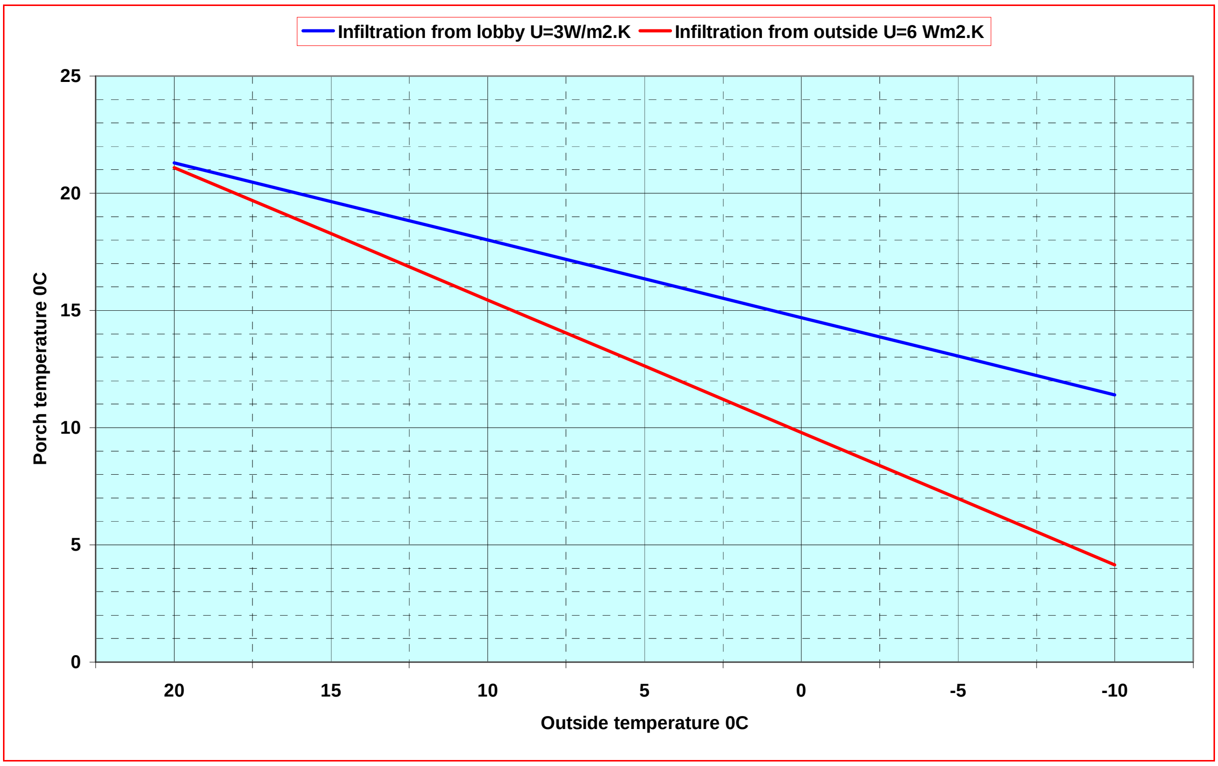

Figure 1 – Room temperature of smoking porch in relation to outside temperature

The floor area of the smoking porch is 29 m^2. The height is 6.85 m and the gross volume is approximately 200 m^3. The porch can accommodate 24 standing smokers at four smoking tables. The wishes of the staff for seating to be included were not met. See figure 2, in which the points of measurement are also shown.

VENTILATION AIRFLOW RATE OF THE SMOKING PORCH

The airflow rate was chiefly determined by the room temperature desired in the porch during the winter months; the temperature of the supply air is a given factor. On the basis of overall calculations, a total airflow rate of 1600 m^3/h has been assumed, equivalent to 8 air changes per hour and 67 m^3/h per person at maximum occupation. Dutch Building Regulations only require a ventilation rate of approximately 500 m3/h for a room such as this.

PRESSURE HIERARCHY AND AIR BALANCE

The smoking room has an east-facing glass outer facade with a surface area of approximately 140 m^2. The infiltration of this facade by a strong east wind cannot be ignored, because it could result in the extraction capacity becoming too small and in the most serious case even create overpressure in the smoking room.

The infiltration flow can be calculated using the formula in [1]:

$$\boxed{V = \sum (a.l) * \sqrt[3]{\Delta p^2}} \qquad (1)$$

Wherein:
V = infiltration flow m^3/h
a = flow coefficient in $m^3/m^1.h.\Delta p^{2/3}$ (For unopenable sections to be set at 0.20)
l = length of cracks in m (Ca 104 m)
Δp = pressure difference across the outer facade in Pa (At 5 Beaufort[1] estimated at ca 50 Pa).

Using formula 1 the maximum infiltration is estimated at ca 300 m^3/h. On this basis the air balance is set as follows:

	Calm	Wind 5 m/s
Exhaust airflow	1600 m^3/h	1600 m^3/h
Infiltration from lobby	300 m^3/h	0 m^3/h
Infiltration from outside	0 m^3/h	300 m^3/h
Supply airflow	1300 m^3/h	1300 m^3/h

The ventilation system is constructed as an upflow system (see below), so in principle it could function as displacement ventilation. For this to work it is important that the airflow is greater than the upward air current of 20 dm3/s (72 m3/h) which is the natural thermal caused by the human body [2]. A supply airflow of 1300 m3/h is therefore sufficient for (1300/72=) 18 persons. At the full capacity of 24 persons, (24*72=) 1.728 m3/h would be needed. However, whether and to what extent the ventilation system is suitable to function as displacement ventilation is discussed below.

THE VENTILATION SYSTEM

The smoking porch is equipped with four round tables mounted on steel tubes Ø 323.9 x 4 mm with perforations Ø 10 mm, centre to centre 60 mm, through which air is supplied. The speed at which air is blown in through the perforations is 5 m/s, aiming at a fast reduction in air temperature. If the temperature of this air were too high it would adversely affect the upflow system. On the other hand, the high speed of the air blown in and the inherent induction of room air counteract the intended displacement effect. Partly due to the cold downdraught along the outer facade of the porch, despite the supply of ventilation air at a low level, hardly any displacement effect has been found to occur (see below).

The air is extracted high up in the porch via steel tubes Ø 193.7 x 4 mm measuring ca 6 m in length, which are installed concentrically in the air supply tubes – see figure 3.

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[1] An east wind at force ≥ 5 Beaufort is rare in the Netherlands.

ANALYSIS OF THE VENTILATION CONCEPT

Separation effectiveness:

The separation effectiveness is defined as

$$\eta_s = \frac{C_i - C_e}{C_i} = 1 - \frac{C_e}{C_i} \qquad (2)$$

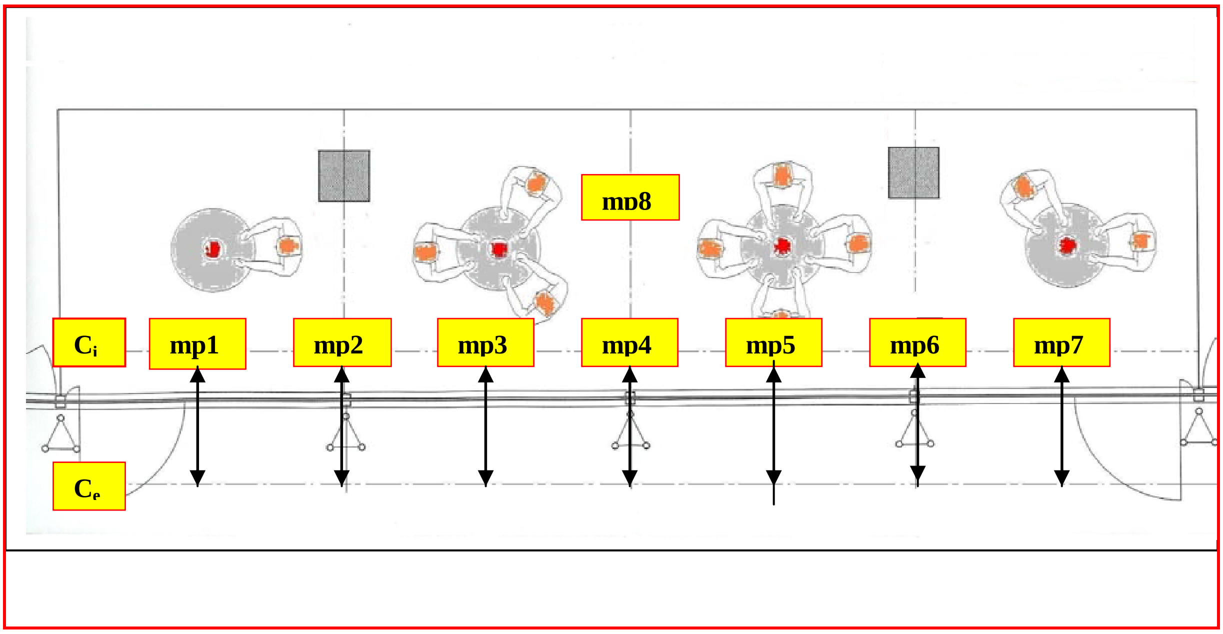
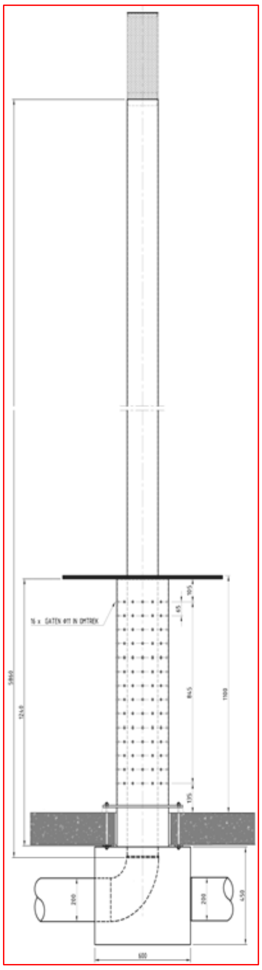


Figure 2 – Floor plan of the smoking porch with measuring points



Wherein

C_i = ETS concentration in the porch

C_e = ETS concentration in the lobby outside the porch

The ETS concentrations in the smoking room are measured at breathing height between 1.1 en 1.8 at respectively $C_{i.1,1}$ en $C_{i.1,8}$

It is not easily possible to develop an analytical model to predict the separation effectiveness. The dissipation of ETS to the lobby can perhaps be modelled, but the ETS concentration in the lobby is dependent to a large extent on the volume of the room, the air movement and the ventilation of the lobby. The separation effectiveness can therefore only be determined on the basis of measurements.

Figure 3.

Ventilation effectiveness:

The ventilation system is set up as an upflow system, but does not meet the conditions for displacement ventilation because:

- The temperature of the supply air is higher than the room temperature. After all, the air supply also serves to warm the porch.
- The doors of the porch are frequently opened and closed for smokers to leave and enter.
- The cold downdraught along the outer facade causes a downward current of contaminated air.

The smoking porch is fitted with hinged glass doors, which when opened and closed 'pump' a quantity of air from the porch into the central lobby and vice versa. Under isothermal conditions this quantity is approximately 672 dm^3 a time [3]. With an average occupation of 15 smokers, each of whom spend 15 minutes in the porch, the doors are opened and closed 120 times an hour (twice per smoker), whereby ($120*672*10^{-3}$=) 80 m^3/h ETS is transferred into the lobby. In relation to the ventilation capacity of 1600 m^3/h this is a comparatively small amount, which in itself need not disturb the upflow of displacement ventilation too seriously.

The cold downdraught along outer facades that are made only of single glazing is more of a problem. The convection flow that arises along a vertical surface can be calculated using the following formula [3]. The formula applies to turbulent flows which in this case will certainly occur.

$$\boxed{q_{v,z} = 2.75\Delta\theta^{0.4} z^{1.2}} \qquad (3)$$

Wherein:

$q_{v,z}$ = vertical airflow -$dm^3/s.m$

$\Delta\theta$ = temperature difference -K

z = height of the vertical surface -m

The outer facade of the porch has a surface area of (H*B= 6.65*16.40) 109 m^2. The temperature difference $\Delta\theta$ between the air in the smoking porch and the glass surface depends on the outside temperature and the wind speed. The lower the temperature outside and the higher the wind speed, the more the temperature in the porch will fall and with it the inner surface temperature of the outer facade. The cold downdraught has been quantified using formula (3); figure 4 shows the results.

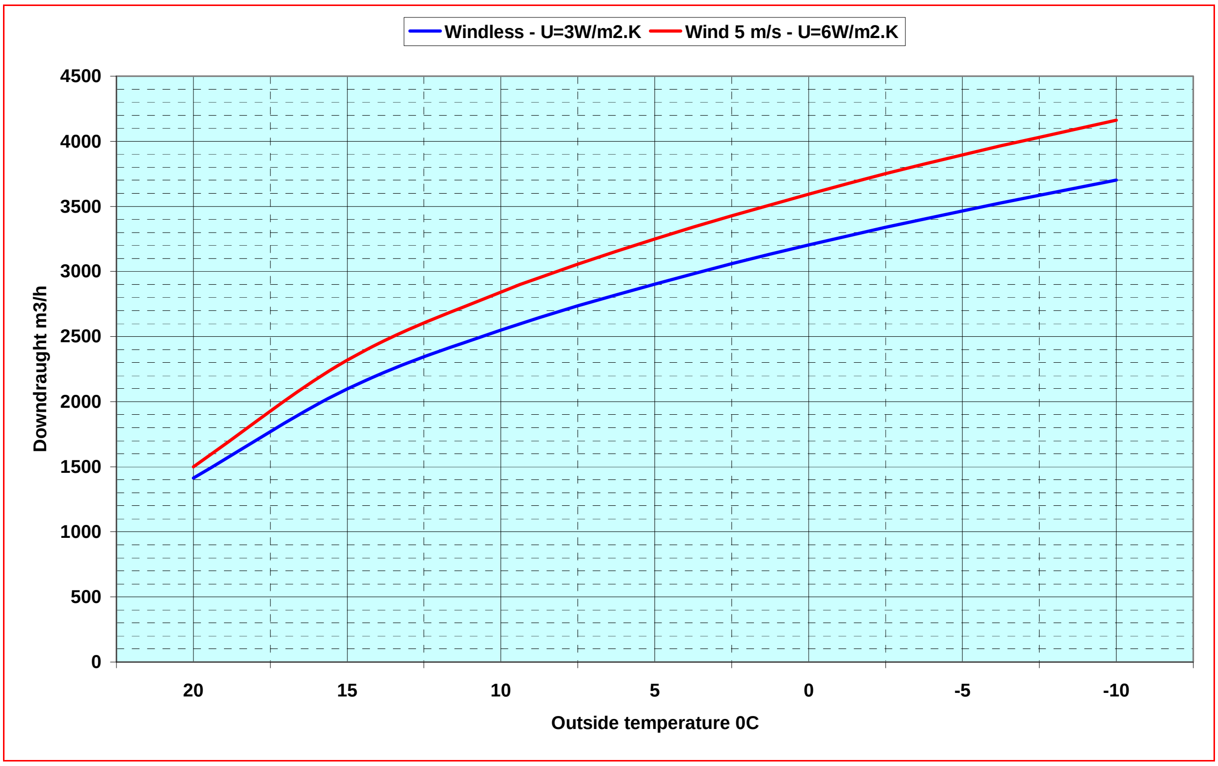


Figure 4 – Downdraught in m^3/h in relation to outside temperature

At an outside temperature of -3^{0}C, the lowest occurring during the day, a downdraught in the order of 3500 m^3/h occurs. But also at a more moderate outside temperature such as 15^{0}C the downdraught is still greater than 2000 m^3/h.

This downdraught is considerably greater than the supply flow, so displacement ventilation hardly occurs. The lower the outside temperature falls, the more the system assumes the character of mixing ventilation. This is not important for separation effectiveness, but it is for the air quality in the porch, as the downward airflow set in motion by the cold downdraught is contaminated with ETS.

DETERMINATION METHOD FOR ETS CONCENTRATION

The determination of ETS concentrations according to the international standards ISO 18144 and ISO 18145 is a time-consuming and expensive matter and moreover it cannot be carried out online. The ETS concentration has therefore been determined with the aid of an Ultra fine Particle (UFP) Sensor [4] developed by Philips Research in Eindhoven. Philips has made a prototype of this innovative sensor available for the measurements to be conducted. They therefore provide no data on nicotine, 3-EP and solanesol concentrations in the ETS, which are also not required by the present measuring programme given that both for separation effectiveness and ventilation effectiveness relative values are concerned.

CONCENTRATION MEASUREMENTS

During three measurement sessions, first the separation effectiveness and then the ventilation effectiveness were determined with the aid of the UFP sensor. The values measured by the sensor are in pA. It should be noted that a considerable inhomogeneity was recorded, both in time and in place. The values measured are ± 10%.

For the measurements of separation effectiveness a mobile table was used with two surfaces at 1.0 and 1.6 m high, on which the measuring pump and analysis equipment was placed. Separation effectiveness was calculated from the values measured using equation (2).

For the measurements of ventilation effectiveness a hand-operated lift table was used. Measurements were taken at heights of 0.1 – 1.1 – 1.8 – 3.0 en 3.5 m; this was the greatest height it was possible to reach. It was therefore not possible to take the desired measurement next to the extraction point. The ventilation effectiveness, expressed in Contaminant Removal Effectiveness CRE [5], has not therefore been calculated on the basis of the concentration in the exhaust C_a, but of the concentration at a height of 3.5 m. The CRE value has been calculated respectively for a sitting height of 1.0 m and a breathing height of 1.6 m using the formula

$$\boxed{CRE = \frac{C_{3.5}}{C_{1.0}} \ and \ \frac{C_{3.5}}{C_{1.6}}} \qquad (4)$$

The measuring points are shown in figure 2. The ventilation flows, temperatures and number of cigarettes burnt are shown in table 1. For each cigarette the doors were briefly opened and closed either one or two times.

Table 1 – Measurement conditions

No.	Supply flow m^3/h	Exhaust flow m^3/h	Room temp. ^0C	Outside temp. ^0C	No. of cigarettes
1	1.310	1.640	20.6	16.9	49
2	720	1.060	20.6	16.4	42
3	215	500	20.6	16.4	41

MEASUREMENT SESSIONS

Session 1A – Separation effectiveness η_s

The values measured in pA are shown in table 2:

Table 2 – Values measured in measurement session 1A

concentr.	height	mp1	mp2	mp3	mp4	mp5	mp6	mp7	gem.
C_i	1.0 m	1.00	0.95	0.65	0.70	0.70	0.51	0.36	0.70
C_i	1.6 m	1.10	1.50	1.10	1.50	1.35	1.20	1.72	1.35
C_i	av.	1.05	1.23	0.88	1.10	1.03	0.86	1.04	**1.03**
C_e	1.0 m	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04
C_e	1.6 m	0.03	0.03	0.03	0.03	0.03	0.03	0.03	0.03
C_e	av.	0.035	0.035	0.035	0.035	0.035	0.035	0.035	**0.035**

Separation effectiveness $\boxed{\eta_s = \frac{C_i - C_e}{C_i} = 1 - \frac{C_e}{C_i} = 1 - \frac{0.035}{1.03} = 0.97}$

Session 1B – Ventilation effectiveness CRE:

The values measured in pA at mp 8, and the calculated CRE values $CRE_{1.0}$ *and* $CRE_{1.6}$ are shown in table 3.

Table 3 – Session 1B measurements and results for Contaminant Removal Efficiency CRE

height m	concentration at mp 8	
0.1	1.30	$CRE_{1.0} = \frac{C_{3.5}}{C_{10}} = \frac{1.4}{1.1} = 1.7$
1.0	1.10	
1.6	1.35	$CRE_{1.6} = \frac{C_{3.5}}{C_{1.6}} = \frac{1.4}{1.35} = 1.04$
3.0	1.35	
3.5	1.40	

Session 2A – Separation effectiveness η_s

The values measured in pA are shown in table 4:

Table 4 – Values measured in measurement session 2A

concentr.	height	mp1	mp2	mp3	mp4	mp5	mp6	mp7	av.
C_i	1.0 m	1.38	1.40	1.23	1.35	1.32	1.31	1.28	1.32
C_i	1.6 m	1.74	1.60	1.30	1.65	1.55	1.65	1.75	1.60
C_i	av.	1.56	1.35	1.27	1.50	1.44	1.48	1.52	**1.46**
C_e	1.0 m	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05
C_e	1.6 m	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05
C_e	av.	0.05	0.05	0.05	0.05	0.05	0.05	0.05	**0.05**

Separation effectiveness, calculated using the average values for C_i en C_e from table 2, is calculated as follows: $\boxed{\eta_s = \frac{C_i - C_e}{C_i} = 1 - \frac{C_e}{C_i} = 1 - \frac{0.05}{1.46} = 0.97}$

Session 2B – Ventilation effectiveness CRE

The values measured in pA at mp 8, and the calculated *CRE* values *$CRE_{1.0}$ en $CRE_{1.6}$* are shown in table 5.

Table 5 – Session 2B measurements and results for Contaminant Removal Efficiency CRE

height m	concentration at mp 8	
0.1	1.20	$CRE_{1.0} = \frac{C_{3.5}}{C_{1.0}} = \frac{2.00}{1.45} = 1.38$
1.0	1.45	
1.6	1.50	$CRE_{1.6} = \frac{C_{3.5}}{C_{1.6}} = \frac{2.00}{1.50} = 1.33$
3.0	1.65	
3.5	2.00	

Session 3A – Separation effectiveness η_s

The values measured in pA are shown in table 5:

Table 6 – Values measured in measurement session 3A

concentr.	height	mp1	Mp2	mp3	mp4	mp5	mp6	mp7	av.
C_i	1.0 m	1.95	1.80	1.80	2.20	2.10	2.05	2.25	2.02
C_i	1.6 m	2.38	2.28	2.13	2.05	2.03	2.18	2.25	2.19
C_i	av.	2.17	2.04	1.97	2.13	2.07	2.12	2.25	**2.10**
C_e	1.0 m	0.06	0.08	0.08	0.08	0.08	0.08	0.08	0.077
C_e	1.6 m	0.06	0.07	0.07	0.07	0.07	0.08	0.08	0.071
C_e	av.	0.06	0.075	0.075	0.075	0.075	0.08	0.08	**0.074**

Separation effectiveness, calculated using the average values for C_i en C_e from table 2, is calculated as follows: $\boxed{\eta_s = \frac{C_i - C_e}{C_i} = 1 - \frac{C_e}{C_i} = 1 - \frac{0.074}{2.10} = 0.965}$

Session 3B – Ventilation effectiveness CRE

The values measured in pA at measurement point 8, and the calculated CRE values CRE1.0 en CRE1.6 are shown in table 6.

Table 7 – Session 3b measurements and results for Contaminant Removal Efficiency CRE

height m	concentration at mp 8	
0.1	2.50	$CRE_{1.0} = \frac{C_{3.5}}{C_{1.0}} = \frac{3.00}{2.50} = 1.20$
1.0	2.50	
1.6	2.56	$CRE_{1.6} = \frac{C_{3.5}}{C_{1.6}} = \frac{3.00}{2.56} = 1.17$
3.0	3.00	
3.5	3.00	

DISCUSSION

The concentrations measured in pA using the UFP sensor in the smoking porch are thrown into relief when they are compared with typical values in reasonably clean outside air of 0.02 pA. When it is warm outside, with moderate traffic the sensor reading rises to 0.05 – 0.06 pA. In a car on the motorway peak concentrations occur in the range of 0.2 – 0.5 pA. The air quality in the porch can therefore be regarded as being relatively poor, even at higher ventilation capacities. ETS is an infamous source of air contamination in indoor environments!

The concentration in the porch decreases sharply with increasing ventilation capacity; see figure 5. Extrapolating logarithmically, the concentration at 3.000 m3/h would come out at 0.5 pA, which would undoubtedly reduce the ETS absorbance in the clothing of visitors to the porch and hence the distribution of ETS in building. The question is whether it would also improve the air quality measurements. ETS is a complex cocktail with thousands of components. In most cases ETS concentration is below the chemical/physical detection limit. [6]

For the rest, the clear relationship between ETS concentration and ventilation capacity provides an excellent possibility for a demand controlled ventilation system. The signal from the UFP sensor can be included in the control circuit of an air quality controller. Having made enquiries within the industry, the author is not aware of any other sensors with which this possible.

The potential field of application of the UFP sensor is not limited to smoking rooms. This sensor for ventilation control can be applied anywhere where ultra fine particles ($PM_{2.5}$) form a risk. For future applications a multifunctional room sensor for temperature, relative humidity, CO_2 and $PM_{2.5}$ is also being considered.

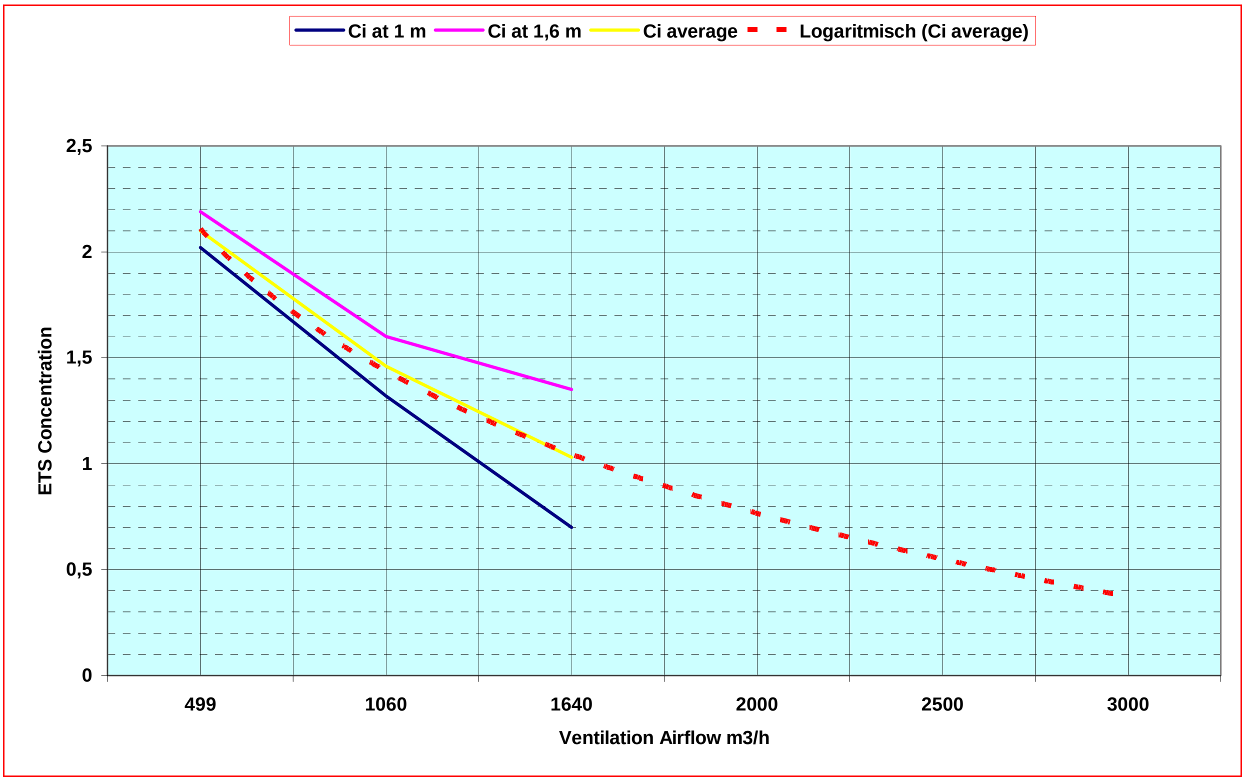


Figure 5 –ETS Concentration in smoking porch in relation to ventilation capacity

CONCLUSIONS

- At a value of 0.97 the separation effectiveness is extremely high, also at low ventilation capacities.
- The ventilation effectiveness is low. This can be attributed to the cold down draught along the porch outer facade, and to the ventilation system as such.
- To realise a high separation effectiveness, a difference between supply and exhaust flow of -1.9 $dm^3/s.m^2$ is sufficient.
- To improve the IAQ in the smoking porch the cold down draught should be decreased, combined with an increase in ventilation flow.

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Ventilation Design in Large Enclosures for Sports Events using CFD: the Halls of the "Città dello Sport" in Rome.

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SUMMARY

Buildings with large enclosure often present unresolved problems related to energy and air flows such as unwanted thermal stratification, local overheating, uncontrolled contaminant spreading. These kind of constructions are often found in unique buildings where innovative designs are developed. Consequently, there exists no previous experience and very careful analysis on the ventilation design is advisable [1]. This is particularly true for the halls of the "Città dello Sport" in Rome, designed by Santiago Calatrava, because of their singularity in term of dimension and shape. The CFD simulations here presented are playing a significant role in the on to go design of these unique large enclosure buildings for the optimisation of the HVAC strategies.
The simulated heat, cooling and ventilation system defines three different climatic zones in order to minimize the energy consumption:

- the swimming pool plane and the playground where optimal condition for the athletes have to be assumed;
- the stands where the people watching the event is placed;
- the huge volume under the roofing where particular climate conditions aren't required.

The simulated system provides a particular ventilation system using a controlled airflow to heat and cool the mass of the stands, which afterwards operates directly in the zone occupied by the spectators.
The swimming pool level and the basketball playground are heated, cooled and ventilated by a dedicated HVAC system.
Air velocity, temperature and relative humidity fields in the indoor space of the building, in condition of maximum external and internal thermal loads have been obtained from simulations, in order to evaluate suggested value for the designing process of the HVAC system.

INTRODUCTION

Large enclosures have become a major feature of modern building design. Spaces such as atria and covered areas are used in all varieties of buildings including office complexes, shopping malls, airports and sport halls. Essentially, they create an environment protected from the outdoor climate in which a wide range of activities is possible. However such spaces demand very careful design to ensure good indoor air quality and thermal comfort.
Common problems in poorly designed buildings can include oversizing of equipment, excessive energy requirements and poor indoor environment. These problems are usually caused by a lack of knowledge and guidance at the design stage.
Ventilation of large enclosures deserves careful attention by designer for the following reasons [1]:

- usually only a small portion of the entire volume is occupied, therefore energy efficiency may be achieved by special strategies aimed at directing ventilation air and thermal conditioning to the occupied zone;
- vertical air streams driven by temperature differences gather large momentum in large spaces: resultant cold down-drafts from vertical surfaces may have severe comfort implication;
- large enclosures are often found in unique buildings where no previous experience exists.

The “Città dello Sport” will be realized in Rome, located within the University Campus of “Tor Vergata”, to host the World Swimming Championships in 2009 and Volley Championship in 2010. The complex could be also presented as a main infrastructure for the Olympic Village for Rome Games candidature in 2016.
The complex, designed by the architect Santiago Calatrava, includes a large covered building for indoor competitions, composed from two equivalent, but functionally independent, pavilions: one with a polyvalent function (basket, volley) and the other as an Aquatics Stadium for swimming [2]. Volumes and shapes of Pavilions are similar Of but the presence of different occupancy both with the installation of pools on the case of swimming pools need two different HVAC system strategies.

Occupant comfort in these types of environments is affected by the air temperature and humidity, the air movement and solar radiation. The air temperature in the large space is mainly dependent on ambient conditions but is also affected, during summer, by direct solar radiation warming building elements including the roof. The extent of natural ventilation also affects the air temperatures, as greater air exchange will reduce the effect of localised heat sources.
The design has to include specific design issues which have been addressed as multidisciplinary team programs with close liaisons between engineer and architect, often utilising state of the art techniques to resolve them. The key design issues that need to be solved by integrated design solutions are:

- Materials selection
- Air flow analysis
- Energy efficiency
- Acoustic strategy
- Lighting

It is increasingly becoming common to use Computational Fluid Dynamics (CFD) techniques as a cost effective and innovative method to assist in the engineering design process to achieve the desired outcomes [2]. By using CFD at an early stage of the engineering design process, a large number of investigations related to spectator comfort aspects by the natural or mechanical (HVAC) ventilation of the indoor environment in sport complexes can be performed.

A preliminary analysis of the air ventilation in the two large spaces has been performed using a commercial CFD program.
The simulated heat, cooling and ventilation system defines different climatic zones in order to minimize the energy consumption:

- the swimming pool plane and the playground where optimal condition for the athletes have to be assumed;
- the stands where the people watching the event is placed;

- the huge volume under the roofing where particular climate conditions aren't required.

BUILDING DESCRIPTION

The complex will extend on an area of approximately 50 hectares within the university campus of Tor Vergata, Rome and includes, among other systems, the realization of a great covered structure for the indoor sports (Fig. 1). The Campus is organized in order to accommodate great events. The main building is realized with two equivalent pavilions (Fig. 2), located symmetrically and characterized by the same shape. Inside of a pavilion is the playground and in the other is localized the covered swimming pools. The two structures have independent function and they can offer different events at the same time.

Fig. 1 – "Città dello Sport", Rome – General view and main building [1]

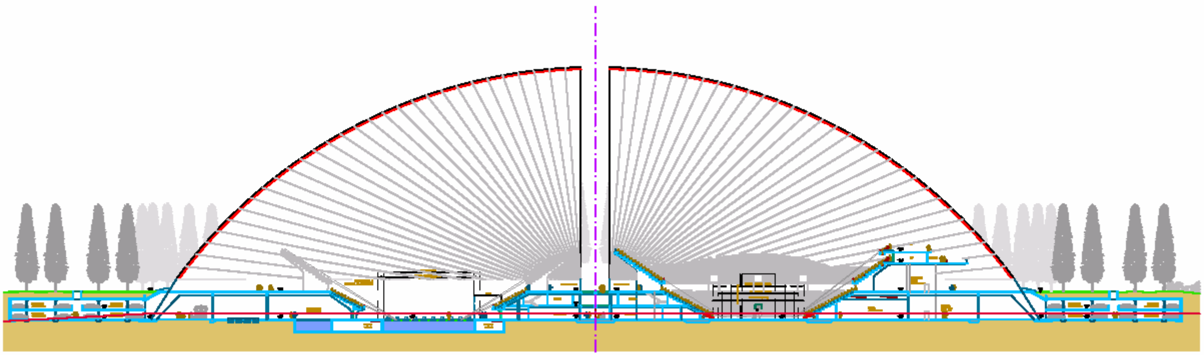


Fig. 2 – Section of the Swimming Hall and the Palasport [1]

The structure of the cover is quite light, based on steel with opaque and transparent panels (the latter being 30% of the total surface). 4,000 places are located in the Swimming Hall and 15,000 spectators are foreseen in Palasport. The structure has a maximum height of about 70 m and the main axes of the two bases are 130 x 90 m, for a volume greater than 500.000 m^3 for each pavilion. The game field of the Palasport is 30 x 45m. Two pools are placed in the Swimming Hall (53 x 25m and 25 x 25m for the dives and a swimming pool 50 x 10m for training. All around the main building are located areas of service, arenas, laboratories, didactic classrooms, structures, stores, offices, warehouses. The structures are completed and connected with a great park, polyvalent fields and athletics tracks.

METHODOLOGY

To analyze the air flow inside the two structures a commercial CFD solver based on the finite volume technique has been used.
CFD simulation of the flow field in large enclosures presents many difficulties due to its large size, the complexity of the flow field and of the geometries. Furthermore, a high number of cells is needed to represent the flow. This adds to the complexity of simulating phenomena such as turbulence and increases the risk of numerical errors. In large spaces, airflow tends to be dominated by buoyancy forces. These are stimulated by high degree of coupling with the external volume environment, relevant to represent accurately surface heat transfer processes such as natural/mixed convection, solar radiation, long wave thermal radiation, heat transfer through external walls and structural heat storage.
Accurate prediction of such a flow field is very difficult and, the popular k-ε turbulence models have to be carefully controlled step by step during the calculation to achieve reasonable results.
In the present calculations the fluid (air-water mixture) has been considered as an "incompressible" perfect gas: the pressure does not influence the density of the fluid, while the effect of the temperature is properly considered.
The geometric models of the buildings includes almost 150,000 nodes, approximately 800,000 cells and 200 faces for boundary conditions for the Swimming Arena, and about 220.000 nodes with 1,000,000 cells for the basketball hall. The nodalization is of triangular type for faces and tetrahedral for the cells. The calculation volume is, as already said, approximately 554,000 m^3 for each hall, the envelope area in the model is 25,000 m^2, 8,000 m^2 of which transparent glass and 17,000 m^2 as opaque surface (Fig. 3).

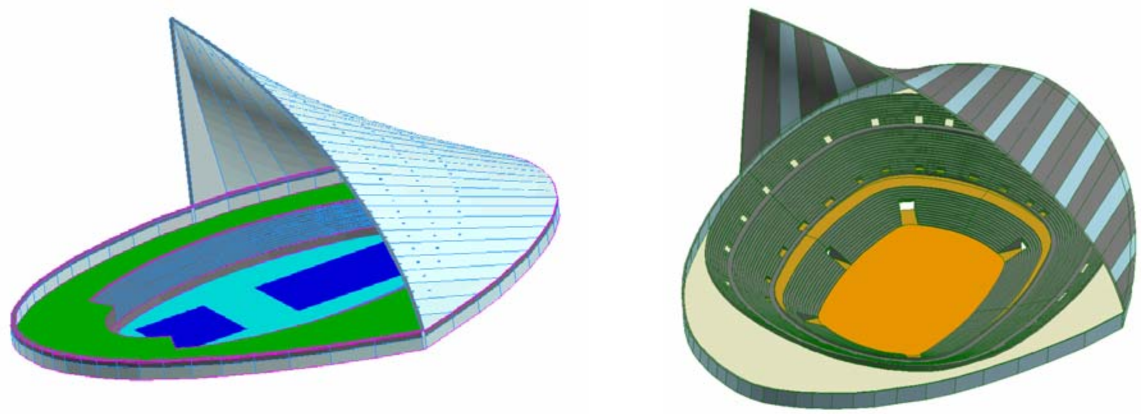


Fig. 3 - Swimming Hall (left) and the Palasport (right) CFD models.

The lateral external walls area outlined under level 8 m are of approximately 4,200 m^2. Inner surfaces thermally connected to temperature controlled environment have been also considered.
The cover of the buildings has been characterized by an equivalent thermal conductivity to take into account the different thermal resistances (opaque and transparent panels).
Also spectators have been considered as sensible and latent heat sources.
Risk of condensation on the internal part of the cover is also evaluated in order to prevent dangerous corrosion formations on the steel structures.

HVAC System: in the Swimming Hall two mechanical systems are present: 88 high velocity nozzles for the pool bowl, located under the lower row of seats, 2.5 m above the pool level. This system has a total air flow capacity of 132,000 m^3/h. A second system allow to introduce air at controlled temperature and humidity along the perimeter of the building, above the spectators level; it is mainly devoted to control the humidity in the large space to avoid condensation on the cover inner surface. The total air flow capacity of this system is 128,000 m^3/h.
The humid air at the higher levels pass through gratings strategically located in the upper part of the cover, and is partially recirculated by return channels to reduce energy load.
The basketball hall is equipped with four different dedicated HVAC systems: five inlet groups of four nozzles each supplying an overall flow rate of 30,000 m^3/h for the playground.
A plenum under the seats of the lower order is installed to control the environment conditions for the spectators providing a total flow rate of 96.000 m^3/h at low velocity. A second plenum with a flow rate of 104,000 m^3/h is installed under the three last rows of the medium order of seats. Peripheral injections (64 units) with a total flow rate of 120.000 m^3/h are placed along the base of the roof in order to heat the "ring" around the arena (Fig. 4).
In the slab surrounding the tribunes a radiant heating floor is installed to increase the mean air temperature in the enclosure of the building.
Windows can be opened along the main arc of the building envelope from the highest point, each with an area of about 6 m^2.

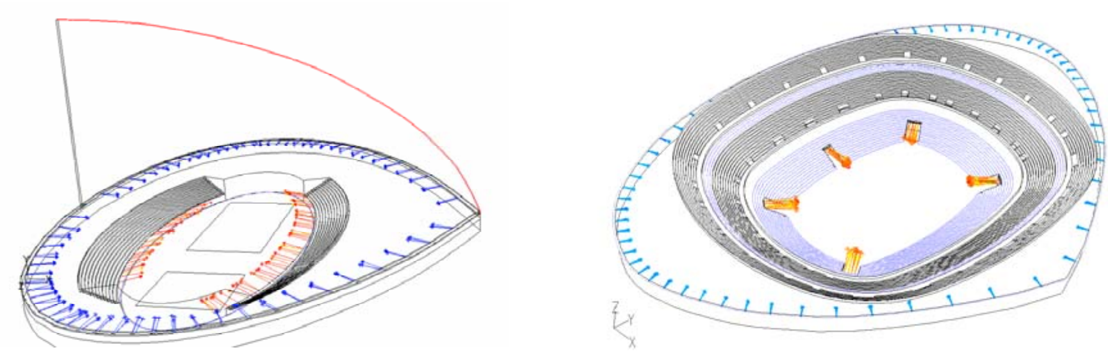


Fig. 4 - Swimming Center (left) and Palasport (right) HVAC systems.

The analyses presented concern the air flow simulation in winter conditions and in two different configuration: 1) natural ventilation in the closed space, to evaluate the air flow due only to the buoyancy driven by the heat sources in the lower zone and the cold envelope surface; 2) HVAC system in operation, to evaluate the effects of the comfort conditions for spectators.

RESULTS

SWIMMING POOL HALL - To analyze the air distribution, a CFD analysis of the space was conducted establishing air flows, temperature and relative humidity.
In addition to the normal concerns of adequate temperature comfort control the ventilation system design within an indoor swimming pool complex also needs to address the issue of maintaining sufficient fresh air ventilation within the occupied zones of the pool hall. This is in order to provide the necessary removal of water vapour (for relative humidity control) and chemical vapours (eg, chlorine) that are evaporated from the pool surfaces [1].

In the winter season, a first calculation was performed considering the enclosure with HVAC system not operating. The cold surfaces of the envelope and the heat source at pool and public levels generate a main vortex above the deck level behind the left public stands (Fig. 5a). Above the pool, hot air is buoyancy driven upwards and towards public stands, with velocities generally lower than 0.3 m/s with some spots up to 0.5 m/s (Fig. 5b) . Temperatures in the spectators zone are around 15 °C (Fig. 5c) at 50 cm height.

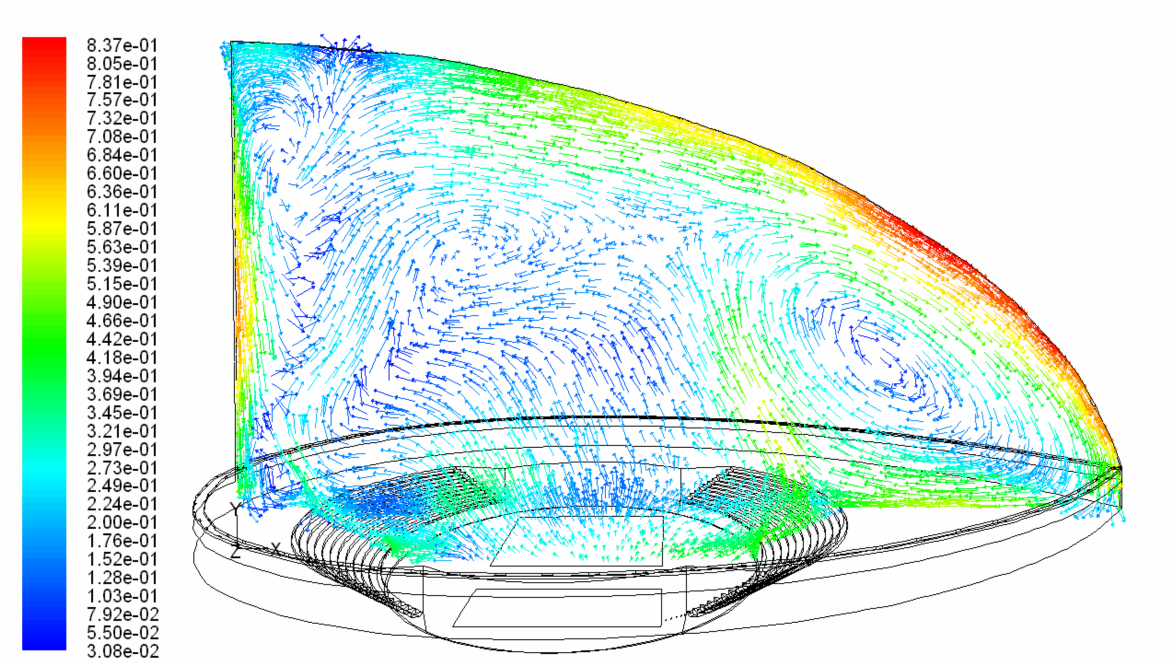
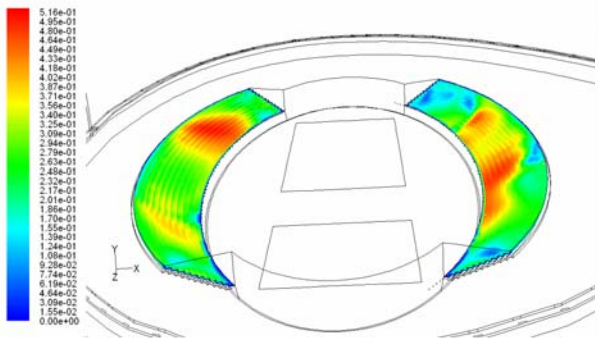


Fig. 5a – Swimming Center: Winter and HVAC off – Air flow in the symmetry plane



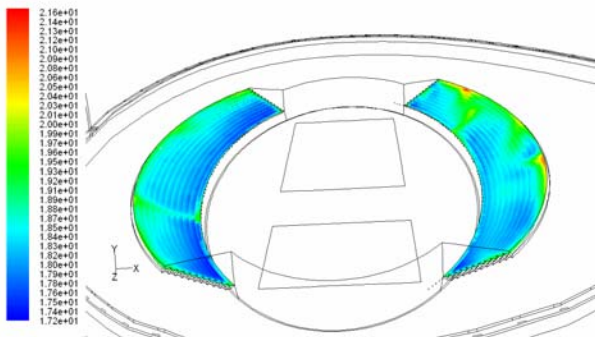


Fig. 5b – Velocity map at public stands Fig. 5c – Temperature map at public stands

The effect of HVAC system is to enhance comfort for spectators. The high speed nozzles in the pool region are foreseen to separate the pool bowl from the huge higher volume. Return air should be drawn from the pool level grating surrounding the pool to draw off air at its most contaminated point in terms of odours, particularly chloramines and particulate matter. Flow inlet velocity and nozzle orientation need to be adjusted because of direction of air velocities coming from the cover, which is forced towards the right public stands. It could be convenient to verify the elimination of inlet nozzles placed on left tribune and/or their substitution with outlet ones. (Fig. 6a). Air velocities in the spectators zone are now higher than in the previous calculations (> 0.4 m/s, Fig. 6b), and temperatures are much more higher (around 29°C) (Fig. 6c).

Air from the perimeter of the building, above the public level, helps to control humidity below 60% to avoid the risk of condensation on the inner surface of the cover during the winter season.

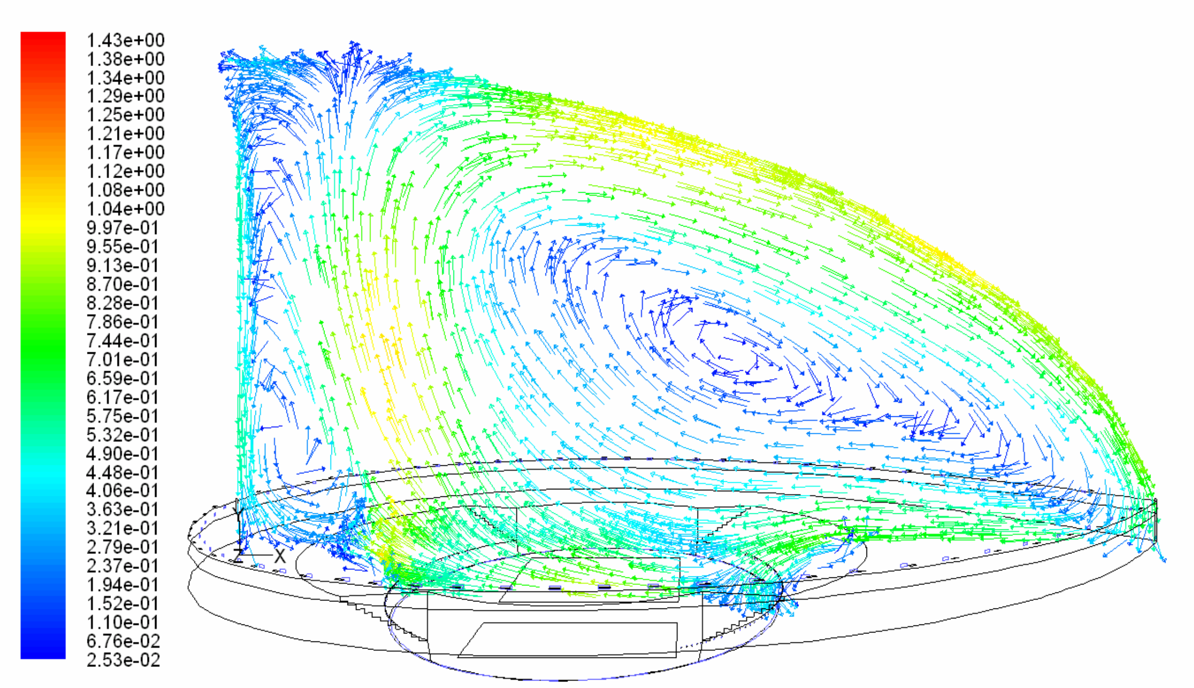


Fig. 6a – Swimming Center: Winter and HVAC on – Air flow in the symmetry plane

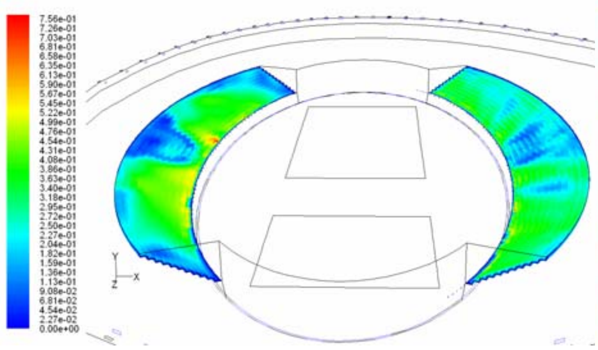


Fig. 6b – Velocity map at public stands

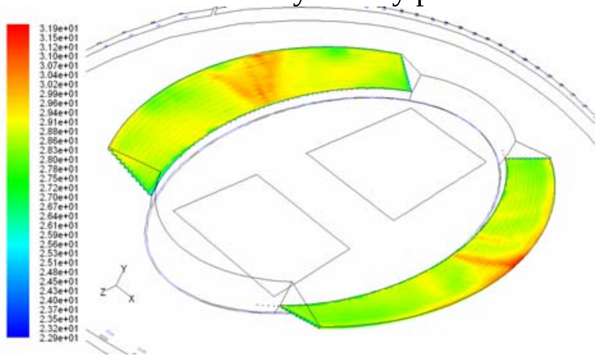


Fig. 6c – Temperature map at public stands

BASKET AND VOLLEY HALL: The matter of acceptable air quality and comfortable environmental conditions in this building where a maximum capability of 15,000 spectators is foreseen, is the major concern. Proper ventilation and supply of fresh air play a significant role in the control of indoor air quality and thermal comfort for the intense metabolism due to the overcrowding of people. Therefore, this study investigates numerically the environmental conditions prevailing in the basketball hall in winter, with full spectators affluence in order to optimize the HVAC system evaluating also the risk of condensation on the inner surface of the cover.

The first simulation presented considers the HVAC disabled to evaluate the flow regime established by the buoyancy forces as in the swimming hall. A boundary layer of cold air descending along the vertical wall and a warm air column upward in the centre of the playground generate a main vortex in the large enclosure of the hall. The air downward the cold wall separates in two different flow on top of right tribunes: one investing the spectators at 0.85 m/s first row of the seats and the other reaching the floor level at 8.00 m generating a zone with positive pressure under the top tribunes. This generates an inflow of air trough the entrances of the top level which combines with the other descending from the upper levels.

The air velocity measured at 1 m over the tribune varies between 0 and 0.85 m/s with a temperature around 13°C. This fluid dynamic regime established spontaneously separates the volume of the enclosure in two zones: the big hall where spectators and athletes stands, and the ring zone which surrounds the tribunes.

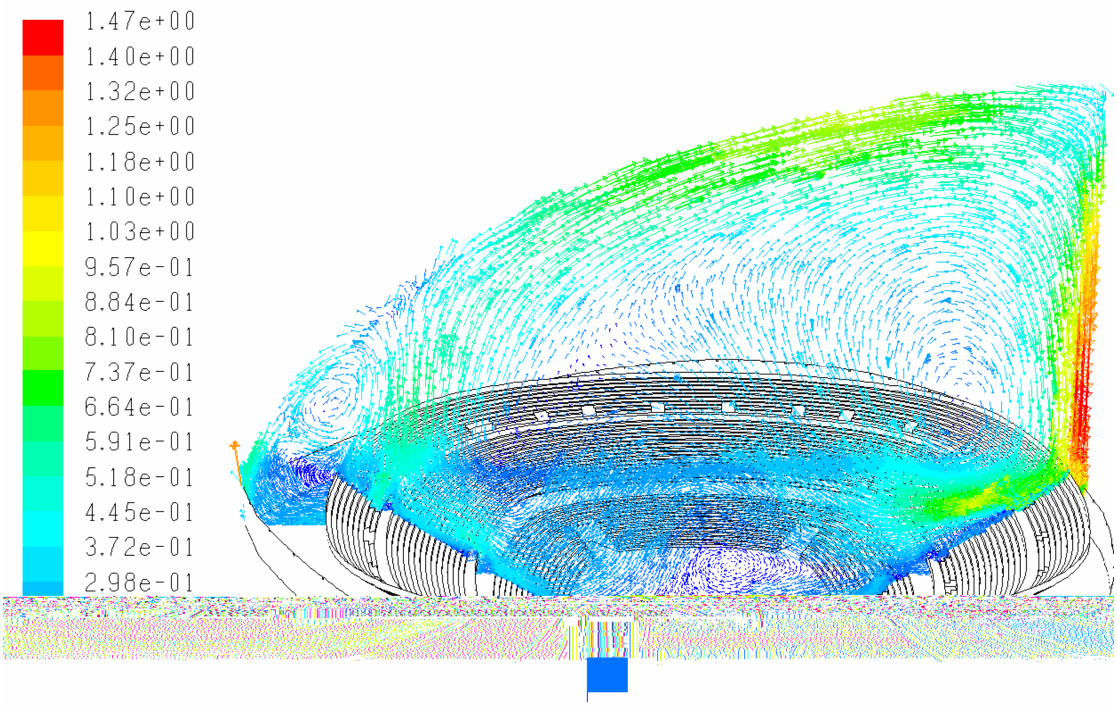
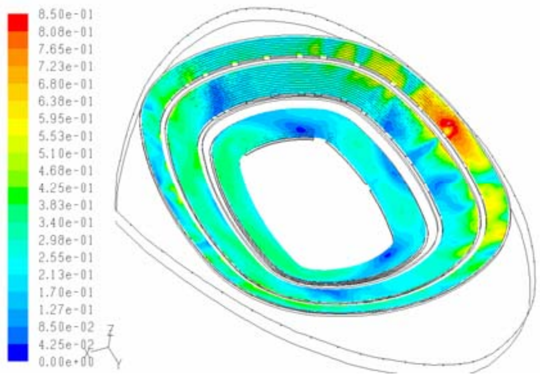


Fig. 7a – Palasport: Winter and HVAC off – Air flow in the symmetry plane



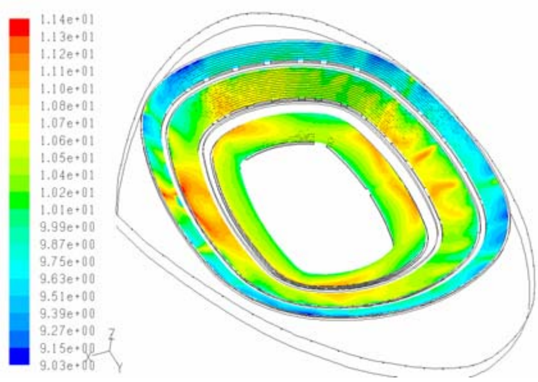


Fig. 7b – Velocity map at public stands   Fig. 7c – Temperature map at public stands

These results show that a HVAC system is needed to obtain proper comfort conditions for people standings on the tribunes in term of temperature.
The activation of the mechanical ventilation system increases the fluid dynamic regime established by the buoyancy forces. In general, the air velocity values are higher ranging from 0.5 m/s to 1.5m/s. Also temperatures increase up to 15°C in the upper order of the tribune and 22°C in the parterre seats. Optimal values of temperature and humidity near people is assured by the plenum inlets under the seats. Part of this controlled flow rate is exhausted by the outlet at the top of the parterre tribune and under the stairs of the upper order of the seats. The air injected by inlet units at the base of the roof generates a turbulent motion which mixing and heating the air coming from the cold boundary layer of the cover in the zone under the

tribunes. The main vortex in the hall sweeps the zone of spectators assuring a good renovation of the exhaust air. The behaviour of the nozzles placed at the top of the vomitoriums depends on the prevalent flow regime. Indeed the units placed at the vertex entrances sweep efficiently the playground, differently by the injections on the symmetry access which reach immediately rises higher zones over the field.

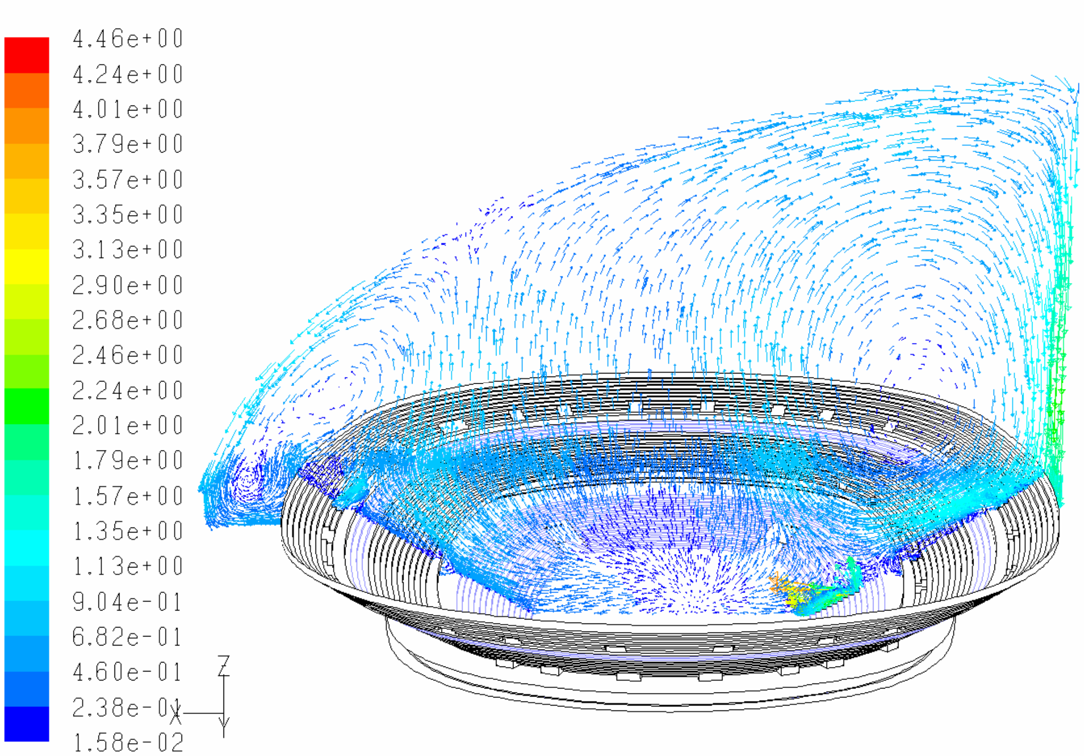


Fig. 8a – Palasport: Winter and HVAC on – Air flow in the symmetry plane

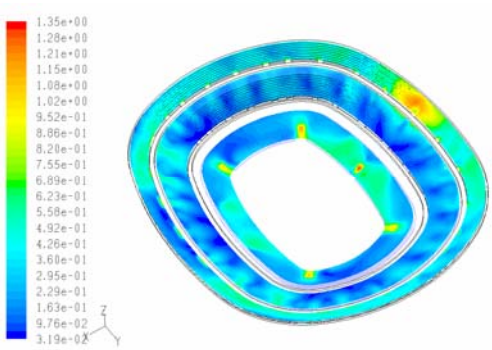


Fig. 8b – Velocity map at public stands

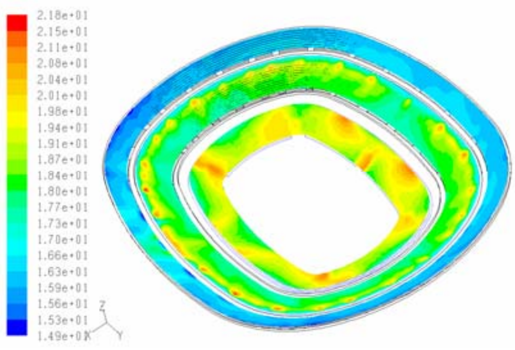


Fig. 8c – Temperature map at public stands

DISCUSSION

SWIMMING HALL: During the winter season, humidity control within the large space and envelope could be assured by the HVAC system. Also public and athletes comfort can be well controlled, with minor adjustment in flow rates and orientation of nozzles.
Summer conditions simulation is actually in progress. During summer, the solar radiation and the heat gains for public and lighting makes the design of HVAC system more challenging. In

fact, the main 50 m competition pool is kept at 26-29°C. During a major event pool water temperatures in the competition pool are likely to around 28°C. To ensure comfort for public and athletes and in consideration of energy use, the air temperature is generally kept 1°C above the water temperature and the consequent environmental control conditions are around 30°C, 60% relative humidity. The spectators are seated up to 3-5m above the pool and the consequent heat stratification could increase the temperature above 30°C, making conditions relatively uncomfortable. On the other hand, the air flow could be better controlled by the mechanical system, in absence of the drop of cold air from the cover towards the public and the pool zone.

BASKET AND VOLLEY HALL: as expected the vertical air streams driven by temperature differences gather large momentum in the hall, prevailing on the flow path imposed by the inlet units of the mechanical system. Therefore flow rates, temperatures and humidity of the air injected in the enclosure have to be optimized in relation to the existing buoyancy induced flow regime, in order to obtain a local climatic control in the zones occupied by persons. The requested climatic conditions for the athletes in playground are assured by the air injected by the high velocity nozzles at the corner of the field. Their performance is strongly influenced by the injection temperature which governs the trend of air path. The value of 22°C assumed in the simulation, has to be considered carefully for the injections over the principal entrance of the parterre in the installing phase.

Non uniform comfort condition are established in the tribunes: optimal conditions are obtained for the two lower order of seats due to the plenum inlet units and acceptable values of temperature and air velocity in the upper levels are reached naturally only by the mixed air in the environment.

Summer conditions simulations, critical for the high value of the external and internal thermal load, are in progress in order to optimize the cooling system.

Middle season and different affluence of public conditions are also object of study to find alternative low energy consumption HVAC strategies.

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Analytical Prediction of Carbon Dioxide Concentrations in Variable Air Volume Systems

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SUMMARY

A typical commercial/institutional building is modeled using a computer program developed by ASHRAE as research project number RP-590. The RP-590 program output was used as input in a separate program written to calculate the CO_2 concentration levels in the various zones of this typical building on an hourly basis for a one year cycle. This data allow one to see how VAV A/C systems affect the CO_2 levels in each zone as the VAV boxes modulate in response to the thermal load. The CO_2 concentration has been suggested to be a good indicator of indoor air quality (IAQ) in typical commercial and institutional settings. IAQ is a hot legal liability issue in the building industry.

The results show that a VAV A/C system, that is adequate in all other respects, can adversely affect the indoor air quality of the building. Factors responsible for this are discussed and recommendations and limitations are given for the utilization of this work.

INTRODUCTION

After the oil embargo, energy conservation became a key issue as companies faced rising operating costs due to the increasing energy cost. One quick way to slow down this rise in operating cost was to reduce the amount of outside air delivered to the occupied zones. Also, architects and designers began to apply Variable Air Volume (VAV) systems more and more in place of traditional constant volume systems. This allowed the total system airflow to be modulated relative to the thermal load demands thereby saving fan energy, duct sizing, etc. Buildings began to be designed and constructed more tightly reducing the amount of infiltration of outside air. These things together contributed to the increase in cases of sick buildings and building related illnesses. These problems had not previously manifested themselves, due to the generous use of outside air and relative “looseness” of building construction. There has been much information published concerning the types and sources of contaminants commonly found in buildings with poor indoor air quality. The ASHRAE current standard for ventilation is ASHRAE Standard 62.1-2004 [1]. This standard specifies the required ventilation rate for many types of buildings from 8 L/(s-person) to 25 L/(s-person). It retained many features from earlier standards including allowing two different procedures for ventilation design. The first procedure, “The Ventilation Rate Procedure” specifies minimum amounts of quality outside air for particular occupant densities. The second procedure, “The Indoor Air Quality Procedure”, allows innovative solutions by other means, such as special filtration, to obtain acceptable indoor air quality. Carbon dioxide levels are recommended for use as an indicator to determine the quality of the air and a limit of 1000 ppm is established.

Objective

The objective of this research is to determine the effect that a Variable Air Volume (VAV) air-conditioning system has on the indoor air quality of the building or zones which it serves. Specifically, this analysis is concerned with the variation with time of the CO_2 concentration levels as the VAV system modulates relative to the thermal load.

Scope

The objective is accomplished by modeling a prototypical building that was developed for use in ASHRAE research project RP-590. The occupant density is 30 persons/100 m^2 which is typical of the several commercial/institutional examples found in ASHRAE Standard 62.1-2004. This model is analyzed for two ventilation design criteria: (1) 3L/(s-person)/person outside air, and (2) 10L/(s-person) outside air. Design criterion (1) was used in the 1970's and 1980's when energy conservation was a priority and indoor air quality (IAQ) issues had not yet surfaced. Criterion (2) comes from the most recent standards issued by ASHRAE concerning acceptable indoor air quality.

ASHRAE Research Project 590, *Control of Outside Air and Building Pressurization in VAV Systems*, developed a personal computer program, in Turbo C™, that models seven basic types of VAV control systems in a prototypical four zone building. The results available from this program include pressures throughout the system/building, zone airflow rates, cooling/heating energy requirements, and certain warnings when pressures or air flow rates are outside prescribed limits. In this paper an algorithm is developed to calculate carbon dioxide levels in each zone based on (1) the amount of outside air delivered to each zone on an hourly basis as calculated by the modified RP-590 program, and (2) the amount of CO_2 generated in each zone during the same time periods.

Description of the Building

The building parameters used in this paper are such that they provide a convenient reference point for comparison with the facilities listed in ASHRAE Standard 62.1-2004, *Outdoor Air Requirements for Ventilation*. The building (see Figure 1) is single story, of medium mass, has an occupancy of 30 persons/100 m^2, and the occupants are the only source of carbon dioxide. This prototypical building, as developed in RP-590, is occupied 24 hrs/day. Each zone has identical thermal loads except for the external loads due to orientation. The interior zone is not connected to this system.

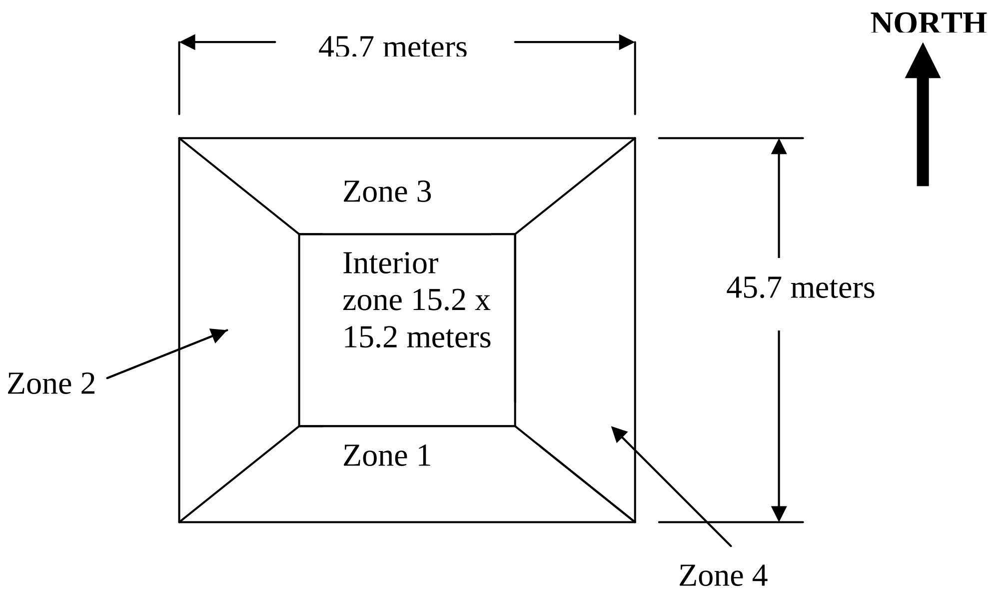

Figure 1. Floor Plan of the Building Used in the Analysis

Description of the VAV System

Of the seven types of VAV systems modeled in RP-590, system 2 is the system utilized in this research (see Figure 2). It has a return fan whose capacity is controlled by a signal from the return duct pressure. The supply fan capacity is controlled by a signal from the supply duct pressure.

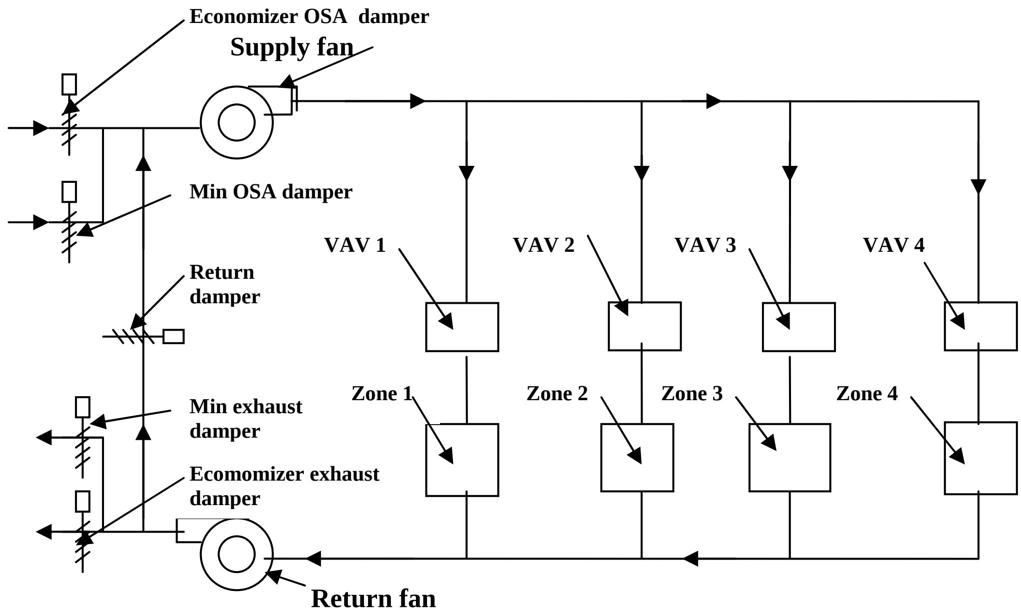

Figure 2. Schematic of VAV system Analyzed in this Work

Under the 3 L/(s-person) design criteria, the total amount of outside air is 1500 L/s (+0, -500). Under the 10 L/(s-person) design criteria, the total amount of outside air is 6,000 L/s. The desired zone pressure is 0.004 cm of H_20 (±0.004) in all cases. Economizer operation is controlled by the mixed air temperature, which has a setpoint of 23 C. The total air flow to a zone is modulated between a specified minimum turn-down ratio and wide open flow. Below that point reheat is added via baseboard heaters in order to regulate the zone temperature. VAV boxes are usually set to deliver some minimum amount of flow regardless of the room temperature. Turndown ratio is defined as the minimum percentage of wide open flow to which the VAV box will modulate. If the room is still too cool then a source of heat is activated to maintain the setpoint temperature.

For the 10 L/(s-person) design, the sizes of the minimum OSA damper (as compared to the ASHRAE RP-590 report) and the corresponding exhaust dampers were increased so that they would maintain their same respective pressure drops at the new airflow. The sizes of the other dampers were not changed.

A summary of the assumptions used follow:

1. Thorough mixing of outside air with return air takes place
2. dilution air mixes perfectly and instantaneously with the zone space air
3. Perfect control is assumed so there is no effect due to instrumentation lead/lag
4. The system attempts to maintain a slight positive pressure in each zone;
5. CO_2 generation and dilution in a zone remains constant (i.e., in a steady state)

Description of CO_2 Algorithm

The CO_2 concentration in each zone can be modeled, as it varies with time throughout the day, by considering the conservation of mass equation for a control volume. The resulting 1st order linear differential equation can then be solved. The general form of the conservation of mass equation can be simplified based on the assumptions stated above to :

$$\frac{\partial}{\partial t}\rho V = \rho_i V_i A_i - \rho_o \vec{V}_o A_o + \vec{MG} - \rho_A V_A A_A - \vec{\rho_R} V_R A_R \qquad (1)$$

The following symbols apply to the equation.:

A = area, ft^2	V = conditioned space volume, ft^3
C = CO_2 concentration, lbmol/ft^3	$\vec{V}$ = velocity, ft/min
E = filter CO_2 removal efficiency, decimal	P = density, lb/ft^3
G = CO_2 generation rate, lbmol/min	*Subscript* i = entering conditions
M = molecular weight, lbm/lbmol	*Subscript* o = exiting conditions
Q = volumetric flowrate, ft^3/min	*Subscript* A = absorbed air conditions
T = time, min	*Subscript* R = return air conditions

This equation can be solved subject to the initial condition that at t = 0, $C(0) = C_0$. If it is assumed that the amount of CO2 absorbed or filtered is negligible, the resulting solution to Equation (1) is

$$C(t) = C_i + (C_o - C_i)\exp\{-Qt/V\} + (G/Q)(1 - \exp\{-Qt/V\}). \qquad (2)$$

Note in Equation (2) under the assumptions of this work that $C_o = C_i$.

This solution to equation (2) was incorporated into a separate program written in Turbo C™ . The airflow output file from RP-590 is read into this program where it is used to calculate the CO_2 levels in each zone. A listing of the program code is given in Reference [4]. The program prompts the user for the following inputs:

1. Output filename from RP-590 that is to be used as the input for this program.
2. The filename for the output generated by this program.
3. CO_2 concentration of the outside air, ppm.
4. Conditioned volume in cubic meters of each zone.
5. Initial CO_2 concentration of each zone, ppm
6. Number of sources in each zone that generate CO_2. (Each source for a particular zone must generate the same amount of CO_2).
7. Generation rate of CO_2 in L/s of each source.

The following values were used as input to calculate the CO_2 levels in each zone:

- 400 ppm CO_2 in outside air
- 4,669 m^3 of space in each zone (from RP-590)
- 400 ppm CO_2 initial concentration in each zone
- 150 people/zone (from RP-590) = only CO_2 sources
- 0.0053 L/s CO_2 generated per person [2]

The computer program used in this work was written specifically for system 2 from RP-590; however, it can also be used to model system 1. To model the other systems, minor modifications would have to be made to properly scan the input files.

RESULTS

The output of the computer model produces some very interesting results. Tables 1 is a summary of the raw data for the case of 3 L/(s-person) outside air criteria for zone 3 (CASE A). Table 2 is a summary of the raw data for the case of 10 L/(s-person) outside air criteria for zone 3 (CASE B). (Tables for all zones for both outside air requirement cases are included in reference[4].The top twelve rows of data in each table show, for a particular turndown ratio (TD=fraction of maximum air flow), the number of hours for a typical day of the corresponding month that the 1000 ppm limit was equaled or exceeded. The bottom five rows show (1) the percentage of the yearly hours that the zone exceeds or equals the CO_2 limit, (2) the average value of the CO_2 concentration, (3) the standard deviation, (4) the maximum value, and (5) the minimum value. The American Conference of Governmental Industrial Hygienist has adopted a Threshold Limit Value – Time Weighted Average (TWA) of 5000 ppm for CO_2 [3]. Due to space considerations raw data can not be included in this paper but is included in reference [4]. The *maximum* data values listed in Tables 1 and 2 give evidence that this TWA may be exceeded. By referring to the raw data it can be seen that the TWA is indeed exceeded. For example for Zone 3 the TWA is exceeded for 32 consecutive hours for CASE A.

By referring to the *maximum* data values in Table 2 (for CASE B) it can be seen that the largest values are much less than the TWA.

On average for CASE A, the 1000 ppm limit is exceeded for all turndown ratios in all four zones. Even under constant volume operation, this design fails to maintain the CO_2 concentration below the 1000 ppm limit approximately 30% of the time. Surprisingly for CASE B, the percentage of time that this system equals or exceeds the limit is very close to the results for CASE A. However, the amount that the CO_2 concentration exceeds 1000 ppm for CASE B is much less than that for CASE A. These details can be seen in reference [4].

Although space does not allow the details for all zones to be presented, the complete results shown in reference [4] show that during mild months when Zones 1, 2, and 4 are well ventilated, Zone 3 (North) is not. This occurs because of the low thermal load in Zone 3, which requires that the VAV box modulate down.

Table 1. Zone 3-Number of hours (in bold) that CO2 is above 1000ppm @ 3 L/(s-person)

Month	Days/Mon	TD=0.4	TD=0.5	TD=0.6	TD=0.7	TD=0.8	TD=0.9	TD=1
Jan	31	***24***	***24***	***24***	***20***	***17***	***12***	***0***
Feb	28	***24***	***24***	***21***	***18***	***15***	***10***	***0***
Mar	31	***19***	***19***	***17***	***15***	***12***	***5***	***0***
Apr	30	***13***	***13***	***9***	***5***	***0***	***0***	***0***
May	31	***12***	***12***	***12***	***12***	***11***	***11***	***11***
Jun	30	***16***	***16***	***16***	***16***	***15***	***15***	***15***
Jul	31	***22***	***22***	***22***	***21***	***21***	***21***	***21***
Aug	31	***24***	***24***	***24***	***24***	***24***	***24***	***24***
Sep	30	***12***	***12***	***12***	***12***	***10***	***10***	***10***
Oct	31	***17***	***17***	***6***	***6***	***5***	***5***	***5***
Nov	30	***18***	***18***	***15***	***11***	***6***	***0***	***0***
Dec	31	***24***	***24***	***24***	***24***	***19***	***16***	***11***
Yearly % time out Of compliance		78	78	70	64	54	45	34
Average CO_2 ,ppm		2388	2356	2312	2272	2176	2052	1922
Std Dev CO_2,ppm		1855	1889	1956	2034	1985	1863	1721
Maximum CO_2,ppm		6821	6840	6982	7199	6498	5653	5253
Minimum CO_2,ppm		828	824	828	821	821	787	749

Table 2. Zone 3-Number of Hours (in bold) that CO_2 is above 1000ppm @ 10 L/(s-person)

Month	Days/Mon	TD=0.4	TD=0.5	TD=0.6	TD=0.7	TD=0.8	TD=0.9	TD=1.0
Jan	31	***24***	***24***	***24***	***20***	***16***	***13***	***3***
Feb	28	***24***	***24***	***21***	***18***	***15***	***10***	***0***
Mar	31	***20***	***19***	***17***	***15***	***11***	***5***	***0***
Apr	30	***13***	***13***	***9***	***4***	***0***	***0***	***0***
May	31	***12***	***11***	***10***	***11***	***11***	***11***	***11***
Jun	30	***16***	***16***	***16***	***15***	***15***	***15***	***15***
Jul	31	***21***	***22***	***21***	***21***	***21***	***21***	***21***
Aug	31	***24***	***24***	***24***	***24***	***24***	***24***	***24***
Sep	30	***12***	***12***	***11***	***10***	***10***	***10***	***10***
Oct	31	***16***	***17***	***6***	***5***	***5***	***5***	***5***
Nov	30	***18***	***18***	***15***	***11***	***6***	***0***	***0***
Dec	31	***24***	***24***	***24***	***24***	***19***	***15***	***10***
Yearly % time out Of compliance		78	78	69	62	53	45	35
Average CO_2 ,ppm		1759	1713	1653	1592	1514	1424	1334
Std Dev CO_2,ppm		840	847	879	910	893	836	769
Maximum CO_2,ppm		3570	3593	3600	3507	3197	2870	2586
Minimum CO_2,ppm		834	823	828	822	823	789	753

DISCUSSION

The HVAC system analyzed was designed for the older ASHRAE recommendation of 3 L/(s-person) of outside air and consequently will not adequately keep the zone CO_2 levels below the current ASHRAE recommendation of 1000 ppm. However, even with the minimum outside air and corresponding minimum exhaust air dampers increased in size (maintaining the same pressure drops) for 10 L/(s-person) required by ASHRAE 62.1-2004, this system allows a significant number of excursions above the 1000 ppm limit. Simply increasing the size of the outside air dampers will not cure the problem. The thermal load can still dictate how much outside air is delivered to a zone. Even if the internal thermal loads are identical, the loads due to the external exposures can cause significant differences. The prudent engineer will utilize ASHRAE Standard 62.1-2004 that describes a method to calculate the required amount of outside air for multiple zone systems. This manifests itself as an increase in the total amount of outside air for the system over and above what would be calculated by simply multiplying the total number of people by the required ventilation per person as tabulated in the standard. This still leaves open the issue of how to maintain the minimum amount of outside air at the minimum turndown ratio for the zone VAV box. One strategy could be to install a CO_2 sensor in the critical zone that would override the thermostat and drive the VAV box open to deliver more air to the zone during low load situations. If the CO_2 sensor were still not satisfied, it could override the speed modulating controller for the supply fan and speed up the fan to call for more air. The supply duct pressurization would then have to be kept within allowable limits. This could be accomplished by proper placement of relief dampers or through utilizing a hi-limit pressure controller to open the other VAV boxes in some order to relieve the pressure. The thermostat would continue its normal operation and call for reheat to maintain the space temperature.

For both design cases, in each of Zones 1, 2, & 4 a turndown ratio of 0.7 performs about as well as a turndown ratio of 1.0. Zone 3 shows a significant percentage improvement with increasing turndown ratio. This would seem to indicate the need for other system design modifications. For the design analyzed in this paper, the return ductwork offers less resistance than the exhaust ductwork. Less resistance in this branch would allow more return air to re-circulate and keep the corresponding portion of outside air from being brought into the system. In this case the return air damper should be decreased in size, thereby, increasing the pressure drop and restricting the return air flow. Increasing the resistance in this branch would allow more outside air to enter the system.

Even with a turndown ration of 1.0, each zone in both design cases allows the CO_2 concentration to exceed the limit. This indicates that the outside air dampers and exhaust dampers are undersized.

The system modeled in ASHRAE Research Project RP-590 was not designed with a load diversity factor. Load diversity is a factor that takes advantage of the fact that the different zones reach their peak thermal load at different times of the day. When utilizing the load diversity factor, smaller equipment can be purchased resulting in lower initial cost. However, the engineer must be very careful to compare the ventilation load required by the CO_2 generators with the thermal load and select equipment capable of satisfying both loads at all conditions.

This model, though quite simple, provides valuable insight to those interested in analyzing the variations in CO_2 levels in many types of buildings. The steady state assumption will not be valid in all types of buildings and occupancies, but should be adequate in settings where the occupancy remains fairly constant throughout the day.

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Energy efficient ventilation systems

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SUMMARY

Changes of actual load often lead to trouble with the operation of the ventilation system. The design data are no longer correct if, for any reason, the actual load has permanently changed during the usage. The proposed ventilating system maintains the most commonly developed harmful gaseous contaminant content below the permissible level in spaces with forced ventilation. Sensors are mounted in every room, in a given height. By following the demand, the supplied fresh air volume matches the momentary load. Thus, possible health hazards can be mitigated and acceptable life- and working conditions can be maintained in these spaces. The requested indoor air quality is assured in any ventilated space without disturbing the air supply to other spaces. One of the main advantages of this system is that no energy is wasted for the handling of any excess air when the system load is below the design load.

1. INTRODUCTION

1.1 Background

When the air exchange rate in a room is maintained at the nominal design value during periods of excess load, the concentration of air contaminants is likely to exceed the permissible limits. Some of these contaminants can be identified easily, but there might be some that can not be perceived with senses. Continuous inhalation of the air of such compounds can cause harmful physiological effects[1]. Proper indoor air quality can be maintained if the fresh air supply rate corresponds to the load imposed by the number of people, by their activity in the room, and by the altered technology's requirements. Of course, if the air supply rate is high enough to match the highest load, the excess circulation wastes energy. A ventilating system is usually selected for a given, constant scenario, for the demand expected at the time of the design. These data are no longer correct if for any reason, the actual load has changed during the usage. Experience shows that the differences between the design and the real load of ventilation systems can be significant. Without listing all the possible reasons, they are the following:

- The customer is not aware of the exact load, the number of people, and the equipment heat load in the spaces at the time of the design.
- The load, the number of people or the equipment heat load changes considerably during operation of the ventilation system.
- The layout of the space is altered resulting in changes of the volume and the air flow or the function of the space is changed.

Usually, it is not simple to predict these variations, and it is not possible to mitigate the effect of the load distribution changes during operation. Thus the whole ventilation system should be developed in a way that it would be capable of furnishing sufficient fresh air according to the varying load. Applying adequate control and suitably modified air duct layout, the gas contaminants' concentration can be kept below the health hazard limit.

1.2 Theoretically demanded air flow

Requirements of indoor air quality comprise the physical characteristics of the air, and its composition. Because of the natural usage, harmful gaseous contaminants, most commonly carbon dioxide, carbon monoxide, and different nitrogen based mixtures develop in spaces that have no windows or doors opening directly to the fresh air. The concentrations of these gases might reach critical values[2] during natural usage of any space if it is not compensated with sufficient fresh air.

People exhale 14-30 l/h carbon dioxide[3] depending on their activity. In practice, the average of this range of carbon dioxide is considered to be the determinant of fresh air needed per person. According to the customary proportional control formula the fresh air demand due to respiration is:

$V_s = K/(p_{er} - k_{out})$ [m³/h] /1/

Where V_s – the supplied fresh air quantity [m^3/h]; K - the generated air contamination quantity [g/h]; k_{per} - the permitted level of the contaminant in the room [g/m^3]; k_{out} - the average content of air contaminant in the outside air [g/m^3][4].

There are professional recommendations proposing the fresh air supply as a function of the activity between 25 - 90 m^3/h per person[5] for office work.

1.3 The task to solve

The load often changes during operation of a ventilating system. The load might diminish or rise depending on the number of people in the room, on the time they spend there, on their activity level and on the technology's requirements. In the first case, energy will be wasted because of the excess air supplied. In the latter case, it is vital to provide fresh air, to prevent the contamination's possible harmful effect on the health of the occupants.

There is need for a general method or system that can operate in a reliable way in every air conditioning or ventilating system to suit the altered demand. There is need for such a system that assures the appropriate indoor air quality in order to protect people from the mentioned harmful effects. The ventilating system should follow the load between certain limits, neither less than necessary to maintain acceptable working conditions, nor more, to avoid wasting energy.

The reason of application of contaminant controlled ventilation is twofold. On the one hand, the supplied fresh air quantity can be controlled independently from varying human load. On the other hand, energy efficient operation should be assured.

2. METHODS

2.1 Motivation

In addition to the requirements on the thermodynamic properties of the indoor air, for example temperature and humidity, there are requirements on its chemical composition. Up until now, no regular attempt were made to monitor the variation of the inside air's carbon dioxide content in ventilated spaces during the operation of the ventilation system. During normal usage, different types of gaseous contaminants enter the air that are hazardous to health. One of these contaminants is carbon dioxide. It is used as the most common indicator of the indoor air quality, is not poisonous but in continued excess concentration might lead to oxygen deprivation. Since this contaminant gas is colorless and odorless, the occupants are not aware of its presence. It affects the occupants to varying degrees, depending on their individual susceptibility, particularly their alertness and effectiveness.

The equation in 2.2 would only be valid in reality if the level of the inside air's carbon dioxide content /k_{per}/ remained below the design limit. In reality, this value differs from the

theoretically proposed one. This content is determined by the load. That is, its quantity depends on how many people are staying in a room and for how long or how much fresh air is demanded by the altered technology.
While the physical characteristics of the indoor air are directly measurable, its quality is harder to monitor. For example, it can be measured indirectly by following the change of the CO_2 content. With suitable instrumentation and appropriate controls, its concentration can be reliably measured, and the required indoor air quality can be maintained in the ventilated space.
There is another issue that requires special concern. The ventilation system seems to be the only among building services that might cause direct harm on the surrounding, on the occupants health when it performs poorly. It is quite likely, that people working inside a building, spent about 90 percent of their lives inside. During this time they might be exposed to the mentioned harmful effect. Offices for example can be regarded as “continuous operation”, where people stay mostly at the same location. One can assume that the individual’s performance depends on the surrounding conditions as well as on their own capability. The office environment along with the suitable indoor air quality is an important part of these surroundings. That is why the proper ventilation is so important especially where the forced ventilation is the single source of supplying fresh air. That is in spaces having no doors or windows directly to the fresh air.
Because of the nature of public building, continuously moving and varying number of people can be expected. The consequence is the change in ventilation load. Either people sojourning there change their place, leave or arrive, or visitors arrive or leave in unknown number and time frame. Between certain limits this change should be predictable for most of the systems at the time of the design based on the building’s function. The building’s owner, the user, and the operator should be aware of the design limits.
Recent professional studies have examined the main advantages of CO_2 based control applied in ventilation systems. According to the published articles, CO_2 based control assures adequate indoor environment. The symptoms reported because of inappropriate inside environment reduced or disappeared. There are also indications that buildings installed with CO_2 controlled ventilation show reduced demand under part load conditions[6].

3. RESULTS

3.1 Proposed arrangement

The arrangement on Figure 1. is one among the numerous recommendations with the purpose to assure the required indoor air quality. There are two prospects considering the most economic operation of the proposed ventilating system. Energy might be saved when the transitory surplus air heating- or cooling energy can be prevented, and also when the fan is working at its highest efficiency. The control system is tracking the actual load, and supplies the necessary supplemental fresh air. By monitoring the instantaneous load, the fresh air supplied to the space is adjusted. Consequently, the air contaminant's quantity is maintained below the critical value as the supplied fresh air quantity suits the actual load, the number of people the activity in the room or the technology requirement. There are two prerequisites concerning this system; individual spaces should be separated from each other regarding the ventilation, and the heat load generated by people and the heat loss due to transmission is not considered.
Sensors are mounted in every room at an appropriate height in the breathing zone. Following the sensors' signals, the room is supplied with fresh air until the concentration of the contaminants decreases below the threshold limit. The control system gives alarm signals,

when the monitored gaseous contaminant's concentration constantly exceeds the harmful limit. These alarms may be digital, acoustic or visual. This control system does not replace the control system based on the air physical characteristics. It is proposed to ensure the air supply within V_s = 0,2-1,2 V_{nom} intervals referring to the whole system. There is need for

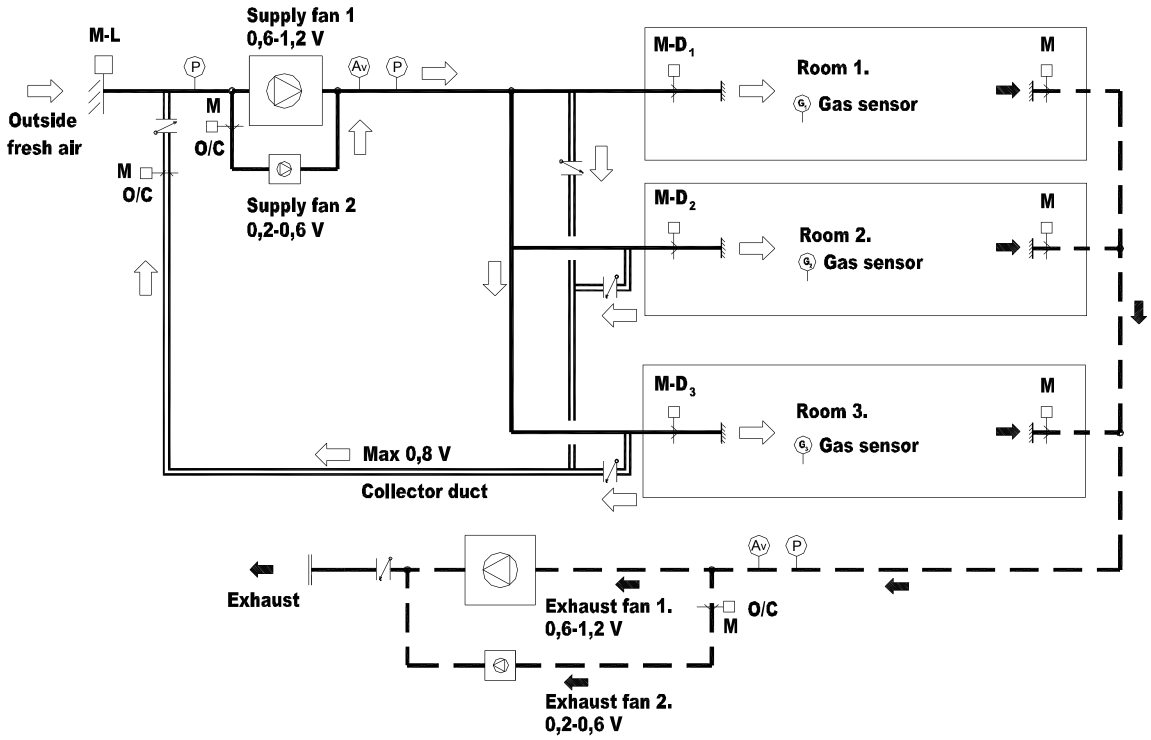

Figure 1. System layout

supplemental air when the load in the space exceeds the design conditions. The high limit of the air flow has been chosen at 1,2 $V_{nominal}$ regarding the whole system. During periods of low loads, through the third duct the excess air recirculates to the fan inlet, or the surplus air is flowing ahead the fan into the system through a central by-pass of the supply duct. The discussion entirely disregards the control in the air-heating and air-cooling process. Room temperature supposed to be constant. One of the main purpose of this arrangement is the stable operation of the fan on the designed or on the nearby operating point and with the possible favorable efficiency. The smaller fans indicated on Figure 1. are provided for extended periods, potentially several days, of low load conditions. Then the controlled data, pressures, air volumes determines when to shift fans.

All the proposed limits are arbitrarily chosen. Their values might vary depending on the size of the room, and the building, on the type or function of the building. In any case, it has to be considered during the design stage. It has to be mentioned that the load tracking control system does not substitute the control system based on the indoor air's temperature. The suitable air quality in each supplied space can be assured solely by the sufficiently harmonized operation of both control systems.

3.2 Proposed steps of sizing

The proper operation of the ventilation system can not be achieved or maintained in the ventilated spaces without adequate control. The determination of the exact or the near exact quantity of fresh air is important in order to avoid any probable health's risks on occupants' and also to provide the energy efficiency of the ventilation system. The basis of calculation is not a specific air volume any more, but an interval of air volume. There are upper and lower limits upon sizing and selection should be based.

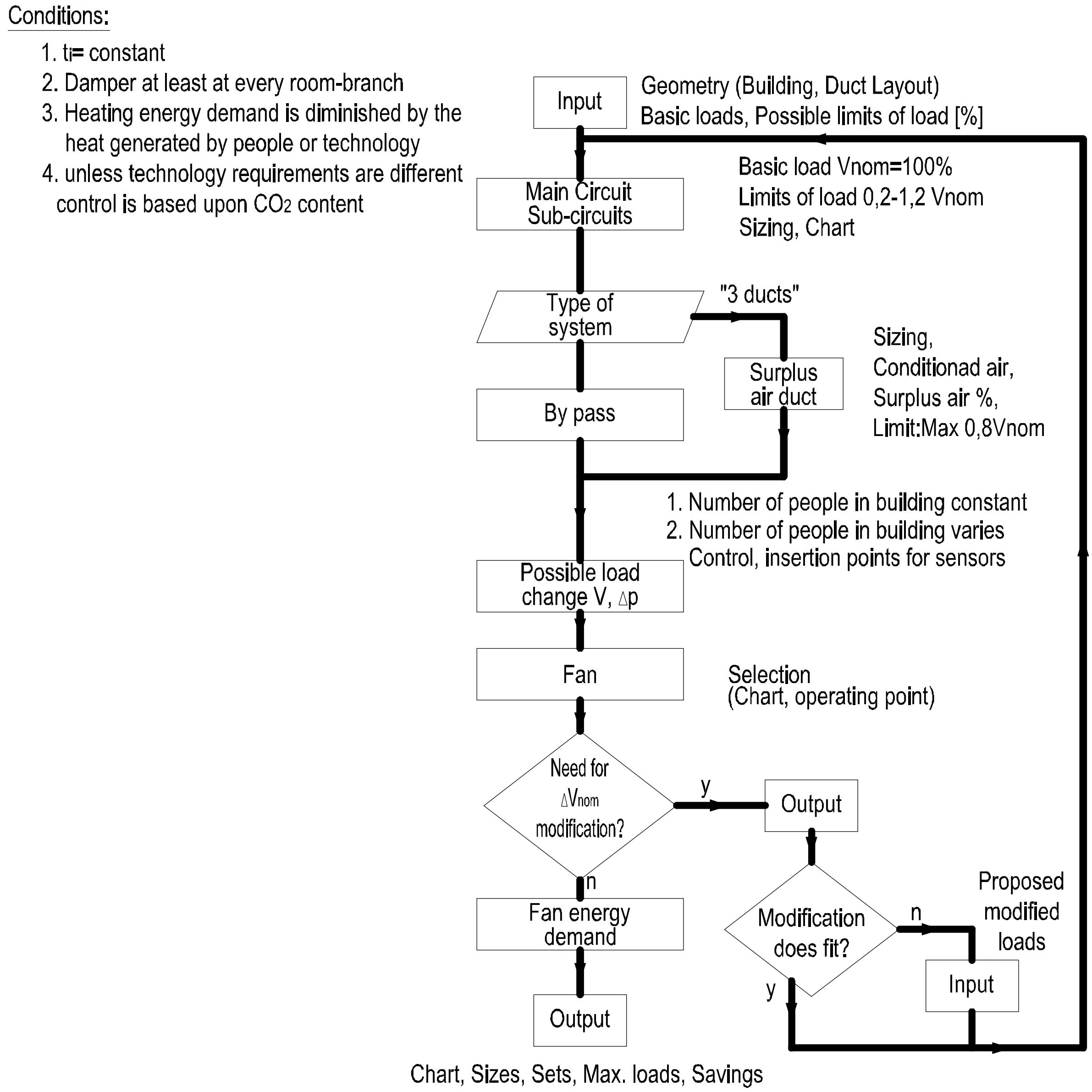

Figure 2. System flow chart

The basic data of sizing is similar to that of any other calculation, geometric and building data, tracing of air duct, spaces available and the nominal loads. New types of data are the predicted surcharges by rooms and the predicted lowest loads. The predicted surcharges comprise the maximal number of occupants by ventilated spaces, and the duration they can sojourn there without being exposed to any harmful effect. As the calculation, it is essential to be aware of the permissible surcharge of the ventilating system. When its exact value is not known at the time of the design, the proposed limits of the supplied air volume are $0,2\ V_{nominal} - 1,2\ V_{nominal}$. The main steps of sizing are shown in Figure 2.

Accordingly, after reading the INPUT data, succeeds the preliminary duct sizing based on the usual limits, as the velocity limit in the air duct. After completing this cycle the next is the air duct sizing, the pressure condition of the by-pass or that of the 'third duct'. Afterwards the proposed sizing made by using the predicted load limits of every room, and the controlled values succeed. The next is the determination of simultaneity loads for the whole system, the lowest and permissible highest loads by rooms, and also the calculation of the preset value for each room's damper. The process continues with the fans' selection for the most disadvantageous loads along with the estimation of their operational costs. This is the time for revision and entering the revised values. Re-controlling is based on the revised data is the next step. Then the selection of the fan of smaller capacity comes after if it is necessary, when the load is below 0,6 $V_{nominal}$. The result, the OUTPUT of the calculation is the air ducts' sizes, the permissible extreme load limits, the sets of air dampers for 100% and for the extreme loads, the selection of sensors insertion points in the duct, and that of the measuring devices, the operating characteristics of the fans, and forecast for energy-saving operation. It is also highly recommended to be marked on the design the maximal permissible load per ventilated spaces, the supplied surplus air volumes along with other values. The floor plan and the scheme should contain the maximum number of occupants per spaces, the insertion points of all the control and measuring devices. Besides checking the system and the definition of the room's permissible load it calculates, how much energy can be saved by operating this increased valued ventilating system.

3.3 Example for theoretically possible energy savings

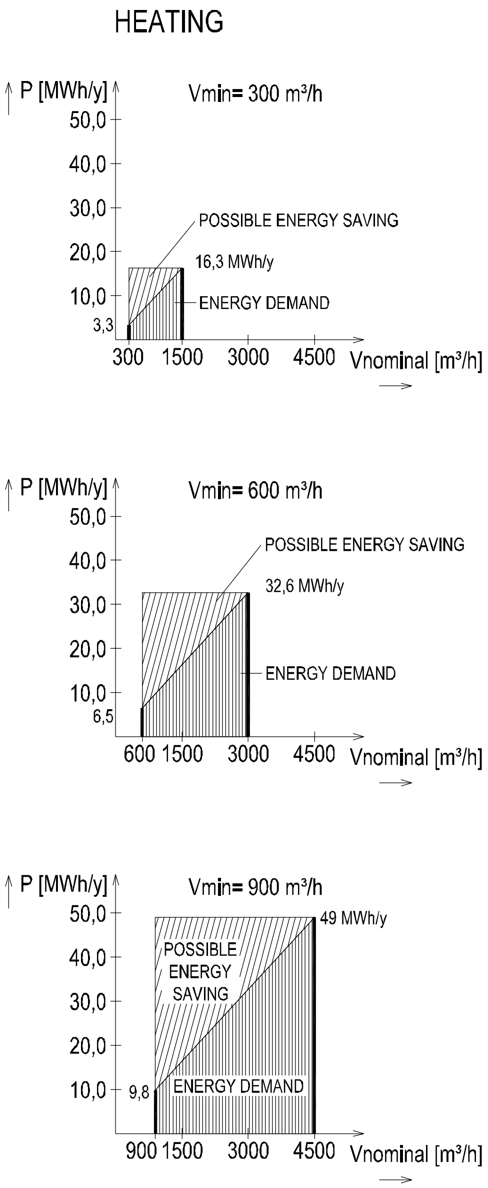

Figure 3. Theoretically possible energy saving of air heating

Due to variable load the difference between the design and the actual fresh air need resulted also a difference in energy use of the operation of the ventilating system. This is the difference at low load conditions that can be saved by using the contaminant controlled ventilation system. There are two prospects considering the most economic operation of the proposed ventilating system. Energy might be saved when the transitory surplus air heating- or cooling energy can be prevented, and also when the fan is working on its highest efficiency.
The next example will discuss the theoretically possible energy savings under varying air supply. The energy need is directly proportional to the transported air volume, to the pressure difference consumed by the air distribution system itself. Concerning energy savings, the example is studying the change of energy demand in winter season. The data do not contain any heat gain generated by people, or heat loss through the building structure. It refers solely for the heating of the supplied fresh air. Otherwise, the energy need for cooling is disregarded in this discussion.
Conditions are the next: space inside temperature $t_i = +20^oC$; outside temperature $t_o = -15\,^oC$; system operation: 5 days a week; duration of daily operation: 10 hours; fan1 supplied fresh air quantity V_1 =1500 m3/h; fan 2: V_2 =3000 m3/h; fan 3: V_3 =4500 m3/h.
On Figure 3. are indicated the yearly energy needs for fan No.1, No.2, and No.3 transporting the different air volumes, from $V_s = 0{,}2\ V_{nominal}$, up to $V_s = V_{nominal}$. There are indicated the yearly heating energy needs. The yearly fresh air heating energy demand is determined according to [7]:

$$Q_a = G_a \text{ x } V \text{ x } c \text{ x } 10^{-6} [\text{ GJ/year }] \qquad /2/$$

where Q_a –heat load of the transported fresh air [GJ/year]; G_a – yearly heat gap of ventilation [kh/y]; V – transported fresh air [m3/h]; c – specific heat [kJ/kgK].
Since the proposed ventilating system is capable to supply varying quantity of air from the lowest load $0{,}2\ V_{nominal}$, the theoretically possible energy saving during heating season might reach 80%. Although this study does not contain operating energy savings possibilities, beyond this probable energy savings, the fan's operation should be mentioned too. There are estimations showing that fan operating costs takes almost half of the energy consumptions of a building. It means that the fan's energy demand might influence considerably the building energy consumption. While energy saving in cooling season is not comprised in this paper, it has to be mentioned that heating is seasonal operation, while the fans are working continuously, almost year round. This is one of the features that do augment the achievable energy savings. The values in the example are theoretical that in reality might vary slightly because of different constructing, operating or simultaneity conditions.

4. DISCUSSION

4.1 Differences from traditional ventilating systems

The proposed arrangement is based on the traditional ventilating system's element, as e.g. air handling unit, the air distributing duct system, the supply and exhaust grids, exhaust ducts and exhaust fans. There are a few differences from the traditional ventilating systems:

- $V_{supply} = 1{,}2\ V_{nominal}$, oversized system /grids, duct, AHU/
- Air volume controlling devices at every room supply connection
- Third duct to reveal actual surplus air $V_{supply} = 0{,}8\ V_{nominal}$

- By pass in the main supply duct with possible air transport of $V_{supply} = 0,6\ V_{nominal}$
- A supplemental fan of smaller capacity in the supply and exhaust main duct
- Carbon dioxide and/or other sensors installed in the ventilated spaces
- Pressure and air volume sensors in the air duct

4.2 Advantages

The advantages of the proposed ventilating system are: proper life- and working conditions can be maintained in the ventilated space; appropriate indoor air quality can be assured in any distinct room with varying load, without disturbing other space's ventilation; the possible health hazard effect of gaseous contaminants to people is prevented; demand controlled ventilation can be assured in every spaces; beyond the nominal ventilating fresh air, between certain limit, sufficient amount of fresh air can be supplied to any ventilated room; none of the parts or element of the ventilating system is operating unnecessarily; the cooling, heating or transporting energy of the momentary superfluous air can be saved; occupants are continuously informed of the momentary air quality; occupants and operator are warned when indoor air quality is unacceptable in any space;

This arrangement requires slightly larger duct, grids or possible air handling unit, and an additional control. All of these augment the value of the ventilation system. What is more, these are one time expences that might recover soon not only in energy savings but also in the enhanced working ability the occupants. Depending on architectural conditions, it can be installed in existing ventilating systems too. Although the control system can be developed according to the description, its efficiency can only be proved after successful experimental testing, by an installed and reliable working system.

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Heating and ventilation of the contemporary warehouse complexes

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INTRODUCTION

Cargo handling industry is an important part of the economy. Construction of warehouse complexes utilizing the most modern storage, cargo reception and dispatch technologies is rapidly developing now.

In 2002-2005 OJSC "NPO Termek" and OJSC "TSNIIPromzdaniy" designed and installed heat and ventilation systems for industrial & trade complex "Sherland', where the warehouse area is 26 000 m^2.

Figure 1. Industrial & Trade Complex "Sherland".

METHODS

The complex is situated in the vicinity of Sheremetyevo-1 airport, 8 kilometers off the Moscow Ring Road. Warehouse facilities of the complex represent dry heated premises and are provided with the up-to-date equipment. Automated system of stock control allows for high dynamic cargo handling at all stages of the logistic chain – from cargo receipt into the warehouse up to its dispatch.

All warehouses are equipped with six level rack structures, intra-depot equipment, automated computerized system of stocking activity as well as with monitoring, control and alarm systems, etc.
Warehouse premise block represents a single-store, four aisle blocked building shown on Figure 2. Each aisle contains rack storage. Basic specification of each warehouse is given in Table 1.

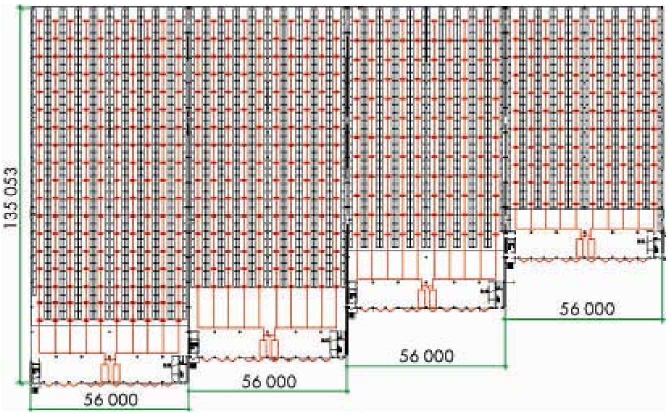

Figure 2. "SHERLAND" Rack Storage Complex. Layout.

The main requirement to the air parameters inside the warehouses is a uniform air temperature distribution in the plan and, especially, throughout the height. Analyses of eventual heating schemes and systems of the rack storages demonstrated that the most rational heating system is a combined heating system, which utilizes local heating registers placed close to the floor alongside the walls and a hot-air heating system with intensive air baffling in the space of the premise. Main heating load comes to the hot-air heating.

A hot-air heating system having deflecting nozzles is supported by a perimeter hot-water heating system with heat registers. Additional heating system was provided to prevent cooling of wall-adjacent zones. As the racks are placed virtually right up to the walls it was impossible to supply the hot air into the wall adjacent zone.

Table 1. Basic Specification of Industrial & Trade Complex "Sherland" Warehouse Premises.

Pos.	PremiseWare-house No:	Dimensions a*b*h, m	Area, A, 1000 m²	Volume, V, 1000 m³	Rack Space, Vrack, 1000 m³
1	1	90*56*17	5.04	85,68	44.12
2	2	!08*56*17	6.05	102.22	52.95
3	3	126*56*17	7.06	119.95	61.78
4	4	135*56*17	7.56	128.52	68.19

Hot-air heating system with deflecting nozzles presented at Figure 3 [1] is a modification of the widely used «Direvent» system [2] and provides for virtually uniform air temperature distribution throughout the height of the serviced premises. The system enables to provide an efficient heating at minimized consumption of the air supplied by the nozzles at high velocity and with a considerable temperature differential to secure specified microclimate parameters in the working area. The deflecting nozzles are placed between the racks in the upper zone of the premise and supply the hot air vertically down towards the working zone. Energy efficiency of the system with the deflecting nozzles is secured by a uniform air temperature distribution throughout the height and by elimination of the premises' upper zone overheating.

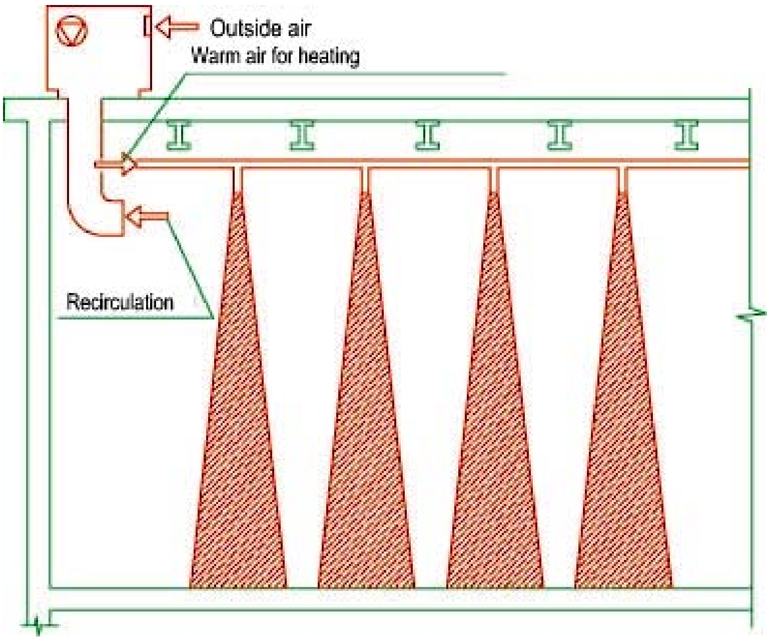

Figure 3. Outline of the Air Heating System with Deflecting Nozzles.

Table 2 presents the heat load design results for the heating and ventilation systems of the storage premises.

Table 2. Calculated Heat Load on the Heating and Ventilation Systems of Industrial & Trade Complex "Sherland" Warehouse Storage Premises.

Pos	Premise Ware¬house No	tн, °C*	t_B, °C**	Heat Consumption, kWh			
				Hot-Water Heating	Ventilation ***	Hot-Air Heating	Total
1	1	−28	12	174.3	402.0	46.7	623.0
2	2	−28	12	164.8	486.3	46.7	697.8
3	3	−28	12	189.1	567.2	46.7	803.0
4	4	−28	12	390.3	607.7	109.7	1107.7

* Outside air temperature;
** Inside air temperature;
*** Heat consumption for ventilation is designed at the assumption of a single air renewal in the premise space portion having the height of 6 m.

Each warehouse is equipped with two forced ventilation units placed at the mezzanine floors. The forced ventilation installations contain air mixing chambers with valves on both outer and recycle air ducts, which allows for control of outer and recycle air ratio in the course of the operation. During non-heating season the outside air only is supplied into the warehouse in the ventilation mode. During transitional period and heating season the flow of the outside air is decreasing in the heating mode combined with the ventilation depending on the outside air temperature and cargo storage conditions. The forced ventilation units are equipped with multi-speed electrical motors, which enable to provide quantitative adjustment of the systems and secure their high energy efficiency.

Removal of the exhaust air form the warehouse is natural via exhaust shafts on the roof combined with the smoke removal shafts.

Air ducts of the forced ventilation system are laid down in the roof-truss space alongside the passageways between the racks. Height from the inlet nozzle edge to the floor of the premise is 13.5 m. Air ducts networks are combined in pairs by a bypass duct, which allows for 50% hot-air heating back-up in each warehouse.

Diagram of the warehouse hot air heating system is presented at Figure 4.

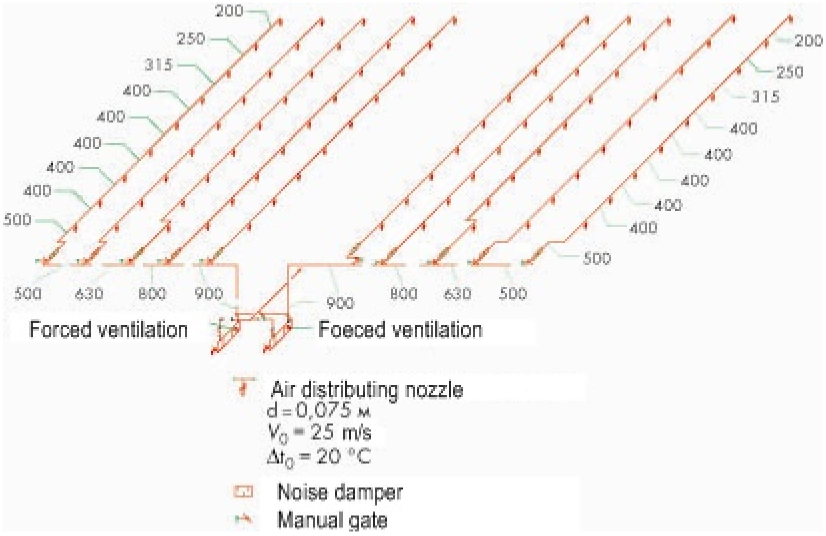

Figure 4. Diagram of the Warehouse Hot Air Heating System.

Design of the hot-air heating system with deflecting nozzles is executed in conformity with “Recommendations for Design of Heating and Ventilation Systems with Deflecting Nozzles” [3].

When designing the air distributing system we have accepted the following provisions as criteria:

- Minimal number of nozzles, which was provided by a maximal air supply velocity;
- Maximal supplied air temperature, which provided minimal consumption of the fresh air;
- Minimal deviations of air temperature and velocity in the working zone during the heating season under fresh air temperature variations.

Design results are presented in Table 3.

Table 3. Design Results of the System with Deflecting Nozzles

												$t_н$ = -28 °C
Pos.	Premise	Δt_0, °C	d_0, m	LΣ, m³/h	V_0, m/s	L_{nozzle}, m³/h	n, off	Cell Width, m *)	H, m	X_{max}, m	$K_н$	$V_{w.z.}$, m/s
1	Warehouse No.1	20	0.075	35250	25	396	90	9	21.6	11.9	0.58	0.5
2	Warehouse No.2	20	0.075	57030	25	396	140	7.1	21.6	11.9	0.58	0.5
3	Warehouse No.3	20	0.075	71750	25	396	180	6.5	21.6	11.9	0.58	0.5
4	Warehouse No.4	20	0.075	77290	25	396	200	6.75	21.6	11.9	0.58	0.5

The following notations are used in the Table (Respectively):

Δt_0 – Temperature difference between the fresh air and the air inside the premise;
d_0 – Nozzle diameter;
L_Σ – Total air consumption for the system;
V_0 – Exhaust air velocity;
L_{nozzle} – Air flow rate through the nozzle;
H – Geometry parameter of the jet;
X_{max} – Jet range capability;
$K_н$ – Jet non-isothermic factor;
$V_{w.z}$ – Air velocity in the working zone.

Width of the cell, where the air jet is developing that is coming out of the nozzle was selected having in view the requirement to provide a uniform air temperature and velocity distribution in the zone to be serviced and to provide the jet development without its cross-sectional constraint.

Installation and commissioning works were conducted in 2004-2005.

Air temperature and velocity were measured at the jet discharge spot and in the working zone alongside the fresh air jet in warehouse No.4 in the course of the commissioning works.

At the moment of measuring the outside air temperature was $t_в$ = - 0.4 °C; air discharge velocity from the nozzle and its temperature was respectively: V_0 -22.6 m/s; t_0 = 17.6 °C.

The measurements proved that:

- Air temperature throughout the warehouse height and working zone area is virtually constant ($t_в$ = 19.6 ÷20 ° C);
- Air velocity in the working zone does not exceed 0.35 m/s;
- Air velocity distribution alongside the jet axis is close to the design one (Figure 5).

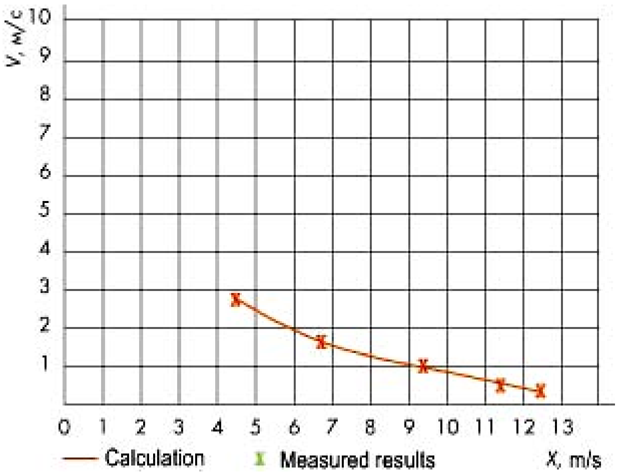

Figure 5. Air Velocity Distribution alongside the Jet Axis.

THE CONCLUSION

Utilization of the hot-air heating system combined with ventilation that is equipped with the qualitative and quantitative control and with air supply by the deflecting nozzles is perspective for large rack-store warehouses.

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Ventilation conditions of different indoor environments in a university

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SUMMARY

Limited data exist on indoor air and environmental quality (IEQ) in schools, and how IEQ affects students' health or performance in China. Research was conducted in different types of indoor environments in a university to explore possible relationship. Indoor temperature, relative humidity (RH), and CO_2 concentration were continuous monitored while outdoor parameters combined with on-site climate conditions were recorded. Questionnaire concerning time-activity patterns, judgment about IAE in campus and comfortlessness possibly relative to bad ventilation was distributed to undergraduate students. Dormitories (n=20), classroom (n=20), reading rooms (n=5) and meeting rooms (n=5) were selected. Average air exchange rates were calculated using CO_2 as the tracer gas when indoor environments were not occupied. Results indicated that the best ventilation was achieved in reading rooms, and the worst situation was found in dormitories, classrooms and meeting rooms, which was accordant with occupants' more complaint. In dormitories, factors such as outdoor climate conditions, deficiency of building design, room usage and living habits of students, were considered to be essential on ventilation and indoor CO_2 concentration.

INTRODUCTION

Indoor environmental quality may affect physiological and psychological process, which may affect activities that may affect performance at tasks that may interact with other factors to affect overall productivity [1]. Ventilation consists of unintentional air infiltration through the building envelope, natural ventilation through open doors and windows, and mechanical ventilation. Adequate ventilation has the potential to mitigate concentrations of indoor-generated air pollutants. Ventilation within schools is important because there is varying evidence on the effect of low ventilation rates on performance and health [2].

The CO_2 concentration of exhaled air is higher than that of the outdoor air, leading to an increase in the CO_2 content in the closed spaces. Elevated indoor CO_2 concentrations may indicate inadequate ventilation per occupant and elevated indoor pollutant concentrations, leading to SBS symptoms [3].

The indoor concentration of carbon dioxide (CO_2) has also been used as a surrogate for the ventilation rate per occupant, including in schools [4]. The current American Society of Heating, Refrigeration, and Air-conditioning Engineers (ASHRAE) recommended for adult office workers, assuming a ventilation rate of 7.5 l/s per person and a typical outdoor CO_2 concentration of 350-400 ppm, a steady-state indoor CO_2 concentration of 1000 ppm has been used as an informal dividing line between 'adequate' and 'inadequate' ventilation. It also specifies a minimum ventilation rate of 7.5 l/s (15 ft^3/min) per occupant for classrooms [5].

Up to 2005, there are approximately 2273 colleges and universities in China, where more than 2300 million college students study and live [6]. Many students spend most of their time in classrooms, libraries, laboratories, dormitories, and other indoor environments. So there is a question of indoor environment quality and its effect on students' health and performance. In response to this concern, the study was conducted in a university in north China for current status of ventilation, as well as students' perception on indoor environment quality on campus.

METHODOLOGY

Field measurement

Tianjin, 137 kilometers east to Beijing, is the second largest city in Northern China after Beijing. As a metropolis of commerce, trade and industry, Tianjin has a resident population of approximate 10 million. This study was undertaken from June to August in 2006, with an ambient temperature of between 18 and 39 °C, a relative humidity of 35- 90%, and wind speeds up to about 10 m/s. The climatic condition was quite comfortable. All the meteorological measurements were done on-site while the field investigation was conducted.

Previous research on CO_2 in school classrooms [7] indicated a single monitoring location was appropriate for characterizing such indoor contaminant levels when HVAC systems were on, i.e. air was well-mixed. Considering probably inadequate full-mixing, 3 sets of equipment were placed in each classroom, reading room and meeting room.

Dormitories (n=20), reading rooms (n=5), air-conditioned classrooms (n=10) in an old building and classrooms with an air circulation cooling system (n=10) in a new building were selected for field measurement. CO_2 measurements were conducted by field technicians using Vaisala GMW20 series CO_2 sensors (Vaisala Inc., Finland) coupled with Hobo data logger (Hobo, USA). There sets of equipment were placed in different indoor environments for at least 12 hours, approximately 1.5 m high from the floor and away from windows, doors and fans, and at least 1 m from students. Indoor temperature, relative humidity (RH), and CO_2 concentration were continuous monitored while outdoor parameters combined with on-site climate conditions were recorded simultaneously.

One set of equipment was also placed out of the windows about 1.5 m high from the indoor floor. The occupants were interviewed at the end of investigation period about the number of people indoor. Participants were asked to record their daily activities in a time/location diary and field technicians administered a recall questionnaire at the end of field investigation to each participant.

Average air exchange rates were calculated using CO_2 as the tracer gas by decay method based on box model assumptions when indoor environments were not occupied 8. Air exchange rates were obtained under static conditions (no occupants) to reflect conservative, but not worst-case ventilation, while, in reality, the real ventilation rate may be lower because of the activities of occupants, for example, keeping windows and doors closed when staying indoor.

Questionnaire

Questionnaire concerning time-activity patterns, students' judgment about IAE on campus and comfortlessness possibly relative to bad ventilation was distributed to 1000 undergraduate students with different academic majors. In addition, 40 students (20 males and 20 females) were recruited to assess the relationship between indoor environment quality and performance. The subjects were randomly selected from 8 dormitories; they were undergraduate students from Department of Environmental Science aged between 18 and 22 years and in good health; none of them was smokers. Self-reported evaluation of how ventilation rate affected comfortlessness was collected. It was expected that respondents' performance data is less influenced by the subjects' expectations or biases and, it can be considered more reliable.

Because the experiments were performed only once a week, the whole experiment lasted for one and a half months. Indoor environment conditions (dormitories and classroom) were recorded when they stayed indoors while subjective experience was investigated in weekdays in a week. Room characteristics, ventilation methods and numbers of occupants and their activities were recorded.

RESULTS AND DISCUSSION

Time-activity patterns of undergraduate students

In weekdays, students spent 8.68% of their time outdoors while 91.32% of a day indoors including 42.66% in dorms, 48.66% in the classrooms and reading rooms. In weekends, the time students stayed indoor (88.8%) was less than it spent in weekdays, but significantly longer than the time spent outdoor (11.23%). Longer time was also spent in dorms (72.8%). When studying by themselves in scheduled time, 74.4% students selected classrooms, 14.7% students in library (mainly in the reading room), and 7.8% students in dorms. Obviously, undergraduate students surveyed spent over 90% of their time indoors.

Indoor CO_2 concentration, temperature, and relative humidity

Table 1 presents descriptive statistics for indoor carbon dioxide concentration levels, temperature, and relative humidity in different indoor environments.

Table 1 Indoor and outdoor CO_2 concentration, temperature and relative humidity

	CO_2 concentration (ppm)		Temperature (°C)		Relative humidity	
	Average	Range	Average	Range	Average	Range
Dorms	1278±702	612-2466	26.8±5.4	20.1-38.3	68.1±17.8%	37.5-90.0%
Classroom	2580±1854	812-5346	24.0±3.1	20.1-30.4	65.1±15.5%	38.6-76.5%
Reading rooms	767±164	629-994	22.1±3.9	18.9-28.2	60.2±19.4%	30.0-90.0%
Meeting rooms	2394±1106	762-3946	23.5±2.7	19.2-29.4	60.4±14.3%	38.5-75.6%
Outdoor	501±21	478-534	25.2±5.7	18.0-39.0	60.0±21.0%	35.0-90.0%

In this study, almost half of the CO_2 concentrations were above 1000 ppm. In classroom and meeting rooms, over 80% were above 2000 ppm. If the measured CO_2 concentrations had been maximum or steady-state values, a substantially larger proportion would be expected to exceed 1000 ppm. Thus, it is likely more than half of the classrooms in this study had ventilation rates less than specified in current minimum ventilation standards.

Air exchange rates

Based on the measured CO_2 data, we computed the air exchange rates using a decay model. Dormitories and reading rooms monitored were ceiling fans ventilated with opened windows/doors. Meeting rooms were without any ventilation appliance but air-conditioning. Because the CO_2 measurements in this study were short-term (at most 2 days), and AER calculation was made on weekdays at variable times of occupied period, they should be considered only rough surrogates for the long-term average indoor ventilation rates, and may be much lower in some spaces.

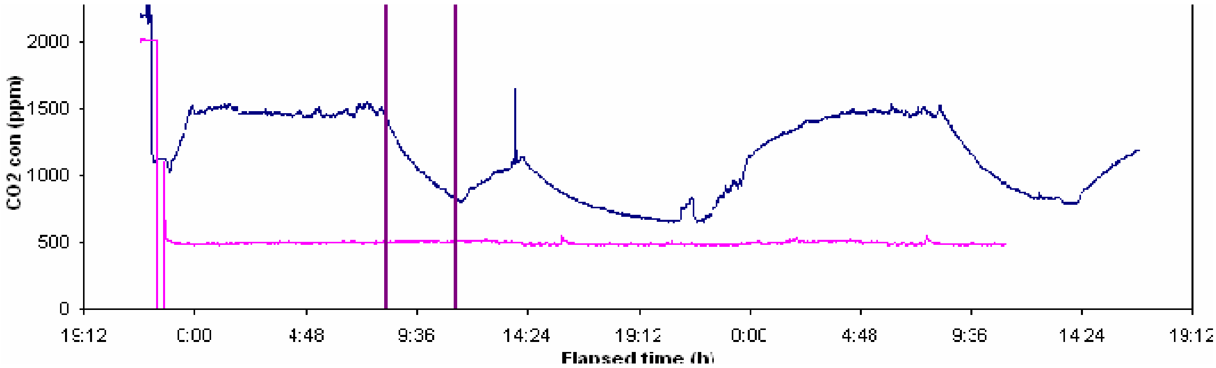

Figure 1a CO_2 concentration monitored in a room where two students lived in.

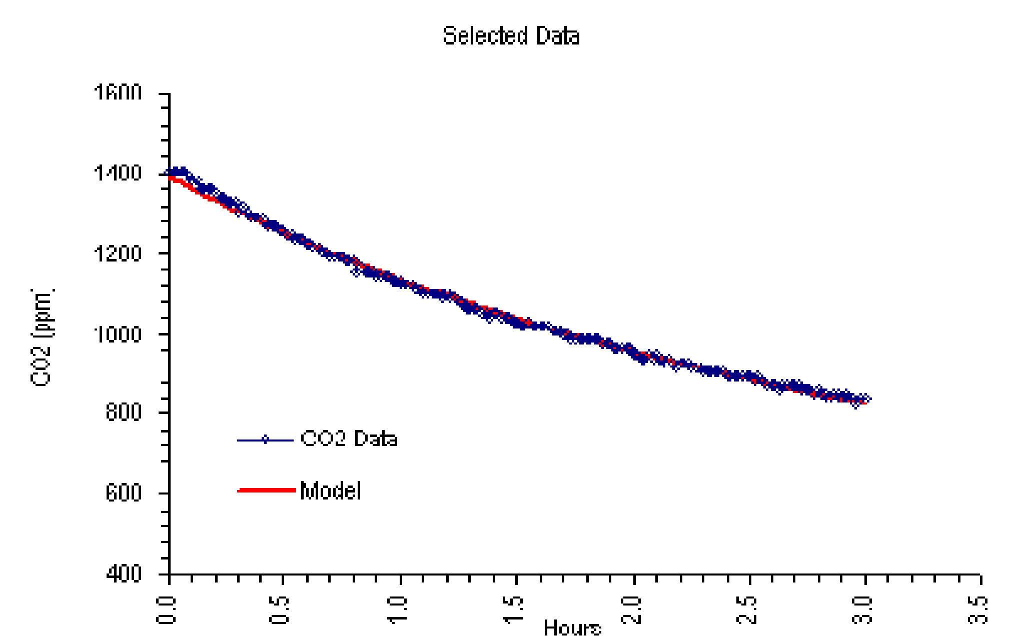

Figure 1b Air exchange rate calculation for selected duration. 1.2, 2.9, 1.6, 1.1 and 0.7 hr^{-1} in dormitories, reading rooms, old classrooms, new classrooms and conference rooms

Table 2 Air exchange rates in different indoor environments on campus

	Air exchange rates (h^{-1})		
	n	Average	Range
Dorms	20	1.22±0.67	0.30 - 2.84
Classroom I	10	1.37±0.68	0.68 - 2.30
Classroom II	10	1.10±0.55	0.15 - 2.16
Reading rooms	5	1.91±0.87	1.10 - 3.11
Meeting rooms	5	0.73±0.48	0.15 - 1.35

Classroom I: air-conditioned classrooms in an old building
Classroom II: classrooms with an air circulation cooling system in a new building

In dormitories, factors such as outdoor climate conditions, deficiency of building design, room usage and living habits of students, were considered to be essential on ventilation and indoor CO_2 concentration. Under same indoor conditions (windows and doors were fully or partially opened, no occupants), the higher the floors were (less than 6 floors), the lower the ventilation achieved in the dorms ($P<0.01$). It illustrated the negative relationship between increase of floor height and outdoor climate conditions, mainly because the climate conditions were uneven and variational vertically.

In classrooms I, ventilations were always low for windows and doors were closed, especially for air-conditionings began to work from the end of June. While weather was cool and ceiling fans were turned on, ventilation conditions became much better. Although there wasn't significant difference between two types of classroom with different ventilation devices, the air circulation cooling system newly employed was not of great help for improving ventilation in classrooms. Available data always indicated many classrooms with ventilation rates below the code minimum or with CO_2 concentrations above 1000 ppm [9, 10].

All the meeting rooms investigated were located in new buildings. Due to the deficiency in design, most meeting rooms had few windows and ventilators for ventilation. Together with occupants' activities, such as keeping the doors closed, indoor CO_2 accumulated quickly under bad ventilation contions.

Judgments of indoor environment quality and performance

Analysis results indicated that the best ventilation was achieved in reading rooms (AERs>1.0 hr^{-1} and [CO_2]<1000ppm), and the worst situation (AERs<1.0hr^{-1} and [CO_2]>1600ppm) was found in dormitories, classrooms and meeting rooms, which was accordant with occupants' more complaint.

Although the indoor RH and temperature were comfortable (40~70%, 22~28°C), students in classrooms easily felt absent of mind, out of breath, smelly, even fidget and headache, especially for the female students. CO_2 concentration is regarded as one key factor. In former researches, higher CO_2 concentrations have been associated with increased frequency of health symptoms and increased absence in studies of office workers [11, 12]. This phenomenon was not obvious in dormitory. A significant impact of temperature and humidity on the perception of air quality was also found. The air was perceived as less acceptable with increasing temperature and humidity, in particular in dormitories.

For all the indoor environments, ventilation conditions had negative relationship with unpleasant smell ($P<0.05$). Female students' judgment on indoor environment quality showed positive relationship with air exchange rates ($P<0.01$) while those of male students didn't. Female students were also more sensitive to bad ventilation conditions and showed aggravating symptoms.

CONCLUSIONS

This study disclosed that high CO_2 concentrations in indoor environments on campus, which indicated that classroom ventilation rates were often below the minimum rates specified in codes. CO_2 concentration is regarded as one key factor for students' judgments on IEQ. In case of a constant pollution level indoors, the perceived indoor environment quality decreases with increasing indoor temperature and humidity. Temperature and humidity also have a strong and significant impact on the perception of indoor environment quality in dormitories.

ACKNOWLEDGEMENT

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Measuring Air Exchanges Rates Using Continuous CO_2 Sensors

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SUMMARY

Measuring AERs in an effective, real-time, easy and low-cost way is still a challenge, especially in China where rooms were usually naturally ventilated but not with the Heating, Ventilating and Air Conditioning (HVAC) system. A new AER monitoring method using continuous CO_2 sensor was validated through both laboratory experiments and field studies. Controlled laboratory simulation tests were conducted in a 1 m^3 environmental chamber at different AERs (0.1~10.0 hr^{-1}). AERs were determined using steady-state method and decay method based on box model assumptions. Field tests were conducted in apartments, offices, classrooms, dormitories and meeting rooms in during 3-5 weekdays with four sets of sensors coupled with data loggers. Statistic results indicated that good laboratory performance was achieved. In spite of limitations, this new method, along with mini-sized and easily operational instrumentations which provide sufficient continuous data, demonstrated an effective way to measure AERs in various indoor environments.

INTRODUCTION

Ventilation (Air exchange rates, AERs) is a critical ventilation parameter that affects thermal comfort and air contaminant exposure in civil buildings and other indoor environments. The concentrations of indoor pollutants mainly depend on the ventilation rates, source strength, and the emission rates. For many indoor-generated air pollutants, ventilation with outside air is usually the dominant removal process.

Many methods have been established to measure air exchange rates. Some studies reported AERs using perfluorocarbon tracer (PFT) compounds [1-4]. The most commonly used method is sulfur hexafluoride (SF6) decay tests which has been reported in many studies [5, 6] when commercial availability of instruments for SF6 tests has also made possible AER measurements. However, the instrumentation for these methods are usually big sized and expensive.

Although the Chinese national Indoor Air Quality Standard (GB18883 - 2002) regulates that the new air quantity for occupants is 30 m^3 per hour per person, the requirement of new air is not strictly meet based on experiences. The ventilation rate or the air exchange rate was not measured at all when indoor air quality was investigated because of limits of measurement equipment available and convenient for tracer gas such as CO_2, CO, CH4, SF6 and so on. Because the cost of conventional PFT technique in China is high (at least $1,500 only for a bottle of SF6 gas), and its analysis equipment, the GC-ECD is not so popular in China, a convenient, accurate, and low-cost measurement method for ventilation or AERs needs to be set up and employed.

Carbon dioxide (CO_2), although a ubiquitous compound in air, is one of the bioeffluents. Occupants are usually the main indoor source of CO_2, resulting in an increase of indoor CO_2 concentrations compared with outdoor levels. In this study, CO_2 was employed as a tracer gas for determining AERs under certain circumstances satisfied for a box model (described in details in the method part). Commercially available CO_2 sensor have the ability to measure real-time CO_2 concentrations, and therefore semi real-time AERs can be determined using both steady-state and decay methods. The instruments are usually small sized, quiet, easy to use and relatively cheap.

The objective of this study is to validate a method for measuring AERs using continuous CO_2 sensor in both controlled environmental chamber and real settings.

METHOD

Box model for determining AERs

A compartment can be simplified as a box with a specified volume V (m³) and a box model can be established assuming that the inside air is fully mixed (Figure 1). The carbon dioxide in a compartment results from outdoor concentration and exhaled carbon dioxide.

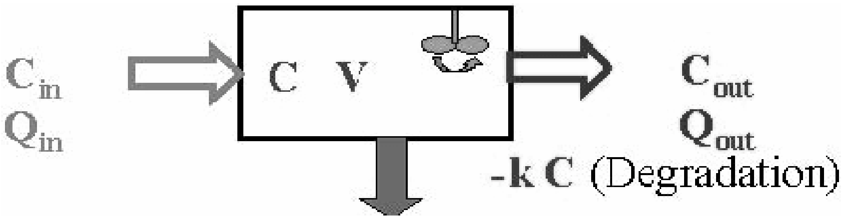

Figure 1. Schematic diagram of a fully mixed box model

The rate of change in concentration of the monitored gas depends on the concentration of the in-flowing air, the concentration of the out-flowing air and the internal generation rate of the gas in question. The time derivative of the monitored concentration is given by:

For a contaminant of interest, the governing equation for this model is:

$$V\, dC/dt = Q_{in}C_{in} - Q_{out}C_{out} + S - kC \qquad (1)$$

i.e. dC/dt = inflows -outflows + sources - degradation where C_{in}, C and C_{out} are concentrations of the contaminant in the inflow, indoor air and outflow (mg/m³), $Q_{in} = Q_{out} = Q$ are air flows into/out of the building (m³/h), S is the indoor emission source of the contaminant (mg/h) and k is the first-order degradation constant (m³/h). Solving (1) by integration yields the final equation, which describes the generation and decay of CO_2 as a function of time:

$$C(t) = W/(ß\, V)[1 - \exp(-ß\, t)] + C0\exp(-ß\, t) \qquad (2)$$

Where:
C_o = concentration at t = 0, $W = Q_{in}C_{in} + S$ and ß = AER + k.
C(t) = internal concentration of carbon dioxide at time t (ppm)
For a conservative contaminant, i.e. k = 0, AER can be determined in two ways:

(1) Steady-state method
When t -> infinity:

$$AER = W/(CV) \qquad (3)$$

(2) Decay method
When there is no source, i.e. S = 0, AER can be resolved from the equation:

$$C = QCin/(ß\ V)[1\text{-}exp\ (\text{-}ß\ t)] + C0exp\ (\text{-}ß\ t) \qquad (4)$$

Environmental chamber simulation

The laboratory simulation is evaluated in a simplified model chamber using a tracer gas technique of CO_2 gas injected into a supply duct. Ventilation systems, which consist of a air supply pump, a CO_2 gas generator, a CO_2 gas analyzer and a small stainless steel environmental simulating chamber (1m×1m×1m, 1.5 mm thick), are set up to evaluate the performance of this box model for determining AER using CO_2 concentrations at steady-state and decay processes. A fan is installed inside the chamber to provide the fully mixing condition. The CO_2 level in the environmental chamber was measured at a height of 3 ft above the floor level using a Vaisala GMW20 series CO_2 sensor (Vaisala Oyj, Helsinki, Finland). This sensor has an accuracy better than ±1% full scale ±1.5% of the reading with a repeatability better than ±1% of full scale and temperature dependence of , 0.1% full scale/°C. An air pump (Yinhu Company, China) is installed inside the chamber and emits CO_2 at a constant rate. One-channel data loggers equipped with temperature and relative humidity sensors Hobo sensors (Hobo H08004-02, Onset Computer Corporation, USA) are used in conjunction with the CO_2 sensor to capture the CO_2 concentration every 10 seconds. Three are placed inside and one outside of the chamber to record ambient temperature and relative humidity simultaneously. Data from the data acquisition system were periodically downloaded via a cable to a PC for processing.

An air pump ((Xinma Electronic Enterprise Co., Ltd., China)) is installed at the downstream of the chamber to control the AER. Dupliate simulations are performed at 10 AERs, i.e. 0.1, 0.6, 1.0, 2.0, 3.2, 4.0, 5.2, 6.0, 7.2 and 10.0 hr^{-1}, respectively. In each test, the chamber temperature and relative humidity are equilibrated with the room temperature at 20 0C and 50%, respectively. The CO_2 pump is running for few minutes. Then the CO_2 pump is shut down but the AER is kept unchanged for about 10 hr.

Calibration of the carbon dioxide sensors was weekly accomplished using zero gas (N2, purity=99.999%) and a series of carbon dioxide standard gas (297, 632, 914, 1212, and 1501 ppm CO_2). Instruments were also cross-compared during short-term (about 15 min) outdoor air CO_2 measurements at each outdoor location distant from potential CO_2 sources. Calibration of the temperature and humidity sensors had been performed by the manufacturer immediately prior to this research.

Field study

The field study will be conducted in 2 residential apartments, 1 office building, and 1 school. In each apartment, CO_2 sensors are placed in each room for 3-5 weekdays. In the office building and school building, CO_2 sensors are placed in selected offices or classrooms. There sets of equipment were placed in different indoor environments for at least 12 hours, approximately 1.5 m high from the floor and away from windows, doors and fans, and at least 1 m from occupants. One set of equipment was also placed out of the windows about 1.5 m high from the indoor floor. A questionnaire and walkthrough survey is completed for each location examining basic building characteristics, occupancy and other information potentially related to AERs.

A daily journal was kept to record activities that might affect the air change rate, for example, when windows were open or closed, and how much the actual width/area of each opening was. The normal practice of the occupants for indoor ventilation was to open windows/doors and fans.

RESULTS

Laboratory simulation

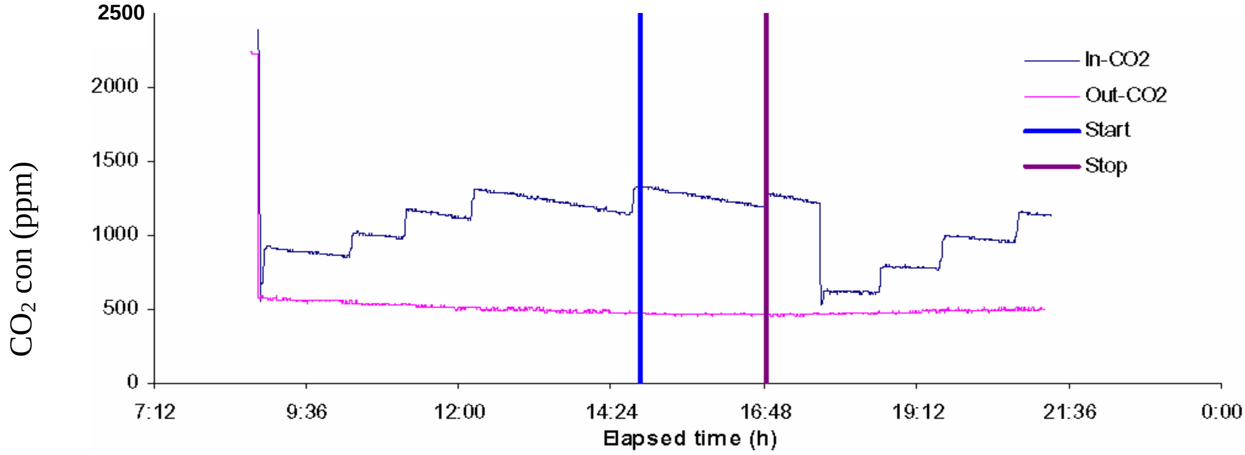

Figure 2a Decay curves of CO_2 under AER=0.1 conditions in chamber

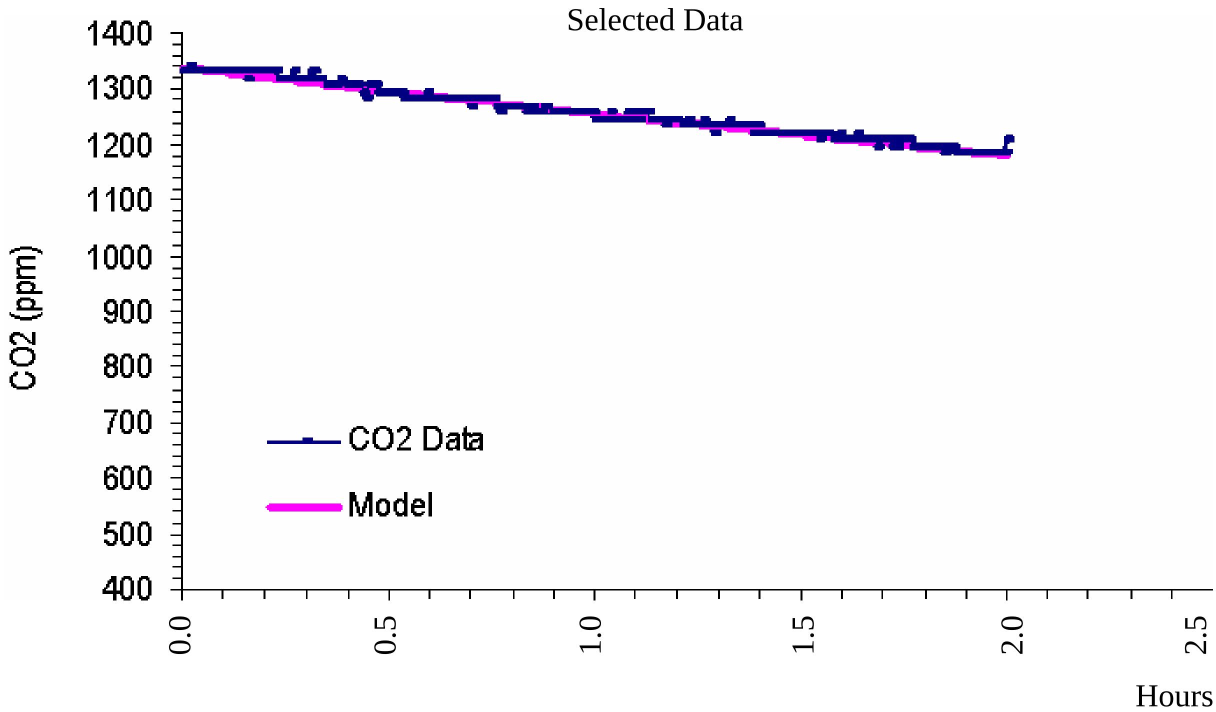

Figure 2b Data simulation, $AER_{nominal}$=0.1, $AER_{simulation}$=0.1 (slope).

Table 1 Simulated AERs and actual AERs using steady-state and decay methods.

	Steady State Method		Decay Method	
Nominal AERs (hr^{-1})	Simulated AER(hr^{-1})	Duplicate precision (%)	Simulated AER(hr^{-1})	Duplicate precision (%)
0.10	0.12	9.2	0.11	9.2
0.60	0.65	8.8	0.63	7.5
1.0	1.2	4.7	1.1	3.3
2.0	2.2	4.2	2.0	3.4
3.2	3.5	3.9	3.3	3.1
4.0	4.6	3.8	4.3	3.0
5.2	5.6	2.8	5.7	2.0
6.0	6.3	2.7	6.0	2.5
7.2	7.9	2.5	7.6	2.5
10.0	11.2	2.5	10.4	2.4

Field study

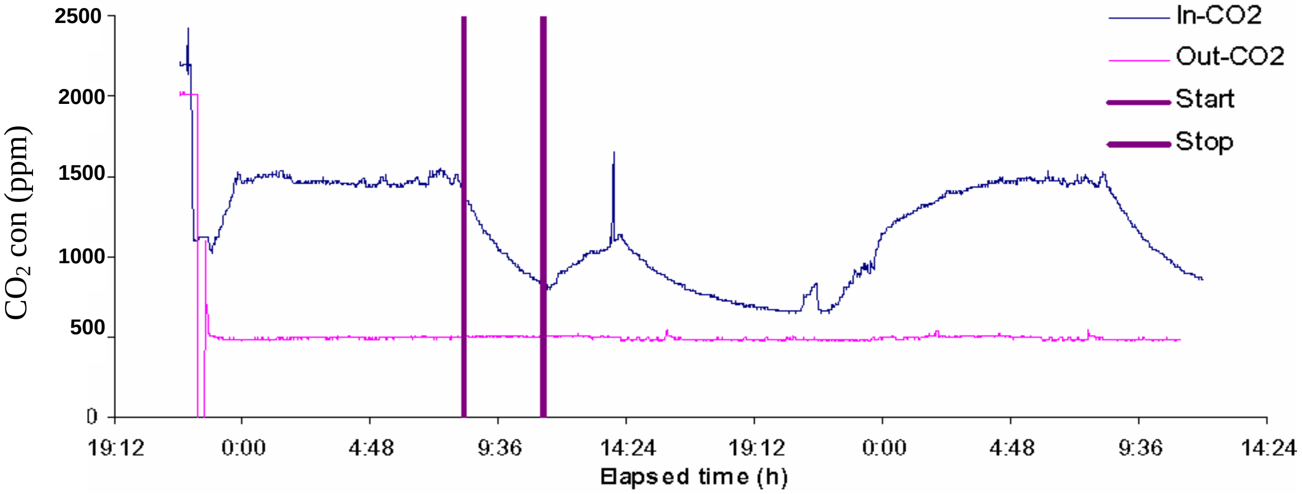

Figure 3a Decay curves of CO_2 in a apartment.

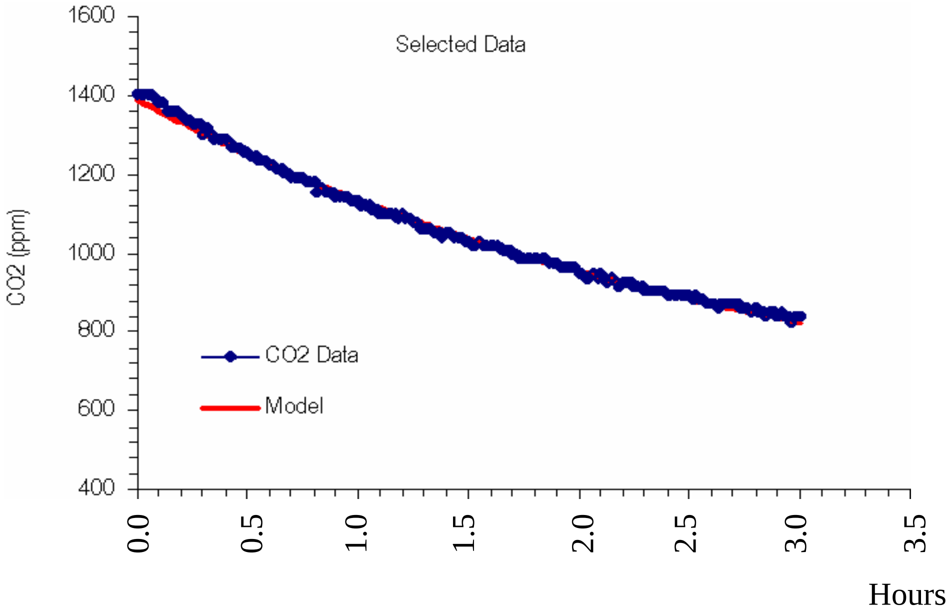

Figure 3b Simulation for selected data, $AER_{simulation}$=0.34.

DISCUSSION

Advantages and applications

CO_2 can act as an indicator of ventilation efficiency, showing whether the supply of outside air is sufficient to dilute air contaminants. As examples, to maintain odor-free environments acceptable contaminant levels, China ventilation standards specify a CO_2 ceiling of 800 ppm higher than outdoor levels. Besides bioeffluents, CO_2 has been used or suggested as a surrogate for carbon monoxide (CO) and other combustion products, and radon.

In former researches, it was always found that obtaining the required mixing of CO_2 and air is often difficult. Multipoint injection of tracer gas into the airstream is frequently necessary and multipoint measurements from different locations within the airstream are essential to confirm mixing7. In our study, because of high occupancy, indoor activities/movement of occupants, as well as the full application of ceiling fans, adequate full-mixing was obtained. The AER in each was constant during the sampling period in the light of small variation (COVs < 20%). CO_2 levels were similar within the same building (COVs < 20%), indicating good mixing conditions.

Statistic results indicated that good laboratory performance was achieved: duplicate precision was within 10%, differences between the two methods were within 10%, and the measured AERs were 90 -120% of the real AERs. Fully mixing conditions were observed in most indoor environments with fan operation.

The AERs measured in selected indoor environments were comparable to those reported in literature. Average AERs were 1.4, 1.7, 1.1, 1.0 and 0.6 hr^{-1} in apartments, offices, classrooms, dormitories and meeting rooms, respectively. Two day-long experiments, although air exchange rates higher than 2 h^{-1} were achieved for a brief time, the average over 12 h was never more than $1h^{-1}$

Figure 3 shows that the CO_2 level returned to about 800 ppm in about 3 hours from the time the occupants left the house while the initial indoor level was 1, 400 ppm. This defines the complete air change cycle of the apartment. Thus, a building that takes a longer time for the CO_2 to decay is tighter than that which takes a shorter time.

Limitations

Although CO_2 emission rates have been very well characterized and average 18 L/hr-person or 35.3 g/hr-person for people engaged in typical office work, the main problem with the method is that it can not be well employed when the spaces were occupied or it requires accurate information about the rate of CO_2 production within the space. This is not a problem during the unoccupied period, when the production rate will be zero.

ACHNOWLEDGEMENT

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Prediction of Annual Energy Consumption of Office Building Taking Account of Improved Ventilation Effectiveness

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SUMMARY

In this study, energy consumptions associated with the change in outdoor air flow rate are analyzed based on numerical methods. By employing the concept of ventilation effectiveness, possible reductions in the outdoor air load through reductions in the amount of outdoor air flow rate has been estimated. In addition, the effects of optimizing the capacity of HVAC equipment on annual energy consumption and annual CO_2 emissions have been analyzed.

INTRODUCTION

Against the background of a growing concern about global environmental problems, reductions in building energy consumption are required. Reducing heat load, which account for a majority of energy consumption in the operation of a building, is particularly important. One of the measures designed to conserve energy in buildings and facilities is the lowering of outdoor air flow rate (Here after O.A.) to reduce energy consumption.

In this study, energy consumptions associated with the change in O.A. are analyzed based on numerical methods. By employing the concept of ventilation effectiveness, possible reductions in the outdoor air load through reductions in the amount of O.A. has been estimated. In addition, the effects of optimizing the capacity of HVAC equipment on annual energy consumption and annual CO_2 emissions have been analyzed.

VENTILATION EFFECTIVENESS

As shown in Fig. 1, this analysis defines the space up to 1.8 m from the floor as the occupied zone. In this study, the normalized concentration in an occupied zone (C_n) is adopted as the index for ventilation effectiveness.[1],[2] The C_n is an index that represents the extent of incomplete mixing within the room and is defined as the ratio of the average concentration in an occupied zone (C_a) to the perfect mixing concentration (C_p) under conditions in which pollutants are uniformly generated in a room, as shown in equation (1) in Fig.2, where C_o denotes the pollutant concentration in outdoor air.

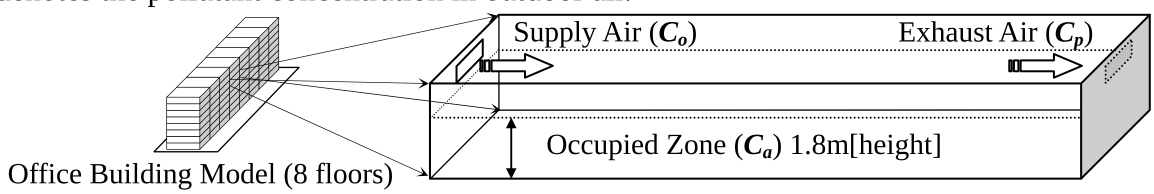


Figure 1 Definition of Occupied Zone in a Room

The equation to estimate the amount of O.A. flow rate using the C_n index is shown as equation (2) in Fig.2.

(1) Investigation of Ventilation Effectiveness into HVAC design
The index for evaluating the ventilation effectiveness

$$C_n = \frac{C_a - C_o}{C_p - C_o} \qquad (1)$$

C_n : Normalized Concentration in an Occupied zone
C_a : Average pollutant concentration in the occupied zone
C_o : Outdoor Air concentration
C_p : Completely-mixed concentration

$$Q = Q_p \times C_n \qquad (2)$$

Q_p : O.A. which defines the perfect mixing concentration
Q : Final O.A.

Q_p = 30 m^3/h/person × 16 persons in the room = 480 m^3/h (in case of this study)
C_n = 0.9 ($C_a < C_p$) Q = 432m^3/h (in case of decreased O.A. by good Ventilation Effectiveness)
C_n = 1.0 ($C_a = C_p$) Q = 480m^3/h
C_n = 1.1 ($C_a > C_p$) Q = 528m^3/h (in case of increased O.A. by bad Ventilation Effectiveness)
Q = Optimization of Outdoor Air Flow Rate by C_n index

(2) Set up Conditions for office Building Model (See Fig. 1)

(3) Set up conditions for outdoor climate
3-1 Sapporo (Cold Climate Area : Temp. - 9.4 - + 29.0˚C , R.H. 66 – 70 %)
3-2 Tokyo (Standard Area : Temp. + 0.6 - + 32.6˚C , R.H. 36 – 64 %)
3-3 Fukuoka (Warm Climate Area : Temp. + 1.5 - + 33.3˚C , R.H. 56 – 63 %)

(4) Optimization of HVAC System in Target Building

(5) Analysis of Energy Consumption, CO_2 Emission

Figure 2 Flowchart of analysis of energy consumption

Here, Q_p is the amount of O.A. when perfectly mixed conditions are assumed. The amount of O.A. when ventilation effectiveness is taken into consideration in the equation represents the amount of O.A. needed to maintain the average concentration in an occupied zone (C_a) below the reference concentration. The value of C_n varies with the placement of outlets and inlets in a room, conditions of supplied air flow, and other factors. The European Standards put C_n value within the range from 0.7 to 5.0 [1], and the authors' previous studies based on CFD an alysis report that the value comes within the range from 0.9 to 1.2 [2].
This paper sets the value of C_n at 0.9, 1.0, and 1.1, and looks into the effects of changes in the outdoor air load on the optimization of the capacity of air-conditioning heat-source equipment as well as on annual energy consumption.

ANALYSIS OF ANNUAL ENERGY CONSUMPTION

Fig. 2 shows the flowchart of annual energy consumption analysis. In general, the outdoor air load accounts for a substantial portion (30% to 40%) of the loads associated with HVAC equipment in office buildings. Accordingly, optimizing the amount of O.A. is an effective energy-saving measure. In the present analysis, an office space with personnel density of 0.2 people / m^2 as shown in Fig.1 is assumed and the amount of outdoor air required per person is set at 30 m^3 /h /person. As a result, the amount of O.A. in the standard office space when perfectly mixed conditions are assumed works out to 480 m^3/h (with 16 persons present in a room).

Table 1 Conditions for the air-conditioning system

Supplied Air Condition (Difference Of temp.)	Target Temp. [˚C]	Heat Load [W/m^2]	Air Change Rate [times / h]	Supply Air Temp. [˚C]	Supply Air Flow Rate [m^3/h]
Cooling (–10˚C)	26	+114	3.65	16	2786
Heating (+10˚C)	22	–108	3.65	32	2639

Table 1 shows conditions of supply inlet. The supplied air flow rate is determined by the heat balance based on the target air temperature and is set at 2,786 m^3/h during cooling and 2,639 m^3/h during heating (with air change rate of 3.65 times/h). When ventilation effectiveness is applied, the amount of O.A. is increased or decreased according to equation (2) and if C_n = 0.9, the amount of O.A. comes to 432 m^3/h and if C_n = 1.1, it works out to 528 m^3/h. With respect to climate conditions, Sapporo area representing cold regions and Fukuoka area representing warm regions were covered in addition to Tokyo area. Fig. 3 shows data on the daily average temperature and humidity on a monthly basis in the areas covered in the analysis. Table 2 shows the cases analyzed. A total of 18 cases are established as different combinations of the climate conditions, types of HVAC equipment, and C_n values.

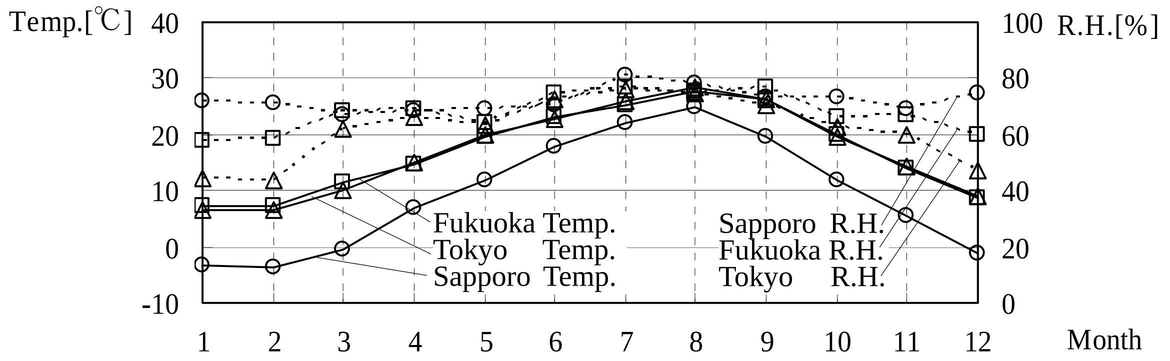


Figure 3 Weather Data of analysis target area

Table 2 Analysis Cases (H.E.=HVAC equipment)

Climate Area	HVAC equipment	C_n
Sapporo (Cold area)	Direct fired double effect absorption water chiller boiler (= H.E.(A))	0.9 [432 m^3/h] 1.0 [480 m^3/h] 1.1 [528 m^3/h]
Tokyo(standard area)		
Fukuoka(Warm area)		
Sapporo (Cold area)	Engine driven heat pump(=H.E.(B))	
Tokyo(standard area)	Air cooled packaged air conditioner(= H.E.(C))	
Fukuoka(Warm area)		

C_n=Ventilation Effectiveness, [ ] = outdoor air supply rate(=O.A.)

Table 3 shows specification of the HVAC equipment. As for HVAC equipment, HVAC equipment (A) (an absorption water heater/cooler unit is adopted (Here after H.E.(A))) is used as a base, and HVAC equipment (B) (an engine heat pump (Here after H.E.(B))) and HVAC equipment (C) (an air-cooled, packaged air conditioner (Here after H.E.(C))) are chosen depending on regional characteristics when analyzing data.

Table 3 HVAC equipment specifications

Type of H.E.	The ratio of Heating capacity to Cooling capacity[-]	The Ratio of Rated Output[-]	
		(supply child water)	(supply hot water)
H.E. (A)	0.84	1.0(=standard)	1.0(=standard)
H.E. (B)	1.2	1.05	1.44
H.E. (C)	1.12	0.26	0.34

Table 4 HVAC equipment Adopted Rate

Area	Adopted Rate [%]	
	1st	2nd
Sapporo (Cold Area)	H.E.(B) 57.1	H.E.(A) 14.3
Tokyo (Standard Area)	H.E.(C) 31.9	H.E.(A) 28.6
Fukuoka (Warm Area)	H.E.(A) 38.0	H.E.(C) 32.0

Table 4 shows adopted rate of each HVAC equipment in each area in Japan.

Building Model and its Conditions for Analysis of Annul Energy Consumption

Table 5 gives basic data on the building that is covered in the analysis. Based on the previously published paper [2], the covered building is assumed to be a central HVAC system-installed eight-storied, 4,000 m^2 office building that has 48 office modules each with space volume of 6.4 × 12.8 × 2.7 m. It is a typical medium-sized building found in Japan. Table 6 shows numerical conditions for annual heat loads analysis. Heat load varies from region to region and from season to season, and it impacts the energy consumption of HVAC equipment. In this analysis, heat loads for 12 months and for every 24 hours have been calculated using commercial software (MICRO-PEAK). Based on the results of heat loads calculated and calculation conditions given in Table 7, annual energy consumption and CO_2 emissions have been determined using commercial software for calculation of the energy consumption of heat sources in buildings (E-pass plan).

Table 5 Target building Size and HVAC system

Target Room Size	6.4 $_{(Wide)}$ × 12.8 $_{(Depth)}$ × 2.7 $_{(Height)}$
Number of Floors, Target Rooms	8 floors , 48 rooms
Building Size	Total floor area: 4000 m^2
Air-conditioning System	Constant volume single duct system

Table 6 Conditions for Annual Heat Load Analysis

Building Data	Area	(See Table2, Area) Analysis without thermal load from infiltration leave and overhang
	Over-all heat transfer coefficient [W/m^2 K]	Out wall = 0.22 Inner wall = 0.44 Window = 5.9
Schedule Data	Cooling / Heating	Jun.-Sep. / Dec.-Mar.
	Mode of operation [%]	8 am - 9 am = 50 12 pm - 13 pm = 50 9 am - 12 pm = 100 13 pm - 18 pm = 100 ,

Table 7 Conditions of annual office building consumption

Electric power	Commercial power service (Special power rate system for commercial use)
Gas	(Special power rate system for commercial use)
Energy Consumption Rate	Electricity:10.26 MJ/kW CO_2 Emission Rate:0.378 kg-CO_2 /kW Gas :46.5 MJ/m^3 CO_2 Emission Rate:2.11 kg-CO_2/m^3

RESULTS AND DISCUSSION

Results of Peak Load

Fig. 4-1 and 4-2 shows reductions in maximum cooling and heating loads by regions and by amounts of O.A..

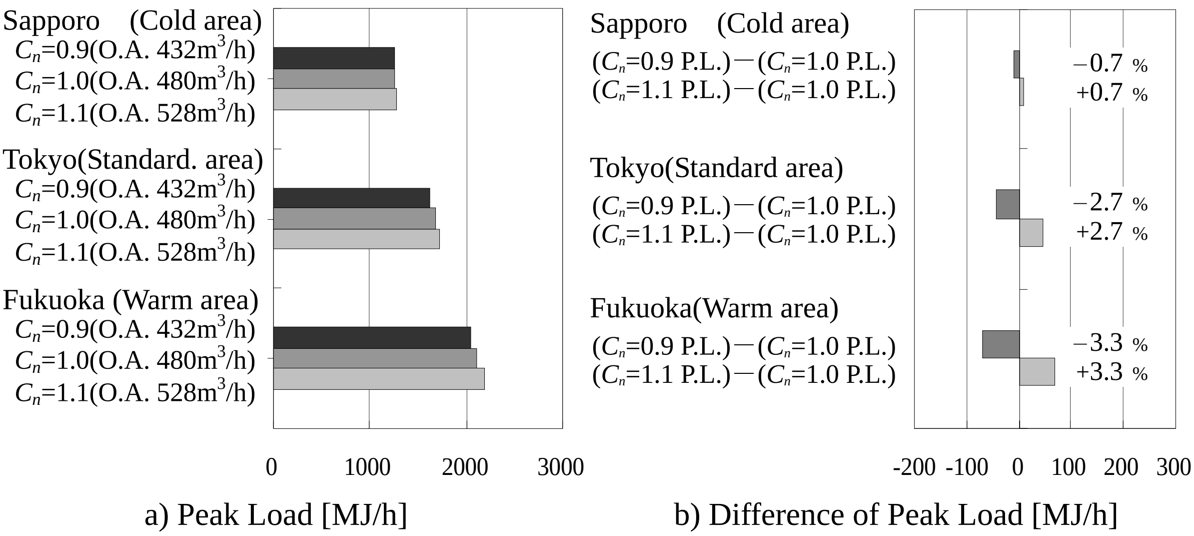


Figure 4 - 1 Cooling Load

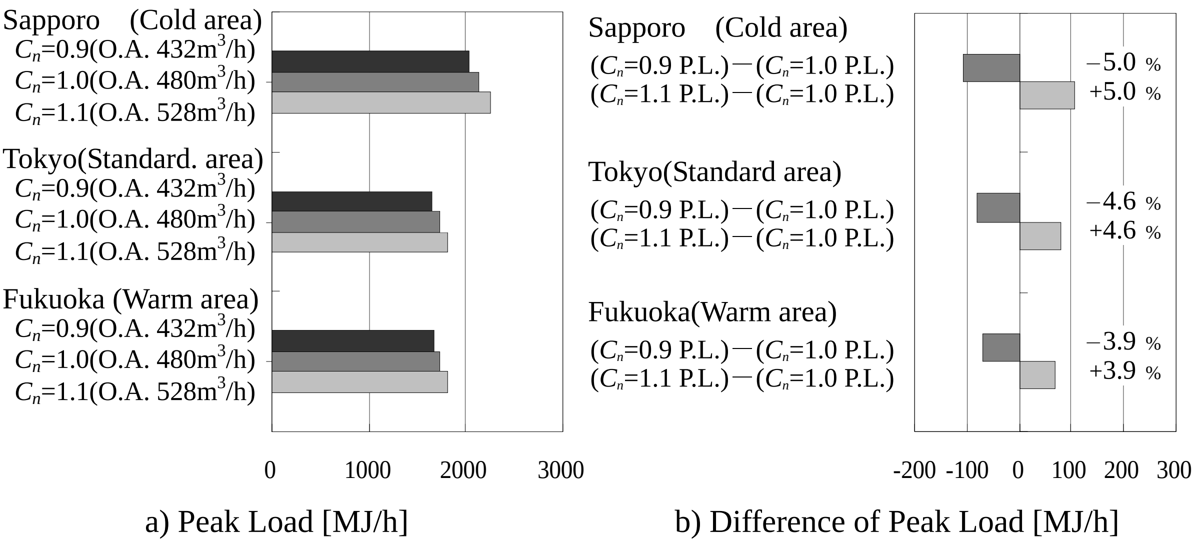
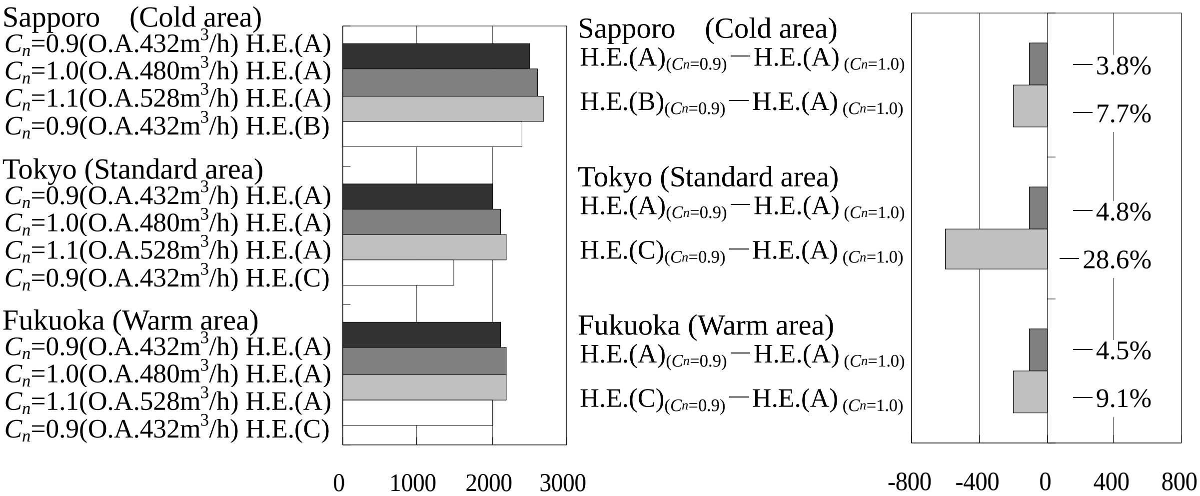


Figure 4 - 2 Heating Load

In the Tokyo area, an improvement of 0.1 in ventilation effectiveness C_n (= 0.9) leads to reductions in maximum heating and cooling loads of 45 MJ/h (a 2.7% reduction if the standard is C_n = 1.0) during cooling and 80 MJ/h (a 4.6% reduction when C_n = 1.0) during heating. By the same improvement, in the Sapporo area, it resulted in reductions of 9 MJ/h (0.7%) in the cooling load and 108 MJ/h (5.0%) in the heating load, and in the Fukuoka area, it resulted in reductions of 69 MJ/h (3.3%) in the cooling load and 68 MJ/h (3.9%) in the heating load.Taken together, a reduction in the O.A. has brought about the largest reduction in the heating load in Sapporo, followed by Tokyo. Presumably there is a large difference between room and the outdoor air temperature during heating in Sapporo, a cold area. Likewise, this is similar reason why a difference between room and outdoor air temperature in Tokyo.

Results of Equipment Capacity

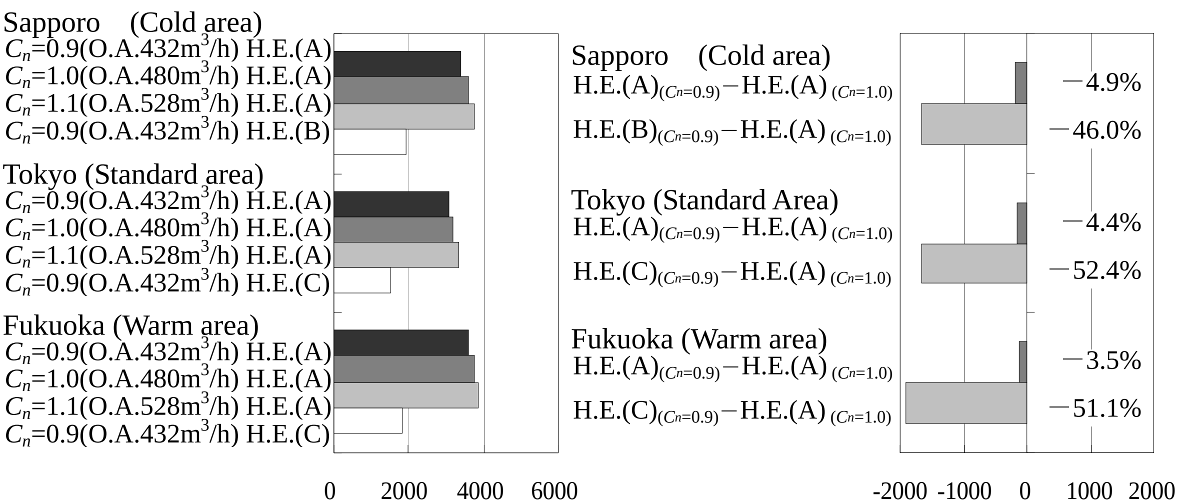


a) Equipment Capacity [MJ/h] b) Difference of Equipment Capacity [MJ/h]

Figure 5 Prediction Results of Equipment Capacity

Fig. 5 shows changes in the capacity of HVAC equipment. Assuming that the same H.E.(A) is adopted in all the areas covered, an improvement of 0.1 in the ventilation effectiveness C_n (=0.9) led to a reduction of 100 MJ/h of equipment capacity in all climate areas. Also shown in Fig. 5 is the effect of the selecting HVAC equipment that is commonly used in each area, in terms of changes from the basic case (C_n =1.0), which adopts H.E.(A). An improvement of 0.1 in C_n (=0.9) reduced 200MJ/h of equipment capacity (a 7.7% reduction when C_n = 1.0) in Sapporo area with H.E.(C), and reduced 600MJ/h of equipment capacity (a 28.6% reduction when C_n = 1.0) in Tokyo area with H.E.(C),and reduced 200MJ/h of equipment capacity (a 9.1% reduction when C_n = 1.0) in Fukuoka area with H.E.(C). H.E.(C) has hardly difference between heating capacity and cooling capacity, and the rated output is 1/4 times smaller than the H.E.(A). Therefore, in Tokyo area with 2% larger heating load than cooling load, equipment capacity became large reduction. Also in Fukuoka area with cooling load 20% larger than heating load, equipment capacity of H.E.(C) by small rated output became smaller than H.E.(A).

Results of Annual Energy Consumption

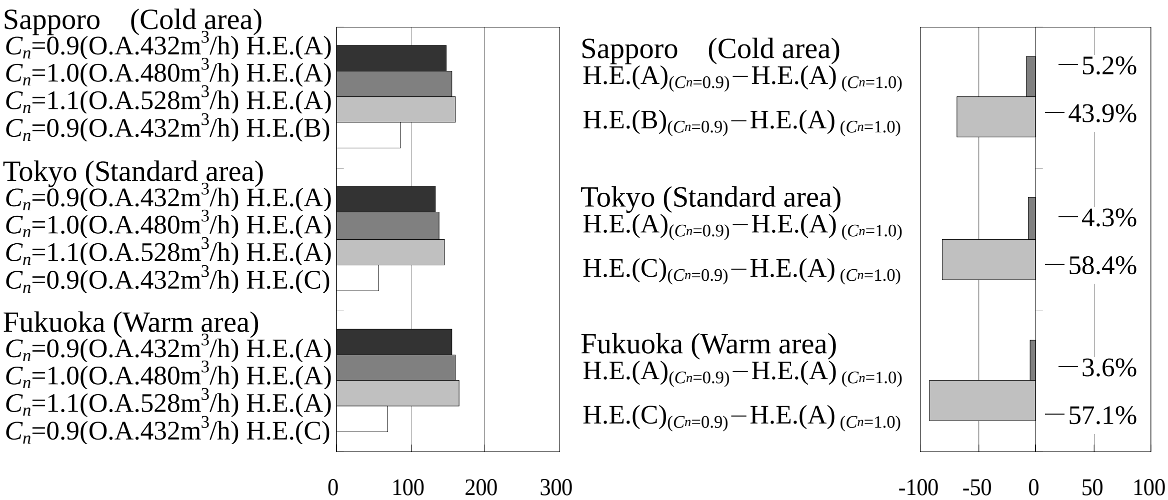


a) Equipment Capacity [MJ/h] b) Difference of Equipment Capacity [MJ/h]

Figure 6 Prediction Results of Energy Consumption

Fig. 6 shows the analysis results of the Prediction results of annual energy consumption. The adoption of the same H.E.(A) in all three climate areas resulted in a reduction of 176 GJ/year (4.9%) in annual energy consumption in Sapporo area, a reduction of 140 GJ/year (4.4%) in Tokyo, and a reduction of 130 GJ/year (3.5%) in Fukuoka. Sapporo achieved the largest reduction in energy consumption, followed by Tokyo, then Fukuoka. This result suggests that the considerable difference between outside air temperature and room temperature during the heating season causes large amounts of energy to be consumed to raise the temperature of outside air, indicating that a reduction in the O.A. helps to greatly reduce energy consumption in cold areas.
Adoption of the H.E.(B) in Sapporo area was reduction of the energy consumption of 1,408 GJ/Year (a 46.0% reduction when C_n = 1.0). This is because gas volume hardly changes between each HVAC equipment, and H.E.(B) in using electric energy reduce to 0.5 times under heating period, and reduce to 0.15 times under cooling period than H.E.(A). Adoption of the H.E.(C), in Tokyo reduced 1,676 GJ/Year (a 52.4% reduction when C_n = 1.0), and in Fukuoka reduced 1,917 GJ/Year (a 51.1% reduction when C_n = 1.0). This is because the H.E.(C) which works only with electric power without using gas.

Results of Annual CO_2 Emissions



a) Carbon Dioxide Emission [t- CO_2/Year] b) Difference of Carbon Dioxide Emission [t- CO_2/Year]

Figure 7 Prediction Results of Carbon Dioxide Emission

Fig. 7 shows reductions in annual CO_2 emissions. The adoption of the same H.E.(A) in all the three climate areas resulted in the largest CO_2 emissions reduction of 8 t- CO_2/Year (5.2%) in Sapporo area. The annual CO_2 emission reduction is 6 t- CO_2/Year (4.3%) in Tokyo and 5 t- CO_2/Year (3.6%) in Fukuoka. This result corresponds to the result obtained in regard to energy consumption. [3],[4]

Also shown in Fig. 7 are the effects of selecting HVAC equipment that is commonly used in each area, in terms of changes from the basic case (C_n = 1.0), which adopts H.E.(A). In this case, the reduction of CO_2 emissions is the largest in Fukuoka at 92 t- CO_2/Year (57.1%), followed by Tokyo at 81 t- CO_2/Year (58.4%) and Sapporo at 55 t- CO_2/Year (43.9%). This result corresponds to the results of analysis of energy consumption.

CONCLUDING REMARKS

The reduction effects on equipment capacity and annual energy consumption of introducing the index for ventilation effectiveness in the design of air-conditioning systems and optimizing outdoor air flow rate as a means to conserve energy in buildings and HVAC equipment have been estimated.

(1) Based on the adoption of the same HVAC equipment for all three climate areas covered, when O.A. is reduced by 10% using the index for ventilation effectiveness, annual energy consumption declined by 4.9% in the cold Sapporo area, 4.4% in the Tokyo standard area, and 3.5% in the warm Fukuoka area. Likewise, annual CO_2 emissions declined by 5.2% in Sapporo, 4.3% in Tokyo, and 3.6% in Fukuoka.

(2) Furthermore, when HVAC equipment that is extensively used in each specific area was adopted and O.A. was reduced by 10%, annual energy consumption declined by 46.0% in cold Sapporo area, 52.4% in Tokyo standard area, and 51.1% in warm Fukuoka area compared to the case in which the same HVAC equipment was adopted but no reduction in O.A. was made. Likewise, annual CO_2 emissions declined by 43.3% in Sapporo, 61.4% in Tokyo, and 60.1% in Fukuoka.

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A study of Variable Air Volume (VAV) systems in foundries

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SUMMARY

The focus on implementing cost efficiency energy efficiency measures will in all probability increase in the future, but it has been shown that trustworthy, site specific information are key features is increasing the adoption of such measures.

This study shows that Building Energy Simulation (BES) software gives trustworthy predictions of energy use and average temperature in the studied case, making it possible to study different HVAC control strategies. When coupled with CFD, it is also possible to study thermal comfort, ventilation efficiency, ventilation effectiveness and draught, giving an even wider range of decision support.

This study also shows that a VAV system is an interesting HVAC control technique for the studied foundry. In this case, the technical potential for reducing energy use in terms of both heat and electricity is predicted to be 30.3% and 28.9% respectively.

INTRODUCTION

An increasing focus on environmental issues, energy prices and competition will in all probability make the implementation of cost efficient energy efficiency measures within the energy intensive foundry industry even more necessary, especially for a country like Sweden, where industry has historically enjoyed one of the lowest energy prices in Europe [1]. The Swedish foundry industry accounts for about 2% (300,000 tons) of the entire European casting production. This makes it a relative large casting producing country per capita, with some 7,000 people employed in the industry.[2] Total energy use by Swedish foundries is about 1 TWh. [2] Support processes such as heating, ventilation and air conditioning (HVAC) within industrial premises are an important issue, as they are related to both energy cost and indoor climate management as well as to the health of the occupants. HVAC in industrial premises account for nearly 30% of the total industrial energy use in Sweden. Furthermore, it has been shown that energy efficiency measures are more easily adopted if they are related to support processes, i.e. HVAC, lighting and compressed air, and not to the production process, as the largest technical barriers to energy efficiency are often related to the production process. This has been studied by Rohdin et al in [3-4]. This stresses the need for accurate and trustworthy information as they are key features in increasing the adoption of energy efficiency measures. One way to obtain this type of information is to use validated simulation methods.

The aim of this paper is to study energy use and thermal climate in a large Swedish light alloy foundry by means of energy simulation, CFD and measurements. The technical potential for using Variable Air Volume (VAV) systems is investigated using these methods.

The energy simulation software IDA ICE [5, 6] was used to perform the numerical energy simulation and the CFD software Fluent 6.2 [8] to perform the numerical flow simulations.

OBJECT DESCRIPTION

Energy use at a light allow casting facility

The relative proportions in the different processes are in line with the ”average” non-iron or steel foundry where melting and holding is estimated to be 40%, molding about 20% and support processes about 40%. [2] The light alloy casting facility presented in this paper has 44% melting/holding, 19% molding and about 37% support processes, see Table 1.

Table 1. Presentation of energy use, energy split on different processes and basic data for the foundry.

Description	Data	Process	Energy use [MWh]	%
Total floor Area	2 744 m2	Melting, holding	7 700	44
Ceiling height	12 m	Casting etc.	3 200	19
U-value walls	~2 W/m^2K	Cooling (product)	650	4
U-value floor	0.3 W/m^2K		11 550	67
U-value roof	~0.6 W/m^2K			
Airflow	45.5 m^3/s	Ventilation	900	5
Q_{trans}	~6 600 W/K	Space heating	4 500	26
Q_{vent}	55 200 W/K	Lighting	250	2
		Compressed air	Not allocated	
Total Energy use	17 200 MWh/year		5 650	33
Energy use/m^2	3 865 kWh/m^2			
Production	~2 100 000 kg/year			

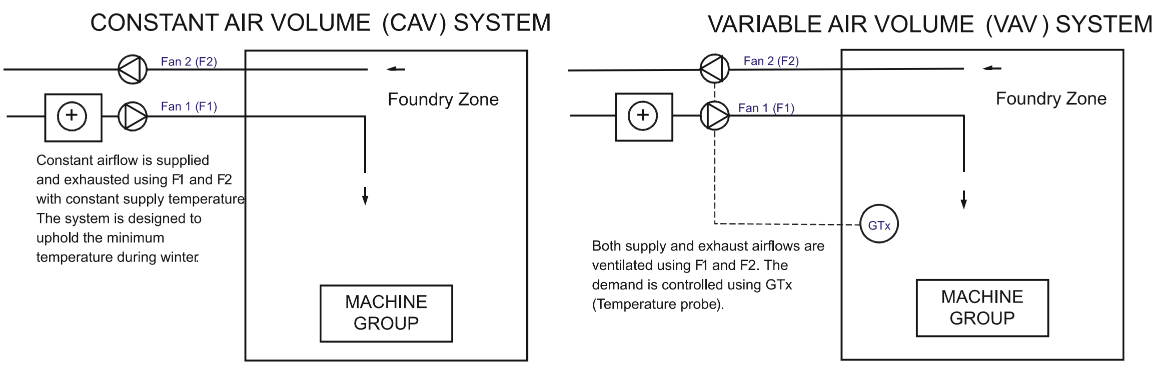


Figure 1. Schematic description of the CAV used today and the purposed VAV system.

THE MODELS

Geometry

The casting facility is 28 m wide and 98 m long, giving an area of 2 744 m^2, see Figure 2. The ceiling height is 12 m. In the facility, there are 16 machine groups (M in the figure) with their associated furnaces (F), and two melting furnaces (MF). Two types of metal are processed in the facility: aluminum and magnesium. Machine groups M1, M2, M4, M12, and M14 process aluminum and the rest magnesium. Air is supplied through 20 displacement supply devices (SD) along the long walls. The exhaust air is evacuated by local exhaust systems located over the machine groups and melting furnaces. All walls are interior walls as the foundry is surrounded by assembly, storage, and other facilities.

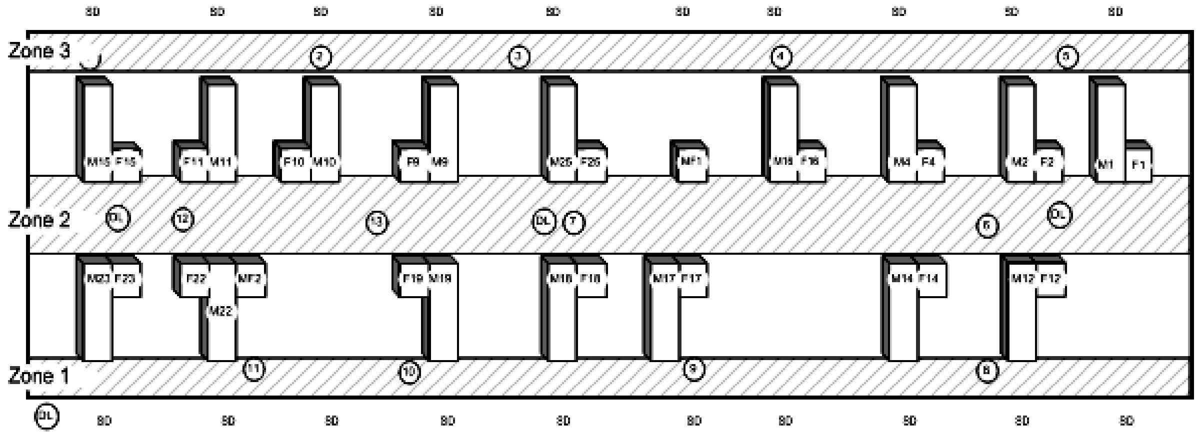


Figure 2. Machine plan of the casting facility.

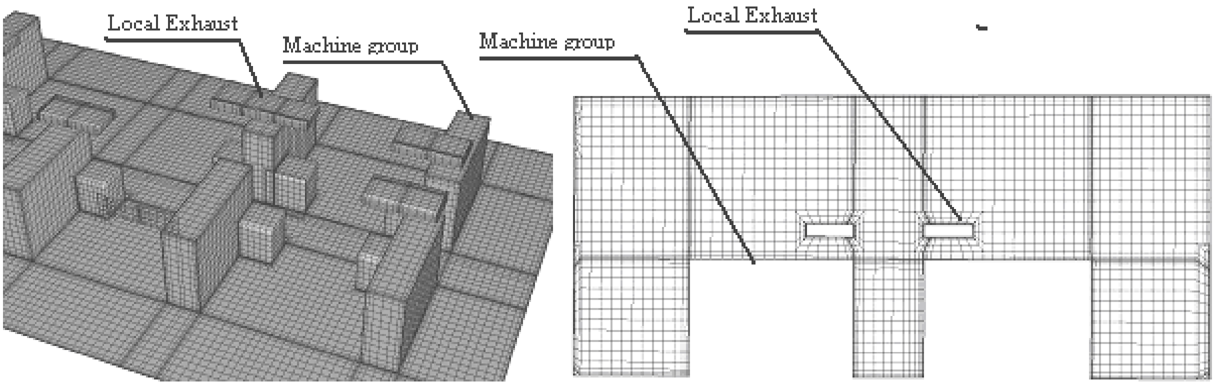


Figure 3. Part of the mesh shown for part of the facility and in section.

CFD Model

Icepack [7] was used to generate the 3-D structured grid shown in Figure 3. The mesh is non-conformal at the interface between wall and coarse region of the model. Fluent 6.2 [8] was used to numerically simulate the thermal climate in the casting facility. The governing equations were solved with a segregated scheme and discretized spatially with a second order upwind scheme. The near-wall treatment used in this model was the standard wall function. The first numerical point was always located at y+>8. The wall boundaries were modeled using the no-slip condition with constant wall temperature. The supply air was modeled as a mass flow inlet with a constant flow rate, and the outlets were modeled using mass flow and pressure outlets. For a more extensive description of the CFD model, where turbulence models and meshing are explained in detail and the model is compared with measurements, see [8]. The Predicted Mean Vote (PMV) and Predicted Percentage Dissatisfied (PPD) are found in ISO 7730 [9], and were implemented in FLUENT by means of a user defined interface (UDF). The calculation of mean age of air was also implemented using a UDF. The Draught Rating (DR) and ventilation efficiency and ventilation effectiveness has been implemented using custom field functions.

Building Energy Simulation Model

IDA ICE 3.0 was used to numerically simulate the energy use aspects of the foundry's construction and HVAC systems. Measured power for the different machine groups and other gains was used as input. The CAV HVAC system, illustrated in Figure 4, was modeled using measured values for supply air temperature, air flows and outdoor temperature. The model was then compared with measured values of the indoor air temperature measurements. The comparison is seen in Figure. 4. A study of time-step independence was made and a maximum of 5 minutes was chosen.

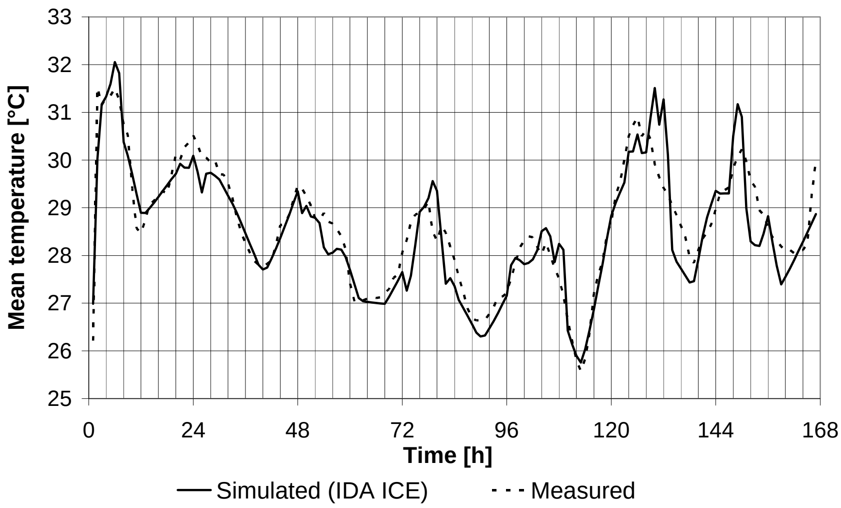


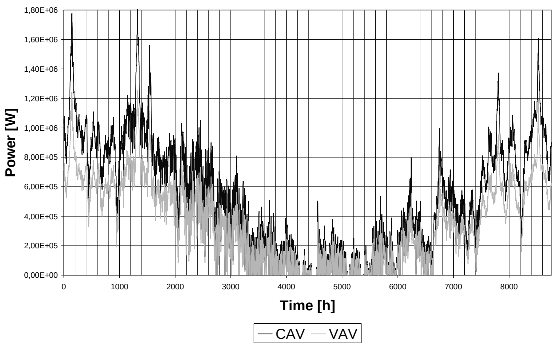
Figure 4. Simulated and measured average temperatures for the entire volume in the facility.

It's important to note that the mean temperature in the occupied zone is lower than the mean temperature as the temperature stratification in this facility is rather high.

VAV VS. CAV IN A FOUNDRY

Two different HVAC control strategies are compared, a CAV system and a VAV system, described in Figure 1. To compare these, both the energy use of the different strategies and their impact on comfort and ventilation effectiveness are studied. Two different methods are used: BES to study energy and power use and a CFD model to study indoor air related parameters.

Energy use aspects



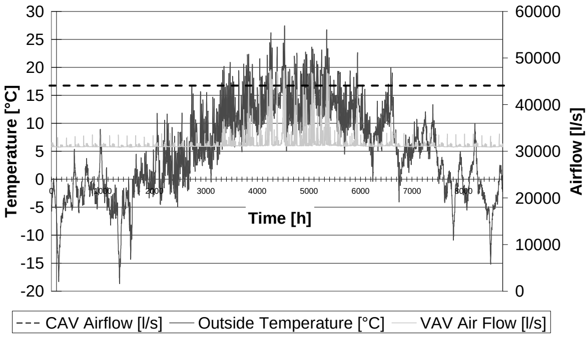


Figure 5. Power used by the heater for the CAV and VAV case (left). Airflows CAV and VAV case.

The mean power for the CAV case is 507 kW and for the VAV case 354 kW. The maximum power for the CAV case is 1.8 MW and 1.25 MW for the VAV case, indicating a decrease in maximum power of 31% from the present case. The energy used for heating the supply air decreases from 4.45 GWh (CAV) to 3.10 GWh (VAV). In addition, the power used by fans is predicted to decrease from 874 MWh to 621 MWh.

Ventilation and thermal comfort aspects

Two different cases have been identified from the energy simulations for further study:
Outside temperature below 14°C, when the supply temperature is 16°C. The VAV system delivers 31 m^3/s.
Outside temperature is 21°C, and the supply temperature 23°C. The VAV system delivers up to full flow of 45.5 m^3/s

To predict changes in comfort for the different cases, both PPD, PMV, and DR [9], were calculated and are shown in Figures 6-9. To predict changes in the function of the ventilation, both ventilation efficiency and ventilation effectiveness were calculated. The Air Exchange efficiency is defined as:

$$\eta_a = \frac{\tau_n}{2 \cdot \tau_i}, \qquad (1)$$

where t_n is the nominal time constant and t_i the average time a particle resides in the facility. The ventilation's effectiveness for heat distribution or heat removal (ε_t) is a measure of how effective the ventilation system is in removing the heat produced internally in the occupied zone and is defined as:

$$\varepsilon_t = \frac{T_o - T_i}{T_{local} - T_i}, \qquad (2)$$

where T_i and T_o are the supply and exhaust air temperatures respectively and, T_{local}, represents the local temperature in the occupied zone. The ventilation effectiveness is presented in Figures 10-12.

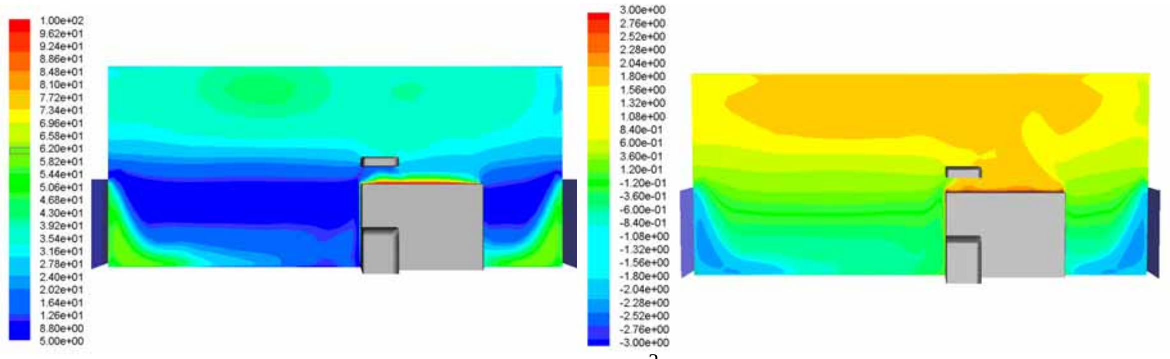


Figure 6. PPD and PMV 16 °C CAV with 45.5 m^3/s airflow.

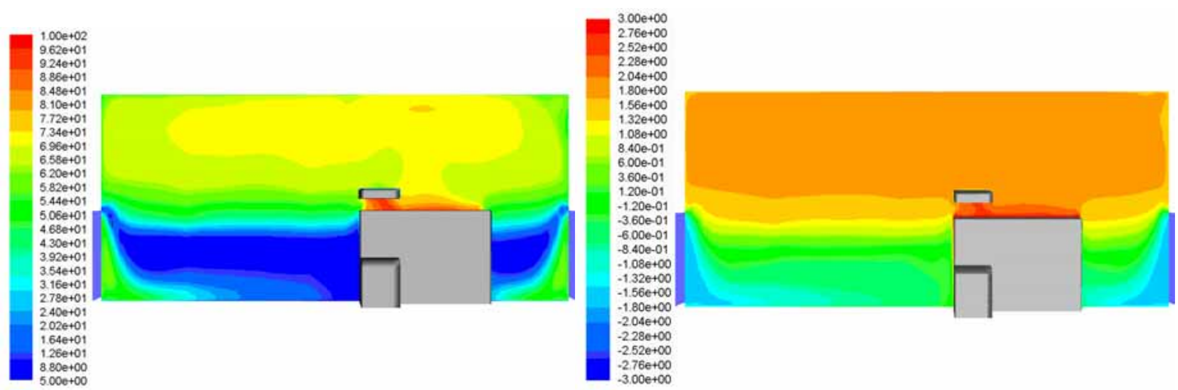


Figure 7. PPD and PMV 16 °C VAV with 31,0 m^3/s airflow.

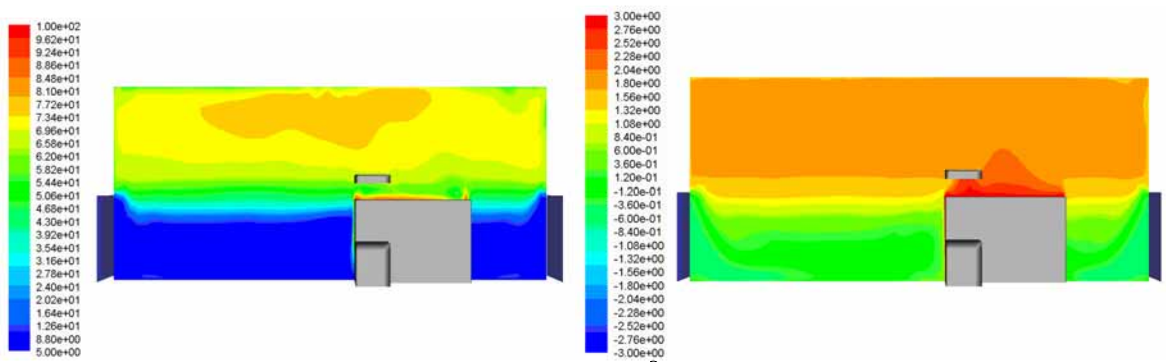


Figure 8. PPD and PMV 23 °C CAV with 45.5 m^3/s airflow.

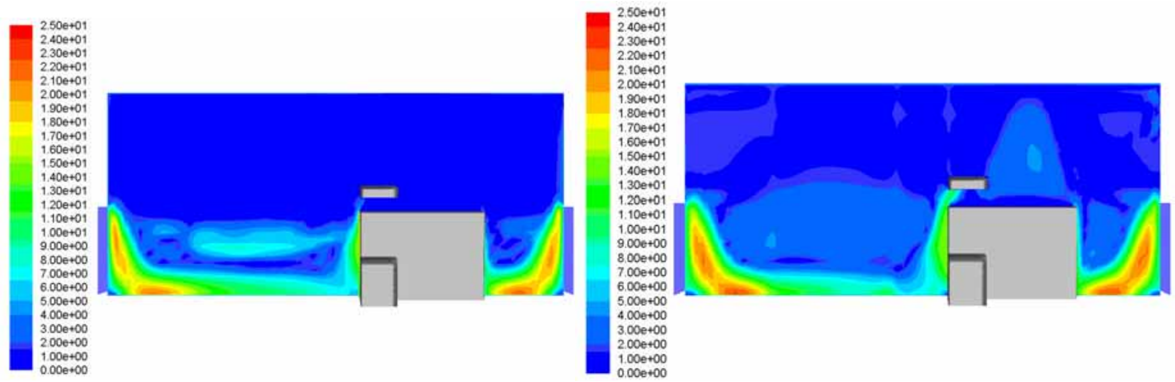


Figure 9. DR (Left) 16°C CAV, (Right) 16°C VAV. The draught decreases slightly when decreasing the flows.

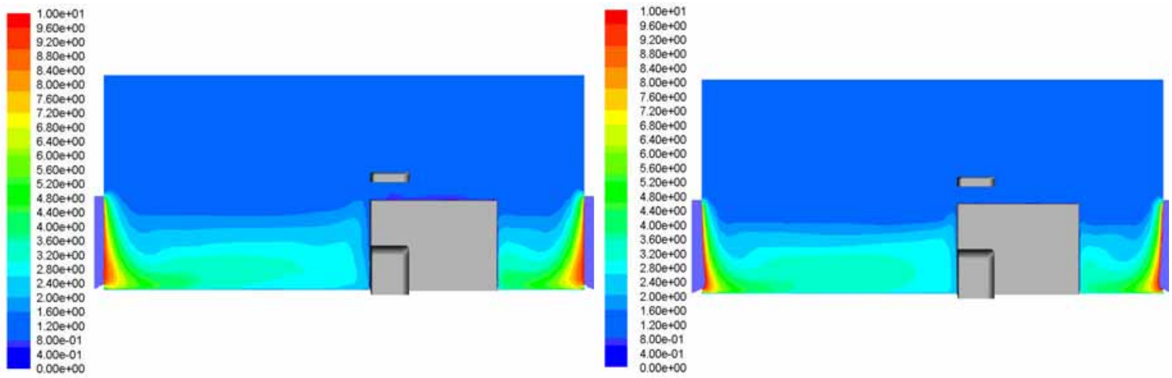


Figure 10-12. ε_t (left) CAV 16 °C (right) VAV 16°C.

For the full flow winter case the air exchange efficiency is 70%, for the 31 m^3/s winter case 76%, and for the full flow summer case 60%. The lower efficiency for the summer case reflects the lower stratification, as the difference between supply air temperature and ambient air temperature is smaller.

The PPD index decreases slightly in the occupied zone when the air flows are lowered when the supply temperature is 16°C. The PMV index indicates that the air is too cold and the air velocity too high. For the summer case when the supply temperature is 23°C the PPD index decreases drastically, indicating a positive thermal sensation. The DR-index indicates that the draught in the supply region in the occupied zone decreases slightly when flows are decreased.

The ventilation’s effectiveness for heat distribution or heat removal decreases slightly, see Figures 10 and 11, when the airflow is reduced. The ventilation effectiveness for the summer case is substantially lower due to a smaller difference between mean air temperature and supply temperature, indicating better functioning ventilation in terms of ventilation effectiveness during winter.

CONCLUDING DISCUSSION

The management of an industrial company requires decisions related to the management of production, maintenance, facilities, energy and labor issues, etc. This management is strongly related to the investment decision the organization has to make in order to achieve and maintain profitability, which is the overall goal of any company. The ability to make the ‘correct’ decision in relation to the overall goal is what separates a successful business from a failure.

This study shows that Building Energy Simulation software such as IDA ICE gives trustworthy predictions of energy use and average temperature in the studied foundry, making it possible to study different HVAC control strategies. When coupled with CFD, it is also possible to study thermal comfort, ventilation efficiency, ventilation effectiveness, and draught, providing an even wider range of decision support. The use of these simulation methods thus gives trustworthy and site specific information which can be used as effective decision support. The importance of site specific and trustworthy information is stressed by, for example, Stern and Aronsson (1984). [11]

This study also shows that a VAV system is an interesting HVAC control technique for the foundry industry. In this case the technical potential for reducing energy use in terms of both heat and electricity is predicted to be about 30% (heat 30.3% and electricity 28.9%). It is also shown that neither the thermal comfort nor the ventilation efficiency is predicted to be negatively affected. The thermal comfort is even predicted to increase. An additional positive effect of the VAV system is the reduced power usage during the cold season, when the demand for district heating and electricity is highest, which in some cases can reduce power related costs even further, since the maximum power demand is reduced.

ACKNOWLEDGEMENT

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Convection Flows from an Overhead Projector and a Data Projector

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SUMMARY

When room ventilation is based on the stratification or zoning strategy, the ventilation air flow rate is determined on the grounds of the convection flows of the heat sources. Thus, in the design phase of office ventilation, the convection flows of common office devices should be known. Especially, to correctly dimension the ventilation air flow, it is important to know the volume flow rates. However, even if the convective power of an office appliance is known, the cooling fan makes the prediction of the characteristics of the convection flow very difficult. Therefore, direct measurement data is needed. In this study, the convection flows of an overhead projector and a data projector were examined. Cross sections of the velocity and temperature fields were measured and analyzed to obtain the volume and momentum flow rates.

INTRODUCTION

In the stratification and zoning strategies of ventilation, the ventilation air flow has to overrun the combined convective flows everywhere in the occupied zone to prevent them from returning. Then, when dimensioning the ventilation in a space with numerous (human or non-human) heat sources, the designer must know the characteristics of the flow generated by each source. Mere knowledge of, *e.g.*, the power consumption of a source is not sufficient. The relative proportions of convective and radiative heat transfers have to be known. (Heat conduction is not significant.) In addition, a warm electric device usually has a fan, generating forced convection with properties completely different from those of natural convection. Therefore, experimental information of the convection flows of typical heat sources in ventilated rooms is needed. This paper reports the results obtained from two common office appliances; an overhead projector and its modern counterpart, a data projector. Both devices have a fan, making the prediction of their convection flows very challenging. Velocity and temperature distributions on different cross sections of the flows were measured, and the results were used to compute the air flow rates, momentum flow rates, and convective powers. The results were compared to a basic heat source, which was a heated, black metal cylinder with 40 % convection.

METHODS

The electric power consumed by an electric device dissipates by two basic mechanisms; convection and radiation. Since the devices usually have a fan, convection is further subdivided in two separate portions. In forced convection heat is carried away by a jet generated by the fan, and in natural convection warm surfaces generate a rising plume, also carrying heat with it. This division is illustrated in Fig. 1. The blow from the fan is most often directed horizontally, whereupon the forced and natural convection flows remain separate.

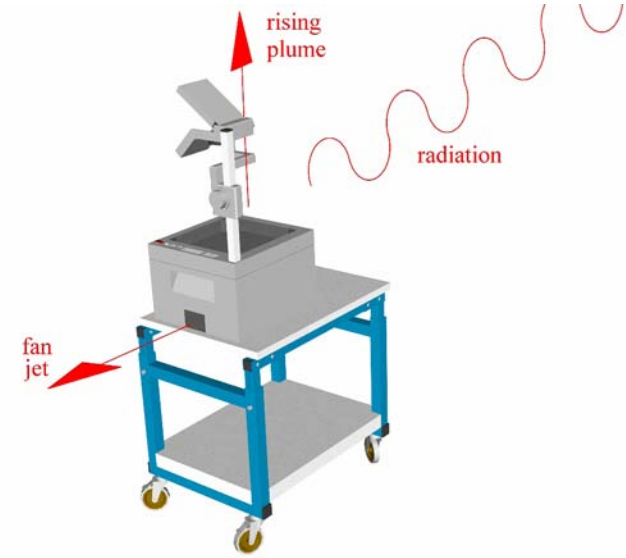


Figure 1. Mechanisms of heat loss from an office device.

The device under examination was placed in a test room shown in Fig. 2. The supply air was delivered through six perforated pipes on the floor to minimize the disturbances inflicted to the plumes, and the exhaust was dispersed at the ceiling. Moreover, the ventilation air current was set as low as possible (~245 l/s) without causing return flow of the plume air. The flow field was measured by two directional sensitive ultrasonic anemometers, attached in a traversing robot, also displayed in Fig. 2. The robot was able to move in all three directions and was programmable, so that measurements could be performed without human presence. In each point of measurement, the anemometers stayed one minute gathering data.

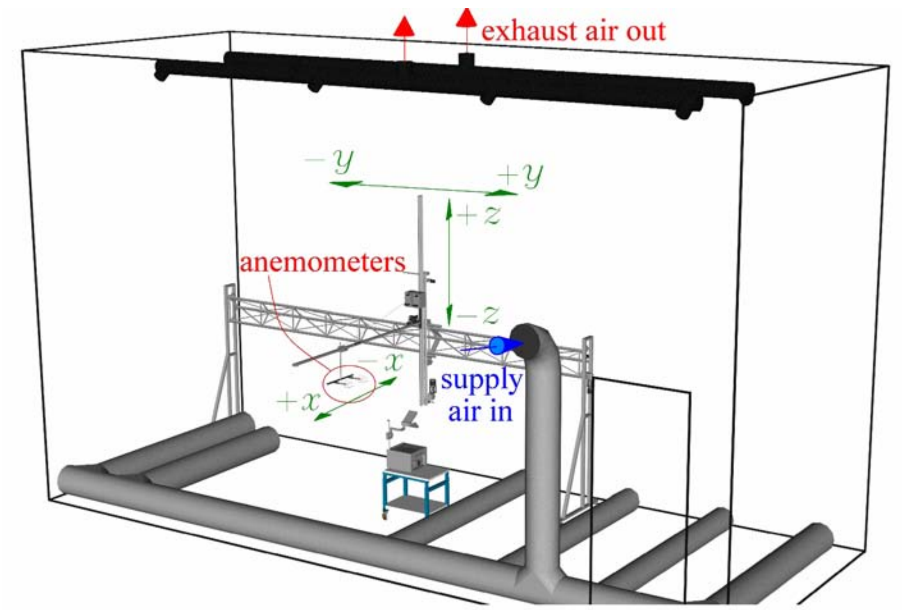


Figure 2. Test room and the measurement robot, carrying the anemometers.

The anemometers were used to measure horizontal and vertical cross sections of the flow fields of the plumes (jets). Figure 3 illustrates one such cross section (the near field in front of the exit opening of the data projector). These cross sections were used to integrate the volume, thermal energy (*i.e.* heat), and momentum flow rates through them. This was possible, since there was a thermistor attached to both anemometers, giving the temperature distributions also.

Figure 3. Measurement grid for the near field in front of the exit opening of the data projector fan. Points of measurement are marked by blue balls. An illustration of the outcome of this measurement is displayed in Fig. 6 (bottom row).

When integrating the air volume flow rate through a plane of measurement, the background velocity has to be subtracted from the velocity measurements. Let $\boldsymbol{v}$ be the velocity vector, and $\boldsymbol{w}$ its background component. Further, let subindex $\perp$ denote the direction perpendicular to the plane of measurement. (This plane can be either horizontal or vertical.) Then the volume flow rate through the plane is

$$q_{\mathrm{pl}} = \int_{\substack{\text{plane of}\\ \text{measurement}}} \left(v_{\perp} - w_{\perp}\right) \mathrm{d}A \ . \qquad (1)$$

The background component $w_{\perp}$ was deduced from the fringe area of the measurement plane.

The heat power P_{c}, conveyed away from the device by the convection flow (plume or fan jet), can be calculated, if at every point we know the flow velocity and temperature rise ΔT of the air. Here ΔT is the increase in temperature, experienced by the air after entering the plume (*i.e.*, the part of the temperature originating from the device). The heat power conveyed through an area element dA is then $\mathrm{d}P_{\mathrm{c}} = c_{\mathrm{p}}\,\rho\,\Delta T\,v_{\perp}\mathrm{d}A$, and the entire power conveyed through the plane of measurement is

$$P_c = \int_{\substack{\text{interior of}\\ \text{conv. flow}}} c_p \, \rho \, \Delta T \, v_\perp \mathrm{d}A \; . \qquad (2)$$

Note that the background velocity $\boldsymbol{w}$ is not subtracted now, since it is the total air flow flushing the device surfaces that is conveying the heat. Also note that the area of integration covers merely the interior of the flow region. This may be difficult to delimit in practice, and if the integration is extended over the entire plane of measurement, it may be necessary to replace $v_\perp$ by $v_\perp - w_\perp$ to reduce the errors accruing from integration outside of the convection flow. Therefore, and for the results to be of general use, the background component of the flow velocity should be insignificant. As for ΔT, its calculation is complicated by temperature stratification. New air is entrained into the flow from all heights, and ΔT should be the present temperature minus the average original temperature of all the air in the flow, which we denote by T_0. When the flow is an upward travelling, natural plume, the effect of stratification can be approximated in a simple way. If at height z above the virtual origin of the plume the ambient temperature is $T_{\mathrm{amb}}(z)$ and plume volume flow rate is $q_{\mathrm{pl}}(z)$, then at an arbitrary height h (above the virtual origin)

$$T_0(h) = \frac{1}{q_{\mathrm{pl}}(h)} \int_0^h T_{\mathrm{amb}} \frac{\mathrm{d}}{\mathrm{d}\,z} q_{\mathrm{pl}} \mathrm{d}\,z$$

and $\Delta T = T - T_0(h)$. If we insert the point source formula $q_{pl}(z) = AP_c^{1/3} z^{5/3}$ and assume linear temperature gradient, *i.e.* $T_{\mathrm{amb}}(z) = T_{\mathrm{amb}}(0) + \beta\, z$, we obtain
$T_0(h) = T_{\mathrm{amb}}(0) + \frac{5}{8}\beta\, h = T_{\mathrm{amb}}(h) - \frac{3}{8}\beta\, h$ and

$$\Delta T = T - T_{\mathrm{amb}}(h) + \frac{3}{8}\beta\, h \, . \qquad (3)$$

Stratification correction (3) was not used when integrating over a vertical cross section of an exhaust jet (*i.e.*, β was set to 0). It was applied after the jet had veered almost vertical and a horizontal cross section was measured. This procedure somewhat underestimates the convection power far from the device, since it underestimates the proportion of cool air entrained into the flow from low heights. However, P_c was only used to judge whether the convection flow has been captured well enough by the measurement. If far away from the source the measured values of P_c have diminished too much, then the flow has probably become too wide or ragged to be analyzed, and the other results are not reliable either.

In addition to volume and energy flow rates, momentum flow rate through each plane of measurement was also calculated. Momentum flow rate $\boldsymbol{M}$ tells the total momentum vector $\boldsymbol{p}$ conveyed per unit time across the piece of plane measured, *i.e.*

$$\boldsymbol{M} = \dot{\boldsymbol{p}} = \mathrm{d}\,\boldsymbol{p} / \mathrm{d}\,t = \int_{\substack{\text{plane of}\\ \text{measurement}}} \boldsymbol{v}\,\mathrm{d}\dot{m} = \int_{\substack{\text{plane of}\\ \text{measurement}}} \rho\,\boldsymbol{v}\,v_\perp \mathrm{d}A \; .$$

This quantity is useful in CFD modelling. The near field momentum flux is a necessary boundary value, and momentum flow rate elsewhere in the flow is useful in validating CFD solutions. The most interesting component of ***M*** is usually that perpendicular to the plane. It gives the perpendicular force that would be impinged by the flow on the plane in case the plane absorbed the flow. It is given by

$$M_z = \int_{\substack{\text{plane of}\\ \text{measurement}}} \rho v_\perp^2 \,\mathrm{d}A \,.$$

The measurement robot only provided one minute time averages $\langle v_\perp \rangle$ and the corresponding standard deviations $s_\perp$. These were then used to calculate the squared average of $v_\perp$ as $\langle v_\perp^2 \rangle = \langle v_\perp \rangle^2 + s_\perp^2$. The deviation originates from the turbulence and fluctuations of the flow. The background flow, which was assumed constant and laminar, was subtracted, so that the final formula for the flow of $p_\perp$ was

$$\langle M_\perp \rangle = \int_{\substack{\text{plane of}\\ \text{measurement}}} \rho \left[\left(\langle v_\perp \rangle - w_\perp \right)^2 + s_\perp^2 \right] \mathrm{d}A \,, \qquad (4)$$

where angle brackets stand for (one minute) time averaging. When computing the above integral, however, the term $s_\perp^2$ was added only in points where $s_\perp < \left(\langle v_\perp \rangle - w_\perp \right)$ to avoid inclusion of mere turbulence in quiescent air.

Finally, the radiative powers of the overhead projector and the data projector were estimated. This was done by measuring the surface temperatures in several points, taking the means of their fourth powers, and applying the formula

$$P = \varepsilon \sigma A \left(\overline{T_\mathrm{d}^4} - \overline{T_\mathrm{r}^4} \right) , \qquad (5)$$

where subindices d and r stand for device surfaces and room surfaces, respectively, overline means spatial average, and emissivity of the device surfaces $\varepsilon = 0.95$. The radiative power of the black cylinder, used as a reference heat load, was calculated more accurately by dividing surfaces into smaller sections and calculating the view factors.

Above formulas (1) – (5) were used to analyze the raw data measured along the flow cross sections. In eqs. (2) and (4), air density ρ is left inside the integrations, since it varies locally along with temperature.

RESULTS

The basic characteristics of the convection flows can be seen from the smoke visualizations in Figs. 4 and 5. It is evident that the convection flow from the data projector is purely forced, but the overhead projector also has a weak plume of natural convection. The jets of forced convection turn more and more vertical on their way, beginning to behave more and more like natural plumes. The jet of the data projector is hotter and slower, whereby it turns faster. The figures also show, how its jet is pasted on the table top (Coanda effect). Therefore, when performing the robot measurements, both the devices were set at the edge of the table. Since the forced convection flow of the overhead projector is slow, cool and wide when it finally rises up, measuring its volume, energy, and momentum flow rates is very inaccurate at high altitudes. The near fields of $v_\perp$ and T of both devices are seen in Fig. 6. They are vertical cross sections in front of the exhaust opening, either 0.2 m or 1.3 m apart from it. They show the fields when looking against the flow, towards the device.

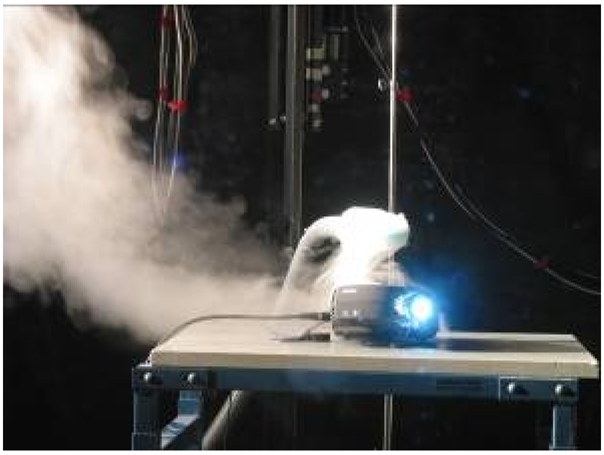
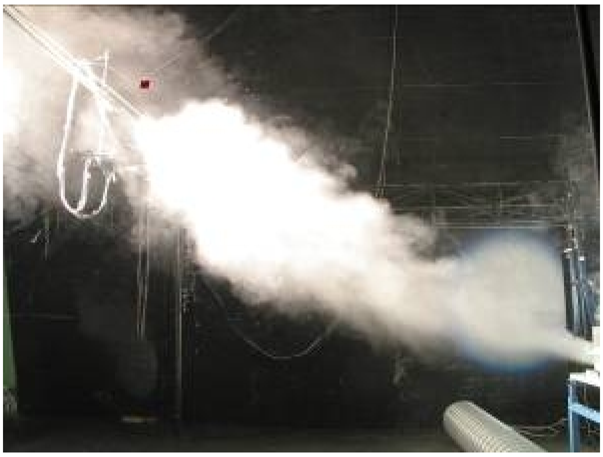


Figure 4. Smoke tests reveal how the convection of the overhead projector (on the left) is divided between forced and natural convection, whereas all convection flow from the data projector (on the right) is forced convection.



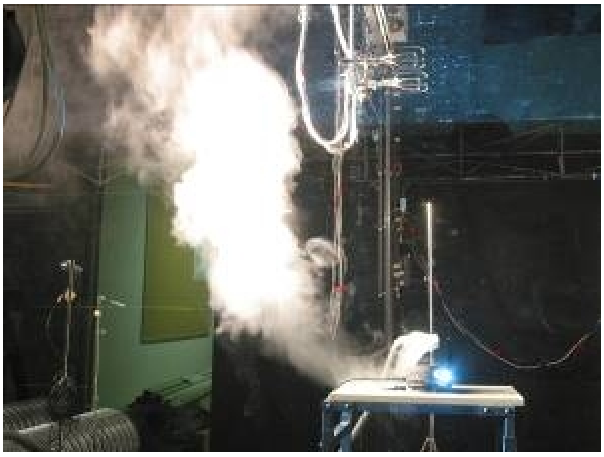


Figure 5. The fan jet from the overhead projector (on the left) is faster and cooler than that from the data projector (on the right), therefore rising up farther away from the device.

The results of the analyses are gathered in Figs 7 – 9. The office devices examined are always compared with a reference plume source, which was a black metal cylinder, heated through its full length. Its properties were known from our earlier measurements, and a measurement with the heating power as close as possible to that of the office appliance was chosen. The height of the cylinder was 1.2 m, and it was standing on a 50 cm high piece of insulator. The height of the table, carrying the projectors, was 0.7 m. The first of the figures, Fig. 7, displays the convection powers (*i.e.*, the heat flow rates) flowing through the cross sections. They should be considered as a test of reliability. For the measurements to be reliable, the sum of the (forced and natural) convection and radiation powers (kinetic energy flow rate is not significant) should be near the heating power used, written in black. Thereby, a reduction in the measured convection power during the rise of the flow tells that the precision is deteriorating.

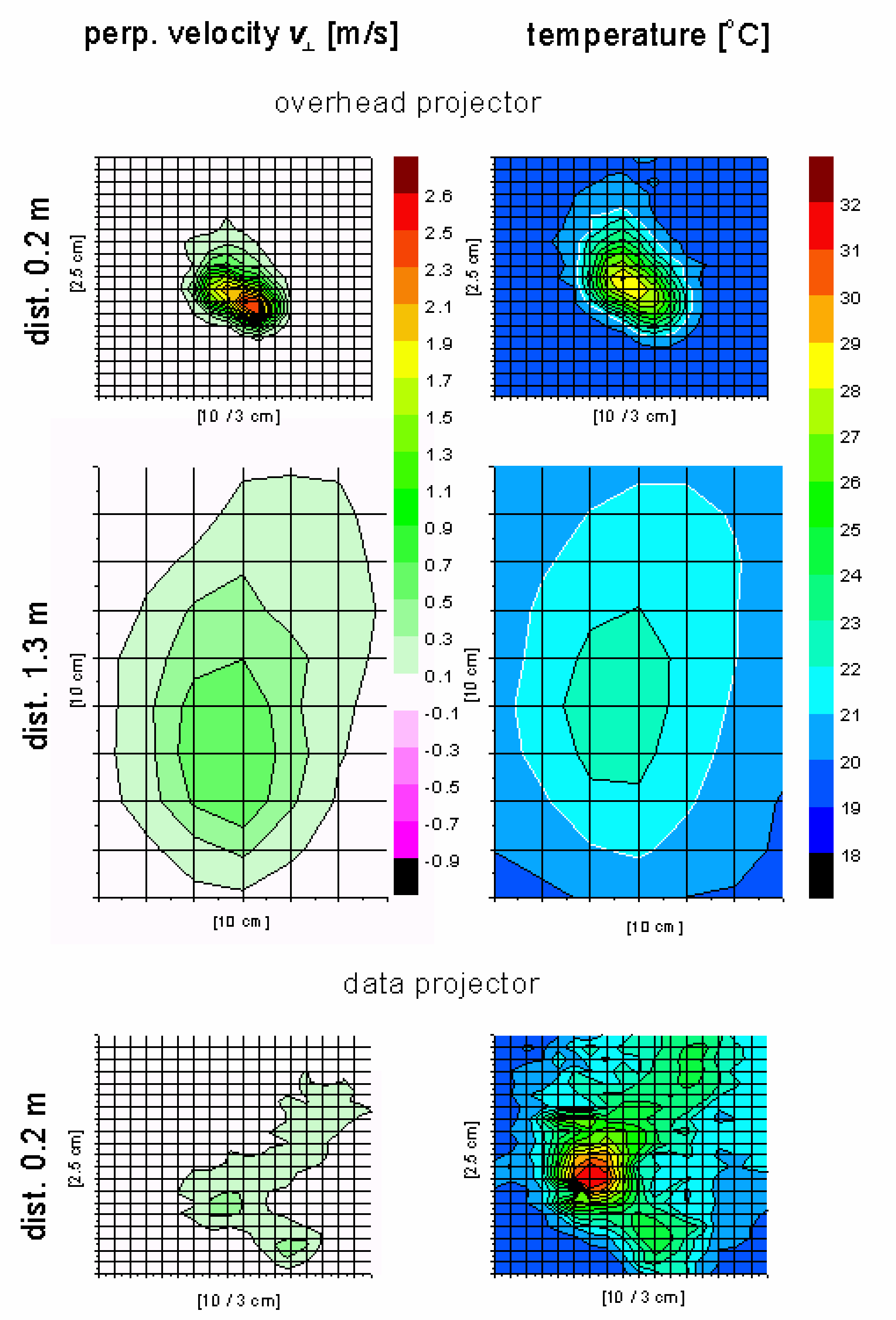


Figure 6. Vertical cross sections of the jets of forced convection. The velocity maps display the component $v_\perp$, parallel with the original direction of the blow and perpendicular to the plane of measurement. All the maps are drawn in the same scale, the grid of measurement points is shown, and the grid size is given in centimeters.

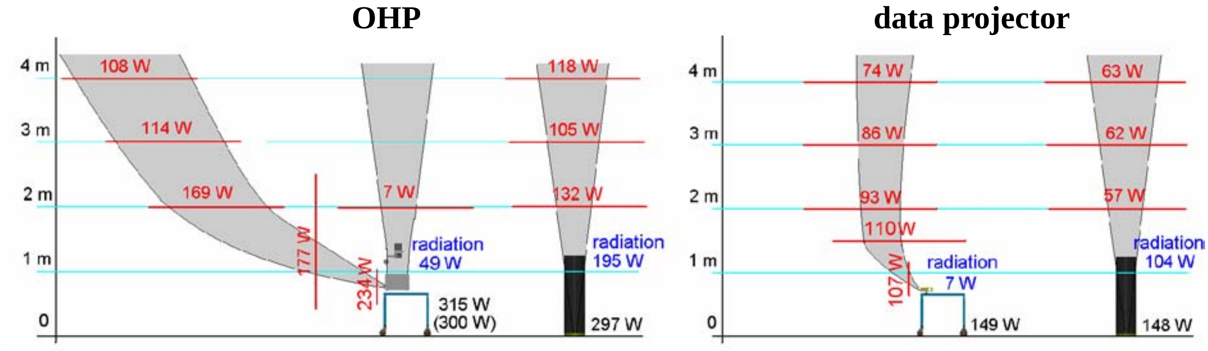


Figure 7. Measured convective and radiative powers of the office devices as compared with those of a black metal cylinder. The power consumption of the overhead projector was normally 315 W, but when measuring the near field 0.2 m apart, a reserve lamp had to be changed, after which the power consumed was 300 W. OHP = overhead projector.

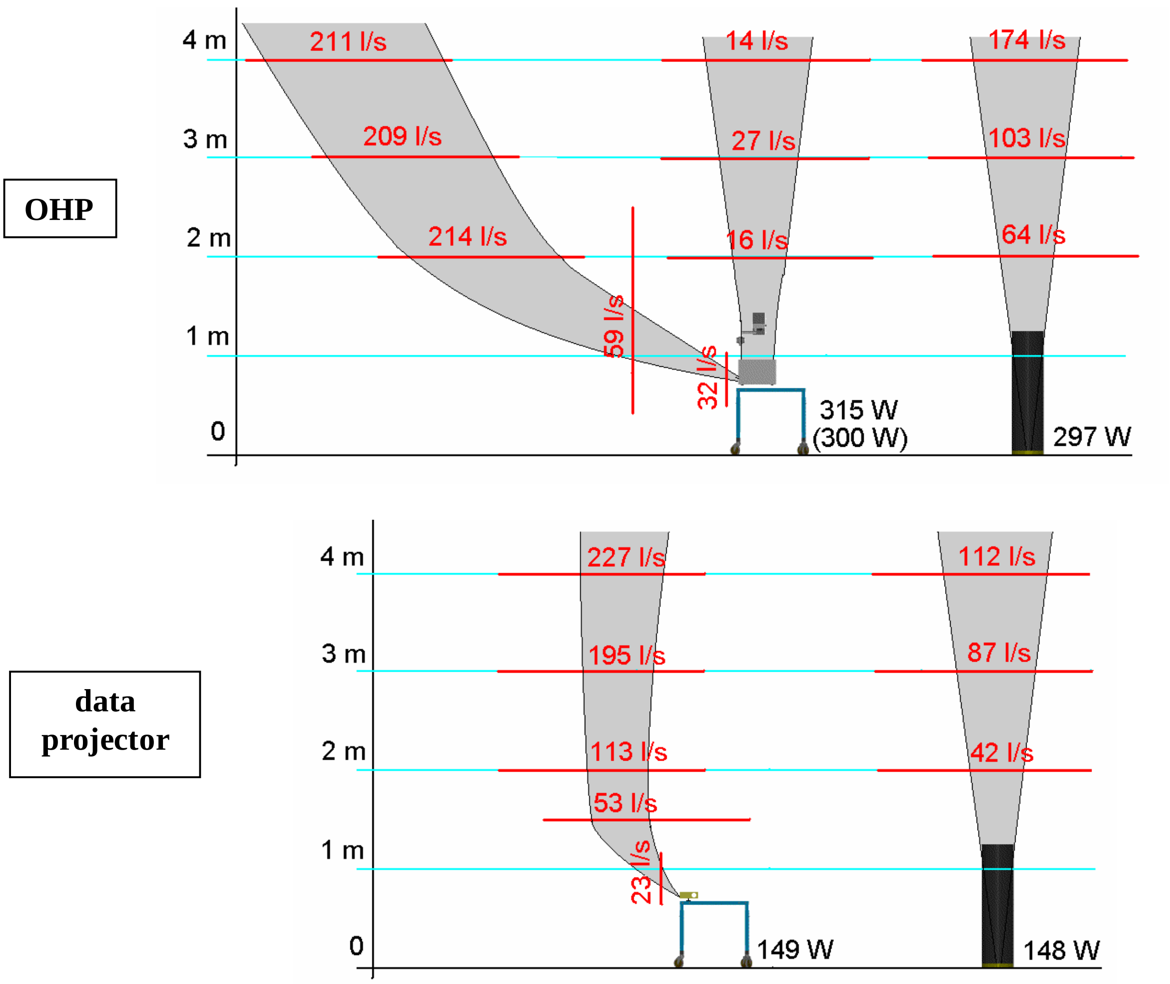
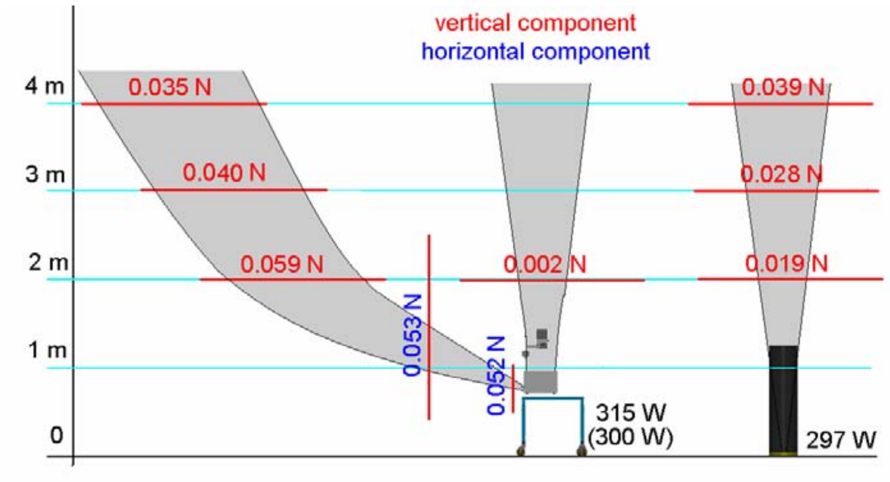


Figure 8. Measured convective volume flow rates of the projectors as compared with those of the black cylinder.

OHP



data projector

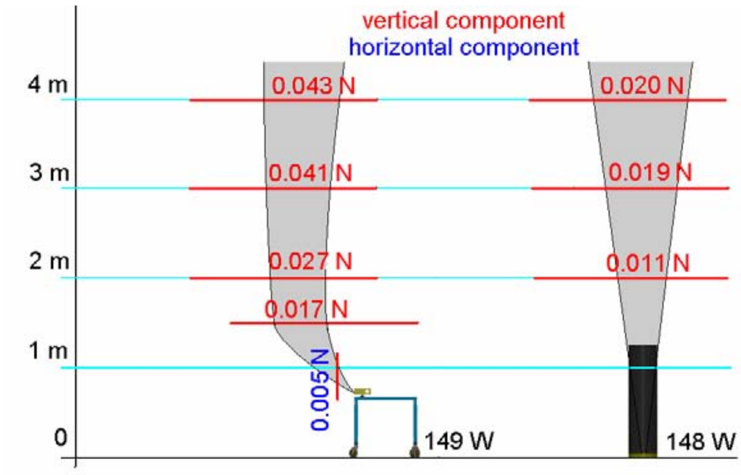


Figure 9. Measured convective momentum flow rates of the projectors v. the black cylinder.

According to the results, the most important of which are shown above, we can say the following:

- In the presence of device fan, a great majority of thermal losses takes place through convection.
- The fan and the increased buoyancy due to increased convection power multiply the convection flow (volume flow rate) by a factor between 2 and 4 as compared with a black cylinder with the same heating power. Therefore, it is a wise idea to attach a data projector to the ceiling.
- The vertical momentum flow rate is increased by a factor between 2 and 3. This can partly be explained by the increased buoyancy, but the most important reason to this are probably the pressure forces at the top surface of the reference cylinder, hindering the growth of the momentum flow rate. These kinds of forces do not confine the exhaust jets of the projectors. A totally new feature is the horizontal momentum flow rate due to the horizontal fan jet (blue numbers in Fig. 9).
- The temperatures of the plume air are not much different between the projectors and the cylinder with the same heating power. At the height of 2 m the maximal temperature in the plume of the overhead projector is 23.9 ºC, when for the cylinder heated by the power of 297 W it is 23.5 ºC. For the data projector and the cylinder heated by 148 W the corresponding values are 21.3 ºC and 22.2 ºC.

DISCUSSION

Common electric office appliances generally have a cooling fan producing a powerful flow of forced convection, which easily absorbs the natural convection flow along. This flow is a horizontal buoyant jet, which first behaves like a jet, and gradually changes to a buoyancy-driven plume. Prediction of the volume and momentum flow rates of such a flow is problematic, and in any case the near field has to be measured to obtain the initial values. In this study the behaviour of the (forced) convection flow in the jet, transition, and plume regions has been measured for two common office devices. The results were compared to results measured for a black cylinder, generating only a natural, buoyant plume.

The farther away the flow has traveled from its origin, the more challenging is its measurement. The reason to this is that the flow becomes cooler, slower, wider, and more disassembled on its way, making it more difficult to distinguish from the background. To provide a criterion to decide, how reliable the results are, the (thermal) energy flow rate (or the convection power) was always calculated together with the volume and momentum flow rates, see Fig. 7. Comparison of the electric power with the sum of the measured radiation and near field convection powers tells us, how the measurement of the initial values has succeeded. After this, the conservation of the measured convection power during the course of the flow gives a conception of how fast the measuring accuracy is deteriorating. Note, however, that when calculating the convection power, the convective temperature rise ΔT is also needed. It is obtained in poorer accuracy than velocity, and therefore the accuracy of momentum flow rate and, especially, volume flow rate is higher than that of the convection power.

As Fig. 8 shows, at the height of 2 m the volume flow rate of the overhead projector is 230 l/s. This is more than ten times the air flow above the head of a person (20 l/s, see [1]). The data projector, despite its relatively small power, also produces a convection flow rate corresponding to several persons. In addition, each person usually has a computer, producing a larger convection flow than the person himself [1]. Therefore, when dimensioning the ventilation air flow rate in stratification or zoning air distribution strategy, the office appliances should definitely be taken into account. Nonetheless, the ventilation air flow rate required can be reduced by attaching the data projector to the ceiling, above the occupied zone.

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11 June 2007 at 15:30 - 17:00

A03
Filters and air cleaning

The Engineering of Practical Gas Phase Air Cleaning

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SUMMARY

This is one of two CLIMA 2007 papers on gas phase air cleaning by this author. The Science of Gas Phase Air Cleaning' covers aspects of air quality, gas phase air cleaning (particularly adsorption) limitations and opportunities, capital and energy saving impacts, and the role of Standards.

This paper discusses application parameters, deals with testing of gas phase air cleaners and presents odour removal efficiency test results for a product. Economics of use are discussed.

ACKNOWLEDGEMENT

This author gratefully acknowledges the financial assistance of the Australian Capital Territory Government in developing gas phase air cleaning apparatus and testing facilities.

This author gratefully acknowledges the assistance and cooperation of the Australian National University – particularly in allowing use of its facilities.

INTRODUCTION

The use of gas phase air cleaning (GPAC) in heating ventilation and air-conditioning (HVAC) applications - to remove building/occupant generated pollutants - is rare.

This contrasts with the use of particulate air filters which is near universal.

As indoor air pollutants are either particles or gases this contrast of technology use is stark.

The main barriers to GPAC use in HVAC are:
- Understanding of GPAC
- Product availability
- Testing
- Application approach.

OPPORTUNITY

Presently the HVAC contribution to good indoor air quality (IAQ) is by particulate filtration and use of ventilation to dilute odours (always, for practical purposes, in the gas phase) and other gas phase contaminants like volatile organics.

GPAC use presents the opportunity to reduce ventilation without reducing IAQ.

If ventilation is reduced energy use is reduced, HVAC plant size is reduced, kVA charges are reduced, plant efficiency may be improved, energy infrastructure needs are reduced and greenhouse gas production is reduced – refer to the other paper for more detail.

GAS PHASE AIR CLEANING

Only adsorptive (physisorptive and chemosorptive) air cleaners are dealt with here, as this author sees them as having most potential. Absorptive and (photo)catalytic approaches are not addressed.

Absorptive processes are usually water based, with the obvious problems – humidity control, legionella, chemical contamination of air etc.

(Photo)catalytic media is a possible solution but has the problem that it can sometimes adsorb reasonably harmless chemicals and catalyse them into more harmful species and release these into the airstream (perhaps this problem will be solved by developers).

This author favours adsorption – the pollutant is trapped and held in the media.

Typical adsorptive GPAC ‘media’ are activated carbon, activated alumina, zeolites, silica gel etc. These media may be ‘pure’ or they may be impregnated with chemically active ingredients to allow chemisorption.

A GPAC is usually a bed, of chosen thicknesss, of adsorbent material granules contained between metal mesh or like ‘porous to airflow’ material.

UNDERSTANDING GAS PHASE AIR CLEANING

Adsorption is qualitatively addressed in more depth in the other paper.

Quantitative (mathematical) treatment of adsorption is extremely difficult and the outcomes are exceptionally complex. Also sorbent material performance information is scant.

With application of considerable resources, the matter can be resolved but developed methodologies are largely kept confidential. This author has developed a predictive model. Useful published design recipes are rare, perhaps nonexistent. Industrial adsorption process design seems to be based on considerable experience.

A good (but not simple) overview and basic design data source is Perry’s Chemical Engineers Handbook 7th Edition Section 16 ‘Adsorption and Ion Exchange’.

GPAC BEHAVIOUR

Numerical examination of behaviour of possible and existing adsorptive GPAC’s reveals useful things. Descriptive material addressing adsorption is available – but, without numbers, few useful design conclusions can be drawn.

The efficiency (percentage removal of contaminant from air) of a specific adsorptive bed at a specific time depends on many factors - including contaminant concentration, contaminant nature, adsorbent nature, face velocity, temperature, effect of other contaminants, bed depth.

The actual proclivity of a sorbent to sorb a contaminant is paramount but after this the prime characteristic of interest, when considering performance of practical GPAC units, is depth of the media bed (dimension in the direction of airflow). This can be qualitatively referred to by calling these beds 'thick' or 'thin'.

Media bed depth is not to be confused with the overall size of a commercial GPAC. Such units are often composed of thin sub units arranged in 'v', 'w', etc formation so that much media face area is presented to the airflow. These composite units thus have a large dimension in the direction of gross airflow but the individual sub units have a smaller dimension in the direction of local airflow.

Variation, over time, of efficiency is particularly dependent on bed depth. This is now addressed, but in a simplified manner.

When presented with a substantially constant pollutant removal challenge a thick bed GPAC operates thus:

- At air inlet side of the bed there is part of the bed thickness (say ¼ of the thickness – as an aid to thinking) referred to as the mass transfer zone
- As air moves through the zone the pollutant concentration decreases until it becomes the practical minimum at the end of the zone (i.e. practically, as clean as it is likely to get – often close to 100% pollutant removal)
- As time passes, and more air is cleaned, the upstream part of the zone becomes increasingly saturated with pollutant and the downstream edge of the zone moves further downstream
- As more time passes the bed divides into three zones – an increasingly wide saturated zone, followed, by a largely constant width mass transfer zone and a increasingly smaller 'clean' zone
- The mass transfer zone may widen or narrow as it moves, depending on the adsorption relationship, A narrowing zone will asymptotically approach a constant width.

The practical outcome of the above is that a thick bed GPAC will have a substantially <u>constant pollutant removal efficiency</u> for a period of time and then this efficiency will decrease. Finally, when the bed is saturated the efficiency will be zero.

If a thin bed GPAC is used (i.e. the thickness is less than that of the mass transfer zone - the 'critical thickness') then the initial <u>efficiency</u> will be a chosen value and it <u>will decline from the time of first use</u>. This is usually an undesirable characteristic.

REQUIRED GAS PHASE AIR FILTER PERFORMANCE FOR HVAC

Example (typical in many places) HVAC conditioned airflow into an occupied space is about 50 litres per second per person (l/s/pp) – this is determined by thermodynamic needs i.e. it is the airflow required to deliver the required heat (or "coolth").

Typical "fresh air" flow into a space is about 10 l/s/pp – this is what is needed to maintain odour at acceptable levels – given conventional design practice.

Say 7.5 l/s/pp of this fresh air is replaced by recirculation air that has been 100% cleaned in a GPAC. And, also, that 2.5 l/s/pp of "fresh" air is delivered to dilute CO_2 and other pollutants etc to acceptable levels. In this scenario the 10l/s/pp of ventilation air is replaced by 2.5 l/s/pp of ventilation air plus 7.5 l/s/pp of cleaned air with the cleaned air being, in the case of many pollutants, cleaner than the ventilation (outside) air that otherwise would have been used.

Now, consider that the same outcome is to be achieved by use of a GPAC treating all conditioned air flow (i.e. 50 l/s/pp). The question arises "what is the required efficiency of this GPAC".

The answer: 7.5/50 or 15%. i.e. the required gas phase air cleaner odour removal efficiency is 15%. The required efficiency would be 10% if 5 l/s/pp was cleaned.

So, the required efficiency of a GPAC used for odour control in reasonably normal HVAC systems will be in the order of 10% to 20%.

AVAILABLE GAS PHASE AIR FILTER PRODUCT

Presently GPAC product is available for HVAC application. Mostly this product contains activated carbon or potassium/sodium permanganate impregnated activated alumina. A wide range of other impregnations (for carbon and alumina), including some with catalytic activity, are also available.

Generally, the % efficiency of this product (at any stage of its operational lifecycle) for removal of odours, or many other pollutants, will not be stated by the manufacturer.

Designers must rely on their experience etc and, perhaps, their 'blind faith'.

Most of this presently available commercial product is, in the context of claimed HVAC use for the removal of body odours etc and trace organics, thin bed. There is a small amount of thick bed product available but it seems to be aimed at industrial use.

This product will thus, generally, offer pollutant removal performance that will decline (from what value?) immediately and continuously, or soon and then continuously, after installation.

The presumed reason for one aspect of this undesirable situation is that a GPAC user must limit the air pressure drop to that he/she finds acceptable and GPAC manufacturers provide product to match this presumed requirement.

At the heart of this issue is the balance that must be made between the advantages to be obtained by GPAC use in HVAC (lowered ventilation, plant size, energy use etc) and the costs – particularly fan energy use. In some cases the problem is illusory, in most it deserves attention.

An optimal approach to many, if not most, practical situations is to be found in a recent Patent application[1]**.**

GPAC TESTING

A user of GPAC properly requires adequate performance data – though, to date, much use has been with rule-of-thumb design methods or pure faith.

GPAC may be used to remove pollutants that are undesirable from a health or like perspective. They may also be used to remove odours from air.

As many ventilation standards/requirements are based on odour control (with a smaller ventilation rate set for health reasons) e.g. Australian Standard AS1668.2: 2002[2], the major opportunity for economic application to HVAC use is in odour control – though VOC etc control is of importance.

A number of organisations (ASHRAE, ASTM, CETIAT etc) and some companies have done thorough and advanced work in the development of standards and apparatus for testing the removal efficiency of GPAC or GPAC media for specific chemicals – e.g. toluene.

If theory and data collection were sufficiently advanced then information from such testing may be applicable to calculation of odour removal by GPAC. However, it seems that we are presently a very long way away from this goal.

Accordingly, it is desirable that GPAC be directly tested for odour removal efficiency.

GPAC ODOUR REMOVAL EFFICIENCY TESTING

This author has developed GPAC odour removal efficiency test apparatus (a test rig) at the Australian National University. This apparatus is suited to the testing of full scale GPAC product. The apparatus is designed to test GPAC so that they may be used to gain 'outdoor air' concessions in accordance with the requirements of AS1668.2[2] (1991 and 2002 editions). Product has been successfully tested.

The mentioned test rig is believed to be the only one in operation or under consideration – anywhere. The test approach is now described.

Odour removal efficiency removal testing requires the use of human noses. Current sensor technology is not up to the task. The described methodology uses human odour assessors.

Traditionally, in respect of olfactory assessment of indoor air, there has been debate about whether assessors are to be (a) selected for olfactory capability or selected as being representative of the general population and/or (b) trained or untrained. Assessors selected for testing of devices in accordance with the methodology <u>are selected</u> for adequate olfactory capability and <u>may or may not</u> be trained for the specific or general task at hand. Assessors are selected in accordance with AS2542.1.3[3].

The method is conceptually straightforward:

- A panel of assessors samples two air streams and declares which is the most odorous
- In olfactory science terms this is a bilateral paired comparison test in accordance with AS2542.2.1[4] (tests other than the bilateral type may be chosen)

- Contaminated air is drawn from an enclosure reasonably representative of one for which the GPAC may be used (eg a nominated type of air-conditioned room with people in it)
- "Fresh' air is drawn from outside. Source/treatment conditions are specified
- Cleaned air is drawn from downstream of the GPAC
- A minimum GPAC odour removal efficiency is postulated
- A mixture of contaminated and fresh air (chosen to mimic air that has been cleaned by a GPAC of the postulated efficiency) is delivered to assessors
- Cleaned air is delivered to assessors
- The two airstreams are assessed and the most odorous is declared
- If the cleaned air is less odorous than the mixed airstream then the minimum efficiency of the GPAC (in the tested application) is as postulated
- The test is repeated, at a different postulated minimum efficiency, if required.

In operation the test rig has proved to be robust, efficient and effective.

The rig has few limitations, and these will be soon remedied. Presently the rig will only test a postulated efficiency up to a level just above 90%. At higher efficiencies one airflow falls outside the parameters of the airflow test measurement standard used (AS 2360.1.1: 1993[5]).

TEST RESULTS

A GPAC device for the removal of body odour and other building occupancy odours (established building) has been developed. The composition of the media bed is proprietary (a trade secret).

The GPAC has been tested. It has a minimum tested body odour etc removal efficiency of 90.4% with $P<=0.05$ error probability (criteria that is customary in sensory science, and other scientific and engineering applications). The calculated actual efficiency is in excess of 98%.

ECONOMICS

The economics of GPAC use to reduce ventilation costs is illustrated in the following table. Here the savings to be made by GPAC use for 14 Australian cities are tabulated. The calculations have used precise hourly outdoor dry/wet bulb frequency data. The cities represent most climate types but no really cold climate is represented so conclusions cannot be drawn for many north American and north European places. Note that the estimated savings in the table below are based on energy use at Australian prices. European prices are considerably higher.

Next table below assumes:

Ventilation rate reduction: 1000 litres/second (perhaps a 200 person building)
Cost of electricity for cooling: A$0.138/kWH, Cooling coefficient of performance: 3.0
Heating gas energy cost: A$0.0372/kWH, Heating system efficiency: 76%
Marginal capital cost of cooling effect: A$800/kW
A$=Australian dollar (in early 2007, A$1.0~0.6 Euro~USA$0.8)
6am to 6pm, 6pm to 6am or 24hr operation: 24C summer inside, 21C winter inside.
Economy cycle not used, no humidification, dehumidification only to 60% RH.

Australian City	*Climate*	*Marginal cooling plant cost (A$)*	*Annual cool+heat cost, day (A$)*	*Annual cool+heat cost, night (A$)*	*Annual cool+heat 24 hr (A$)*
Adelaide	Mediterranean	12,480	887	1290	2077
Alice Springs	Hot arid	15,440	1644	1461	3104
Brisbane	Sub tropical	18,240	1417	847	2264
Canberra	Temperate	4,880	1565	2406	3970
Cloncurry	Semi arid tropical	16,080	2622	1751	4373
Darwin	Monsoon	21,280	5011	4350	9361
Hobart	Cool temperate	3,040	1274	1913	3187
Melbourne	Temperate	10,160	1020	1511	2531
Mildura	Semi arid	14,880	1171	1531	2710
Perth	Mediterranean	12,096	893	978	1870
Port Hedland	Semi arid	15,360	3609	2603	6212
Sydney	Temperate oceanic	6,800	1018	1973	2991
Townsville	Wet tropical	25,680	3603	2405	6008
Wagga Wagga	Temperate	12,560	2224	1706	3930

The operation cost (media+energy) of the mentioned 1000 l/s unit, used as shown in reference 1, is likely to be in the region of $A1000 per annum for a typical office space (2500 hrs/year) use. The capital cost is not given here.

Clearly, GPAC use is cost effective in many climates. In many places it is cost effective in capital cost terms regardless of energy savings. It may be used only or principally to reduce capital and kVA etc. charges - thus reducing GPAC fan energy to negligible levels and extending media life to, perhaps, decades.

CONCLUSION

Now is an opportune time for the widespread introduction of gas phase air cleaners into HVAC applications. Various monetary savings and environmental benefits can be had worldwide. There is present ample opportunity for interested commercial entities to leverage recent developments into benefiting the greater good through improving indoor air quality and reducing greenhouse gas emissions.

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4. Australian Standard 2542.2.1: Sensory Analysis – Paired comparison test.
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Breakthrough time of activated-carbon filters used in residential and office buildings – Modelling and comparison with experimental data

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ABSTRACT

While being used for years in industrial applications, gaseous contaminant sorption units such as activated-carbon filters remain marginal in office and residential buildings. This study deals with models that could help building designers and administrators to design and maintain activated-carbon filters depending on the environmental conditions (pollution load, airflow rate, temperature, humidity) they operate. More precisely, as a necessary first step of the development, the emphasis is put here on demonstrating the relevance of the models by comparing the predicted and measured breakthrough curves of various contaminants (isolated or as a mixture in the air) in dry and isothermal conditions. The results globally show a good agreement between predicted and measured data, while highlighting some interesting trends for future model developments. Beyond the question of model validation, the results also underline great discrepancies in the breakthrough time of the filter from one contaminant to another: the filter breaks within few tens of hours for some species when it breaks after several months for others. Considering the number of contaminant found in indoor settings, this shows that the question of the efficiency of the filter cannot be answered globally, and that the question of how often the filtering medium must be changed is not as simple as one can think.

INTRODUCTION

In the context of climate change, reducing drastically the energy consumption and related CO_2 emission of buildings has become a really challenging problem for most countries. For instance, the European directive on Energy Performance in Buildings (EPBD) estimates that the energy consumption of buildings should be reduced by 22% by the year 2010. Since ventilation can represent from 30 to 40% of the energy loss in new buildings, reducing the air change rate by using air cleaning systems may be an interesting solution to save energy while maintaining an acceptable indoor air quality. Two broad air cleaning technologies exist: oxidation techniques such as photocatalysis or cold plasmas have the advantage of decomposing organic compounds but face the problems of possible deactivation of the catalyst as well as possible yield of harmful secondary chemicals. On the other hand, adsorption systems such as activated carbon filters only trap gaseous species which means that the adsorption medium must be changed periodically to keep the filter efficiency. Then the question becomes to know how often the medium must be changed depending on the conditions the filter operates. Most existing methods are intended for industrial applications and cannot correctly account for the conditions encountered in buildings, especially time-varying mixtures of contaminants at low concentrations in the air, and in most cases fluctuating temperature and humidity at the filter inlet. The present paper addresses this problem. More precisely, the results presented here are part of a larger study aimed at developing physically-based models capable of predicting the dynamic change in the efficiency of a packed-bed activated carbon filter operating in such conditions. As a necessary first step of the development, the emphasis is put on demonstrating the ability of the models to accurately fit the breakthrough curves of any gas. After describing the models for isolated contaminants as well as for mixtures of contaminants in the supply air-flow, the paper presents a comparison between the predicted and measured breakthrough profiles of six contaminants pertaining to different chemical classes.

MODELS DESCRIPTION

The adsorption filter models used for the study have been developed based on contributions from Axley [1] and Yu and Neretnieks [2] in the early nineties, as well as theories and equations proposed for the dynamic modeling of filters used in industrial applications (see for instance Do [3] or Tien [4]). All these studies consider that the sorption dynamics in the filtering medium are governed by four elemental transport phenomena, namely advective transports (contaminants are carried by the airflow passing through the filter), diffusion through the boundary layer separating the bulk air-phase from the adsorbent surface, porous diffusion in the adsorbent, and adsorption/desorption processes at the pore surfaces. The existing models have first been extended for a better representation of the sorption equilibrium of individual species in the field of low concentrations. Then, a multi-component adsorption isotherm model was implemented to account for the interactions between species at the pore surfaces.

Filter model for single contaminants

Under isothermal conditions, the sorption dynamics of a single contaminant in a packed-bed of spherical activated-carbon pellets is described by two equations, one representing the mass balance of the contaminant in the inter-pellet air-phase, and the other representing the mass balance within the pellet:

$$\frac{dC}{dt} = D_x \frac{d^2C}{dx^2} - \frac{d(uC)}{dx} - \frac{1}{\rho_{air}} h_m A_s \left(C - C^*\right) \qquad (1)$$

$$\rho_{air} D_e \left(\frac{d^2 C_p}{dr^2} + \frac{2}{r}\frac{dC_p}{dr} \right) = \rho_s \frac{dC_s}{dt} + \rho_{air} \varepsilon \frac{dC_p}{dt} \qquad (2)$$

In equation (1), the first term of the right-hand side represents the contribution of turbulent axial dispersion. D_x ($m^2.s^{-1}$) is the axial dispersion coefficient, x (m) the axial dimension and C (kg/kg_{air}) the inter-pellet concentration. The second term, where u ($m.s^{-1}$) stands for the interstitial (inter-pellet) air velocity, represents the contribution of advective transports. Finally, the last term represents the contribution of boundary layer diffusion; ρ_{air} ($kg.m^{-3}$) is the air density ($kg.m^{-3}$), h_m ($m.s^{-1}$) the convective mass-transfer coefficient of the contaminant, A_s (m^2) the surface area of the pellets exposed to the bulk air, and C^* (kg/kg_{air}) the contaminant concentration at the pellet surface.

In equation (2), the left-hand side represents the radial diffusion within the adsorbent pellet. The terms on the right-hand side represent the contaminant accumulation in the air-phase of the pores, and the contaminant accumulation at the pore surface due to adsorption/desorption. C_p (kg/kg_{air}) is the pore air-phase concentration at radius r (m), D_p ($m^2.s^{-1}$) the diffusion coefficient in the pores, C_s ($kg.kg^{-1}$) the sorbed-phase concentration, and ρ_s ($kg.m^{-3}$) the activated-carbon density.

The continuity between the two equations is given by $C_p(r=R) = C^*$, where R (m) is the pellet radius. Moreover, since the adsorption/desorption transports at the pore surfaces are much faster than the diffusion transports, C_s and C_p can be considered to be always in equilibrium and related by the adsorption isotherm of the contaminant and adsorbent system:

$$C_s = f(C_p) \text{ or } C_p = f^{-1}(C_s) \qquad (3)$$

The Dubinin/Radushkevic equation is usually taken as the adsorption isotherm model for its predictive characteristics (*i.e.* the model parameters for any gas can be assessed from its physical properties, and the temperature dependence of the sorption equilibrium is included in the model). However, this model is not suitable for many indoor air applications since it isn't defined at $C_p = 0$. It will therefore fail in representing the behavior of filters at the first times and/or when the contaminant air-phase concentrations approach zero. To get round this problem, as well as probably better fit reality, the simulations presented hereafter were carried out considering a linear adsorption isotherm model, $C_s=K_pC_p$, where K_p (kg_{air}/kg) is the so-called partition coefficient of the contaminant / adsorbent system.

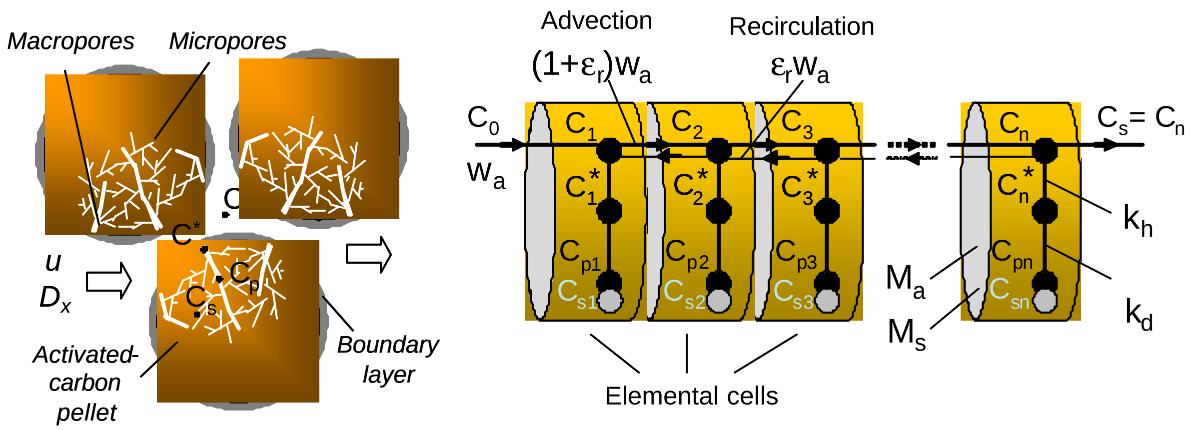

Figure 1: Schematic representation of the discrete filtering medium

To correctly account for the concentration gradients in the direction of the airflow on the one hand, and within the spherical pellets on the other hand, equations (1) and (2) are first discretized by decomposing the filter into n elemental cells connected in series (Figure 1); the concentrations C, C^*, C_p and C_s are assumed to be uniform within a same cell, but are of course different. Then, by using the linear driving force (LDF) assumption to describe the contaminant diffusion in the pores, equations (1) and (2) transform to the following equations where the only unknowns are the inter-pellet air-phase concentrations and the sorbed-phase concentrations in each cell, C_i and C_{si}, respectively:

$$\left|\begin{array}{ll} M_{a1}\dfrac{dC_1}{dt} = w_a C_0 - w_a(1+e_r)C_1 + w_a e_r C_2 - \dfrac{K_d K_h}{K_d + K_h}\left[C_1 - f^{-1}(C_{s1})\right] & \\ M_{ai}\dfrac{dC_i}{dt} = w_a(1+e_r)C_{i-1} - w_a(1+2e_r)C_i + w_a e_r C_{i+1} - \dfrac{K_d K_h}{K_d + K_h}\left[C_i - f^{-1}(C_{si})\right] & \text{for } i=2,n-1 \\ M_{an}\dfrac{dC_n}{dt} = w_a(1+e_r)C_{n-1} - w_a(1+e_r)C_n - \dfrac{K_d K_h}{K_d + K_h}\left[C_n - f^{-1}(C_{sn})\right] & \end{array}\right.$$

$$M_{si}\frac{dC_{si}}{dt} = \frac{K_d K_h}{K_d + K_h}\left[C_i - f^{-1}(C_{si})\right], \qquad (4)$$

M_{si} (kg) and M_{ai} (kg) are the mass of adsorbent and the inter-pellet mass of air in cell i, respectively; w_a (kg_{air}/s) is the airflow rate passing through the filter, and C_0 (kg/kg_{air}) the contaminant concentration at the filter inlet. Given that concentrations are thought to be very low compared to industrial applications, only the possible air recirculations between adjacent cells are considered for axial turbulent diffusion. The latter are characterised through e_r, a user-defined parameter ranging from 0 to infinity. $K_h = r_{air}A_s h_m$ and $K_d = 15 M_s D_p s/R^2$, with s being the slope of the adsorption isotherm at concentration C_p, ($s = K_p$ for a linear isotherm), represent the boundary layer diffusion rate and the diffusion rate in the pores of the pellets, respectively [1]. Each of the filter cells described by equations (4) was implemented as a mass transport component using the dynamic system simulation program Matlab/Simulink.

Filter model for contaminant mixtures

Except in the case of water vapour, contaminant concentrations in the outdoor as well as the indoor air seldom exceed few tens of $\mu g/m^3$. Consequently, it can reasonably be assumed that species do not interact when diffusing, but only compete for sorption at the pore surfaces of the adsorbent. In the frame of a study dealing with the ability of activated-carbon filters to clean the fresh air in heavily-polluted environments, Blondeau et al [5] used the volume exclusion theory as multi-component adsorption isotherm model. This model is more or less an extension of the Dubinin/Radushkevic isotherm for single gases and thus also fails numerically when the concentrations approach zero. Here, multi-component facilities were added in the model described above by implementing the extended Langmuir equation in equations (4). This equation expresses that the sorbed-phase concentration of a given species j, depends on the air-phase concentrations of all components k of the contaminant mixture through the relation:

$$C_{sj} = f\left(C_{pk,k=1,n_p}\right) = \frac{C^0_{sj} K_{Lj} C_{pj}}{1 + \sum_{k=1}^{n_p} K_{Lk} C_{pk}}, \qquad (5)$$

where C^0_s (kg.kg^{-1}) and K_L (kg_{air}.kg^{-1}) are the Langmuir equation parameters for the individual species. The slope of the adsorption isotherm for a contaminant j is then given by:

$$s_j = \left(\frac{\partial C_{sj}}{\partial C_{pj}}\right)_{C_{pk,k\neq j}} = C^0_{sj} K_{Lj} \left(\frac{1}{1 + \sum_{k=1}^{n_p} K_{Lk} C_{pk}} - \frac{K_{Lj} C_{pj}}{\left(1 + \sum_{k=1}^{n_p} K_{Lk} C_{pk}\right)^2} \right) \qquad (6)$$

Practically, implementing the extended Langmuir equation thus contribute to couple the mass transport equations: for n elemental cells and n_p contaminants in the air, the problem to solve is a set of 2 x n x n_p coupled differential equations.

Model parameters

As described by equations (4) to (6), the parameters of the models are the boundary layer mass transfer coefficients, h_m, the diffusion coefficients, D_p, and the sorption coefficients, K_p (single contaminant model) or K_L and C^0_s (multi-contaminant model).

Various correlations between dimensionless numbers can be found in the literature to determine the boundary layer diffusion coefficients in packed-beds of adsorbents. Two were considered here, the Ranz-Marshall (RM) and the Wakao & Smith (WS) correlations [4]:

$$\text{Ranz-Marshall}: Sh = 2.0 + 0.6Sc^{1/3}Re^{1/2} \qquad \text{Wakao \& Smith}: Sh = 2.0 + 1.1Sc^{1/3}Re^{0.6} \quad (7)$$

In these expressions, the Sherwood (Sh), Schmidt (Sc) and Reynolds (Re) numbers are defined by:

$$Sh = \frac{2Rh_m}{D_m},\ Sc = \frac{\mu_{air}}{\rho_{air}D_m} \text{ and } Re = \frac{2RG}{\mu_{air}}, \qquad (8)$$

where D_m ($m^2.s^{-1}$) is the molecular diffusivity of the contaminant in the air, μ_{air} ($Pa.s^{-1}$) the air viscosity, and G ($kg.m^{-2}.s^{-1}$) the mass flux per unit cross-sectional area of empty bed.

If neglecting both solid and surface diffusion transports, diffusion in the adsorbent pellets reduces to air-phase diffusion in the porous interstices. The latter is characterized by the effective diffusion coefficient of the contaminant/adsorbent systems, D_e ($m^2.s^{-1}$), which can be assessed from the relations [3]:

$$D_p = D_e = \frac{1}{\tau}\left(\frac{1}{D_m} + \frac{1}{D_k}\right)^{-1} \text{ and } D_k = 97r_p\sqrt{\frac{T}{M}}, \qquad (9)$$

where D_m and D_k ($m^2.s^{-1}$) are the molecular and Knudsen diffusivities of the contaminant, respectively; T(K) stands for temperature, M ($kg.mol^{-1}$) for the molecular weight of the contaminant, r_p (m) for the macropore mean radius, and τ for the tortuosity factor of the pores.

COMPARISON BETWEEN EXPERIMENTAL AND NUMERICAL RESULTS

As a way to assess the reliability of the models, and to identify the needs for future developments, experiments carried out on a packed-bed of commercial activated-carbon pellets (mean radius R = 1.1 mm) were simulated under Matlab/Simulink. These experiments were small-scale laboratory experiments aimed to determine the breakthrough curves of six contaminants pertaining to different chemical classes. The tests were first carried out with each gas isolated in the air, and then with all the gases in mixture. In all cases the airflow rate was 0.610 kg/h and the temperature was set to 20°C±2°C. In the absence of reliable sorption data for the activated-carbon tested, the partition coefficients K_p were determined from the sorption equilibrium obtained experimentally: after the filter breaks, the air-phase concentration is the same throughout the filter and equal to the concentration at the filter inlet, C_0. The corresponding equilibrium sorbed-phase concentration, and subsequently the partition coefficients, can be assessed from the total mass of gas adsorbed by integrating over time the contaminant masses entering and leaving the filter, that is:

$$K_p = \frac{C_s}{C_0} = \frac{m_s / M_s}{C_0} = \frac{1}{C_0 M_s}\left(\frac{w_a}{\rho_{air}}C_0 t_p - \frac{w_a}{\rho_{air}}\int_0^{t_p} C(t)dt\right) \qquad (10)$$

Equivalent K_L and C_s^0 values were also determined from the computed K_p to feed the multicomponent models. Table 1 summarizes all the model parameters corresponding to the single contaminant and multi-contaminant tests. To simulate the sorption dynamics of a new filtering media for each test, the concentrations in all cells were set to zero at time t=0 (initial conditions of the models). In all cases, the filtering medium was divided into 50 elemental cells of same length in the direction of flow. The re-circulation factor was set to 0 (ideal piston flow). Given the strong uncertainty on 1) the convective mass transfer coefficients (the coefficients obtained from the RM correlation are about three-fold the ones returned by the WF correlation, see Table 1), 2) the actual convective surface area (A_s is lower than the surface area of a pellet times the number of pellets due to contacts between pellets), and 3) the sorption coefficients, five simulations were carried out for each isolated contaminant: first, the uncertainty on K_h was accounted for by considering the reference values of K_p (and resulting K_d, see Table 1), and repeating the simulations for $K_{h,\max} = K_{h,ref} = \rho_{air} A_{s,\max} h_m^{WF}$ and $K_{h,\min} = \rho_{air} \frac{A_{s,\max}}{2} h_m^{RM}$. Then, three other sets of simulations were carried out considering $K_h = K_{h,ref}$ and $K_p = K_{p,ref} \pm 25\%$ to account for the presumed uncertainty on the latter parameter. The predicted breakthrough curves as well as the experimental data for the six contaminants are depicted on Figure 2. For the multicomponent case, the simulations were carried out considering the K_h values computed from the RM correlation and the Langmuir coefficients presented in Table 1 (Figure 3).

Table 1 : Model parameters corresponding to the tests

	R134a	Acetaldehyde	Ethanol	Acetone	Cyclohexane	Toluene
M (g.mol^{-1})	102	44	46	58	84	92
C_0 [(1)] (mg/m^3)	2.80 (4)	1.08 (1)	15.65 (20)	1.06 (1)	1.25 (1)	1.07 (1)
D_m (m^2/s)	1.00 10^{-5}	1.28 x 10^{-5}	1.24 x 10^{-5}	1.21 x 10^{-5}	7.80 x 10^{-6}	8.04 x 10^{-6}
D_k (m^2/s)	2.60 x 10^{-4}	3.96 x 10^{-4}	3.87 x 10^{-4}	3.45 x 10^{-4}	2.86 x 10^{-4}	2.74 x 10^{-4}
D_e (m^2/s)	2.41 x 10^{-6}	3.10 x 10^{-6}	3.00 x 10^{-6}	2.92 x 10^{-6}	1.90 x 10^{-6}	1.95 x 10^{-6}
K_p (kg$_{air}$/kg$_{AC}$)	86	1331	610	8314	25784	123642
$C_s^{\,0}$ (kg/kg$_{AC}$)	0.542	0.2667	0.6372	0.3864	0.5315	0.7136
K_L (kg$_{air}$/kg)	260	5000	970	25000	50000	200000
Tests with the contaminants isolated in the air						
M_s (g)	21.35	22.3	16.3	16.3	8.41	8.43
M_a (g)	0.0168	0.0084	0.0116	0.0116	0.0072	0.0072
h_m (m/s) - WF	1.24 x 10^{-1}	1.49 x 10^{-1}	1.45 x 10^{-1}	1.43 x 10^{-1}	1.04 x 10^{-1}	1.06 x 10^{-1}
h_m (m/s) - RM	4.75 x 10^{-2}	5.69 x 10^{-2}	5.56 x 10^{-2}	5.46 x 10^{-2}	3.96 x 10^{-2}	4.05 x 10^{-2}
K_h (kg/s)[(2)] - WF	0.0193	0.0242	0.0172	0.0170	0.0064	0.0065
K_h (kg/s)[(2)] - RM	0.0074	0.0092	0.0066	0.0065	0.0024	0.0025
K_d (kg/s)	228	4710	1528	20328	20967	103886
Test with the mixture of contaminants – M_s = 12.95 g, M_a = 0.0078 g						
h_m (m/s) - RM	4.28 x 10^{-2}	5.31 x 10^{-2}	5.21 x 10^{-2}	4.72 x 10^{-2}	4.04 x 10^{-2}	4.03 x 10^{-2}
K_h (kg/s)[(2)] - RM	0.0040	0.0050	0.0049	0.0044	0.038	0.0038
K_d (kg/s)	*Calculated at each time-step as a function of the predicted concentrations*					

[(1)] Mean concentration measured during the tests - the concentration sets are put in brackets
[(2)] Calculated based on $A_{s,max}$ (surface area of a pellet times the number of pellets)

If focusing first on the only numerical results, Figure 1 shows that decreasing K_h tends to level the predicted breakthrough curves. Increasing K_p also contribute to level the profiles but overall to delay the predicted breakthrough time of the filter. Then, it can be noted that the predicted concentrations at the filter outlet correctly fit the experimental data: the predicted breakthrough time agree with the measured ones, and the predicted concentrations are within the combined uncertainty range of K_h and K_p. It is important to emphasize here that only the sorption equilibrium state obtained experimentally were used to draw the K_p coefficients, and therefore the breakthrough profiles are not mathematical fits of the experimental data. The only contaminant for which the results aren't fully satisfactory is R134a: the breakthrough time is correct but the predicted concentration profile clearly do not resemble the measured one. Of all the possible explanations, the most likely is that the linear adsorption model fails for R134a due to its very high vapor pressure. Despite the strong uncertainty on the parameters of the Langmuir adsorption isotherms, the predicted profiles also correctly fit the experimental data for the contaminants in mixture. Exceptions are acetone (breakthrough time overestimated of about 200 h) and acetaldehyde. In the latter case, it is interesting to note that the predicted breakthrough time seems to be correct but the filter surprisingly breaks at a concentration which is about half the inlet concentration. There are few doubts that this phenomenon originates from a binary surface reaction which contribute to consume acetaldehyde (and which is of course not accounted for in the model!). Unfortunately the tests were stopped two early to determine if toluene could be the other reactant involved in the reaction.

Beyond the problems of model validation and future numerical developments, all the results presented here underline the strong discrepancies in the breakthrough time of the filter from one contaminant to another. These breakthrough times vary from few hours for R134a and ethanol to several thousands of hours for toluene, which inevitably poses the question to know which contaminant to consider for maintenance issues. Should the life of the filtering media be based on high affinity contaminants such as toluene, with the consequence that the filter will be ineffectual in removing many other species contained in the air, or should it be based on lower affinity contaminants, with the consequence of much higher operating cost?

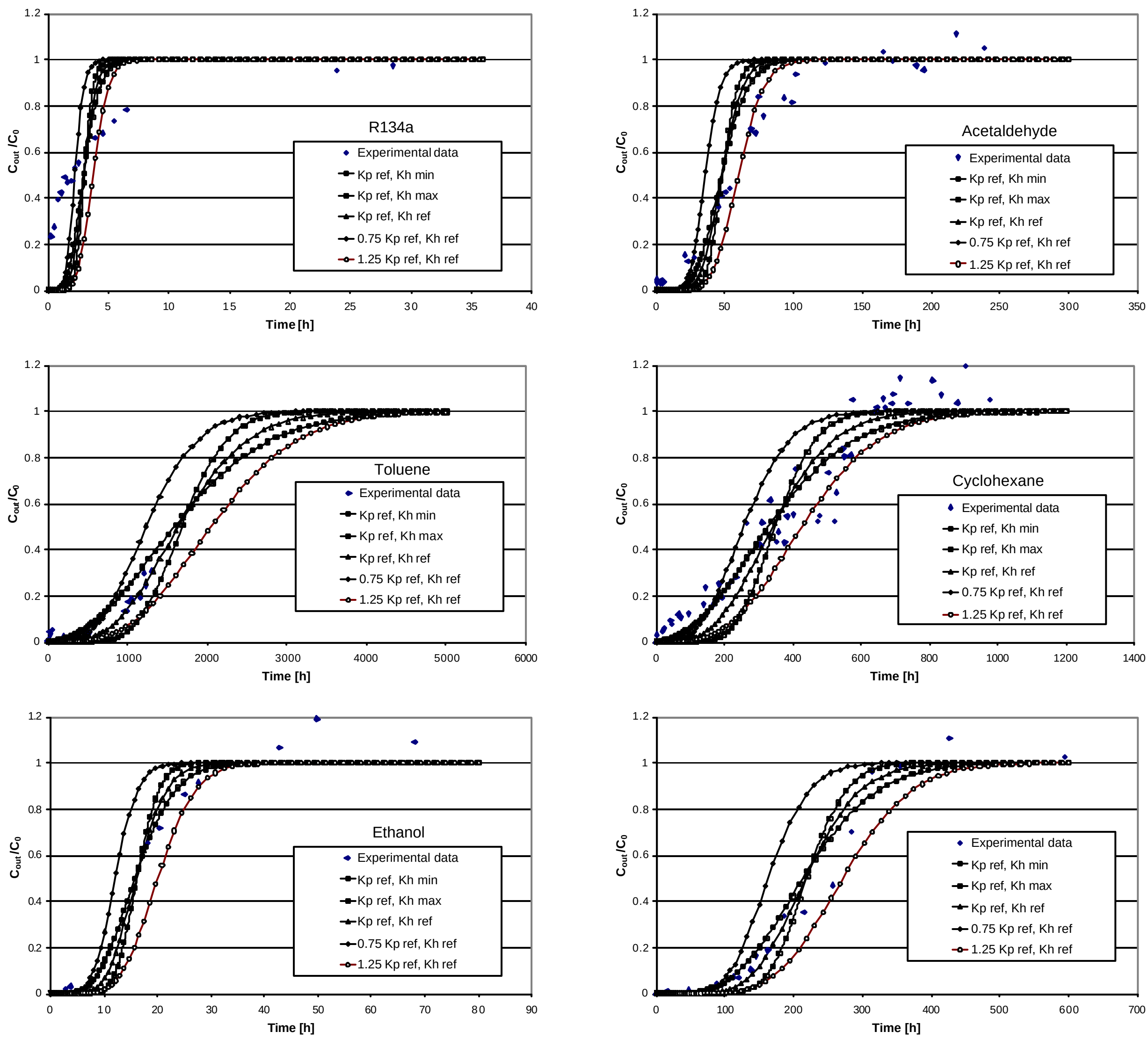

Figure 2: Breakthrough profiles of the contaminants isolated in the air

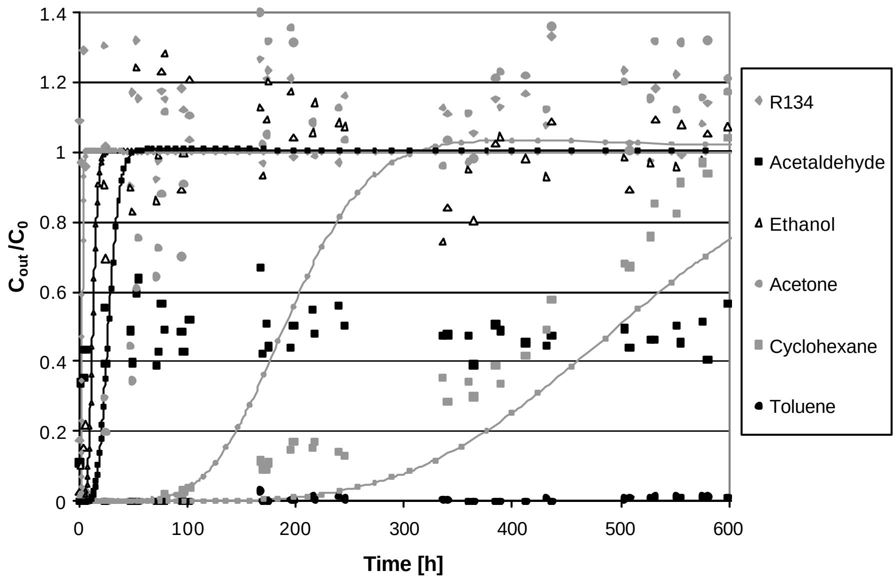

Figure 3: Breakthrough profiles of the contaminants as a mixture in the air

CONCLUSION

The results presented in this paper are a first step in developing dynamic adsorption filter models capable of accounting for all the elemental phenomena involved in the sorption process on the one hand, and the operating conditions characterizing building applications on the other hand. The predicted breakthrough profiles under dry and isothermal conditions globally fit the experimental data, which is a necessary condition for further developments of the models. Priority studies will now be directed toward coupling the mass –transfer equations with equations representing the thermal transfer in the filtering media, implementing functionalities to represent the influence of air humidity (absorption/desorption models, humidity-dependent diffusivities, …), and adding surface chemistry models. It is important to note that progressing toward reliable design tools also needs more experimental data on surface reactions (models exist but very few is known about the species involved and the reaction kinetics) as well adsorption isotherms of contaminants in the range of (very) low concentrations.

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Filtering ultrafine particles and gases – targeted clean air

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SUMMARY

We have performed studies onboard cruise ship to study the particle number concentrations, the commonly used filters and the utilization of novel filtration technologies. We observed that the majority of particles in the indoor air of a ship - and this applies when the ship was sailing in clean or polluted outdoor air - are smaller than 1 micron, and mostly of the of the sizes of combustion particles (ultrafine particles). We also observed that the commonly used filters mainly filtered particles larger than 1 micron. Thus we were able to improve the indoor air quality dramatically by introducing a new filtration technology that filters efficiently not only particles but also gas/vapour phase impurities. The same technology was further applied in an urban office to generate a clean breathing zone, and the results were promising: The breathing zone air was cleaned with up to an efficiency of 73,6%. Also subjective results from the people involved in the test were positive

INTRODUCTION

Indoor particulate and gaseous pollutants cause health problems and make living or working uncomfortable. Recently researchers have shown that there is a direct relationship between the productivity of employees and the indoor climate. Even though studies of outdoor pollutant concentrations and their drifting to indoor environment are abundant, and the awareness of the adverse health effects increase, the utilization of efficient filtration does not.

Ambient particle concentrations and their health effects

Studies of aerosol number size distributions show that the majority of the particles are in the so-called ultrafine mode, i.e. their diameter is less than, or of the order of, 100 nm [1,2]. E.g. Car engines and burning processes emit small particles. Especially the diesel exhaust particles have been shown to cause health problems: the ultrafine diesel particles increase response to allergens and have been proven to be carcinogenic [3,4]. Filtration classes better than EU 7, rather even HEPA class, should be used to efficiently remove the ultrafine particles.

Gaseous phase pollutants

Gas and vapor pollutants are in most buildings not filtrated at all, and thus compounds originating outdoors or emitted indoors are carried around the premises through the HVAC system. Compounds that have recently been studied are e.g. ozone and volatile organic compounds (VOC). Ozone episodes are regular in European cities and VOCs are emitted when e.g. fossil fuels are burned. There are also several indoor sources of VOCs. Much studied VOCs are the benzenes, which are known carcinogens – exposure to benzene is e.g. related to the development of leukaemia and lymphoma.

Re-circulation

In many locations circulation of air is considered the most energy efficient method. Re-circulation, however, also circulates the impurities emitted indoors. By filtering air at the return air units or in a Fan Coil Unit (FCU) it possible to increase the level of comfort and to protect the HVAC system. We have studied different levels of return air filtration and the consequent effect on particle concentration and e.g. sensed odours.

Studies in real environment

Researchers have reported high particle numbers (Aitken and Accumulation mode) measured indoors in urban environment. Several studies also report that these are the most abundant modes in marine air. Thus we saw a need to study indoor air of a cruiser ship. In a cruiser ship the ratio of re-circulated air is usually high – the objective of our study was to determine whether there are significant indoor sources and to what extent are the ambient particles transported indoors. And further to study the effect of different filters in reducing the particle concentrations.

The marine studies also offered us a kind of a micro-environment whose results can directly be applied to land applications and especially to cases where re-circulation of air is used: In some buildings it is not possible to increase the power of the HVAC system to meet the clean air requirements of e.g. people with allergies. For example in a premise with a suspected mould problem, where the actual source of the spores has not been located, clean air should be supplied to the person's breathing zone without a need to increase general air supply. Such an approach has also been studied in an office with good results. We wanted to study the actual particle concentrations, but also the sensed effect of the improved indoor air: does the decrease in indoor pollutants affect the subjective results i.e. make the test persons feel well in their current workplace/station.

METHODS

Ship measurements

The ship study was conducted on a large cruiser ship operating on the Caribbean seas. The trial project phases:
1.) Pre-study and installation of the filters and sampling lines: a) the pre-existing situation was studied through measuring the particle concentrations in airflows of four fan coil units. For this sampling lines were installed into the fan coil units; b) improved particle and gas filters were installed into two of the selected fan coil units; c) the aerosol particle measurements were repeated for each of the four fan coil units to define the short-term effect of the filters.
2.) The concentration of aerosol particles was measured every 1-3 weeks by the ship personnel for a period of six months.

Four sampling lines were installed into each fan coil unit: 1) air that was returned from the cabin to the fan coil unit, 2) return air after the filter in the fan coil unit; 3) the air coming from the main air handling unit (AHU); 4) air mixed of the filtered return air and the air from the air handling unit (supply air to the cabin).

The advanced filter technology was a filter bag that consisted of filter media for both particles and gases. The particle filter media is pre-charged, and thus capable of collecting also submicron particles. The gas (and vapour) media is based on activated carbon. Both filter medias are manufactured in a way that enables a very low pressure drop (filtration efficiency EU7). In one of the cabins also a particle charger was used to gain an even higher filtration efficiency; a particle charger in front of the filter bag results in a HEPA-class efficiency.

Aerosol particle number concentrations were measured with an optical particle counter. The device calculates the number of 0.3, 0.5, 0.7, 1.0, 2.0 and 5.0 mm particles per a litre of sample air, it also measures the relative humidity and temperature. The main focus of the trial was, on sub-micron particles that penetrate traditional filters. Such particles are most abundant in all environments and have been proven to cause adverse health effects.

Measurements in an office

Measurements of the effect of a local clean air breathing zone were conducted in an office in Helsinki, Finland. The office is going to be renovated in the near future, due to suspected mould problems. The office room is used by 4-5 people, out of whom two have asthma and one allergies.

A office desk, with an integrated air filter system, was installed into the office room for the duration of the tests (5 weeks). The filtration system re-circulated air, ths acting as a local air cleaner. The desk had two clean air nozzles leading air from the filter unit to the breathing zone of the worker using the desk. The distance between the nozzles was 120 cm, and they were situated at even distances on both sides of the computer keyboard (which was considered the centre point of the desk).

The effect of using the filter unit was measured using an optical aerosol particle counter, with main focus on 0,3 μm particles. Measurements were performed above the keyboard at the height of the face of the person using the desk. Effect of different flow velocities through the filter system were studied.

During the study the users of the desk was interviewed weekly to find out the subjective response to the filtering of air.

RESULTS

Ship measurements

The installation took place during a dry dock of the cruise ship. The shipyard is located in the vicinity of a European city. During dry dock the measurements the median ratio of the concentration of 0.3 micrometer particles of the supply air to the cabin ("to the cabin" = filtered + air from AHU) compared to the concentration of the particles in the air from the AHU was 0.90 for cabin A (min 0.81, max 4.14), 0.41 for cabin D after installation of the filter (min 0.21, max 0.51), 0.71 for cabin B (min 0.50, max 0.88), and 0.46 for cabin C after the deployment of the filter (min 0.36, max 0.58). This shows that at the dry dock the air from the AHU was usually the biggest source of small aerosol particles. It also shows that even though the filtration efficiency of the advanced filters was on the average 95% for the filter bag, and 97% for the filter bag & charger, the air entering the cabin can only be filtered up to

the ratio defined by the air flows: the amount of air from the AHU with respect to the amount of circulated air from the cabin.
Already during the pre-measurement phase few occasions with very high emissions of particles within the cabins were observed. They were related to people getting ready to go ashore: Perfumes, hairspray, fixing of clothes and similar sources are the likely causes of the detected high particle numbers. Similar events occurred occasionally through out the study period, but mostly the particles of the air from AHU outnumbered the indoor emissions.

We observed that at all times the number of the particles increases dramatically with decreasing particle size. This is true for all measured samples: air that enters the FCU from the cabin, air that has been filtered in the FCU, the so called fresh air that comes from the main AHU and the air supplied to the cabin (mixed of filtered air and fresh air). Fluctuations in the particle numbers are due to the different environments the ship has sailed at: the number of 0.3 micron particles in the fresh air (i.e. the air that came from the AHU) ranged from 2 000 particles per litre up to more than 250 000 particles per litre. During the trial the AHU contained filters that are comparable to the standard filters used in the fan coil units. Examples of the measured particle number concentrations are shown in Figures 1 and 2.

Filtration efficiencies for the studied filters are shown in Table 1. The median is used due to the few occasion when the filters seemed to emit particles – such data points would affect the calculation of averages to an unrealistic level: e.g. for the standard filters in cabin A, the median filtration efficiency for 5 um particles was 67%, but the average only 42%. For the advanced filters the average and the median did not differ remarkably.
Please note that the calculated filtration efficiencies for the largest particle sizes (1, 2 and 5 micrometers) were not representative at all times: occasionally the sample air only contained very few of these particles and thus the statistics during such times is very poor. The largest particles might also in some cases deposit on the surfaces of the FCU and then the observed particles are rather due to re-suspension than penetration through the filter at the measurement time.

Table 1. Median filtration efficiencies for the filters in the fan coil units

Cabin & filter	0.3 μm	0.5 μm	0.7 μm	1.0 μm	2.0 μm	5.0 μm
A, Standard	2%	2%	5%	12%	20%	67%
B, Standard	1%	2%	5%	9%	17%	50%
C, Advanced filter bag	91%	95%	96%	97%	98%	100%
D, Advanced filter bag &Charger	97%	97%	98%	98%	99%	100%

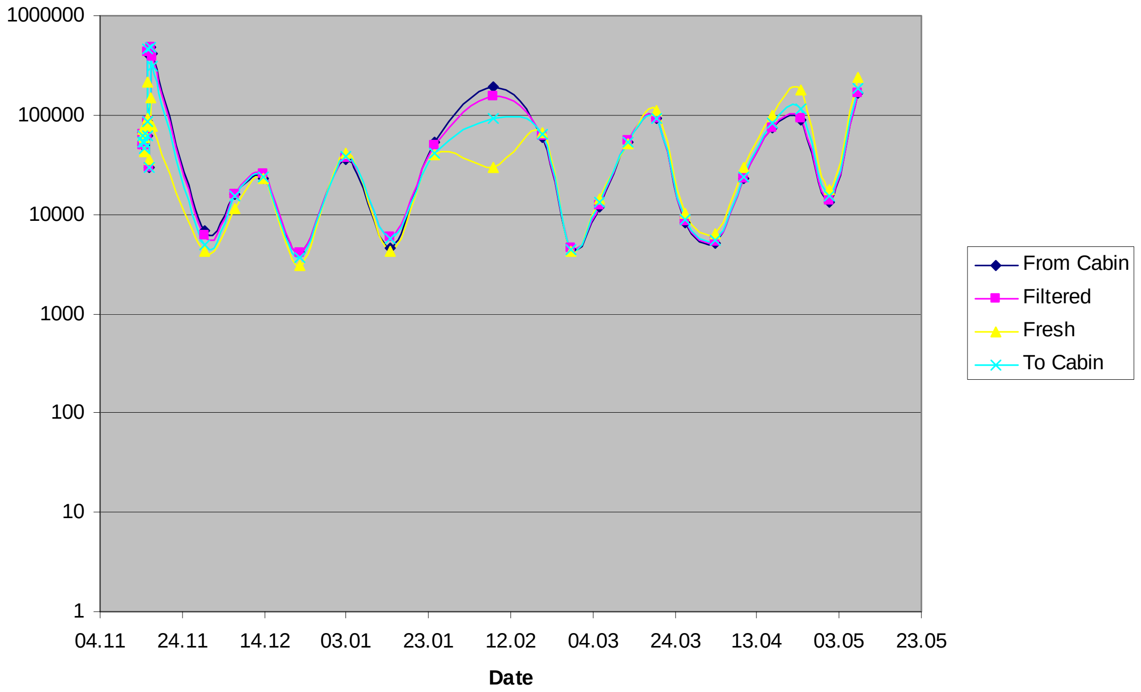

Figure 1. Measured concentrations of cabin A (with standard filter): on vertical axis the number concentration of 0,3 µm particles in a litre of air, on the vertical axis the measurement time.

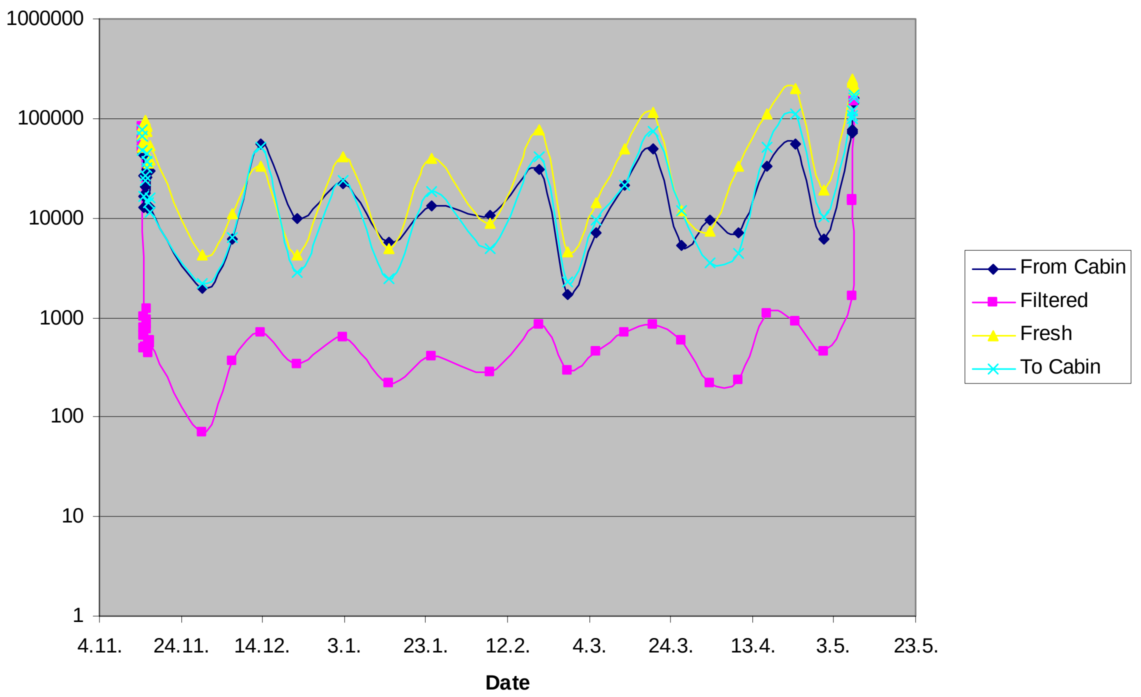

Figure 2. Measured concentrations of cabin D (with advanced filter and the particle charger): on vertical axis the number concentration of 0,3 µm particles in a litre of air, on the vertical axis the measurement time.

Measurements in an office

The effect of the air filtering desk was remarkable. The highest cleaning efficiency (for 0,3 μm particles) within the breathing zone was 73.6% when the system was running on maximum air volume. The cleaning efficiency is defined as the ratio of particles in the breathing zone when the filter system is running compared to the number concentration of particles in the breathing zone when the system is switched of.

Table 2. Average cleaning efficiency of the filter unit integrated to the office desk.

	Cleaning efficiency	Filtration efficiency of the filter unit
Lowest volume flow through the filter	52.1 %	99.55%
Highest volume flow through the filter	73.6%	99.45%

The interviews revealed that during the trial period while working by the desk the symptoms the worker suffered from occurred less frequently or vanished completely.

DISCUSSION AND CONCLUSION

All the studies showed that the effect of using good air filters is dramatic. In the ship, especially when the ship visited urban harbours, ambient pollution is remarkable and thus good filtration should be placed on both the main air handling unit and in to e.g. fan coil units. Filters in the fan coil units reduce the negative effects of recycled air (i.e. removes the indoor emissions)

The office study showed that air in challenging indoor air environments (with respect to particle sources that can not be located & removed) a local clean air zone drastically improves the air quality. The breathing zone air was cleaned with up to an efficiency of 73,6%. Also subjective results from the people involved in the test were positive: they had less symptoms and hence less sick-leaves even though they continued to work in an environment that earlier caused the symptoms to occur. Whether the positive subjective results are due to the reduced air impurities or the psychological effect of the efforts to improve the situation, can not be determined based on the short study. However, if the intention is to minimize the amount of particles in the breathing zone of the work place and hence improve the well being of an employee, then the results are conclusive both based on measurement data and subjectively.

ACKNOWLEDGEMENT

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Combining air filtration with ultra-fine particle sensing for an enhanced energy-efficient indoor air quality optimization

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SUMMARY

A satisfactory indoor air quality (IAQ) relies, amongst other things, on the availability of clean ventilation air. The outdoor air cleanliness in many urban environments is far from optimum. Fine particles (FPs ≤ 2.5 μm) and certainly ultra-fine particles (UFPs ≤ 0.3 μm) feature prominently as hazardous constituents of common urban air pollution. Installed filters in ventilation units of buildings and cars can clean mechanically supplied ventilation air. However, because affordable particle sensors of sufficient reliability and robustness are not commercially available, the degree to which the applied ventilation and filtration strategy is able to control the absolute indoor pollution level at any time remains usually unknown. This paper introduces a UFP sensor capable of recording the ambient UFP pollution level, yielding a sensor signal that relates to the relative inhalation-induced health hazard of airborne UFPs. The monitoring functionality of this sensor can be used for creating awareness with regard to the ambient UFP pollution level. For control purposes, UFP sensors can be used in addition to common T, RH and CO_2 sensors. Furthermore, a new type of electrostatically-enhanced particle filter is presented that accomplishes its air cleaning functionality in a more energy-efficient way than a comparable mechanical filter. Providing air handling units with particle filters as well as with UFP sensors downstream of these filters and/or in indoor spaces allows for a versatile control of air handling units that explicitly takes the indoor UFP pollution level into account as a decision factor for the ventilation strategy. This enables an enhanced sensor-controlled fine-tuning of the IAQ with anticipated savings in power consumption.

INTRODUCTION

In the Western World, most people spend over 90% of their time in indoor environments such as homes, offices and cars. The indoor air quality is therefore very relevant for human health and comfort. Important indoor air quality parameters comprise the temperature (T), the relative humidity (RH), the CO_2 level and the air pollution level [1]. In recent years human productivity has also been found to depend on the indoor air quality [2 - 3], thereby further emphasizing the economic significance of optimized air quality parameters. Several air treatment/handling devices (fans, heaters, coolers, (de) humidifiers, ventilation systems, cleaning systems) are often present to help maintaining an acceptable indoor air quality. However, the energy-hungriness of these devices and rising energy prices have since long promoted energy saving measures such as better home/building insulation and partial air re-circulation by air handling units. An accompanying aspect of the latter measures is a generally decreased ventilation rate. Ample evidence has come forth [4] that the decreased ventilation rates are frequently insufficient for maintaining an acceptable if not optimum indoor air quality, and are at least partly responsible for an overall increasing indoor air pollution level.

Indoor air pollution is directly related to various diseases, feelings of discomfort, reduced work performance and a diminished learning capability at school [4]. Part of the indoor pollution comes from indoor sources such as outgassing building materials and equipment, human bio-effluents, and activities like cooking and smoking. Supplied ventilation air and/or exhaust indeed diminish the pollutant concentration derived from indoor sources through dilution, however ventilation also introduces pollutants from outdoors. Hazardous outdoor pollutant species comprise allergenic particles (e.g. pollen), bioaerosols, and particles/gases originating from automobiles and industrial activities. Especially the fine particles (FP ≤ 2.5 μm) and even more so the ultra fine particles (UFPs ≤ 0.3 μm) derived from combustion processes (e.g. soot particles) are currently receiving much attention from various legislative authorities because of their demonstrated long-term impact on human health [5]. Outdoor particle concentrations vary considerably in time and place but are relatively highest in urban areas and/or close to motorways. As of today, the use of air filters in air handling units possessing a substantial filtration efficiency towards all particle sizes down to 100 nm diameter and below is still the exception rather than the rule. Any existing passive ventilation is anyway accompanied by an unhindered intrusion of pollutants from outdoors.

A rigorous filtration of sufficient amounts of ventilation air is the most direct way of dealing with polluted indoor air, however it is not the most economical approach. An introduction of (localized) ventilation on demand and air cleaning on demand offers better prospects in this regard but depends on the availability of practical and reliable air pollution sensors. Only CO_2 sensors and motion sensors detecting human presence are currently used for enabling ventilation on demand in spaces like meeting rooms and school classes wherein many people reside. Reliable and robust particle sensors or gas (VOC, NO_x) sensors capable of measuring the (relatively very low) absolute gas concentration levels in polluted indoor air have up till now not become available in the market place at an acceptable price level.

The present paper discusses a UFP sensor that can be installed in air handling units and/or in separate indoor rooms. Its output signal can be used to adjust the ventilation and air cleaning strategy to existing needs in a more sophisticated and economical way than what has been possible up till now. Ideally, the UFP sensor is used to complement the information obtained from T, RH and CO_2 sensors and can be integrated in a building automation system.

Whenever the outdoor air has an unacceptable cleanliness level, it is desirable to avoid any natural ventilation and pass all mechanically supplied ventilation air through one or more filters to accomplish a sufficient degree of air cleaning prior to its release indoors. The novel embodiment of the electrostatically-enhanced particle filter presented in this paper can be of help to also accomplish the particle filtration step in a more energy-efficient way than what is achievable when conventional mechanical filters of similar size and performance are used.

EQUIPMENT

UFP sensor

All common particle sensors are currently optical sensors capable of detecting particles down to 300 nm diameter, the most expensive ones down to 100 nm. Because of their price (at least a few thousand Euro’s) and maintenance requirements, their use is limited to monitoring clean

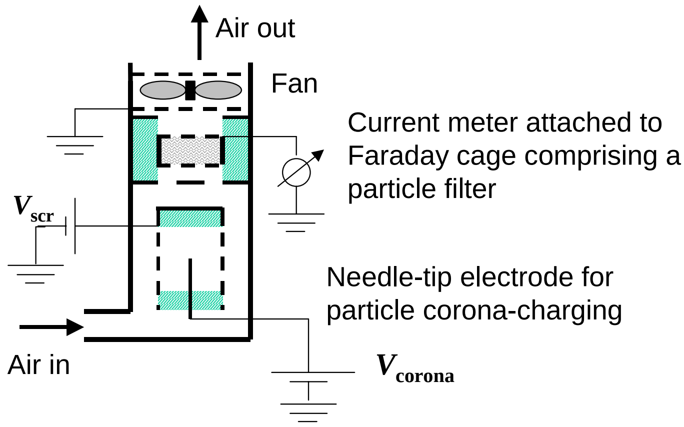

Figure 1:- Schematic design of the UFP sensor

room environments and professional air pollution measurements. Optical sensors are incapable of detecting particles smaller than 100 - 300 nm, even though most combustion-related ultra fine particles, and also cigarette smoke particles, are smaller than 100 nm.

To fill this technology gap, we realized an electrical UFP sensor that is particularly sensitive for detecting hazardous ultra fine particles down to sizes as small as 10 nm. A schematic design of the home-built UFP sensor is depicted in Figure 1. A small airflow ϕ is drawn through the sensor by means of a mini-fan situated atop the sensor. Airborne particles entering the UFP sensor are first charged via diffusion charging [6] with a corona discharge from a needle-tip electrode (set at V_{corona}) that is surrounded by a separate screen electrode whereupon a screen voltage $V_{scr} << V_{corona}$ is imposed. Subsequently, the charged particles are captured within a filter situated inside a conducting Faraday cage. A current meter attached to this cage records the amount of particle-bound charge that deposits inside the filter per unit time as a current I_{sensor}. I_{sensor} constitutes the sensor signal.

Because particles are charged via diffusion charging, one has for the average number of elementary charges "$p(d_p)$" that attach to a particle of diameter d_p the proportionality

$$p(d_p) \propto d_p \tag{1}$$

[6]. In practice, a particle size distribution $\frac{dN(d_p)}{d\ln d_p}$ is encountered in the ambient air wherein $dN(d_p)$ denotes the particle concentration for particles of diameter d_p. An integration of the particle size distribution over all particle diameters d_p yields the total airborne particle number concentration N_{total} (particles/cm^3) according to

$$N_{total} = \int_{d_p=0}^{\infty} \frac{dN(d_p)}{d\ln d_p} d\ln d_p \tag{2}$$

The sensor signal I_{sensor} relates to the ambient particle size distribution and the airflow ϕ (m^3/s) through the sensor according to

$$I_{sensor} = \int_{d_p=0}^{\infty} \phi p(d_p) e \frac{dN(d_p)}{d\ln d_p} d\ln d_p \propto \int_{d_p=0}^{\infty} d_p \frac{dN(d_p)}{d\ln d_p} d\ln d_p \tag{3}$$

$$\propto L_{total}$$

L_{total} (in units m/m^3) denotes the total particle length concentration, i.e. the total length of the string of particles that is created when all airborne particles within a unit volume would be lined up side by side. It is well known that in ambient air, the overwhelming contribution to both N_{total} and L_{total} comes from particles sized between 10 nm and about 200 - 300 nm, i.e. the combustion-derived UFPs. Larger particles ($\geq$ 1 μm) hardly contribute to both N_{total} and L_{total} but are almost exclusively responsible for all airborne particles mass M_{total} (μg/m^3).

The question is now what is learned from knowing L_{total}. It is generally believed that in as far as solid UFPs like soot particles are concerned, the relative health hazard H_{ufp} of inhaled UFPs relates primarily to the deposited UFP surface area $S \sim d_p^2$ in the deep alveolar region of the lung where gas exchange with blood occurs. In contrast with the upper head airways and the intermediate tracheo-bronchial part of the lungs, the alveolar region has significant difficulty cleaning itself from particulate deposits. The UFP surface carries carcinogenic substances like poly-cyclic aromatic hydrocarbons (PAH) and a long residence time of deposited UFPs in the alveolar region adds to their hazardousness. Other than the surface area d_p^2 of a lung-deposited UFP particle, an assessment of H_{ufp} must also take into account the alveolar deposition efficiency of inhaled UFPs as a function of d_p. Figure 2 shows the respiratory deposition efficiency of inhaled particles as a function of d_p in different parts of the airways (see e.g. ref. [6], Chapter 11). Concerning the alveolar region, it can be inferred from Fig. 2 that the alveolar UFP deposition efficiency is approximately proportional to $d_p^{-0.5}$ within the 10 – 300 nm particle size range. The relative health impact H_{ufp} then follows from

$$H_{ufp} \propto \int_{d_p=0}^{\infty} d_p^2 d_p^{0.5} \frac{dN(d_p)}{d \ln d_p} d \ln d_p = \int_{d_p=0}^{\infty} d^{1.5} \frac{dN(d_p)}{d \ln d_p} d \ln d_p \qquad (4)$$

which is more closely related to L_{total} than to N_{total} or M_{total}. Thus, to a first approximation, a measurement of $I_{sensor} \propto L_{total}$ with the UFP sensor can be used to approximately infer the health-hazardousness of the ambient UFP pollution level.

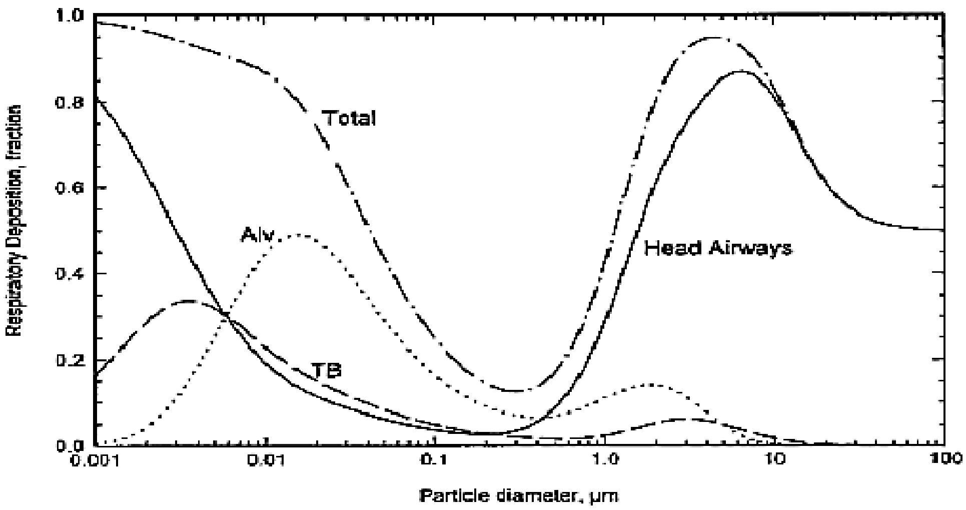

Figure 2:- Predicted total and regional deposition fraction of airborne particles in the human airways during nose-breathing with light exercise according to the International Commission on Radiological Protection (ICRP) deposition model (see e.g. ref. [6] Chapter 11). TB = tracheo-bronchial region; Alv = alveolar region. The total deposition fraction is the sum of the regional deposition fractions in the head airways, the TB region and the alveolar region.

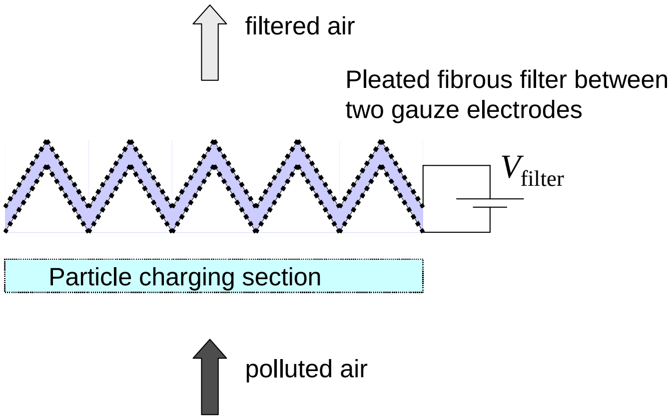

Figure 3:- Schematic design of the electrostatically-enhanced particle filter featuring an upstream particle charging section and a downstream particle filtration section.

Electrostatically-enhanced particle filter
To improve the ventilation air cleanliness, mechanical particle filters can be installed to try removing a substantial portion of airborne particles from air. The incurred pressure drop ΔP across these filters is directly proportional to the HVAC power required for air filtration. It is desirable to have filters of a given size that enable a desired air cleaning efficiency at a minimized power consumption. Figure 3 shows the schematics of a home-designed electrostatically-enhanced fibrous filter. A corona-charging unit is located upstream from the pleated fibrous filter and serves to provide all airborne particles with an electrostatic charge through local air ionization. Importantly, the pleated filter features a filter cloth that is sandwiched between two gauze electrodes possessing a carefully chosen conductivity. By imposing a voltage difference between the gauze electrodes, an electric field is set up across the filter cloth, which was experienced to markedly improve the efficiency with which charged particles can be filtered from air.

RESULTS

UFP sensor
Some typical UFP sensor recordings in indoor air as a function of time are shown in Figure 4 for the extremes of (pre-filtered) laboratory air and of cabin air (pre-filtered by an ordinary cabin filter) in a driving automobile. In all cases, the UFP sensor signal and thus the indoor UFP concentration varies significantly in the course of time, presumably in parallel with the outdoor UFP concentration variation. However the rate of change of the UFP sensor signal in laboratory air is relatively limited whereas fast variations are encountered inside a car cabin due to the rapidly changing proximity of (the exhausts) of other cars. It is evident from Figure 4 that the measured I_{sensor} inside a car and thus the cabin UFP pollution level can be up to 100 times higher than in the “clean” laboratory environment. This is evidence of the relative seriousness of the cabin air pollution level, a situation that most people are not aware off.

The air sampling rate of our UFP sensor ϕ = 0.5 liter/min while the sensor signal I_{sensor} is measurable down to a sensitivity of about 1 – 2 fA within a measuring range 0 – 10 pA. This approximately conforms to a measuring sensitivity of the UFP number concentration of 1000 /cm^3 for UFPs sized in the d_p = 10 - 300 nm diameter range. The average UFP diameter in ambient air is usually between 30 and 70 nm. A UFP concentration as low as 10^3 /cm^3 is encountered only in very clean air (e.g. high in the Swiss Alps). In ordinary sub-urban air,

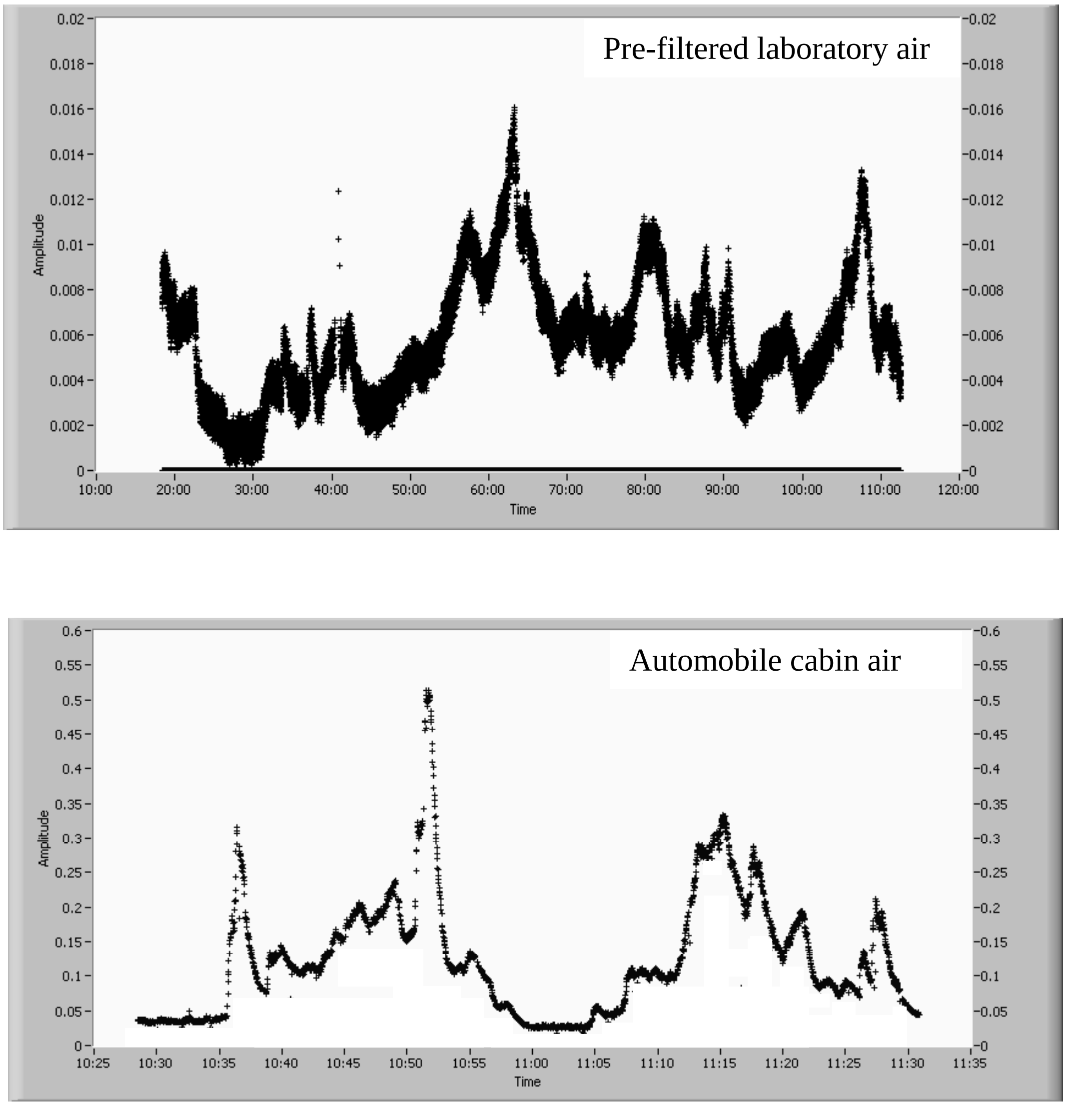

Figure 4:- Measured amplitude of the UFP sensor signal (in units pA) as a function of time (in hours) in a laboratory environment and in the cabin of a driving car.

UFP concentrations are typically close to 10^4 particles/cm^3 (leading to $I_{sensor} \approx 0.03$ pA). Near busy roads, this can rise up to $5*10^4$ particles/cm^3 and up to $5*10^5$ inside the cabin of a car stuck in a traffic queue. The UFP sensor is therefore able to readily discriminate between clean air, moderately polluted air and seriously polluted air.

Electrostatically-enhanced particle filter

An electrostatically-enhanced fibrous filter designed according to the schematics shown in Fig. 4 was realized and subjected to several filtration efficiency tests both in the absence and presence of electrostatic enhancement. For this purpose, the particle size distribution for particle diameters from 10 – 300 nm was measured with a Scanning Mobility Particle Sizing system (Type 5403, Grimm GmbH, Germany) upstream and downstream of the filter at various air flows through the filter. The results are shown in Figure 5.

In the absence of electrostatic enhancement, the filter behaves as an ordinary mechanical filter with a filtration performance that approximately conforms to that of a EU 6 filter, characterized by a poor filtration performance with respect to UFPs larger than 50 nm.

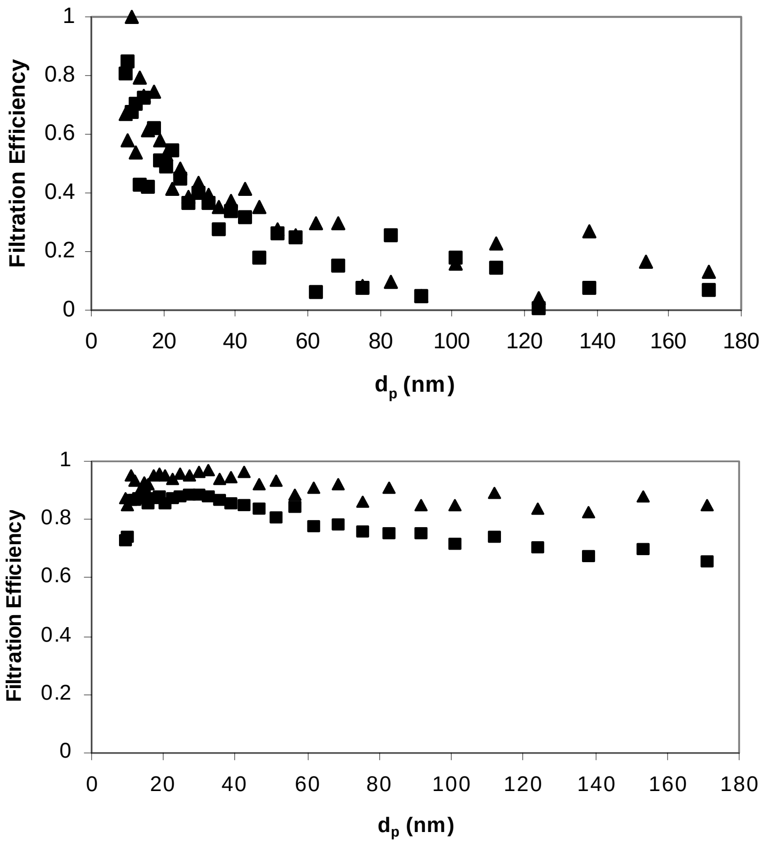

Figure 5:- Measured filtration efficiency of a particle filter before electrostatic enhancement (top) and after electrostatic enhancement (bottom) as a function of the UFP diameter d_p. The superficial airflow speed was set at either 1 m/s (▲) or 2 m/s (■) while the electrostatic enhancement involved both particle charging and the set-up of an electrostatic field E_{filter} = 1 kV/mm across the depth of a fibrous filter cloth.

For particles sized between 100 nm and 200 nm, the filtration efficiency does not exceed 20% at an apparent airflow speed of 1 – 2 m/s through the filter cassette. In the presence of electrostatic enhancement wherein the particles are electrostatically charged and subsequently passed through a fibrous filter across which a field strength $E_{filter} \approx 1$ kV/mm is applied, the filtration efficiency significantly increases up to 70 – 80% for 100 – 200 nm sized particles. Note that this efficiency increase is accomplished at a constant power consumption, airflow speed, and size of the filter cassette. The specific power consumption required for airborne particle charging and filter polarization only amounts to a few W for an air flow of several hundred m^3/hr and is therefore negligible with respect to the total HVAC power consumption.

DISCUSSION

The described UFP sensor yields a real-time output signal proportional to the ambient UFP pollution level according to a metric that approximately corresponds with the relative health-hazardousness of the UFP pollution level. At any locality where the dose of air pollution is received from a more-or-less constant set of pollution sources (e.g. automobile traffic) in the course of time, the UFP pollution level may be expected to at least roughly reflect the total airborne gaseous/particulate pollution level. The UFP sensor readily distinguishes clean air from moderately or seriously polluted air and can be used for assessing pollution levels in outdoor and indoor air. Because the local outdoor and indoor air pollution vary strongly in time and place, it is convenient to continuously monitor these pollution levels with UFP sensors in order to acquire pollution-related decision factors for optimizing the ventilation strategy.

In case all ventilation air is received from a mechanical air handling unit that is supplied with an air filter, it is recommended to position UFP sensors directly downstream of the filter and/or in the indoor enclosure itself away from the supplied ventilation air flow. This allows for a monitoring of the residual UFP pollution level in the supplied ventilation air and/or in the indoor air itself, respectively. Electronic feedback from the UFP sensor, preferably in combination with feedback from other sensors (T, RH, CO_2, human presence), can then be used to enable ventilation on demand, to adjust the total ventilation air flow and/or the ratio between supply air and return air such as to minimize the absolute indoor pollution level while retaining comfortable and productive indoor conditions in an energy-efficient way. The application of an electrostatically-enhanced particle filter in the air handling unit instead of a conventional mechanical filter will help to further minimize the required power consumption necessary for creating a healthy and comfortable indoor air quality.

In case the ventilation air is a mixture of naturally supplied ventilation air and mechanically supplied filtered ventilation air, it is recommended to continuously monitor the UFP pollution both outdoors and indoors. Received sensor signals can then be used to adjust the ratio between natural ventilation and mechanical ventilation either automatically or manually, aiming to minimize the indoor pollution level in an energy-efficient way under all environmental circumstances. It is obvious that the recommended ventilation strategy will usually be the outcome of a judiciously chosen compromise.

Various additional application modes of UFP sensors can be readily construed and applied in homes, offices and cars. A common aspect is always that the very presence of UFP sensors allows for a direct visualization of the UFP pollution level. Except through the application of professional expensive instrumentation, UFP pollution visualization has up till now not been possible and can therefore be of much help in both raising human awareness with respect to the pollution level in inhaled air and in reducing human exposure to indoor air pollution.

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Indoor deposition of fine and ultrafine particles: A full-scale study

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SUMMARY

The aim of this study was the experimental determination of particle deposition for both different particle size fractions and different indoor surface materials. The selected surface materials were glass, gypsum plate, carpet, and curtain. These materials were tested vertically in a full-scale test chamber. Experiments took place in a 32 m^3 chamber with walls and ceiling made of glass. Prior to each experiment the chamber was flushed with outdoor air to reach an initial particle concentration typical of indoor air in buildings with natural ventilation. The decay of particle concentrations was monitored. Seven particle size fractions were studied. These comprised ultrafine and fine particles. Deposition was higher on carpet and curtain than on glass and gypsum plate. Particles in the range from 0.3 to 0.5 µm had the lowest deposition. This fraction also has the highest penetration and its indoor concentration is expected to be closest to outdoor concentrations.

INTRODUCTION

In recent years, exposure to fine and ultrafine airborne particles has been identified as an important risk factor for human health [1]. Deposition of particles onto indoor surfaces such as floors, walls, ceilings, and furniture may be thought of as a phenomenon protecting humans from some of the dangers caused by exposure to particles [2, 3].

Measures to reduce exposure to these particles generally fall into three main categories: source control, ventilation control, and removal control. Source control should always be the first strategy examined. When source control is not feasible or practical, ventilation control should be the next option. But in many of our urban environments today, the outside air itself is one of the main sources of indoor particle pollution. Therefore, when it is clear that neither of the first two control strategies mentioned above would reduce the level of contaminants in the affected space, removal of particles should be applied.

One approach to improve indoor air quality is to select such surface materials as are able to remove indoor particles through increased particle deposition. Understanding particle deposition phenomena provides useful information on human exposure indoors. Particle deposition onto indoor surfaces are complex phenomena and varies with particle size and surface roughness. The deposition rate for each particle size can vary over a few orders of magnitudes [5]. For particle sizes larger than 1 µm, gravitation is dominant, while diffusion is dominant for particles of sizes smaller than 0.1 µm. For particle sizes between 0.1 and 1 µm, a mixture of both mechanisms may be assumed [4].

Experimental and simulation studies reported that several other parameters can influence indoor particle deposition. These parameters are chamber size, airflow rates, ventilation strategies, air velocity and degree of turbulence, furniture, roughness of surfaces, orientation and electrostatic forces [6, 7]. Most of the previous experimental studies were carried out in real buildings. Consequently, it is difficult to control the many factors that influence particle deposition rates simultaneously. Therefore, as a complement to field studies, it is necessary to carry out experimental studies in the chamber under controlled experimental conditions to understand the mechanisms behind deposition phenomena.

These previous studies have gathered information about deposition of generated fine particles onto rough and smooth vertical surfaces in small and full-scale chambers. However, few studies have examined deposition of ultrafine particles from ambient air onto the surfaces of building materials under controlled experimental conditions. The aim of the present study is quantification of deposition rates of fine and ultrafine particles onto three building materials often found indoors.

METHODS

The experiments were carried out in the air quality laboratory of the Danish Building Research Institute (SBi). A full-scale chamber with a volume of 32 m^3 was used. The walls and the ceilings of the chamber consist of panes of glass mounted in aluminium frames and the floor is made of high-pressure laminated fibreboard. Low concentration of polluting gases and particles in the supply air was ensured by the use of a fine filter of class F7, a charcoal filter followed by another fine filter of class F7 and finally a HEPA filter. The full-scale chamber was located in an 1800 m^3 hall that was ventilated with an outdoor air change rate of approximately 4 h^{-1}.

The air change rate in the chamber was 0.66±0.1 h^{-1}. Three table-fans were in operation during all experiments in order to ensure complete mixing of the chamber air. The table fans were located at the same places during all measurements ensuring an average air velocity in the chamber around 0.15-0.5 m/s. A tracer gas decay method was used to determine the air change rate and for verification of the air mixing conditions in the chamber. The tracer gas decay concentration was measured continuously over four hours with a photo-acoustic spectroscopy (PAS) instrument from Innova, type 1302. The temperature and relative humidity were recorded during the experiments. The temperature ranged from 22 to 25 ^{0}C, and the relative humidity ranged from 37 to 53 % RH, see Table 1. An over-pressure of 10 Pa was maintained in the test chamber relative to the surrounding hall throughout the experiments. The air velocity was measured to be 0.37 m/s at the centre of the sample, 5 mm above the material surfaces. The air velocity in the chamber was measured to be 0.16 m/s at 1.30 above floor level.

The selected surface materials for the present study were window glass, gypsum plates, wall-to-wall carpet and cotton curtain. The area of each tested material was 10.4 m^2. It should be noted that the walls of the chamber consisting of glass panes were treated as a surface material, i.e. the window glass. It means that a new window glass was not tested in the chamber. The other surface materials were tested vertically on eight places of the walls of the chamber. Test materials were placed directly on the glass panes of the chamber, thus replacing glass pane with test material. The other tested materials were placed at the same locations on the wall of the chamber.

Supply air was particle free, and an initial high concentration was obtained by flushing the chamber with unfiltered heated outdoor air. Prior to each experiment, the chamber was flushed with unfiltered outdoor air to reach an initial particle concentration typical of buildings with natural ventilation. The door of the chamber was open until the concentration of ultrafine particles in the chamber reached approximately 5000 particles/cm^3. During the experiments the decay of particle concentrations caused by ventilation and deposition was monitored.
The outdoor climate conditions were rather similar in all measurements. The air temperature ranged from 11 to 14 ^{0}C, and relative humidity ranged from 60 to 70% RH.

After flushing, the decay of the particle concentrations were monitored using both a condensation particle counter for ultrafine particles (UFP) counter (TSI model CPC 3007) and an optical particle counter for fine particles (PMS model Laser II-310B). The deposition rates were obtained by linear regression of the measured particle concentration for different size fractions while compensating for particle removal by ventilation. Seven particle size fractions were studied. These comprised ultrafine (particles larger than 0.01 µm) and fine particles in the fractions 0.3-0.5, 0.5-1, 1-3, 3-5, 5-10 and particles larger than 10 µm. Particle concentrations were monitored every minute.

The concentration measurements of UFP were made in the centre of the test chamber, 1.30 m above floor level. The CPC 3007 instrument enables real-time measurement of particle number concentrations and data collection. The size range for particle detection of the instrument is between 0.01 and about 1.0 µm. Software is available for later retrieval of measured data. According to the manufacturer the concentration accuracy of up to 100,000 particles per cm^{-3} is ± 20% of the reading. In this concentration range coincidence is low and no correction is required. At concentrations of up to 4 x 10^5 particles per cm^3, the data could be corrected for coincidence loss, e.g. according to Hämeri et al., (2002).

The concentration measurements of fine particles were also placed at the same location as the UFP. The Laser II-310B is an optical particle counter capable of measuring different particle fractions ranging from 0.3 to larger than 10µm in size. Six size fractions were chosen according to the above-mentioned sizes. According to the manufacturer the concentration, accuracy is approximately ± 20% of the reading.

Table 1. Air change rate, relative humidity and temperature in the chamber during the experiments.

Material	Air change rate h^{-1}	Relative humidity % RH	Temperature (^{0}C)
Empty chamber, glass	0.65	53	24.6
Wall-to-wall carpet, polyamid	0.65	42	24.9
Gypsum plate	0.67	37	22.8
Cotton curtain, recently washed	0.67	42	22.8

Calculation of deposition rate

Analyses of the particle concentration during the step-change of the surface materials with different properties provide information both on the particle concentration and the quantity of deposition effects. The particle decay, observed after the concentration of fine and ultrafine particles reached predetermined levels, was used to assess the time constant for the particle decay, T_P. The time constant was obtained through regression of the measured particle concentration decay. The ventilation time constant, T_v , was obtained by tracer gas measurements. The difference between the inverse values of the ventilation time constant and the particle decay time constant represents the particle deposition rate constant, r, which is given in the figures using the unit h^{-1}.

$$C_{(t)} = C_{(0)} \times e^{-\left(r+\frac{1}{T_V}\right)\times t} = C_{(0)} \times e^{-\frac{t}{T_P}} , \qquad (1)$$

where *C(t)* is concentration at time t (particles/m^3 or particles/cm^3), and *C(0)* is concentration at time zero (particles/m^3 or particles/cm^3). This method has been used earlier in another study (Abt et al., 2000).

The particle deposition rate for each material ($r_{Material}$) was calculated according to the following:

$$r_{Material} = r_{Material+chamber} - r_{Chamber} , \qquad (2)$$

where $r_{Material+Chamber}$ is the sum of the deposition rate for material and chamber, and $r_{Chamber}$ is the deposition rate for chamber surfaces.

Where material samples replaced chamber glass, the test procedure did not change the main geometrical properties of chamber and flow patterns. But as a consequence, it should be noted that the deposition rate for materials ($r_{Material}$) actually represents the material deposition minus the previous deposition on the glass replaced by test specimens. It was decided that the other way of conducting the experiments with freely hanging specimens not replacing chamber glass, would be impossible to analyse since the impact of modifications to flow patterns could impact deposition rates.

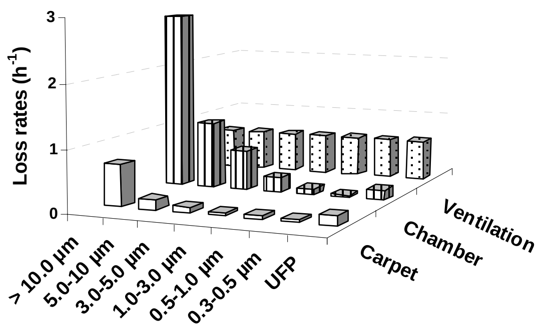
RESULTS

The deposition rates for materials ($r_{Material}$), chamber ($r_{Chamber}$) and ventilation rates are shown in Figures 1a, 1b and 1c. The deposition rates for all three materials together are shown in Figure 1d. Figure 2 shows the deposition rates for sum of material and chamber, i.e $r_{Material+Chamber}$. Figure 2 also shows the deposition rates of the chamber when it was empty.

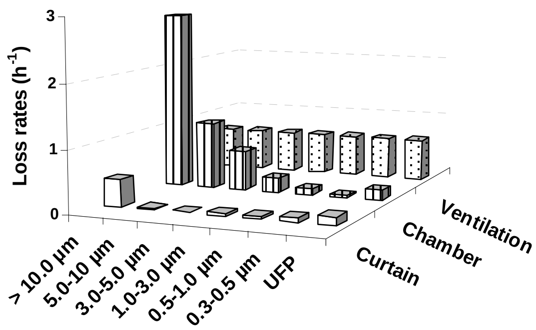
Figures 1a-1d illustrates the deposition rates for the seven particle fractions onto the chamber surfaces when it was empty and onto the tested materials when the materials were placed in the chamber.

Figure 1a shows that the particle deposition onto the chamber surfaces and the fixed carpet on the walls follow a similar pattern and also that the deposition rate varies with the particle size. The deposition rate increased both for larger particles and for smaller particles. The measurements for particles with diameters of 0.3 and 0.5 µm give a smaller loss rate than for the other particle sizes.

Figures 1b and 1c illustrate the deposition rates for the seven particle fractions onto the curtain and the gypsum fixed on the walls, respectively. The results show that the particle deposition onto the curtain and the gypsum do not follow the similar pattern as the deposition rates of the chamber and the carpet. The deposition rates for curtain increased for particles larger than 10 µm and UFP. Particles with diameter 3-5 µm and 5- 10 µm give a smaller deposition rate than for the other particle sizes. The deposition rate for gypsum varies from particle size to particle size. The deposition rates of particle sizes 3-5 µm and 5-10 µm is marked with a circle. Their deposition rates are negative after calculation. It may depend on the fact that gypsum and the glass the samples are replacing have very similar deposition properties. The negative numbers may simply be caused by uncertainties.



a)



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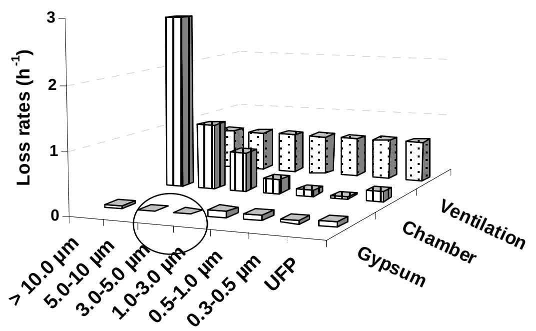

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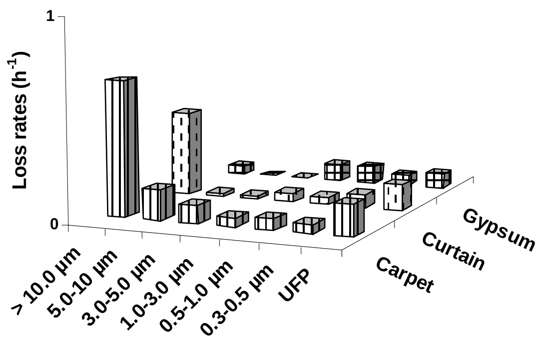

d)

Figure 1. Particle deposition rates onto the surfaces of the chamber and three materials. a) wall-to-wall carpet, b) cotton curtain, c) gypsum plate, d) all three materials

DISCUSSION

The experimental data reveal three different patterns for the tested materials. For the carpet, the deposition rates decrease as particle size increases from the UFP to 0.3 µm, and then begin to increase again. For the curtain, the deposition rates decreases as particle size increases from UFP to 10 µm, then begins to increase. For the gypsum, the deposition rates are very similar to the glass the samples are replacing and no clear trend is observed. The experimental data reveal also that both the surface properties of the materials and the particle sizes have an effect on the deposition rates. The effect of particle size is much stronger than the effect of surface properties. The effect of surface properties is stronger for the UFP and for the particles with diameters larger than 10 µm.

The rather high air velocity in the chamber may have caused an elevated deposition of the larger particles. This may be caused by impaction on surfaces when these particles with some mass inertia have difficulties to follow the flow patterns of the air near the surfaces. At the other end of the particle size range the ultrafine and fine particles have a higher air velocity rate partly caused by collisions with molecules. These Brownian movements cause higher diffusion of the smallest particles and this also causes higher deposition. These two different mechanisms of increasing deposition affecting either the very small or the largest of the particles are causing deposition of the particles in the median range of 0.1-1 µm to be the lowest.

Based on the experimental data illustrated in Figure 2, the deposition rates vary with particle sizes. Figure 2 shows that the measurements for particles with a diameter of 0.3 µm give smaller deposition rates than the other particle sizes. It may depend on particle diffusivity that is dominant for ultrafine particles and sedimentation that is dominant for particles larger than 1 µm. This is in agreement with results from other studies that were carried out in real buildings [9, 10]. A similar pattern found for wall-to-wall carpet, despite that the deposition rates for carpet was over-represented. However, the pattern was not similar for gypsum board and cotton curtain when the deposition rate for carpet was over-represented.

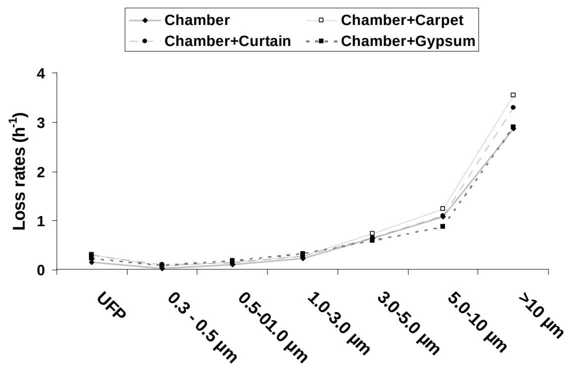

Figure 2. Particle deposition rates onto the surface of chamber and three materials.

The relative importance of particle losses due to ventilation, chamber properties and test specimens follow a typical pattern. For the larger particles down to 5 µm, gravitational settling and other deposition are more important than ventilation and deposition on specimens. For the smaller particles, ventilation losses are more important than deposition on chamber and test specimens.

CONCLUSION

The experimental data reveal that both the surface properties of the materials and the particle sizes have an effect on the deposition rates. The effect of particle size is much larger than the effect of surface properties.

The effect of surface properties is larger for the UFP and for the particles with diameters larger than 10 µm. The measurements for particles with diameters of 0.3 and 0.5 µm give less deposition than for both bigger and smaller particle sizes.

There is a need for further research aiming at the development of an improved method for assessment of deposition of indoor particles on different surface materials.

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The Science of Gas Phase Air Cleaning

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SUMMARY

This paper reviews indoor air quality needs; then it addresses gas phase air cleaning (a little understood, little used in HVAC, high opportunity technology) - particularly by adsorption. Opportunities and limitations are discussed and the capital and energy cost impacts of use are detailed. The role of ventilation/IAQ standards is addressed.

INTRODUCTION - THE NEED FOR AIR CLEANING

It is desirable that people breathe clean air. It is desirable that the air does not have things in it which are harmful if inhaled and it is desirable that the air that is breathed in is not unpleasant i.e. that it does not contain smelly things and irritating things. Air is made up of gases and particulates so control of air quality requires control of both.

Health

Gases *and* Particulates affect health. In extreme cases gaseous components (sulphur oxides, phosgene, cyanides, chlorine etc) can be deadly. In other cases particulates can be deadly (arsenical smoke, plutonium dust, infectious agents etc). In ordinary circumstances (when considering fairly average indoor and outdoor conditions) the most health influencing substances are probably particulates.

Amenity

Particulates can be irritating, they can obscure vision they can make surfaces dirty and they can otherwise be a nuisance (eg. pollens irritate many hay fever sufferers). Particulates, also, can affect systems that keep people comfortable – dirty cooling coils etc. However, virtually all odours are gases and probably a gross majority of oronasal irritants (irritating in the sense of affecting the trigeminal nerve system) are gases. When considering amenity in buildings (those where ambient pollution is not excessive) the main technical opportunity for improving amenity is in control of gaseous pollutants by ventilation or gas phase air cleaning.

BASICS OF GAS PHASE AIR CLEANING

Gaseous pollutants are removed from air by ‘gas phase filters’ or ‘gas phase air cleaners (GPAC)’. ‘Filter’ is a word without universally agreed meaning so ‘GPAC’ is used here.

GPAC’s come in wet and dry types. A wet GPAC is usually an air washer. A dry GPAC is usually a sorptive device (physisorptive, chemisorptive, catylitic or photocatylitic).

An air washer (wet type GPAC) is an Absorptive air cleaner. As the polluted air contacts the water spray some gaseous pollutants will absorb into liquid water and then be sent to disposal or recovery. If a pollutant is water soluble an air washer may use ordinary water but treated waters are sometimes used eg. use of acidic water to remove alkaline pollutants from air. In non air-conditioning applications air washers may be very useful but 'the devil is in the details' e. g. when treating gaseous pollutants in road tunnel exhausts (a thing not often done) it is not particularly difficult to wash sulphur oxides out of the air but it is difficult to wash nitrogen oxides out- the solution and chemical kinetics are inherently unfavourable.

In air-conditioning applications air washers have disadvantages. They, generally, will humidify the air (they may dehumidify), and this may not be desired. If the water is treated with chemicals then these may get into air and be breathed in. Where there is water there is the possibility of Legionella etc. hazard.

Wet GPAC's are a niche product in HVAC so, here, they are not considered further. Dry GPAC's have more potential. For convenience they are hereafter referred to as GPAC's.

Gas Phase Air Cleaners

Presently, GPAC are rarely used in the air-conditioning industry and useful information about their design and use is rare. Such poor commercial information is available that reasonable HVAC engineers may suspect that there is no basis to claims about GPAC efficacy. Consequently there is a need for material that will assist HVAC practitioners.

GPAC Types

There are four principal types of Gas Phase Air Cleaners – Chemisorptive, Physisorptive, Catalytic and Photocatalytic. In each cases the primary mechanism is adsorption i.e. the capture of gaseous pollutant molecules by (onto) GPAC material.

In physisorptive GPAC's the capture of the pollutants is by weak electrical forces and the process is reversible (by change of temperature or pressure). In chemisorptive GPAC's pollutant is attracted to the adsorptive by physisorptive forces and then it chemically reacts with the adsorptive so that it is permanently and irreversibly (for practical purposes) attached. In a sense, chemisorptive GPAC's are a subclass of physisorptive GPAC's.

A well known Chemisorptive GPAC is 'Purafil'. In 'Purafil' the physisorptive is activated alumina but the alumina pellets are impregnated with potassium or sodium permanganate etc (the amount is judged – too little and service life is reduced, too much and the holes in the pellet become clogged and efficiency is reduced). The permanganate will trap some chemicals and not others: so, care in application is needed. Activated carbons may also be impregnated.

In catalytic adsorption, chemical reactions are encouraged by the nature of the surface on which they are adsorbed thus pollutants may be removed from an airstream (a catalyst increases the rate of the reaction influenced but it does not affect the end point). In air conditioning applications catalytic air cleaners are uncommon. One product uses gold/iron oxide on zeolite substrate to deodorise toilet exhaust. Photocatalytic adsorption is essentially the same as catalytic adsorption but with light energy added to accelerate the reaction, probably photons assist dissociative adsorption prior to recombination into the final species.

When air is passed through a photo-catalytic converter many objectionable pollutants are turned into harmless substances and the air is thus ‘cleaned’; unfortunately these devices can also cause the formation of troublesome pollutants from otherwise less objectionable air pollutants. These devices are presently in ‘demonstration’ stage and, basically, the jury is still out where their utility is concerned.

The archetypal physisorption GPAC is the “activated carbon” filter so this filter type is examined here. Many things to be said about activated carbon are also applicable (to varying degrees) to other adsorbents – activated alumina, silica gel, zeolites, organic synthetics etc.

Activated Carbon – its Properties

The most special feature of activated carbon (& other adsorbents) is that it has a very large surface area though the size distribution of the ‘pores’ in it can be of great relevance. Chemically, ‘activated carbon’ is close enough to being ‘pure’ carbon.

A piece of activated carbon may be thought of as a pure carbon object with holes in it and the walls of these holes having smaller holes in them and the walls of these holes having holes that are even smaller etc. This geometry allows a very large accessible surface area.

The specific surface area of a commercial activated carbon will typically be in the region of *1100 square meters per gram of carbon***.** This is extremely large and this can be well appreciated by observing that if *one gram of carbon* is formed into a sheet that is one atom thick then *the combined area of both sides of the sheet will be about 2630 square metres*.

It is possible to debate these observations (how was the surface area measured or calculated? etc) but it remains that in surface area terms ‘activated carbon’ is quite remarkable.

An activated carbon filter is usually a mass (‘pad’, ‘bed’ etc – the descriptor varies to suit the circumstances) of activated carbon fragments that are often granular, cylindrical or flat particles about 4 mm across and these particles are usually enclosed in a metal screen casing so that air can easily be directed through the ‘bed’ so formed. Some other quite clever types of carbon filters are available.

The large surface area of Activated carbon allows it to remove pollutants by *Adsorption*. *Physisorption* is now further described but, for simplicity, being called *adsorption.*

As this carbon may have a surface area of about 1100 $m^{2/}$g there is an opportunity for contaminants to adhere to the surfaces and thus be ‘filtered out’ of the airstream.

Adsorption

Lightly polluted air consists of numerous separate molecules of various gases (oxygen, nitrogen, carbon dioxide, water, trace organics etc).

These molecules are vibrating *and* rotating (on all axes) *and* travelling in straight lines. They ‘hit’ each other quite often and when they hit a surface (eg. the walls of a gas storage cylinder) the aggregate effect of their impacts is called ‘pressure’.

Some rough data about the movement of molecules in air (at normal atmospheric pressure) is

Speed of H_2 molecules: about 1800 metres/sec
Speed of molecules like N_2, O_2, H_20: about 500 metres/sec
Average distance travelled before colliding with another molecule: about 10^{-5} cm
Average number of collisions per molecule per second: about $5*10^9$
Number of molecules per cubic centimetre: about $25*10^{18}$
Number of molecules striking a surface per second: about $3*10^{23.}$

Many people will have been told, in their high school (or university) physics classes, that when molecules strike a surface they usually 'bounce off' the surface in a symmetrical way and this can be true but the more usual case is different. Usually, when a molecule hits a surface it attaches to the surface for, on average, an infinitesimally small time and then it is emitted back into the space. This time of attachment is important, as is the 'sticking probability'. If this time is small (as it normally is) then for practical purposes the behaviour of the gas is much the same as if the molecules always bounced symmetrically off the wall (as, in reality, they sometimes do).

If, in the case of a particular molecule, this time is large (say months, years, centuries etc, on average) then that molecule is removed from the gas for a useful time (if not forever) and we may have the phenomenon of useful *Adsorption.*

Why do molecules adhere to surfaces

At base this is a quantum electrodynamic process but the analysis of the process is usually done at a simplified level.

Molecules adhere to surfaces by electrical forces. These are varied, have various names and there are inconsistencies in how the names are used. The expression 'Van de Waals force(s)' is often used for the various forces, sometimes it is used more specifically.

A useful way of classifying the forces is:
Coulomb force: where a charged molecule (an ion) is attracted to an oppositely charged surface
Ion - Induced Polarisation force: Where an ion induces an opposite charge near a surface so it is attracted
Dipole - Dipole force: Where a polarised molecule is attracted to a polarised surface of opposite polarity

- Some molecules while electrically neutral have an unsymmetrical distribution of electrons so, statistically and rather roughly speaking, one end of the molecule is constantly negatively charged and the other end is positively charged
- Surface molecules, whilst electrically neutral, may have an unsymmetrical charge distribution
- These considerations vary with the type of surface molecule and the surface bonding – metals are quite different.

Dipole-Induced Dipole force: Where a permanent dipole molecule induces a dipole at a surface and is attracted
Induced Dipole - Induced Dipole force, and the forces between various induced dipoles, quadrapoles, hexapoles, octapoles etc: Of these various forces the Induced Dipole - Induced Dipole force and the Dipole - Induced Dipole forces seem to be the ones more commonly emphasised in academic discussion of adsorption.

The Induced Dipole- Induced Dipole force occurs when the fluctuating electron distribution around one (otherwise nonpolar) molecule produces a temporary dipole and that dipole induces a dipole in a neighbouring (otherwise nonpolar) molecule – then the molecules are attracted to each other.

A significant complication is that the various induced forces are subject to ‘retardation’ effects. When a molecule induces a dipole in another molecule the ‘message’ between the molecules moves at the speed of light so if the time of travel of light between the molecules (ie the distance) is significant (when compared to the time taken for valency electrons to move around the molecule) synchronisation is reduced and the attracting forces are attenuated.

Why do Some Gases Adsorb Better than Others

Gas molecules have inherent physical and electrical properties – shape, size, permanent dipole moment and polarisability are quite important.

Gases with particular physical/electrical properties tend to be preferentially adsorbed so a study of these properties may indicate adsorption proclivities.

Larger molecules (molecules with higher molecular weights) may be preferentially adsorbed.

Again ‘the devil is in the detail’ and, though larger/heavier molecules tend to be better adsorbed exceptions abound, e.g. carbon dioxide is much less adsorbed on activated carbon than is acetaldehyde – yet these two chemicals have essentially identical molecular weights.

Practical Outcome

Usefully, the broad trend is for larger molecules to be more readily adsorbed. Also ‘normal air’ molecules are small and many air pollutant molecules (particularly the ones that are smelly) are larger thus as a general rule odorous gaseous air pollutants are attracted to and trapped by activated carbon etc. filters. Many dangerous pollutants are so adsorbed.

OPPORTUNITIES AND LIMITATIONS

Opportunities

Indoor air quality (IAQ) targets are presently achieved by ventilation and particulate filtration and designers can choose commercial particulate filtration product to deal with almost any situation thus the modern IAQ problem can be reduced to the provision of a good mixture of ventilation and gas phase air cleaning – and the problem is usually addressed without reference to gas phase air cleaning.

If, in practice, gas phase air cleaning can be employed to remove odorous and other air contaminants (eg volatile organics) then ventilation may be reduced.

If ventilation is reduced savings may result:
- Ventilation energy cost is reduced (less air is heated and cooled, and there may be a minor air transport cost saving)
- Peak energy charges (eg kVA charges) are reduced
- Plant capital cost is reduced (peak cooling/heating loads are reduced) and there may be some plant efficiency and infrastructure gains from making the cooling/heating loads less ‘peaky’.

Also, gas phase air cleaning (GPAC) use reduces greenhouse gas generation.

Limitations

The following factors limit the application of adsorptive GPAC in HVAC application:
- Misunderstanding about acceptable maximum Carbon Dioxide levels indoors abounds - confusion between health based limits and odour indication levels: also, much quoted odour indication values seem to have a very poor research base
- Little useful information, about the use of adsorption filters, seems to be available – commercially or in design literature (except in the case of a few isolated industrial applications).
- Commercial HVAC application data tends, even in the case of product from well established companies, to be a list of various chemicals that may be adsorbed by the product with the performance in adsorbing each chemical described as “excellent”, “good” etc but few useful numbers being evident: performance curves tend to have meaningless axes.
- The common adsorptive (activated carbon) is a heterogeneous thing manufactured in various ways from various precursors thus the product varies greatly
- Product data is usually basic – at best a notional ‘specific surface area’ or an equilibrium uptake of one particular chemical (eg. ‘65% by weight of CCl_4’) whereas, ideally, ‘uptake Vs. challenge concentration’ data is needed
- Notwithstanding that little data available there is no agreed theory for creating isotherms (uptake Vs. challenge concentration curves) from equilibrium single chemical values or for predicting the performance of an adsorbent with a particular chemical from its performance with a different chemical (there are many isotherm theories and some are more popular than others: in fact, particular theories suit particular chemical species). The fairly popular BET theory (circa 1938) is useful, but, its parameters may be varied to, to some extent, get the desired result!
- Available theories may suit particular (single) chemical species but their application to mixtures (body odour being an obvious example) is not clear.
- Actual GPAC design or application methodology/data is scanty and either crude or hideously complicated (mathematically) and essentially unproven: the better information appears to come from the chemical process industry where
    - most parameters are very different from those in the air-conditioning industry (species, concentration, cycle times, cost structures etc)
    - the available literature suggests that there is a fair bit of ‘suck it and see’ to the design of adsorptive processes (though, presumably, there will be useful tightly held information in some places)

- There is a need to balance the economic advantage to be gained by GPAC use against the cost of fan horsepower energy used to drive air through a GPAC.

A good (but not simple) overview and design data source is Perry's Chemical Engineers Handbook 7th Edition Section 16 'Adsorption and Ion Exchange".

Overcoming the Limitations

The limitations are primarily intellectual (understanding the physics/engineering) – rather than based on fundamental barriers. However the matter of balancing GPAC fan energy cost against overall energy/capital cost savings can be real problem in many normal situations. This matter is addressed in a recent Patent application[1]**.**

ECONOMICS OF APPLICATION

Given the paucity of *current* satisfactory GPAC product (this should soon change) the economics of providing GPAC apparatus is not dealt with here. However the advantages to be gained by ventilation reduction are presented below for a number of Australian cities. The calculations have used precise hourly outdoor dry/wet bulb frequency data. These cities are in different climates (though a truly cold climate is not to be found in Australia) so they may be indicative of the situation in some other places.

The table below assumes:

Ventilation rate reduction: 1000 litres/second (perhaps, a 200 person building)
Cost of electricity for cooling: A$0.138/kWH, Cooling coefficient of performance: 3.0
Heating gas energy cost: A$.0372/kWH, Heating system efficiency: 76%
Marginal capital cost of cooling effect: A$800/kW
A$=Australian dollar (in early 2007, A$1.0~0.6 Euro~USA$0.8)
6am to 6pm, 6pm to 6am or 24hr operation: 24C summer inside, 21C winter inside.
Economy cycle not used, no humidification, dehumidification only to 60% RH.

Australian City	*Climate*	*Marginal cooling plant cost (A$)*	*Annual cool+heat cost, day (A$)*	*Annual cool+heat cost, night (A$)*	*Annual cool+heat 24 hr (A$)*
Adelaide	Mediterranean	12,480	887	1290	2077
Alice Springs	Hot arid	15,440	1644	1461	3104
Brisbane	Sub tropical	18,240	1417	847	2264
Canberra	Temperate	4,880	1565	2406	3970
Cloncurry	Semi arid tropical	16,080	2622	1751	4373
Darwin	Monsoon	21,280	5011	4350	9361
Hobart	Cool temperate	3,040	1274	1913	3187
Melbourne	Temperate	10,160	1020	1511	2531
Mildura	Semi arid	14,880	1171	1531	2710
Perth	Mediterranean	12,096	893	978	1870
Port Hedland	Semi arid	15,360	3609	2603	6212
Sydney	Temperate oceanic	6,800	1018	1973	2991
Townsville	Wet tropical	25,680	3603	2405	6008
Wagga Wagga	Temperate	12,560	2224	1706	3930

The operation cost of the mentioned 1000 l/s unit, used as shown in reference 1, is likely to be in the region of $A1000 per annum for a typical office space (2500 hrs/year). The capital cost is not given here.

Much can be concluded from examining this table (but, bear in mind that no really cold climate is represented so conclusions cannot be drawn for many North American and European places):

- Installing GPAC will be wise in many climates
- Very often the motivation for installing GPAC will be capital plant cost saving (new and retrofit installations)
- Energy savings are a strong motivation in consistently hot/humid climates & this may be so in cold climates
- Some places offer large capital savings but small energy savings ('peaky' climates).

GPAC may be used only or principally to reduce capital and kVA etc. charges - thus reducing GPAC fan energy to negligible levels and extending media life to, perhaps, decades.

Note that the above refers to energy use at Australian prices. European prices are considerably higher

VENTILATION STANDARDS ETC.

Often, national ventilation/IAQ requirements are prescribed in ventilation standards. The Australian ventilation standard (AS1668.2[2], 1991 and 2002 editions) sets ventilation rates based on odour and health criteria. Odour is the controlling variable so the standard allows significant reduction of required ventilation rate so long as enough for health is provided.

The Australia Ventilation (IAQ) standard contains specific and detailed provisions for the use of GPAC and particulate air cleaners (filters). These provisions include full calculation procedures for situations in which a 'body odour' GPAC of a known efficiency is used.

Practical odour removal devices will tend, also, to remove a large range of health related contaminants – though not, normally, the smaller molecular species such as carbon monoxide and carbon dioxide - so the gaining of economies by using GPAC odour control is likely to be accompanied by much increased healthiness of breathing air.

The USA standard ASHRAE 62 also appears to offer scope for GPAC use.

Some other national standards may need improvement if the economic and environmental (indoor and global) benefits of GPAC are to be gained.

CONCLUSION

Gas phase air filters are seldom well understood and seldom used. There is much economic and environmental advantage to be gained by their use. Proper attention to this potentially most beneficial technology is overdue.

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Accumulation Behavior of Intermediate Products on The Surface of TiO_2/ACF Catalysts during The Photocatalytic Oxidation of Toluene: Study of The Influence of Environmental Relative Humidity

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SUMMARY

Photocatalytic oxidation (PCO) tests have been carried out for toluene adsorbed on the activated carbon fibers (ACFs) supported TiO_2 photocatalyst in an environmental condition controlled chamber. Through exploring the remnant of toluene and the accumulation of intermediates on the TiO_2/ACF catalyst including species, amount and their change processes under different relative humidity (RH), this study aimed to explore the influence of RH on the PCO of toluene and the role of water vapor in the PCO process: PCO reaction paths and the accumulation of intermediates on the TiO_2/ACF catalyst. Results showed that: (1) with the increase of RH in the chamber (15%, 30%, 45% and 60%) the PCO conversion rate of toluene was positive correlated and no catalyst deactivation was observed under all RH; (2) beside the intermediates of side chain oxidation including benzyl alcohol, benzaldehyde and benzoic acid, which had been reported, byproducts of aromatic ring oxidation including p-toluquinone and o (m, p)-cresol were also observed on the TiO_2/ACF catalyst during the gas-solid PCO process of toluene; (3) although benzaldehyde was the primary intermediate under all RH level, amounts of the byproducts of aromatic ring oxidation were increased with the increase of RH; (4) elevated RH increased the accumulation of benzyl alcohol but assuredly decreased the accumulation of benzaldehyde. These results suggested that: (1) RH affects both the PCO rate and the PCO reaction path of toluene; (2) although methyl group oxidation is the major path, aromatic ring oxidation, which is not the expected path for the PCO of toluene, is enhanced when the RH increases; (3) apart from the role of hydroxyl radical (·OH) arose from water by TiO_2, water molecular also directly takes part in the PCO process. A hypothesis has been proposed: transition species comprised of benzaldehyde, hydroxyl and water molecular exists in the PCO conversion process from benzaldehyde to benzoic acid, though the hypothesis has not been confirmed.

INTRODUCTION

Volatile organic compounds (VOCs) are one group of the main pollutants in indoor air released from many kinds of sources and contribute negatively to human health. Despite the success of adsorption as a means to remove VOCs in indoor air, it does not actually destroy the pollutants but merely remove them for recycling elsewhere in the biosphere; moreover, saturation adsorption is another troublesome problem existing in the adsorption technique. Heterogeneous photocatalytic oxidation (PCO) [1, 2] has been proposed as an advanced oxidation technique for mineralization of various environmental organic pollutants, especially for air decontamination. However, there are still issues unanswered in the application of the

PCO technique. Some of these issues are: (1) lower PCO rate than expected; (2) intermediates environmental toxics and behavior; and (3) catalyst deactivation. Environmental relative humidity (RH) plays an important role influencing on the gas-solid PCO process of VOCs. A large number of studies [3-21] related to RH have been devoted to this field in the last two decades to address three issues mentioned above. The work presented here also addresses this topic.

Toluene is a major indoor air pollutant and various studies have tested the potentiality in the use of gas-phase PCO of toluene for air decontamination [6-12]. In this paper, the PCO of toluene has been researched using nano-TiO_2 immobilized on activated carbon fibers (ACFs). ACFs, a new formulation of activated carbon with high BET surface area and uniform micropore structure, were used to support nano-TiO_2, adsorb toluene and catch hold of all intermediates during the PCO process. Toluene was adsorbed on the TiO_2/ACF catalyst in sample vial firstly, and then adsorbed toluene was photocatalytic degraded in an environmental condition controlled chamber. Elimination of toluene and accumulation of the intermediates under different RH were studied in order to explore the influence of RH on the PCO of toluene and the role of water vapor in the PCO process.

METHODS

Reagents

The grade of all reagents was chromatogram purity grade (Guangfu Reagent Corporation, China). Water used for solution and experimental preparations was ultra-pure water (18.2 Megohm.cm resistivity; <10 ppb TOC) from Milli-Q plus (Millipore, USA).

TiO_2/ACF catalyst preparation

Nano-TiO_2 (Degussa P-25: surface area 50 m^2/g, non porous, about 80% anatase) was used as a photocatalyst without any pretreatment. ACFs (Nantong Senyou Carbon Fibers Corporation, China), in the form of felt, were produced from rayon precursor. ACFs were washed with ultrapure water and dried at 105 ºC before use.

Pieces of ACFs (10 mm×12 mm×1 mm) were put into nano-TiO_2 water suspension (2% wt), under ultrasonic treatment for 60 min. Then the pieces (TiO_2/ACF catalysts) were dried at 105 ºC for 2 h. In all experiments, the amount of TiO_2 of 5.2 mg ± 5% loaded on one piece of ACFs was determined by the difference of weight before and after the coating procedure.

Experimental setup

Adsorption was conducted in the EPA-sample vials with PP Screw Seals (20 ml, ND 24, La-Pha-Pack, Ger.). A stainless steel environmental condition controlled chamber (1 m × 1 m × 1 m, 1.5 mm thick, 1Ni18Cr9Ti) was employed for PCO degradation studies. Illumination was provided by a 15 W UV mercury lamp (Nanjing Jinling Lamp Corporation, China) which emits ultraviolet radiation with a primary wavelength at 254 nm. The UV lamp tube (length 50 cm) on a holder was vertically placed on the bottom of the chamber. A stainless steel circle plate (diameter 10 cm) with eight equiangular and spokewise stainless steel thread rods (diameter 2 mm, length 8 cm) was horizontally fixed in the middle of the lamp tube. Eight metal clamps were fixed on each rod by two hexagon thin nuts respectively with a horizontal distance of 5 cm from the lamp tube surface. Surface of clamps was coated with Teflon film.

Constant temperatures (T, 25 ± 0.5 ºC) and 4 RH levels (15%, 30%, 45% and 60%) in the chamber were achieved by a thermostatic and humidifier controller.

Absorption-degradation experiments

Absorption procedure: 9 TiO_2/ACF samples were put into 9 sample vials respectively, and then vials were sealed. Each vial was injected 2.0 ul toluene carefully by a syringe (10UL, Agilent Corp., USA), and then kept in darkness.

Degradation procedure: After 12 h, 8 TiO_2/ACF samples were taken out from the vials, and fixed on the clamps respectively. Lamp was turned on when the T and RH in the chamber reached the set condition. 6 TiO_2/ACF samples were collected respectively at 30 min, 60 min, 90 min, 120 min, 150 min and 180 min as the experimental samples. 2 TiO_2/ACF samples were collected at 90 min and 180 min respectively as the parallel samples. The 9th TiO_2/ACF sample, as the control sample, was analyzed directly without exposure to UV light.

GC-MS and GC-FID analysis

Accumulated intermediate organic products were extracted from the TiO_2/ACF sample by carbon disulfide (CS_2) and ultrasonication. Qualitative analysis of the filtrate was carried out by the use of an Agilent 6890 Gas Chromatograph equipped with a HP-5MS capillary column and a HP 5975 mass analyzer detector.

Accumulated intermediate organic products and remained reactant were extracted from each TiO_2/ACF sample by 2 ml carbon disulfide with 10 min ultrasonication. Quantitative analysis of the extraction was done by GC (Agilent 6890, Agilent Corp., USA) equipped with a HP-5MS capillary column and a FID detector.

RESULTS

Toluene removal

The conversion of toluene in 180 min under different RH was shown in Fig. 1. It showed that the oxidation conversion of toluene increased from 9.5% to 14.2% when the RH in the chamber increased from 15% to 60%. In the control experiments no peak other than toluene was found through GC-FID analysis and no loss of toluene was found. These results indicated that elevated RH increased the PCO conversion rate of toluene. The conversion rate of toluene was not significantly diminished in 180 min, which indicated that no deactivation of catalyst occurred during the gas-solid PCO process in this study.

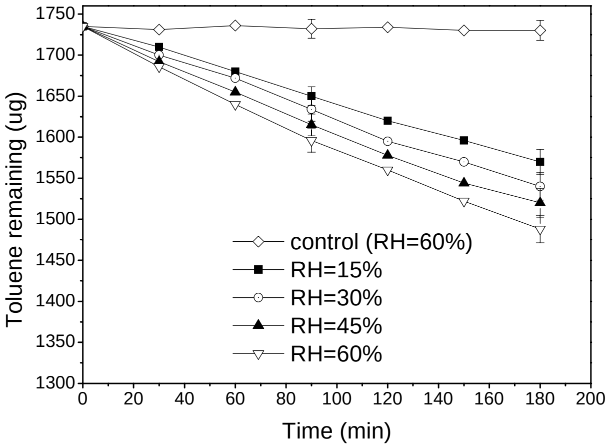


Figure 1. Photocatalytic degradation behavior of toluene adsorbed on TiO_2/ACF catalyst

Intermediate organic products

Intermediate products accumulated on the TiO_2/ACF catalysts during the PCO of toluene were analyzed. Five intermediate products were detected as benzyl alcohol, benzaldehyde, benzoic acid, p-toluquinone and cresol by GC-MS. The accumulation of the intermediate products varied with the increase of RH. Firstly, different levels of intermediate products were found accumulated on the TiO_2/ACF catalysts. Under all RH levels, benzaldehyde was the most abundant intermediate (about 74~89% by mass) and benzyl alcohol was the secondary (about 9~19%). Benzoic acid (about 1%), p-toluquinone (about 1~3%) and cresol (about 1~2%) were trace intermediates. Secondly, the changing trends of the accumulation amount of these intermediates were different. With the increase of the RH in the chamber, the accumulation amount trends of five intermediates: benzaldehyde decreased a lot; benzoic acid increased a little; benzyl alcohol, p-toluquinone and cresol increased a lot.

To explore the detailed reason for the influence of the RH in the chamber, the accumulation process of benzaldehyde and benzyl alcohol were studied and the results were shown in Fig. 2. It can easily be found that the accumulation behaviors of benzaldehyde and benzyl alcohol were opposite. The accumulation rate of benzaldehyde on the catalyst decreased significantly with the increase of RH, however, the accumulation rate of benzyl alcohol increased a lot.

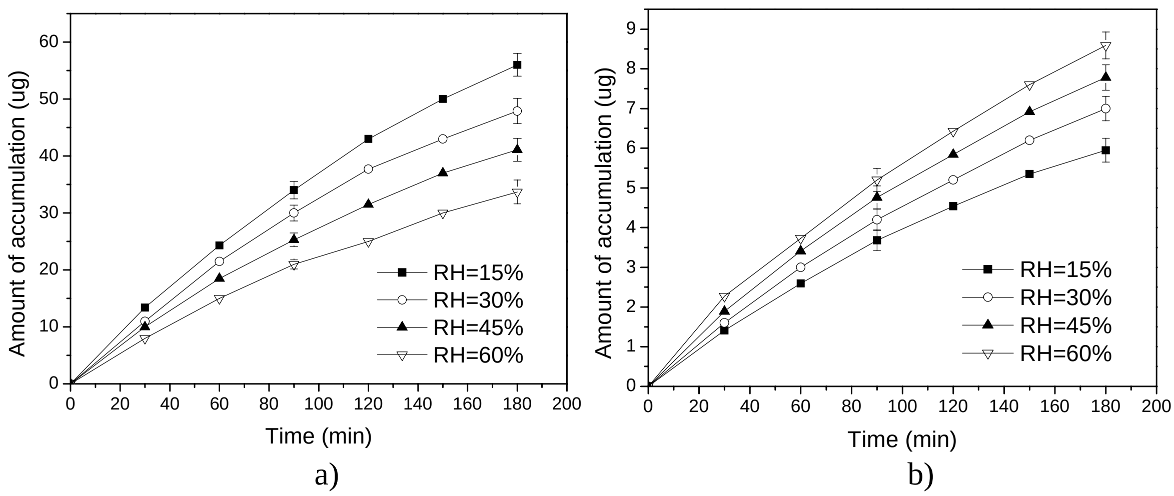
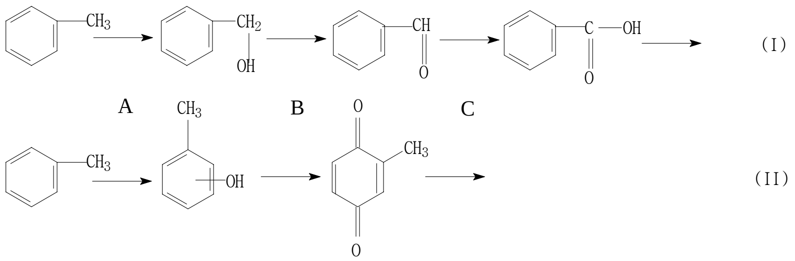


Figure 2. Amounts of intermediate accumulated on TiO_2/ACF catalyst. a) Benzaldehyde, b) Benzyl alcohol

DISCUSSION

Reaction path

There are two kinds of hydrogen atom in a toluene molecule: one is in the methyl group, the other is in the aromatic ring. It is accepted that the oxidation of toluene takes place on the methyl group and proceeds in the following reaction chain: toluene→ benzyl alcohol→ benzaldehyde→ benzoic acid → … → carbon dioxide [16, 17]. At the same time the oxidation of toluene taking place on the aromatic ring as a minor path has been proposed in aqueous solution [20] without much certainty during the gas-solid PCO process [17]. A schematic of the possible degradation paths of toluene is shown:
Path I: Side chain (methyl group) oxidation
Path II: Aromatic ring oxidation



In this study, the formation of benzyl alcohol, benzaldehyde and benzoic acid that have been reported by other researchers [3, 5, 13, 16-17] confirmed the occurrence of path I. At the same time, some intermediate products of the aromatic ring oxidation including p-toluquinone and cresol were also observed in the extraction of the TiO_2/ACF samples, which confirmed the occurrence of path II. However, the intermediate products of the aromatic ring oxidation are usually more poisonous and difficultly oxidized than those of the side chain oxidation. Larson and Falconer [17] reported that the adsorbed m-cresol was photocatalytically oxidized difficulty by TiO_2 catalyst and the catalyst also turned black due to the polymerization of m-cresol to polyphenolic compounds. So, path II is an unexpected path for the PCO of toluene.

Benzyl alcohol, benzaldehyde and benzoic acid were found in all experiments and benzaldehyde also was the most abundant intermediate under all RH. These results indicated that the methyl group oxidation is the main PCO reaction path of toluene. The aromatic ring oxidation is a minor PCO reaction path of toluene because only trace byproducts of this reaction path were observed. However, a fact must be noticed that the accumulation amount of the intermediates of the aromatic ring oxidation increased with the increase of the RH. The distribution sketch map about the intermediate products of side chain oxidation and aromatic ring oxidation after 180 min reaction at 15%, 30%, 45% and 60% RH levels was shown in Fig. 3. It indicated that the condition of RH assuredly changed the PCO reaction process of toluene. The aromatic ring oxidation was enhanced when the RH increased. In view of the environmental toxicity and PCO behavior of their intermediates, it deserves more attention to the PCO of aromatic compounds under high water vapor level.

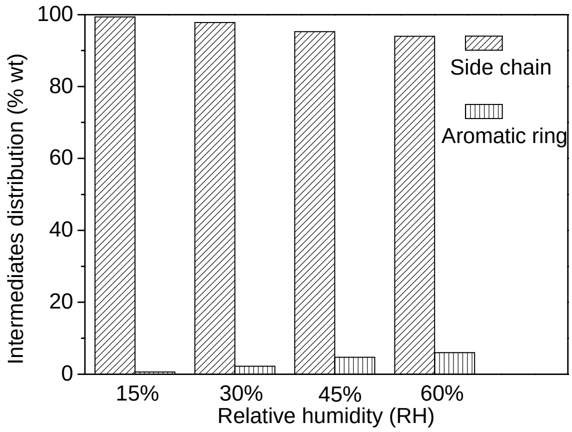


Figure 3. A distribution sketch map about intermediate products after 180min reaction

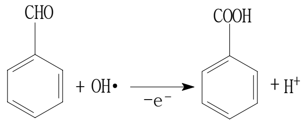
Roles of water vapor

With the increase of RH, the intermediates of side chain oxidation displayed quite different accumulation behaviors on the TiO_2/ACF samples (Fig. 2). Elevation of RH in the chamber increased the accumulation of benzyl alcohol but significantly decreased the accumulation of benzaldehyde.

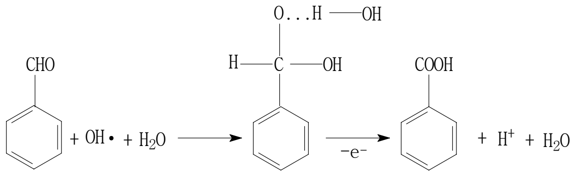
A competitive adsorption between water and organic compounds on catalyst surface had been proposed as a cause of the decrease of adsorption or the increase of desorption of organic compounds under high RH condition [9]. However, two experimental phenomena in our study indicated that it was not the competitive adsorption between water and benzaldehyde that caused the reduction of benzaldehyde accumulation on the TiO_2/ACF photocatalyst when environmental RH was high. Firstly, in the control experiment (using TiO_2/ACF without illumination at 60% RH level), no elimination of toluene by desorption was found within the range of the experimental error (Fig. 1). Secondly, no elimination of the accumulation amount of benzyl alcohol was observed in the experiments though it would be more easily desorbed than benzaldehyde.

In our experimental system, the rate of the conversion of toluene displayed as a quasi-first-order reaction and the increase of RH in the environmental chamber enhanced the PCO conversion rate of toluene, which indicated the apparent reaction rate constant (k) of the conversion of toluene increased (Fig. 1). Hydroxyl radical arose from water by TiO_2 is often assumed to be the major active species to oxidize pollutants without selectivity for aqueous PCO process, however their role in gas-phase PCO is debated [13]. Under fixed illumination intensity, increase of RH in the environmental chamber caused an enhancement of the concentration of OH radical, which should be responsible for the increase of the apparent k. Meanwhile, the increase of the apparent k also suggested that OH radical is the active species in gas-solid PCO system. According to the reaction dynamic principle, the increase of the apparent k of reactant should enhance the accumulation rate of the intermediates. In our experimental system, the accumulation rate of the intermediates increased with the increase of the RH except that of benzaldehyde. So, beside the role of the OH radical, there must exist another role arose from water vapor that significantly affect the dynamic behavior of benzaldehyde.

The photocatalytic oxidation from benzaldehyde to benzoic acid can be simply shown as reaction C1.



(C1)



(C2)

Based on following two facts that: (1) the radical of $C_6H_5CO\cdot$ is unstable and (2) free radical substitution reaction is more difficult than free radical addition reaction for the oxidation of aromatic aldehyde to aromatic acid, we propose a possible PCO mechanism from benzaldehyde to benzoic acid shown in reaction C2. Although no water exists in reaction C1, reaction C2 is consistent with our experimental data. Here, we propose that a transition species comprised of benzaldehyde, hydroxyl, and water may exist in the reaction process though it has not been confirmed yet. We believe that although OH radical is the real active species to oxidize benzaldehyde, water molecular also directly takes part in the PCO process from benzaldehyde to benzoic. The increase of the RH benefits the formation of the hydrated transition species, which enhances the conversion of benzaldehyde and causes the visible elimination of this intermediate compound. In conclusion, water vapor plays two kinds of roles during the PCO of toluene: one arises from the hydroxyl radical which acts as the active species to oxidize pollutants; the other arises from water molecular which improves in eliminating accumulated benzaldehyde.

Species for deactivation

Some researches suggested that it is the nonvolatile carboxylic acid or carboxylate that leads to the deactivation of catalyst [22]. These authors thought that carboxylic acids, usually nonvolatile and with strong bond with catalyst, would accumulate on the surface of photocatalysts and block the active sites on the photocatalysts. However, data from our experiments are inconsistent with this conclusion. If benzoic acid blocks the active sites on the photocatalysts and causes the decline of catalyst activity, a significant accumulation of benzoic acid in the experiment should be observed. However, only trace benzoic acid was observed during the PCO process. Meanwhile, benzaldehyde was the most abundant compound among all intermediate products accumulated on the photocatalyst under all RH. It indicated that benzaldehyde is more difficult to be oxidized than benzoic acid and the step from benzaldehyde to benzyl acid should be the slowest step during the PCO process of toluene. Although there was no deactivation in our experiments, we believe it is the benzaldehyde that should be responsible for the deactivation of photocatalyst in gas-solid system. Besides, due to their quite inert PCO behavior, the accumulated intermediates of aromatic ring oxidation also will be the species for the photocatalyst deactivation especially when the environmental RH is very high.

Merit of TiO_2/ACF

Deactivation of the photocatalyst has been regarded as a fatal disadvantage of the PCO technique in practice and it has involved the attention of any researches aimed at the development of this new technique. When using the bare nano-TiO_2 as the photocatalyst, catalyst deactivation concomitantly exists in the gas-solid PCO process of aromatics especially under low RH condition [3, 6-7]. However, in our study, no catalyst deactivation was observed under any RH conditions. Nano-TiO_2 particles agglomerated only on the outside surface of ACFs, while the inside surface of ACFs was still exposed to the air. The exposed ACFs served as the adsorption centers of pollutants, water and intermediates by physical or chemical process, allowing intermediate accumulated on the catalyst surface without hampering the interaction between pollutant and TiO_2, and reducing the competitive adsorption between pollutants and water [23-27]. So, we believe that lacking active sites is the reason for the deactivation of catalysts and negative effect of water under high RH. TiO_2/ACF composite material, with large surface, is a good photocatalyst suitable for the abatement of catalyst deactivation problem.

ACKNOWLEDGEMENT

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Effects of Ionisation Air Purifiers on Indoor Air Quality

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SUMMARY

Three different air purifying devices are compared in terms of their influence on indoor air quality. Two systems use ozonisation and ionisation. One system uses ionisation and special filter devices (manufacturer's specifications). In the case of ozone production the perceived intensity and the PD value will increase to inacceptable values. The chemical analyses shows that the increase aldehyde concentration is probably caused by surface reactions between material and ozone.
Purifying devices running with ionisation only have slightly no effect on the perceived air quality. Only one test case, when emissions of human beings are an additionally pollution source in the test room, a small increase in air quality can be recognised.

INTRODUCTION

A common method to achieve good indoor air quality is to reduce concentrations of odorous substances by ventilation. Presently an increasing number of air cleaning devices are offered on the market. The manufactures promise a reduction of odour-active substances in the room air.
Up to now, only very few data is available [1, 2] on the odour removal efficiency. Most of the time the efficiency of these devices refers only to manufacturer's specifications. Therefore systematic tests are made in a new experimental set up.

EXPERIMENTAL SET-UP

With two identical, ventilated test rooms the efficiency of different air cleaning devices are examined. Figure 1 shows the location of the two test rooms (room 1 and room 2) in the air quality lab. The dimensions are 2.5 m x 2.5 m x 2.5 m. Both rooms are supplied with outdoor air by the same supply air duct and two fan coil units.

The rooms are built from the same building materials and have the same furnishing, similar to common office rooms. A controllable air flow of the polluted indoor air of the test rooms is extracted by a potential free tubing of stainless steal. It is presented with diffusers to a panel of subjects to assess the air quality. In addition, the air contaminants are analysed chemically by using GC/MS technology via TENAX probes.

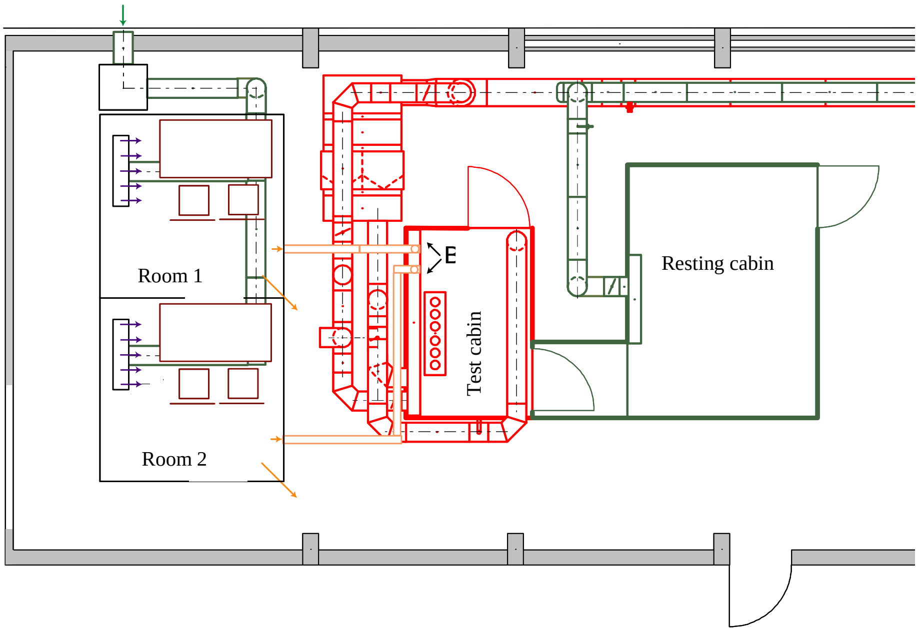

Figure 1: Sketch of Test Setup in the Air Quality Lab with two test rooms and the cabins for the panel [4]

METHOD

The air in the two test rooms is polluted by the equipment (standard building materials like carpet, furniture and wallpapers). In the first step of the experiment the perceived odour intensity, the acceptability and the hedonic impression of both rooms is evaluated.

The perceived intensity is measured with a selected panel of 10 persons [3, 4]. When assessing perceived intensity of unknown samples, panel members can rely on a comparative scale of acetone/air mixtures, the so-called markers, which help to determine the intensity. The perceived intensity Π can only be determined with trained panels using a comparative scale.

Unlike the acceptability method with untrained panels, the intensity of odorous substances in the air is determined by a comparison with different specified intensities of the reference material acetone. The smelling capability varies from human to human. Training and use of comparative sources ensure that the influence of subjective perception of the test result is reduced since all panel members evaluate air quality based on the same scale.

A naive panel of 40 persons is asked about the acceptance of the air quality. Therefore a continuous scale is used for the acceptability. It is converted into values of +10 (clearly acceptable) to –10 (clearly not acceptable). An average value is calculated from all answers to calculate the acceptability [5]. In addition a question to the hedonic impression is answered by both panels.

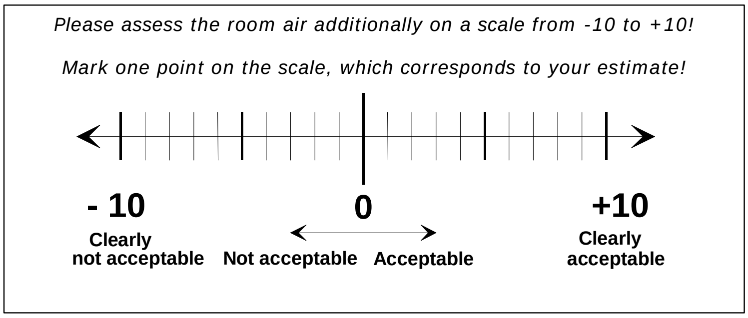

Figure 2: Question to determine acceptability

Three different purifying devices are tested separately. In one of the test rooms the device is installed, the other room is running without any purifying for comparison. For each device a test series over several days is done. In intervals the perceived intensity Π, acceptability and the hedonic impression is assessed for both rooms at the same time in a blind test.

PURIFYING DEVICES

In Table 1 the technical specifications are listed. Type 1 and type 2 have a built in purifying device, which produces ozone and ions. For type 1 there is only the option to run both modes n the same time. Type 2 can produce ozone and ions separately. The stand alone device type 3 has four options for different modes. The manual shows the “turbo” mode as the mode for fast air cleaning. Therefore this mode is tested, additionally all other options are switched on.

Tabele 1: Technical specifications of the tested air purifying devices (manufacturer's specifications)

Type	**1**	**2**	**3**
	Decentralized fan coil	**Decentralized fan coil**	**Stand alone in room**
Purifying Method	Ionization and ozonisation	Ionization and ozonisation	Pre filter Plasma ioniser Bio antibody filter Ionising frame Ionised wire Streamer discharger Opposing pole plates Plated filter (electrostatic) Photo catalyser
Tested working modes	Combination of ionisation and ozonisation runs in out door air modus	Combination of ionisation and ozonisation runs in air recirculation modus	**Auto** (automatically adjusted air volume flow) **Turbo** (fast air cleaning), **Anti Pollen** (removing pollen) **Relax** (negative ions)
	---	ionisation without ozonisation runs in air recirculation modus	---

RESULTS

In the first step a comparison of the two test rooms was done. Figure 3 shows a good analogy in the air quality of both rooms.

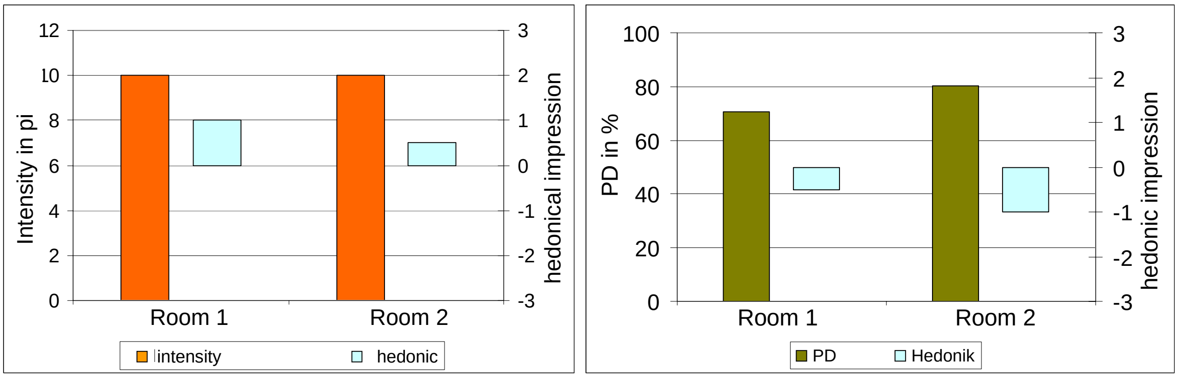

Figure 3: Perceived intensity and hedonic impression, PD and hedonic impression

The perceived intensity is similar for both rooms. The value of 10 pi characterises a strong odour impression. The hedonic impression of the rooms is also quite similar. The acceptability measurements show as well very comparable results for both rooms. During the time the PD-value and the intensity decreases equally in both rooms because of normal emission of the materials and the ventilation.

This similar air quality state in both rooms is the starting point of all investigations of the three different purifying devices.

Device 1

This device runs with a combination of ozone production and ionisation. The ozone concentration in the room is in the range of 50-70 ppb (alternating because of internal control loop).

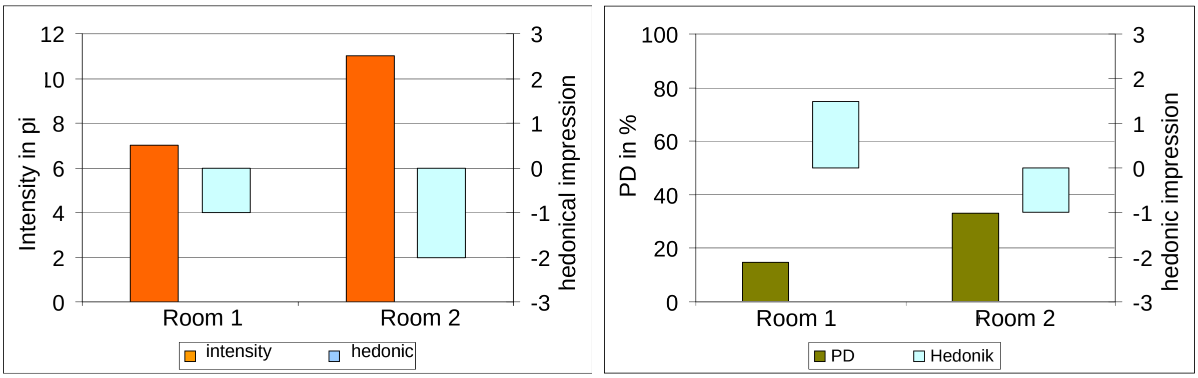

Figure 4: Perceived air quality, hedonic impression and PD for device 1

Figure 4 shows that the purifying device increases the perceived intensity about 4 pi in comparison to the room without purifying device. The hedonic impression becomes negative when the device is switched on. Measurements with the naive panel show similar results. The question of the acceptability gives a PD level of about 15 % for the room without purifying device and about 35 % with purifying device.

Device 2

First device 2 runs in the ionisation mode only with recirculating air. Figure 5 shows that there is slightly no effect on the perceived intensity Π and the acceptance values. Only a positive tendency in the hedonic impression of the panel that perceives the intensity can be recognised.

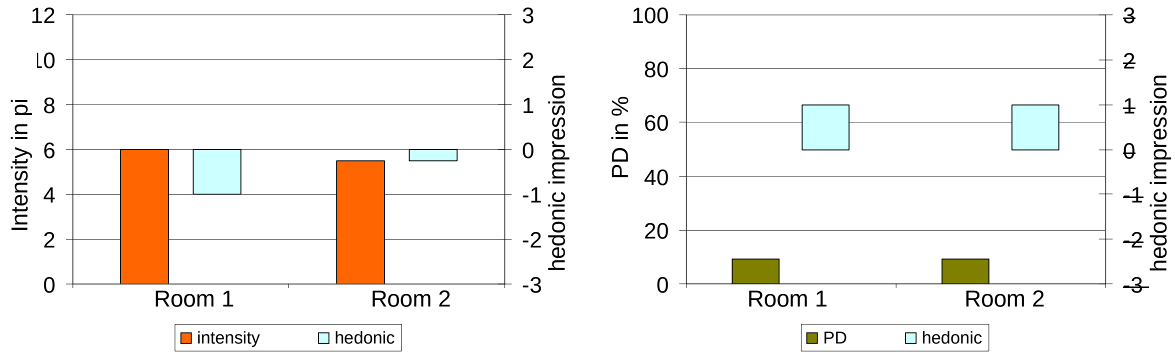

Figure 5: Perceived air quality, hedonic impression and PD for device 2 in ionisation mode (test 1)

Second, device 2 runs in a mode that combines ionisation and ozone production. In Figure 6 the influence of the ozone production (20 ppb) on the perceived intensity is shown. The intensity increases from 6 pi to 9.5 pi.

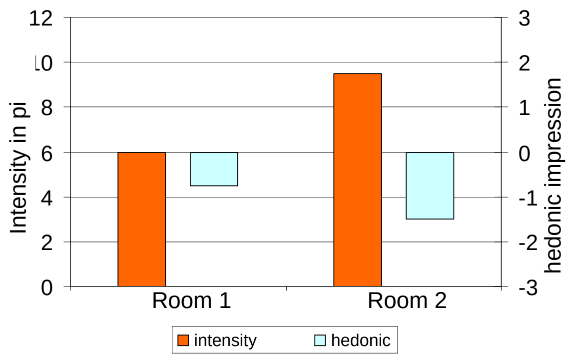

Figure 6: Perceived air quality and hedonic impression for device 2 in ionisation and ozone mode (test 2)

A third test run was made with persons as additional pollution source in the test room. The results are given in Figure 7, where a little increase can be recognised in the hedonic impression. The perceived intensity is nearly the same for both rooms.

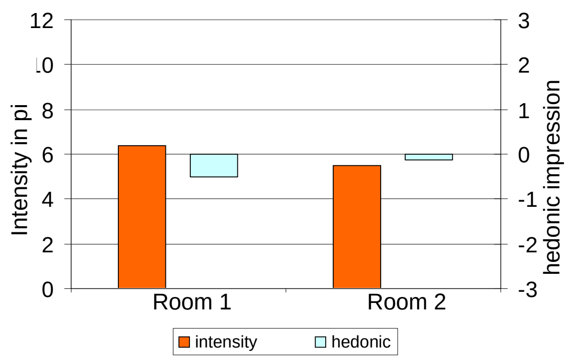

Figure 7: Perceived air quality, hedonic impression for device 2 in ionisation mode, person as pollution source in the test room (test 3)

Device 3

For this stand alone device the internal functions are not documented properly, it runs in the "turbo" mode with all options switched on.

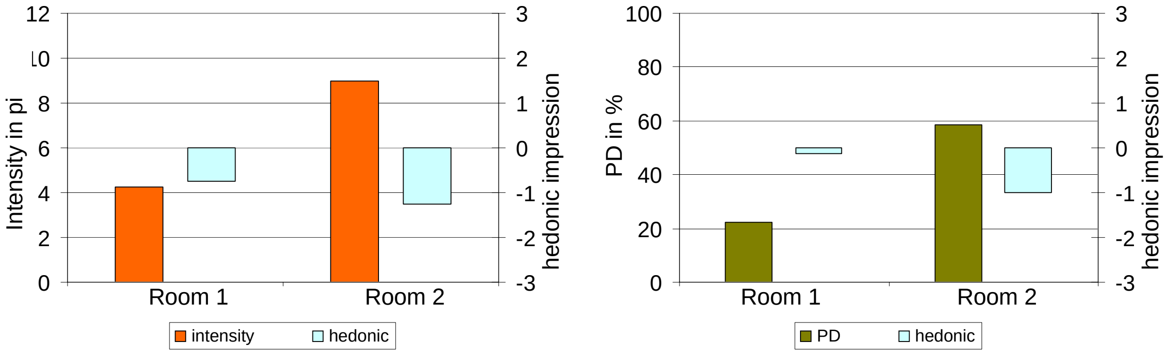

Figure 8: Perceived air quality, hedonic impression and PD for device 3 in automatic mode

Figure 8 shows that perceived intensity and hedonic impression is worse in the case with purifying device also the PD value raises from about 25 % up to nearly 60 %.

Chemical analyses

As an example for all of the chemical measurement the data of device 1 are discussed. The results from chemical analyses using GC/MS technology show the influence of ozone concentration in the room. Figure 9 indicates an increase of the concentration of aldehydes. The dark part of the bar shows the concentration in the test room without air purifier, the whole bar stands for the concentration in the room with running device. One reason of the rising concentrations are chemical reactions of ozone on the surfaces in the room [6]. Products of these reactions are aldehydes. Aldehydes as nonanal and hexanal have a strong odour impression and this might be a reason for the worse perceived air quality. Other concentrations of chemicals are low, but in most cases there is an increasing of concentration according to the ozone concentration. The addition of several chemicals in low concentration can cause odorous irritations.

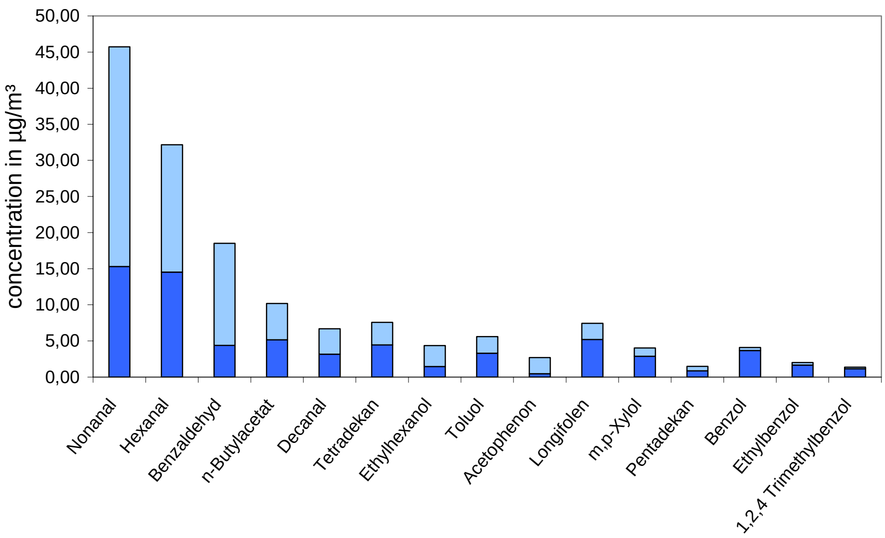

Figure 9: Increasing concentration of aldehydes caused by ozone

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AIRSECURE- safety through filtration and detection

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SUMMARY

AIRSECURE project develops a protective solution against airborne threats for airport environment. The development efforts are based on risk analysis, and the developed technologies include high efficiency particle filtration, chemical filtration, detection of aerosol particles and hazardous chemicals

INTRODUCTION

The AIRSECURE project is a European Research (CRAFT) project, where the Commission supports the work by providing participating small companies the possibility to utilize the work of large research institutes. The AIRSECURE project will develop a system that consists of filtration and detection solutions against airborne threats at airports. The design and operation of the system is based on Risk analysis and Risk management. Thus the complete AIRSECURE system can be divided into three main component groups: risk management, filtration solutions, and detection solutions.

The fear of terrorist attacks against civil targets has increased recently. One of the most frightful scenarios is the use of airborne chemical, biological or radiological (CBR) weapons against unprotected civilians. Of particular concern are airports where such an attack may cause extensive injury and severe impact on the aviation industry and the whole economy of the European Union.

In a possible attack with CBR agents it is credible that the mechanical ventilation system will be used for agent delivery since any release into the ventilation system would be rapidly spread throughout the ventilated spaces. High efficiency filtration and real-time detection of hazardous agents are among the potential measures that can be implemented in advance to reduce the consequences of intentional CBR agent release. However, while it is possible to rapidly detect chemical compounds, it is a very challenging task to develop fast and sensitive sensors for biological agents.

In principle, high efficiency filtration can be used to reduce the risk of airborne threats delivered through mechanical ventilation systems. However, because of the high pressure drop, large space requirements and the high initial and operating costs the present high efficiency filters are not practical solutions for existing buildings. Moreover, their installation would need costly and time-consuming renovation of the whole ventilation system. The AIRSECURE system will improve the security of passengers and workers at airports by a comprehensive approach including risk analysis for identifying the high-risk areas, novel protective filtration systems, proper air distribution, and detectors for early warnings of threat.

The main idea of the AIRSECURE solution is to combine promising new filtration technologies for removal of both biological and chemical agents with a protective filtration unit. These distributed units can be flexibly and quickly installed in the supply or exhaust air ducts of the high-risk areas. The very low flow resistance of the filter allows its installation without extensive modifications to the ventilation systems. New particle detectors will be developed to monitor the performance of the filtration system for maximum security. The optimum number and location of both particle and gas detectors and protective filtration systems are based on risk analysis. The secure air-filtration and advanced warning systems can deter the attacks, and reduce the effects of a CBR agent release by removing the toxic agents from supply air of the building.

MAIN INNOVATIONS OF AIRSECURE

The main challenges of a protective system and the ways AIRSECURE tackles these challenges are presented in Figure 1. In order to overcome preceding technological barriers, the AIRSECURE system consists of

1) Chemical detectors with centralized monitoring
- Fast evacuation
- Real-time information of the dispersion of toxic chemicals (impurities)
- Protecting large spaces (halls)

2) High efficiency filtration combined with particle detectors
- Continuous protection against bio- and other aerosols in selected premises
- Particle detectors ensure safe operation of the filters at all times and give information about abnormal particle concentration levels
- Particle detectors are monitored through the same centralized system as the chemical detectors
- Versatile chemical filters applied in areas of sensitive operations

The principle of the AIRSECURE system with the possible locations of detectors is shown in Figure 2. Among crowded populations, airborne transmission predominates the spread of several diseases in enclosed spaces. Thanks to the improved filtration, the AIRSECURE system also provides the added benefit of minimizing the transmission of natural diseases through the ventilation system.

In the presentation results of the laboratory tests of the developed components and preliminary field tests will be presented.

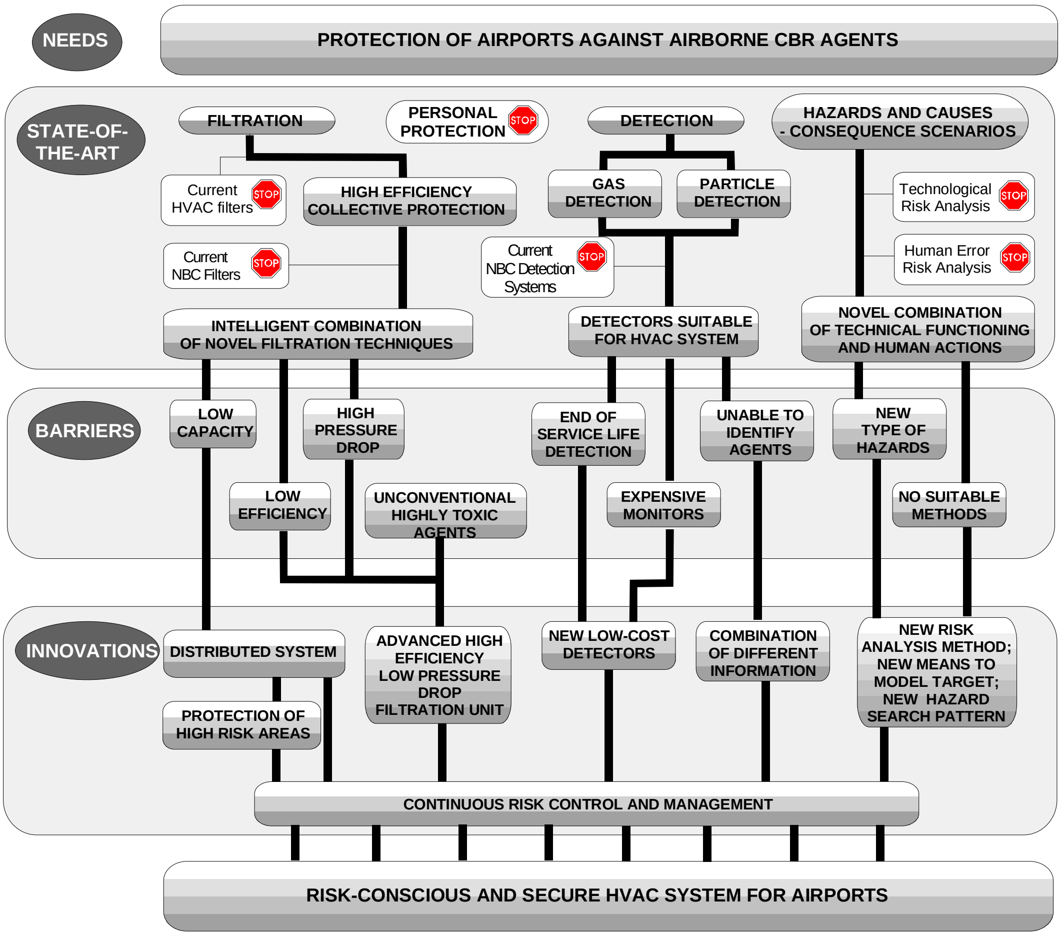

Figure 1: Technology map for risk-based detection and protective filtration system.

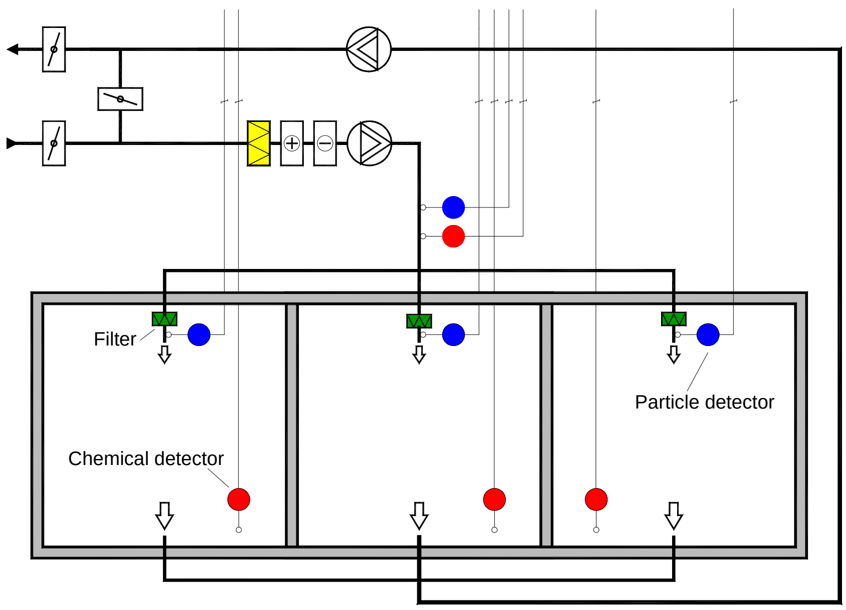

Figure 2. Principle of the AIRSECURE system. The exact location of components is based on the risk analysis and required protection levels.

ACKNOWLEDGEMENT

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Numerical Simulations For Particle Penetration Through Buildings Cracks

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SUMMARY

This study is in the frame of indoor air quality from the particle pollution point of view and its goal is to determine the particle penetration factor for different common types of leaks which deteriorate the filter capacity of building envelope. The most important parameters for the particle penetration are: the crack geometry, the pressure difference and the particles' diameter. CFD numerical simulations have been done for rectangular cracks with 0.2mm and 1mm height for 40mm and 94mm length, also changing the pressure difference between 4Pa and 10Pa. The penetration coefficient showed to be bigger than 90% for particle diameters less than 3µm for cracks with 1mm height regardless their length or pressure difference. For the diameter span from 3µm to 8µm the penetration factor varies from 90% to 15% depending on the pressure difference and only the particles bigger than 8µm are completely filtrated by the studied cracks. We have found good agreement between the numerical simulations and bibliography.

INTRODUCTION

People spend most of their time indoors being exposed to the pollution level determined by the inside sources as well as by the change realized between the inside and outside environment through the building envelope. Outside air enters generally by three means: mechanical ventilation, natural ventilation and infiltration. If the first two mechanisms can be almost controlled, the last one refers to the undesired air flows through the cracks of the building envelope. The infiltrated outside air is charged with particles characterized by diameters in the most dangerous domain for human health. The leaks are present in the building envelope, even if they are undesired, around the windows, doors and at the connections between the walls and ceiling. Common cracks are characterized by changing properties in time and space being so very difficult to establish a relation between them and the ones that can be reproduced in laboratory tests or in numerical simulations.
In order to determine the characteristics of infiltration it is necessary to establish the summa of the parameters that influence it and the correlations between them. An overall characteristic number for the process of infiltration it is called the "coefficient of penetration" and states the degree of outside pollution level which is encountered in the indoor climate. The parameter is defined as the ratio between the number of particles encountered at the inside end of the crack and the number of particles assumed to be entered in the outside end of the crack for a certain particle diameter or a granulation range. The shell filtration efficiency can be then easily determined as the difference to unit of the coefficient of penetration.
In residential environments without the operation of mechanical ventilation, infiltration becomes the dominant ventilation mode, while penetration determines how much of ambient

particle can be brought from outside into the indoor environment [1]. The effect of particles size from 0.02µm to 10µm on penetration coefficient was evaluated [1] by Chao et al. and they established a peak of 0.79 at the size range of 0.853-1.382µm. For the domain of 4.698-9.647µm the penetration coefficient showed the minimum value of 0.48.
Liu and Nazaroff proposed a physical model for the evaluation of the penetration factor [2]. Its main components were defined as partial factors and were due to gravitational settling, surface adherence by means of van der Waals forces and Brownian diffusion and impact settling induced by crack bends. They also estimated the air flow characteristics in cracks by respect to the crack geometry and the pressure difference between the crack's ends. The penetration coefficient was calculated for a range of particle diameters from 0.001µm to 100µm, for a pressure difference domain between 4Pa and 10Pa and they concluded that the domain where the penetration factor is different of 0 or 1 is for the particle diameter between 0.5µm and 10µm for simple rectangular cracks.
Liu and Nazaroff later presented the results of laboratory experiments [3] meant to confirm the accuracy of the physical model and to determine its applicability range. In their paper [3] they experimentally studied the same simple rectangular cracks in order to determine the correlation between the physical theoretical model and experiment. Generally speaking the correlation was in good terms especially for the most sensible domain of particle diameters ranging from 0.5-7µm but some discrepancies were found. They explained their origins in the influence of crack material roughness versus boundary layer thickness and boundary layer distribution functions. One of the conclusions was linked to the recommended mesh dimension that must be of the same order of magnitude with the boundary layer determined as a value of 100-400µm for particles averaging 0.3µm in a crack of 0.25mm height at a pressure difference of 4Pa. They also stated that for particles larger than 0.4µm, where gravity became to control deposition, roughness appears to be relatively unimportant, and the smooth surface model generally confirms well to the experimental data.
Regarding the influence of the pressure difference applied on the crack upon the coefficient of penetration, Mosley et al. [4] showed out for the particles of 2µm an increase of the percent value of the penetration factor from 2% at 2Pa to 40% at 5Pa reaching 85% at 10Pa. They linked those results to the variations of the speed distribution in the crack, as the increased value of the aerosol speed determines the advection of the deposed particles to the indoor climate. So, the flow characteristics in the crack as flow and speed distribution are critical in the pressure difference domain of 1-10Pa for cracks averaging 0.5mm in height.
Liu and Nazaroff applied a general model based on mass conservation law to particle transfer from one chamber to another through real windows (both domestic and industrial types)[5]. The measurements were used to quantify general factors introduced in the equations with the purpose of easily evaluating other functioning circumstances and determining a quality factor for the window by respect to the penetration coefficient. They stated that the distribution of window leakage dimensions determines the overall performance of particle filtration rather than the leakage area itself as a global value. The results indicated that airborne particles of 0.2-3µm have greater penetration factors than particles of less or bigger diameters.
Nowadays, as the requests for indoor comfort became more and more demanding, a good approximation of the pollutant transport through the building shell is needed. Those models should be able to describe the complex geometry of the real joints for construction elements, windows and doors. Most of the data found in the literature come from experimental works or general physical models computation, but those means can be applied only on simple geometry cracks which do not characterize the real life case. The idea of applying numerical simulations for the evaluation of penetration factor of the pollutants for the real cracks offers the advantage of rapidity and low costs modeling, especially when a group of different geometries of the cracks is studied in the same project. The present paper aims to determine

the most appropriate input conditions for the numerical simulations as the air flow parameters, the turbulence model, the pollutant injection model, the boundary conditions, the mesh geometry requirements and other specific conditions in order to obtain the pollutant penetration factor. In order to evaluate the accuracy of the numerical simulations, existent experimental data were considered as a reference and compared with the numerical simulation results.

METHODS

A Computational Fluid Dynamics (CFD) program was used for the numerical simulations. The program is able to determine the trajectories and speed distribution fields and the heat and mass transfer for certain imposed boundary conditions, for a linear or non linear mesh distribution. The numerical solution techniques employed in solving the equations of mathematical models use the finite volume method.
The accuracy of the results depends on the conditions imposed in the input section of the program. So, for the general geometry of the model we chose two identical volumes of air linked by a wall provided with a single crack. The volumes were dimensioned as to be large enough in order to be considered an infinite space by respect to the flow disturbances generated by the crack in the flow. The flow engine through the crack is the pressure difference established between the two volumes separated by the solid wall. The pressure difference between the two volumes is obtained with constant pressure boundary conditions imposed on the volume fluid limits. The presence of the solid wall generates wall boundaries conditions specific for fluid flow. A non linear mesh distribution was chosen in a way to allow sufficient precision for the calculus in the zones of interest like boundaries or, more important, close to the crack, but also to diminish the total number of cells having a low refined mesh in the zone were the effects of the crack are not important. In the figure 1 with red color (darker) is represented the fluid zone and with the green color (lighter) the solid wall. It can be noticed the different zonal mesh refinement.

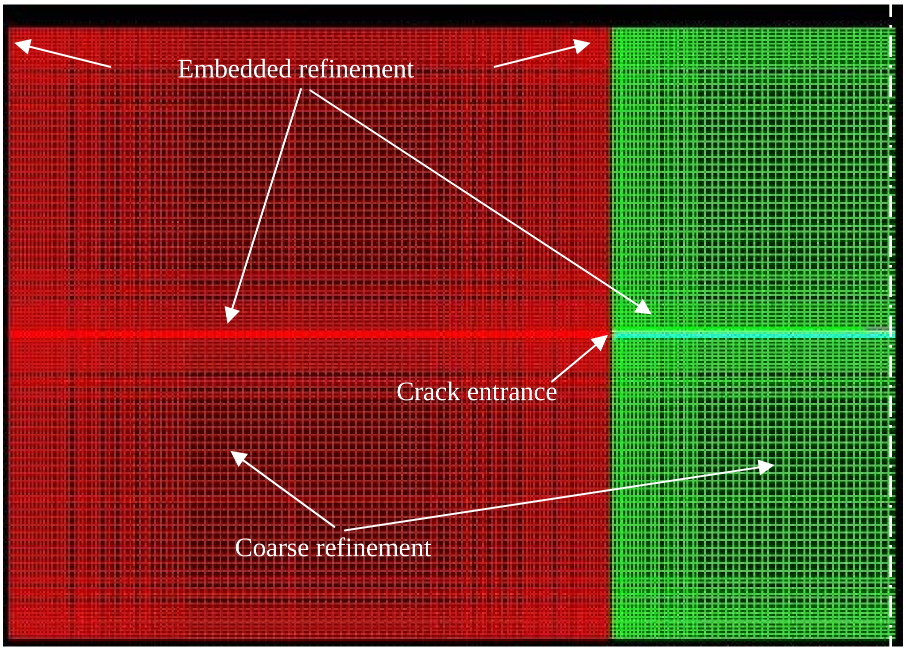


Figure1 – Zonal mesh refinement of the model

For the first step of the study we chose to simplify the model by considering a bi-dimensional case partly because the simulations are to be obtained faster and also because the data used as reference are from bi-dimensional type cracks.

The studied cracks had a simple rectangular geometry. The chosen cracks' height was 1mm for two different lengths of 94mm and 40mm. The solid wall material was defined as well finished concrete with normal roughness. According to the reference data three levels of pressure difference were studied: 4Pa representative for weather induced pressure, a value also considered as reference for the definition of Effective Leakage Area by ASTM (E779-99) and by ASHRAE Standards, 10Pa as the most common wind driven difference pressure and 7Pa as an intermediate value necessary to determine the curve evolution between the two marginal points.

The most dangerous particles' diameters span for the human health reaches from 0.5 to 10μm, so than we have chosen for our case this particles' dimensions domain. Although for this range the penetration factor for the studied cracks' conditions covers almost the entire span. For the numerical simulations the values of 0 and 1 for the penetration factor are not considered representative in the validation of the model. The injected particles were made of a solid with the density of 1000kg/m^3 and had the spherical shape.

Three different types of particles injection were studied and compared among: a punctual injection in the entrance of the crack, some punctual injection points disposed on the height of the crack and the third one, considered being the closest to real case, some points of injection situated on the isodynamic lines at the entrance of the crack. The initial speed of each particle was equal in direction and value to the air speed in the injection point (isocinetical injection).

The Lagrange model was considered for the dispersed phase flow as being the most suitable because of its characteristic of explicitly describing the interactions between particles and fluid. The complexity of the equations and the high number of unknown parameters involved is a disadvantage but did not influence the stability of the numerical solution. In this model the trajectory of the particle is given by the vector equation of the resulting force which drives the particle. The resulting force (R) generally represented by equation 1 has the components expplained in equation 2: the drag force (F_f), the fluid displacement force $(\vec{F}_{dizl.fluid.})$ and the small particle specific forces (F_{sp} - gravity force and diffusion force).

$$\vec{R} = \vec{F}_f + \vec{F}_{dizl.fluid.} + \vec{F}_{sp} \ , \qquad (1)$$

$$\frac{\pi}{6} d_p^3 \rho_p \frac{dU_p}{dt} = C_d \frac{\pi d_p^2}{4} \rho \frac{\left|U - U_p\right|\left(U - U_p\right)}{2} + \rho \frac{\pi d_p^3}{6} \frac{dU}{dt} + F_{sp}, \qquad (2)$$

Where d_p is the particle diameter, ρ_p is the particle density, C_d is the discharged coefficient of the particle, U is the mean air speed, t is the time and U_p is the relative speed of the particle by respects to the carrier fluid.

Three turbulence models were available for the present case: low Reynolds number model, high Reynolds number model and a relaxation model known as V2F. For the high Reynolds number turbulence model the wall function representation of the near-wall turbulent behavior can be not accurate, depending on the degree to which the assumptions and the approximations embodied in the distribution functions correspond with the reality of the application. Due to the lack of information regarding the distribution functions for the present case and also because the number of iterations needed to accomplish the required precision (see figure2) is higher than using other models, the high Reynolds number model was not chosen. The V2F model is a relaxation model based on the equations for the kinetic energy and its dissipation rate. The model is claimed to be valid throughout the flow domain while automatically modeling the region close to solid surface. It was designed to handle wall effects in turbulent boundary layers and to accommodate non-local effects. It has been improved to give a more accurate prediction of near wall flows including separation, heat transfer and wall friction. This model would fit very well the present application, but as it is shown in figure 2 its convergence is slower than in the case of using the low Reynolds number turbulence model. The low Reynolds model also treats the near wall region, in most respects, the same way as the interior flow, with no slip conditions imposed at the boundary surfaces. Therefore, in this model, the law of the wall is not required. In addition to these advantages the smaller number of iterations is accomplished also for low Reynolds number turbulence model and that’s why this model was chosen for the numerical simulations.

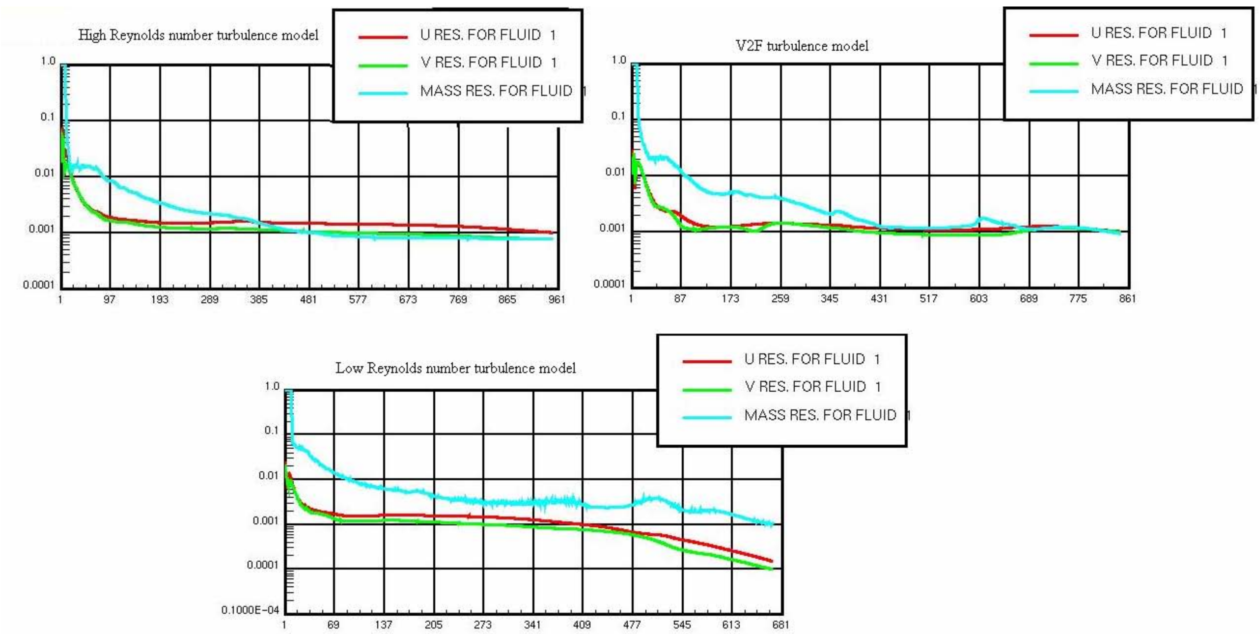
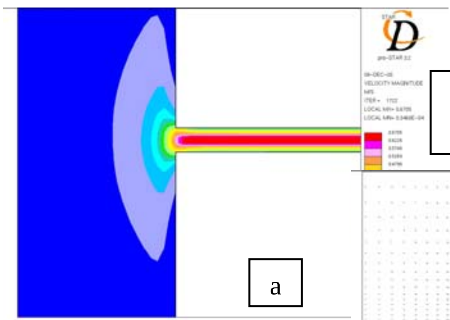


Figure 2 – Convergence residuals as functions of iteration steps for the equations matrices for the three turbulence models considered for the application: high Reynolds number, V2F and low Reynolds number

RESULTS

In order to establish the penetration coefficient for a large number of real situations the main parameters were varied and their influence quantified comparatively for sets of simulations. The parameters varied are: the crack length (the dimension of the crack along the main flow direction) 94mm or 40mm, the crack height – 1mm or 0.2mm, the particle diameter – from 1 µm to 15µm and the pressure difference 4Pa, 7Pa or 10Pa.

In figure 3 is presented an example of the calculated flow at the entrance of the crack.



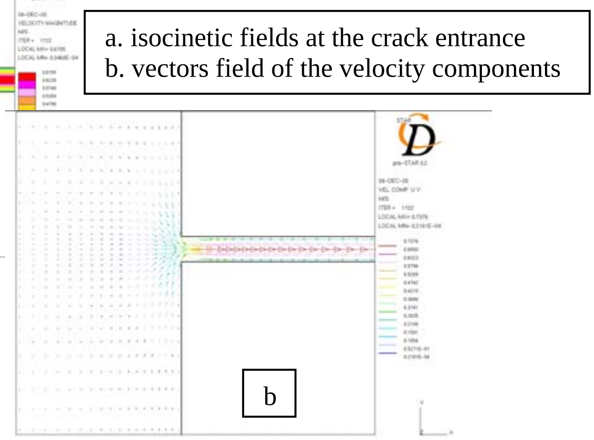


Figure 3 – Example of velocity fields at the entrance of the crack

It is important to mention both the coherence of the velocity fields and the fact that the influence of the crack in the flow is local and does not reach the boundaries where initial conditions were imposed suggesting that the overall dimensions of the chambers are correctly designed.
At first we aimed to verify the validity of the numerical model by comparing the results obtained for a crack of 94mm length, 1mm height and a pressure difference of 10Pa with the data from other authors' calculations [2] and experimental investigations [3]. In figure 4, four sets of values are represented: two of them from literature, one from the bi-dimensional CFD model and one from a simplified 3D CFD model (a small number of cells for the third dimension).

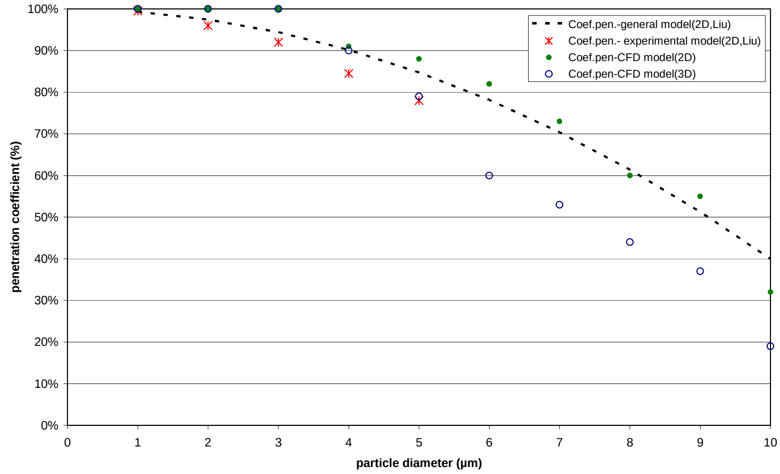


Figure 4 – Compared values for the penetration coefficient for a crack of 1mm x 94mm, 10Pa

For the CFD 2D-model the values are in accordance with those calculated or experimentally determined by other authors. That leads to the conclusion that the flow results and particles' trajectories are correct meaning that the model geometry, mesh definition, boundaries conditions and flow model were correctly set. For the CFD 3D-model the obtained results are in a logic range of values by respect to 2D case but there are no referential data to be compared with.
In figure 5 are compared the results for the penetration factor for a crack of 1mm x 94mm when the pressure parameter is varied among three referential values: 4Pa, 7Pa and 10Pa.

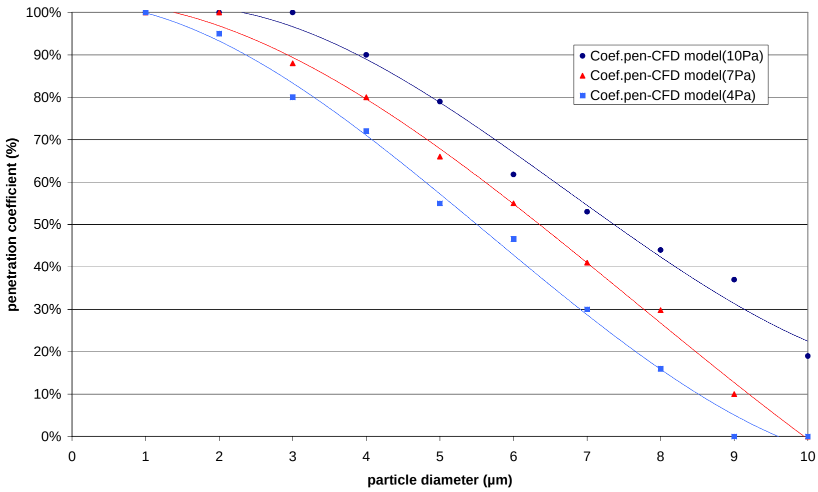


Figure 5 – Compared values for the penetration coefficient for a crack of 1mm x 94mm, at 4Pa, 7Pa and 10Pa

The values obtained clearly reflect the influence of the pressure on the process. For the particles' diameter less than 2µm the penetration coefficient is 100% regardless of the pressure difference. For the particles' diameter in the range 3µm - 8µm a decrease of the penetration coefficient with 10-60% is observed suggesting that this is the domain the most sensible to the variation of the pressure difference. For the bigger particles the penetration coefficient has low values or even reaches 0%.

The next influential parameter that was varied is the crack's length from 94mm to 40mm. The results presented in figure 6 show out the major influence of this parameter for this domain of pressure difference and crack heights. So, the ratio between the crack length and the crack height is one of the most important parameters for the filtration capacity of a construction element as confirmed by the results presented in figure 7 which represents the variation of the penetration coefficient at a fixed crack length for two different crack heights at the same pressure difference.

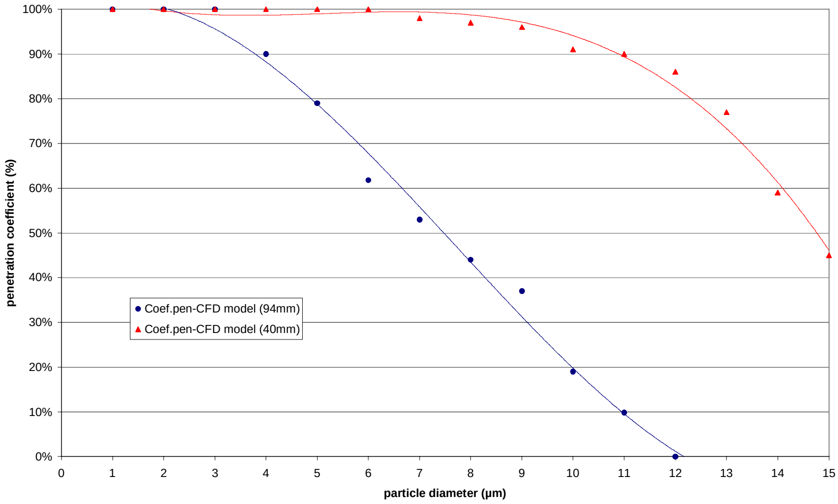


Figure 6 – Compared values for the penetration coefficient for a crack of 1mm height and 94mm or 40mm length at 10Pa

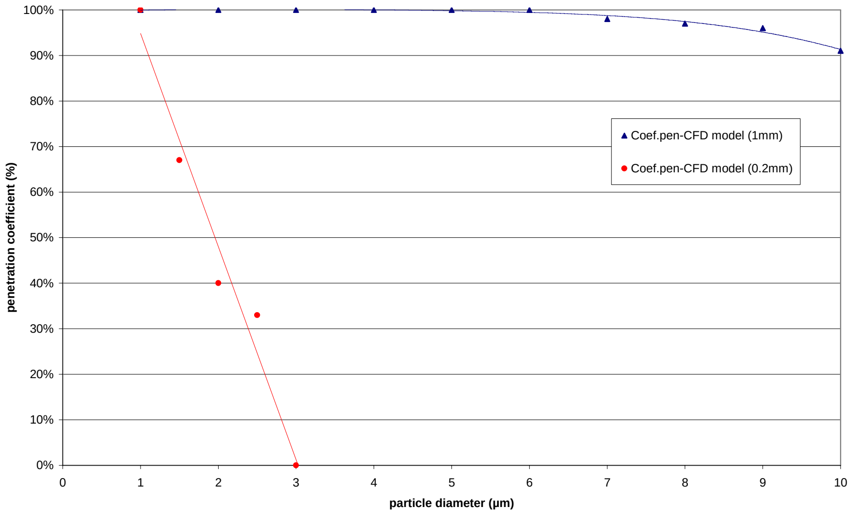


Figure 7 – Compared values for the penetration coefficient for a crack of 40mm length and 0.2mm or 1mm height at 10Pa

DISCUSSIONS

The numerical simulations showed to be in good agreement with the data from literature meaning that an extent of the application can be obtained for more complicated, close to real geometries, for a wider span of particle diameters and pressure differences. The biggest advantage is the quick and costless change of input data by respect to experimental works, but of course we intend to experimentally validate a few future more complex cases of numerical simulations.

The penetration coefficient showed to be higher than 90% for particle diameters less than 3µm for cracks with 1mm height regardless their length or pressure difference for the studied domains. For the diameter span from 3µm to 8µm the penetration factor varies from 90% to 15% depending on the pressure difference, only the particles bigger than 8µm are completely filtrated by the studied cracks.

The data obtained indicated the major importance of geometrical and physical parameters prior considered as important and permitted a global evaluation of their influence upon the infiltration phenomenon. So, for the same pressure difference, the dimensionless parameter (narrowness number) defined by the report between the crack's height and the crack's length has a major influence on the filtration efficiency of the crack. For a particle of 2µm the filtration effect decreases from 60% to 0% for an increase of the narrowness number from 1/200 to 1/100. Increasing the particle diameter to 4µm the effect becomes more evident the filtration effect decreasing from 100% to 10%. These observations determine us to further study the influence of the narrowness number on the filtration efficiency for a wider span of building cracks.

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Ozone Removal, ultra fine particles, and VOC levels on sooty supply air filters in the presence of alpha-pinene

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SUMMARY

Ozone removal, concentration of ultra-fine particles (2 to 64 nm), and VOCs were measured on sooty ventilation filters. A F8 class filter loaded by diesel soot particles in a motor laboratory and a heavily loaded F5 class filter used for 8 months in a bus service terminal were used in the tests. In addition, both filters were saturated with alpha-pinene vapor to examine possible formation of secondary aerosols by heterogeneous reactions. Both filters removed ozone effectively in the beginning. Then, the removal efficiency declined until it reached a steady state level in three hours.
Some particle formation was observed on both filters in the beginning of the test. Alpha-pinene disappeared from the air after F5 filter within three hours, whereas, its concentration remained almost on a constant level after the sooty F8 fine filter during the whole four hour test period.

INTRODUCTION

Ozone (O_3) is a strong oxidation agent in ambient air and it can cause adverse health effects in sensitive people already at low concentrations [1]. Concentration of ambient ozone exceeds health based limit values in several days annually even in Finland where concentrations are lower than in more densely populated southern areas. Ozone reacts on dusty ventilation filters and small fraction of formaldehyde is produced in these reactions [2,3]. Diesel soot increases the removal of ozone. This is beneficial because of toxic effects of ozone. However, more irritating reactive species may form in these reactions.

Ultra fine particles (i.e. secondary organic aerosols, SOA) are formed when VOCs are partially oxidized in air [4]. The reaction products are partially condensed onto existing aerosols and partially form new particles. Ultra fine particles are especially formed in homogeneous reactions between terpenes and ozone in indoor air [5, 6]. These reactions and their health significance are under intensive study.

The aim of the present study is to investigate whether ozone and terpenes can react on sooty supply air filters. Ozone removal and possible formation of secondary aerosols are investigated.

METHODS

A F8 class fine filter loaded by diesel soot particles and a heavily loaded F5 class filter used 8 months in a bus service terminal in Kuopio were used in the tests. Fine (F8) filter materials were made sooty by a vertical 3-cylinder four-stroke turbo-charged 1123 cm^3 (28.5 kW) industrial diesel-engine (Kubota V1105-TE) in laboratory at constant conditions. The engine were fuelled with regular low sulfur diesel oil (sulfur content ≤ 8 mg/kg) (Fortum DIKC 0/-10) and lubricated with fully synthetic lubrication oil of quality SAE Diesel turbo 5W-40 (Valvoline). Particle size distributions and mass concentrations of diesel soot in the dilution tunnel were analyzed by L-SMPS (Long - Scanning Mobility Particle Sizer) system consisting of a DMA3071 differential mobility analyzer (size range 15-740 nm) and CPC 3022A condensation particle counter (Thermo Systems Inc.). The sampling time was one hour. The dilution tunnel was designed according to the ISO-8178 standard [7]. Surface areas of the soot/dust samples were analyzed by BET-method (Micromeritec, model 2200) [8].

Filters were exposed in 48 L volume chamber (steel and glass) which contained 3 ml of alpha-pinene (6 petri dishes at the bottom of chamber 0.5 ml of alpha-pinene/dish). Filters were kept in the chamber for a three days to assure their saturation to the model compound.

Sooty filters were installed into a small scale ventilation system using compressed air. Compressed air was first cleaned from VOCs and NOx by activated carbon and Purafil filtration. The face velocity of air was 0.07 m/s on the filter and it was controlled by mass flow meters. Ozone was produced by a UV lamp (5W) and its concentration was adjusted to 100-250 ppb. Ozone was measured continuously (Dasibi 1008-RS) 0.5 m before and after the filters. The accuracy of the O_3 analyzer was reported to be ±0.5 ppb by the manufacturer. The device was calibrated by the Calibration Laboratory of the Finnish Meteorological Institute and its functionality was checked with an in-built calibration system. Data logger (Grant) was used to collect the measured data (Ozone, temperature, and RH).

Particle size distributions were analyzed by TSI (Thermo Systems Inc.) N-SMPS (Nano - Scanning Mobility Particle Sizer) system consisting of a DMA3085 differential mobility analyzer (size range 2-64 nm) and CPC 3025A condensation particle counter. Particle concentrations were measured in the middle of a duct about 0.5 m upstream and downstream of the filters. Sampling location was changed by using a 3-way ball valve. A half minute stabilization and three minute analyzing periods were used for a typical data collection cycle. Transport efficiency of particles undergoing diffusional deposition flowing through a sampling tube was estimated to be over 90% for the particles greater than 10 nm [9].

VOCs (Tenax GR) (sampling time 10-30 minutes) were collected simultaneously 0.5 m upstream and downstream of the filters. Samples were analyzed with an automated thermal desorption coldtrap injector (Perkin Elmer ATD 400) connected to a gas chromatograph (HP 6980) equipped with a mass selective detector (MSD 5973). The ozonolytic decomposition of the sampling cartridges was prevented by potassium iodide coated copper tubes. The analysis method has been described previously [3].

RESULTS

Ozone was removed more effectively on the sooty F8 filter than on the sooty F5 filter. Mean removal of ozone was 44% on the F8 filter and 22% on the F5 filter. Also, unused F8 filter (saturated with alpha-pinene) removed ozone effectively in the beginning of the ozone exposure, but the removal decreased to 12% within two hours and was less than 10 % after four hours. Typical ozone removal curves on sooty F8 and F5 filter are presented in figure 1.

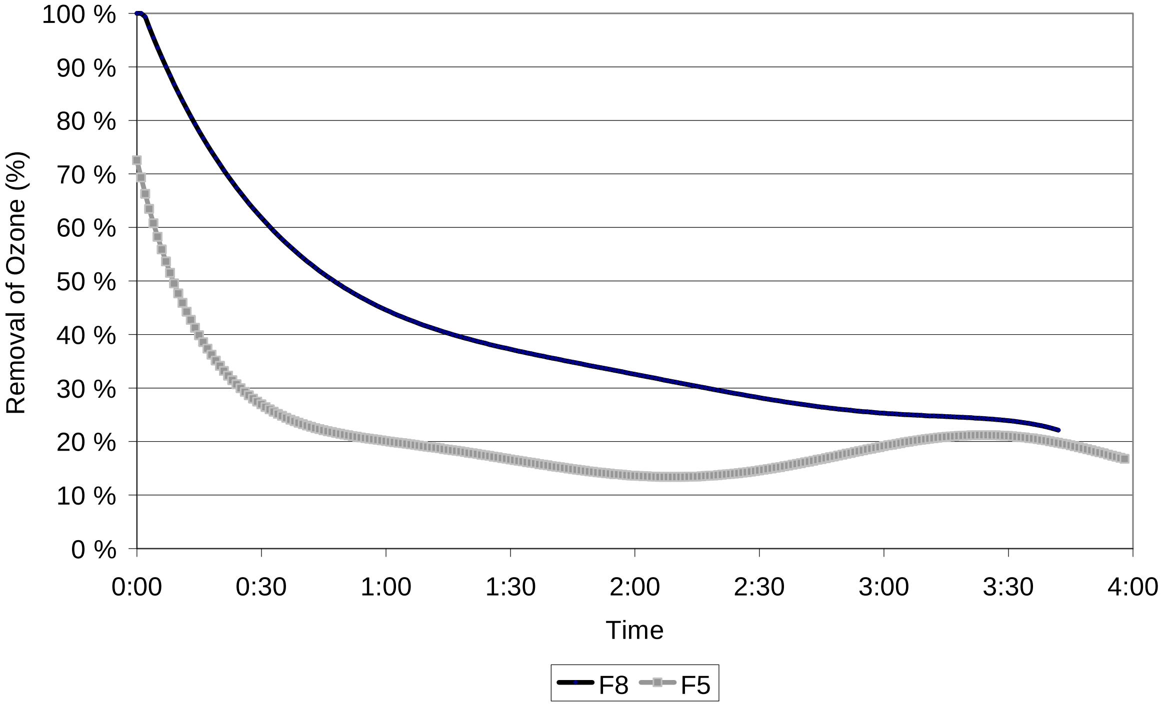

Figure 1. Removal of ozone (%) on sooty F8 and F5 filters saturated with alpha-pinene, face velocity during the test was 0.07 m/s. Conditions in F8 test: O_3 upstream 182±7 ppb, downstream 103±36 ppb, mean removal of ozone (3 hours) 44±19%, RH 10±1%, temp. 23.5±0.3°C; in F5 test: O_3 upstream 198±5 ppb, downstream 156±25 ppb, mean removal of ozone (3 hours) 22±11%, RH 16±4%, temp. 24.0±0.2°C.

Effective surface area of the dust taken from the filters in bus terminal was ca. 4 m^2/g of dust whereas soot made in motor laboratory had as large effective area as 100-103 m^2/g. Properties of soot also affected to the adsorption of the alpha-pinene onto the surfaces of the sooty filter material. Concentration of alpha-pinene decreased from 12 µg/m^3 to undetectable approximately in three hours after the sooty F5 whereas it remained at a relatively constant level of 41±4 µg/m^3 after the sooty F8 filter during the whole test period of four hours.

Particle concentrations were slightly higher downstream of the alpha-pinene saturated F5 filter (figure 2). However, there were no clear particle formation peaks during the test period of four hours.

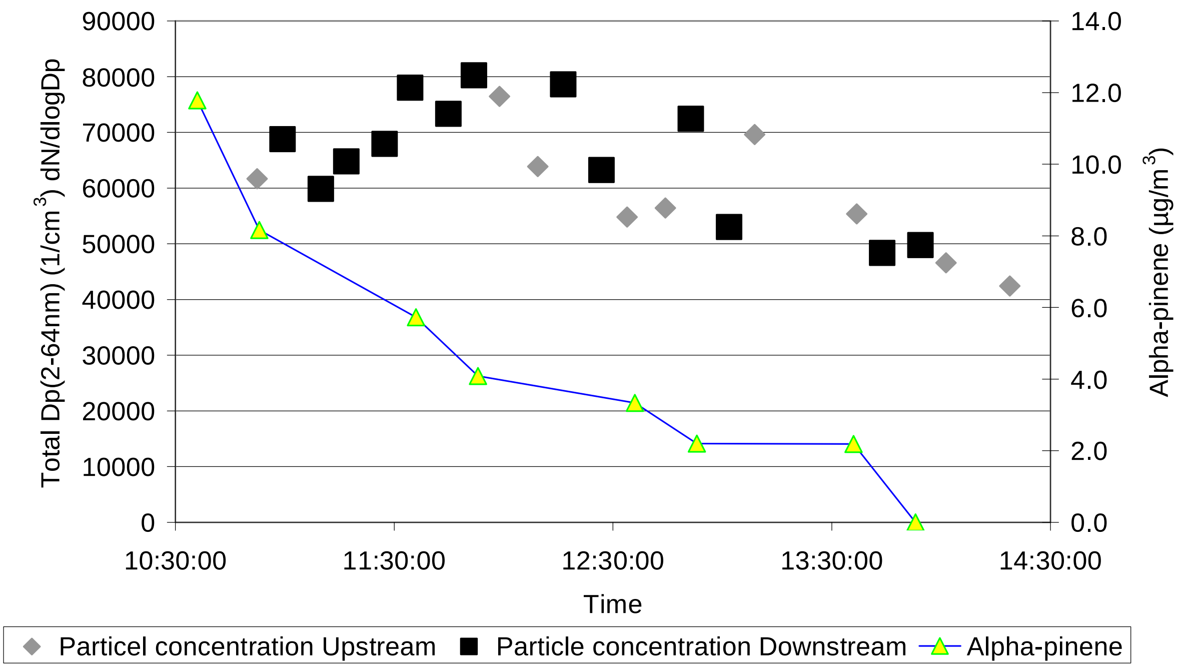

Figure 2. Particle concentrations upstream and downstream of a F5 filter (loaded for 8 months in a bus service terminal) and alpha-pinene concentration downstream of the filter. O_3 concentration 198±5 ppb upstream, 156±25 ppb downstream. Face velocity on the filter was 0.07 m/s.

Particle concentrations behaved similarly upstream and downstream of the sooty F8 filter (saturated with alpha-pinene). Slightly higher concentrations were occasionally observed downstream of the filter. The unused F8 (saturated with alpha-pinene) filter also gave similar results. When there was no ozone present, the particle concentration was negligible, only single particles were detected downstream of the filter and no typical size distribution curves were found. Overall, no clear formation of the ultra-fine particles was detected. However, temporary formation of ultra-fine particles was observed in the beginning of a few tests. An example of such phenomenon is given in figure 3. The duration of the peak was less than 3-4 minutes.

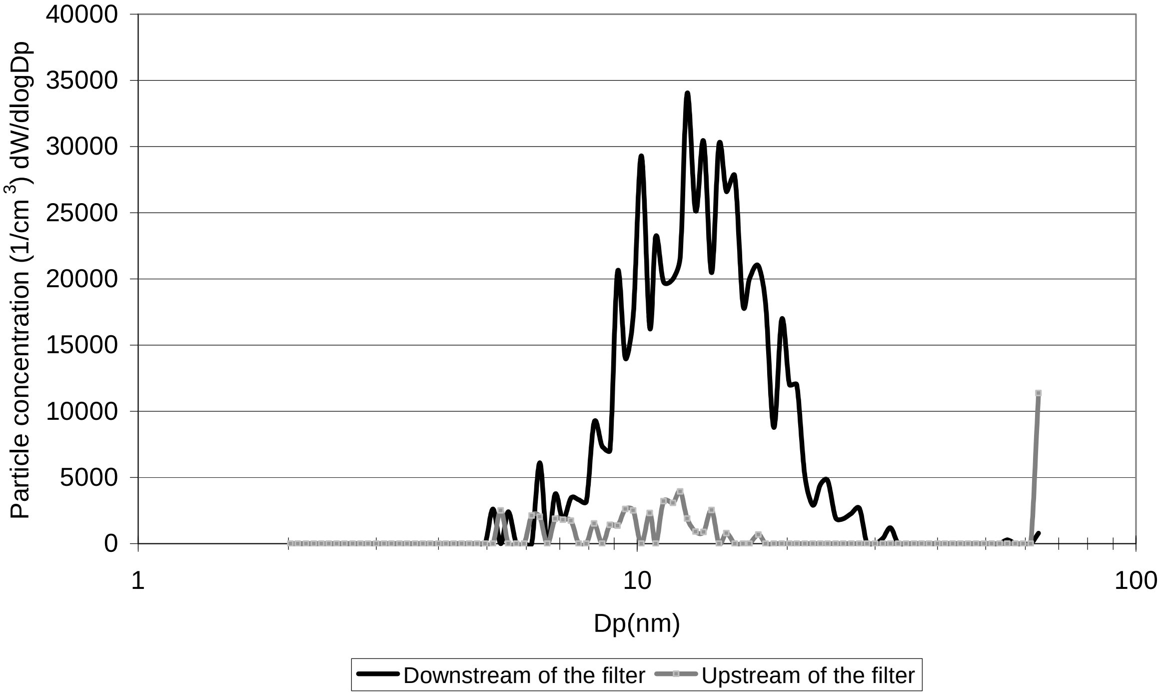

Figure 3. An example of temporary particle formation after sooty F8 filter in the beginning of ozone feeding (saturated with alpha-pinene) (2-64 nm). O_3 concentrations: 115±10 ppb upstream and 103±10 ppb downstream. Temperature 21.6±0.3°C, RH 1.4%.

DISCUSSION

Ozone was removed on sooty filters. In agreement with previous studies [10, 3], the reaction was strong in the beginning and decreased then rapidly during the first few hours. Removal of ozone was also observed on unused F8 filter.

Soot obtained from the bus service terminal had much smaller effective surface area than soot collected in the motor laboratory. This also affected the adsorption and desorption of alpha-pinene and removal of ozone on filters. This is consistent with the fact that effective area is of crucial importance in surface reactions. In addition, bus terminal filters were accumulated 8 months; and ageing of the particles may have changed their reactivity.

Sudden peak emissions of particles were occasionally observed at the start of ozone feeding when filters were still almost saturated with alpha-pinene. The results suggest that formation of secondary aerosols is not practically important on supply air filters. Pinto et al. [11] did not observe any secondary aerosol formation when terpenes were emitted into a test chamber from the plants and ozone concentration was 100 ppb, whereas clear particle formation was observed at higher ozone concentrations (200 and 400 ppb). Concentrations of the six terpenes were 2-15 µg/m^3 (alpha-pinene and limonene were the most abundant compounds). It should be noted that average residence time of air in the chamber was about 17 minutes which is much higher than in the present study where possible particle formation time was only a few seconds.

ACKNOWLEDGEMENT

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Comparison Of HEPA/ULPA Filter Test Standards Between America And Europe

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SUMMARY

This paper compares the difference of HEPA/ULPA filter test standards between America and Europe from test procedures, aerosol types and its size, to air filter classification. Both of them adopt MPPS method as their trend. According to EN1822, it is recommended to combine the test rigs for efficiency test and leakage test in practice. Influence of different scanning velocity on sampling accuracy, leakage and efficiency test accuracy need further studied and reasonable scanning velocity should be fixed. According to Rongyi Zhao's theoretical analysis, dilution method to test high test aerosol concentration may not be correct, and reasonable test methods instead of dilution need to be further studied.

INTRODUCTION

High efficiency particulate air filters (HEPA) and ultro-low penetrating particulate air filter (ULPA) are widely used in pharmacy, hospital, food processing and microelectronics, they are one of the most important terminals in clean space. In order to guarantee product quality and environmental control, every country makes its strict test standards.

During the late 1950s, China adopted oil mist method from USSR. At the early 1960s, oil mist method was improved to meet requirement of collective safeguard equipment and the development of atomic energy engineering. In 1963 and 1965, sodium flame test rig was successfully developed respectively to test filter media and HEPA. In 1970s, especially after the economic reform and opening-up policy, oil mist method and sodium flame method were used widely, therefore in 1980s national standard GB6165 was made using both methods.

As the development of microelectronics, the orginal standards using sodium salt and oil mist are old-fashioned. Before standard revising, review of test standards is very needed. This paper compares HEPA and ULPA filters between America and Europe from test procedures. Some problems and future development are proposed here for benefit of new standards.

DEVELOPMENT OF HEPA/ULPA FILTER TEST STANDRDS

America

In 1956, MIL-STD-282<Filter Unit, Protective Clothing, Gas-mask Components and Related Products: Performance-Testing Methods> using DOP method was published. DOP liquid is heated and vaporized, then condensate to become DOP particle with mean diameter 0.3μm, which is called thermal generation. It is used to test HEPA filters on Q-127, Q-76 and Q-107 test rigs. Till now it has been revised for 4 times and the latest one is MIL-STD-282:1995.

In 1959, Underwriters Laboratories Inc. published ANSI/UL 586:1959<Air Filter Units>. It is mainly used to test penetration under ambient environment and after hot air, wet air and low temperature treatment. After 8 times revision, the latest version is ANSI/UL 586:2004.
In 1975 and 1976, ANSI N45.8 committee published ASME N509 and ASME/ANSI N510 respectively, and revised them in 1980 and 1981. In 1976, this committee was recognized as “American Society of Mechanical Engineers Committee on Nuclear Air and Gas Treatment”. In 1986, ASME AG-1-1986<Code on Nuclear Air and Gas Treatment> was issued. After several revisions, the latest version is ASME AG-1-2003.

In 1980s, Los Alamos National Laboratory (LNAL) reviewed various HEPA filter testing practices, and pointed out the pro and con of MIL-STD-282 method. LNAL proposed alternative method “High Flow Alternative Test System”, which uses standardized Laskin nozzles and impactors instead of DOP generator to produce aerosols, and uses laser aerosol spectrometer instead of aerosol measuring instrument on Q-107. The test result is equivalent with that of MIL-STD-282 method, and notably efficiency corresponding to certain particle size is determined. However, because the long test time and increased data error caused by particle sizing instrument, it wasn’t popular[1].

During 1992 and 1993, Institute of Environment Sciences and Technology (IEST) issues a series of HEPA/ULPA filter test standards for different requirements and applications, such as IEST-RP-CC007.1:1992<Testing ULPA Filters>, IEST-RP-CC001.3:1993<HEPA and ULPA Filters> and IEST-RP-CC-006.2:1993< Testing Cleanrooms>.

In 2005, IEST issued IEST-RP-CC001.4 <HEPA and ULPA Filters> and IEST-RP-CC034.2<HEPA and ULPA Filter Leak Tests>

Europe

In 1984, EUROVENT issued EUROVENT 4/4 < Sodium Chloride Aerosol Test for Filters Using photometers>. This standard tests efficiency and pressure drop of HEPA filters using sodium flame methods, and then filters are classified. The method was originated from BS3928-1969, which uses mass mean diameter 0.6μm NaCl aerosols. Different concentration of salt mist will cause different color from hydrogen gas flame, so according to the sodium flame color from upstream and downstream samplings, efficiency is determined.

However, since sodium flame methods are not suitable for testing filters with large air flow rate and high-tech requirement, working group 2 of the Technical Committee 195 from CEN began to draft a new standard from 1990. At first many people inclined to accept BS 3928 for its wide use, but at last a completely new test method was preferred. The reasons are[2]:

(1) A consistent and logical classification system covering HEPA/ULPA filters is needed;
(2) The method can satisfy the need of clean room users, where room classes always differ by a factor of 10;
(3) The relationship between leak sizes and efficiency of a specified class is required;
(4) The correlation between different test rigs and fewer errors should be met.

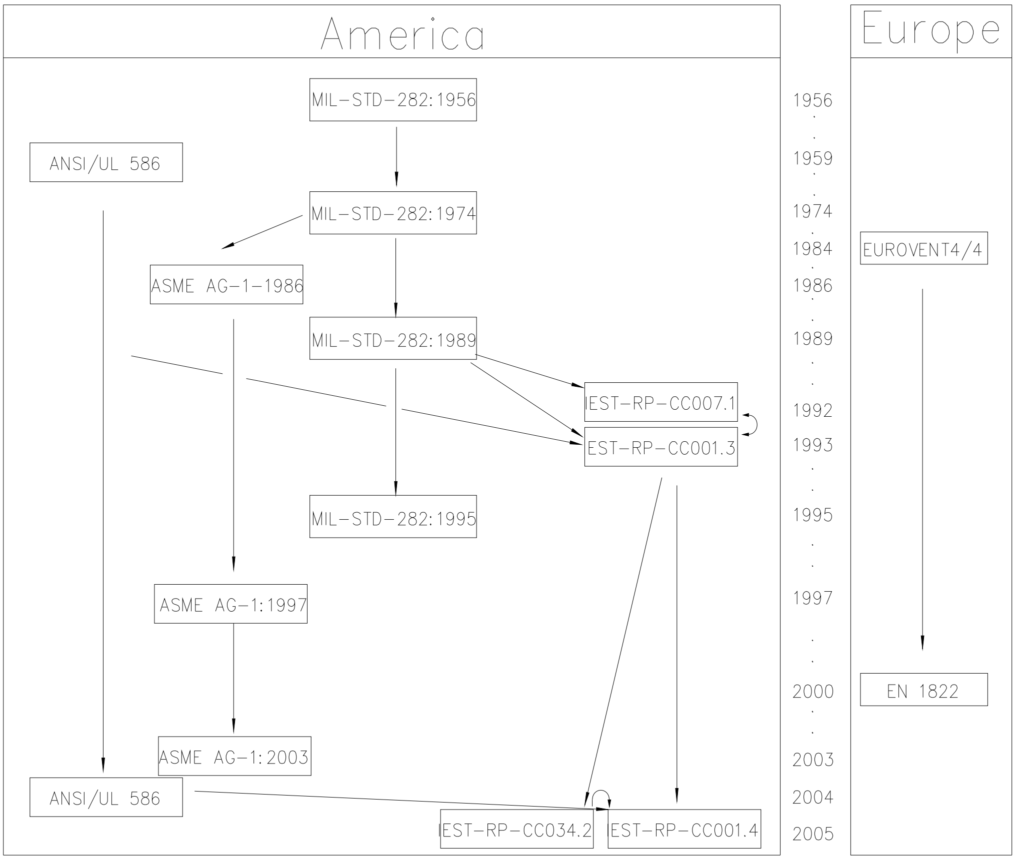

Figure 1. Development of standards for HEPA/ULPA filters.

Through about 10 years' endeavor, during 1998 and 2000, CEN issued EN 1822 in succession. Since EN 1822 is issued, disorder situations have been change in Europe. Compared with former standards, the biggest feature is that Most Penetrating Particle Size[3] (MPPS) efficiency is measured for filter medium and filters.

COMPARISON AND DEVELOPMENT TENDENCY

At present, main HEPA/ULPA filter test standards among Europe and America are MIL-STD-282:1995, IEST-RP-CC001.3:1993, IEST-RP-CC007.1:1992, ANSI/UL 586:2004, IEST-RP-CC001.4:2005, IEST-RP-CC034.2:2005, EN 1822. In the following standards from IEST and CEN will be compared.

Table 1. Comparison of main HEPA/ULPA filter test standards.

	IEST-RP-CC001.3	IEST-RP-CC007.1	IEST-RP-CC001.4	IEST-RP-CC034.2	EN 1822
Test Aerosol	Thermal DOP (HEPA filters) or any aerosol of IEST-RP-CC007(F type filters)	Mineral oil, corn oil, olive oil, oleic acid, DOP, DOS or PSL	Thermal DOP (HEPA filters) or any aerosol of IEST-RP-CC007 (F~K type filters)	DOP, DOS, mineral oil, oleic acid, PAO or PSL	DOS, DOP or paraffin oil
Aerosol size	0.3μm (HEPA filters), 0.1~0.2μm (F type filters)	Polydisperse or 90% in 0.1~0.2μm	0.3μm (A~E filters), MPPS (G type filters), others 0.1~0.2μm and 0.2~0.3μm	n/a	MPPS
Sampling Instrument	Photometers (HEPA filters), particle counter(F type filters)	Optical counter, electrical mobility particle sizer, diffusion separation instrument	n/a	n/a	Optical counter or condensation nucleus counter
Temperature and relative humidity	n/a	10～37.8°, 30～70%	n/a	n/a	23±5°, ＜75%
Pressure	n/a	Positive pressure	n/a	n/a	n/a
Qualification Requirement	(1)air velocity uniformity across duct cross*; (2)aerosol uniformity across duct cross**; (3)pressure drop under rated air flow; (4)test duct leakage test; (5)inlet HEPA filters; (6) velocity uniformity measures; (7)cleanliness of test air; (8)aerosol concentration of upstream; (9)scanning velocity; (10)sampling rate; (11)particle loss; (12)system response time; (13)correlation ratio; (14)aerosol neutralizer;				
	(1)(2)(3)(4)(8)(9)(10)	(1)(2)(4)(5)(6)(7)(11)(12)(13)(14)	(1)(3)	(1)(2)(9)(10)	(2)(7)(10)(11)

* air velocity uniformity of IEST-RP-CC001.3 and IEST-RP-CC001.4 is ±20%, while that of IEST-RP-CC007.1 is ±10%;
** aerosol uniformity of IEST-RP-CC007.1 and IEST-RP-CC034.2 is ±20%, while that of EN1822 is ±10%.

Test Procedure Comparison

IEST-RP-CC001.3 uses thermal DOP method for A~E type filters and “laser test” for F type filter. DOP method uses photometers while the latter uses particle counters. IEST-RP-CC001.4 determines penetration in the same way and are both suitable for testing submicron particle efficiency≥99.97%. By comparison, the main difference includes:

(1) IEST-RP-CC001.3 includes scanning leakage test, while IEST-RP-CC001.4 doesn’t and IEST-RP-CC034.2 is made for the leakage test only;

(2) The leakage test of IEST-RP-CC001.3 is used to test HEPA/ULPA filters in the lab, while IEST-RP-CC034.2 requires testing HEPA/ULPA filters during factory

manufacture, field installation, and after installed in the clean room and unidirectional air cleaners. The methods in IEST-RP-CC001.3 includes: photometer scanning method (for filters with penetration lager than 0.001%) and particle counter scanning method (for filters with penetration smaller than 0.001%), but the method is limited if the background counts are less than 10% of the maximum acceptable leakage count. IEST-RP-CC034 not only introduces the two different scanning methods, another special leakage test method is introduced for filters difficult to access: Aerosol photometer total leakage test method. By introducing aerosol into the test duct, concentrations of upstream and downstream and total leakage are determined.

(3) Compared with IEST-RP-CC001.3, A~F type filters classification are mainly the same. The differences are:
1. the test aerosol for C and D type filter can be PAO or DOP, because the traditionally used DOP poses potentially treat to health, and both the Food and Drug Administration and US Surgeon General have accepted PAO to replace DOP as a test aerosol.
2. Filters with 99.999% minimum efficiency related to 0.1~0.2μm particles are F type filters in IEST-RP-CC001.3, while the same minimum efficiency related to 0.1~0.2μm and 0.2~0.3μm particles are F type filters in IEST-RP-CC001.4.
3. Except for the particle counter, IEST-RP-CC001.4 allows photometers for F type filters.

(4) Compared with IEST-RP-CC001.3, super ULPA (G type)~ ULPA (A type) are added. In Europe, EN 1822 is used to test HEPA/ULPA filters upon MPPS efficiency. The procedures in the standard include 3 sections: filter medium test, filter leakage test and filter overall efficiency test. Firstly, the relationship curve between filter medium efficiency and particle size is established under rated air flow, then MPPS value is fixed; Secondly, under corresponding air velocity, monodisperse or multidisperse aerosols are used to test filters' local penetration, so as to determine whether the filter has leakage, but the precondition is that the particle size of monodisperse aerosols and the number mean size of multidisperse aerosols are MPPS; At last, static measuring method or scan method are used to test the overall efficiency, of which the static measuring method use stationary sampling probes to test the concentration of upstream and downstream, and the scan method uses a scanning probe on the downstream side together with a stationary sampling probe set upstream to get the local MPPS efficiency and then the overall efficiency.

Difference[1] between EN 1822 and IEST-RP-CC001.4 includes the following:

(1) Filters with efficiency≥99.95% in EN 1822 are tested for MPPS efficiency and leakage, while in IEST-RP-CC001.4 most filters should be tested using number mean diameter (NMD) 0.3μm thermal DOP which is approximately near MPPS.

(2) Filters with MPPS efficiency≥85% in EN 1822 are called HEPA filters (in total 7 types), while Filters with NMD 0.3μm thermal DOP efficiency≥99.97% in IEST-RP-CC001.4 are called HEPA filters (in total 14 types);

(3)IEST allows photometer and particle counters (both OPC and CNC), while EN1822 only accept particle counters (both OPC and CNC) and Different Mobility Analyzers (DMA).

(4) The test range for OPC in EN 1822 is 0.1~2μm, which is divided into 6 groups; and for CNC is 0.05~0.8μm; and for DMA is 0.01~0.8μm. In IEST-RP-CC001.4, the test range for OPC is 0.1~0.3μm, while for CNC is also 0.1~0.3μm.

(5) In EN 1822, the efficiency can be tested using scanning method of leakage test, so the efficiency test and leakage test can be one test, while in IEST they are two independent tests.
(6) In EN 1822, filters with MPPS efficiency≥99.95% must be tested through scanning method, while in IEST, whether to scan the filters depends on the performance: A and H type filter only need efficiency test, while B, E and I filters need be tested through dual flow rate leakage test, and the remaining are required to be tested both.
(7) In EN 1822, percentage is usually used to reflect penetration of filters, while in IEST introduces new Par Per Million (PPM).
(8) In EN 1822 MPPS efficiency is used, while in IEST only G type filter uses, and for A~E type filters, mass mean diameter 0.3μm aerosol is used, whose NMD is close to MPPS.

Type selection of test aerosol

Since DOP was first applied by MIL-STD-282, it has been popular and most standards recently use DOP, furthermore, definition of HEPA filter is based on the DOP efficiency.

However, the vapor of transparent liquid DOP is irritative for eyes and respiration channel. When high concentration of DOP vapor is inhaled, people may become uncomfortable, such as headache, nausea, and hypoesthesia. When exposed to DOP for a long time, people tend to be poisoned, or it may even be carcinogenic. Therefore people are concerned about DOP.

America and Europe are both active in seeking new substitute, for example, in IEST-RP-CC001.4 PAO will replace DOP. As for the toxicity and safety, PAO is affirmed to be noncorrosive, stable and cheap. Now it has been used in field filter test and present instrument can produce and deal with it. However, the remaining liquid on the filters will be released gradually after test like DOP and poses pollution. In some HEPA filter test standards of America, PSL is recommended to avoid the above situation. PSL is usually used for calibrating OPC, so it can satisfy the test requirement, but it’s expensive compared with PAO.

The most important properties of a possible aerosol substance are index of refraction, vapor pressure and density, so during the seeking process, the values of the substitute should not differ too much from Table 1 of EN 1822-2.

Filter classification

In IEST-RP-CC001.4, A~E type filters are classified using DOP method of MIL-STD282; F, H~K type filters are based on the method of IEST-RP-CC007.1; G type filter is classified according to the method of IEST-RP-CC021. In EN 1822 H10~U17 type filters are all based on MPPS efficiency. Different test methods, test aerosols and test conditions will bring about different performances for the same filter, which makes the comparison between different standards difficult. IEST-RP-CC001.4:2005 lists the comparative performance.

From comparison, performances of various types of filters in IEST-RP-CC001.4 are closer to each other. The overall efficiency of A type filter is 99.97%, the same as that of B, E, H, and I type filters, while that of C and J type filters is 99.99%, and that of D and F filters is 99.999%, but test aerosols and leakage test requirements are different. In EN 1822, except that the ratio of total penetration of H10 to that of H11 is 3, others between the neighboring type are 10. In

EN 1822, except that the ratio of local penetration to total penetration, of U17 is 20, others are 5. In IEST-RP-CC001.4, the local penetration ratios are different, because errors during scanning leakage test are very big, and the permitted leakage penetration during leakage test should be equal or larger than that of efficiency test.

Existing problems and development tendency

(1) Former HEPA filter test standards were used to test 0.3μm particle efficiency, because it is believed that with the combined effect of diffusion and interception, 0.3μm particle is the most difficult one to collect, so it became the base size of DOP method. However, later research denied this conclusion, which shows that the MPPS range is 0.1~0.25μm and MPPS is different for different types of filters and air velocity [3]. From above comparison between EN1822 and IEST-RP-CC001.4, as test condition changes, MPPS efficiency and filter classification change according to EN1822, but not in IEST-RP-CC001.4. EN1822 has the characteristic of high sensitivity and accuracy, but the test procedure is complicated and test time is long, so a more efficient, simple and reliable test method should be exploited.

(2) In EN1822, both the stationary sampling probe or movable probe can be installed to test the overall efficiency, and for H13~H17 filters scanning probe is needed to test the local penetration, because the test rig and instruments between efficiency test and leakage test are alike, it is recommended to combine the two test rigs in practice for high resource utilization efficiency. C~K type filters in IEST-RP-CC001.4:2005 are the same.

(3) Rrequirements in EN1822 and IEST-RP-CC034.2 are different. In EN1822, the probe inlet air speed should less than 25% of filter face velocity, and be placed 10~50mm from the filter surface, while in IEST-RP-CC034.2, the corresponding values are±10% and 25mm. In EN 1822, the upper limit for scanning velocity is 10cm/s, while in IEST-RP-CC034.2, the value is 5cm/s for photometer scanning and 2.5cm/s for particle counter scanning. As different scanning velocity will influence sampling accuracy and then leakage and efficiency test accuracy, hence further study on reasonable scanning velocity is needed.

(4) EN1822 allows to use two counters simultaneously at upstream and downstream, or one counter with sequential sampling between upstream and downstream[4]. As for the former, the same types should be used and calibrated, and as for the latter, aerosol concentration, size distribution and uniformity in the test duct should be kept constant.

(5) Because particle counters have test range, aerosol concentration in the upstream should be controlled so as to avoid aerosol coagulation and overlap loss in particle counters. Both standards allow the use of dilution system. However, in order to verify the accuracy of tested concentration after dilution, theoretical analysis is made by Rongyi Zhao, showing that the result after dilution is incredible[5].Until now there’s no other methods to deal with high concentration test except, reasonable test methods need to be further studied.

(6) In order to control the sampling error of particle counter, non-isokinetic sampling error and particle loss due to diffusion, sediment, coagulation and impaction in the sampling tube, together with overlap loss should be reduced. Special probe should be designed to keep isokinetic, and the length of sampling tube should be shortened, and the electric performance of sampling tube should be improved, and no other parts are installed in the tube, and the sampling tube should be kept away from noise source[6].

(7) As for field leakage test for HEPA/ULPA filters, EUROVENT issued EUROVENT 4/8:1985 "In Situ Leak Test of High Efficiency Filters In Clean Spaces" using DOP method in early 1985. Because the process from manufacture, transportation and field installation is a system, each part will influence the cleanliness of clean room. Because field test is especially important for CMOS chip workshop and biosafety, IEST makes IEST-RP-CC034 specially, which covers the leakage test of each part, and it's the complete leakage test standard now.

CONCLUSIONS

In America, DOP method has a profound influence, but now particle sizing method is the trend. In Europe, test method changes from Sodium Flame method to MPPS method. It is recommended to combine the test rigs for efficiency test and leakage test in practice.

Different scanning velocity between EN1822 and IEST-RP-CC034.2:2005 will influence sampling accuracy, leakage and efficiency test accuracy, hence, further study on reasonable scanning velocity is needed.

According to Rongyi Zhao's theoretical analysis, reasonable test methods instead of dilution need to be further studied.

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Experimental Study on the Effect of Foliage Plants on Removing Indoor Air Contaminants

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SUMMARY

The objective of this paper is to demonstrate an ability of foliage plants quantitatively to remove chemical contaminants by experiments using a small desiccator for the different kinds of plant under the different luminescence and light sources, fluorescent, incandescent and light-emitting-diode (LED) lamps. The foliage plants, *Benjamin, Spathiphyllum, Areca palm* and *Concinna*, cultivated in hydroponics were used. As a result of the experiments, the effective performance of contaminant removal was obtained under lighting conditions with the different intensity distribution by wavelength for the case of pulse injection of toluene into the chamber without ventilation. The removal efficiency of toluene under an incandescent light was larger than that under a fluorescent lamp. The different removal efficiency was obtained according to the kind of LED with different intensity distribution and foliage plants. The blue LED lamp showed the highest removal efficiency among the LED lamps evaluated in this study.

INTRODUCTION

An ability of foliage plants to absorb chemical compounds from the indoor air has been demonstrated in many studies. In the report of the National Aeronautics and Space Administration (NASA) Wolverton et al. [1] showed that low-light-requiring houseplants such as *Bamboo palm, English ivy* and so on have demonstrated the potential for improving indoor air quality by removing trace organic pollutants from the air in energy-efficient buildings. In this study they revealed that the root-soil zone appeared to be the most effective area for removing volatile organic chemicals, and proposed an indoor air purification system combining houseplants with an activated carbon filter.

Wood et al. [2] carried out an investigation of the biological decontamination capabilities of indoor plants. The work showed that *Kentia palms* (*Howea forsteriana Becc.*) had capacity to remove several times the maximum occupational levels of n-hexane and benzene in indoor air, and their capacity to do so was increased by exposure. Also they suggested that the intimate association between VOC metabolizing bacteria and the roots system was highlighted by hydroponic conditions.

The removal of airborne toluene by means of the phyllosphere of *Azalea indica* augmented with a toluene-degrading enrichment culture of *Pseudomonas putida* TVA8 was studied by Kempeneer et al [3]. The results presented were promising and could be practical importance in the field of indoor air pollution control.

Sawada et al. [4] measured TVOC and odors to examine purification effect of plants by using *Golden pothos* and *Peace lily*. The results showed that the removal rate for TVOC was 74% and the one for odor was 68%. It was confirmed that plants had high purification capability in the real environment.

Wood et al. [5] obtained the results by the office field-study that the presence of potted-plants brought about highly significant reduction in TVOC levels, of up to 75% when indoor average concentrations rose above 100ppb. The findings also indicated that the potted-plant microcosm represented an adaptive, self-regulating, low-cost, sustainable system for bioremediation of VOC pollution in indoor air.

The authors has been investigated the VOC removal performance of foliage plants under the different lighting conditions, kinds of plant and light, and the illuminance [6]. It was found that the VOC removal efficiency was strongly affected by the illuminance and the intensity distribution by wavelength of light.

The processes of diffusive exchange between air and leaves through stomata and cuticle, metabolism and dilution by growth may be involved with the effect. Nevertheless the valid mechanism of biological decontamination and the quantitative performance to remove indoor air contaminants have not been established. Therefore further studies concerning the effect need to be done; especially the effects of light source on the removal efficiency of indoor plants should be investigated.

The objective of this paper is to investigate an ability of foliage plants quantitatively to remove chemical contaminants by the experiments using a desiccator for the different kinds of foliage plant under the different luminescence and light sources, fluorescent, incandescent and LED.

METHODS

An overview of the measurement system used here is shown in Figure 1. The experiments were carried out for each plant put in a transparent desiccator made of acrylic resin with dimension L560× W 483×H986 mm. Toluene gas was used as a chemical contaminant to be removed from the test chamber. The concentrations of toluene and carbon dioxide were analyzed by B&K Multi Gas Monitor. Sampling points were fixed in the lower part of the desiccator and the outside air. Artificial light was set at the above the desiccator. The illuminance was measured by the illumination photometer (Tokyo Koden). Spectrometer (SOLAR Laser Systems) with the operational spectral range, 190 to 1,100 nm, was used to investigate the intensity distribution by wavelength of lightings used here.

The test plants were set in the desiccator after exposing fresh air for 24 hours and more, and then a small quantity of VOC gas adjusted the initial concentration was injected by a syringe three hours after the measurement start. The measurement data of the VOC and carbon dioxide was obtained every 75 seconds for 24 hours. To avoid the air exchange between the desiccator and the outside, the exhaust air from the desiccator used for detecting the concentration of sampling air was returned to the desiccator. Air temperature and relative humidity in the desiccator and the outside air were also measured every 10 minutes during the experiment.

Experimental items evaluated are shown in Table 1. Toluene, one of VOCs, was used as a chemical contaminant evaluated. Fluorescent light, incandescence light and LED lights such as blue, red and blue mixed with red LED were evaluated. Each LED light was composed of 1,200 LEDs with the maximum rating 100mA per a LED set on a flat board of dimension L50 ×W45 cm. The desiccator was set in a laboratory where the room air temperature was controlled at 20-30 degree C. The illuminance of lighting was set at 350, 700, and 1,050 lx. The foliage plants, *Benjamin* (scientific name: *Ficus benjamina L.*), *Spathiphyllum* (*Spathiphyllum clevelandii*), *Areca palm* (*Chrysalidocarpus lutescens*) and *Concinna* (*Dracaena concinna*), which are considered to remove VOC effectively, cultivated in hydroponics were used.

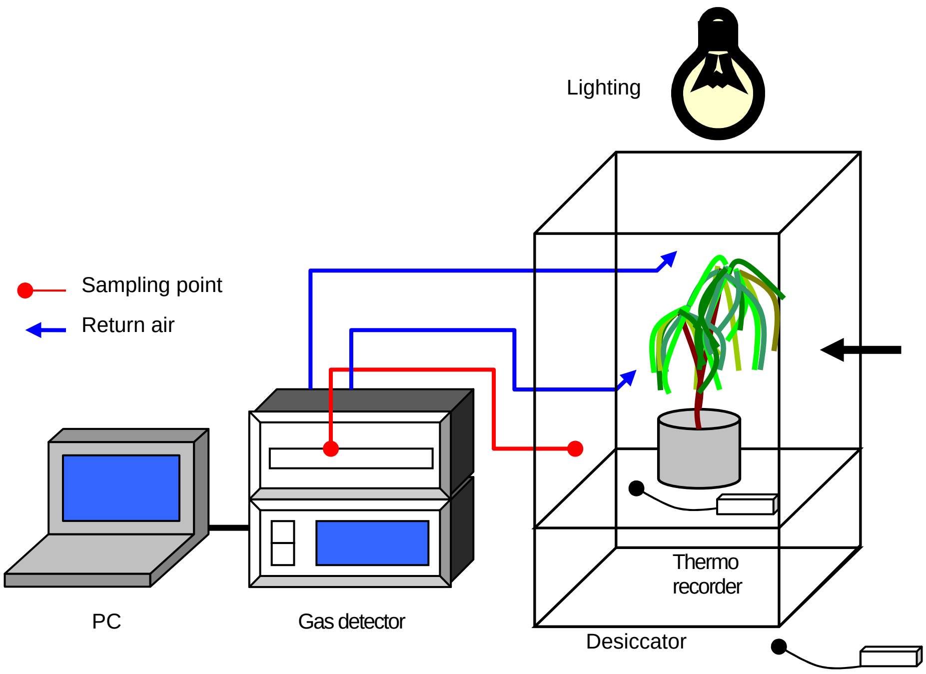


Figure1. Measurement system.

Table 1. Experimental items.

Items	Contents
Chemical contaminant	Toluene
Light sources	Fluorescent light
	Incandescence light
	Light-emitting diode (LED)
Foliage plants	Benjamin
	Spathiphyllum
	Areca palm
	Concinna
Illuminance	350 lx
	700 lx
	1050 lx
Wavelengths of the light	Red LED
	Blue LED
	Red and blue (B/R ratio 0.25)

RESULTS

Effect of light sources

To investigate the effect of light sources on VOC removal of foliage plants, fluorescent and incandescent lights with the relative intensity distribution by wavelength as shown in Figure 2 were used. The incandescent light has relatively continuous intensity distribution through the wide range of wavelength, but the fluorescent light has discrete intensity distribution and some peaks for the specific wavelength.

Figure 3 shows the cumulative removal of toluene for a pot of *Concinna* at 10 and 20 hours after a small amount of toluene injection using a syringe under the different lights, incandescent and fluorescent lights, in comparison with no lighting. The illuminance on the middle plane of the desiccator was controlled at 356 lx. The removal of toluene was strongly affected by the lighting. In this experiment the removal efficiency of toluene under an incandescent light was larger than that under a fluorescent lamp.

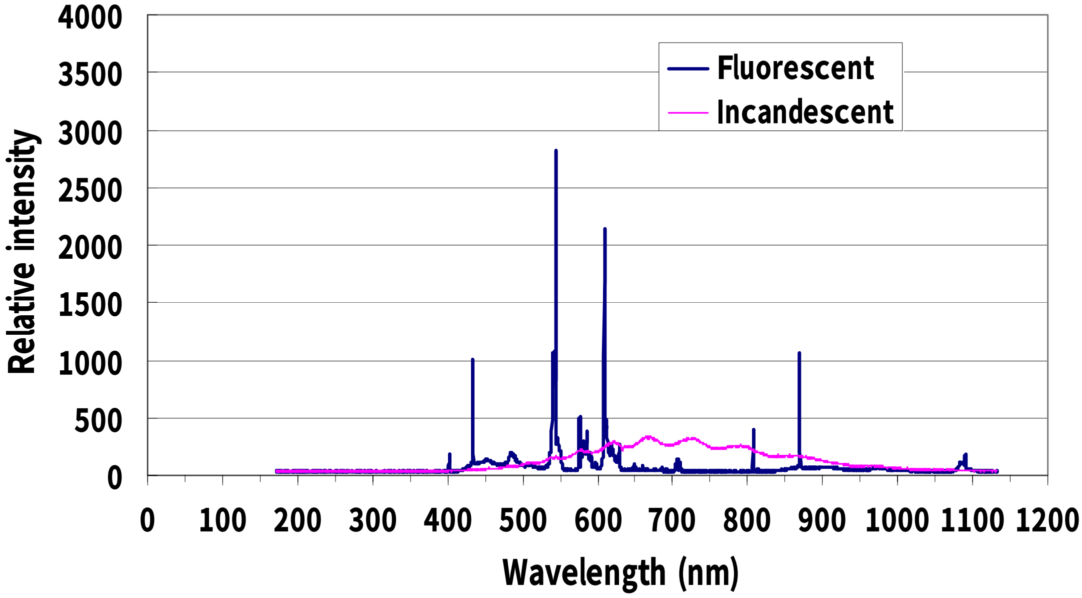


Figure 2. Spectrum distributions of the fluorescent and incandescent lights.

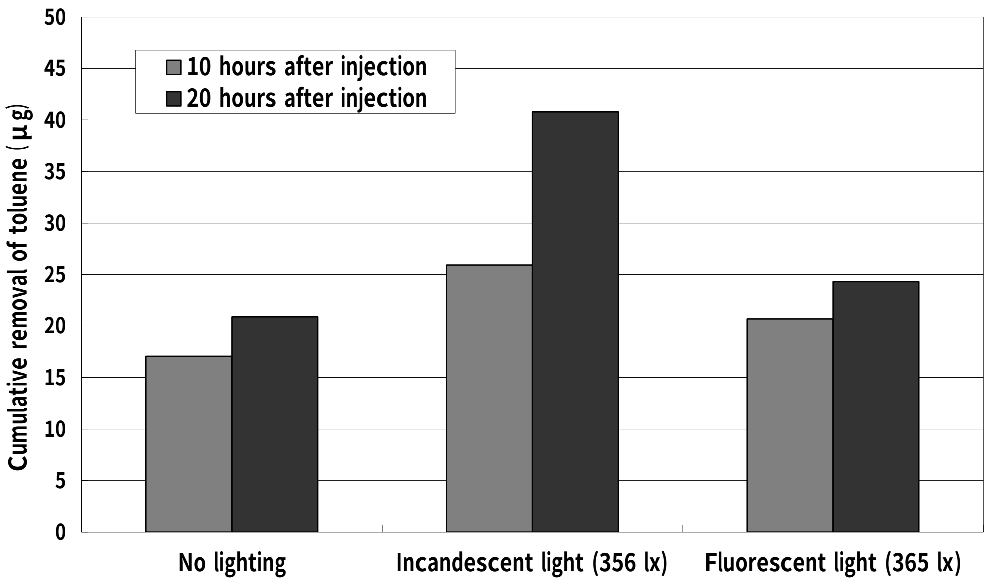
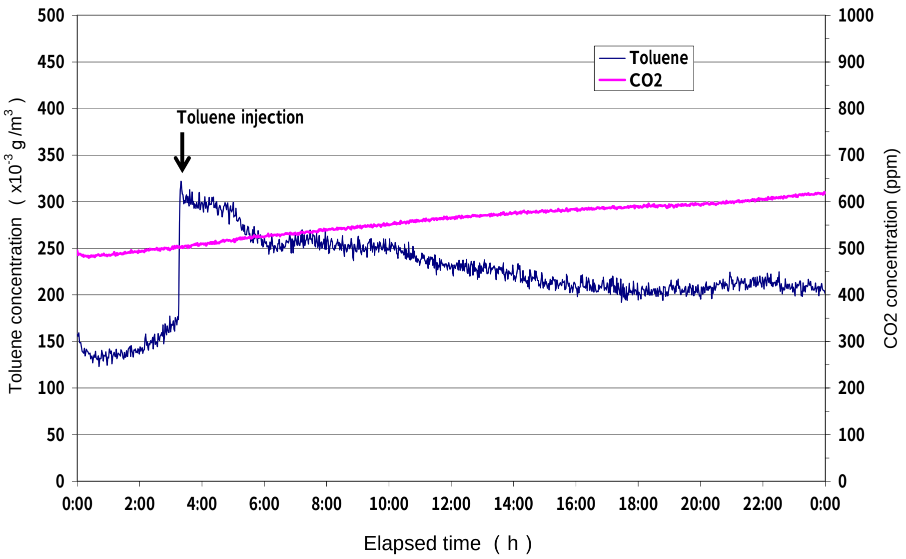


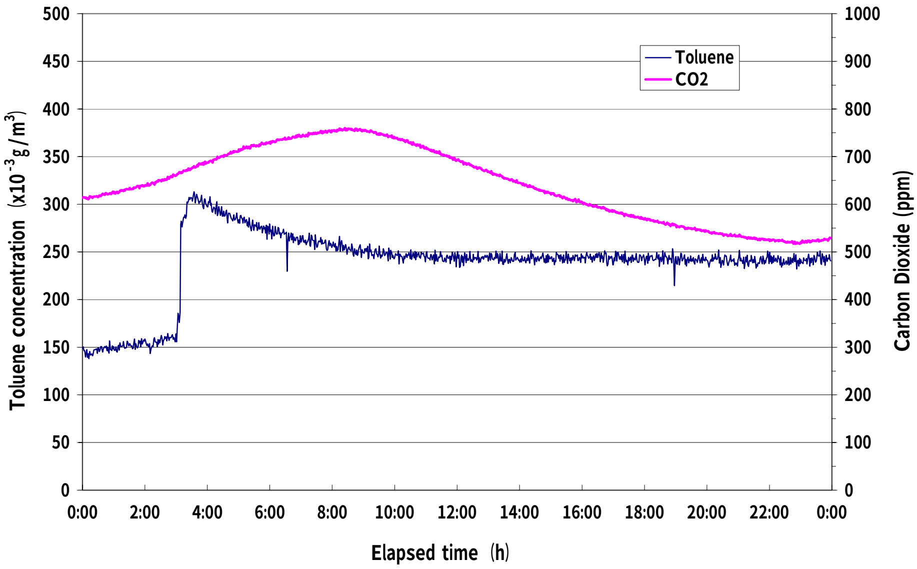
Figure 3. Cumulative removal of toluene under the different lights.

Kinds of plant and illuminance of lights

The foliage plants, *Benjamin*, *Spathiphyllum*, *Areca palm* and *Concinna*, which are considered to have high removal performance of VOC, cultivated in hydroponics were used. Figure 4 shows an example of the measurement results of toluene and CO_2 concentrations for *Concinna* under the fluorescent illuminance 700 and 1,050 lx, to identify the removal of VOC under the different illuminance. In this case the air temperature inside the desiccator was about 23 degree C and the relative humidity was changed from 30 to 80 %. Under the illuminance 700 lx, toluene concentration was decreased from 0.32 to 0.2 mg/m^3 at 15 hours after toluene injection. CO_2 concentration was slightly increased as the elapsed time. For the illuminance 1,050 lx, toluene concentration was decreased from 0.31 to 0.25 mg/m^3 at 15 hours after toluene injection. CO_2 concentration was decreased after 9 hours from the measurement start. This data shows that the photosynthesis of the plant was not active in spite of remarkable VOC removal under the illuminance 700 lx, while photosynthesis was active under the illuminance 1,050 lx.



(a) Fluorescent illuminance 700 lx



(b) Fluorescent illuminance 1,050 lx

Figure 4. Toluene and CO_2 concentrations vs. elapsed time.

Figure 5 shows the VOC removal efficiency, defined by the ratio of the concentration difference between the initial concentration and the concentration at the evaluated time to the initial value, for the different foliage plant under the different fluorescent illuminance. For *Spathiphyllum* and *Concinna*, the removal efficiency under low illuminance was over 80% and larger than that under high illuminance. *Benjamin* and *Areca palm* showed the highest efficiency above 80% for the illuminance 700 and 1,050 lx, respectively. On the other hand the efficiency for *Spathiphyllum* under the illuminance 1,050 lx was very low. These results can be considered that each plant has the particular illuminance to remove VOC efficiently.

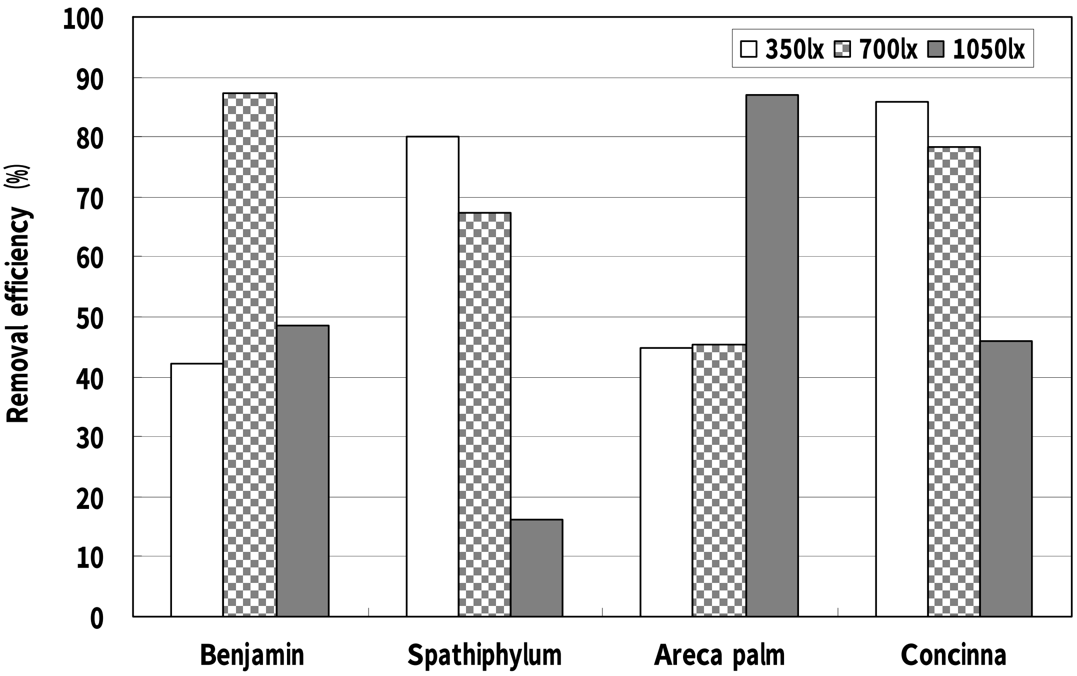


Figure 5. Removal efficiency of toluene under different illuminance.

Wavelength of the lights

Relative intensity distribution by wavelength of LED lights, blue, red and the mixed light (B/R ratio 0.25), used here is shown in Figure 6. The most part of intensity including the peak of the blue LED was covered within the wavelength range 420 to 470 nm, which range is considered to be wavelength range required for normal growth of plants [7, 8]. On the other hand, the most part of intensity distribution of the red LED is close to the wavelength range 640 to 690 nm, which is considered to be wavelength range for effective photosynthesis.

Figure 7 shows the removal efficiency of toluene under the different LED light. The illuminance of the lights on the middle of the plane in the desiccator was controlled at 350 lx. All the cases showed the removal efficiency of 60% and more. Especially the highest removal efficiency was the case under the blue LED, and the efficiency over 100 % means that the plant reduced VOC concentration in the desiccator less than the initial value before setting the plant. For the red LED and the mixing of red with blue, both of the cases were almost similar.

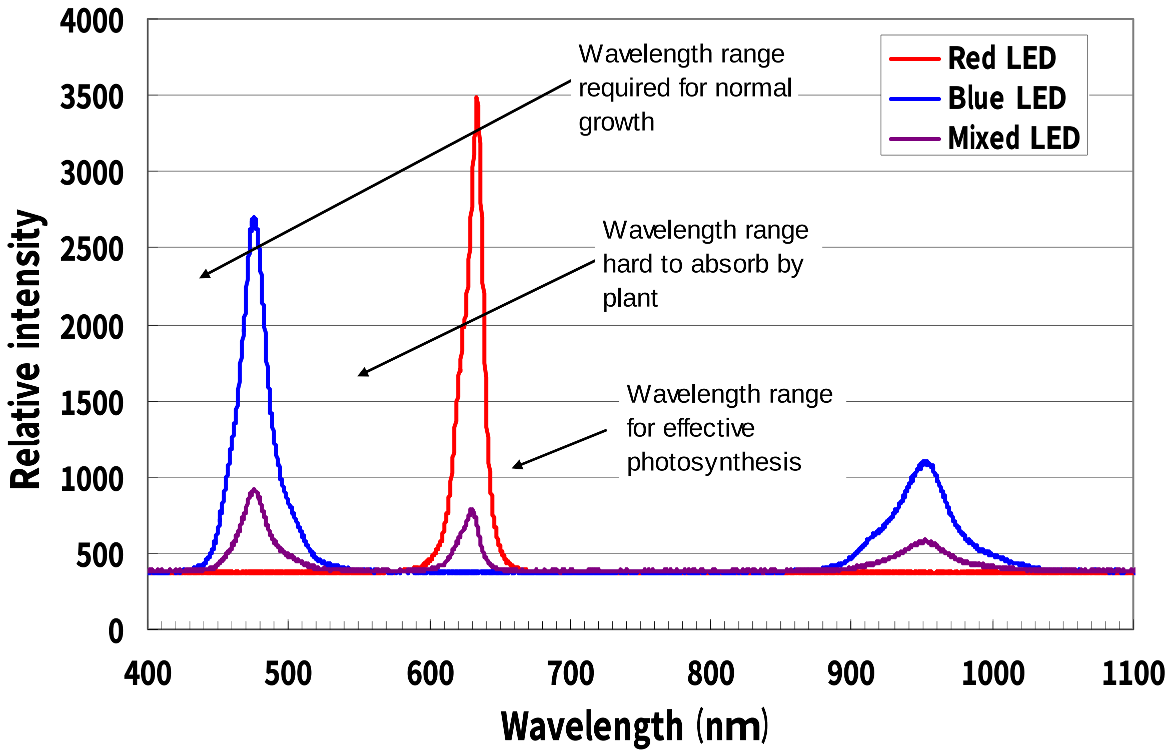


Figure 6. Relative intensity distributions by wavelength of the LED lights.

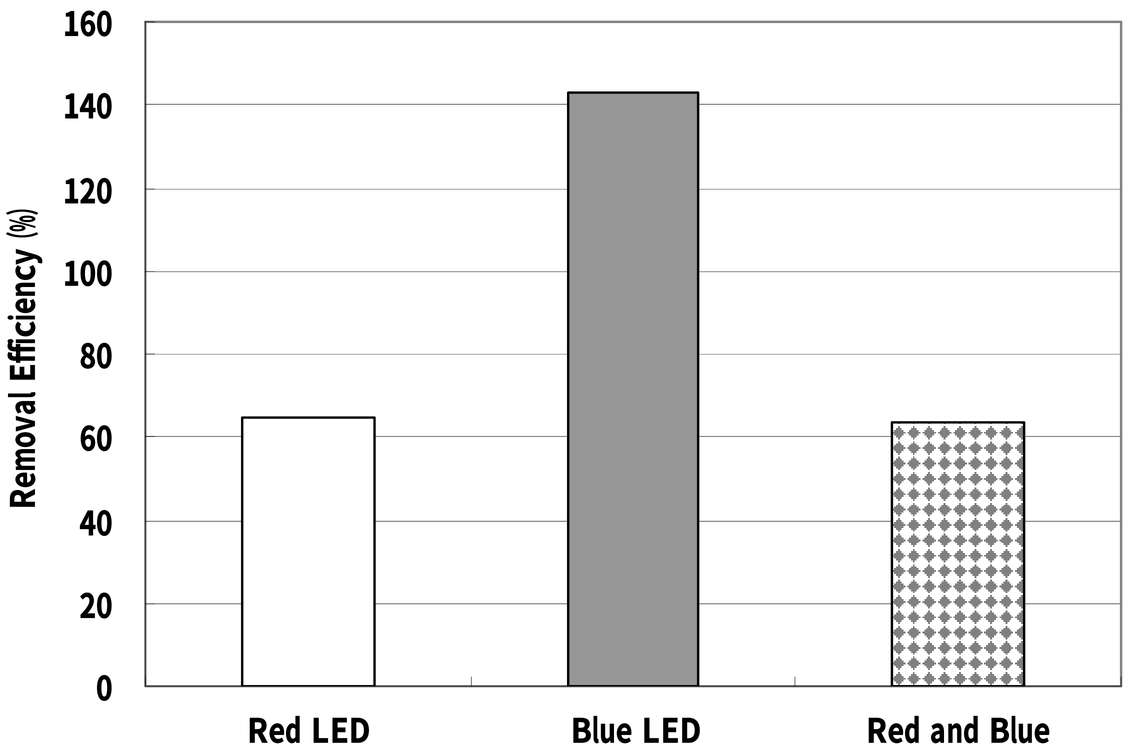


Figure 7. Removal efficiency under the different LED lights (350lx).

CONCLUSIONS

As a result of the experiments, the effective performance of contaminant removal was obtained under lighting conditions with the different intensity distribution by wavelength for the case of pulse injection of toluene into the chamber without ventilation. The removal efficiency of toluene under a fluorescent light, illuminance 356 lx on the upper part of the desiccator, was larger than that under incandescent lamps.

For *Spathiphyllum* and *Concinna*, the removal efficiency under low illuminance was over 80% and larger than that under high illuminance. *Benjamin* and *Areca palm* showed the highest efficiency above 80% for the illuminance 700 and 1,050 lx, respectively. This result suggests that each plant has the particular illuminance to remove VOC efficiently. Also it was found that the highest removal efficiency was obtained under the lighting condition with the blue LED out of the LEDs evaluated in this study.

In the next stage, experiments using a full scale room model to investigate VOC removal performance of foliage plants and further experiments using a desiccator to identify the fundamental characteristic of plants for removing VOC as well are required.

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A Study on Improvement of Indoor Air Setting by Korean Native Plants

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SUMMARY

This study conducted the experiment of concentration reduction effect if indoor air pollutants to Korean native plants, Fatsia japonica Decne. et Planch. and Ardisia pusilla DC. The two plants are advantageous in that they are highly available as they grow wild, and being easy to get. Fatsia japonica Decne. et Planch. is a plant of its wide and large leaf diverged 7 or 8 parts, which is thought to have a high effect of air purification. Ardisia pusilla DC. has a smaller leaf than Fatsia japonica Decne. et Planch., which is characterized by more leaves and beautiful. Field measurements were performed in models where the plants were placed and were not. The dimensions of the two models were equal. The concentration of Benzene, Toluene, Ethlybezene, Xylene, Formaldehyde were monitored, since they were known as most toxic materials. The concentration of VOCs was monitored three hours after the plants were placed and three days after the plants were placed. As a result, they had all an effect of reducing pollution. Especially, Fatsia japonica Decne. et Planch. was more excellent in reducing Toluene and Formaldehyde with a lot of pollutants.

INTRODUCTION

The experiment purifying air using plants was conducted by NASA since 1980s. Of late in Korea, as concern about well-being has risen, there has appeared a movement to make clean air, and improve residential environment using already purified air. This study attempts to the most Korean and effective plant by which we feel beauty growing and maintain pleasant indoor air. For this, the study grasped VOCs reduction effect produced indoors to Ardisia pusilla DC, and Ardisia pusilla DC, Korean native plants. The grasp the effect, the study compared the room with and that without plants, and measured VOCs concentration. Also to compare the effect of the two plants, the study alternated the plating and growing volume and placement, and experimented it.

METHODS

This experiment measured the concentration change of VOCs in the room with plants, and that without them in order to seize the VOCs reduction effect by plants. The kinds of measured VOCs were Benzene, Toluene, Ethylbenzene, Xylene, Formaldehyde on the measured results of which for three days reduction effect was grasped. The experiment room measures 3.5m wide, 5m long, and 2.4m long in a form of reduced apartment, which had a living room and a veranda, with the plants arranged based on the veranda part.

In the experiment regarding planting and growing amount, the plants were divided by 10% and 5% of the experiment space, and that to the placement of planting and growing, sunny spot placement close to the veranda part, and scattered placement were measured respectively. The measurement was made on the apartment measuring method(process testing method) at three in the afternoon for three days in a row after 5 hour closed state for 5 hours after ventilation for 30 minutes in the morning. The experiment room was not operative with an air-conditioner, with the condition of both rooms in all the same. BTEX was sampled with a Charcoal tube, and analyzed using GC-FIID, and HCHO was sampled with a DNPH-cartridge, and analyzed using HPLC.

Figure 1 shows the ground plan of the laboratory, and Figure 2 shows the planting amount and placement.

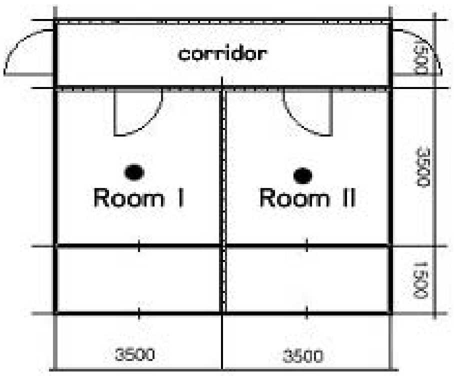

Figure 1. Plan of the space

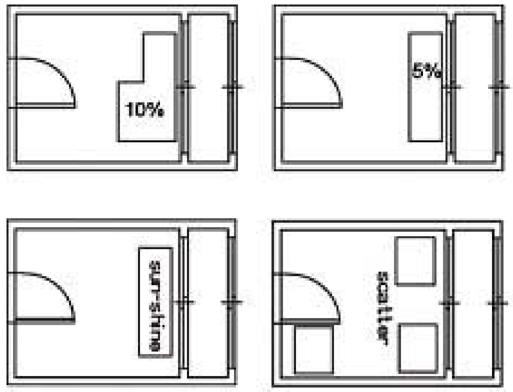

Figure 2. Layout of plant

RESULTS

3.1 Assessment to Planting and Growing Amount

1) Fatsia japonica Decne. et Planch

In case of Fatsia japonica Decne. et Planch, the reduction capability of Formaldehyde was excellent. The more plating and growing volume there was, the more reduced volume there was and as a result of plating Fatsia japonica Decne. et Planch by 10%, Formaldehyde was reduced to 212.33 μg/m³ in total from 251.00 μg/m³.

In planting Fatsia japonica Decne. et Planch by 10% in Toluene, Toluene was reduced by 12.90 μg/m³, which was less than that of Ardisia pusilla DC, showing a reduction effect.

In addition, Benzene was reduced by 6.20~11.60 μg/m³, Ethylbenzene was by 3.40~5.10 μg/m³, Xylene was by 10.40~11.80 μg/m³, Stylene was by 7.80~9.30 μg/m³. In case of setting up all plants, which was effective, where more plants, more effective.

2) Ardisia pusilla DC

Ardisia pusilla DC was excellent in reduction capability in Toluene. In Ardisia pusilla DC planted by 10%, Toluene was most reduced by 15.10 μg/m³, which was more effective than

Fatsia japonica Decne. et Planch. In Ardisia pusilla DC planted, Benzene was reduced by 9.00~12.80 μg/m³, Ethylbenzene by 5.60~7.00 μg/m³, Xylene by 7.90~10.00 μg/m³, Stylene by 4.90~5.80 μg/m³.

Also for Formaldehyde, it was less reduced than for Fatsia japonica Decne. et Planch, but in case of 5%, the reduced volume was great as 96.25 μg/m³, in case of 10%, as 149.58 μg/m³.

Figure 3 indicates the concentration change of Benzene in case of Fatsia japonica Decne. et Planch and Ardisia pusilla DC planted, Figure 4 does the concentration change of Toluene, Figure 5 does the concentration change of Ethylene Benzene, Figure 6 does the concentration change of Xylene, Figure 7 does the concentration change of Stylene, and Figure 8 does the concentration change of Formaldehyde, which is the comparative graph of the set-up case and not set-up case of all plants.

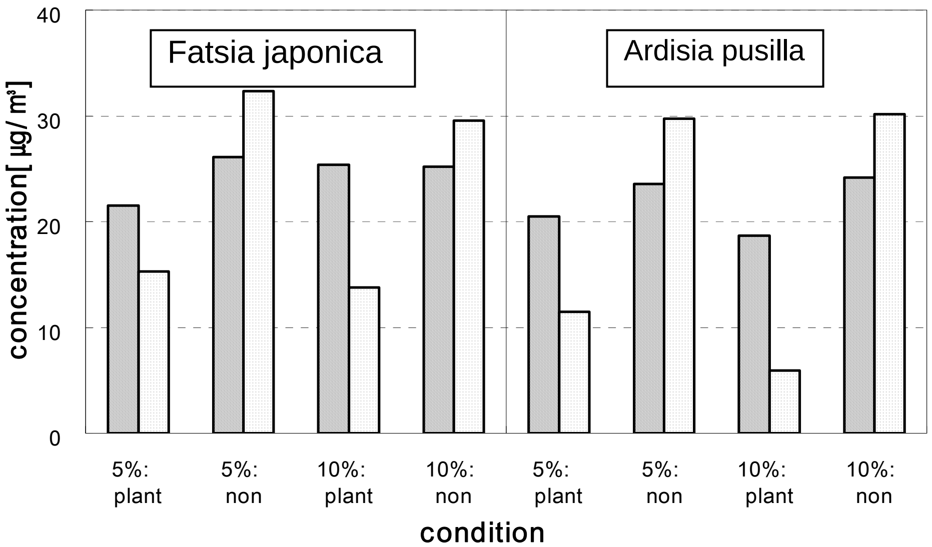

Figure 3. Variation of Benzene concentration according to the amount of plant

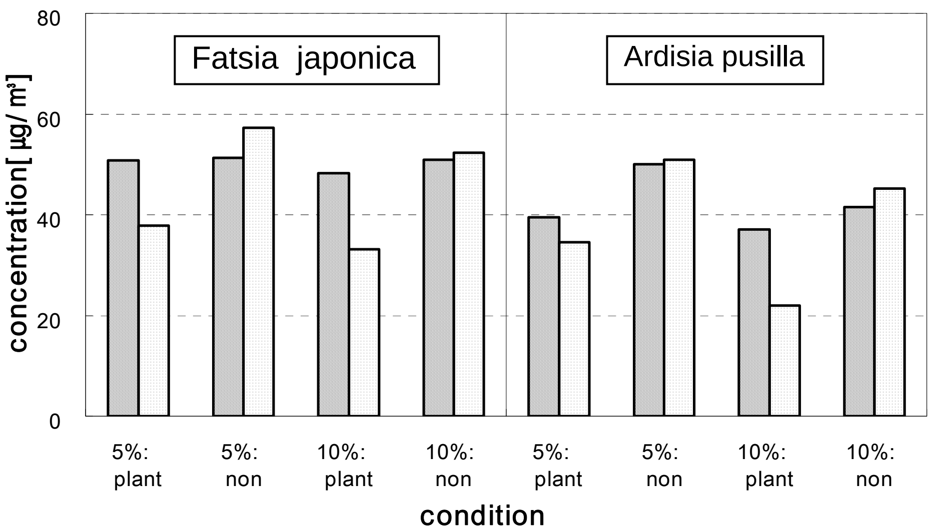

Figure 4. Variation of Toluene concentration according to the amount of plant

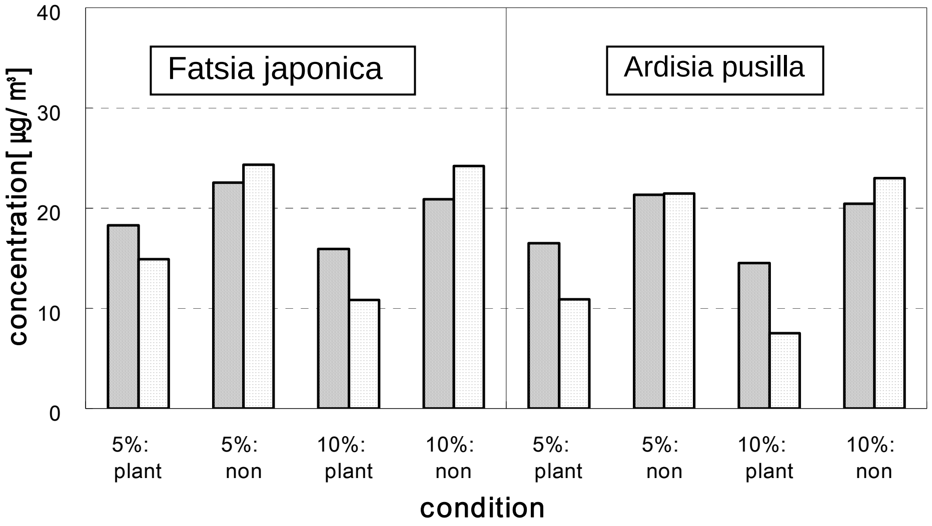

Figure 5. Variation of Ethylbezene concentration according to the amount of plant

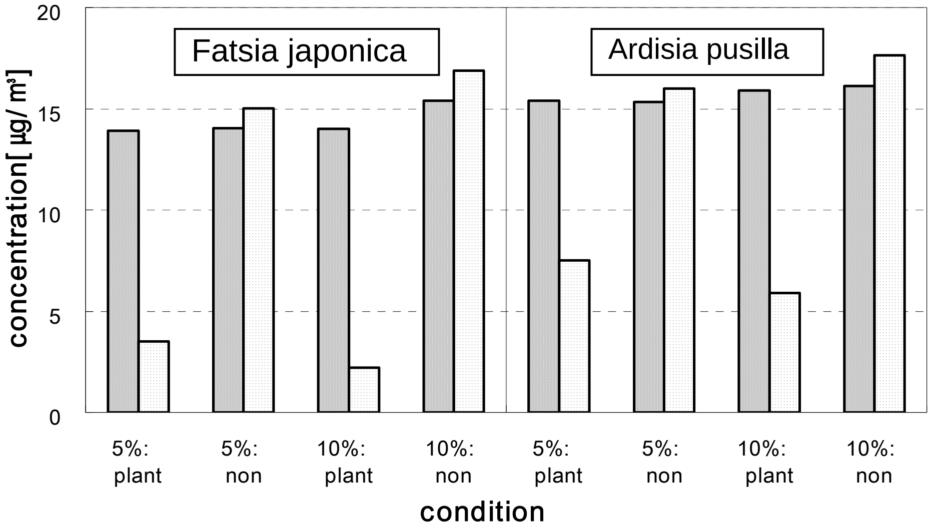

Figure 6. Variation of Xylene concentration according to the amount of plant

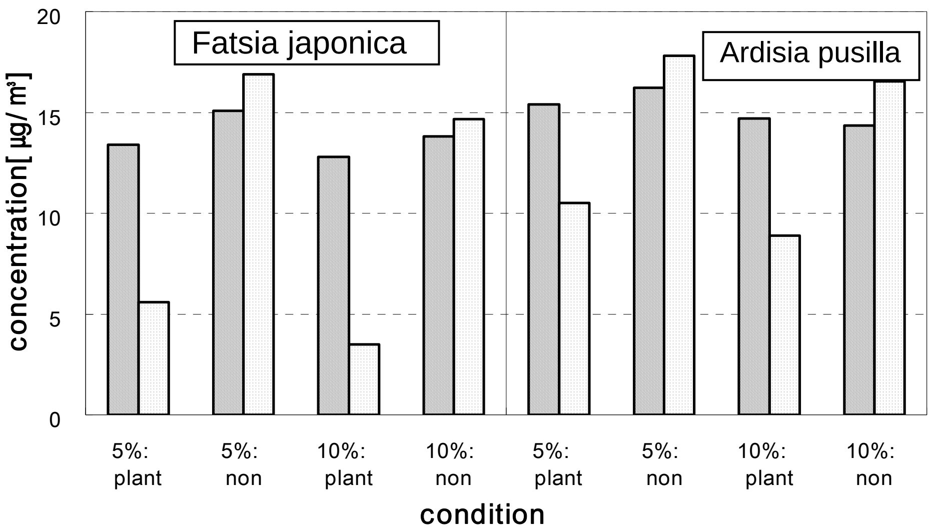

Figure 7. Variation of Stylene concentration according to the amount of plant

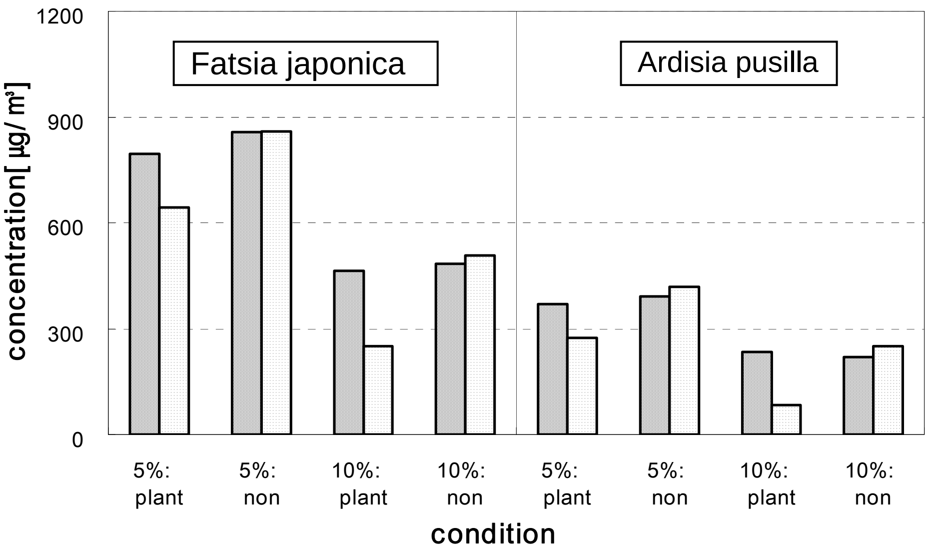

Figure 8. Variation of Formaldehyde concentration according to the amount of plant

3.2 Assessment to Planting and Growing Placement

1) Fatsia japonica Decne. et Planch

In case of planting Fatsia japonica Decne. et Planch, placing it at sunny spot was more effective than that at scattered planting. Benzene was reduced by 6.20 μg/m³ in case of placing at sunny spot, Toluene was reduced by 12.90 μg/m³, which was a bit less than Ardisia pusilla DC. Ethylbenzene was reduced to 1.80~3.40 μg/m³, and Xylene was reduced by 10.40 μg/m³, and Stylene was reduced by 6.00~7.80 μg/m³. Formaldehyde was more effectively reduced in Ardisia pusilla DC and 150.83 μg/m³ was largely reduced in sunny spot placement, and 107.92 μg/m³ was reduced in scattered placement.

2) Ardisia pusilla DC

In planting Ardisia pusilla DC, placing it at sunny spot was more excellent. Benzene was indicated by 9.00 μg/m³ at sunny spot placement, by 5.60 μg/m³ at scattered place, Toluene was indicated by 15.00 μg/m³ at sunny spot placement, by 5.40 μg/m³ at scattered place. Ethylbenzene was indicated by 5.60 μg/m³ at sunny spot place, by 3.90 μg/m³ at scattered placement, and Xylene was indicated by 7.90 μg/m³ at sunny spot, by 4.50 μg/m³ at scattered placement. Ardisia pusilla DC was more excellent than Fatsia japonica Decne. et Planch in Benzene, Toluene, Ethylbenzene. Formaldehyde was more effective in sunny spot placement than scattered placement, where the former was indicated to be 96.25 μg/m³, and the latter was 86.25 μg/m³.

Figure 9 indicates the concentration change of Benzene in case of Fatsia japonica Decne. et Planch and Ardisia pusilla DC planted, Figure 10 does the concentration change of Toluene, Figure 11 does the concentration change of Ethylene Benzene, Figure 12 does the concentration change of Xylene, Figure 13 does the concentration change of Stylene, and Figure 14 does the concentration change of Formaldehyde, which is the comparative graph of the set-up case and not set-up case of all plants.

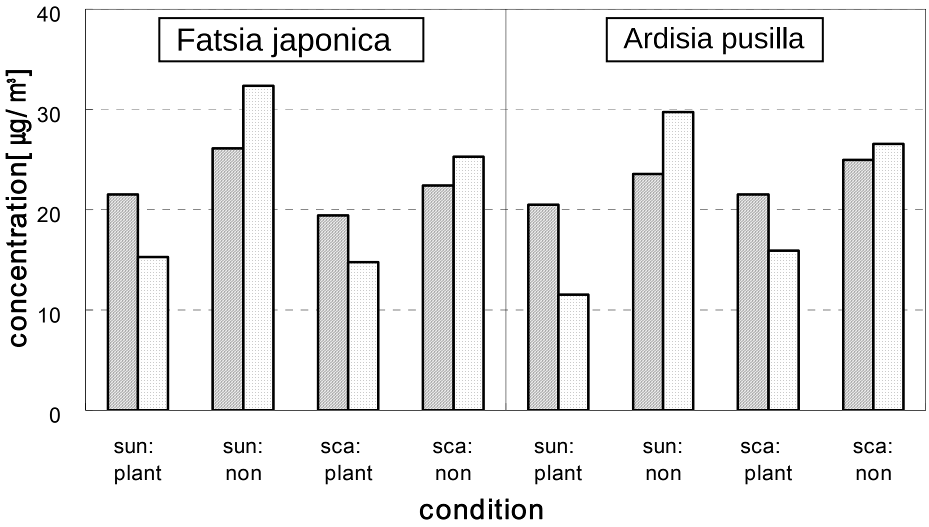

Figure 9. Variation of Benzene according to the positioning of plant

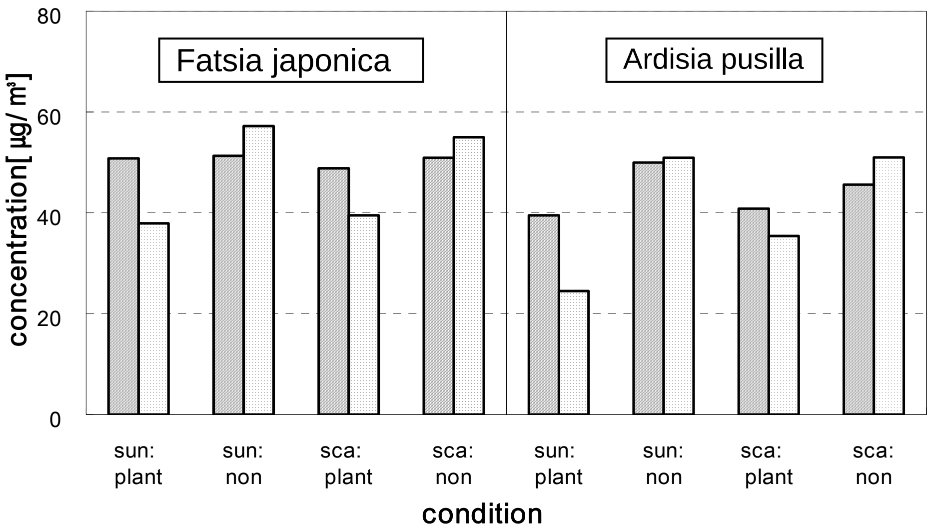

Figure 10. Variation of Toluene according to the positioning of plant

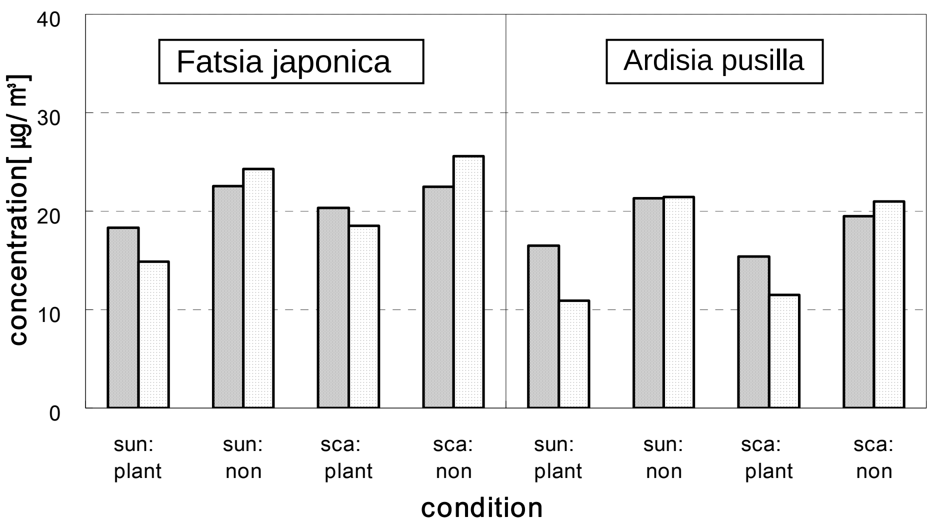

Figure 11. Variation of Ethylbezene according to the positioning of plant

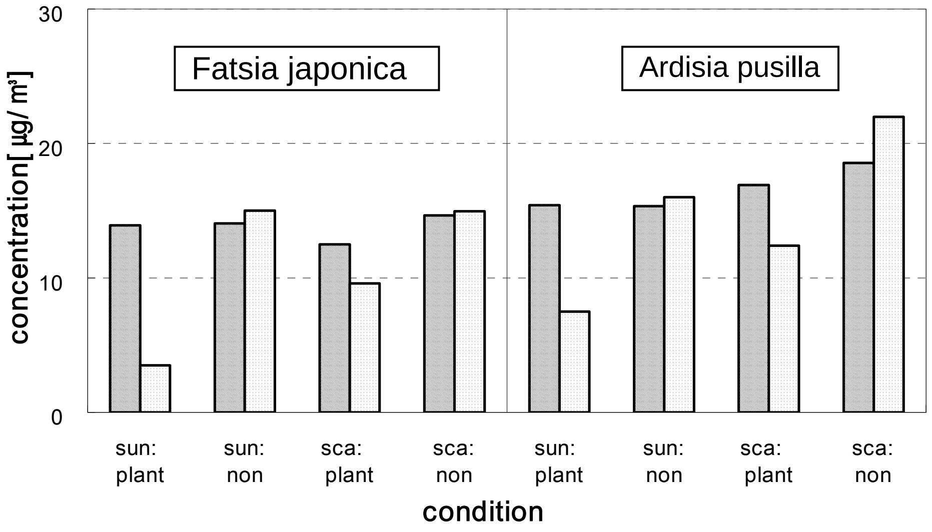

Figure 12. Variation of Xylene according to the positioning of plant

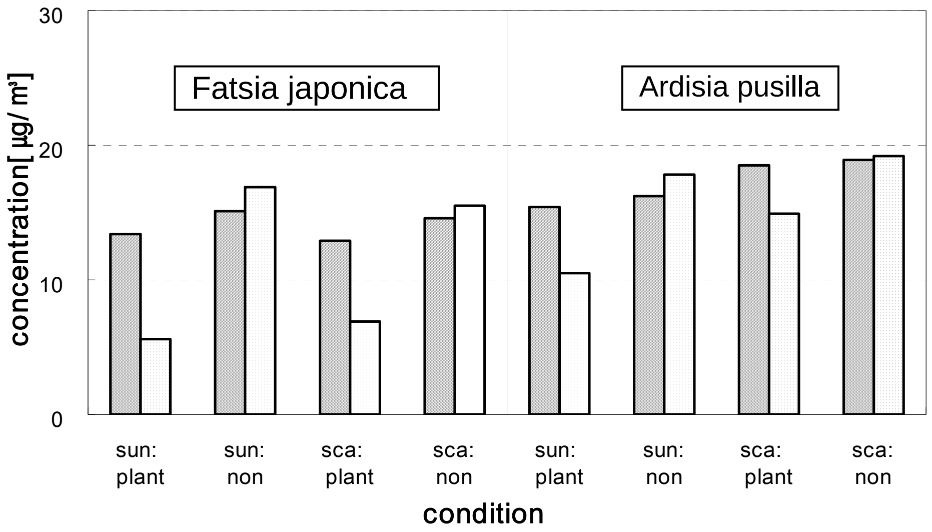

Figure 13. Variation of Stylene according to the positioning of plant

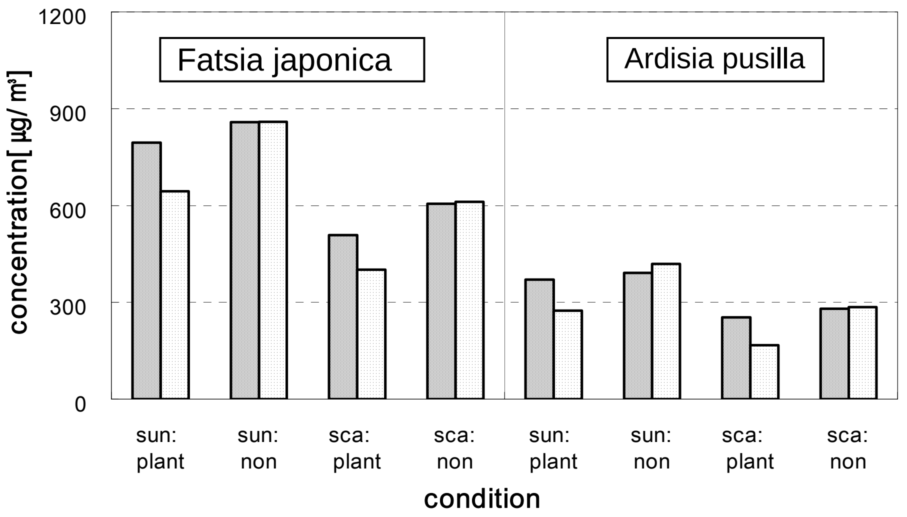

Figure 14. Variation of Formaldehyde according to the positioning of plant

DISCUSSION

This explored the effect of planting and growing volume and placement of Korean native plants, Fatsia japonica Decne. et Planch and Ardisia pusilla DC. whose findings are as follows.

(1) In all experiments, the room with plants indicated higher reduction effect of VOCs compared to that without plants.
(2) In all cases of experiment of planting and growing volume, the more planting volume, the more excellent the effect.
Toluene was more effective in Ardisia pusilla DC planted, Formaldehyde was more effective in Fatsia japonica Decne. et Planch planted respectively.
(3) In planting and growing and placing experiment, the placement at sunny spot was more effective than that at scattered growing.
When Fatsia japonica Decne. et Planch was placed at sunny spot, the reduction effect of Formaldehyde was the most excellent, and when Ardisia pusilla DC was placed at sunny spot, the reduction effect of Toluene was the most effective.
(4) In all experiments, Toluene was effective in Ardisia pusilla DC, and Formaldehyde was effective in Fatsia japonica Decne. et Planch.

ACKNOWLEDGEMENT

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A06
Residential ventilation

Energy Impact of Residential Ventilation Norms in the United States

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SUMMARY

The first and only national norm for residential ventilation in the United States is Standard 62.2-2004 published by the American Society of Heating, Refrigerating and Air-conditioning Engineers (ASHRAE). This standard does not by itself have the force of regulation, but is being considered for adoption by various jurisdictions within the U.S. as well as by various voluntary programs. The adoption of 62.2 would require mechanical ventilation systems to be installed in virtually all new homes, but allows for a wide variety of design solutions. These solutions, however, may have a different energy costs and non-energy benefits. This report uses a detailed simulation model to evaluate the energy impacts of currently popular and proposed mechanical ventilation approaches that are 62.2 compliant for a variety of climates. These results separate the energy needed to ventilate from the energy needed to condition the ventilation air, from the energy needed to distribute and/or temper the ventilation air. The results show that exhaust systems are generally the most energy efficient method of meeting the proposed requirements. Balanced and supply systems have more ventilation resulting in greater energy and their associated distribution energy use can be significant.

INTRODUCTION

In the United States (U.S.) residential ventilation was not traditionally a major concern because policy makers believed that between operable windows and envelope leakage, people were getting enough outdoor air. Over the past three decades, houses have become much more energy efficient and a key contributor was from reductions in envelope leakage. At the same time, the types of materials used in furniture, appliances, and building materials in houses have changed resulting in more indoor pollutants. People have also become more environmentally conscious not only about the resources they were consuming but about the environment in which they lived.

Historically, people have ventilated buildings to provide source control for both combustion products and objectionable odors [1]. Currently, a wide range of ventilation technologies is available to provide ventilation in dwellings including both mechanical systems and sustainable technologies. Recent residential construction has created tighter, energy-saving building envelopes that create a potential for under-ventilation [2,3]. As a result, new homes often need mechanical ventilation systems to meet current ventilation standards. These standards and related factors have been reviewed in previous studies [4].

ASHRAE Standard 62.2-2004 [5] is the U.S. standard for ventilation for acceptable indoor air quality in low rise residential buildings that (together with its companion Standard 62.1 for all other buildings) represents the current standard for setting ventilation rates. It is the objective of this study to evaluate the energy impacts of meeting this standard in U.S. houses.

ASHRAE Standard 62.2 has requirements for whole-house ventilation, local exhaust ventilation, source control and system requirements. The whole-house mechanical ventilation requirement of 62.2 is given by Equation 1:

$$Q\left(L/s\right) = 0.05 A_{floor}\left(m^2\right) + 3.5\left(N+1\right), \qquad \textbf{(1)}$$

It is assumed that an additional 0.1 L/s/m^2 is supplied through infiltration. Standard 62.2 also requires local mechanical exhaust in kitchens and bathrooms. Kitchens must have the capacity to exhaust at least 50 L/s through a range hood or provide 5 kitchen air changes per hour. Bathrooms must have the capacity to exhaust 25 L/s intermittently or have 10 L/s of exhaust continuously.

METHOD

In order to evaluate the energy impacts of the ASHRAE 62.2 requirements, we used a simulation tool to perform minute-by-minute ventilation, heat and moisture calculations that allowed for the dynamic performance of buildings and Heating, Ventilating and Air Conditioning (HVAC) components. Details of the simulation tool can be found in [6] and a summary of model validation can be found in [7].

Climates Evaluated

Six locations were used that cover the major climate zones:

- **Hot humid**: This zone represents many tropical and semi-tropical areas and is represented by weather data for the U.S. city of Houston,
- **Arid**: This zone represents hot, dry (desert) conditions and is represented by weather data for the U.S. city of Phoenix,
- **Warm humid**: This zone may be typical of some warm temperate zones such as in southern Europe and is represented by weather data for the U.S. city of Charlotte,
- **Continental**: This zone is unmoderated by large bodies of water and has both heating and cooling loads; it is represented by weather data for the U.S. city of Kansas City,
- **Marine**: This zone is typically a cool zone with long, but mild winters; it is represented by weather data for the U.S. city of Seattle,
- **Cold**: This zone had severe winters and short or mild summers typical of Nordic climates; it is represented by weather data for the U.S. city of Minneapolis.

Houses Evaluated

The houses were chosen to be typical of new US homes and complaint with all relevant norms. Different sizes (from 93 m^2 2-bedroom to 372 m^2 5-bedroom) and air tightness levels were used, leading to mechanical ventilation rates from 15 to 40 L/s. Further details of the houses can be found in [8]. For brevity only results for a medium sized house of 186 m^2.

Heating and Cooling Equipment

Equipment sizing was based on ACCA Manuals J and S [10, 11]. Equipment sizing was important in this study because several ventilation systems used the furnace blower to distribute ventilation air. Further details of the equipment can be found in [7 and 8].

Systems Evaluated

We considered nine different ventilation systems derived from previous reviews [11] and rated performance. Fan energy use and HRV/ERV efficiencies were taken from the Home Ventilating Institute directory [12]. The following ventilation systems were simulated:

Standard House (no whole-house mechanical ventilation)

This was the base case for comparison to the other ventilation methods and was simulated for all six climates and three house sizes. This was the same house as the mechanically ventilated cases, except it had no whole-house mechanical ventilation, only bathroom and kitchen source control exhaust. This house was ***not*** 62.2 compliant.

Leaky Envelope that meets 62.2 (no whole-house mechanical ventilation)

This represented existing homes and was simulated for all six climates and three house sizes. This house was 62.2 compliant because of its leaky envelope.

Continuous exhaust

Continuous exhaust was simulated using a bathroom exhaust fan.

Intermittent Exhaust

Intermittent exhaust was simulated using bathroom fans that were on for 20 hours and off for 4 hours during peak space conditioning load. The fan flow was increased from the continuous exhaust case to account for the intermittent operation using algorithms from 62.2.

Heat Recovery Ventilator (HRV) & Energy recovery Ventilator (ERV)

An HRV was used in the cold, marine, continental and arid climates. The hot humid and warm humid climates used ERVs. The HRV and ERVs were connected to the forced air duct system and the air handler fan operated at the same time to distribute the air. Listed recovery efficiencies were applied to the air flow through the unites when calculating the energy use.

Continuous Exhaust plus Air Distribution

The continuous exhaust system was augmented with a central fan integrated (CFI) supply that used the furnace blower to intentionally draw outdoor air through a duct into the return and distribute it throughout the house using the heating/cooling supply ducts. The outdoor air duct was only open to outdoors during heating and cooling blower operation and had a damper that closed when the blower was off. The flow through the outdoor air duct into the return was the same as the continuous exhaust fan flow.

Continuous Supply

The continuous supply system used a fan to supply filtered air from outside to the duct system that then distributed the air throughout the house without using the furnace blower. The supply fan was sized to be four times the ASHRAE Standard 62.2 outdoor air requirements to allow for a 3:1 mixing ratio of indoor air to outdoor air for tempering.

RESULTS

In order to focus on the impacts of the ventilation system and not the details of the house, we have elected to show the results relative to a base case. This approach subtracts out common factors, allowing the results to reflect the difference due to the ventilation treatment chosen. We have elected to use the *unvented house* as the base case, but it is important to remember that that case is a non-compliant one.

The figures are stacked bar charts where we have broken up the total energy use into a piece that represents the energy to induce the desired ventilation (Ventilation), the energy to distribute air (Distribution) and the energy to condition the air (Space Conditioning that combines heating and cooling). The energy use data was converted to dollars ($) using current local energy prices for the US cities chosen.

Figure 1 shows the results for the Hot humid climate where the total energy use for the standard house was 16,500 kWh (cost was $1125). The exhaust systems had the least energy increase of about 800 to 1000 kWh (5% of the total space conditioning) and the intermittent system saved 200 kWh ($18) relative to the continuous system. Both the ERV and Continuous Supply systems used significant distribution energy that made them the least energy efficient. The higher ventilation rates of the continuous supply and continuous exhaust plus intermittent supply lead to greater space conditioning energy use.

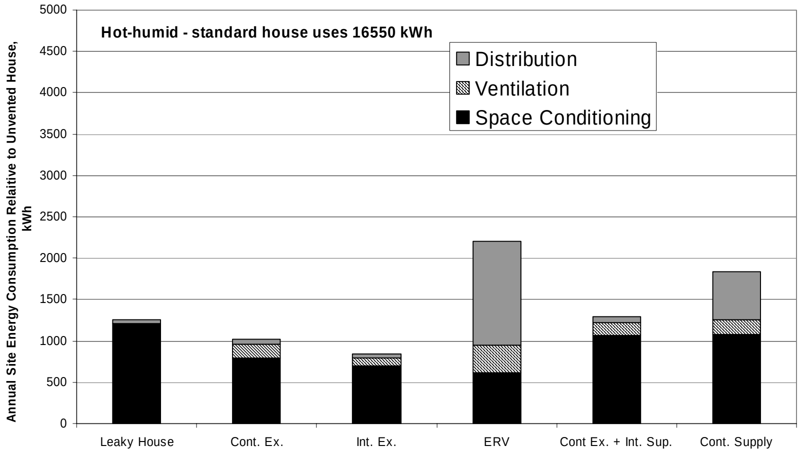

Figure 1: *Energy use of ASHRAE 62.2 compliant ventilation systems relative to standard unventilated house in Hot humid climate.*

Although the ERV had the greatest air change rates, the recovery of energy was shown by the having the least space conditioning energy use. However the distribution energy requirements and ventilation fan power made the ERV the greatest energy user (costing an additional $240 compared to the standard house). If the ERV had not used the central fan for distribution the energy use would be between the continuous and intermittent exhaust cases. In comparison to the leaky house, both the exhaust only systems actually used less energy despite providing greater effective ventilation. The leaky house suffers from having higher ventilation rates when indoor outdoor temperatures are greatest leading to excess energy use.

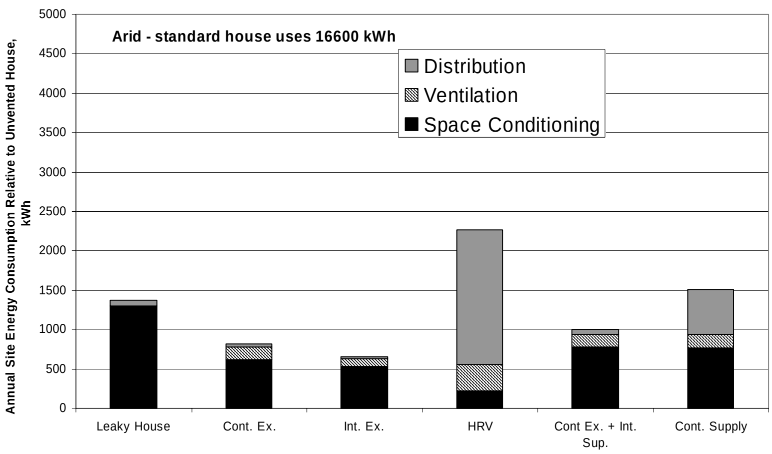

Figure 2: *Energy use of ASHRAE 62.2 compliant ventilation systems relative to standard unventilated house in an Arid climate.*

Figure 2 shows the results for the Arid climate where the total energy use for the standard house was 16,600 kWh (cost was $1170). Although this is considered a cooling dominated climate (with almost four times as much cooling operating time than heating), the energy used for heating (9600 kWh) is still greater than that used for cooling (5500 kWh). The Arid climate results had the same general trends as the Hot humid climate, but the milder climate meant that the space conditioning energy use related to ventilation is lowered. The exhaust system used 800 kWh ($60). The intermittent exhaust saved 160 kWh ($14) relative to the continuous system. With less space conditioning energy, the distribution energy use was even more dominant for the HRV and continuous supply than in the Hot humid climate and made them the least energy efficient. The higher ventilation rates of the continuous supply and continuous exhaust plus intermittent supply lead to greater space conditioning energy use.

Although the HRV has the greatest air change rates the recovery of energy is shown by the having the least space conditioning energy use. However the distribution energy requirements and ventilation fan power make the HRV the greatest energy user: requiring an additional 2250 kWh compared to the standard unvented house (costing an additional $240 compared to the standard house). If the HRV did not use the central fan for distribution the energy use would be between the continuous and intermittent exhaust cases. As with the Hot humid climate, the leaky house used more energy than either of the exhaust systems.

Figure 3 shows the results for the Warm humid climate where the total energy use for the standard house was 24,750 kWh (cost was $1410). For the Warm humid climate, the less benign climate led to the energy for space conditioning increasing significantly - in particular for the higher ventilation rate systems and the leaky house.

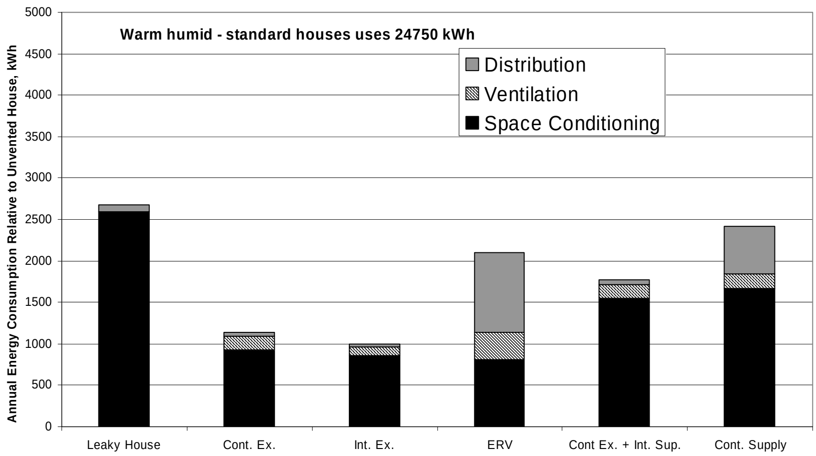

Figure 3: *Energy use of ASHRAE 62.2 compliant ventilation systems relative to standard unventilated house in a Warm humid climate.*

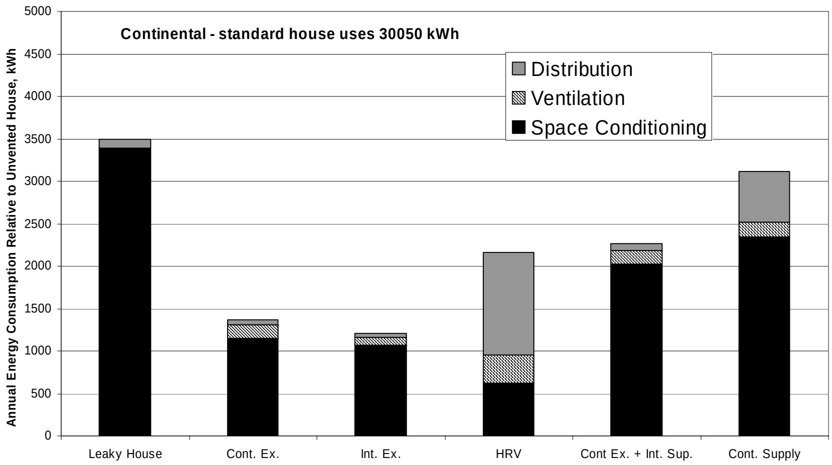

Figure 4: *Energy use of ASHRAE 62.2 compliant ventilation systems relative to standard unventilated house in a Continental climate.*

The exhaust system used 1150 kWh ($75) more than the unvented house. The intermittent exhaust saved only 45 kWh (less than $10) relative to the continuous system. The leaky house was the greatest energy user, followed by the continuous supply. The ERV had the lowest space conditioning energy use that made it more energy and cost competitive with the other systems.

The Continental climate is more heating dominated and cooling energy use is reduced to about 5% of the total. The colder climate is reflected in the space conditioning energy use that in increased relative to the distribution and ventilation fan components. The same trends are seen as for the Warm humid climate, with the leaky house requiring the most additional energy (3500 kWh ($190)) and the continuous exhaust system the least (1365 kWh ($85)).

The marine climate has very little air conditioning - compressor energy was less than 1% of the gas energy used. However, the milder winter climate compared to the Continental climate led to less total energy use and less difference between the unvented standard house and the ventilation systems.

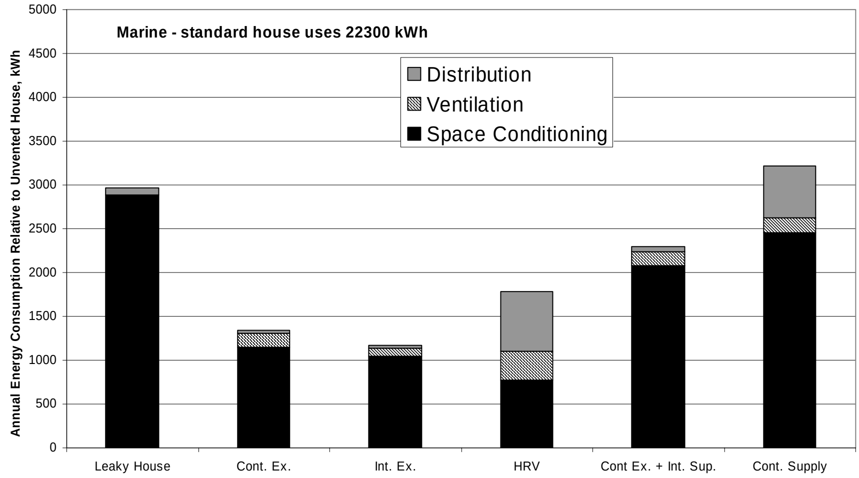

Figure 5: *Energy use of ASHRAE 62.2 compliant ventilation systems relative to standard unventilated house in a Marine climate.*

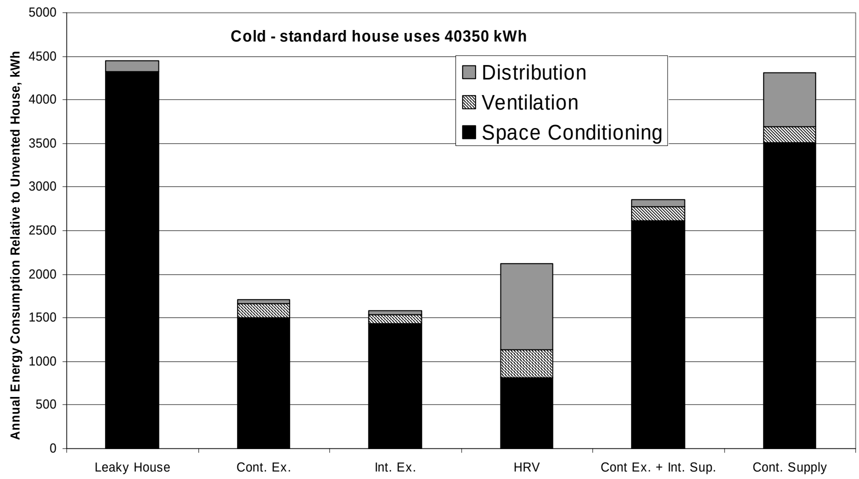

Figure 6: *Energy use of ASHRAE 62.2 compliant ventilation systems relative to standard unventilated house in a Cold climate*.

The distribution energy for the HRV and Continuous exhaust with intermittent supply were reduced due to the small capacity furnace and air conditioner required in the Marine climate with their smaller blower.

The Cold climate was the harshest climate investigated in this study and used the most energy. The low outdoor temperatures in winter make the leaky house a big energy user - requiring 4400 kWh ($240) more than the standard house. Although the space conditioning energy use for the HRV is about half that for the exhaust systems, the blower energy used to distribute the air is such that it requires 400 kWh ($60) more energy. As stated previously this could be reduced if the central furnace blower was not used to distribute the air, in which case the HRV would use the least energy. The higher air change rates of the continuous exhaust plus intermittent supply and the continuous supply lead to much more energy use than the exhaust only cases in this extreme climate.

Exhaust systems were found to be more energy efficient for several reasons: they had the lowest ventilation rates, they used the least ventilation fan power and they did not use fans to mix or distribute air. The most important reason was that requirements of 62.2 do not account for the differences in combining of infiltration with balanced and unbalanced mechanical ventilation.

Air distribution systems are used to distribute heating and cooling, to distribute fresh air, and to filter, clean and recirculate indoor air. As such they can be seen as a separate building function from the rest of the ventilation system and could be treated separately both in regulation and design. One advantage of treating distribution as a separate function is that it will allow efficiency advances in distribution to be made independently from, for example, cooling systems. Other studies [13, 14] have shown that using more efficient variable speed motors at low speed to circulate and mix air in a house results in significant energy savings relative to the blowers in most systems like the ones used in this study. Other options include using the HRV/ERV fans to also distribute the air in the house either through a dedicated duct systems or using the existing forced air system ducts.

CONCLUSION AND RECOMMENDATIONS

In this study we used a simulation approach to determine the likely energy impacts of residential ventilation requirements associated with ASHRAE Standard 62.2 and ventilation strategies for six different climates. The energy to provide such minimally acceptable ventilation was in the range of 800 to 1700 kWh per year ($50 to $100) for the most efficient options, which typically represented about 5% of the heating and cooling energy. The space conditioning energy associated with the extra ventilation provided by the ASHRAE 62.2 ventilation systems was much larger than the fan energy needed to supply or exhaust the air. In every climate the intermittent exhaust system proved to be the most energy efficient system for meeting the proposed requirements—followed by the continuous exhaust system.

Homes with leaky envelopes paid a significant energy penalty without necessarily providing more ventilation then a house with a better envelope and mechanical ventilation. Exhaust systems were found to be more energy efficient than the alternatives because the requirements of 62.2 do not correctly account for the addition of infiltration and mechanical ventilation. Since the exhaust system had a lower impact on the total air exchange, its space conditioning impact was smaller and it was the more energy-effective approach by virtue of producing fewer air changes.

Although the ventilation related energy requires changed by about a factor of three between mild and harsh climates, the general trends for relative energy use between systems was consistent.

Finally, systems that used the furnace or air conditioner blower to distribute air used substantially more energy than other systems. In particular for HRVs and ERVs, the use of the central blower counteracted the space conditioning energy savings of these devices such that they used more energy than a simple exhaust system. This energy penalty may be acceptable if mixing of air is considered desirable for thermal or health concerns.

ACKNOWLEDGEMENTS

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Room Airflow Rates in Finnish Houses

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SUMMARY

Airflow rates were measured in one hundred and two newly built single-family houses during 2002-2004. Of the measured houses, 10 % used natural ventilation, 28 % used mechanical exhaust and 61 % used mechanical supply and exhaust. Exhaust airflow rates were measured during the summer period from terminals. Supply airflow rates and sound pressure levels were measured in master bedrooms. During a 2-3 week period in winter, air change rates were measured with a passive tracer gas technique to determine the overall air change rate.

An average actual air change rate in the summer period was 0,40 1/h in houses with mechanical supply and exhaust ventilation and 0,37 1/h in houses with mechanical exhaust ventilation. Winter period values were respectively 0,41 1/h, 0,34 1/h and 0,3 1/h in naturally ventilated houses. All these values are below the code value of 0,5 1/h. Similarly, the average supply airflow rates in bedrooms were lower than the recommended value in the building code. Both exhaust and supply airflow rates will be reported.

Actual air change rates, room airflow rates together with sound pressure levels compared to airflow rates at the design speed of the ventilation units show possible improvements needed in the ventilation of houses.

INTRODUCTION

In Finnish houses, mechanical supply and exhaust ventilation with heat recovery has been the most common solution during the last 20 years. In apartment buildings, the centralized mechanical exhaust system has been the prevailing system since the 1970s [1]. The situation changed in 2003, when the national building code began to require at least 30 % heat recovery from the exhaust air (D2 2003). Due to this requirement, practically all new residential buildings are today equipped with mechanical supply and exhaust ventilation.

The building code requirement for the ventilation is 0.5 ach in general; the code also gives guideline values for outdoor and exhaust airflow rates for common rooms. These Finnish values follow quite well the second category specified in prEN 15251 [2]. The guideline value for an outdoor airflow rate is 6 l/s per person (4 l/s, per person before 2003) or 0.5 l/s,m^2 (0.35 l/s,m^2 before 2003) for the bedrooms and living rooms. Occupancy by two persons is a common design assumption for bedrooms, giving a 12 l/s design supply airflow rate.

In this study, airflow rates were measured in 102 single-family houses in Helsinki and Tampere area. Most of the measured houses were equipped with mechanical supply (28%) or mechanical supply and exhaust (61%) ventilation. Of the measured houses, 10 % were built with natural ventilation. Only a few houses were older than 5 years in this study. In this article, we concentrate on room airflow rates.

METHODS

The sample of 102 houses was selected according to ventilation system, so that natural ventilation and mechanical ventilation were represented in the study. As mechanical supply and exhaust ventilation has been a standard solution in Finland during last few decades, the sample included more natural and mechanical exhaust ventilation than there is in newly built houses (natural ventilation means here passive stack ventilation). As it was impossible to find newly built houses with natural ventilation, older houses (up to 24 years) were included in the study. Other houses were less than five years old. Half of the houses were located in Tampere and the other half in the Helsinki metropolitan area with a maximum distance of about 200 km between houses and a similar cold climate for all houses.

The houses were selected from databases of companies that manufacture and build houses. In the areas close to the universities in Helsinki and Tampere, announcements were published in local newspapers to get homeowners involved in the study. Also, newsletters were delivered to areas with newly build houses.

In the subgroups according to ventilation system the selection was random. The sample of 102 houses (from which 74 houses in PFT measurement) was according to ventilation system:

- 10% (11%) natural ventilation (passive stack ventilation)
- 29% (28%) mechanical exhaust
- 61% (61%) mechanical supply and exhaust.

The average size of the houses was 156 m^2 of heated area and 386 m^3 volume (medians) and the average age was 3 years (except those with natural ventilation). The average number of occupants were 3.6 and average building leakage value 3.9 ach at 50 Pa (the range was between 0.5 to 8.9 ach at 50 Pa). All the houses were one- or two-storey houses with wooden frame construction.

Dust from outside air can rise filters pressure drop and significalntly reduce airflows of the ventilation unit within a few months. To try to avoid that, tenants were asked to change filters before measurements were taken. In a few cases, measurements had to be repeated because the filters were not changed and there was no airflow to measure.

Exhaust airflow rates were measured in kitchens, bathrooms and saunas during the summer period from terminals. Exhaust airflow rates were measured in the minimum, maximum and normally used fan speed position. The sums of the exhaust airflows were used to calculate the air change rate of the houses. Supply airflow rates to bedrooms were measured with pressure difference from terminals. These air exchange rates were close to the air change rates that were measured in wintertime with tracer gas [3].

Occupants were asked to fill a questionnaire about their perception of indoor climate and the use of ventilation units. Actual use of ventilation units is analyzed based on air change rate and bedroom sound level measurements, as well as occupant complaints and comparison of actual ventilation unit fan speed, in order to design one.

RESULTS

Air exchange rates of the houses were measured from terminals at different fan speeds of the ventilation units. In Figure 1, the air change rates of mechanically ventilated houses can be seen. In the building code, the air change rate is 0,5 change per hour is recommended [4]. The measurements (Figure 1 b) shows that this recommended value can be attained in 90 % of the houses.

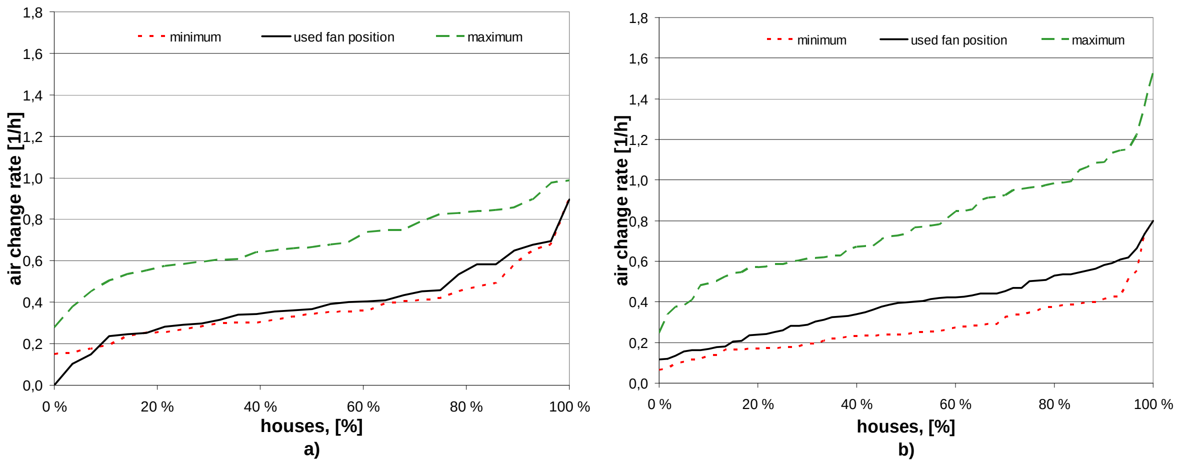


Figure 1 Air change rates in mechanically ventilated houses: a) mechanical exhaust (29) and b) mechanical supply and exhaust (61).

In reality, only 20 % of mechanically ventilated houses exceed the building code value in the normally used fan speed position. With the mechanical exhaust and supply ventilation unit, about 25 % of the measured houses had an air change rate of over 0,5 change per hour.

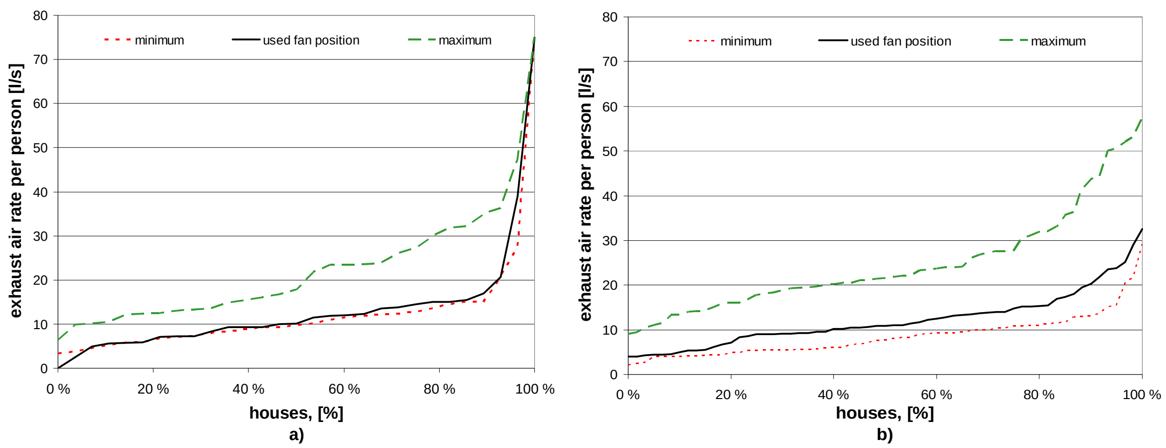


Figure 2 Exhaust air flow rates in litres per person. Overall exhaust air in mechanically ventilated houses a) mechanical exhaust, b) mechanical supply and exhaust.

In Figure 2, we have a graph of the exhaust air rate per occupant. We can see that the average in both systems was near 12 l/s.

Exhaust air rate in kitchens

Finnish building code (D2) recommends a range-hood exhaust air rate of 8 l/s, with a 25 l/s boost if fan speed can be controlled. Without the fan speed control, the airflow rate should be 20 l/s, with 25 l/s boost [4].

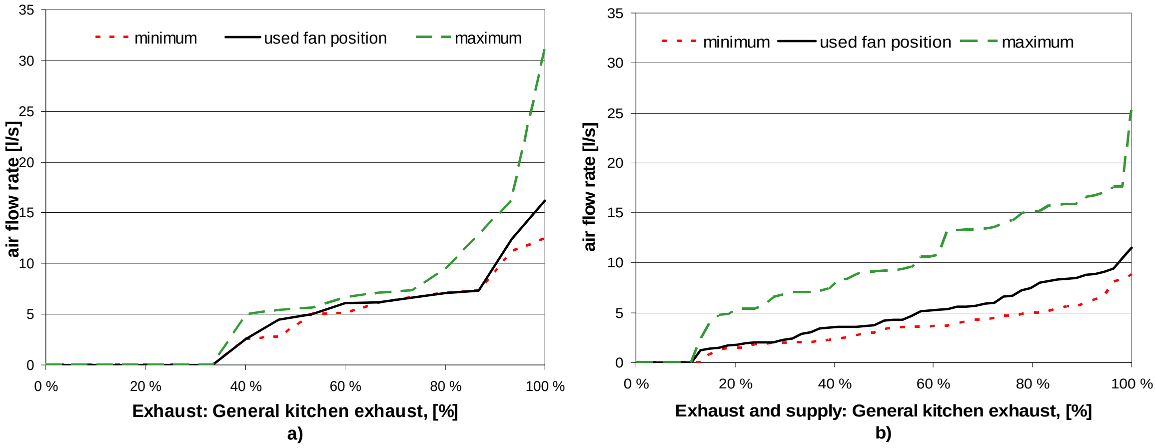


Figure 3 Exhaust airflow rate in kitchens (general ventilation). A) mechanical exhaust (29 kitchens) and b) mechanical supply and exhaust (61 kitchens). Airflow rates in minimum, maximum and normally used fan speed position.

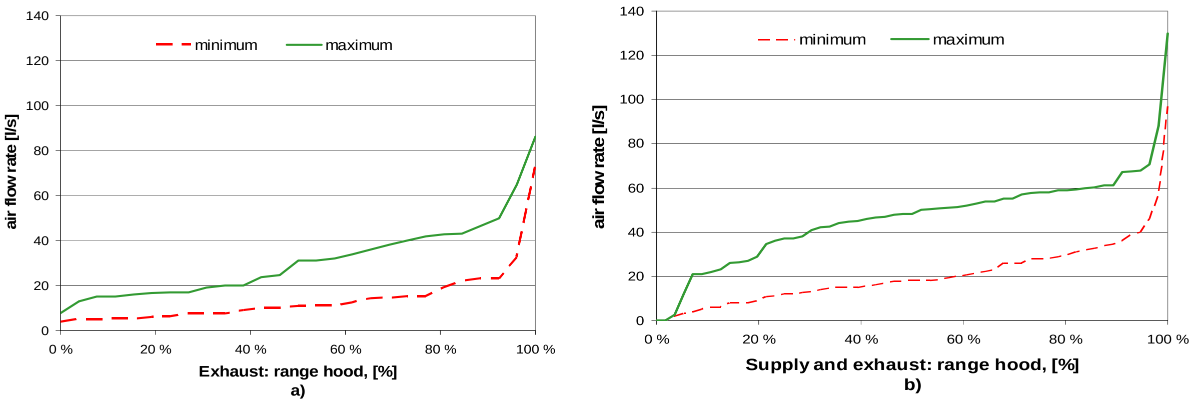


Figure 4 Exhaust airflow in kitchen range hood. A) Mechanical exhaust (27 hoods) and b) mechanical supply and exhaust (57 hoods).

Table 1 General kitchen exhaust and kitchen range hood airflow rates average, median and standard deviation. Mechanical exhaust, and mechanical supply and exhaust.

Mechanical exhaust 29 houses	min	Used fan position	max	Mechanical supply and exhaust 61 houses	min	Used fan position	max
General exhaust 16 kitchens				General exhaust 55 kitchens			
average [l/s] median standard deviation	4 4 4,1	5 5 4,8	7 6 8,2	average [l/s] median standard deviation	3 3 2,2	4 4 3,1	10 9 5,8
27 range hoods				57 range hoods			
average [l/s] median standard deviation	14 11 13,6		31 31 17,6	average [l/s] median standard deviation	21 18 15,7		47 48 21,0

In many houses, the measured airflow was under the 1 l/s (Fig. 3). In mechanical exhaust houses, the 50 l/s was possible to achieve in 10 % of kitchen-range hoods with boost, and in mechanical supply and exhaust houses, in 50 %.

Bathroom and sauna

The Finnish building code recommendation for bathroom exhaust is 10 l/s and for boosted exhaust 15 l/s. In mechanical exhaust ventilated houses, 65 % and in mechanical supply and exhaust ventilated houses, 70 % could have achieved the recommended value of 10 l/s at fan speed normally used.

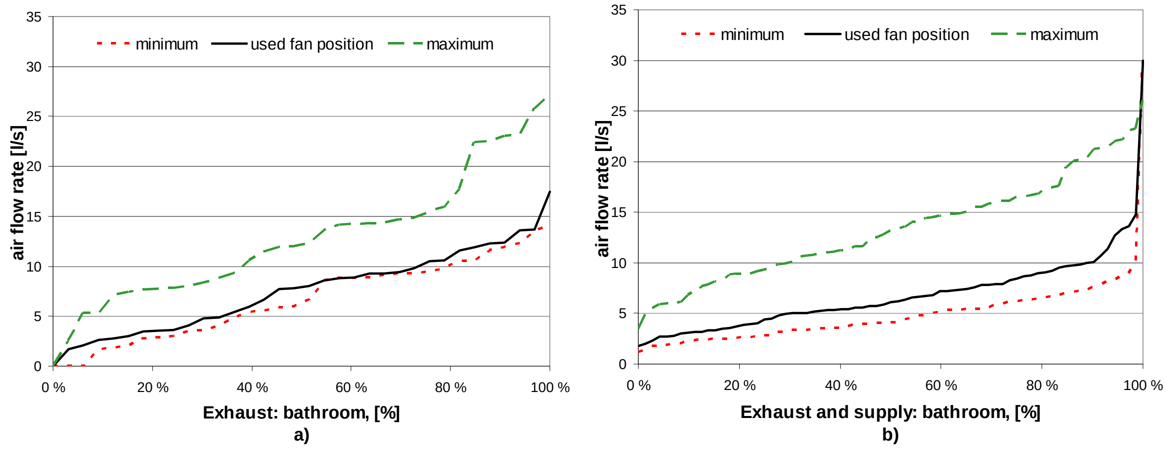


Figure 5 Bathroom exhaust airflow rate. A) mechanical exhaust (34 bathrooms), b) mechanical supply and exhaust (73 bathrooms).

In the sauna, the exhaust should be at least 6 l/s. That was achieved in 70 % of the measured houses using the normally used fan speed position.

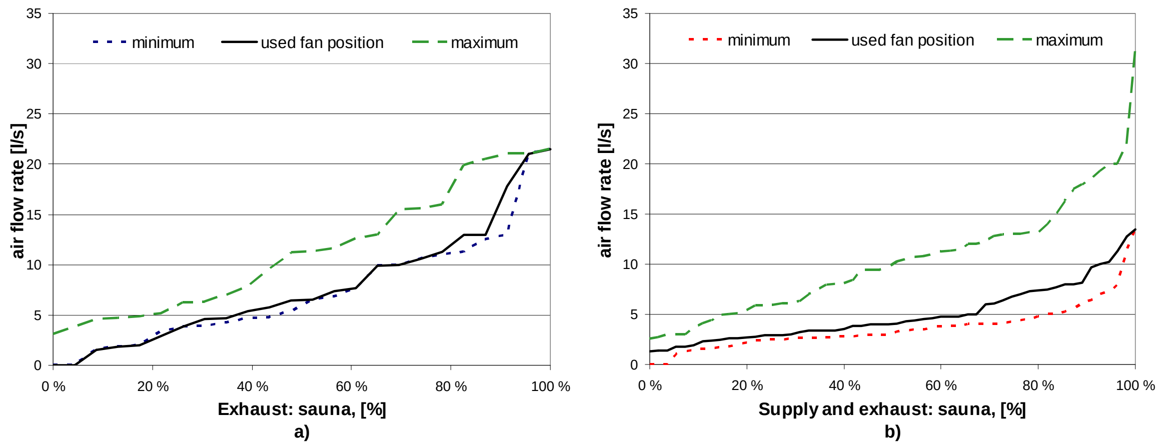


Figure 6 Exhaust air rates in sauna: a) mechanical exhaust (21 saunas), b) mechanical supply and exhaust (56 saunas).

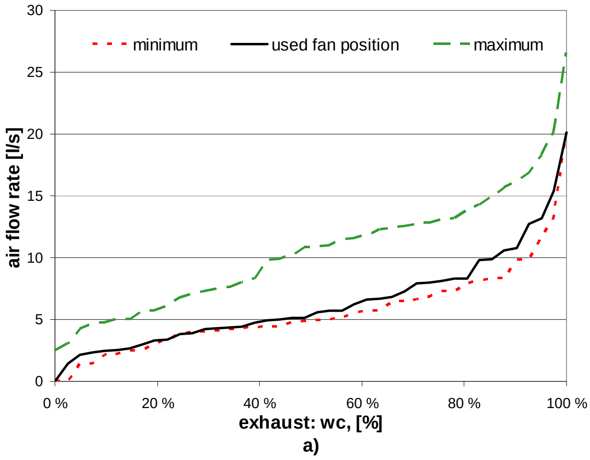
Table 2 Characteristics of mechanical exhaust and mechanical supply and exhaust: bathroom, average, median, standard deviation.

Mechanical exhaust 29 houses	min	Used fan position	max	Mechanical supply and exhaust 61 houses	min	Used fan position	max
34 bathroom				73 bathroom			
average [l/s]	7	8	13	average [l/s]	5	7	13
median	6	8	12	median	4	6	13
standard deviation	4,1	4,1	6,6	standard deviation	3,6	4,0	5,1

Table 3 Characteristics of mechanical exhaust and mechanical supply and exhaust: sauna average, median, standard deviation.

Mechanical exhaust 29 houses	min	Used fan position	max	Mechanical supply and exhaust 61 houses	min	Used fan position	max
24 saunas				56 saunas			
average [l/s]	7	8	11	average [l/s]	4	5	10
median	6	6	11	median	3	4	10
standard deviation	5,8	6,1	6,2	standard deviation	2,4	2,9	5,8

Exhaust airflow rates in toilets



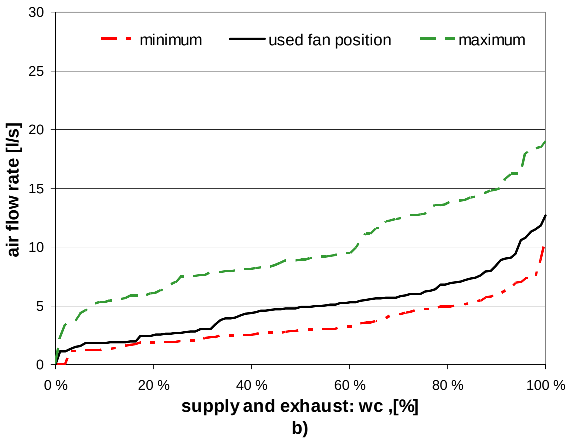


Figure 7 Exhaust airflow rates in toilets. A) Mechanical exhaust 42, normal use position average 6 l/s, median 5 l/s and standard deviation 4,0 at normally used fan speed position. B) Mechanical supply and exhaust 99, normal use position average 5 l/s, median 5 l/s and standard deviation 2,7 at normally used fan speed position.

The Finnish building code recommendation for a toilets exhaust air rate is 10 l/s. In mechanically ventilated houses, this is reached in 15 % of mechanical exhaust and in 5 % of houses with mechanical supply and exhaust ventilation at normally used fan speed position.

Bedrooms

In two-person bedrooms, both supply and exhaust were measured. Normally there is only supply airflow to bedrooms, but, in some cases, there were also exhaust terminal. There were nine houses (3 with mechanical exhaust and 6 with mechanical supply and exhaust) with an exhaust airflow rate of between 2 to 10 l/s.

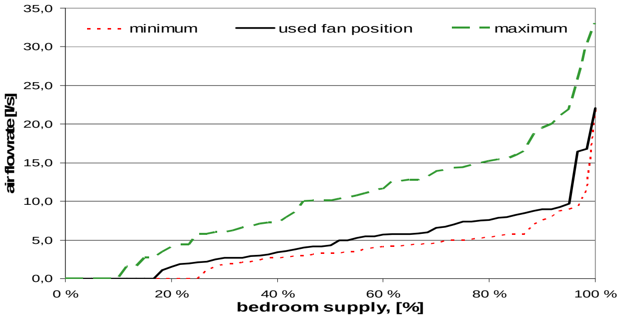


Figure 8 Supply airflow rates in bedrooms (61). Average 5 l/s, median 4 l/s and standard deviation 4,3 at normally used fan speed position.

The recommendation for minimum supply air rate in the Finnish building code is 6 l/s per person. The measured bedrooms were two-person bedrooms, so the minimum recommended supply air rate was 12 l/s.

Figure 8 (right) shows that, in 60 % of the houses, the recommended supply airflow rate could be achieved at some fan speed. It is possible to have a better air change rate for bedrooms with wardrobes (exhaust included). The measured exhaust air rates in wardrobes in Figure 9 shows that 27 % of the measured terminals have over 12 l/s exhaust in the operating range.

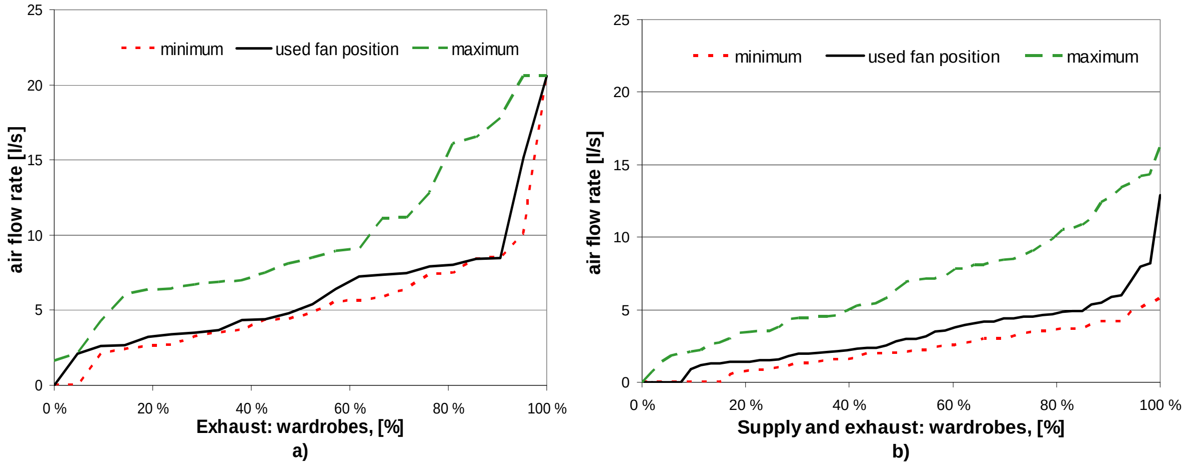


Figure 9 Wardrobe exhaust air rates. A) mechanical exhaust n= 22, normal use position average 6 l/s, median 6 l/s and standard deviation 4,5 at normally used fan speed position. B) mechanical supply and exhaust 54, normal use position average 3 l/s, median 3 l/s and standard deviation 2,4 at normally used fan speed position.

The building code recommended exhaust air rate is only 3 l/s. In mechanically ventilated houses, this value was achieved in exhaust in 77 % of the houses and in supply and exhaust in 48 %.

Sound levels in bedrooms

The sound level of the ventilation unit was also measured in the bedroom and living room. When interviewing the occupants during measurements, only a few of them mentioned the problems with noise from ventilation units.

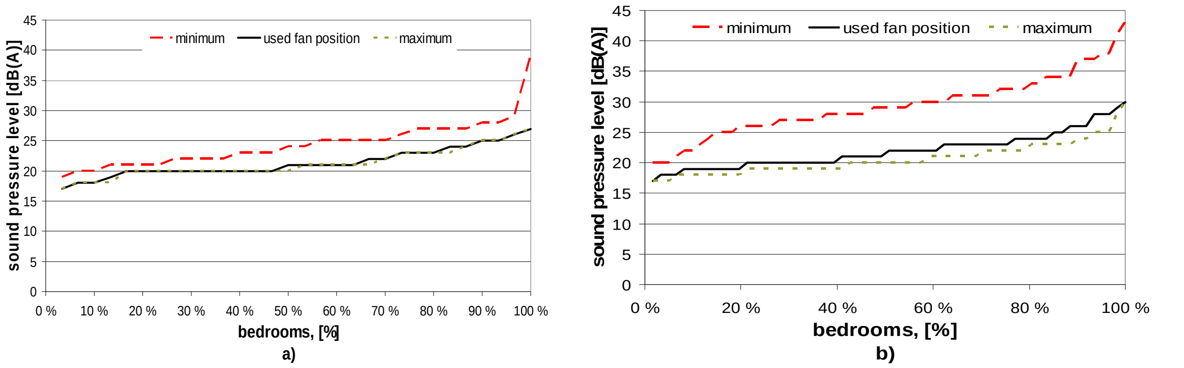


Figure 10 Bedroom sound pressure levels. A) mechanical exhaust, B) mechanical supply and exhaust.

From Figure 10, however, it can be seen that sound pressure levels were low compared to building code value 28 dB(A) [4]. This indicates that the sound pressure levels should be near 20 to 22 dB(A), which is acceptable at night.

DISCUSSION

At normally used fan speed position, room airflow rates were only half of the recommended values of the Finnish building code. Only in a few houses the airflow rate values were higher than 0.5 ach at normally used fan speed position. The recommended value was possible to reach in 90 % of the houses at some fan speed.

Measured exhaust air rates were slightly higher than 10 l/s per person in almost every house at normally used fan speed. In interviews, tenants did not complain about any odour problems, even if the airflow change rate was low.

In many cases, tenants did not have any knowledge about their ventilation system and its operation range. They did not know the best possible way to use it efficiently or how to maintain it properly.

It can be recommended that the design fan speed should be written on the ventilation unit and there should also be instructions how to change the filters and how often that should be done. It would also be useful to provide a guideline with ventilation unit given information about the general principle of the ventilation system, its operation and maintenance.

ACKNOWLEDGEMENT

Measured data analyzed in this study is from the national research project “Moisture-proof healthy detached house”.

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Impact of ventilation systems on energy and IAQ performance

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SUMMARY

The study aims at quantifying the impact of the retrofit of ventilation systems from the energy, IAQ (Indoor Air Quality) and health perspective
In a first part, the French market has been defined (number of each ventilation system installed in existing building). Both qualitative and quantitative studies of malfunctions noticed on ventilation systems have been made.
In a second hand, the influence of ventilation retrofitting on the building energy performance has been evaluated. The impact of implementing innovative techniques (i.e. decreasing the fan absorbed power…) has also been studied.
Although the study has been held on the French market, it gives an idea of the main trends and results that regulators may expect when acting on ventilation systems in retrofitting strategies.

INTRODUCTION

The aim of the study is first to quantify the impacts of the retrofit of ventilation systems from the energy, IAQ and health perspective then to propose systems upgrades in order to improve the installations.

METHODS

Experienced people from the field (building owners, designers…) analyzed the main causes for poor installation and function of ventilation systems. This has been compared and applied on the building stock in order to determine possible actions, constraints and potential savings when improving the current systems.

RESULTS FOR RESIDENTIAL BUILDINGS

Ventilation systems have been mainly modified in France by different regulations (i.e: whole-house ventilation with shunt in 1969, mechanical exhaust in 1982). There are therefore few retrofits of these systems. Needs for retrofitting ventilation systems while thermal retrofit is being done is a recent trends. It has started recently (after 1985) mainly in public housing because of the condensation problems.

Table 1 : Residential ventilation systems in France

Ventilation systems	**Multi-family house**	**Single-family house**
None	9%	14%
Local grills in wet rooms	34%	36%
Natural and Hybrid ventilation with ducts (whole-house)	17%	22%
Pressure controlled mechanical exhaust	35%	24%
Humidity controlled mechanical exhaust	5%	5%
Supply and exhaust	0%	1%
Type of ventilation		
None	9%	14%
Wet rooms only	34%	36%
Natural & Hybrid with ducts (whole-house)	18%	22%
Mechanical ventilation (whole-house)	40%	28%
All whole-house ventilation systems	57%	50%
Number of dwellings (existing stock)	13 066 266	17 333 069

The survey has investigated the main disorders encountered for each type of ventilation systems. Figure 1 shows overall results, given in frequency of cases encountered; the scale is not mentioned because the survey was only qualitative.

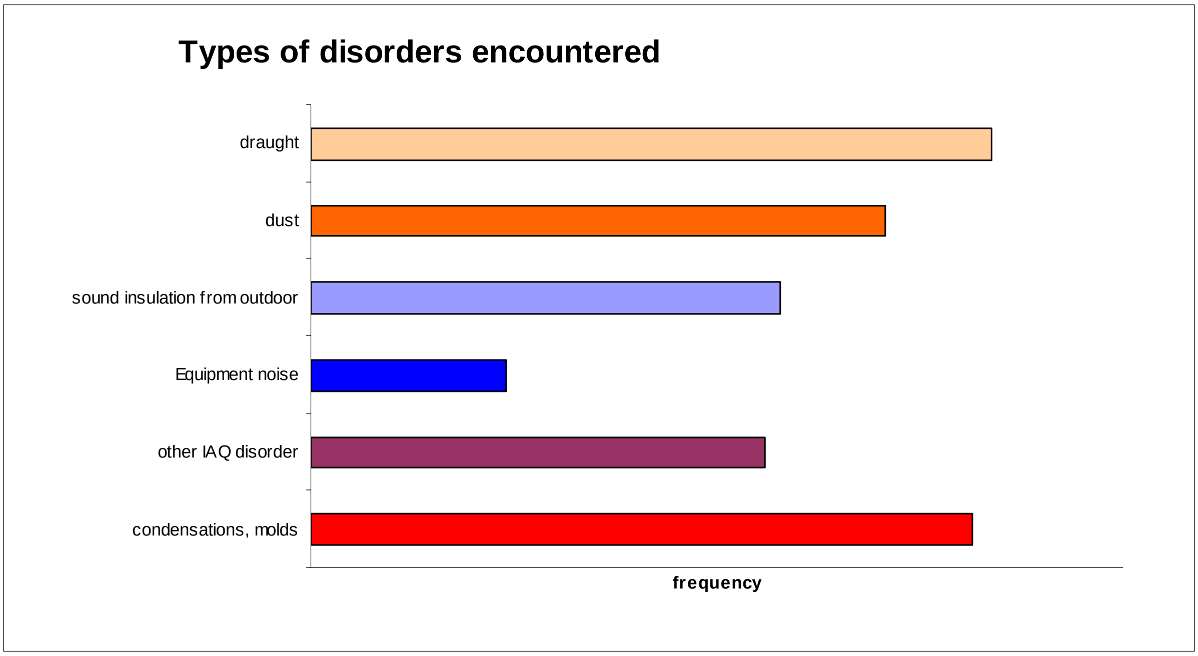

Figure 1 : Type of disorders encountered

More generally, we can note from the survey that:

- 47% of dwellings, mainly built before 1974, are not ventilated or just with some airing of the wet rooms. They are the first target of energy retrofits. Any reinforcement of the building airtightness (windows replacement…) without ventilation system retrofit leads to various disorders: humidity, condensation, moulds…
- 53% of dwellings, mainly from 1969 to the present day, have a whole-house ventilation either natural (20%) or mechanical (33%). Whole-house ventilation has

been mandatory in new buildings since 1969 in order to avoid condensations in bedrooms. The survey showed that these buildings have less air change disorders and better performance than the previous category. Nevertheless, their design and dimensioning as well as their installation are too often poor (duct tightness, fan power, acoustics…).

All disorders have an impact on :

- Durability of the building envelope (insurance companies report that condensation represents 50% of the causes of disorders and repairs cost in average 3300 € per dwelling),
- Occupants' health (as bibliography shows impact on symptoms of asthma, rhinitis…),
- Energy consumption. Uncontrolled air flows in dwellings are the first cause to waste energy, either by leaks, airing or windows opening. Adequate ventilation should provide a controlled ventilation air flow in order to avoid over-ventilation and supply only what is needed where it's needed. The study quantified the potential savings on houses without ventilation system.. .

  On existing stock, some choices of improvements of the ventilation systems have been made and estimation of the savings that would be obtained are shown in tables 2 and 3. Calculations take into account standard dimensioning as applied in France and emissions of CO_2 are calculated with 220 g/KWh as an average. The hypothesis is that all the existing stock would be improved (potential). The scenarios studied were:
  - when no ventilation system is available : switch to whole-house ventilation;
  - when a ventilation strategy is present : improvement of the control of the ventilation strategy (either by Demand Controlled Ventilation (DCV), heat recovery and/or low fan absorbed power).

Table 2 : Energy impact of ventilation improvement in single-family houses in France

	Existing system	**Number of dwellings (millions)**	**Improvement**	**Average savings on the total energy consumption KWh Primary Energy/ house/ year**	**Potential of savings kt eq CO_2**
Old house	none	2.40	whole-house ventilation	4968	2628
	wet rooms only	6.31	whole-house ventilation	4968	***6895***
	natural ventilation with ducts	3.76	mechanical ventilation	4968	**4110**
New house	mechanical ventilation	2.43	single exhaust, low fan consumption	324	173
		2.43	humidity controlled (DCV) balanced system with high recovery efficiency	972	519
	overall	**17.33**			**14326**

Table 3 : Energy impact of ventilation improvement in multi-family houses in France

	Existing system	**Number of dwellings (millions)**	**Improvement**	**Average saving on the total energy consumption KWh Primary Energy/ house/year**	**Potential of savings kt eq CO2**
Old multi-family houses	none	1.16	whole-house ventilation	2340	595
	wet rooms only	4.45	whole-house ventilation	2340	2291
	natural ventilation with ducts	5.23	humidity controlled natural ventilation or mechanical exhaust	2340	2690
Recent multi-family houses	mechanical ventilation	2.22	humidity controlled mechanical ventilation or balanced system	1885	921
		13.07			**6498**

In addition, the study has quantified the savings that could be obtained by a control and audit of installed mechanical ventilation systems (30% are claimed to be not conform to French regulation today). The savings are lower because they apply to a smaller building stock but show that a correct diagnostic of the ventilation systems could lead on the French market to:

- 0,225 Mt CO_2 on single-family houses stock,
- 0,2 Mt CO_2 on multi-family houses.

CEN TC 156 has prepared standards for the diagnostic of ventilation and air conditioning systems (EN 15239 and EN 15240) in the framework of the EPBD directive.

RESULTS FOR NON-RESIDENTIAL (COMMERCIAL) BUILDINGS

Table 4 : Types of ventilation systems in non residential buildings in France

	Offices	Commercial	Schools	Health	Leisure	Hotels	Overall
None or airing (windows)	50%	40%	60%		10%	5%	34%
Natural ventilation with ducts						9%	1%
Local (room) fans	10%	10%		15%	20%		10%
Mechanical exhaust	9%		20%	25%		75%	13%
Mechanical Exhaust with DCV	1%						0.2%
Supply & Exhaust	10%		19%	25%		10%	9%
AHU inc. ventilation	20%	50%	1%	35%	70%	1%	34%
OVERALL	100%	100%	100%	100%	100%	100%	100%
Built 1000 x m^2/year	3026	3732	1747	1700	1666	821	12692
Existing stock 1000 x m^2	176244	193030	167852	100711	67141	58748	763726

Two main conclusions have appeared :

1. A lot of buildings do not have ventilation systems except local exhaust from toilets and bathrooms. Airing by window opening is common although it is not always allowed by French hygiene regulation..

2. In most cases, ventilation air flow rates are not high enough in rooms with high density of occupation (meeting and conference rooms, spectacles…) due to poor dimensioning and sizing and/or change of occupation in time. This leads to IAQ problems although there is less humidity trouble in this kind of buildings than in dwellings. Too high equipment noise is also common and attributed to poor commissioning of the installations.

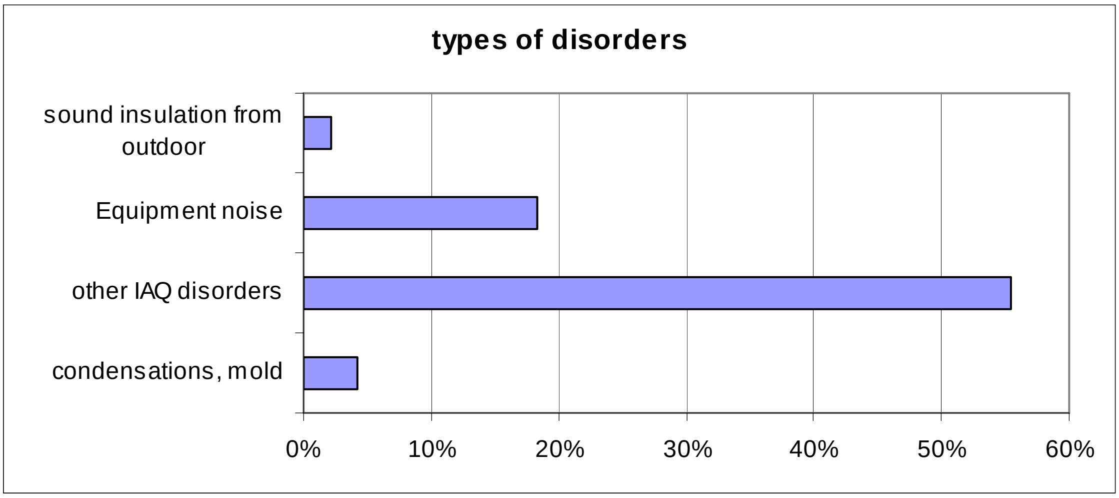

Figure 2 : Main disorders encountered for all types of non residential buildings.

The study has also quantified the potential of energy savings that could be gained from improving the ventilation systems. Where no ventilation was available, the scenario was to install one. When a ventilation system was present, the scenario was to improve the ventilation strategy (timer, heat recovery unit (HRU), DCV...). Results are shown for office buildings stock in table 5.

Table 5 : Energy impacts of improvements in non residential buildings in France

	Existing system	**Number of buildings (millions)**	**Improvement**	**Average saving on the total energy consumption KWh Primary Energy/ m²/year**	**Potential of savings kt eq CO2**
All periods	None	106.2	Mechanical exhaust with timer	32	748
	Mechanical exhaust, no timer	8.0	Mechanical exhaust with timer	32	56
	Mechanical exhaust, with timer	8.0	DCV	7	12
	Balanced AHU without HRU	22.1	Balanced system with timer	52	253
		22.1	Balanced system with HRU or DCV	10	49
	Balanced AHU with HRU	8.9	Balanced systemwith HRU and DCV	7	14
		177			**1131**

GUIDE FOR IMPROVEMENTS

As quantification of potential savings has shown a real impact, it is necessary to give to professionals some guidance and information about improving ventilation systems when retrofitting.
In order to help diagnostic and improvements, the study listed for main ventilation systems and main buildings the different options and proposed some charts to help decision. As an example, figure 3 shows a decision process for offices.

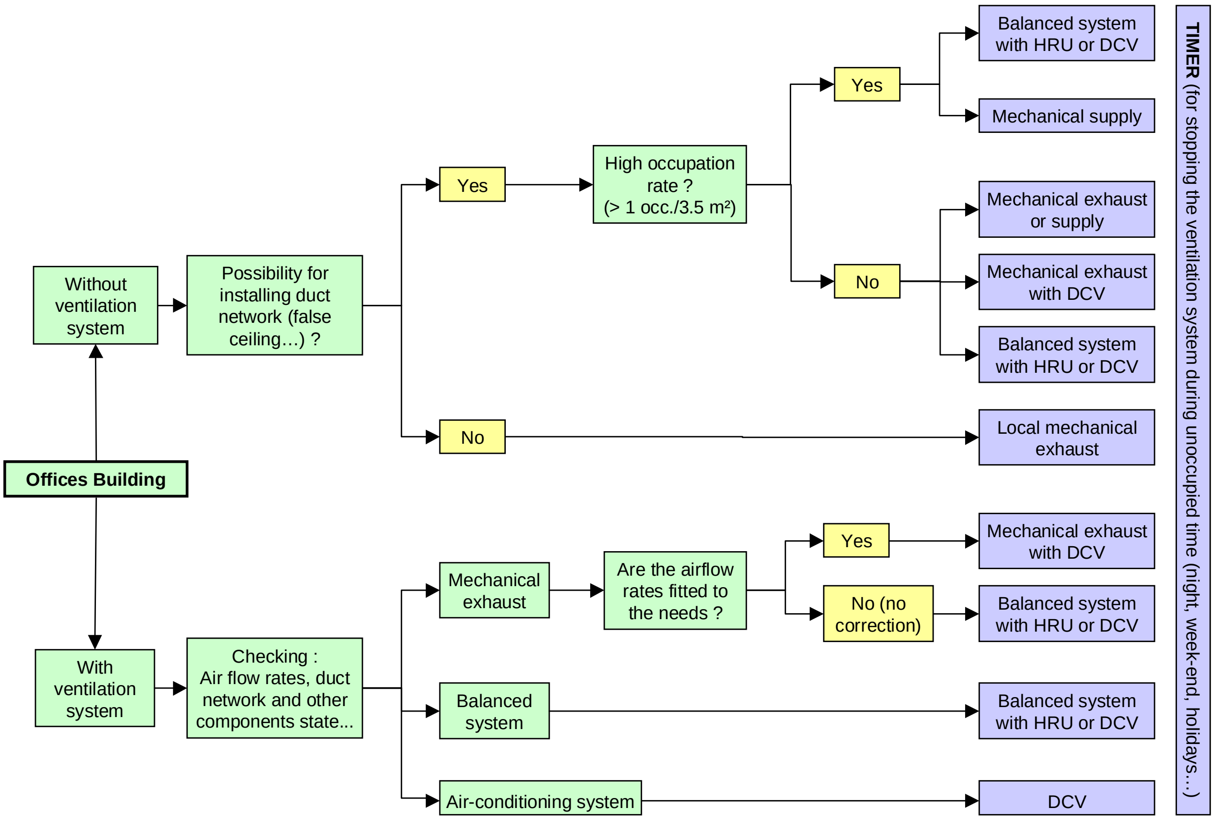

Figure 3 : Decision process for improving ventilation systems in offices

DISCUSSION

All European countries, due to the implementation of the EPBD directive, are studying how to reduce energy consumption on the existing building stock. Ventilation systems are essential when retrofitting:

- To avoid condensations, molds or IAQ disorders too common when the building tightness is reinforced. These disorders have impacts on health, comfort and performance.
- To save energy by the control of air flows in the building. Uncontrolled flows, leakages and airing are too expensive from the point of view of energy. A first step when retrofitting should be to install a ventilation system when none exists. A second step is to check regularly if the performance of the system is correct in order to maintain in time the energy performance. A third one is to improve the ventilation by heat recovery, DCV and/or low fan absorbed power.

The impact of ventilation on the building energy consumption is high enough to be dealt correctly. Poor installations and designs are a cost from the energy point of view.

ACKNOWLEDGEMENT

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Building leakage, infiltration and energy performance analyses for Finnish detached houses

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SUMMARY

This study focuses on the correlation between the airtightness of a building envelope and the average infiltration and energy consumption of a typical modern Finnish detached house. The correlation between tightness and infiltration was determined using an empirically validated dynamic IDA-ICE simulation model of a two-storey detached house. The effect of wind conditions, Finnish climate conditions, balance of ventilation system and leakage distribution on infiltration were studied with the simulation model. According to the results, the average infiltration rate and heat energy consumption increase almost linearly with the building leakage rate n_{50} of the building envelope. Resulting from the linear dependence, the annual average infiltration rate of the detached house in a sheltered suburban area can be approximated with the simple equation $n_{50}/31$. If the average infiltration rate is applied to heat energy calculations, the simple equation is $n_{50}/25$.

INTRODUCTION

This study is part of the cooperation project *Tightness, indoor air and energy efficiency of residential buildings* (AISE), that is being carried out by two cooperating Finnish universities: Tampere University of Technology and Helsinki University of Technology during the period 2005 – 2007. The objective of this study is to find a correlation between the tightness of a building envelope and the average infiltration and energy consumption of a typical modern Finnish detached house in the cold climate of Finland, and to study the effect of other important factors, such as leakage distribution, wind and climate conditions, on infiltration, using an empirically validated simulation model.

Since the late seventies, studies have been conducted on the correlation between airtightness of a building envelope measured with a single pressurization test and annual infiltration rate. In 1982, Kronvall and Persily compared pressurization test results to infiltration rates measured with tracer-gas in detached and terraced houses in Sweden and USA (New Jersey). From their comparison, they obtained the widely used “rule of thumb” for annual infiltration rate: $n_{50}/20$ [1,2], where n_{50} is leakage air change rate per hour at 50 Pa of pressure difference. In 1988, Dubrul found that the infiltration rate can be estimated with n_{50}/K. According to the study of Dubrul, the value of the coefficient K ranges from 10 to 30, depending on, for example, the type of building, wind conditions and leakage distribution. However, the mean value 20 can be regarded as typical [3].

METHODS

Building description

This study was carried out as a sensitivity analysis simulating a single building model in various conditions. The building model describes an existing detached house comprising two floors (Figure 1). The building is situated in the metropolitan area of Helsinki; it was built in 2000. The net floor area of the building is 172 m² and the building structures are of wood-frame construction; the base floor of the house is a concrete slab on the ground. The level of thermal insulation of the house fulfils the requirements of the Finnish building code [4]; the house is equipped with a mechanical supply and exhaust ventilation system with heat recovery. The building corresponds to the typical timber-framed detached house defined by the national project "Moisture-proof healthy detached house" [5].

Figure 1 The object of the study is a detached house.

The building was studied with extensive field measurements, including, for example, a pressurization test of the building envelope and an analysis of the leakage distribution, using infrared photography. The measured leakage air change rate at 50 Pa pressure difference over the envelope (n_{50}) was 3.9 ach. The building was simulated with the measured leakage distribution and two approximated distributions shown in Table 1 [6]. In the "draughty roof" case, most of the leakage openings were at the junction of the roof and external wall, while in the "draughty base floor" case, most of the leakage openings were at the junction of the base floor.

Table 1 The vertical leakage distributions of the building. The measured distribution is based on the infrared photography of the building; the two other distributions are approximated cases.

Place of the leakage routes		Vertical leakage distribution, %		
		Measured	Draughty roof	Draughty base floor
2nd floor	Junction of roof	36	75	12.5
	Upper edge of window frame	4	0	0
	Lower edge of window frame	4	0	0
	Junction of intermediate floor	2	0	0
1st floor	Junction of intermediate floor	21	12.5	12.5
	Upper edge of window	0	0	0
	Lower edge of window frame	24	0	0
	Junction of base floor	10	12.5	75

The dynamic simulation model

This sensitivity analysis was carried out using a simulation model of the preceding detached house. The pressure conditions of this building model were compared against measurement results and the model was found to be suitable for infiltration and energy analyses [6]. The building model was done using IDA indoor Climate and Energy 3.0 (IDA-ICE) building simulation software. This software allows the modelling of a multi-zone building, HVAC-systems, internal and solar loads, outdoor climate, etc. and provides simultaneous dynamic simulation of heat transfer and air flows. It is a suitable tool for the simulation of thermal comfort, infiltration, and energy consumption in complex buildings. A modular simulation application, IDA simulation environment and IDA-ICE, has originally been developed by the Division of Building Services Engineering, KTH, and the Swedish Institute of Applied Mathematics, ITM [7,8]. IDA-ICE has been tested against measurements [9,10] and several independent inter-model comparisons have been made [11]. In the comparisons, the performance of radiant heating and cooling systems using five simulation programs (CLIM2000, DOE, ESP-r, IDA-ICE and TRNSYS) were compared; IDA-ICE showed a good agreement with the other programs.

RESULTS

The effect various factors on infiltration and heat energy consumption of Finnish detached house were simulated. Most of the simulations were carried out with the hourly weather data of Helsinki (1979), which is commonly used as test-reference data for energy calculations in Finland [12]. The temperature dependence of infiltration was studied in two separate cases using the weather data of 1979 from Jyväskylä and Sodankylä. The Finnish climate is cold and the annual average outdoor temperatures of 1979 was 4.3°C in Helsinki, 2.8°C in Jyväskylä and -0.8°C in Sodankylä. The measured annual average wind velocity of these places was between 3 and 4m/s, so the wind conditions were very similar.

Studied cases

The studied factors of the sensitivity analysis and the description of the simulated cases are listed in Table 2. All the cases, except the simulations of different Finnish climate conditions, were simulated with three levels of airtightness. Almost completely airtight detached houses were described with a building leakage rate of n_{50} = 0.15 ach, while the airtightness of typical timber-framed detached houses was simulated using n_{50} = 3.9 ach [5]; the leakage air change rate n_{50} = 10 ach describes leaky detached houses. The effect of wind pressure on infiltration was studied with three levels of wind shielding: an exposed area, a sheltered suburban area and a theoretical case without wind at all. Exposed wind conditions are quite rare in Finland because the country is mostly forested and suburban areas are typically closely built. The sheltered wind conditions describe the typical wind conditions of Finnish suburban areas; the effect of stack-induced infiltration alone is studied in the theoretical case of no wind effect. The effect of the distribution of leakage openings were simulated using the measured and the two estimated distributions shown in Table 1. The effect of the balance of the mechanical ventilation system was studied with equal supply and return air flow rates or with supply air flow rates 15% greater or less than the return air flow rates. The air change rate of the building was 0.56 ach in all the simulation cases fulfilling the minimum requirement (0.5 ach) of the Finnish building code [13].

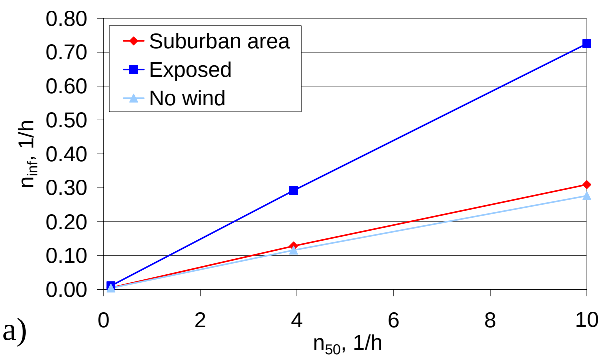
Table 2 The description of the simulated cases.

Focus of the simulation	Studied factors				
	Wind conditions	Leakage distribution	Supply/ return	Climate	Airtightness n_{50}, ach
Wind conditions	Exposed	Measured	1	Helsinki	0.15-3.9-10
	Suburban	— ‖ —	— ‖ —	— ‖ —	0.15-3.9-10
	No wind	— ‖ —	— ‖ —	— ‖ —	0.15-3.9-10
Leakage distribution	Suburban	Draughty roof	— ‖ —	— ‖ —	0.15-3.9-10
	— ‖ —	Measured	— ‖ —	— ‖ —	0.15-3.9-10
	— ‖ —	Draughty base floor	— ‖ —	— ‖ —	0.15-3.9-10
Ventilation performance	— ‖ —	Measured	0.85	— ‖ —	0.15-3.9-10
	— ‖ —	— ‖ —	1	— ‖ —	0.15-3.9-10
	— ‖ —	— ‖ —	1.15	— ‖ —	0.15-3.9-10
Finnish Climate conditions	— ‖ —	— ‖ —	1	Helsinki	3.9
	— ‖ —	— ‖ —	— ‖ —	Jyväskylä	3.9
	— ‖ —	— ‖ —	— ‖ —	Sodankylä	3.9

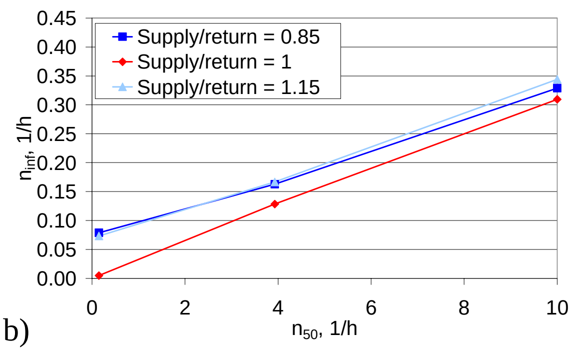
Infiltration

The effect of wind on the average infiltration air change rate is about 10% in the sheltered suburban area (see Figure 2a) of Helsinki. Wind-induced infiltration is more significant than stack-induced infiltration only in the exposed wind conditions, where wind effect is about 60% of the average infiltration air change rate. The effect of the balance of the ventilation system on pressure conditions of the building and infiltration is emphasized in the airtight building (see Figure 2b). If the airtightness of the building is poor, the balance of the ventilation system is not important.

The difference in infiltration between the different leakage distributions is shown in Figure 2c. The average infiltration air change rate is about 10-25% higher in the building with the measured leakage distribution than in the building with the estimated distributions. The effect of the Finnish climate conditions on infiltration is shown for the detached house with a typical level of building leakage rate of n_{50} = 3.9 ach (see Figure 2d). Those cases were simulated with the wind conditions of the suburban area. The average infiltration air change rate slightly increases with decreasing annual average outdoor temperature, but the difference is insignificant (2%) between Helsinki and Jyväskylä. The difference in infiltration between the climate conditions of Helsinki and Sodankylä is 16% according to the simulation results.



a)



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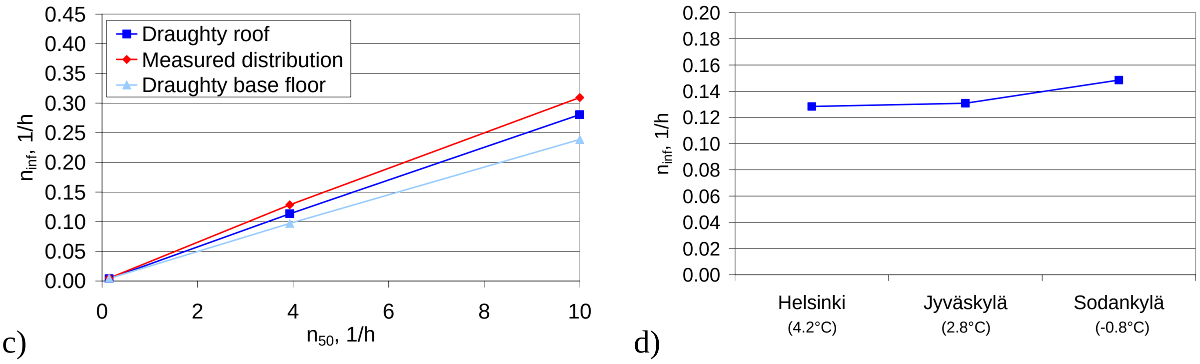

Figure 2 The effect of several factors on the annual infiltration air change rate. Figures show the effect of wind conditions of the building site (a), the balance of the ventilation system (b), the leakage distribution (c), and the Finnish climate conditions (d). (The annual average temperatures are shown in the brackets.)

The results show that the correlation between airtightness and average infiltration air change rate is almost linear when the ventilation supply and return air flow rates are in balance. Then, the annual average infiltration rate n_{inf} can be approximated by dividing the building leakage rate n_{50} by a constant parameter x

$$n_{inf} = \frac{n_{50}}{x}. \qquad (1)$$

If the average annual infiltration air change rate is used in the heat energy calculation, the dependence of infiltration air flows on the temperature-driven pressure difference between indoor and outdoor air should be taken into account in the cold climate. During the heating season in Finland, the infiltration air flow increases with the increasing temperature difference between indoor and outdoor air (see Figure 3).

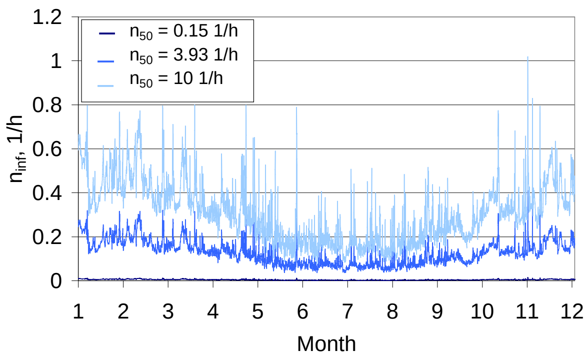

Figure 3 Infiltration air change of the detached house with three different building leakage rates in Helsinki.

The fluctuation of the infiltration air change between the heating season and summer season can be taken into account by, for example, calculating the annual infiltration air change weighted by the temperature difference of the indoor and outdoor air

$$n^{e}_{inf} = \sum_{i=1}^{8760} \frac{(T_{in} - T^{i}_{out})_i \cdot n^{i}_{inf}}{(T_{in} - T^{i}_{out})_i}, \qquad (2)$$

where n^e_{inf} is the average annual infiltration air change rate suitable for heat energy calculation and n^i_{inf} is a normal hourly infiltration air change and T_{in} is a set point temperature of heating, T^i_{out} is hourly outdoor air temperature. If the weighted infiltration air change rate is calculated using the form of Equation (1), the temperature correction could be performed using a correction factor k as follows

$$n^e_{inf} = \frac{n_{50}}{k \cdot x}. \qquad (3)$$

The resultant parameters of the most important simulation cases for the simple calculation methods of Equations (1) and (3) are shown in Table 3. A suitable value of parameter x for the detached house is about 31 in suburban areas in Helsinki. Then, the average infiltration air change can be approximated by dividing the building leakage rate n_{50} by 31. In that case, the correction factor k for the heat energy calculation is about 0.8 and the weighted infiltration air change rate is 20% higher than the mean value of the infiltration air change rate. Substituting the preceding values of parameter x and the correction coefficient k into Equation (3), the simple calculation method for the weighted infiltration air change rate reduces into the form $n_{50}/25$. The determined values of parameters x and k are accurate enough for the climate conditions of Jyväskylä also. In the exposed wind conditions, the average infiltration air change is about two times bigger than in the shielded suburban area and the mean value of parameter x is about 14. If the supply and return air flow rates of the mechanical supply and exhaust ventilation system are unbalanced, the average infiltration rate cannot be approximated using the simple linear calculation method of Equations (1) and (3), especially when the envelope is extremely airtight, because the infiltration air flow rate depends on the difference between supply and return air flow rates.

Table 3 Parameters of the most important simulation cases for the simple calculation methods shown by Equations (1) and (3).

Building leakage rate n_{50}, ach	$x = n_{50}/n_{inf}$	Correction factor for heat energy calculation k
Suburban area, measured leakage distribution, supply/return air flow rate = 1		
0.15	31	0.8
3.9	31	0.8
10	32	0.8
Exposed area, measured leakage distribution, supply/return air flow rate = 1		
0.15	14	0.9
3.9	13	0.9
10	14	0.9
Suburban area, measured leakage distribution, supply/return air flow rate = 0.85		
0.15	2	1
3.9	24	0.9
10	30	0.9

Energy consumption

The effect of wind conditions and the balance of the ventilation system on heat energy consumption are shown in Figure 4. The resultant energy consumption covers the heat energy consumption of the building zones and the ventilation system. The correlation between the tightness of the building envelope and the heat energy consumption is almost linear in simulated wind conditions (see Figure 4a). Because of that, the preceding correlation reduces into a simple rule of thumb: The increase of one unit of the building leakage rate n_{50} gives rise to an increase of about 6% in the heat energy consumption of the zones and the ventilation system. At the same time, the increase in total heat energy consumption is about 4% consisting heat demand of domestic hot water and household electricity. The preceding simple rule is valid for the studied detached house in a sheltered suburban area in the climate conditions of Helsinki. The effect of the balance of the ventilation system on the heat energy consumption of the building is slightly greater in the airtight building (see Figure 4b).

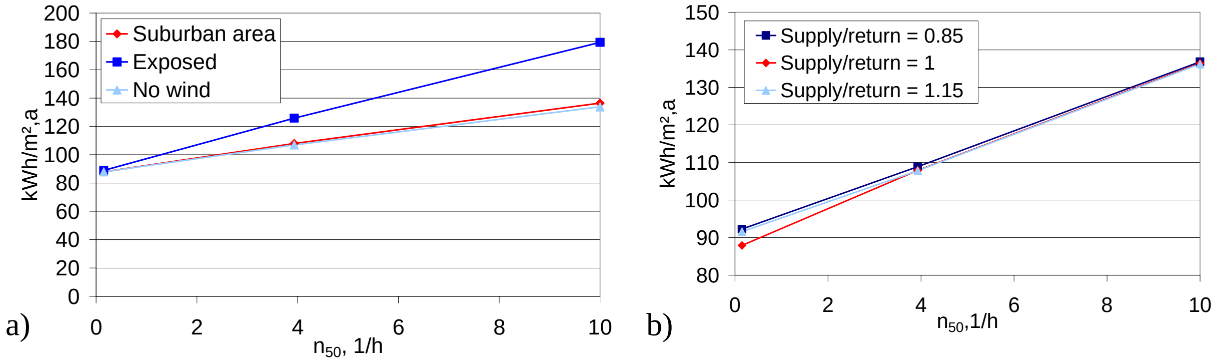

Figure 4 The effect of wind conditions (a) and the balance of the ventilation system (b) on heat energy consumption of the zones and the ventilation system.

DISCUSSION

According to the simulation results, wind has a minor effect on the average infiltration in typical suburban areas of Helsinki and the stack-induced infiltration is dominant in the Finnish cold climate. A correlation between the airtightness of the building envelope and the annual infiltration air change rate is almost linear. In this case, the annual infiltration air change rate can be approximated with the simple equation $n_{50}/31$. The average infiltration air change rate for the heat energy calculations can be calculated by weighting the average infiltration air change rate with the heat demand of a building. The correction for the heat energy calculation can be performed by multiplying the denominator of the preceding equation by the correction factor, which is 0.8 in a typical suburban area. Then the weighted annual average infiltration air change rate can be approximated with the simple equation $n_{50}/25$. These simple approximations are valid in the climate conditions of Helsinki and Jyväskylä, respectively located in Southern and Central Finland. The resultant increase in heat energy consumption regarding zones and the air handling unit is 6% on average, when the value of the building leakage rate n_{50} increases by one unit. Respectively, the increase in total heat energy consumption is about 4%.

ACKNOWLEDGEMENT

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Influence of occupants on the energy use of balanced ventilation

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SUMMARY

In this paper we give an overview of the ways occupants use ventilation systems and describe the results of interviews conducted in households equipped with balanced ventilation. An attempt is made to quantify the effects of occupant behaviour on the final energy use of the household for heating. This energy use is studied for several behaviour scenarios, leading to the conclusion that occupant behaviour may easily reduce the predicted savings to zero, or even may increase the energy use when compared to natural ventilation.

INTRODUCTION

In the Netherlands the energy use of new buildings is subject to legislation and should not exceed a certain value, calculated by using the Energy Performance Coefficient (EPC, [17]). The lower the EPC, the more energy efficient the building. Ever since its introduction in 1996, the EPC-value has become stricter. This has led to an increase in the implementation of heat-recovery balanced ventilation systems in newly built dwellings. In 1996-1997, when the EPC-value was 1.4, almost all ventilation systems applied used a natural air inlet and a mechanical air outlet. In 2000-2001, when the EPC-value was 1.0, 40 percent of all new residential buildings were equipped with heat recovery balanced ventilation systems [1]. The recent lowering of the EPC to 0,8 will probably enhance this percentage even further, because balanced ventilation is one of the most cost-effective ways to achieve the prescribed EPC-value. However, occupant behaviour may have a considerable influence on the yearly energy efficiency of the balanced ventilation systems in dwellings. This energy efficiency depends among others on the flow-rate settings chosen by the occupants and on the way they use natural ventilation openings. This paper aims at a first evaluation of the energy efficiency of heat-recovery balanced ventilation systems when actual use is taken into account.

METHODS

Occupant behaviour

First, a literature study was conducted. There are indications that occupant behaviour often plays a considerable role [2,3,4] in the effectiveness of sustainable building options that have a user component. When occupants use certain techniques in a way that differs from the intentions of the designer, the minimisation of environmental impacts might be counteracted. A post-occupancy evaluation (PoE) of mechanically ventilated heat recovery systems showed that the real CO_2-production was higher than the calculated optimum and also higher than the base case with only natural ventilation [4]. Jeeninga found in [3] that the actual energy use of households living in dwellings with the same EPC-value can vary up to a factor two. Four main factors play a role on ventilation behaviour [5, 6]. These factors are occupant characteristics, characteristics of the dwellings and of its environment and finally the characteristics of the ventilation system itself.

Occupant's characteristics: the size and the composition of the households has an influence on the ventilation behaviour [5,7,8], an particularly the presence of children and the age of occupants. The older the occupants, the less ventilation. The presence of occupants is related to the duration of window-opening [9] as well as household activities like cooking, tumble drying, washing clothes or dishes, showering and smoking [5]. Thermal comfort preferences of occupants play also a role. People who have a preference for high temperature open the windows less [5]. The preferred temperature in the bedrooms is lower than in the living room [5,10], which is one of the reasons why windows are opened more often in bedrooms than in living rooms. Ventilation behaviour is strongly correlated to perceived fresh/stuffy air, dry/humid air, cooking odours and other strong odours ([6], [11]). Knowledge about the ventilation system is another important actor. Occupants may not use the system on the right way, because of inadequate operation instructions [8]. From [8] it appears that there is no difference in the frequency and duration of opening of the windows between occupants using different mechanical and natural ventilation systems. However, a difference is noticed in [10] where it appeared that occupants with mechanical ventilations systems tend to use the windows less. The differences between both studies may arise from differences in knowledge or in poorly working systems. Pulmonary diseases may also play a role in the ventilation behaviour. Occupants with lung diseases ventilate longer and more often [12]. Finally some studies show that less conscious occupants and occupants paying a collective energy bill may open the windows longer ([5],[10]). *Characteristics of the dwelling:* The type and design of dwellings influences the opening of windows [9]. In ground-bounded dwellings windows are less often open in the living room and the kitchen than in apartments. For bedrooms, the inverse is observed. In apartments, the windows are more often ajar and in ground-bounded dwellings wide open. Thermal insulation, air tightness and noise insulation of the dwelling may also influence the use of the windows and of the mechanical ventilation. Occupants tend to ventilate less through mechanical system or windows when the air tightness of the dwelling is low (high infiltration rates) or when the rooms are small ([5],[6]). South orientated rooms tend to be more ventilated than other rooms when the sun shines [5]. The use of the rooms itself may influence the ventilation behaviour. Also the location of furniture or the use of decoration stuffs or plants on the windowsill are of importance [13]. Finally, the type of heating system plays a role. In dwellings with central heating windows are less often open than in other dwellings [9]. *Characteristics of the environment:* The weather plays a main role in opening or not the window. The use of windows is linearly correlated with the outside temperature for temperatures between -10 $^{\circ}$C and 25 $^{\circ}$C and inversed correlated with wind velocities.

When it is raining or snowing, windows are less often used. Next to weather, noise nuisance is an important factor ([9], [13]). In about 30 % of Dutch dwellings, occupants close the windows to avoid noise from outside. Odours from industry, traffic or agriculture produce the same effect [10]. Finally, occupants close their windows when they leave the house or at night as protection against burglary [5]. *Characteristics of the ventilation system:* The ventilation system consists in infiltration trough cracks, ventilation through windows and ventilation through a mechanical system if present. The way the window hangs up in the window frame is important as well as the direction of opening. Windows that are fixed on the bottom of the frame and that open inwards are more often open than other types of windows [9]. Upper windows are open twice more often than windows opening outwards. If the window in open stand cannot be fixed at several positions through a grip, it is possible that the window will never be used [10]. If a mechanical ventilation system is used, the location of the switches may be of importance. If the system is not cleaned regularly, which is the case in half of the households with balanced ventilation in the Netherlands [6], the capacity of the system may be reduced strongly. Occupants of dwellings with balanced ventilation report fewer burdens from draught from windows and grilles, but report draught from supply air inlets [12]. If the air inlets don't fit with the esthetical preferences of occupants they may remove them [10]. Finally, noise produced by the mechanical ventilation system plays a main role ([10], [14]). In 28 % of dwellings equipped with balanced ventilation occupants are "always" or "often" annoyed by noise and 37 % of the occupants "sometimes". Because of this noise nuisance, 17 % of the occupants ventilate less with the mechanical system than they would like [6].

Based on this literature study and on 18 qualitative interviews with occupants in their houses, combined with an inspection, different user scenarios are established. All occupants rent their houses from a housing association. The aim of the in-depth interviews is to get insights into the relationship between the technical characteristics of the balanced ventilation system and its use by occupants. In a following phase of this research, these interviews will be used as basis for a large survey in order to gather statistical data. The results of these interviews are summarized hereafter. Most of the occupants are not aware of the energy saving aims of balanced ventilation. They do not know that they can save energy by closing the windows in the winter. They do know that the system is needed to supply continuous ventilation. To control the system they use the switch, in all cases placed in the kitchen, but are not aware of other possibilities like the use of the bypass in summer situation. This is because of the poor quality of the use instructions from the producer (no difference in instructions for servicemen and for occupants) and of the housing associations. The design of the switches plays an important role in the use of the system. Balanced ventilation systems run almost always in the lowest stand. When the system is equipped with a two-stand switch the lowest stand has a capacity about 50 %. Systems with a three-stand switch have a capacity of only 30 % in the lowest stand. The location of the switch is important too. 22 % of the interviewees are lacking a switch in the bathroom. Because of this the bathroom is less ventilated than needed. The design of the control display on the system itself is not adapted to occupants. Most of them do not understand the function of the comfort temperature and of the bypass. Most of the interviewees are aware of the necessity of cleaning the inlets and the filters and also clean them regularly. Occupants have relatively often the impression that the system does not supply fresh air, whereas the window does. They have doubts whether the system provides enough fresh air and open therefore the windows. Some odours from outside, like barbecue odours are smelled inside. When this is the case, some occupants shut the ventilation system down by disconnecting the plug. The air exhaust in the kitchen is perceived as insufficient. As a reaction some occupants placed a supplementary cooker hood on the existing system. As a result the whole system is disturbed. The supply air is often perceived as being too dry. In this case, they would open the window or use humidifiers on the radiators. Some of the occupants close the air inlet when they feel draught. Mostly the air inlets then remain closed. Avoiding noise from the ventilators is a reason why the system works in a lower stand than the maximal one. For the same reason some occupants shut the supply ventilator down. The use of the switch is linked to specific activities like cooking, showering, using the toilet, using a tumble dryer en receiving visit, but in most cases, the switch remains continuously in the same low stand.

Characteristics of the studied dwelling and ventilation scenarios

For the calculations we used a typical Dutch terraced house with three bedrooms and an attic. It is a three storey house with a gross floor area of 124 m² and a gross storey height of 2.8 m. The use area as defined in the building decree is 63 m². Facades, roof and ground floor have a Rc value of 2.5 m²K/W. The south and north façades have an area of 46 m², with a window percentage of 30%. Roof and ground floor have an area of 52 m². The west and south facades are outer separation walls with other dwellings. High efficiency glazing is used (U-value= 2.1 W/m²K). The dwelling is heated by a high efficiency boiler (efficiency of 0.83 on yearly basis).

The Dutch Building Decree prescribes the minimum flow rates given in table 1 [15]. The floor areas and the minimum ventilation flow rates for this dwelling calculated according to the Dutch building decree are summarized in table 2. At the level of the dwelling the minimum required air flow rate is therefore 259 m³/h.

Table 1: Minimum flow rates.

	Living area (bedrooms and living room)	Kitchen	toilet	Bathroom
Flow rate (m^3s^{-1})	$0,9x10^{-3}$ per square meter with a minimum of 7×10^{-3} total	$21x10^{-3}$ total	7×10^{-3} total	14×10^{-3} total

Table 2: Areas and minimum flow rates for a typical Dutch house.

	Living room + kitchen	Bedroom 1	Bedroom 2	Bedroom 3	Bathroom
Area (m²)	37,1	9,9	9,7	6,0	6,6
Flow rate (m^3h^{-1})	120.2	32.0	31.3	25.2	50.4

To limit costs, balanced ventilation systems in the Netherlands are mostly designed to meet the requirements on the minimum ventilation rates, but not more. Furthermore, the legislation prescribes that these minimum ventilation flow rates must be achieved when the system runs at its highest level (level 3). The systems installed in the inspected dwellings were designed this way. Balanced ventilation systems in dwellings are mostly

equipped with switches that can be put in levels 1, 2 or 3. The interviews showed that the systems are running most of the time in level 2 in which only 50 % of the capacity is available. At night time, the system mostly runs in level 1 due to noise pollution from the ventilator, with only 30 % of the capacity. Furthermore, studies have shown [16] a reduction of the capacity of 10% a year, due to a too low level of maintenance (cleaning). After 5 years, the maximum capacity is reduced to 65 % of the original capacity. The total amount of fresh air supplied to the dwelling is also dependent on the air-tightness of the construction. The air-tightness of Dutch new dwellings should not exceed 200 dm³/s per 500 m³ building volume. For the reference dwelling studied, the air flow rate should be lower than 200 dm³/s. In the calculations we used this value as the upper limit of the infiltration flow rate.This is to take into account the possible opening of windows to ventilate. In the Dutch mandatory energy performance norm [17], it is assumed that an air infiltration flow rate lower than 0.4 dm³/sm² cannot be achieved. We used this value as the lower limit for the infiltration rate.

Five ventilation scenarios, depending on occupant behaviour are. For each scenario, a variant is calculated with low infiltration rate (variant a), high infiltration rate (variant b) and with low infiltration rate and supply ventilator shut down (variant c). In this case, the heat recovery doesn't work. In variant a) en c) the infiltration flow rates are 91 m³/h and in variant b) 720 m³/h. The five scenarios' are:

1 Use of a 5 years-old system with a 3-stands switch. The filters have never been cleaned or not often enough, thus the maximum capacity of the ventilation system is 65 % of the mandatory capacity as given in table 1. The system runs continuously in stand 1, thus 30 % of the maximum capacity is available. The mechanical ventilation flow rate is therefore 50.4 m³/h, which is far below the mandatory value.
2 Use of a 5 years-old regularly cleaned system with a 3-stands switch. The system runs continuously in stand 2, thus 50 % of the mandatory ventilation capacity is available, which is 129 m³/h.
3 Optimal use of the system: 100% of the mandatory capacity is available, that is 258 m³/h.
4 The occupants shut the whole system down and only use natural ventilation. The ventilation flow rate is 258+91=349 m³/h.
5 The occupants shut the whole system down and only use natural ventilation. There is a high infiltration flow rate of 720 m³/h.

Calculation model

The calculations are conducted using the software h.e.n.k. [18]. The energy demand for heating is calculated using a heat balance within the dwelling envelope, based on hourly calculations taking into account transmission losses, infiltration and ventilation losses and heat load through sun entering the building and electrical appliances. We assumed that the heat load from lighting was maximum 4 W/m² and for appliances 3 W/m². Maximum 4 people are present in the dwellings. We developed and used occupancy and heat load profiles as shown in figure 1.These profiles give the fraction of maximum occupancy and heat load for each hour of the year. The profiles are identical for all ventilation scenarios.

The ventilation flow rate was kept on the same level all over the day, which is plausible from the interviews. We also assumed that an average temperature of 18 °C was maintained all over the day. The heat exchanger recovers heat from the exhaust air as long as the temperature difference between outdoor and indoor air is 2 K. Below this limit there is no heat recovery. Hourly calculations are made, using the Dutch average climate year TRYdeBilt.

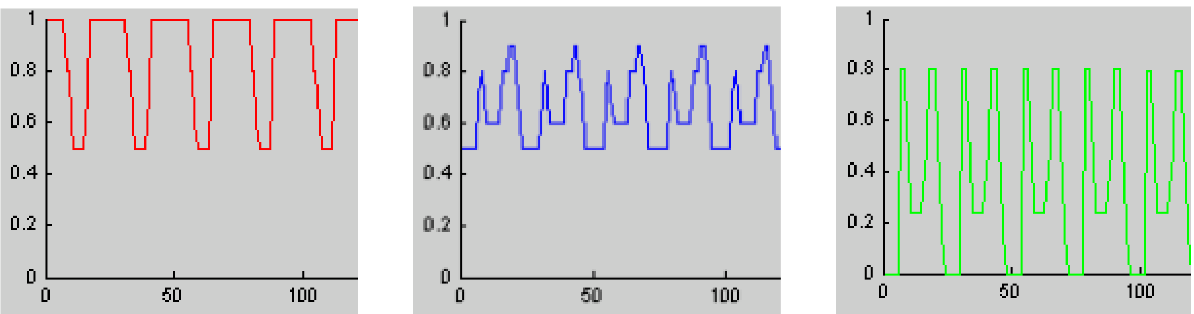

Figure 1: 1) occupancy profile, 2) appliances profile, 3) lighting profile x-as: hours (0 is midnight); y-as: fraction of the maximum occupancy, heat load from appliances and heat load from lighting respectively.

The energy consumption at the meter box, which is the energy delivered to the heating and ventilation system (gas and electricity), is calculated from the energy demand. For room heating a high efficiency boiler is used (η=0.83 on year basis). The electricity needed for the ventilation systems is calculated assuming a pressure drop of 150 Pa in the supply and exhaust ducts. The efficiency of the ventilators is 0.85 and the efficiency of the motor is 0.7. The primary energy use is calculated from the energy consumption. It includes the energy needed to generate electricity. We used an average power plant generation with an efficiency of 0.4.

RESULTS AND DISCUSSION

The results of the calculation are shown in table 3. The total primary energy use of all options is plotted in figure 3. The electricity consumption of the ventilators appears to be minimal in comparison with the gas consumption for heating. If the occupants shut down the supply ventilator, there is no heat recovery (variant c). Compared to the case with heat recovery (variant a), an increase of primary energy use from 11 % (, variant 1) up to 36 % (variant 3) is observed. In variant 3, the system works optimally and the air flow rate is much higher than in variant 1, where the system is sub-optimal. Heat recovery plays a more important role at larger flow rates. That is why the effects of not using heat recovery are stronger in variant 3 than in variant 1. Comparing variant 1, 2 and 3 it appears that the energy use of the sub-optimal systems is lower than the energy use of the optimal system (variant 3). This is logical because of the decrease in ventilation losses arising from the reduced air flow rates. However, the flow rates in variants 1 and 2 are far below the minimum acceptable ventilation flow rate, resulting in a poor indoor air quality and possibly in health problems. It is plausible (see first section of this paper) that the occupants would compensate these too low air flow rates by opening windows more often, resulting in variant 1b, 2b, 4 or 5 depending on the air tightness of the house and on the way occupants use natural ventilation openings. It is therefore possible that no energy savings are achieved by a balanced ventilation systems when compared to natural ventilation. It appears from figure 3 that air tightness of the house and use of the natural ventilation openings will play a major role in achieving or not the aimed savings. The comparison of a well working balanced ventilation system (option 3a) with natural ventilation option 4, shows that the maximum savings could amount 24 %, but could also be reduced to zero or even be negative.

Table 3: Energy consumption and primary energy use.

	1a	1b	1c	2a	2b	2c	3a	3b	3c	4	5
Gas cons. (heating), kWh	5918	23597	6693	7145	24824	9130	9157	26836	13127	13127	22812
Elect. cons. (ventilators), kWh	63	63	31	160	160	80	321	321	160	0	0
Total primary energy use, MJ	21870	85514	24379	27162	90806	33588	35856	99500	48700	47257	82123

1: sever under-capacity (30x65%); 2: under-capacity (50%); 3: full capacity; 4: natural ventilation, same capacity as in 3); 5: natural ventilation, high infiltration.
a: low infiltration; b: high infiltration; c: no heat recovery

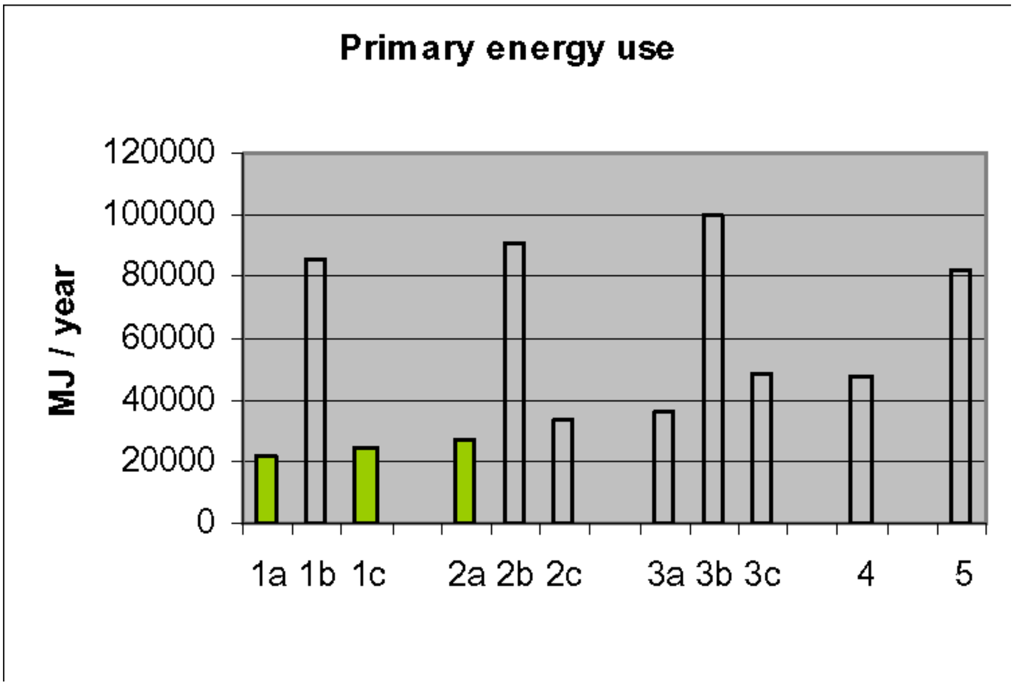

Figure 3: comparison of the primary energy use for several options.

CONCLUSIONS

In this paper we gave an overview of the ways occupants use ventilation systems and made a first attempt to quantify the effects of their behaviour on the final energy use for heating. This energy use was studied for several behaviour scenarios, leading to the conclusion that occupant behaviour may easily reduce the predicted savings to zero, or even may increase the energy use when compared to natural ventilation. From the data we gather it appears that most of occupants use the system in sub-optimal settings, in which energy savings may be high because of the low ventilation flow rates. However this would cause a poor indoor air quality, which may be compensated by a more wide use of natural openings, resulting in an energy performance identical to that of a natural system. More research is needed to obtain statistical data and to validate the calculation model, in order to be able to make more accurate prediction of the possible savings by balanced ventilation systems.

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Air pressure conditions in Finnish residences

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SUMMARY

Air pressure conditions in typical Finnish residences are analyzed using data from field measurements and computer simulations. Field measurements were conducted in a two-storey detached house and in a five-storey apartment building. The effects of airtightness, ventilation rate, air leakage distributions, and outdoor environmental conditions on air pressure conditions in a detached house were simulated on a multi-zone simulation model using the IDA ICE simulation program. There is almost always a positive and negative air pressure difference across the building envelope in detached houses and in apartment buildings. During winter, in detached houses there is continuous positive air pressure at ceiling level on the top floor and negative pressure at floor level on the bottom floor. For detached houses, the design value of air pressure difference across the building envelope for the moisture convection analysis is close to ±10 Pa. The control of air pressure difference in normal and leaky houses is difficult: ± 15 differences in ventilation airflows have only a minor influence on the air pressure difference. To control air pressure differences, the balancing of ventilation systems in airtight houses is very important.

INTRODUCTION

The uncontrolled air movement through a building envelope leads to problems related to hygrothermal performance, health, energy consumption, performance of the ventilation systems, thermal comfort, noise, and fire resistance. Air leakage through a building envelope depends on the result of air pressure differences across the envelope and on the airtightness of the building envelope.

Air and moisture convection through the building envelope may cause severe moisture loads to be imposed on the structure. Indoor air exfiltration in cold climates may cause moisture accumulation or condensation, leading to microbial growth on materials, change of the properties of the material or even to structural deterioration. The relative humidity (RH) at the connection of floors and external walls in multi-storey platform timber-frame houses can be rather high, causing the risk of mould growth and rot fungi when there is positive air pressure inside the building [1]. The simulation results of Janssens and Hens [2] have shown that, even when a roof design complies with condensation control standards, a lightweight system remains sensitive to condensation problems because of air leakage through the discontinuities, joints and perforations, common to most existing construction methods.

Air leakage through a building envelope could introduce outdoor or crawl-space airborne pollutants to the indoor air. Field measurements [3, 4] have shown evidence of fungal spores being transported indoors from a crawl space. Although radon can enter buildings by several mechanisms, the dominant radon entry mechanism is air leakage through the basement floor [5, 6, 7].

Hygrothermal design is one way to assess and predict the long-term hygrothermal performance of the building envelope to prevent moisture damages and to guarantee a healthy indoor climate and longer service life for buildings. Air pressure difference is one load component in the hygrothermal design of building envelopes. Wind, stack effects, mechanical ventilation and distribution of leakage sites in the building envelope influence air pressure difference. Each one of these effects influences the air pressure difference independently and in a specific way.

As many factors affect air pressure conditions, one of the most reliable ways to determine the relevant design values is to perform long-term field measurements in typical residences. To understand how different factors influence the air pressure conditions, the use of simulation is a valuable tool that makes it possible to show the importance of wind, stack effects, mechanical ventilation and distribution of air leakage places. In this study, air pressure conditions in typical Finnish residences are analyzed using data from field measurements and computer simulations.

METHODS

Field measurements

Field measurements were conducted in a two-storey detached house and in a five-storey apartment building, Figure 1. Residences were selected to represent typical Finnish residences. A detached house was selected as having average properties according to a previous field study of timber-fame detached houses [8]. There was mechanical supply and exhaust ventilation and an electrical floor and ceiling heating system in the studied detached house. The apartment building had a centralized supply and exhaust ventilation system that was demand controlled to keep RH and CO_2 below the target values of 50% and 600 or 1000 ppm.

Figure 1 Studied detached house (left) and apartment building (right)

Air change rate, indoor and outdoor temperature and humidity, as well the air pressure difference over the building envelope were monitored during a three-week period in winter. The ventilation airflows were measured from ventilation ducts at different ventilation fan speeds. During the measurement period, the ventilation fan speed was monitored continuously. Whole air change rates of residences were measured with active and passive (only in detached house) tracer gas air infiltration measurement techniques also. The airtightness of the building envelope was measured using the fan pressurisation method [9]. The values of air pressure differences over the building envelope (ΔP), temperature (T), and RH from inside and outside the building were monitored at 15 min. intervals. Table 1 shows the main characteristic of the studied dwellings.

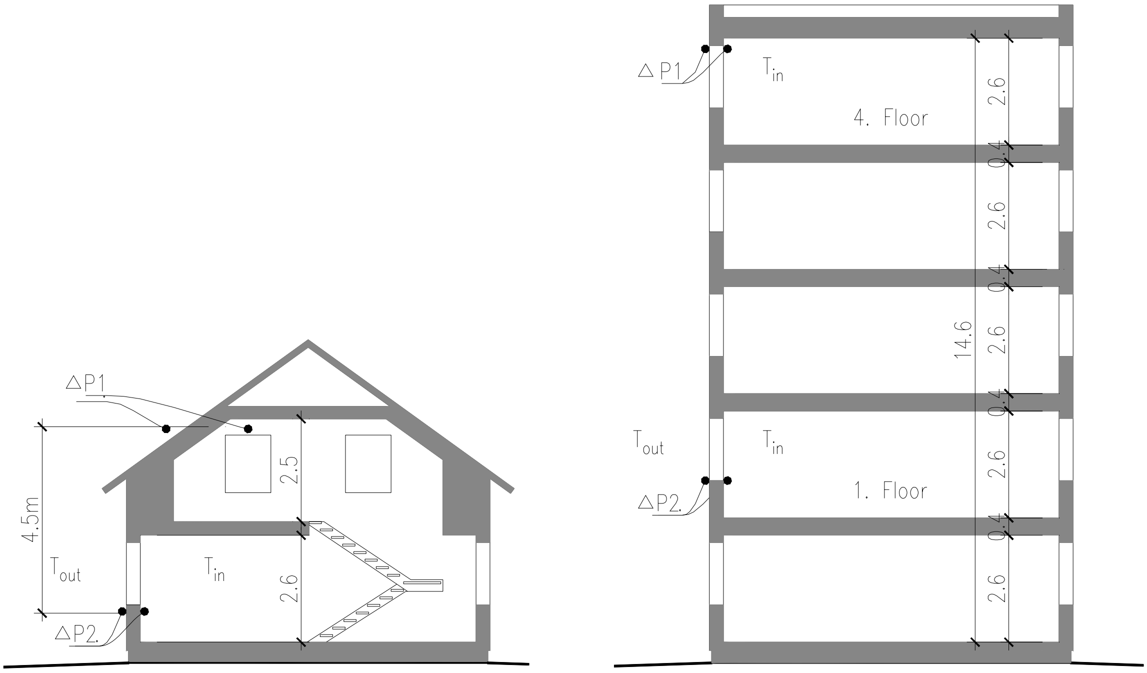

Figure 2 Measurement points in detached house (left) and in apartment building (right)

Table 1 Main characteristic data of the measured dwellings

	Detached house	Apartment on the first floor	Apartment on the fourth floor
Measurement period	4.3…24.3.2005	16.3…06.4.2005	16.3…06.4.2005
Floor area	183 m^2	70 m^2	70 m^2
Volume	452 m^3	183 m^3	183 m^3
Envelope air leakage rate at 50 Pa	3.9 1/h	1.0 1/h	1.9 1/h
Average ventilation rate	0.41 ach 13 l/(s·pers.), 0.29 l/(s·m^2)	0.68 ach 9 l/(s·pers.), 0.49 l/(s·m^2)	0.60 ach 15 l/(s·pers.), 0.44 l/(s·m^2)

Computer simulations

To analyse the effects of airtightness, ventilation rate, air leakage distributions, and outdoor environmental conditions on air pressure conditions, the measured detached house (Figure 1, left) was simulated on a multi-zone simulation model using the IDA-ICE [10, 11] simulation program. The software is an advanced tool for simulation of thermal comfort, indoor air quality and energy consumption in buildings. The simulation model was validated on the basis of field measurements [12]. The effect of the airtightness of the building envelope, distribution of the air leakage places, performance of ventilation, and surrounding conditions (suburban, open country, no wind cases) on air pressure differences were analysed in twenty-one different simulation cases, Table 2.

An air leakage rate per hour at 50 Pa of pressure difference (n_{50}) is changed between n_{50}=0.15…10 ach. In the case of n_{50} = 0.15 ach, a building is almost completely airtight without air leaks. The airtightness value n_{50} = 3.9 ach was the airtightness of the simulated detached house and can be considered typical of Finnish timber-frame detached houses [8]. The airtightness value n_{50} = 10 ach was selected to represent a very leaky house. The effect of the air leakage distribution is simulated with three different cases: 75 % of the leakages at the top of the house and 12.5 % of the leakages at the middle and bottom of the house; equal distribution; and 75 % of the leakages at the bottom of the house and 12.5 % of the leakages at the middle and top of the house. The effect of the performance of ventilation is simulated with three different cases: balanced system (supply airflow = exhaust airflow), 15 % decreased supply airflow, and 15 % increased supply airflow. For the reference case, the "CASE-1 n4" (n_{50} = 3.93 ach, suburban surrounding, normal air leakage distribution and balanced ventilation system) was selected.

Table 2 Simulation cases

Description:		Simulated cases n_{50} = 0.15 ach	 n_{50} =3.9 ach	 n_{50} = 10 ach
Surroundings	Suburban	CASE-1 n0.15	CASE-1 n4	CASE-1 n10
	Open country	CASE-2 n0.15	CASE-2 n4	CASE-2 n10
	No wind cases	CASE-3 n0.15	CASE-3 n4	CASE-3 n10
Air leakage distribution	Most leakages at the top of the house (75/12.5/12.5%)	CASE-4 n0.15	CASE-4 n4	CASE-4 n10
	Normal	CASE-1 n0.15	CASE-1 n4	CASE-1 n10
	Most leakages at the bottom of the house (12.5/12.5/75%)	CASE-5 n0.15	CASE-5 n4	CASE-5 n10
Ventilation performance	15 % decreased supply airflow	CASE-6 n0.15	CASE-6 n4	CASE-6 n10
	Balanced system	CASE-1 n0.15	CASE-1 n4	CASE-1 n10
	15 % increased supply airflow	CASE-7 n0.15	CASE-7 n4	CASE-7 n10

Air pressure differences were calculated from three rooms: from the sauna and living room on the ground floor and the bedroom on the top floor. Air pressure differences were calculated from each room at ceiling and floor levels. The simulations are carried out with hourly Helsinki weather data (1979), which is commonly used as test data for energy calculation in Finland [13].

RESULTS

Field measurements

Field measurements in the detached house showed +3 Pa air pressure at the ceiling level of the top floor and -4 Pa air pressure at the floor level of the ground floor on average, Figure 2, left. The dependency between air pressure difference and temperature difference in the detached house is shown in Figure 3, right. The trend of the measurement results shows good agreement with the theoretical air pressure (red curve) due to stack effect (Eq. 1). The deviation from the theoretical air pressure (red curve) is mainly due to wind effect. The standard deviation from the theoretical curve is 0.4 Pa.

$$\Delta_{ps} = \rho_{in} \cdot g \cdot H \cdot (T_{in} - T_{out}) / T_{out}, \qquad (1)$$

where Δ_{ps} is air pressure due to stack effect, Pa; ρ_{in} is density of the indoor air, g/m^3; g is gravitational constant, 9.8 m/s^2; H is height between measurement points, m; T_{in} is indoor temperature, C; T_{out} is outdoor temperature, C.

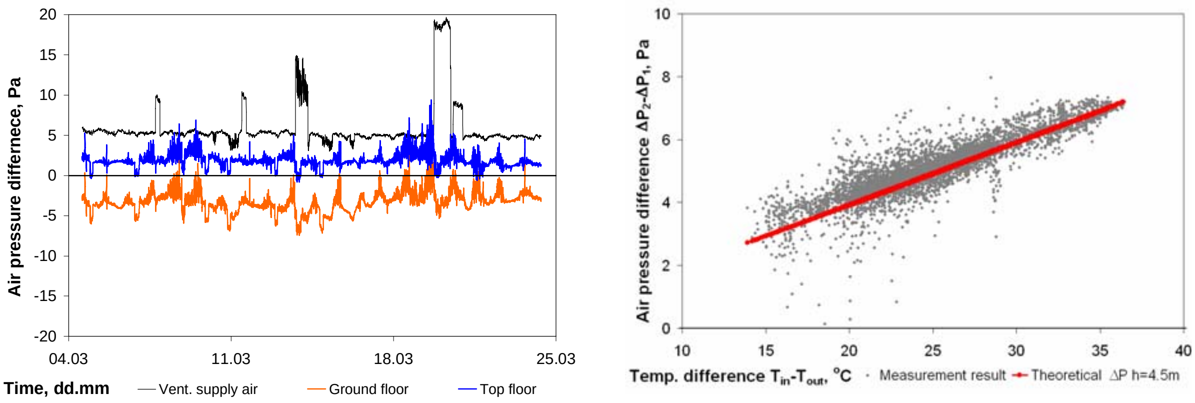

Figure 3 Air pressure difference over the building envelope during the measurement period (left) and dependency between air pressure difference and temperature difference (right) in the detached house

In the apartment building, on the first floor, the daily average air pressure difference over the building envelope was -11 Pa on average, while, on the fourth floor, it was -2 Pa, Figure 4, left. In Figure 4, left, the performance of ventilation on the apartment's second floor is also shown. The black curve is the air pressure measurement from the supply air duct. Ventilation airflows were 20% of the design value when the CO_2 level inside the flat was less than 600 ppm; the design values were switched when the CO_2 level inside the flat was higher than 1000 ppm or when the relative humidity in the bathroom exceeded 50%. This ventilation performance made a 3-Pa air pressure difference across the building envelope (Figure 4,right).

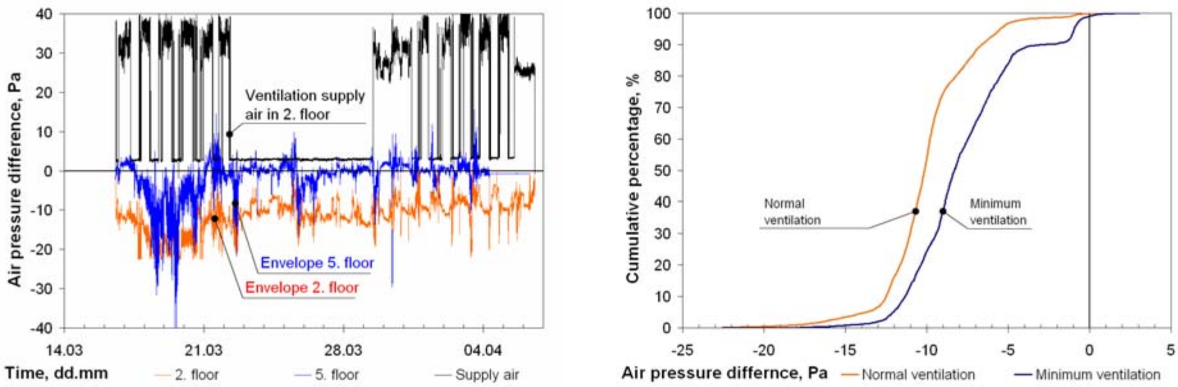

Figure 4 Air pressure difference over the building envelope during the measurement period (left) and dependency of ventilation performance on the air pressure difference (right) in the apartment building

Computer simulations

The annual air pressure difference across the envelope at ceiling level on the top floor and at floor level on the ground floor of the reference case, CASE-1 n4, is shown in Figure 5, left. Distribution of the air pressure difference during the winter season on all six simulation points is shown in Figure 5, right. Figure 6 compares the distribution of the air pressure difference in the average case with 10 % and 90 % percentile cases. There is almost always positive pressure at ceiling level on the top floor and negative pressure at floor level on the ground floor during winter. The intermediate floor can be under either positive or negative pressure.

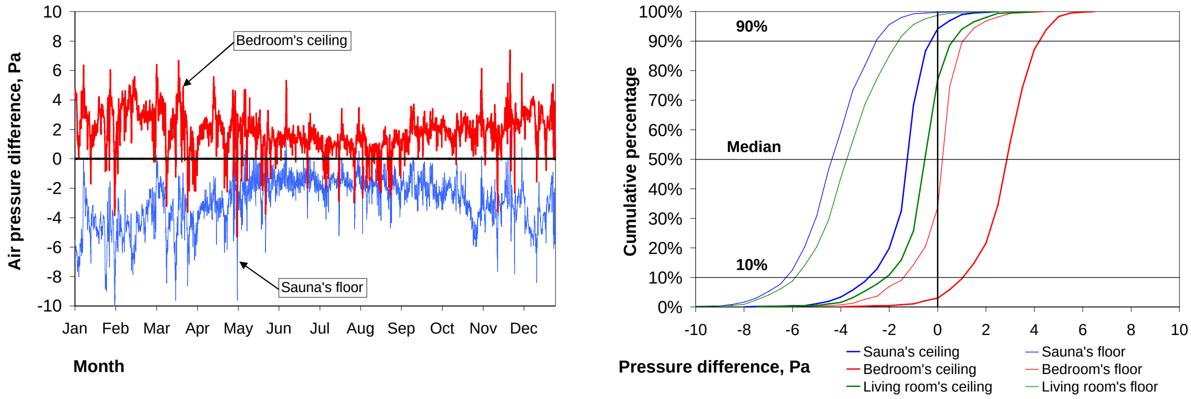

Figure 5 Air pressure difference during whole year (left) and during winter months (right)

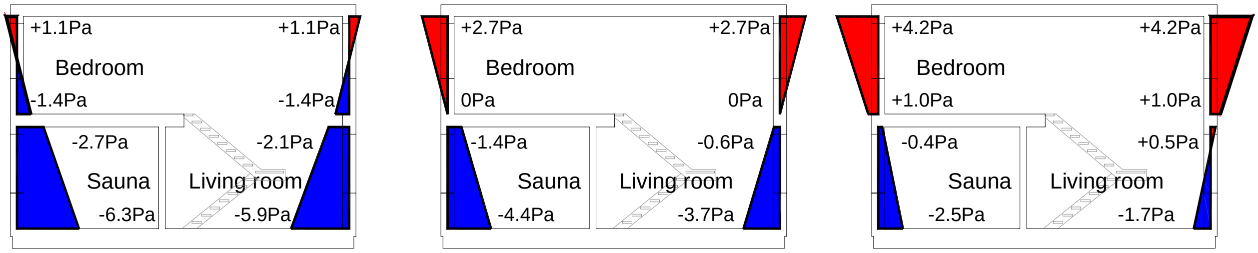

Figure 6 Average (middle), the lowest and highest 10 % percentile (left and right) air pressure difference during winter period

Table 3 shows the simulation results from two points: 10 % percentile value from the sauna at floor level and 90 % percentile value from the bedroom at ceiling level. These two points represent minimum and maximum air pressure differences at the 10 % critical level. The overall consensus [14] is that 10 % is the appropriate level to determine loads for hygrothermal calculations.

Table 3. Simulation results

Description:		Air pressure difference, Pa					
		n_{50} = 0.15 ach		n_{50} = 3.93 ach		n_{50} = 10 ach	
		10%	90%	10%	90%	10%	90%
Surroundings	Suburban	-7	+4	-6	+4	-6	+4
	Open country	-11	+8	-10	+8	-8	+7
	No wind cases	-6	+4	-5	+4	-5	
Air leakage distribution	Most leakages at the top of the house	-10	+2	-9	+2	-8	+2
	Normal	-7	+4	-6	+4	-6	+4
	Most leakages at the bottom of the house	-4	+8	-3	+8	-3	+8
Ventilation performance	15 % less supply airflow	-33	-22	-7	+4	-6	+4
	Balanced system	-7	+4	-6	+4	-6	+4
	15 % more supply airflow	+15	+26	-6	+5	-5	+4

Simulations show that in critical cases value of the air pressure difference across the building envelope at the 10 % percentile level is on the level +8 Pa and -11 Pa. In the most critical cases (airtight envelope with unbalanced supply and exhaust ventilation), the design value of air pressure difference across the building envelope could be up to +26 Pa and -33 Pa.

DISCUSSION

The air pressure difference across the building envelope at the 10 % percentile level in the typical detached house was +8 Pa and -11 Pa. Apartment buildings are much higher, and measurements show that the building envelope of such buildings is more airtight. Both these factors influence the air pressure difference across the building envelope and should be analyzed in addition to the others.

Air pressure simulations in the detached house were carried out during the energy test year in Finland [13]. The annual average temperature is +4.3°C, the average temperature of the coldest month, February, is −9.8°C, and that of the warmest month, August, +16.1 °C. For the period 1971-2000 in Helsinki, the mean annual temperature was +5.6 °C. Since 1900, the warmest annual average temperature in Helsinki has been +7.2 °C (1934) and the coldest +2.7 °C (1902). The annual average temperature in Finland changes from +6 °C in the southern coastal area to -2 °C in the north of Lapland [15]. The influence of different years and different locations, including apartment buildings, on air pressure difference should be studied in future studies.

CONCLUSIONS

There is almost always a positive and negative air pressure difference across the building envelope in detached houses and in apartment buildings. During winter, there is continuous positive air pressure in the detached house at ceiling level on the top floor and negative pressure at floor level on the bottom floor.

Wind influences, first of all, air pressure peak values. Comparing average values, the influence of wind is less.

For detached houses, the design value of air pressure difference across the building envelope for the moisture convection analysis is close to ±10 Pa. In the most critical cases (airtight envelope with unbalanced ventilation), the design value of air pressure difference across the building envelope could be up to +26 Pa and -33 Pa.

The control of air pressure difference in normal and leaky houses is difficult: ±15 differences in ventilation airflows have only a minor influence on the air pressure difference. To control extremely high air pressure differences, the balancing ventilation systems in airtight houses are very important.

ACKNOWLEDGEMENT

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Finding Optimal Airflow in Connection with The Location of Air Inlets And Outlets to Control Odor Dispersion in The High-rise Residential Buildings

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SUMMARY

Food odor dispersion from residential unit to core is one of problems in high-rise residential building. In this study, it was analyzed in terms of stack effect, and the method how optimal air inflow of core was estimated and how the location of air in/outlet were decided were suggested to solve it. A combined CFD(Computational Fluid Dynamics) and CONTAMW analysis was used for stack effect of building, dispersion of food odor, optimal air inflow of core, and the location of air in/outlet in the method.

INTRODUCTION

Recently, the number of high-rise residential building is increasing in Korea. In the building, the opening/closing function of window for natural ventilation is not flexible because the upper part of the building has to be safe from strong wind and energy has to be conserved by strong insulation and airtightness. Moreover, curtain wall systems which usually have small area of openable window are used for it. Accordingly, the mechanical ventilation system is used more than natural one in it [1]. The factors that affect on ventilation performance are indoor air flow by envelop airtightness, backward flow of exhaust due to less of ventilation for upper stories, and stack effect by pressure difference between spaces [2-5]. If the amount of ventilation and the location of air in/outlet are decided to exhaust pollutants without considering the factors, the pollutants can be rather dispersed to non-polluted area. Especially, food odor is used for this study. Residents have direct sense of it, so it can be a good sample among pollutants. Korean food usually generates a lot of odors during cooking, so the odors are dispersed to dining room and core area. Ultimately, the residential unit which generates the odors gives discomfort to neighbors which are another residential unit [6].
The purpose of this study is to control the food odor that is dispersed from a residential unit to core and to show a method how the optimal amount of ventilation and optimal position of air in/outlet are decided. For this study, ventilation performance according to stack effect in high-rise residential building is analyzed by CFD and CONTAMW.

METHODS

In this study, the combined CFD and CONTAMW analysis were suggested to decide the optimal amount of ventilation and optimal position of air in/outlet. There are 5 steps for it, and Figure 1 is a summary (diagram) about it.
First of all, Step 1 is about building descriptions. A boundary condition is explained in the step for combined analysis such as building data, operating schedule of air in/outlet in a residential unit, type and density of a food odor, air leakage data, and weather data etc.

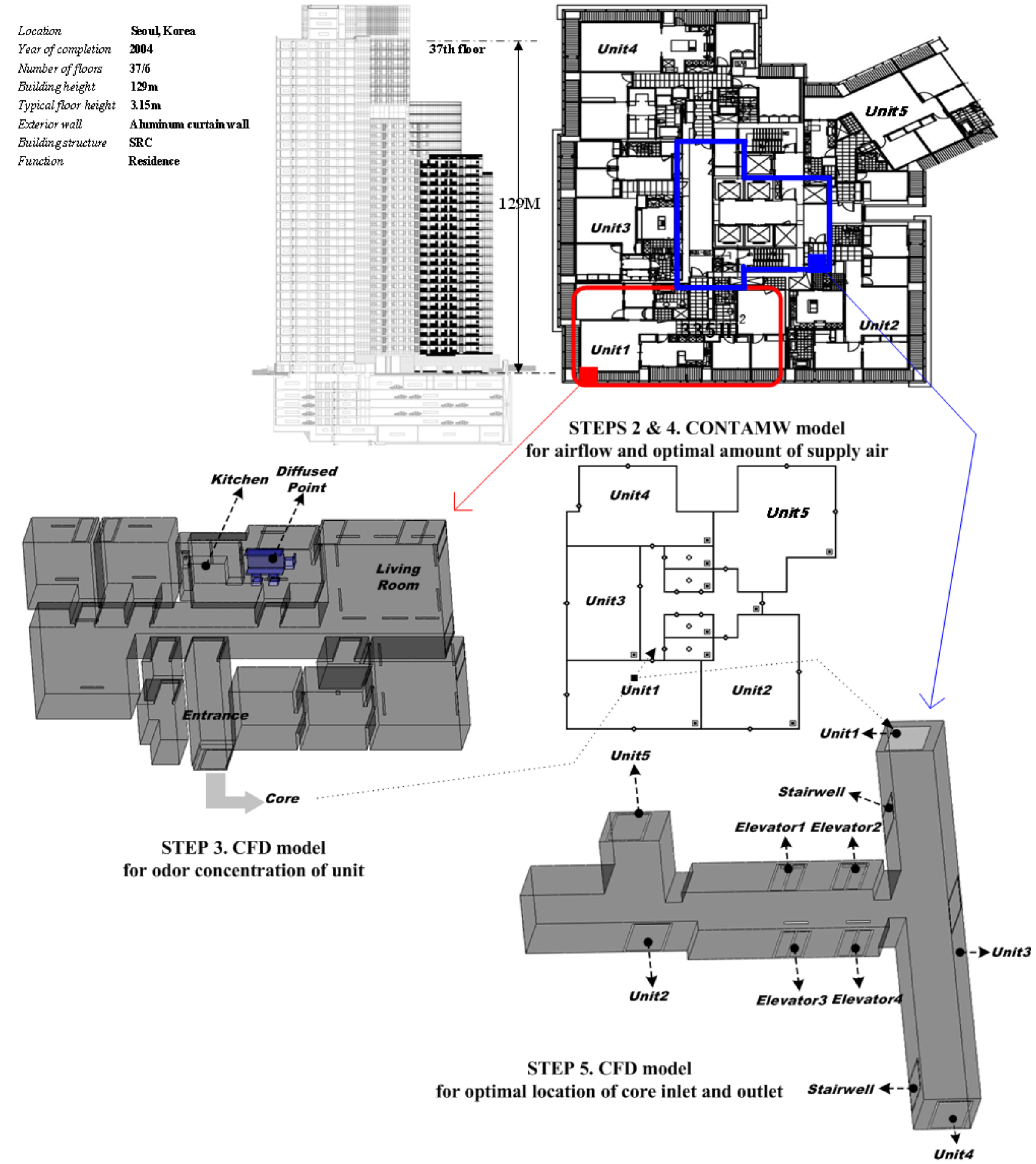


Figure 1. Summary diagram of combined CFD and CONTAMW analysis

Second, Step 2 is to calculate pressure difference of stack effect by CONTAMW and the amount of airflow shifting between spaces. Third, Step 3 is to analyze density distribution of food odor that is dispersed from the residential unit by CFD with the boundary conditions that are result of Step 2 and generated intensity of food odor. Forth, Step 4 is to calculate the optimal amount of ventilation to dilute inflow food odor from the residential unit when air in/outlets are applied in core. For this step, CONTAMW is used, and the result of food odor density that is the result of Step 3 around front door area is used as a boundary condition. Finally, Step 5 is to evaluate the location scheme of air in/outlet that can restrict and isolate the dispersion of food odor by CFD with the boundary condition that is from result of Step 4.

Table 1. Odor elements and intensity

	Food type[a]					Smell	Intensity scale (unit: ppm)[b]				
	Live stock	Fish	Kim-Chi	Starchy	flavors		1	2	3	4	5
Hydrogen sulfide	◐	●	◐	●		rotten egg	0.0005	0.0056	0.063	0.72	8.1
Methyl merchaptan	○	○	○	○		onion	0.0001	0.00065	0.0041	0.026	0.16
Ammonia	○	○			○	urine	0.15	0.59	2.3	9.2	37
Tri-methyl amine		●	◐			fishy	0.00011	0.0014	0.019	0.24	3.0
Acetic aldehyde					○	mold	0.0015	0.015	0.15	1.4	14

[a] ●Major, ◐detected, ○Possible
[b] Intensity scale : 0 None, 1 Threshold, 2 Moderate, 3 Strong, 4 Very Strong, 5 Over Strong

Table 2. Air leakage data

Building components		Air leakage data
Exterior wall	At lobby	$EqLA_{75}$ 2.1 ㎠/㎡ [a]
	At typical floors	$EqLA_{10}$ 1.51 ㎠/㎡
Door	Residential entrance door	$EqLA_{10}$ 70 ㎠/item
	Elevator door	$EqLA_{10}$ 325 ㎠/item
	Stairwell door	$EqLA_{10}$ 120 ㎠/item
	Door for condensing room	$EqLA_{10}$ 23 ㎠/item
	Swing door located on lobby and basement floor	430 CMH at 50 Pa
	Revolving door located on lobby floor	73 CMH at 50 Pa
Etc.	Top of elevator shaft	Equivalent orifice area of 1.0 ㎡

[a]$EqLA_{75}$: Equivalent leakage area at 75 Pa, $EqLA_{10}$: Equivalent leakage area at 10 Pa

Step 1. building description

In this study, a building is selected for combined CFD and CONTAMW analysis. It has thirty-seven stories above and six under the ground with a core system. Figure 1 shows summary, cross-sectional view, and standard floor plan of it. During cooking, generated food odor is collected by force of kitchen hood and outlet, and line inlet supplies air to prevent dispersion of food odor. However, during meal time, residents are usually turn kitchen hood and outlet off for comfortable meal time even though food odor keeps generating. In brief, food odor during meal time is going to be dispersed a lot more than during cooking because food odor can't be collected by kitchen hood or outlet. There is unusual case in Korea when dispersion of food odor is worst; residents set up table in dining room and have meal rather than they have meal time in kitchen that have good air outlet system. In this study, meal time was set as a boundary condition for CFD analysis for food odor dispersion of a unit, and food odor generating point during meal time was set as dining room which was the worst case.

The type of odor and density. Table 1 is about stench intensity according to type and density odor during food production [7]. It is from measurement method of chemical analysis. There are three main causes of food odor that are meat odor, Kim-Chi odor, and fish odor in Korean household. The factors of meat odor are Hydrogen sulfide(H_2S) and Ammonia(NH_3), the factors of Kim-Chi are Hydrogen sulfide(H_2S) and Methyl Merchaptan(CH_3SH), and the factors of fish odor are Hydrogen sulfide(H_2S) and Tri-methyl amine ($(CH_3)_3N$) etc.

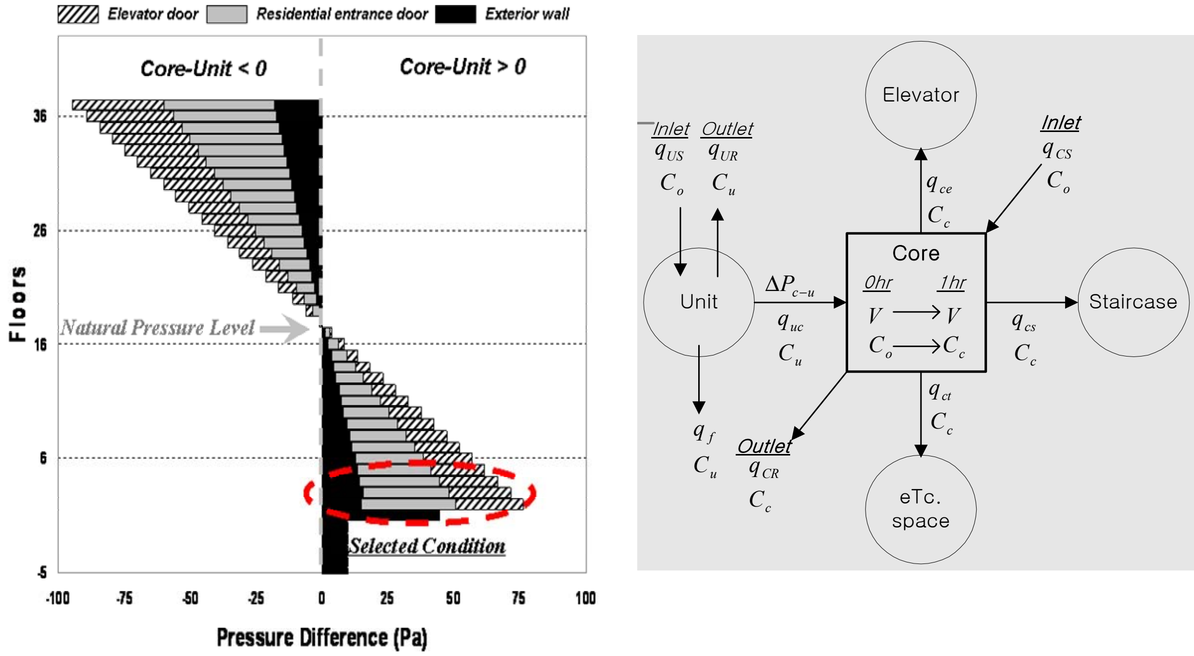


Figure 2. Pressure difference and airflow direction

In this study, fish odor was selected for dispersion of food odor and dilution of it in core was analyzed. In case of fish odor, the main causes of odor are Hydrogen sulfide (H_2S) and Tri-methyl amine($(CH_3)_3N$). Accordingly, the hypothesis was set; Tri-methyl amine($(CH_3)_3N$) of 3ppm which is 5 of stench intensity is generating on diffused point in residential unit in terms of STEP 3. Also, 0.00011ppm of it was set as sensible stench intensity of STEP 4 to decide the optimal amount of air supply that satisfies under sensible stench density.

Air leakage data and weather data. Table 2 is about the air leakage data of exterior wall, the front door, elevator door, a stair hall door, and lobby door etc from the study of Jo et al [5] which analyzed stack effect of high-rise residential building. Stack effect is proportional to difference between indoor temperature and outdoor temperature, so stack effect is going to be bigger as outdoor temperature is decreased. Accordingly, 22℃ and -11.9℃ of TAC2.5 which are standard temperature of heating design in Seoul area were set as indoor and outdoor temperatures, respectively [8]. Also, it is assumed that there was no wind for this study.

Step 2: airflow q

The amount of air in/outflow of residential unit and core, elevator shaft, a stair hall in entire of building that has a standard floor plan as shown in Figure 1, was calculated by CONTAMW. According to Figure 2, the air pressure of core which is above floor from middle floors (17th and 18th floor) is higher than other area, so the air flows from core to residential unit. Also, in case of the floors which are under middle floors, the air flows from residential unit to core because the pressure in residential unit is higher than core. In brief, the dispersion of food odor is happened in the floors that are under the middle floors. In this study, 3 floors which have big difference of the air pressure were selected, and the amount of air in/outflow of residential unit, core, elevator shaft, and a stair hall were calculated. Under the condition that air inflow(q_{US}) is 1369 m^2/h and air outflow(q_{UR}) is 1269 m^2/h, 1761 m^2/h of air flows from residential unit to exterior wall(q_f), 654 m^2/h of air flows from residential unit to core(q_{uc}), 1535 m^2/h of air flows from core to elevator shaft(q_{ce}), 105 m^2/h of air flows from core to a stair hall, and 120 m^2/h of air flows to other places(q_{ct}). There is no information of density of food odor (C_u) which is included in airflows from residential unit to core (q_{uc}) yet.

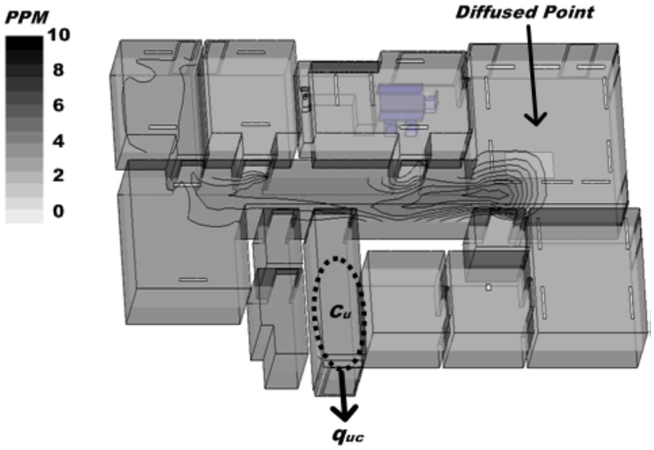
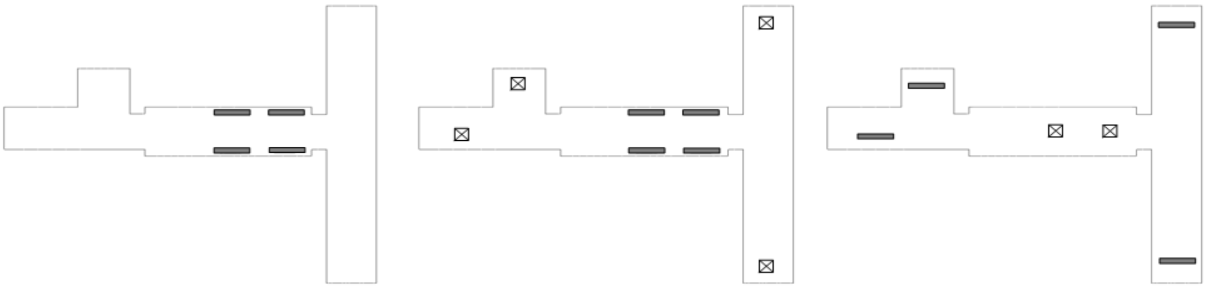


Figure 3. Concentration distribution of the unit

Table 3. Calculation condition of Steps 3 and 5

	STEP 3	STEP 5
Model	Standard k-ε High Reynolds	Standard k-ε High Reynolds
Mesh	700,000	300,000
Wall	Wall function	Wall function



Cases 1 and 2 *Case 3* *Case 4*

Figure 4. Cases according to the location of air in/outlet

Step 3: odor concentration C_u

Dispersion of food odor was analyzed by CFD simulation with residential unit model of Step 3 in Figure 1. In this case, the odor density of the front door area according to dispersion of food odor can be set as C_u. The air inflow of 1369 m^2/h and the air outflow of 1269 m^2/h in residential unit were used as boundary conditions. Also, another boundary condition which were result of Step 2 were used; the amount of airflow of 1761 m^2/h from residential unit to exterior wall and the amount of airflow of 654 m^2/h from residential unit to core. Table 3 is calculating condition. Figure 3 is the result of density dispersion of food odor, and Cu around the front door is 0.495ppm in case of regular ventilation. Accordingly, 89 m^2/h air that has 0.495ppm density flows from residential unit to core. Even though the amount of air and density is small, the stench intensity is very high, so residents can be uncomfortable when they get in or out of the building.

Step 4: Optimal amount of ventilation q_{cs}

Minimum inflow(q_{CS}) that dilutes stench intensity from 1 to 0.00011ppm for each floor can be calculated by CONTAMW with maximum density C_u of a front door area which is the result of Step 3 when Tri-methyl amine($(CH_3)_3N$) is dispersed from residential unit to core. For example, in case of third floor of which pressure difference between residential unit and core were the biggest, the density of Tri-methyl amine ($(CH_3)_3N$) could be kept under the 0.00011ppm when 1000CMH air inflow (q_{CS}) and outflow (q_{CR}) were applied.
However, density of Tri-methyl amine ($(CH_3)_3N$) can be kept under the 0.00011with less than 1000CMH because the pressure difference is decreased as floor get higher until middle floor and the amount of airflow from residential unit to core is getting small.

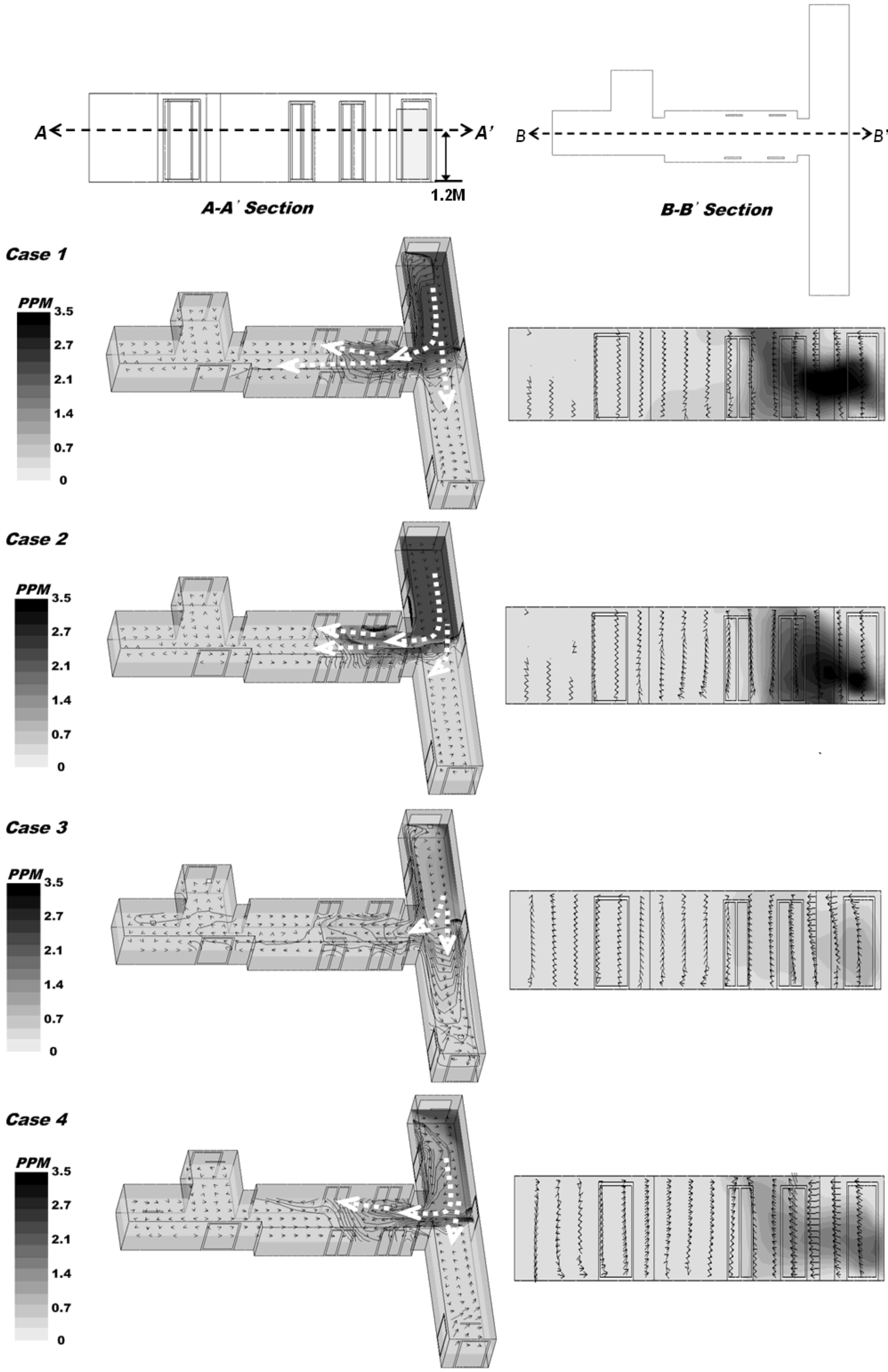


Figure 5. Concentration distribution of each case

With the same way, analyzing air inflows (0, 200, 400, 600, 800CMH) were applied for each floor. After all, to keep the density of Tri-methyl amine($(CH_3)_3N$) under 0.00011ppm, the minimum air inflow were analyzed as 800CMH from 3th floor to 17th floor, 300CMH from 19th floor to 20th floor, 500CMH from 21th floor to 27th floor, and 800CMH from 28th to 37th .

Step 5: location of inlet and outlet
To maximize the efficiency with calculated air inflow (q_{CS}) and air outflow (q_{CR}), the location of in/outlet has to be decided. First of all, Figure 4 is the schemes of ventilating system according to the location of air in/outlet. Four kinds of cases were analyzed: Case1 was about existing scheme, Case 2 was about Case1 with optimal amount of ventilation, Case 3 was about Case1 with air inflow in front of elevator, and Case 4 was about Case1 with air outflow in front of elevator. Figure 5 is about the result of CFD simulation. Case1 shows that the food odor from residential unit was dispersed not only to elevator area but also to stairwell area. Case2 shows that the pollutants were not diluted and dispersed to central core area even though an optimal amount of ventilation was applied. However, Case3 shows the pollutants were diluted by air outlet that is in front of residential, and then the food odor was blocked by airflow which was generated by line air inlet in front of elevator. Finally, Case4 shows that the rest of food odor was dispersed to the core area due to airflow that was made by the air inflow even though the pollutants were diluted by air inflow in front of the residential unit. As a result, the optimal ventilating system was that ventilating system that had air inflow in front of elevator and air outflow in front of residential unit.

RESULTS

This study was about dispersion analysis of food odor which was a serious problem in high-rise residential building in terms of stack effect. Moreover, it was showed the method how to calculate the optimal amount of air inflow and to decide the location of air in/outlet.
As a result of object building, the pressure difference between residential unit and core was big during winter. In case of the floors which were under middle floors, the air flew from residential unit to core because the pressure in residential unit is higher than core. Even though the density of food odor that goes to core area is very small, other residents can be uncomfortable because the stench intensity is very high. Furthermore, the food odor can be dispersed according to the location of air in/outlet even though the amount of ventilation is enough to dilute the food odor.

DISCUSSION

This study brings the stack effect during winter into focus because the winter has the great difference between indoor and outdoor temperatures in South Korea. This difference increases the influence of the stack effect on the airflow between spaces. Also, a phase of stack effect during summer appears in different way from that of the stack effect during winter. In case of summer, the influence decrease because of the small difference between indoor and outdoor temperatures. The air pressure of core, which is above floor from middle floors-neutral pressure level, is lower than residential unit, so the air flows from residential unit to core. Also, in the floors under middle floors, the air flows from core to residential unit because the pressure in residential unit is lower than core. Unlike winter, the dispersion of food odor is happened in the floors that are above the middle floors. Therefore, in hot climate region, this odor problem is less likely to occur because of the small stack effect. Also, intensity of food odor is carefully studied with relation to the human behavior in the residential buildings.

ACKNOWLEDGEMENT

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Survey of ventilation systems in Europe

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SUMMARY

Improving ventilation performance both for energy and IAQ creates a trend of installing more heat recovery supply and exhaust systems in new buildings and controlling better the airflows. Many studies have shown the impact of improved ventilation system on achieved performance but information about the market status and the difficulties in installing for instance supply ducts in collective dwellings are still lacking.

This studies gives the results of a survey in 6 European countries on ventilation systems installed in new and old building and gives an overview of the market of the different technologies, different of practice in different countries and how the problem of collective ducts, both for ventilation and combustion appliances, have been dealt in these countries.

INTRODUCTION

The study is to determine a state of the art of ventilation in Europe in order to know which systems are installed, in detail, and understand better the constraint and barreer to the implementation of supply and exhaust systems (i.e. installing supply ducts…).

METHODS

The study has been made through a survey. The questionnaire was asked to several European experts in ventilation who answered on the subject for their own country. These experts were from Finland, UK, Netherlands, Denmark, Italy, Poland and France. The survey was conducted in 2006.

RESULTS

The first part aimed at determining the ventilation systems installed in houses and collective dwellings in each country interviewed. Table 1 shows results for new buildings and Table 2 for the existing stock. This panorama has shown that :

- Scandinavian countries are for a long time using whole-house mechanical ventilation systems but in the existing stock, there are still buildings without ventilation or local ventilation (wet rooms)
- Netherlands used to have a lot of natural ventilation but since the energy regulation of 1998 are developing mechanical supply and exhaust systems
- UK used a lot of trickle fans ventilation but has recently renovated its regulation on ventilation showing some trend to both improve performance of trickle fans and start to implement whole-house ventilation

- Italy uses a lot of local ventilation by fans and airing
- Poland has a lot of natural ventilation with collective ducts (90%)
- France is using mechanical exhaust ventilation systems in new buildings and whole-house ventilation is mandatory since 1969.

Table 1 : ventilation systems in new buildings

	Finland	UK	NL	Denmark	Italy	Poland	France
supply and exhaust	90% (house), 30% (coll)	<1%	60%		Yes	5%	5%
exhaust only mechanical ventilation	10% (house), 70% (coll)	<1%	40%	flats	Yes, env 5%(5)	7%	95%
Natural ventilation	0	40%		houses	No	87%	0%
Local ventilation	0	20% (1)			Yes	1%	0%
Airing (window)	0	100% (3)			most common	-	0%

Table 2 : ventilation systems in existing building

	Finland	UK	NL	Denmark	Italy	Poland	France
supply and exhaust	30% (house), 5% (coll)	<1%	10%		Few	1%	1%
exhaust only mechanical ventilation	30% (house), 75% (coll)	around 10%	50%	flats <15 year (4)	Yes	5%	40%
Natural ventilation	<40% (house (1)), 20 (coll)	2%	30%	houses	No	93%	19%
Local ventilation	<10% (house (2)), 0% (coll)	20% (1)	10%		Yes	1%	30%
Airing (window)	0	100% (3)		flats >15 year (4)	most common	-	10%

(1) generally, kitchen hood and fan
(2) generally fan in bathroom, hood+fan in kitchen
(3) airing through window is always used for boost ventilation
(4) over 15 years, no ventilation in dwellings
(5) generally airing through window but more texts show the need of mechanical ventilation

Table 3 : main characteristics of mechanical systems installed

	Finland	UK	NL	Denmark	Italy	Poland	France
average airflow in dwelling	216 m3/h	50-100 m3/h	around 1 ach	Min 0,5 ach	generally 0,5 ach, in some areas 0,35	100-150 m3/h depending on energy and baths	50-140 m3/h average (depends on rooms)
Supply & exhaust HRU / temperature ratio	50% (crossplates, some wheels)		90% counterflow		50% Aluminium	crossflow	90% for new systems (counterflow) , 65% (crossflow) for old
Duct	rigid (flexible max 0,5m with kitchen hood)	flexible	rigid	rigid	rigid	flexible or rigid	flexible (house) or rigid (coll)
Fan specific Power exhaustP W/(m3/h)	0,3 exhaust only, 0,7 supply & exhaust	standard	60-80 W both with ECM, 60-120W old supply & exhaust; 60W exhaust	0,55, soon 0,28 *	0,86	200-400 W supply & exhaust, 300-500 W exhaust for houses, 1,5 to 2,5 kW in collective exhaust	120 W old, 70 W new
control		with light and tempo in toilets, manual for hoods	3 speed	variable speed (humidity) with pressure control or constant	variable speed (humidity) with pressure control or constant	constant	2 speed

From the point of view of the systems performance, we can note that:

- High efficiency heat recovery is used in some countries for new buildings but cross-plates exchangers are still very common
- Fan specific power varies a lot from one country to another ECM motors tends to be used in Scandinavia and Netherlands, more recently in France. Fan control (number of speed..) varies a lot depending of regulations and common practice
- There is no specific constraint on ventilation systems in these countries. In Netherlands though, in collective dwellings, taking care of maintenance (easy to use, friendly…) is required when retrofitting. There is nothing specific about the constraints of installing supply duct when retrofitting ventilation systems transforming them in supply and exhaust systems with HRU.

Table 4 : maintenance and cleaning

	Finland	UK	NL	Danemark	Italy	Poland	France
is it done ?	No, around 5 to 10%	No	No	No	No	No	No
is it mandatory ?	Yes, since 2000	recommended	recommended, but rare	rare	recommended/ mandatory	recommended	recommended
Price is	moderate	low	high (because conception has not dealt with an easy maintenance)	low because little is done, 1000 DKK/flat/year (150 €/flat/year)	high	low	moderate
Cleaning is	recommended (visual inspection	recommended (but not done)	rare	rare	rare/mandatory	recommended	recommended
Price is			high (access difficulties)		high	low	moderate
How is it done	by professionnals (ducts and terminals)		robots and sweeping	fan is disconnected and very high depression applied		natural ventilation – sweeping ; mechanical ventilation – sweeping and chemical	sweeping, robots, chemical...
control procedure	possible voluntarily	No	only for filter replacement + recommandations on hoods not connected to ventilation	cameras	No/Yes	No	No, use of cam voluntarily

Finally, the survey studied the maintenance and cleaning. Answers showed uniformly that maintenance and cleaning are not done or very seldom, although it is highly recommended (and mandatory in Finland). We can mention all the actions in Finland about visual inspection and inspections [1].

DISCUSSION

Recent reinforcement of energy regulations in buildings are inducing two trends :

- An increase of supply and exhaust systems with HRU in Scandinavia and Netherlands to reduce heat losses due to ventilation,
- Retrofits of airing and local ventilation systems in wet rooms in whole-house ventilation for all rooms in dwellings. The object of these retrofits is both to avoid condensations when building tightness is improved and to control air flows.

Yet the survey has shown that high efficiency systems are still used mainly in new buildings and the trends of retrofitting are not very large. Existing stock is still linked to older and lower performance systems. Maintenance and cleaning are judged insufficient and rarely done in all countries interviewed.

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We are grateful to all experts who answered the survey

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The performance of ventilation systems in apartment buildings

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SUMMARY

In Finland 12 apartment buildings were investigated as a part of European project HOPE.. There were one building with passive stack ventilation with fan assisted exhaust air ventilation, four buildings with mechanical exhaust ventilation only and seven buildings with mechanical supply and exhaust air ventilation system. Altogether 600 questionnaires were returned. Modern passive stack ventilation system with separate exhaust fans in kitchen and bathroom did not assure sufficient ventilation even in March when ventilation rates and CO_2-levels were measured. Thermal comfort problems (too cold and draught) were common with passive stack ventilation and mechanical exhaust ventilation without any heating of outdoor air. The fresh air radiators gave the best satisfaction. Thermal comfort problems were common also during summertime due to overheating of apartments. The spread of environmental tobacco smoke was the most common indoor air related problem in apartments together with kitchen hoods performing not satisfactory.

INTRODUCTION

Ventilation systems in apartment buildings

Passive stack ventilation was used in Finland until mid 1960's when centralized and time controlled mechanical exhaust ventilation replaced it. Since that mechanical exhaust ventilation system was major solution until 2003 in the Finnish new production of apartment building. Since 1987 kitchen hoods and outdoor air vents in bedrooms and living rooms has been required. Time controlled centralized exhaust ventilation was replace since mid 1990's by constant exhaust from bathroom and WC with user controlled kitchen hood. From the beginning of 1990's centralized and decentralized mechanical supply and exhaust ventilation system penetrated in the Finnish market of the new production of apartment buildings.

Design air flow rates

Design exhaust airflow rates were almost unchanged between 1940 and 2003. The design exhaust airflow rates were; kitchen 20 l/s, bathroom 16 l/s, WC 8 l/s and cloth room 2 l/s. During the wintertime and nighttime those values could be reduced to 50-67 % of the design value. The total exhaust airflow rate was the same in a small apartment flat (30 m^2) as in a medium size apartment flat (70 m^2). Practically exhaust airflow rates were lower because no outdoor air vents were used until in the end of 1980's. After 1987 a minimum outdoor airflow (4 l/s per person) rate of outdoor or supply air via vents has been required in living and bedrooms. That was increased to 6 l/s per person in 2003 [1].

HOPE-project

The final goal of the project HOPE [2] was to provide the means to enhance the construction of energy-efficient buildings that are at the same time healthy, thus decreasing the energy use in buildings and consequently resulting in a reduction of CO_2 emissions from primary energy used for ventilation, heating and humidity control. Within the European research project HOPE, 71 office buildings and 97 residential buildings were investigated and using checklists addressing the building characteristics and questionnaires to occupants asking their perceived comfort (thermal, IAQ, visual and acoustical), health (SBS) and self estimated productivity. In Finland 8 office buildings and 12 residential buildings were investigated.

METHODS

Buildings were selected with the aim that at least fifty occupant questionnaires would be returned per building. In addition, energy consumption and building description information should be available. The buildings should have been in their current state of operation and occupancy for at least one year, there should be energy saving measures present in most of the building.

The evaluation of ventilation systems was made in 21 apartment buildings in southern Finland. 15 out of 21 buildings were new, 5 out 21 buildings had recently modernized ventilation system and one building in an originally conditions. Ventilation systems were; modern passive stack ventilation with user controlled exhaust fans, mechanical exhaust air systems (time controlled or user controlled) (with of without fresh air vents) (fresh air radiators or no heating of outdoor supply air), mechanical supply and exhaust ventilation (centralized or user separate ventilation units in each apartment).

Design outdoor ventilation rates were quite same in all buildings. All buildings were connected to district heating system and had hot water radiators below window. There were no gas stoves in kitchen. 80 % of apartments had kitchen hood with time or user control.

Buildings are described in Tables 1,2 and 3.

Table 1. Apartment buildings with simple mechanical exhaust ventilation.

Building code	Espoo	Helsinki4	Helsinki5	Helsinki7
Ownership	Private	Private	Private	Public
Year of construction	1975	1962	1960	1999
Last modification of ventilation system	Never	Cleaning, adjustment	1997	
Fresh air vents	No	Partially	Yes	Yes
Type of control	Time control	Time control	Time control	
Number of apartments	90	58	83	50
N. of floors	15	7	7	4
Main orientation	S	S and W	S and W	E and S

Table 2. Apartment buildings with advanced mechanical exhaust system.

Building	Helsinki6	Lahti1	Lahti2	Lahti4
Ownership	Public	Public	Public	Public
Year of construction	2000	1953/2001	1995/6	2001
Last modification of ventilation system		2001 new ventilation system		
Fresh air vents	Fresh air window	Window frame	Window frame	Fresh air radiators
Type of control	Exhaust fans in kitchen, WC and bathroom	User controlled kitchen hood	User controlled kitchen hood	User controlled kitchen hood
Number of apartments	38	54	94	83
N. of floors	4	9	8	8
Main orientation	E	E and S and W	E and W	SW

Table 3. Apartment buildings with mechanical supply and exhaust ventilation.

Building code	Helsinki1	Helsinki2	Helsinki3	Lahti3
Ownership	Public	Public	Public	Public
Year of construction	1996	1997	2000	1996
Type of control	User controlled	User controlled	Kitchen hood	Kitchen hood
Number of apartments	151	126	51	95
N. of floors	7	7	5	8
Main orientation	S and W	S and N	E and S	Several

Questionnaires

A questionnaire was distributed to each occupant (one version for adults, one version for children) and covered the following factors [3].

- Years in apartment, hours per day in apartment.
- Environmental comfort in summer and winter. (Temperature, air movement, air quality, light, noise)
- Opinions un heating and ventilation in the apartment.
- Effect of environment Heath, and on ability to curry out necessary work.

A household questionnaire was also distributed, one per apartment..

- Ages of household, length and status of tenancy.
- Window orientation.
- Window opening behaviour.
- Condensation and mold.

Performance of ventilation system was checked with checklist developed during the HOPE-project and with some airflow rate measurements and monitoring of CO_2-levels and air temperatures in bedrooms.

RESULTS

A questionnaire was distributed to all apartments in 21 buildings. The first sets of questionnaires were distribution in March 2003. Questionnaires were distributed to habitants by post. They were asked to fill questionnaires and returned in to post box in their building. Another set of questionnaires were distributed in May into apartments who did not returned the questionnaire in April. The last ones were returned after semester season in August 2003. Altogether 600 questionnaires were returned. The mean response rate was 52 %. The highest response rate 69 % was in building Helsinki6 and the lowest 26% in building Helsinki3. The median age of respondents was 49 years. The median age of habitants living in newest buildings was almost 20 years lower than in buildings built in 1960's and 70's. The proportion of answers by female habitant was 62 %.

Comfort evaluations

Results from the questionnaire are shown in table 4 as mean value of all buildings. The value 1 is most preferable value (comfortable or satisfactory) and 7 means very bad conditions (uncomfortable/unsatisfactory) except for air movement and thermal sensation for which the optimum value is 4, 1= too less air movement 7= too much draught.

Table 4. Comfort assessments in apartment buildings (N=600).

Parameter	Mean value	Standard deviation
Thermal comfort (winter)	3.08	1.70
Thermal sensation (winter)	4.15	1.35
Thermal comfort (summer)	3.43	1.16
Thermal sensation (summer)	2.86	1.70
Air movement (winter)	4.14	1.46
Air movement (summer)	2.99	1.11
Air dryness dry – humid (winter)	2.64	1.15
Air dryness (summer)	3.44	1.13
Air quality fresh – stale (winter)	3.39	1.44
Air quality fresh -- stale (summer)	3.43	1.41
Air quality comfort (winter)	3.08	1.45
Air quality comfort (summer)	3.15	1.50
Comfort overall (w)	2.68	1.50
Comfort overall (s)	2.44	1.36
Affects on health 1=very good impact 7=very bad impact on health	3.65	1.33
Ability to carry out work 1=helps a lot 7=hinders a lot	3.53	1.23

Thermal comfort and thermal sensation was less acceptable during summertime than during wintertime. Air dryness during the wintertime was the most uncomfortable indoor air factor. Air quality sensation and comfort did not differ between winter and summertime. Air quality was judged more fresh than stale. In general indoor climate affected more positively than negatively on health and ability to carry out home work.

Thermal sensation

In figure 1 are shown the distribution of thermal sensation during winter and summer (1= too cold, 4 = neutral, 7 = too warm).

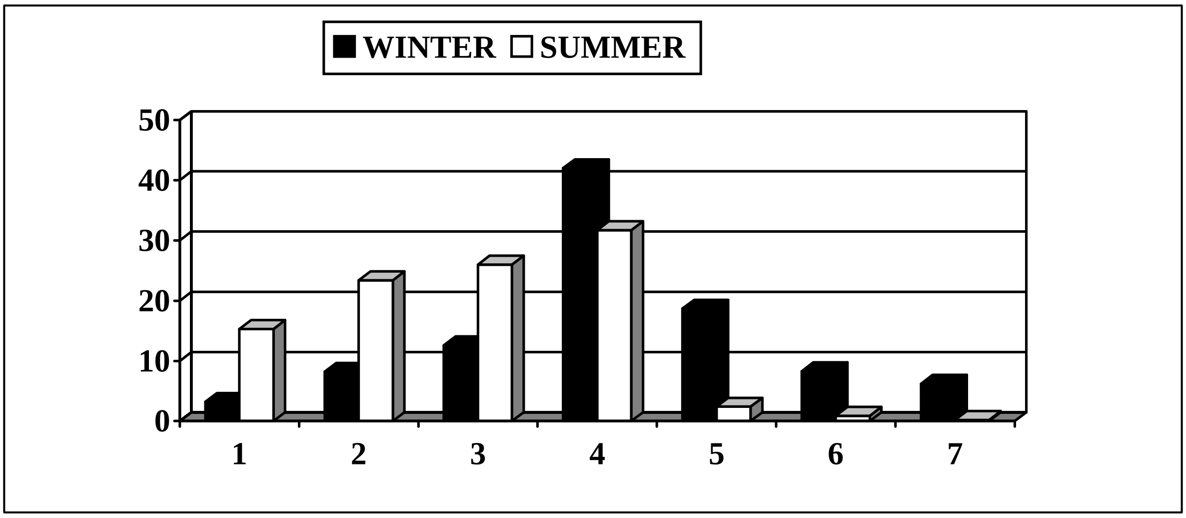

Figure 1.Thermal sensation among habitants during winter and summer (N=600).

Thermal sensations 3, 4 and 5 in the 7-point scale can be asses as acceptable. In that case 73 % of habitants evaluated their apartments as acceptable during winter and 60 % during summer. Even during winter 11.4 % of habitants evaluate room temperatures during winter as too high. During the summertime almost 40 % of habitants evaluate the room temperature too high. There were clear differences between buildings depending on orientation and use of balcony glasses. The worst case was in Helsinki6 were 55 % of habitants evaluated their apartment too hot during summertime.

Thermal comfort and type of ventilation system

Use of preheated (fresh air radiator or mechanical supply with heat recovery and pre heating) supply air improved thermal comfort during the wintertime compared with apartments with supply of unheated outdoor air. In Table 5 are shown statistically significant differences.

Table 5 Ratings of thermal environment and air movement in winter in apartment buildings with and without supply air heating.

Temperature (winter)	Unheated	Pre-heated
Comfort (1=best, 7=worst)	3.35	2.65
Thermal sensation 4=optimum	4.18	3.96
Temperature stability (1=stable, 7=unstable)	3.46	3.10
Air movement (winter)		
1= too less air movement 7= too much draught	4.28	3.72

Odors

As much as 67 % of the habitants did mention at least one source of odors in their apartment. Outdoor air and other apartments were the main source of odors. Other important sources of odors were WC or sewage system inside apartment, stairwell and kitchen. The main odor causing annoyance was tobacco smoke from balconies of neighborhoods. The next common odor source was food preparing but the prevalence was only half of the tobacco smoke.

Performance of ventilation systems

Most of buildings had well adjusted exhaust (supply) air flow rates. Only in building Espoo no adjustment of ventilation rates has been done. In building Helsinki3 with small ventilations inside apartments was a risk for too low air change rates if ventilation units are used with minimum speed.

Modern passive stack ventilation system with separate exhaust fans in kitchen and bathroom in building Helsinki6 did not assure sufficient ventilation even in March when ventilation rates and CO_2-levels were measured. In some bathrooms no ventilation was recorded and in some bedrooms CO_2-levels exceed 2000 ppm. Thermal comfort problems (too cold and draught) were common with passive stack ventilation and mechanical exhaust ventilation without any heating of outdoor air. The exhaust air flow rates were strongly affected by wind speed. Kitchen fans were insufficient to produce exhaust air flow rates required by the building code.

DISCUSSION

Thermal comfort problems were common also during summertime due to overheating of apartments. In several apartment buildings 50 % or more of habitants evaluated their apartment too hot during the summertime.

The fresh air radiators and mechanical supply and exhaust ventilation system with heat recovery gave the best satisfaction against draught. The spread of environmental tobacco smoke was the most common indoor air related problem in apartments together with kitchen hoods performing not satisfactory (too low exhaust air flow or low capture efficiency).

Buildings having hybrid ventilation system should have some kind alarming system showing if CO_2-levels in bedrooms of relative humidity in bathroom are too high.

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Field Studies on the Improvement of Indoor Air Quality by Installing Ventilation System in Apartment Houses

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SUMMARY

Ventilation system is being recommended as an effective tool to improve indoor air quality (IAQ) in apartment houses. Recently, in Korea, apartment house suppliers are required by law to establish ventilation systems in apartment houses. In this study, improvement of IAQ by establishing mechanical ventilation system in apartment houses was investigated through mock-up tests. Seven apartment houses were arranged for the test and concentrations of VOCs and Formaldehyde were examined under various conditions of ventilation rates and duct works. The results of this study show that IAQ of the test houses with mechanical ventilation system was improved from about 30% to 40% compared with IAQ in the apartment houses without ventilation system. However, there were no apparent concentration differences between the cases of changing ventilation rates and ductworks.

INTRODUCTION

Indoor air quality has been deteriorating from bad to worse with a trend towards tighter building triggered by the improvement of technology, and also by increasing use of finished materials containing chemical substances. In Korea, formaldehyde and VOCs are being recognized as very harmful materials to our health, and the law that apartment house suppliers must establish ventilation systems in their apartment houses was enforced on February 13, 2006. However, since there have not been enough studies on improvement of indoor air quality, the reality is that reference data to construct an effective ventilation system is still very insufficient.

In this study, Full scale mock up tests were carried out in order to get a full grasp of improvements of IAQ by establishing mechanical ventilations systems in apartment houses. IAQ was investigated from the point of concentrations of formaldehyde and VOCs. Mock-up tests consist of the improvement of IAQ with changing ventilation rates and ductwork types.

ORGANIZATION OF MOCK-UP HOUSES

Descriptions of examined houses are given in Table 1. Seven apartment houses arranged for the test were constructed with the same interior materials within the same period. One of the seven houses, named ‘NH’, remains blank without a ventilation system, so as to provide comparative criterion for result analysis.

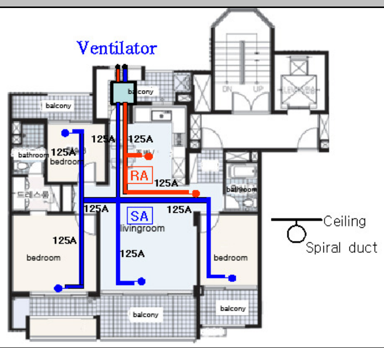
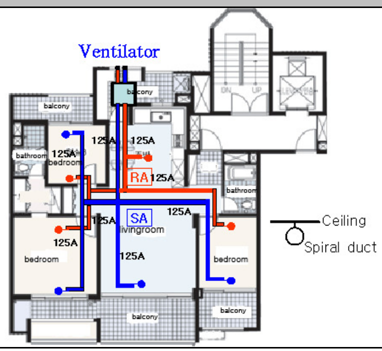
Table 1.Briefs of test houses

Contents	Conditions	Contents	conditions
Site Floor area/height Temperature	Gyeonggi-do, South Korea $85m^2$, 2.3m 20℃ (Ondol system)	Floor Humidity	2nd ~ 5th floor A home humidifier (no adjusting humidity)
Contents	Type	ACH	Ductwork
Ventilator type	No heat exchanger (NH) Total heat exchanger (TH :0.3) Total heat exchanger (TH :0.5) Total heat exchanger (TH :0.8) Total heat exchanger (THIR :0.5)	- 0.3 0.5 0.8 0.5	- Standard* Standard* Standard* Individual room SA &RA

* Ductwork of the standard type is the same as the standard type in Table 2

To examine improvements in IAQ by changing the ventilation rates, total exchanger-type ventilators were installed in 3 houses with respective air change rates of 0.3, 0.5 and 0.8 ACH. To compare and analyze improvements of IAQ caused by different types of ventilators, each total heat exchanger, sensible heat exchanger and alternating current exchanger was installed in each house with the air change rate fixed at 0.5 ACH.
As shown in Table 2, in one of the houses, the supply diffusers and exhaust diffusers were installed in each room, whereas in another house the air supply diffusers were installed in each room while the air exhaust diffusers were installed in the kitchen and near the entrance to examine the improvement of IAQ resulting from different ductworks. Ventilation systems installed in these houses have a total heat exchanger of 0.5 ACH.
Once installations of ventilation systems in each house were done, infiltration rates in two houses were measured to examine the difference in infiltration rates which could affect IAQ. For the test of infiltration rates, constant concentration method was used with SF_6 tracer gas. The result of this test shows a difference of 0.1ACH, as 'NH' house and 'TH : 0.5' house had respective results of 0.75 ACH and 0.65ACH. Although the number should not be ignored, no measure was taken to correct the difference, as it was accepted as a cause of possible error in analyzing IAQ thereafter. The next step was executing TAB(Test, Adjustment and Balancing) which is to control air supply and exhaust rates so that they correspond accurately to determined conditions in each residence.
Indoor temperature was controlled within a range of 20±2℃ by the Ondol system which is Korean traditional floor heating system. When running a ventilation system, it is difficult to

Table 2. Ventilator and Ductwork established in test houses

<table>
<tr><th></th><th>Total heat exchanger and sensible heat exchanger
(standard)</th><th>Total heat exchanger
(Individual room SA & RA)</th></tr>
<tr><td>Ventilator and ductwork</td><td>
</td><td>
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</table>

reach a relative humidity of 50% in indoors, but since the general condition is that a regular house only runs one humidifier, only one humidifier was installed but was run at maximum capacity.

METHODS

Concentrations of formaldehyde and VOCs were measured after operating the ventilation systems. General information and schedule regarding the measurement of the indoor air quality are given in Table 3. Formaldehyde was sampled by using DNPH cartridge for 30 minutes at sampling speed of 500ml/min and VOCs was done by using Tenax tube for 30 minutes at 100ml/min. The measurement started about 2 weeks after the interior construction was completed.

The status 'Ventilator OFF' is the case where the ventilator was always operating except for the day of indoor air sampling, whereas 'Ventilator ON' is the case where the ventilator was operating everyday including the day of sampling.

Table 3. Measurement conditions and schedule

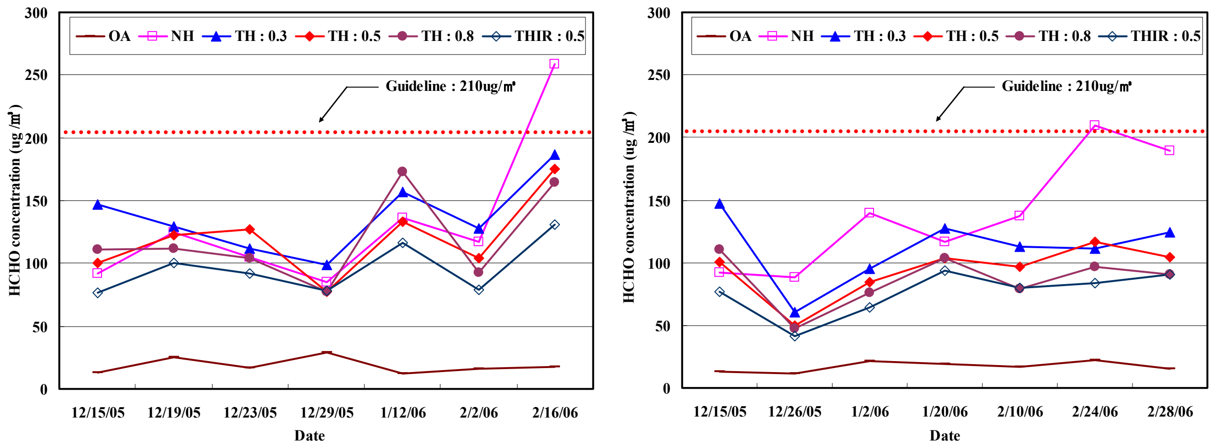
Contents	Conditions
Duration	Dec. 15. 2005 ~ Feb. 28. 2006
Measurement conditions	'Ventilator OFF' : ventilation system was turned on everyday besides the day of sampling the indoor air (date : 12/15/2005, 12/19, 12/23, 12/29, 1/12/2006, 2/2, 2/16) 'Ventilator ON' : ventilation system was turned on everyday inclusive of the day of sampling the indoor air (date : 12/26/2005, 1/2/2006, 1/20, 2/10, 2/24, 2/28)

RESULTS AND DISCUSSION

During the period of measurement, the indoor temperature was under control within a range of 20±2℃ while humidity was maintained differently due to difference in ventilation rates. Change of formaldehyde concentration in both cases (ventilator On/Off) during the measuring period is shown in Figure 1. In both cases, Korean guideline of concentration of formaldehyde (210 $\mu g/m^3$) was mostly satisfied. However, the concentration did not diminish as time went by, but it rather increased gradually. We believe this is due to the gradual increase of emissions coming from the inside of finished materials to the outside, which were constructed in compound.

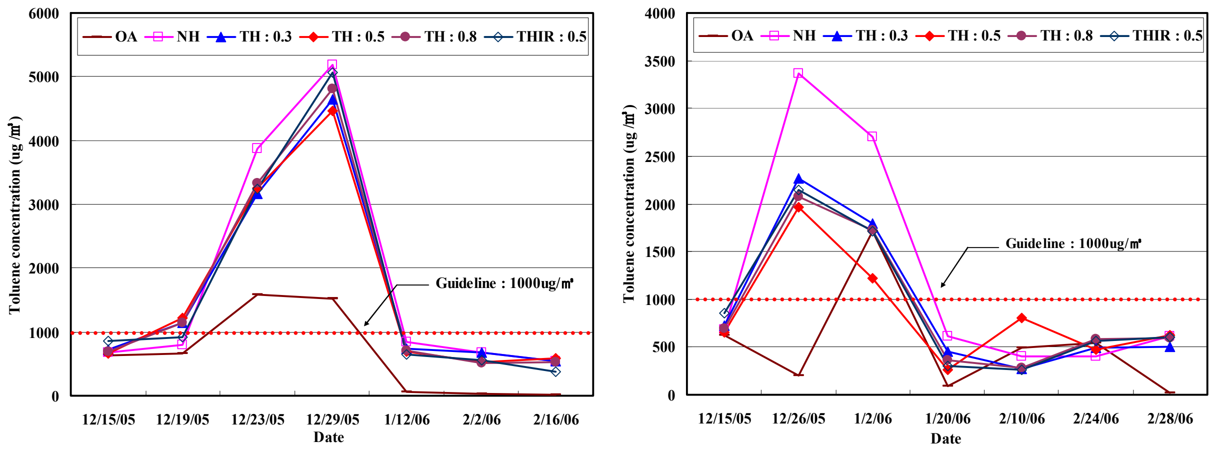
In case of individual VOC, toluene concentration was the highest, and while reduction tendency of other VOC materials were similar to that of toluene, this paper will only cover the analysis on the change of toluene.

The concentration changes of toluene in both cases of 'ventilator OFF/ON' are shown in Figure 2. In case of 'ventilator OFF', the toluene concentration increased largely on Dec. 29; on Jan. 12, which is 1 month after the measurement initiation, it shows a dramatic decrease below the Korean guideline value of 1000 $\mu g/m^3$. From then on, the reduction becomes more gradual. In case of 'ventilator ON', the concentration reaches its peak on Dec. 26 and shows the distribution that satisfies the Korean guideline value from Jan. 20, which is a month after the initiation.



a) Ventilator OFF b) Ventilator ON

Figure 1. Time series HCHO concentration.

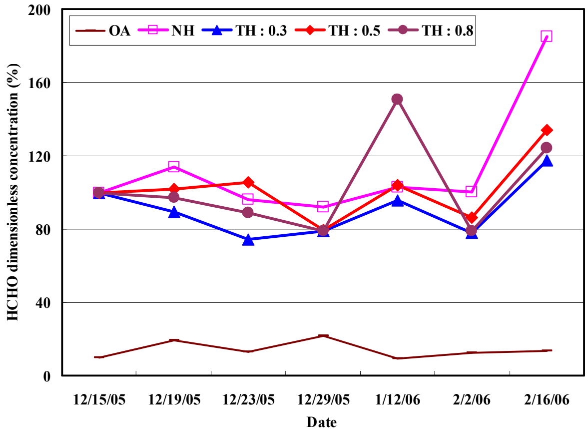


a) Ventilator OFF b) Ventilator ON

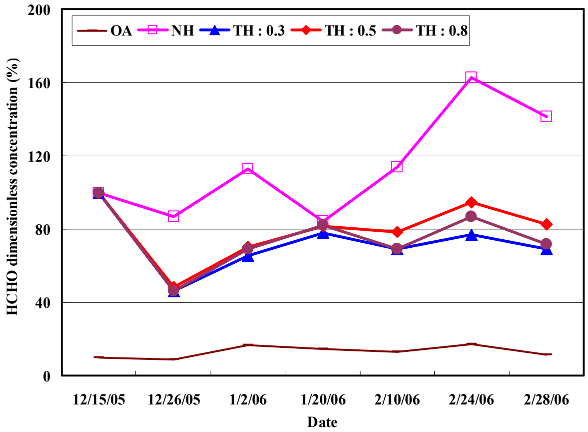
Figure 2. Time series Toluene concentration.

The improvement effect of indoor air quality caused by varied ventilation rates

To compare and analyze the improving effects of indoor air quality among houses, we need to correct the conditions of different temperature and humidity at the time of measurement. We made the corrections according to the formula of INOUE. Also, as initial concentrations differ in houses, as shown in Figure 1, it is difficult to simply compare the diminishing effect of concentration within the time period or the improvement effect of indoor air quality among houses. Therefore, to eliminate the differences in the initial concentration, concentration values earned with INOUE's formula were made dimensionless by the concentration of Dec. 15, and these numbers were multiplied by 100 to get percentage values for comparison and review.
Dimensionless formaldehyde concentration in ventilation rates of 0.3, 0.5 and 0.8ACH is shown in Figure 3. a) of Figure 3 shows the case of 'ventilator OFF', where in the beginning of measurement the house without the ventilator does not show apparent difference in formaldehyde concentration, compared with the case where the ventilator is installed. However, on Feb. 2, 1 1/2 months after initiation, the indoor air quality improved by about 20% and on Feb. 16, 2 months from then, the result went on to improve by about 30%. This

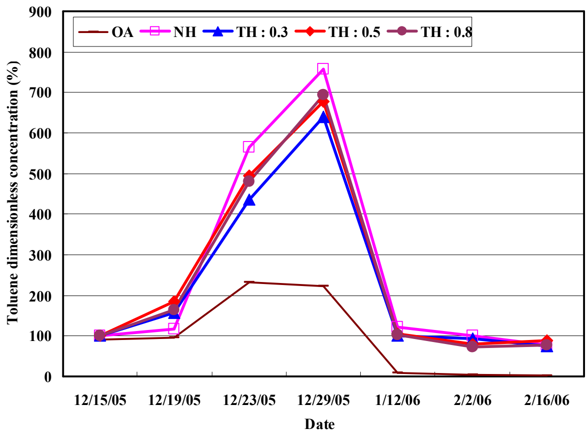


a) Ventilator OFF

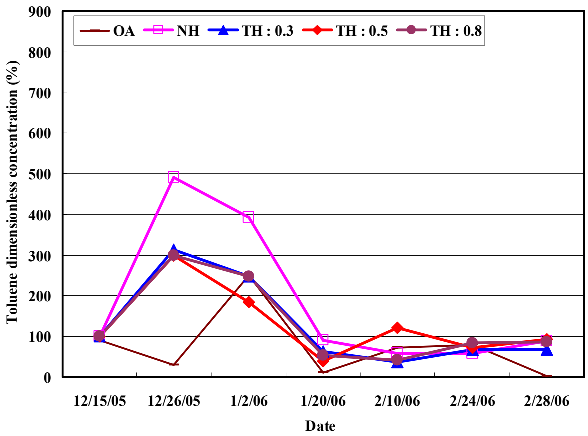


b) Ventilator ON

Figure 3. Comparison of HCHO dimensionless concentration with varied ventilation rates.



a) Ventilator OFF



b) Ventilator ON

Figure 4. Comparison of Toluene dimensionless concentration with varied ventilation rates.

improvement is assumed to have been gained by the all-time operation of the ventilators. In terms of the indoor air quality improvement earned by the change of ventilation rates, the case of 0.3ACH shows a rather lower result in formaldehyde concentration than those of 0.5and 0.8ACH. However, the difference shown (5%) is thought to be in the range of measurement error, therefore, the improving effect of indoor air quality caused by ventilation rates does not show any significant difference.

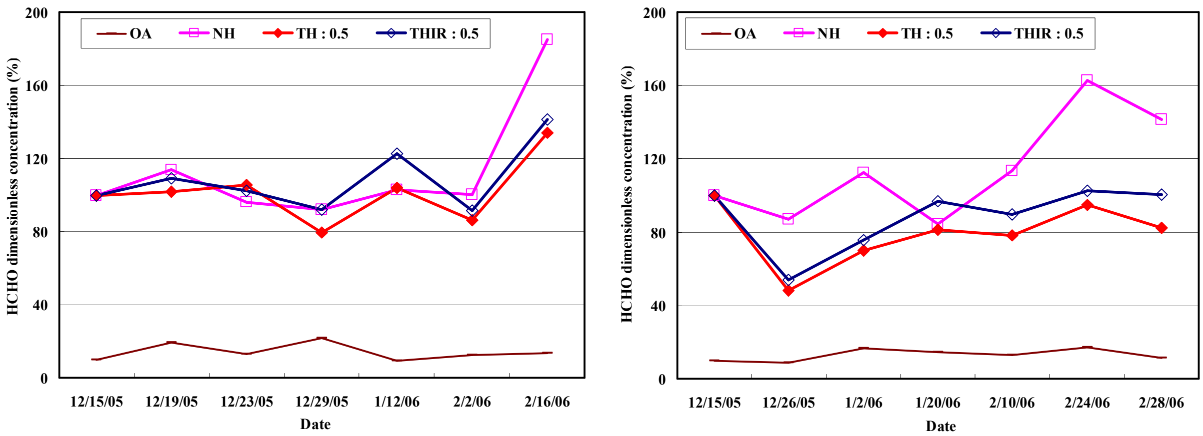
In the case of ‘ventilator ON’, an average of 40% improvement is shown with the exclusion of Jan. 20. However, the difference (5%) caused by ventilation rates is thought to be within the range of measurement error. The following is the presumed cause: since the infiltration rate during the time period (winter time) is about 0.7 ACH, which is relatively high, ventilation rates of 0.3 ~ 0.8ACH became relatively low. Moreover, construction error could have played a role.

In the case of toluene concentration, corrections were not made to the temperature and humidity, but toluene concentration was made dimensionless by the concentration of the first day. Figure 4 shows the dimensionless toluene concentration in both cases of 'ventilator OFF/ON'. From the starting date throughout Dec. 29, which is after half a month, toluene concentration shows relatively high numbers, and the ‘ventilator OFF’ case shows a 15% improved indoor air quality compared with houses without installation. This is considered to be caused by the all-time operation of the ventilator. However, on Jan. 12, which is a month after initiation, the reduction slowed down and improvement caused by the all-time operation

is no longer much effective. Comparing the difference caused by ventilation rates, the case of 0.3 ACH shows numbers that are about 5% lower than the others, but the difference is thought to be insignificant and caused by measurement error.
In case of 'ventilator ON', we will only take into account measured results from Dec. 26, Jan. 20 and Feb. 2, while excluding the remaining dates when pavement construction of the peripheral roads took place and caused high outdoor air concentration.
When concentration was highest on Dec. 26, toluene concentration in houses without installations surged by 500% compared with the initial numbers, whereas in the houses installed with ventilation systems, 300% increase was measured regardless of ventilation rates, thus showing a 40% indoor air improvement. However, from Jan. 20, a month after initiation, the improvement of IAQ caused by operation of the ventilator gradually decreased.

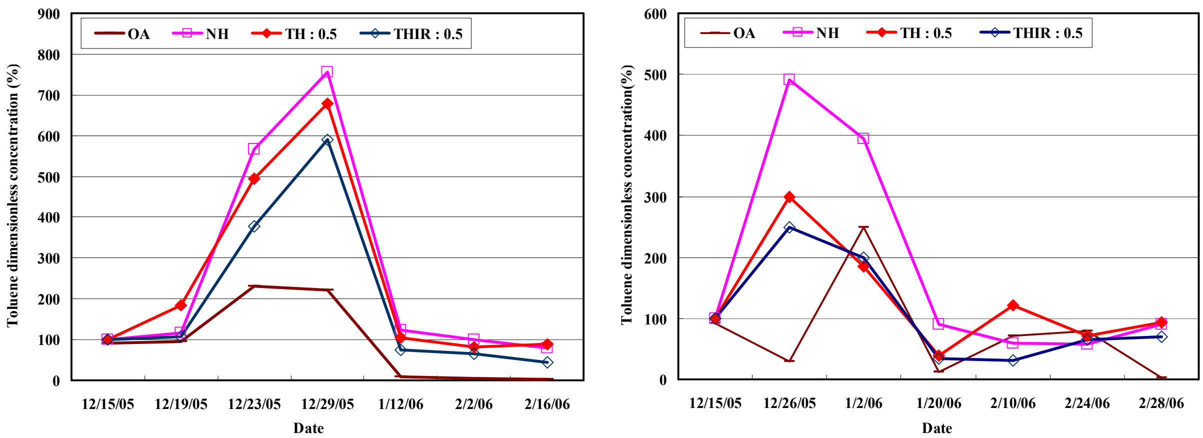
Indoor air quality improvement caused by varied systems of air supply/exhaust duct

Ductwork of ventilation system are classified by 1) whether the air is supplied to each room and exhausted from the kitchen and near the entrance, and by 2) whether the air is supplied to each room and exhausted from each room, and the following shows the indoor air quality improvement caused by varied types of ducts. The dimensionless concentration of formaldehyde is shown in Figure 5.



a) Ventilator OFF b) Ventilator ON

Figure 5. Comparison of HCHO dimensionless concentration by the ductworks



a) Ventilator OFF b) Ventilator ON

Figure 6. Comparison of Toluene dimensionless concentration by the ductworks

In case of 'ventilator OFF', the reduction of formaldehyde concentration is not very apparent until Feb. 2, which is 1 1/2 months after measurement initiation, but afterwards, a slightly diminished concentration is noticeable. On Feb. 16, 2 months after initiation, houses with air supply and exhaust ducts show a 25% improvement compared with houses without ventilator installations. The difference caused by different types of air supply/exhaust is within 5% and is insignificant.
In case of 'ventilator ON', the houses where the air is supplied to each room and exhausted from the kitchen and near the entrance showed a 40% improvement, while houses where the air is supplied to each room and exhausted from each room showed a 30% improvement, showing a 10% difference. However, this is within the range of error, and it is speculated that there is no apparent difference in the diminishing effect of formaldehyde concentration caused by varied duct systems of air supply/exhaust.
Dimensionless concentration of toluene is shown in Figure 6. In case of 'ventilator OFF', houses where the air is supplied to each room and exhausted from each room shows a 25% improvement compared with houses without ventilator installation until Dec. 29, 1/2 month after initiation, while houses where air is supplied to each room and exhausted from the kitchen and near the entrance shows a 10 % improvement.
In case of ventilator ON, checking mainly the results from Dec. 26 when outdoor air concentration was relatively low, improvement of indoor air quality is 50% and 40% respectively for houses where air is supplied to each room and exhausted from each room, and houses where air is supplied to each room and exhausted from the kitchen and near the entrance. However, the 10% difference in improvement effect of indoor air quality caused by varied duct systems of air supply/exhaust is thought to be insignificant.

CONCLUSION

This research aimed to run mock-up tests in order to examine the indoor air quality improvement effect of mechanical ventilation systems in apartment houses. The result only reflects characteristics and field conditions that were applied in the mock-up test, and does not show general characteristics of every ventilation system.
The result of this research data can be summarized as followings:
1) When a ventilation system with 0.3~ 0.8ACH is operated in an apartment house, formaldehyde and VOCs concentrations are reduced by 30~50% compared with those without ventilation systems.
2) When ventilation rates are in the range of 0.3~ 0.8ACH, the difference of indoor air quality improvement effect caused by ventilation rates is insignificant.
3) In terms of indoor air quality improvement caused by different types of supply and exhaust duct systems, there was no significant differences between them.

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Application of The Exergy Concept to Ventilation Using Heat Recovery from Exhaust Air

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SUMMARY

This paper describes steady-state energy and exergy analyses on an instantaneous and on a daily basis for dwelling ventilation systems and domestic hot water DHW production systems. The ventilation uses mechanical exhaust with environmental air supply without heat recovery or balanced ventilation with heat recovery. The exhaust ventilation air is used to preheat DHW, using a heat exchanger or a heat pump. Energy and exergy demands in winter days for De bilt, the Netherlands, are presented at the component level, in terms of heat and electricity, for the systems. The results indicates that for these winter days the total energy demands, but not the total exergy demands, of balanced ventilation and DHW production with preheat by the exhaust ventilation air using a heat pump are lowest. The total exergy demands are dominated by exergy of electricity input to heat recovery unit and the heat pump.

INTRODUCTION

Exhaust ventilation air has been used to preheat inlet ventilation air in balanced ventilation systems and as a heat source in other heating systems. In order to recover heat from the exhaust ventilation air, electric power is required. Since electricity input is small relative to the amounts of thermal energy recovered, such systems are efficient from an energy viewpoint. One important yet often overlooked aspect, however, is the difference in ‘quality’ between the high-grade electricity input and the lower grade thermal energy recovered.

Exergy analysis provides a common basis for evaluating forms of energy (e.g. thermal and electric), considering their abilities to produce work in relation to a given environment [1-3]. Exergy recognizes that energy carried by substances can only be used ‘down’ to the level given by the environment. Unlike energy, exergy is not subject to a conservation law.

This paper presents steady-state energy and exergy analyses on an instantaneous and on a daily basis for dwelling ventilation systems and DHW production systems. The exhaust ventilation air is used to preheat DHW. Analysis results, for De Bilt, the Netherlands, are used for discussion of design options for systems.

SYSTEM DESCRIPTION

The systems to be studied are divided into 2 parts: dwelling ventilation systems and domestic hot water production systems.

Dwelling ventilation systems

Energy and exergy analysis is performed for dwelling ventilation systems, using mechanical exhaust ventilation with environmental air supply without heat recovery (Figure 1a) and balanced ventilation with heat recovery (Figure 1b). The mechanical exhaust ventilation system uses a DC fan, Model: CVE ECO-fan 2 of Itho bv. [4]. The balanced ventilation system uses a DC Heat Recovery Unit, Model: HRU ECO-fan 3 S B of Itho bv. [5], containing two DC fans and a heat exchanger with high thermal effectiveness ε. Thermal effectiveness of the heat exchanger is calculated by interpolating from manufacturer's data [5] in relation to ventilation airflow rates (see Table 1). Figure 1 presents airflow rates and temperatures of ventilation and infiltration airflows of the systems. Air infiltration is also accounted for the analysis.

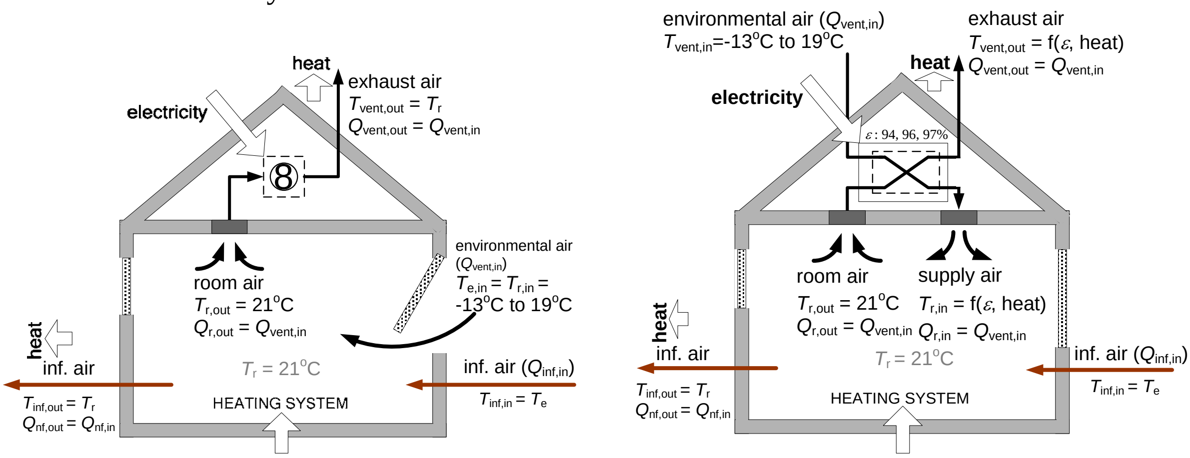


Figure 1. Dwelling ventilation systems. a) Mechanical exhaust ventilation with environmental air supply (left), b) Balanced ventilation with heat recovery (right).

The energy and exergy analysis of the ventilation systems assumes steady state operation and considers only dry air. Pressure difference between air entering and leaving the dwelling is also ignored. General calculation values are given in Table 2.

Table 1. Thermal effectiveness ε versus ventilation airflow rates Q_{vent}, for the DC HRU (left). Table 2. General calculation values (right). [6], [7]

Q_{vent} [m^3/s]	ε [–]
0.063	0.94
0.042	0.96
0.028	0.97

Parameters	Values
Air density (ρ_{air})	1.23 kg m^{-3}
Spec. heat capacity of air ($C_{p,air}$)	1.008 kJ kg^{-1}K^{-1}
Environmental air temperature (T_e)	from -13°C to 19°C
Room air temperature (T_r)	21°C
Ventilation airflow rates (Q)	0.028, 0.042, 0.063 m^3/s

Domestic hot water production systems

The DHW production systems use ground water at a constant temperature (assumed 13°C). The DHW enters the dwelling with/without preheat by the exhaust air from the dwelling ventilation using a heat exchanger or a heat pump. Electricity is needed for DHW preheat if using the heat pump. The thermal effectiveness ε of the heat exchanger and the coefficient of performance COP of the heat pump are assumed 0.90 and 3.7 respectively. The energy efficiencies of the DHW preheat devices are assumed 1.0. The DHW devices operate when temperature of the exhaust ventilation air is higher than the maximum temperature between the inlet water and environment (if using the heat exchanger) and environment (if using the heat pump). The exhaust ventilation air is cooled until a temperature corresponding to the thermal effectiveness ε (if using the heat exchanger) and until environment T_e (if using the heat pump). Air flow and water flow rates are assumed constant at the inlet and the outlet. Flow effects in the systems are also ignored in this study.

Daily operation profiles of winter days in De Bilt, the Netherlands

Figure 2 shows three hourly environmental air temperature profiles, for winter days of maximum, minimum and intermediate mean daily environmental air temperatures. Climate data for De Bilt, the Netherlands are taken from the TMY2 weather data [8].

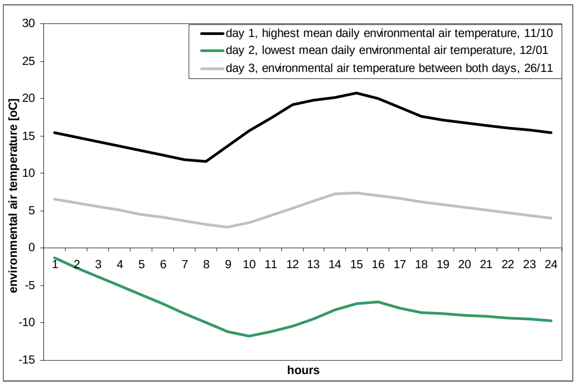


Figure 2. Hourly environmental air temperature profiles in 3 winter days in De Bilt.

The dwelling ventilation and the DHW use in the dwelling for the three days are hourly scheduled, given in Tables 3.

Table 3. Hourly operation plan: the dwelling ventilations (left); and the DHW use (right).

hour	ventilation airflow rate Q_{vent} [m^3/s]
0:00-8:00	0.028
8:00-9:00	0.042
9:00-17:00	0.063
17:00-18:00	0.042
18:00-24:00	0.028

hour	Q_{DHW} [m^3/s]	T_{DHW} [^{o}C]
00:00-07:00	0	0
07:00-09:00	0.00001	65
09:00-18:00	0	0
18:00-20:00	0.00001	65
20:00-24:00	0	0

ENERGY AND EXERGY ANALYSIS

The comparison of design options for combination between dwelling ventilation and domestic hot water production is based on calculation of thermal energy and thermal exergy demands by ventilation and infiltration airflows in relation to environmental air temperature, and electricity input to ventilation unit and to DHW preheating device. The energy and exergy analysis is carried out on an instantaneous basis where T_e is between -13^{o}C and 19^{o}C and on a daily basis for the representative winter days (Figure 2).

Energy and exergy demands by infiltration airflows

Infiltration airflow rate Q_{inf} is calculated based on NEN 2867 [9]. The infiltration airflow rate relies on types of ventilation systems: mechanical exhaust ventilation $Q_{inf,mv}$ (equation 1) and balanced ventilation $Q_{inf,bv}$ (equation 2).

$$Q_{inf,me} = \left(\frac{q_{v10}}{500}\right)\left((A + BCv + D\Delta T)q_{v10}\right) \qquad (1)$$

$$Q_{inf,bv} = 0.8\left((A + BCv + D\Delta T)q_{v10}\right) \qquad (2)$$

where q_{v10} [dm^3/s] is the infiltration airflow rate expressed in a volume airflow rate q_v with pressure difference of 10 Pa, ν [m/s] is the wind speed at 10 m above the ground level and ΔT [K] is the temperature difference between the room air T_r and the environmental air T_e. *A*, *B*, *C* and *D* are the coefficients of the effect from turbulent airflow, shielding, partitioning of the air infiltration concerning the dwelling coating, and temperature difference respectively.

The following values are applied for the study. q_{v10} is 150 dm^3/s for the dwelling with the mechanical exhaust ventilation and 80 dm^3/s for the dwelling with the balanced ventilation. *A*, *B*, *C* and *D* are 0.02, 0.25, 0.20 and 0.004 respectively. These coefficient values are according to NEN 2867 [9] and for dwelling whose volume and height are 250 m^3 and 5.4 m, and having a normal shielding. ν, T_i and T_e are taken from the TMY2 weather data [8].

The infiltration airflow rate Q_{inf} is used for calculation of thermal energy $En_{th,inf}$ and thermal exergy $Ex_{th,inf}$ demands by infiltration airflows, using equations 3 and 4.

$$En_{\text{th,inf}} = \rho_{\text{air}} Q_{\text{inf}} C_{\text{p,air}} (T_{\text{out}} - T_{\text{in}}) \tag{3}$$

$$Ex_{\text{inf,th}} = \rho_{\text{air}} Q_{\text{inf}} C_{\text{p,air}} (T_{\text{out}} - T_{\text{in}} - T_{\text{e}} \ln(\frac{T_{\text{out}}}{T_{\text{in}}})) \tag{4}$$

where T_{out} and T_{in} are temperatures of the infiltration air leaving and entering the dwelling respectively. All the air temperatures are in Kelvin.

Since energy demand by the infiltration airflows accounts only the $En_{th,inf}$, therefore the total demands by the infiltration airflows are equal to the thermal demands for energy and exergy.

Energy and exergy demands by ventilation airflows

Energy and exergy demands by ventilation airflows are obtained from thermal demand by ventilation airflows and electricity input to ventilation unit.

Thermal energy and exergy demands by ventilation airflows are calculated by applying equations 3 and 4. The supply air temperatures to the dwelling $T_{r,in}$ for the balanced ventilation system is calculated by using the exchanger heat transfer effectiveness equation [10] and heat balance equation, assuming the same airflow rates though the HRU for the supply air and the exhaust air.

$$T_{\text{r,in}} = T_{\text{e}} + \varepsilon (T_{\text{r}} - T_{\text{e}}) \tag{5}$$

The inputs of exergy and energy for electricity are identical [11]. The electricity *Pe* is calculated by using the fan manufacturer data and the fan law [10] in equation 6.

$$Pe_1 = Pe_2 (\frac{\phi_2}{\phi_1})^4 (\frac{Q_1}{Q_2})^3 (\frac{\rho_1}{\rho_2}) \tag{6}$$

where ϕ is the fan impeller diameter. The fan law is used to predict performance of the fan when test data (data with subscript 2 in equation 6) are available. The test data are used from the fan producer: Itho bv. [4], [5].

Energy and exergy demands by domestic hot water

Thermal energy and exergy demands by DHW flow are functions of the DHW temperature required to supply to the dwelling T_{DHW} (assumed 65^{o}C) and the preheated DHW temperature $T_{preheat}$. The demands are calculated by applying equations 3 and 4 and using inlet water temperature T_w as reference environment. $T_{preheat}$ depends on types of the DHW preheat devices and is calculated by using equation 7 (if using a heat exchanger, $T_{\text{preheat,ex}}$) and equation 8 (if using a heat pump, $T_{\text{preheat,hp}}$).

$$T_{\text{preheat,ex}} = \varepsilon \frac{C_{\min}}{C_{\text{w}}} \left(T_{\text{vent,out}} - T_{\max} \right) + T_{\text{w}} \tag{7}$$

$$T_{\text{preheat,hp}} = \left(\frac{\rho_{\text{air}} Q_{\text{air}} C_{\text{p,air}} (T_{\text{e}} - T_{\text{e,out}})}{\rho_{\text{w}} Q_{\text{w}} C_{\text{p,w}} (1 - \dfrac{1}{\text{COP}})} \right) + T_{\text{w}} \tag{8}$$

where $C_{\min}$ is the minimum value of the thermal capacities between the exhaust air from the dwelling ventilation and the inlet water, C_{w} is the thermal capacity of the inlet water and $C_{\text{p,w}}$ is the specific heat capacity of the inlet water (4.19 $\text{kJkg}^{-1}\text{K}^{-1}$). $T_{\max}$ is the maximum temperature between inlet water and environment. T_{w} is the inlet water temperature and $T_{\text{e,out}}$ is the air temperature of the exhaust ventilation air.

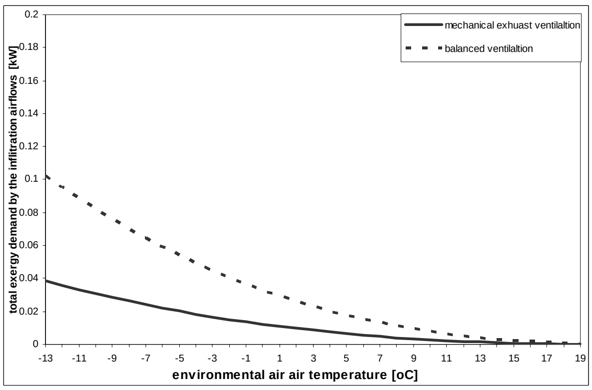
Electricity input to the heat pump is calculated as a different value between thermal energy supplied to heat pump by the exhaust ventilation air and thermal energy released by the heat pump to inlet water, because energy efficiency of the heat pumping process is assumed 1.0.

RESULTS

This chapter presents energy and exergy analysis results for the dwelling ventilation and the DHW production systems on an instantaneous basis and on a daily basis.

Total exergy demands by infiltration and ventilation airflows

Figures 3 and 4 show total exergy demands by the infiltration airflows Ex_{inf} and by the ventilation airflows Ex_{vent}, as a function of environmental air temperature T_{e}. In Figure 4, the mechanical exhaust ventilation with natural air supply and the balanced ventilation with heat recovery are called system A and system B respectively.



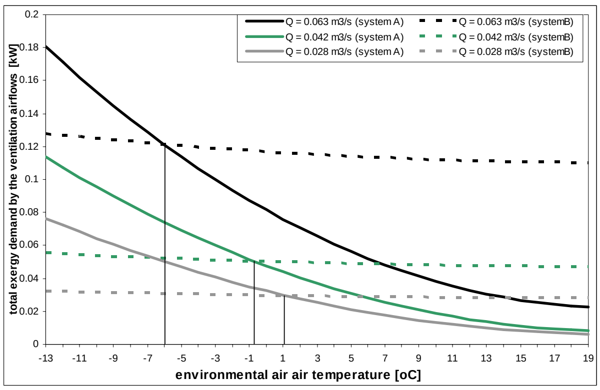


Figure 3. Total exergy demand by the infiltration airflow Ex_{inf} versus T_{e} (left).
Figure 4. Total exergy demand by the ventilation airflows Ex_{vent} versus T_{e} (right).

In Figure 3, the Ex_{inf} lines for the ventilation systems vary nonlinearly with T_{e}. They rely on logarithmic term of ratio between inlet and outlet infiltration air temperatures (see equation 4), since Ex_{inf} is assumed equal to $Ex_{\text{th,inf}}$. In addition, the Ex_{inf} lines for the balanced ventilation with heat recovery is higher than the $Ex_{\text{th,inf}}$ for the mechanical exhaust ventilation due to higher infiltration airflow rate in the entire range of T_{e}.

In Figure 4, the Ex_{vent} decreases for a given Q_{vent}, as T_{e} increases from -13°C to 19°C. The Ex_{vent} lines (smooth ones) for the mechanical exhaust ventilation using the DC fan (called DC fan line) are sharper than the Ex_{vent} lines (dashed ones) for the balanced ventilation with heat recovery using the DC HRU (called DC HRU line). The DC HRU lines are less sensitive in the range of T_e because the DC HRU requires mainly electricity, while the DC fan requires relatively more heat. The DC fan requires more exergy at lower T_e, and relatively less exergy as T_e increases towards the room air temperature T_r. electricity inputs to the DC fan and the DC HRU are given Table 4

Table 4. Electricity inputs to the DC fan and the HRU *Pe* versus ventilation airflow rates Q_{vent}

hour	ventilation airflow rate Q_{vent} [m^3/s]	electricity inputs ***Pe*** **[W]**
DC fan	0.028	6
	0.042	8
	0.063	22
DC HRU	0.028	28
	0.042	47
	0.063	110

The DC fan lines and the DC HRU lines intersect at a given environmental air temperature $T_{e,intersect}$ for a given Q_{vent}. At $T_e > T_{e,intersect}$, using the mechanical exhaust ventilation with the DC fan results in less total exergy demand than using the balanced ventilation with the DC HRU at the same airflow rate. $T_{e,intersect}$ increases when operating the ventilations at lower Q_{vent} since electricity input to the DC HRU is less.

Total exergy demands by domestic hot water flow

Figures 5 shows the total exergy demand by DHW flow with preheat by exhaust ventilation air using the heat exchanger $Ex_{DHW,ex}$ (Figures 5a) and the heat pump $Ex_{DHW,hp}$ (Figures 5b) in relation to T_e, respectively. The mechanical exhaust ventilation with natural air supply and the balanced ventilation with heat recovery are called system A and system B respectively. The thin lines show the total exergy demand by DHW flow without preheat, as a reference for comparison to the $Ex_{DHW,ex}$ and $Ex_{DHW,hp}$.

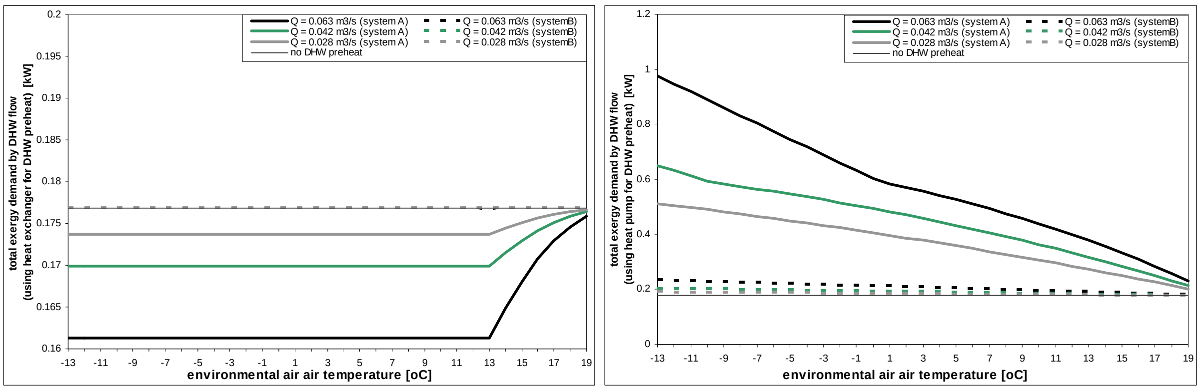


Figure 5. Total exergy demand by DHW flow with preheat by exhaust ventilation air via the heat exchanger $Ex_{DHW,ex}$ (a, left) and via the heat pump $Ex_{DHW,hp}$ versus T_e (b, right).

In Figure 5a, the $Ex_{DHW,ex}$ lines for system B (dashed ones) are almost congruent to the reference line (no DHW preheat), because the En_{th} and Ex_{th} of the exhaust ventilation air are very small. Most of them are recovered in the HRU. The $Ex_{DHW,ex}$ lines for system A (smooth ones) are constant when $T_e < T_w$ according to the assumption that the DHW is preheated until $T_{vent,out} = T_w$. In addition, the $Ex_{DHW,ex}$ depends on the Q_{vent}. For system A operating at Q_{vent}=0.063, 0.042 and 0.028 m^3/s, $Ex_{DHW,ex}$ are 91.24%, 96.06% and 98.24% of the Ex_{DHW} (no DHW preheat) respectively, when $T_e < T_w$. When $T_e > T_w$, $Ex_{DHW,ex}$ increases because the energy and exergy of the exhaust ventilation air decrease.

In Figure 5b, the $Ex_{DHW,hp}$ decreases, as T_e increases from -13°C to 19°C or Q_{vent} decreases. In addition, the $Ex_{DHW,hp}$ lines for system A (smooth ones) are sharper than the $Ex_{DHW,hp}$ lines for system B (dashed ones), because more heat from system A are used in DHW preheat. This costs more electricity, which is high-graded energy, by the heat pump.

The $Ex_{DHW,hp}$ lines for system A (smooth ones) have more than one slope in the range of T_e. At the lower slope (in this case T_e>0°C for Q_{vent}=0.063 m^3/s), the heat pump heats DHW until a temperature less than T_{DHW} and additional heat is then required. On the other hand, at the sharper slope, the heat pump could heat DHW to T_{DHW} only by using electricity. The $Ex_{DHW,hp}$ lines for system B (dashed ones) have only one slope in the range of T_e, since the heat pump could heat DHW until a temperature less than T_{DHW}.

The total energy and total exergy demands by the dwelling ventilation and the domestic hot water production systems in winter days

This item presents examples of cumulative energy and exergy demands for De Bilt (NL) for three representative winter days (Figure 2). Figure 6 shows the energy *En* and exergy *Ex* demands of each design option, in terms of heat and electricity, at the component level (noted that scales of the Figures 6a and 6b are different). The names of the design options consists of an alphabet (A: the mechanical exhaust ventilation with natural air supply; B: the balanced ventilation with heat recovery) and a number (1: no DHW preheat; 2: DHW preheat using the heat exchanger; 3: preheat using the heat pump).

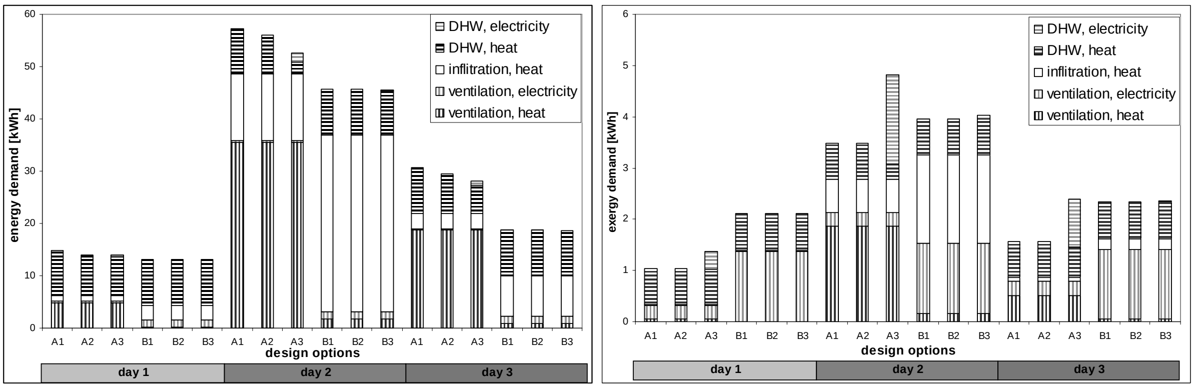


Figure 6. Energy demands (a, left) and exergy demands (a, left) of the design options.

In Figure 6a, option B3 is the most energy efficient one to use in the winter days because it requires the least energy *En* among the other options studied. The *En* of option B3 in the part of ventilation and infiltration depends on environment. The demand is highest on day 2 and lowest on day 1. Option B3 saves 0.45% of the *En* in the DHW production of option B1 (no DHW preheat) because of small heat of the exhaust ventilation air used in the DHW preheat.

In Figure 6b, option A2 is the most exergy efficient one because it requires the least exergy *Ex* among the other options studied, for the winter days. In the part of mechanical ventilation, options A1-A3 require higher thermal exergy than options B1-B3 but much lower electric exergy. In the part of infiltration, options A1-A3 require lower thermal exergy than options B1-B3. In total, options A1-A3 require lower total exergy than options B1-B3 in the parts of mechanical ventilation and infiltration. In the part of the DHW production, the *Ex* of option A2 are smaller than the demand of option A1 (without the DHW preheat), but for option B2 the *Ex* is quite similar to the *Ex* of option A1. However, in the same part the *Ex* of options A3 and B3 (with DHW preheat using the heat pump) are higher than the demand of option A1 (without DHW preheat), due to electricity input to the heat pump. This is because the heat pump uses electricity to pump low-graded heat. Nevertheless, options A3 saves more thermal exergy than options A2. Electric exergy input to the heat pump should be reduced. Besides, options B2 and B3 save very small amounts of the thermal exergy in the DHW production.

CONCLUSION

This paper presents steady-state energy and exergy analyses on an instantaneous and on a daily basis for dwelling ventilation systems and DHW production systems. Analysis results, for De Bilt, the Netherlands, are used for discussion of design options for systems.

On an instantaneous basis, the exergy demands by infiltration airflows for the balanced ventilation with heat recovery are higher than the exergy demands for the mechanical exhaust ventilation in the entire range of environmental air temperature. But this is not for the exergy demands by ventilation airflows because substantial shares of electric exergy input to ventilation unit are big and dominate the total exergy demands. The exergy demands by ventilation airflows for the balanced ventilation with heat recovery are lower than the exergy demands for the mechanical exhaust ventilation, when environmental air temperature close to room air temperature. In the DHW production part, using the heat exchanger for the DHW preheat with the exhaust air from the balanced ventilation with heat recovery saves a small amount of the exergy demands of DHW production, due to a small amount of heat from the air. Preheating the DHW with the exhaust air from the mechanical exhaust ventilation is better according to a bigger amount of the heat. DHW production with DHW preheat using the heat pump requires more exergy than DHW production without the DHW preheat, because the heat pump uses electricity to pump low-graded heat. The exergy of electricity input to the heat pump is larger than the exergy of heat recovered in the DHW preheat.

On a daily basis, for the winter days the total energy demands, but not the total exergy demands, of balanced ventilation and DHW production with preheat by the exhaust ventilation air using a heat pump are lowest. The total exergy demands are dominated by exergy of electricity input to heat recovery unit and the heat pump.

ACKNOWLEDGEMENT

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Residential HVAC System Sizing

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SUMMARY

Heating, ventilating and air-conditioning (HVAC) system sizing for existing single family residents in hot and humid and temperate climates present different problems. In hot and humid climates, the proper sizing of residential air-conditioning systems is an important issue, since if the system is over-sized the resulting mold problems can cause significant problems with occupants who are susceptible to airborne spores like mold that can severely affect their asthmatic illnesses.

In more temperate climates, HVAC system sizing for existing single family residential unit is primarily concerned with heating systems. However, there is often a need to retrofit the residential unit with an air-conditioning system.

The technique for properly sizing residential air-conditioning and heating systems is based on methods described in the ASHRAE Handbook of Fundamentals. Heating and cooling spreadsheet programs that duplicate the methodologies were developed and have been used to evaluate the design cooling load for a number of single family residential units (condominium and homes) in Florida and in New York and Massachusetts. Two examples, a condominium in subtropical Tampa, Florida and a condominium in temperate Albany, New York are presented and discussed. Conclusions and recommendations are made on the sizing criteria for residences in these two diverse climates.

INTRODUCTION

In hot and humid climates, the proper sizing of residential air-conditioning systems is an important issue, since if this is not done correctly the resulting mold problems can cause significant problems with occupants who are susceptible to airborne spores like mold that can severely affect their asthmatic illnesses. The author has examined several hundred residences (single family homes and condominiums) that were constructed in the last 10 to 30 years in the State of Florida where both new and replacement air-conditioning systems were significantly oversized. In a recent seminar presentation by Cochell [1] indicated that a study of 1600 homes in Florida by the University of Florida showed that 78% of the homes had oversized air-conditioning systems by more than ½ Ton (6,000 Btu/hr or 1759 W). As a result, the occupants were exposed to a cooled, damp indoor climate where the oversized air-conditioning system runs for short periods of time to bring the indoor air temperature down to the set-point temperature but does not run long enough time to significantly reduce the humidity levels in the air. The mold accumulates in the air-conditioning ducts that were installed at the time of the original air-conditioning system. In several of the worse over-sized air-conditioning systems that the author has seen, the occupants became quite ill from the resulting mold growth and their entire living quarters had to be completely renovated.

In more temperate climates, many middle-aged (30 to 60 years old) and older (greater that 60 years old) residences have old inefficient heating systems where approximately 50% of the heating fuel is wasted. In most situations, the heating and cooling system sizing exercise is more complex since design drawings are usually not available and many residences have undergone a number of renovations. This makes it necessary to spend a lot of time in developing the building envelope plans and ascertaining the construction materials. In addition, most of these residences did not have air-conditioning systems installed at the time of construction. One recent development in air-conditioning duct systems is the use of properly designed smaller diameter higher velocity air ducts that can be installed in a manner similar to the way smaller diameter electrical wiring can be snaked into the walls, ceilings and floors without disturbing the inside and outside surfaces. Here, in addition to the more difficult air-conditioning sizing exercise, the design of the high velocity air duct retrofit also has to be carried out.

The technique for properly sizing a residential air-conditioning and heating system is based on methods described in Chapter 28 of ASHRAE [2]. Spreadsheet programs that duplicate these heating and cooling load methodology have been developed and used to evaluate the design heating and cooling loads for the residences in both hot and humid and more temperate climates using the specified outside design temperatures given in Chapter 27 of ASHRAE [2]. For cooling load calculations, the estimate of the latent cooling load is the most critical element in the selection of the size and type of retrofit air-conditioning system. For heating load calculations, the estimate of the air infiltration/ventilation rate contribution is the most critical element in the selection of the size and type of heating systems. The sizing results for two residences, one for cooling in a sub-tropical climate and one for heating in a temperate climate are presented in the next section of the paper. Conclusions and recommendations are made on the sizing criteria for residences in these two diverse climates.

CLIMATIC DATA TABLES

Two single family residences in a temperate location in Albany, New York (a second floor condominium) and a subtropical location in Tampa, Florida (a first floor condominium Unit) were studied. Tables [1] and [2] present a small portion of the data available for these two locations from Chapter 27-Climatic Design Information in ASHRAE [2] which presents selected climatic data for the United States, Canada and World locations. More detailed climatic data can be found in ASHRAE [3].

In Table [1], for winter design conditions, the second column gives the latitude, longitude and elevation for the two locations given in the first column. The third column gives the design (99.6% and 99% percentile) winter heating dry bulb temperatures that represent temperatures that are expected to only be exceeded 0.4% and 1.0% of the winter heating season. The fourth column represents design (5%, 2.5%, and 1% percentile) extreme wind speeds that are expected to only be exceeded 5%. 2.5% and 1% of the winter heating season.

In Table [2], for summer design conditions, the second column gives the latitude, longitude and elevation for the two locations given in the first column. The third column gives the design (0.4%, 1% and 2% percentile) summer cooling dry bulb and mean coincident wet bulb temperatures that represent temperatures that are expected to only be exceeded 0.4%, 1% and 2% of the summer cooling season. The forth column gives the mean daily range (difference between the daily maximum and minimum temperatures during the hottest month) of the dry bulb temperature.

The data in Tables [1] and [2] can be used in the determination of the design heating and cooling loads that are presented in the next section. It should be stressed here that the emphasis is on equipment sizing, not annual energy use.

Table 1 Winter Heating Design Conditions

LOCATION	Latitude Longitude Elevation (m)	Design Heating Dry Bulb ºC	Extreme Wind Speed km/h
Albany, New York	42.75º 70.38º 89	-21.7 (99.6%) -16.7 (99%)	30.6 (5%) 35.4 (2.5%) 38.6 (1%)
Tampa, Florida	30.38º 84.37º 21	2.2 (99.6%) 4.4 (99%)	24.9 (5%) 27.4 (2.5%) 30.6 (1%)

Table 2 Summer Cooling Design Conditions

LOCATION	Latitude Longitude Elevation (m)	Cooling Dry Bulb/Mean Coincident Wet-Bulb ºC	Range of Dry Bulb ºC
Albany, New York	42.75º 70.38º 89	32.2/21.7 (0.4%) 30/21.1 (1%) 28.9/20.6 (2%)	13.2
Tampa, Florida	30.38º 84.37º 21	33.3/25 (0.4%) 32.8/25 (1%) 32.2/25 (2%)	8.3

RESULTS

The following Chapters in the ASHRAE Handbook [2] were used in the calculation of the heating and cooling loads presented in the load calculation results tables:

Chapter 25: Thermal and Water Vapor Transmission Data
Chapter 26: Ventilation and Infiltration
Chapter 27: Climatic Design Information
Chapter 28: Residential Cooling and Heating Load
Chapter 30: Fenestration

Heating Load Calculation Results

Table [3] presents a summary table that summarizes the detailed room-by-room spreadsheet results for determining the heating load of a second floor condominium unit located in an approximately 100 year old three story building that was being converted to a two unit condominium association. The second floor unit also includes an unheated attic above it and a first floor unit below it. The ground floor is a partially above ground basement that contains the individual heating systems for the two condominium units. The second floor unit heating system uses oil as the heating fuel and the first floor unit uses natural gas as the heating fuel.

The two Unit Owners own the interiors of their individual units and share ownership of the exterior surfaces of the building and the surrounding land.

Table [3] gives results that follow the procedures presented in Table 12 (Summary of Loads, Equations, and References for Calculating Design Heating Loads) of Chapter 28: in ASHRAE [2]. The wall and floor areas and the ceiling height were obtained using a laser measuring device. The various U-values were obtained from the data and calculation methods presented in Chapters 25 and 30 in ASHRAE [2] for the specified building components. The determination of the construction of the wall, floor and ceiling required the temporary removal of a number of electrical fixtures (wall outlets and switches, ceiling lights) to determine the type and thickness of the various building materials used in the original construction. The data for the Albany, New York location from the Climatic Design Information given in Table 1B of Chapter 27 in ASHRAE [2] is listed in the top portion of Table [3]. The inside to outside temperature difference of 42.7ºC comes from the indoor design temperature of 21.1ºC and the 99.6% winter design dry bulb temperature of -21.6ºC. The temperature differences for the unheated attic were calculated from methods given in Chapter 25 in ASHRAE [2].

The critical heating load element, air infiltration, requires the use of Chapter 26 in ASHRAE [2] to estimate the sensible heating load due to air infiltration for the second floor condominium unit. The 5% design exterior wind speed of 30.6 km/h (19 miles/h) was used in applying two calculation models to obtain estimates of the infiltration flow rates. The “basic model” in Chapter 26 in ASHRAE [2] gave an infiltration rate of 0.62 air changes per hour (ACH) and the “enhanced model” gave an infiltration rate of 0.80 ACH. To be conservative in sizing the heating equipment for the not very well sealed 100 year old building, a value of 1.0 ACH was used. Multiplying the floor (or ceiling in this case) area of 118 m^2 by the ceiling height of 2.6 m gives a volume of 307 m^3. With one ACH, the air infiltration volume flow rate is 307 m^3/h as shown in Table 3.

Figure 3-RESIDENTIAL HEATING LOAD CALCULATION
Location: Albany, New York
Weather Data Location: Albany: Indoor Design Temperature: 21.1ºC
Winter Design Dry Bulb 99.6%: -21.7; 99%: -16.7ºC
Extreme Wind Speeds: 5%: 30.6 km/h; 2.5%: 35.4 km/h; 1%: 38.6 km/h
Ceiling Height: 2.6 m

SENSIBLE LOADS	Area m^2	U-value $W/(m^2 \cdot {}^{o}C)$	Temp Diff ºC	Energy W	Notes
Exterior walls	84.8	1.24	42.7	4501	Wood Stud/No Insulation
Wall to interior hallway	10.3	7.72	12.2	967	Wood Stud/No Insulation
Windows	20.3	3.23	42.7	2789	Single Pane w/Storm Windows
Exterior door	2.3	2.04	42.7	203	Wood
Door to interior hallway	2.2	1.69	12.2	46	Wood
Floor to heated unit	117.8	1.15	0	0	Wood Stud/No Insulation
Ceiling to unheated attic	117.8	0.35	36.1	1472	Wood Stud w/Insulation
Subtotal				9978	
	$\rho \cdot Cp$ $kJ/(m^3 \cdot {}^{o}C)$	Volume Flow Rate m^3/h			
Infiltration	1.205	306.8	42.7	4385	
TOTAL				14363	

Cooling Load Calculation Results

Table [4] presents a table that summarizes the detailed room-by-room spreadsheet results for determining the cooling load of a first floor condominium unit located in an approximately 30 year old four story building that contains 22 condominium units. The ground floor is a parking garage. The Table [4] results are obtained from the procedures presented in Table 9 (Summary of Procedures for Residential Cooling Load Calculations) of Chapter 28 in ASHRAE [2]. In addition, a number of related tables and figures, their footnotes and detailed example calculations in Chapter 28 in ASHRAE [2] are used in the spreadsheet calculations for determining the various elements of the total sensible load. The areas, material properties are determined in the same manner as in the heating load calculations. Glass (windows and doors) are treated differently. A glass factor replaces the U-value and temperature difference to account for the peak solar and sensible heat load. This factor is a function of the type of glass; the orientation, inside and outside shading; the design outdoor temperature (the 0.4% value of 33.3°C given at the top portion of Figure [4] was used in the calculations) and the indoor design temperature (23.9°C was used). For exterior walls and floors, a cooling load temperature difference (CLTD) is determined by orientation; outside shading; the design outdoor temperature and the indoor design temperature. The air infiltration is not as important a factor as it is for heating situations. In Table [4], an ACH of 0.72 was used. It should be noted that the various temperatures have been rounded off to the nearest °C, however the resulting energy values include the non-rounded off values in the detailed spreadsheet calculations.

The critical cooling load element, the latent load, is estimated from the outdoor design humidity ratio and the type of construction (loose, medium, tight) in Figure 1 of Chapter 28 in ASHRAE [2] to determine a load factor (1.16 was used) which is multiplied by the total sensible load to arrive at the latent load as given near the bottom of Table [4]. A properly sized air-conditioning system has to be able to remove a significant amount of moisture from the air in sub-tropical climates.

Figure 4-RESIDENTIAL COOLING LOAD CALCULATION
Location: Multifamily Residential Condominium-Tampa, Florida, FL
Weather Data Location: Tampa: Indoor Design Temperature: 23.9 ºC
Summer Cooling Design Temperature (Dry Bulb/Mean Coincident Wet-Bulb): 0.4%: 33.3/25 ºC; 1%: 32.8/25 ºC; 2%: 32.2/25 ºC
Range of Dry Bulb Temperature: 8.3 ºC (Low Range-less than 8.9 ºC)
Ceiling Height: 2.44m

SENSIBLE LOADS	Area m^2			Energy W	NOTES
GLASS		Glass Load Factor $W/(m^2)$			
Window and Doors	33	84		2737	West facing to Outside with Interior Blinds, and Exterior Shading
Patio Door	2	84		187	North facing to Outside with Interior Blinds, and Exterior Shading
ENVELOPE		U-value $W/(m^2 \cdot °C)$			
OUTSIDE FACING			Cooling Load ΔT (ºC)		
Floor	164	0.82	7	927	To Outside Fully Shaded Garage below
Exterior wall	40	0.82	8	268	North facing to Outside
INSIDE FACING			Temp Diff ºC		
Interior wall	40	1.05	0	0	South facing to Air-Conditioned Unit
Interior wall	33	1.05	0	0	East facing to Air-Conditioned hallway
Ceiling	164	0.82	0	0	To Air-Conditioned Unit above
INFILTRATION & VENTILATION	ρ*Cp $kJ/(m^3 \cdot K)$	Volume flow rate m^3/h			
Infiltration	1.205	287	9	854	0.72 Air Changes per Hour
INTERNAL LOADS					
Lighting/Appliances				352	Value for Multifamily Unit
People				202	3 People for 2 Bedrooms
SENSIBLE LOADS				5527	
LATENT LOAD				6412	1.16 Times Sensible Load for Medium Airtight Construction at 1% Design Humidity Ratio
TOTAL LOADS				11939	

DISCUSSION

The total heating energy required in the condominium in Albany, New York is 14363 W (in I-P units 48982 Btu/h). The oil-fired heating system has low pressure steam delivered to radiators in each of the heated rooms. This system also supplies some water vapor to the rooms to allow a comfortable winter relative humidity level to the occupants. Preliminary spreadsheet calculations indicated that insulating the ceiling/attic floor would reduce the energy to the unheated attic space enough to economically warrant the expense of having fiberglass blown into the cavity between the attic floor and 2nd floor ceiling. The original heating system (boiler) was rated at 75000 Btu/h and the new higher efficiency boiler system was rated at 50000 Btu/h. Both ratings account for the heating system piping losses to the basement. The first season of oil consumption with the new heating system was 40% less than the prior years. However the cost savings was much smaller due to the increase in heating fuel prices.

In addition to the new heating systems for both of the Albany, New York condominium units, new 16 SEER (seasonal energy efficiency ratio), two-stage air-conditioning systems were also added to both units. See Dulley [4] for details of this type of modern high-efficiency, split system air-conditioning systems. Detailed cooling load calculations similar to that for the condominium in Tampa, Florida were performed. For the second floor unit studied here, the evaporator coil/air handler unit was located in the basement near the outdoor compressor/condenser unit. New high velocity smaller diameter duct work (supply and return) was snaked through the first floor unit walls to the second floor unit walls. See Dulley [5] for details of these types of mini-duct, pressurized central air-conditioning systems.

The total cooling energy required in the condominium in Tampa, Florida is 11939 W (in I-P units 40745 Btu/h or 3.4 Tons). The original over-sized air conditioning system was rated at 4 tons or 48000 Btu/h. The new replacement high 16 SEER high-efficiency, heat pump air-conditioning system was rated at 3 tons or 36000 Btu/h. The reason for selecting a lower rating than the calculated load was due to the fact the Unit Owner was away during the summer season and had recently installed hurricane shutters which significantly reduce the sensible cooling load and forced most of the replacement air to come from the air-conditioned hallway at a humidity ratio much lower than that of the outside air that normally came in through the west facing outside windows. In addition, a humidistat was added in parallel to the thermostat to insure that high relative humidity ratios did not occur while the Unit Owner was away. After two summers with the new air-conditioner reduced the Unit Owners electric bill by over 30% and the occurrence of mold growth was reduced significantly. During the short winter heating season, the electric costs were also reduced by using the new air-conditioning system in the heat pump mode which is more efficient that the electric heating in the original system.

In summary, it can be seen that significant energy savings can be achieved by properly sizing heating and cooling system without any reduction in user comfort.

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Moisture convection performance on the joint of external wall and attic floor - laboratory tests and two-dimensional simulation model validation

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SUMMARY

Full-scale laboratory measurements were conducted to determine moisture convection performance on the joint of external wall and attic floor. According to field measurements in previous studies this joint is one of the most typical air leakage paths. On this joint also the highest air pressure difference forms in winter. Two commonly used external walls: timber-frame and autoclaved aerated concrete walls were measured. The attic floor was in both cases a timber-frame structure. Results from the first laboratory measurement series showed that in leaky joint the moisture convection due to positive air pressure remarkable raised moisture accumulation rate on the inner surface of sheathing. A two-dimensional heat, air, and moisture transport computer model was validated for future analysis. The simulation results showed that CHAMPS-BES program is useful tool in assessing the moisture behaviour of building components including moisture convection.

INTRODUCTION

Accurate assessment of hygrothermal performance of building envelope is important to prevent moisture damages and to guarantee longer service life as well better indoor air quality for buildings. Usually, in consulting companies, the hygrothermal dimensioning of building envelope is limited to the moisture accumulation in structures based on the Glaser-method that is a simple method for determining the risk of moisture condensation. Unfortunately, this method has many limitations: the method is the steady state, it does not consider hygroscopic materials, vapour diffusion is the only transport mechanism, condensation is only performance criterion, there are no coupling between the heat and moisture transport, and material properties are constant.

Ventilated facade claddings are commonly used in timber-frame walls that should protected it from driving rain. If timber-frame wall haves a vapour barrier that controls the vapour diffusion, the major humidity load is usually the outgoing airflows due to air leakage. The role of air exfiltration for the hygrothermal performance of the building envelope is analysed in many studies [1, 2, 3, 4]. The uncontrolled air movement through a building envelope is not only hygrothermal problem, but may lead also to problems related to the health, energy consumption, performance of the ventilation systems, thermal comfort, noise, and fire resistance.

Air leakage through a building envelope depends on the result of air-pressure differences over the envelope and air tightness of the building envelope. The positive air pressure difference over the building envelope can be utilised in the control of radon, particulate matter, fungal spore or other contaminant transport to the indoor air. At the same time high indoor humidity load and positive air pressure conditions may cause intensive moisture accumulation in building envelope. As air pressure difference over the building envelope cannot be avoided even in typical residences [5] and buildings with timber-frame envelope are not totally airtight [6, 7], convection should be take into account in the process of hygrothermal design. As air leakages are concentrated to the joints of different envelope parts (wall, roof, floor, window), the air leakage is multidimensional and two-dimensional measurements and simulation should be done.

In this study typical air leakages and their locations were determined from the field measurement data reported in [8]. A full-scale laboratory measurements were carried out to determine moisture convection performance on the joint of external wall and attic floor. A two-dimensional heat, air, and moisture transport computer model is validated for future simulations.

METHODS

Laboratory measurements

Two commonly used external wall (timber-frame wall (Figure 2) and autoclaved aerated concrete (AAC) wall (Figure 3)) joints with timber-frame attic floor is measured in laboratory conditions between climatic chambers. The warm climatic chamber is equipped with heating and humidification units and it simulates the indoor climate. The cold climatic chamber is equipped with refrigeration, heating and humidification units simulating the outdoor climate. The climatic chambers are automatically controlled and continually maintain equilibrium of various climatic parameters, such as air temperature, relative humidity (RH), air pressure difference over the examined structure.

Studied two-dimensional joint details consist of 1.2 by 0.7 m (width by height) external wall part and 1.2 by 0.9 m attic floor part. Studied setup was built as much as possible airtight and special well-defined air leakages were constructed for the joint. In timber-frame wall/attic floor joint the 10 cm wide air leakage was let between two air/vapour barrier foils. In AAC wall/attic floor joint a ~1…2 mm thick and 10 cm wide air gap was let between wall plate and AAC block. The real air inflow through the air leakage part and overall air tightness of the joint with closed air leakage part were measured with an electronic soap film calibrator (bubble flow meter), Figure 1.

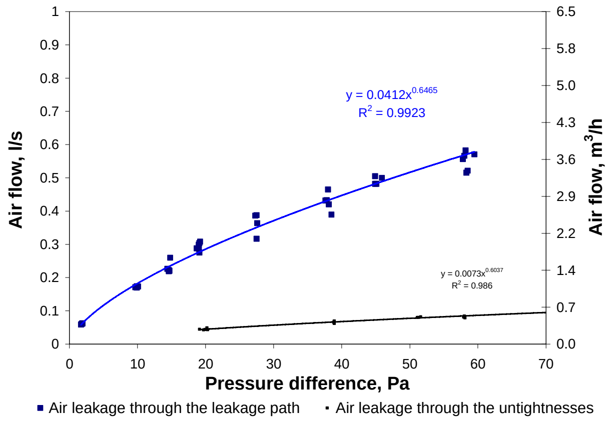

Figure 1 Measured air leakage thought the airgap between two air/vapour barrier foils (timber-frame wall/attic floor).

The laboratory measurements were done at -6 °C and -11 °C outdoor temperature conditions. The moisture excess (Δv, the difference between indoor and outdoor air's humidity by volume) was 4 g/m^3 in both tests. According to measurements in detached houses, this is the design value of moisture excess for dwellings low occupancy [9, 10]. The air pressure difference (ΔP) was +10…+11 Pa. According to measurements and simulations in detached houses the design value of air pressure difference over the building envelope should be at least ±10Pa [5].

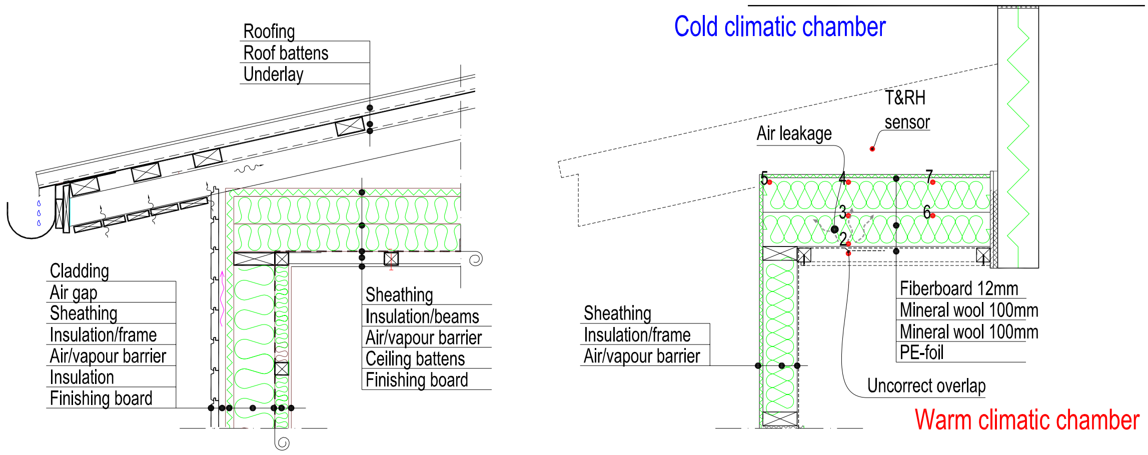

Figure 2 Studied timber-frame wall joints with timber-frame attic floor (left) and the studied joint in laboratory conditions between climatic chambers (right). Temperature and RH measurement points are shown by red dots.

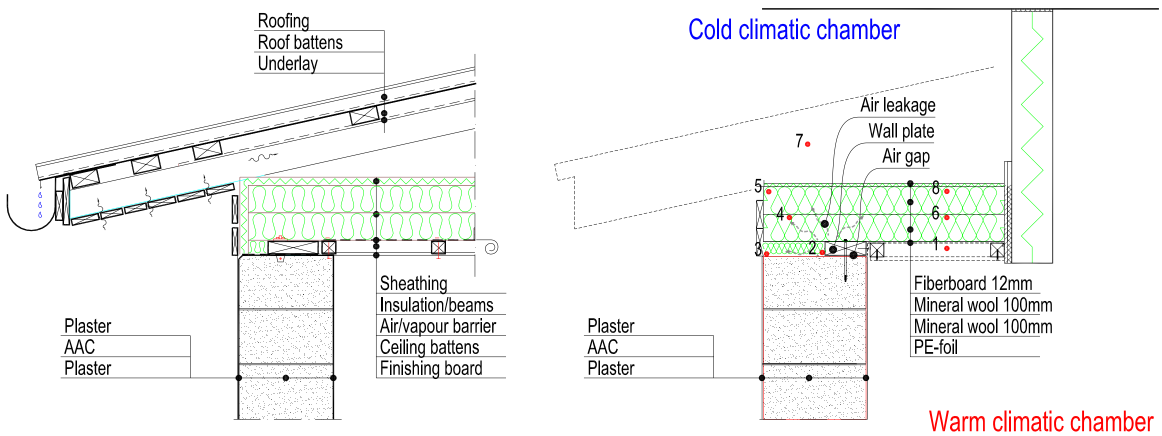

Figure 3 Studied autoclaved aerated concrete wall connection with attic floor (left) and the structure in laboratory conditions between climatic chambers (right).Temperature and RH measurement points are shown by red dots.

Simulation mode

Based on laboratory measurements the simulation model built with CHAMPS-BES program (Coupled Heat, Air, Moisture and Pollutant Simulation in Building Envelope Systems) (version 1.4.1) was validated for future analysis. This program is an outcome of a joint effort between Building Energy and Environmental Systems Laboratory at Syracuse University, U.S.A. and Institute for Building Climatology at University of Technology Dresden, Germany.

CHAMPS-BES modelling comprises the description of fluxes in the calculation domain or in the field (between volume elements including material interfaces) and at the boundary (between volume elements and exterior or interior rooms) by physical models. Also included are models for storage processes like adsorption, desorption and release. The numerical solution is done by semi-discretization in space (using a finite/control volume method) and subsequent integration in time [11].

The governing equation for the moisture balance is:

$$\frac{\partial}{\partial t}\rho_{REV}^{m_{w+v}} = -\frac{\partial}{\partial x}\left[j_{conv}^{m_w} + j_{conv}^{m_v} + j_{diff}^{m_v}\right] \quad (1)$$

where:

$\rho_{REV}^{m_{w+v}}$	Moisture (liquid+vapour) density in reference volume, kg/m^3
$j_{conv}^{m_w}$	Convective liquid (capillary) water flux, kg/m^{2}s
$j_{conv}^{m_v}$	Convective water vapour flux, kg/m^{2}s
$j_{diff}^{m_v}$	Diffusive water vapour flux, kg/m^{2}s

The governing equation for the air mass balance is:

$$\frac{\partial}{\partial t}\rho_{REV}^{m_a} = -\frac{\partial}{\partial x}\left[j_{conv}^{m_a}\right] \quad (2)$$

where:

$\rho_{REV}^{m_a}$	Air mass density in reference volume, kg/m^3
$j_{conv}^{m_a}$	Convective air mass flux, kg/m^{2}s

The governing equation for the energy balance is:

$$\frac{\partial}{\partial t}\rho_{REV}^{U} = -\frac{\partial}{\partial x}\left[j_{diff}^{Q} + u_l\, j_{conv}^{m_l} + u_g\, j_{diff}^{m_g} + h_v\, j_{diff}^{m_v}\right] \quad (3)$$

where:

ρ_{REV}^{U}	Internal energy density in reference volume, J/m^3
j_{diff}^{Q}	Heat conduction, W/m^2
u_l	Specific internal energy of liquid phase, J/kg
u_g	Specific internal energy of gas phase, J/kg
$h_v\, j_{diff}^{m_v}$	Specific enthalpy of water vapour, J/kg

In the simulation for the material properties such as density, thermal conductivity, moisture permeability, and moisture storage function the measured material properties from Tampere University of Technology [12] were used. The example of the construction view of the simulated timber-frame walls connection with attic floor is shown in Figure 4.

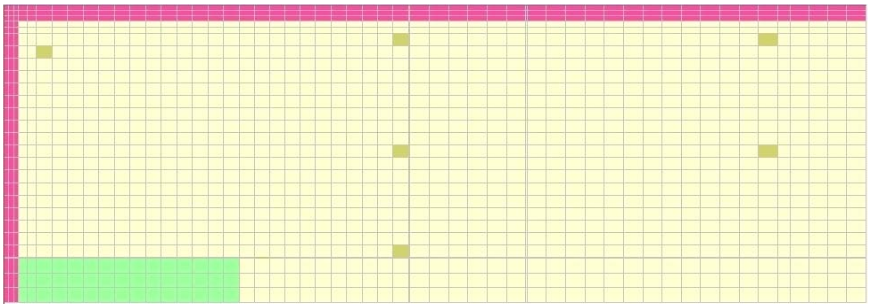

Figure 4 The grid of simulated timber-frame walls connection with attic floor. Temperature and RH measurement points are shown by shaded points. (green, yellow and red colours mark sheathing, mineral wool, and wood materials respectively).

RESULTS

Laboratory measurements

Full-scale laboratory measurements were performed to analyse moisture convection performance of the joint of external wall and attic floor and to get reference data to compare simulation results. Boundary conditions for the first two measurement series are shown in Table 1. Laboratory tests were made in two cycles: first, to stabilise the joint of external wall and attic floor, the test was done with closed (taped) air leakage path. This cycle continued 1…2 weeks. After that, the 10 cm long air leakage was let between air/vapour barrier foils (timber-frame wall/attic floor joint) and between wall plate and AAC block (AAC wall/attic floor joint). The measurements are continued with other indoor humidity loads, air pressure differences, air leakage conditions, and other sheathing materials.

Table 1 Boundary conditions for the first two measurement series

Test	T_{in}, °C	RH_{in}, %	T_{out}, °C	RH_{out}, %	ΔP, Pa	$\Delta \nu$, g/m^3
1	21	32	-11	77	+11	4
2	20	37	-6	75	+10	4

There was condensation and frost formation on the inner surface of the sheathing of the attic floor in the end of the tests 1 and 2, Figure 5. During 30-day measurement period in Test 1 the increase in the mass-related moisture content of the wood fibreboard sheathing of the attic floor on the test 1 was 10 % and the final moisture content was 22 %.

Figure 5 Condensation and frost with mineral wool pieces on the inner surface of the sheathing of the attic floor on the timber-frame wall (left) and the AAC wall (right) cases.

Comparison of laboratory tests and results of simulations

To develop the model for the future analysis, the simulation program CHAMPS-BES was validated based on full-scale laboratory measurements of Test 2.

Comparison of measured (Test 2, Table 1) and simulated temperature, RH, and the humidity by volume on the outer surface of insulation (behind the sheathing of the attic floor: T&RH sensors 4, 5, 7 in Figure 2) are shown in Figures 6-8.

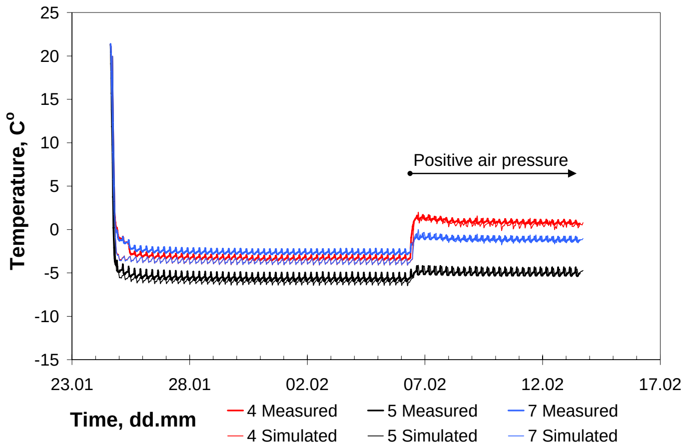

Figure 6 Simulated and measured temperatures behind the sheathing of the attic floor in the case of timber-frame walls connection with attic floor.

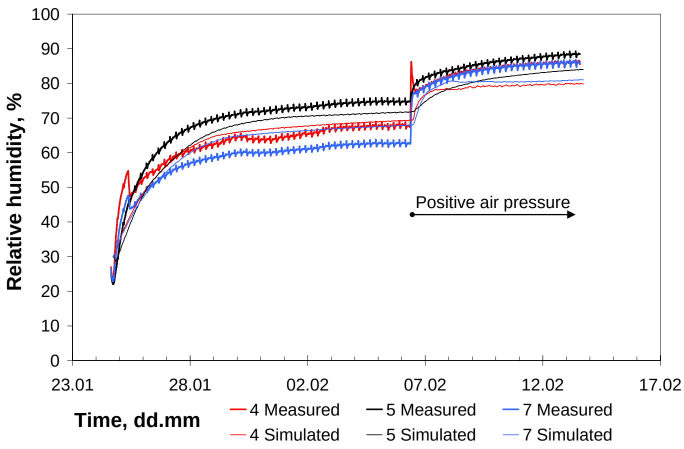

Figure 7 Simulated and measured RH behind the sheathing of the attic floor in the case of timber-frame walls connection with attic floor.

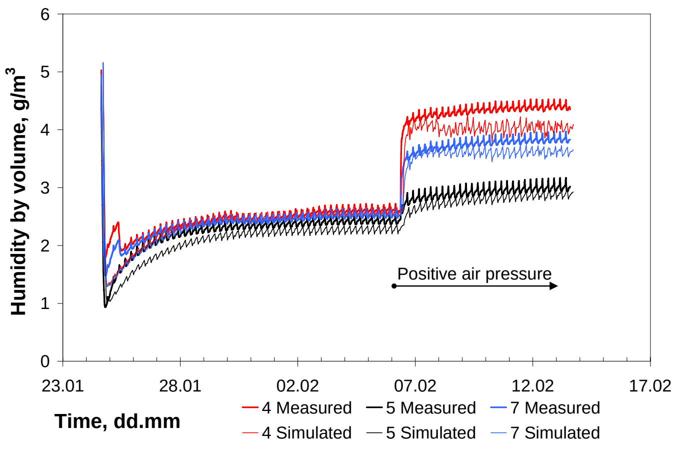

Figure 8 Simulated and measured humidity by volume behind the sheathing of the attic floor in the case of timber-frame walls connection with attic floor.

DISCUSSION

There was condensate and frost formation on the inner surface of the sheathing of the attic floor in case of positive air pressure difference +10…11 Pa and moisture excess +4 g/m^3. Moisture convection remarkable raised moisture accumulation rate: during the Test 1 (T_{out} -11°C) the increase in the moisture content of the wood fibreboard sheathing of the attic floor after 30 day period was 10 % and the final moisture content was 22 %. During the Test 2 (T_{out} -6°C) the frost formation took just longer time. The high moisture content can offer suitable conditions for mould, wood decay fungi, reduce the thermal resistance of building materials and change building physical properties of materials and deform materials.

Field measurements in Finnish detached houses [8] showed that one typical air leakage place was in the joint of attic floor with the external wall. Measurements in Estonian detached houses [7] showed, that more typical air leakage place was in the joint of interstitial floor with the external wall. The RH at the connection of intermediate floor and external wall can be rather high causing the risk of the growth of mould and rot fungi when there is positive air pressure inside the building [13]. As the air pressure difference over the building envelope on the joint of attic floor with the external wall is higher than in the joint of intermediate floor with the external wall, this joint was analysed in current study.

It is not possible to take into account the geometrical change of materials in simulation programs. Nevertheless during measurements the fiberboard sheathing swelled due to moisture accumulation. The joints of timber-frame and AAC walls with the timber-frame attic floor were simulated as two-dimensional joint. In reality, the convection in joint was surely three-dimensional (look condensation areas in Figure 5). These two facts as well as the fact that studied joints consist small imperfections that cannot be modelled may be the main reasons for the difference in comparison of simulation and measurement results. The main difference between measurements and simulations was that the moisture accumulation continued in measurements, while in the simulations the moisture content remained to the fixed level, Figure 8.

CONCLUSIONS

The moisture convection performance on the joint of external wall and attic floor was studied by full-scale laboratory measurements. A two-dimensional heat, air and moisture transport computer model was validated for future analysis.

Results from the first laboratory measurement series showed that in leaky joint the moisture convection due to positive air pressure remarkable raised moisture accumulation rate on the inner surface of sheathing.

Despite of some discrepancies between the results of laboratory measurements and computer simulations, the simulation program CHAMPS-BES showed good performance and is useful tool in assessing the moisture behaviour of building components including moisture convection.

The results will be utilized in further analyses aiming to specify air leakage properties allowing to expose studied structures to defined positive pressure and indoor humidity load. If necessary, the studied structures will be further developed so that the moisture convection problem can be controlled in the construction.

ACKNOWLEDGEMENT

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Different strategies to control humidity in Swiss low energy buildings

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SUMMARY

For the approximation of the internal loads of humidity, too high values are often chosen in Swiss low energy houses. Later on in reality, the occupation and the presence in the dwellings are significantly lower. These leads to high specific air volume rates and consequently to low humidity levels. Humidification by vapour generators, humidity recovery and optimized air volume rates are currently applied. Engineers can now calculate the internal humidity loads without humidity storage in walls etc. They can chose of humidity recovery and/or a vapour generator and set minimal and maximum values for relative humidity. The calculation of the resulting relative humidity and the consumption of electricity and water for the vapour generation are presented in figures. Simulations with IDA ICE have shown that the control strategy with humidity recovery had higher resulting relative humidity than the control strategy with the reduced and optimised air volume rates.

INTRODUCTION

A comparative study on Swiss low energy buildings indicated three important results: Heating energy consumption was reduced by a factor of five compared to conventional buildings, electricity consumption surprisingly remained at the average of Swiss households, and the humidity was below 30 % R.H. for thousands of annual hours. To enhance humidity level manufacturers have now started to promote centralized vapour generators even in low energy buildings. First calculations in the comparative study showed that vapour generators require more specific energy for humidifying than for heating a low energy building. At this point the project “Humidity in Swiss low energy buildings”, funded by the Swiss federal office of energy, has been started. Furthermore six partners from industry are involved in the project. The purpose of the project is to determine how the energy consumption for humidifying in low energy buildings can be reduced or even avoided. Further important parameters are thermal comfort and hygiene. The project was started in June 2006 and will be finished in June 2007.

METHODS

Equipment testing

The first part of the project focuses on the equipment. Rotary and plate heat exchangers in compact ventilation units are now capable of moisture transfer and are tested at the testing facility at the HTA Lucerne. A description of the testing facility with its entire possibilities can be found in the test guideline [2]. The testing facility was designed and built to satisfy present and future requirements as completely as possible, especially key data regarding air conditions. With this testing facility all important operating conditions for comfort ventilation systems can be replicated. The following types of appliances (and combinations thereof) can be tested within a volume flow range from 50 m^3/h up to 1200 m^3/h:

- Units with supply and return air and heat/moisture recovery (recuperative and regenerative)
- Units with supply and return air, heat recovery and exhaust-air heat pump to heat the supply air
- Units with return air, exhaust-air heat pump to heat domestic hot water
- Units with return air, exhaust-air heat pump which heats domestic hot water and/or transfers heat to another water-bearing system

a)

b)

Figure 1. Test center for compact ventilation units a) Outside view b) Internal view

As the test facility is especially designed to accommodate acoustic measurements, the acoustic characteristics of compact units can be tested during thermal performance. Thanks to the two testing chambers with double walls, the base sound level can be minimised, allowing measurements of the sound power level radiated by the casing into the testing chamber as well as the sound power level emitted through the air connections into the ducts. The air can be conditioned to the following specifications for testing purposes:

Table 1. Range of air conditions

Quantity	Outside air	Return air	Testing chamber
Temperature [°C]	-12 to +21	21 to 26	10 to 26
Humidity [% R.H.]	70 to 80	30 to 60	-
Air volume flow [m^3/h]	50 to 1200	50 to 1200	-

For the different strategies to control humidity in Swiss low energy buildings the existing test guideline to test compact ventilation units [2] had to be enhanced. Up to now compact ventilation units were not able to recover humidity. Therefore suitable test conditions in winter and summer had to be determined.

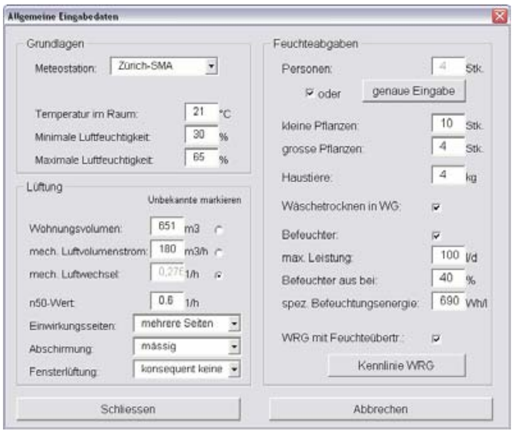
Table 2. Test conditions for humidity recovery

	1	2	3	4	5	6	7	8	9
$\vartheta_{Return.}$ [°C]	21	21	21	21	21	21	21	25	25
Φ_{Return} [%]	25	30	37	45	50	50	50	50	50
$\vartheta_{Outside}$ [°C]	-7	-3	+2	+7	-3	+2	+10	35	35
$\Phi_{Outside}$ [%]	80	80	80	80	80	80	80	50	35
Δx [kg/kg]	0.0022	0.0023	0.0023	0.0021	0.0054	0.0044	0.0017	0.0082	0.0074

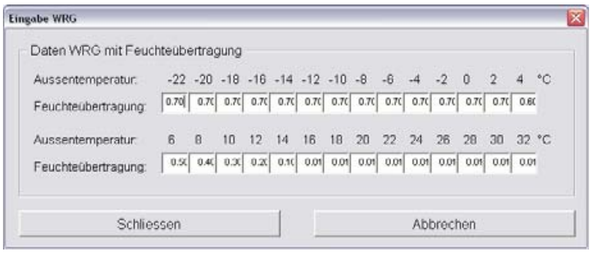
For the second part of the equipment testing the performance of a special type of vapour generator currently used in low energy buildings is determined in laboratory tests. Especially the control characteristics and the use of electricity are of interest.
Furthermore another vapour generator is under investigation by field measurements. In a location in Central Switzerland the behaviour of a commercially available vapour generator designed for low energy buildings is measured. Parameters are temperature, relative humidity, and electrical energy consumption. The same type of vapour generator is also under investigation in the laboratory tests.

Calculation tool

The second part of the project is the development of a calculation tool for engineers to estimate internal humidity loads, performance of humidity recovery, energy consumption of vapour generation and the resulting annual hours of relative humidity during the Swiss heating season from October to March. Actually eleven metrological stations located in Switzerland can be chosen by the user. Earlier studies [1] revealed that the internal moisture loads by plants, animals, and human is overestimated in some cases by a factor of two in calculation standards. Later on low relative humidity values are often measured. By the way it has to be mentioned that high specific air exchange rates – as a result of low occupancy – are frequent in Swiss low energy buildings. More appropriate values for internal moisture loads were once again reviewed in literature and are now fully implemented in the calculation tool. With this development state of the calculation tool the engineer is able to predict the moisture behaviour of the future dwelling. There is only one limitation in this calculation tool: Internal moisture storage capacity of e.g. walls is not implemented. In that case, the use of a simulation tool is recommended. The application of this software package is described in the next part of this paper.



a)



b)

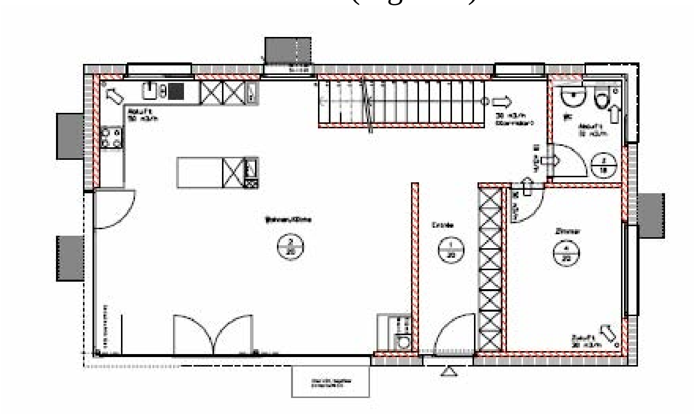
c)

Figure 2. Input data. a) Basic data, b) user-defined values for humidity recovery functions, c) values for occupancy

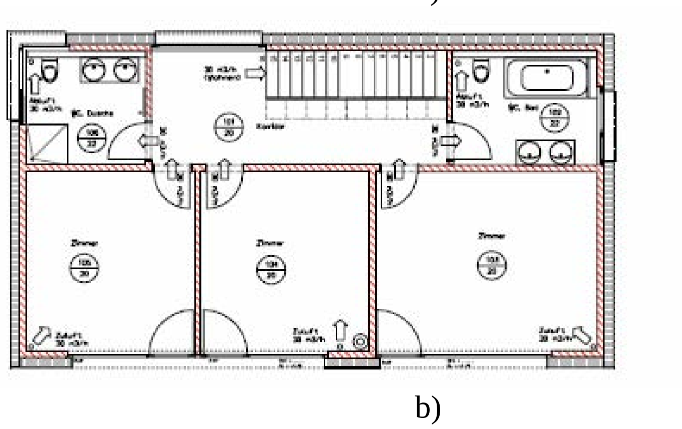
As shown in Figure 2a the user has to select the appropriate metrological station, to determine values for parameters like temperature, minimal relative humidity, maximal relative humidity, room volume, air volume rate, air exchange rate, air tightness of building envelope, number of plants, number of animals, and finally the specific data for the vapour generator. Figure 2b shows the possibility of user-defined values for the humidity recovery functions.

Simulations with IDA ICE

The third part of the project deals with the control of air flow rates to enhance humidity levels. Too high specific air exchange rates – as a result of low occupancy – are frequent in Swiss low energy buildings. The heating load in low energy buildings can be covered by preheating the supply air. A reduction of the air flow rate is therefore limited by the capability to fulfil the heating load requirement by the ventilation system. The control of air flow rate strategy is evaluated by the building simulation tool IDA ICE.
The fundamental input data was taken from the reference dwelling of the Swiss standard SIA 384.201. It is a three floor building located in Central Switzerland. The energy reference area is totally 160 m^2 respectively 80 m^2 for the cases with low occupancy. The floors were divided into thermal zones (Figure 3).



a)



b)

Figure 3. Input data. a) Ground floor with thermal zones, b) 1st floor with thermal zones, c) Ground floor with one respectively two occupants

By definition the following parameters where changed for each calculation:
- Number of occupants (1, 2 or 4)
- Capacity of moisture storage (low, medium, high)
- Control strategy (CO_2, combination of CO_2 and humidity)

Simulations for six cases of parameter combinations were done:

Table 3. Definitions for ten cases of IDA ICE simulations.

Case	**Number of occupants**	**Moisture storage capacity**	**Control strategy**	**Moisture recovery by wheel/plate- HEX**
1	4	low	CO_2	no
2 (Ref.)	4	medium	CO_2	no
3	4	high	CO_2	no
4	2	medium	CO_2	no
5	1	medium	CO_2	no
6	4	medium	CO_2 and humidity	no
7	4	medium	Constant air volume rate	yes
8	4	medium	Constant air volume rate	yes
9	4	low	Constant air volume rate	no
10	4	low	Constant air volume rate	yes

The following border conditions were used in the simulations:
Energy reference area: 80 respectively 160 m^2
• U-value roof: 0.13 W/m^2K
• U-value exterior wall: 0.15 W/m^2K
• U-value ground: 0.21 W/m^2K
• U-value window: 0.91 W/m^2K

Operation data for the simulations (adjusted):
• Air temperature living/kitchen, entrance, corridor, sleeping rooms: 20 °C
• Air temperature WC: 18 °C
• Air temperature bath: 22 °C
• Basic air volume rate depending on occupancy: 130, 80, 60 m^3/h
• Outdoor- / exhaust-air was divided and adjusted to the occupancy

Comparative study

The final part of the project is a comparative study on different ways to control humidity in low energy buildings: Energy, thermal comfort and hygiene are parameters.

RESULTS

Equipment testing

Four different compact ventilation units are currently being tested for this project. The measurements are not yet finished. In Figure 4 some recovered moisture coefficients in function of the condensation potential are shown as an example. The condensation potential is the difference between the absolute humidity of outside and exhaust air.

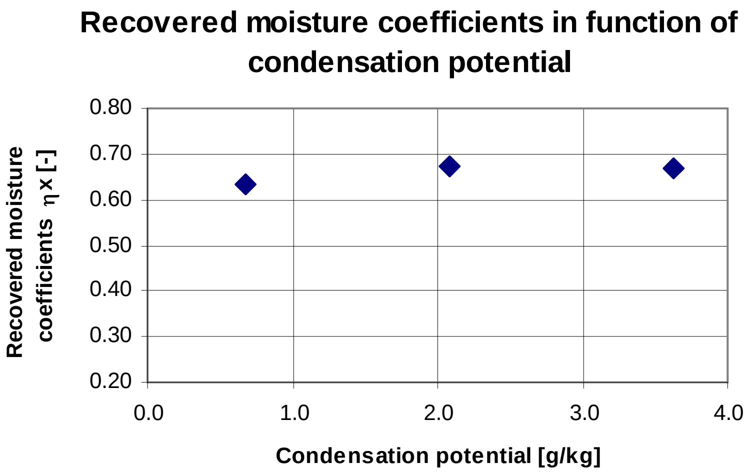
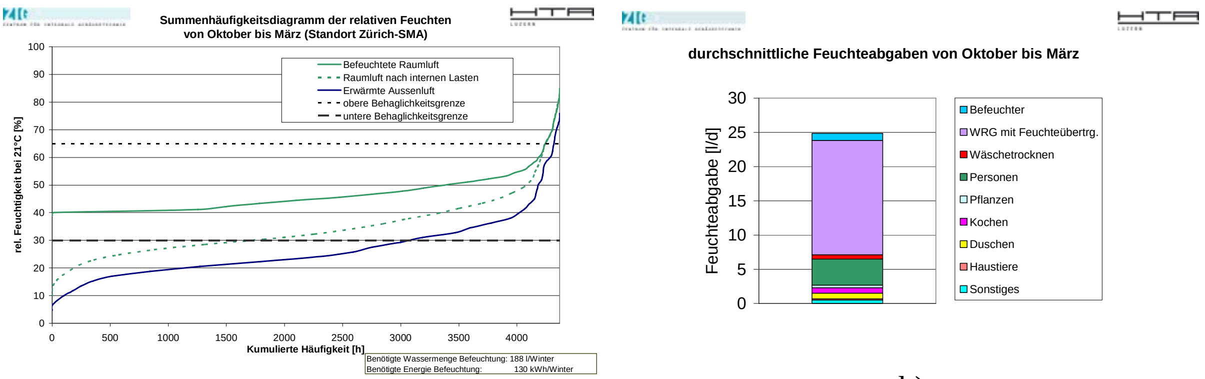


Figure 4. Recovered moisture coefficients in function of condensation potential.

Calculation tool

HVAC Engineers can now calculate the internal humidity loads without humidity storage in walls etc. They can apply the use of humidity recovery and a vapor generator and set minimal and maximum values for relative humidity. The calculation of the resulting relative humidity and the consumption of electricity and water for the vapour generator are presented in Figure 5. The results obtained by the calculation tool will be compared later on to those obtained by simulation in a validation procedure.



a) b)

Figure 5. Output data. a) Annual hours for relative humidity, b) internal moisture loads

Simulations

Simulations have shown that the control strategy with humidity recovery and constant air volume rates had a higher resulting relative humidity than the control strategy with the reduced and optimised air volume rates (CO_2). Obviously the constant air volume rate strategy without recovery or vapour generation leads to the lowest humidity level in January.

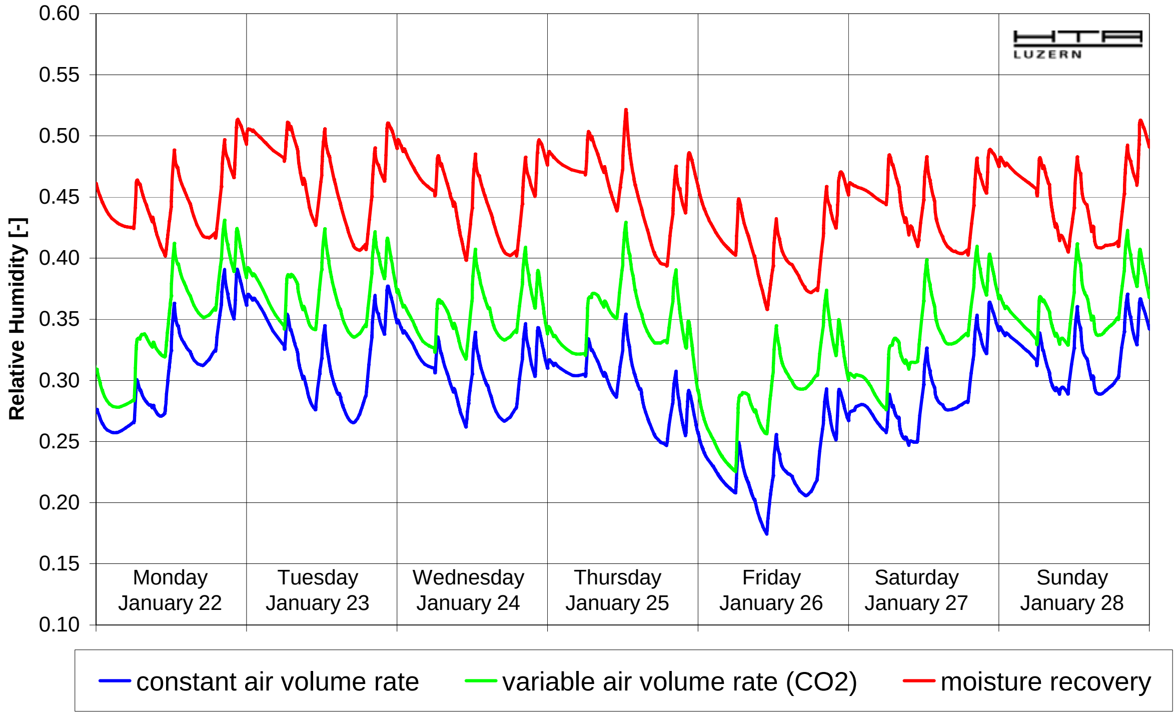


Figure 6. Simulations of three different control strategies for the living room.

Comparative study

From a general point of view, it is clear that every system has its own advantages and disadvantages. The use of vapour generators is certainly not the optimal solutions for low energy buildings. It is demanding that the additional use of electricity has to be minimized. Unfortunately, up to now it was not possible to significantly reduce the consumption of electricity in Swiss energy houses. Often the owners of these houses are not aware of the relatively high electricity consumption compared to the low energy consumption for heating. In newer publications, it was stated that the use of rotary heat exchanger has influences on the hygiene. Obviously, the best solution is the reduction of air volume rates. Surprisingly not in the resulting relative humidity, but it is still the best solution for the low occupancy and presence of people experienced during daytime.

DISCUSSION

Due to technical problems, there is a delay in the project especially in equipment testing. The lack of appropriate standards and definitions for the testing of humidity recovery in compact ventilation units has made the test procedure more time consuming than expected. Currently ISO 13417 is under revision to fit the new developments in compact ventilation units.

Simulations revealed that there is a potential in the application of humidity storage materials in walls to enhance the humidity levels in a longer range in low energy houses.

ACKNOWLEDGEMENT

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Hybrid ventilation tests in houses following the Spanish Technical Building Code (CTE)

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SUMMARY

The present communication forms part of the conclusions to emerge from the research project "Development of a methodology for the estimation of ventilation efficiency in residential houses" financed jointly by the Spanish Ministry of Education and Science and by ERDF, and carried out from 2003 to early 2006.
The aim is to present the results obtained for ventilation efficiency in systems of hybrid ventilation for houses, which is about to be implemented in Spanish legislation. We will provide a summary of studies conducted on air extraction rooms, such as bathrooms or kitchens, and transit or air circulation areas such as corridors, halls, or distributors.
For this purpose, representative typologies have been chosen, and a twin study has been performed using an experimental model in a laboratory with photoacoustic techniques of tracer gases, and a numerical model with CFD (Computational Fluid Dynamics) software techniques.
Numerical procedure validation has enabled us to emulate a variety of common situations different to the tested model and obtain conclusions on the architectural design of houses, thus allowing us to improve indoor air quality (IAQ).

INTRODUCTION

At the Higher Technical School of Architecture in the University of Valladolid we have been researching into the study of ventilation in houses for several years. This study has emerged to a large extent due to the new legislation on Indoor Air Quality (IAQ) introduced by Spain's "Technical Building Code" in which we have been involved.

The last research project has served as the basis to study efficiency in areas where air extraction in houses is common, such as bathrooms and toilets. This work has also served to define the performance protocols for the remaining rooms in houses.

The ultimate goal of the study is to increase ventilation efficiency in order to seek a reduction in volumetric flow rates with the desired energy savings. The immediate objective for toilets and bathrooms would be to specify design criteria to obtain ideal ventilation for these areas.

METHODS

The experimental model was performed inside a 4.80 x 6.00 x 3.60 m (width, length, height), test chamber designed expressly for this project and fitted with the following equipment and instrumentation:

- to obtain the appropriate volumetric flow rate of the air, centrifugal fans UPE 145/200 cm of 37 W and CAB-125 were used. To provide a homogeneous mixing of the gas tracer inside the enclosure, another independent fan BSH-PAE C_VT01 of 30 W was used.
- to regulate the volumetric flow rate of the air, a frequency inverter was used, and to control it two digital anemometers RS 180-7111 and a nozzle TG-40 of calibration were used.
- to control the airtightness of the room the fans were in use with the same frequency inverter and a manometer for measuring the pressure of the interior of the chamber.
- to monitor air movement, we used a smoke machine, model Fz-1200 of 1.2 kW and 500W halogen cycle projectors, together with a video camera for recording.
- to measure air age, we used equipment with a gas analyzer model 1302, a sampler-doser model 1303 and sulphur hexafluoride as tracer gas, managed by 7620 software on a PC computer.

Figures 1, 2 and 3 respectively show a general outside view of the test chamber, the subchamber that represents the bath, and the most important instrumentation, comprising the multi-gas monitor with the multipoint sampler and doser.

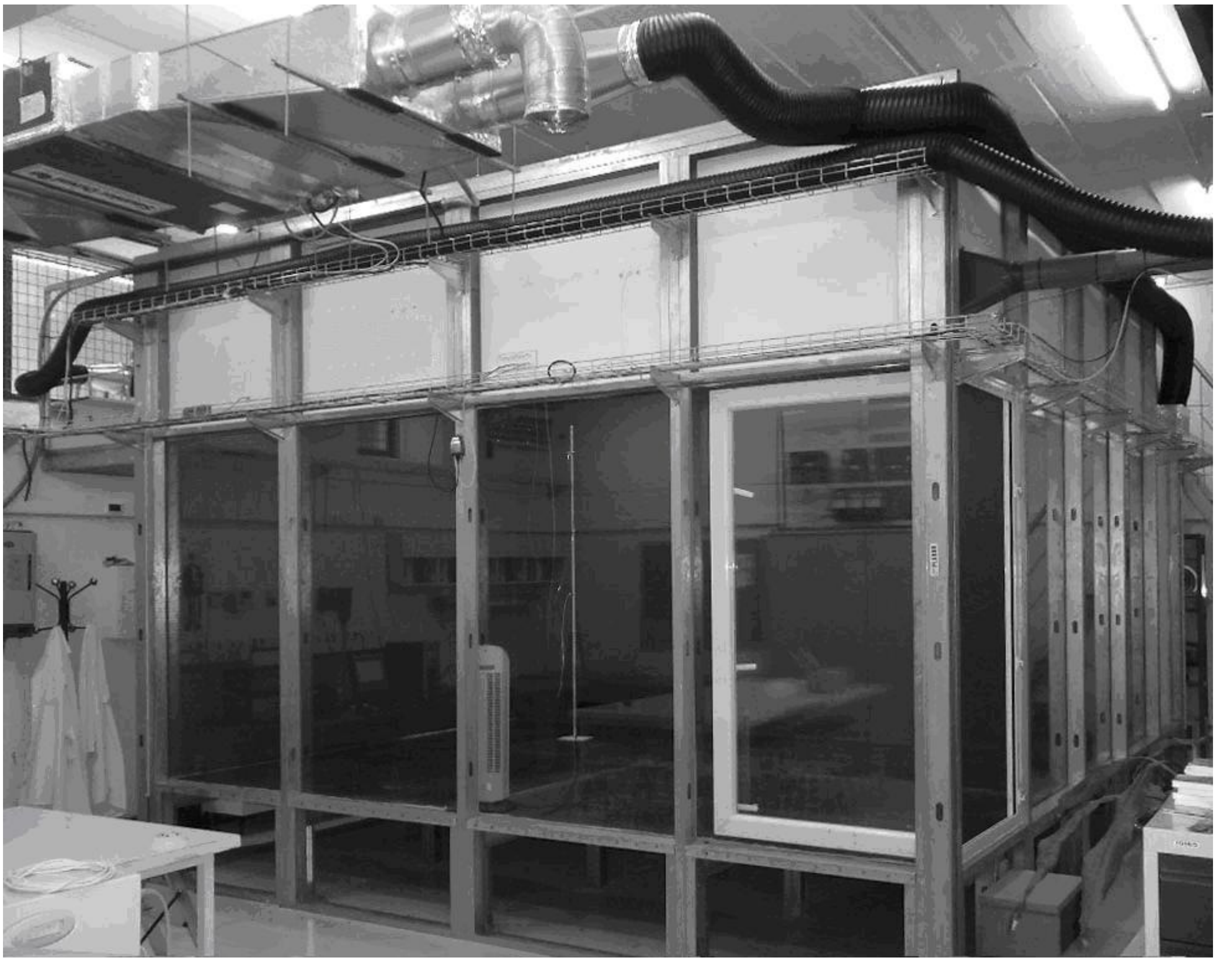

Figure 1. Test Chamber. Exterior view from the air conditioning laboratory.

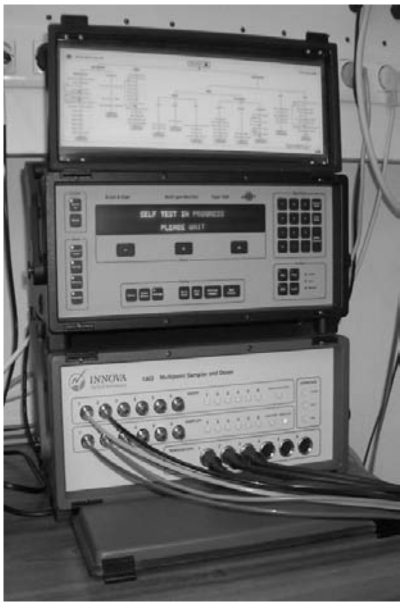

Figure 2. Subchamber reproducing a bathroom. View from the interior of the chamber.
Figure 3. Equipment with multipoint sampler and doser and multi-gas monitor.

The numerical model was carried out using Fluent CFD software with help of the Gambit program for the introduction of the geometry and the meshing of the bath. The process for each test after CFD software adjustment was the following:

• introduction of the geometry, meshing and contour constraints, and the turbulence model.
• validation of the numerical model for comparison with the results of the experimental model, through non-dimensional values.
• numerical study of variants on the same volume and the obtained results.
• graphical results.

PROCESS

We study a prototype model bathroom with a maximum number of alternatives for the distribution of the supply and exhaust openings for the air. The inlet was located in the lower part of the access door and the outlet at the top of the walls. The different distributions were accompanied by the corresponding variations in the locations of the sanitary systems.

Two procedures were employed: the numerical procedure with CFD software (Computational Fluid Dynamics) and the experimental procedure with the mentioned chamber where the size of the bathroom with a specific distribution was reproduced on a real scale. The former allows greater versatility when performing a virtual study of multiple cases, while the second, which is more complex, provides the essential validation of the study in CFD. In both cases the procedure used to calculate ventilation efficiency is based on the expression relating room air age, represented in Equation (1):

$$\varepsilon = \frac{\tau_{exhaust}}{2.\bar{\tau}}, \qquad (1)$$

where ε is the efficiency of the ventilation in the studied room, $\bar{\tau}$ is the room-average age and $\tau_{exhaust}$ is the local mean air age of the outlet.

Since air was entering from other rooms in the house, we were dealing with a common isothermal process, both for the experimental and numerical models. The experimental model applied the concentration decay method using a gas tracer (SF_6) and the averaged data of concentrations obtained were treated in detail by the appropriate spreadsheets.

Figure 4 represents a typical graph showing the evolution of the concentration decay of a gas tracer obtained in one of the numerous tests performed. Conversion from concentrations of gas tracer to mean air ages was carried out by equations (2) and (3).

The experimental results allowed us to validate the results obtained with the numerical model, built with a turbulence model k - ε. The left side of Figure 5 shows a perspective of the model bathroom. The right shows two sections with the mean air ages represented by different colour plots to indicate different ages in seconds.

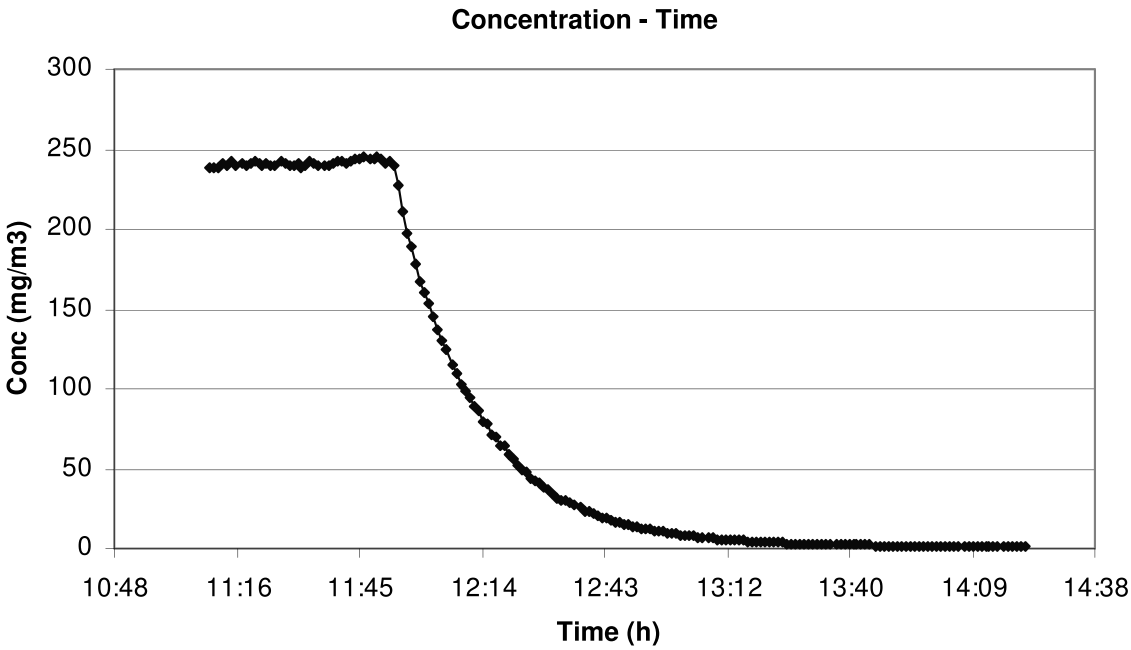

Figure 4. Example of obtained concentration history with SF_6 as tracer gas

Local mean age of air $$\bar{\tau}_p = \int_0^\infty \frac{C_p(r_o,t)}{C(0)} d\tau , \qquad (2)$$

Room average age of air $$\tau = \frac{\int_0^\infty t \cdot C_e(t)dt}{\int_0^\infty C_e(t)dt} , \qquad (3)$$

where:
$C_p(r_o,t)$ tracer gas concentration at point r_o and time = *t*.

$C(0)$ tracer gas concentration at time = 0, where we begin to measure concentration decay

$C_e(t)$ tracer gas concentration at outlet

τ_p local mean age of air (at a random measurement point)

$\bar{\tau}$ room average age of air

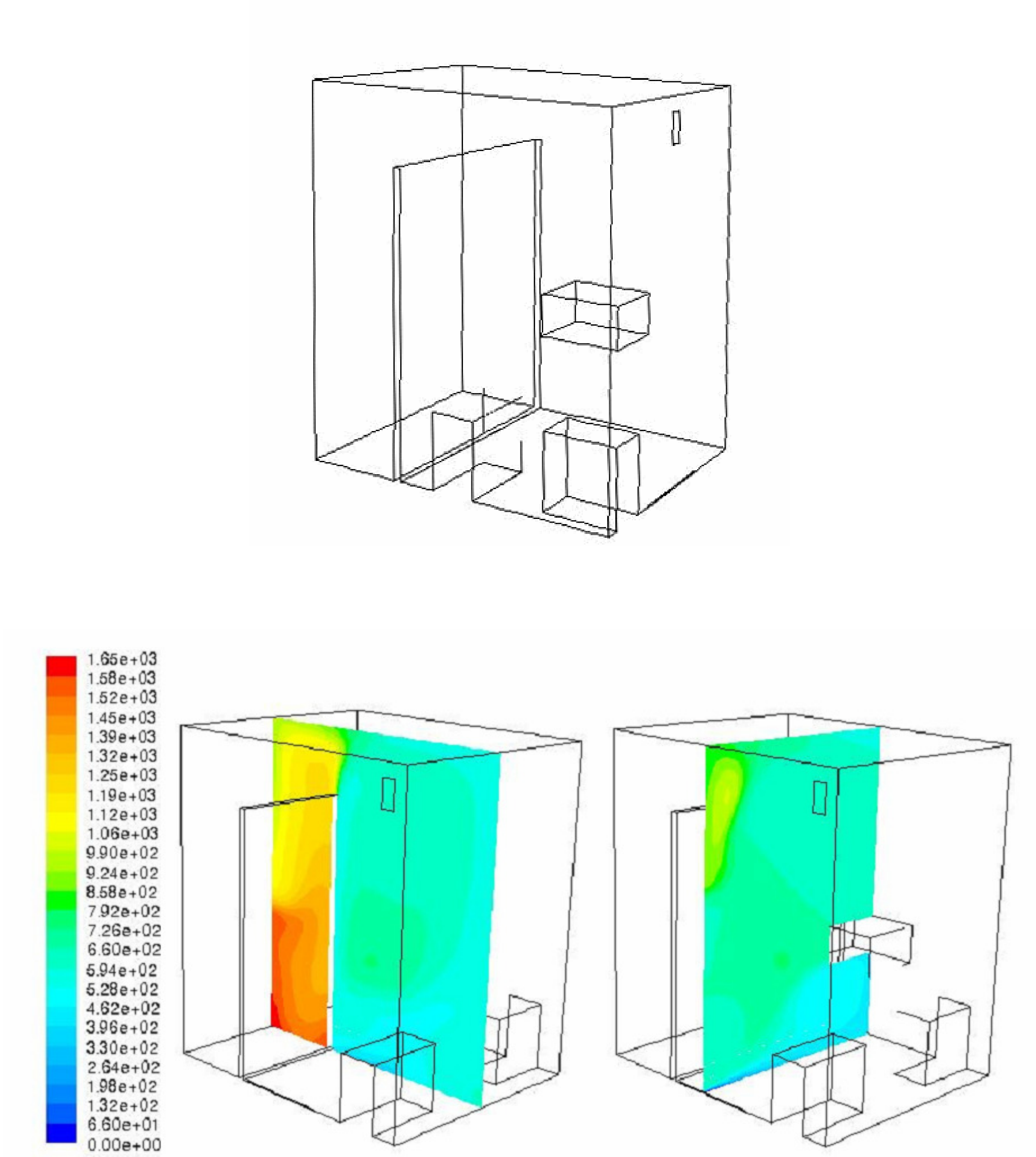

Figure 5. Graphics of the bathroom with graphic results of air ages in certain sections of the room (air age in seconds)

With the obtained results and thinking fundamentally about the possible practical repercussions, efficiencies were sorted into three groups. The first contained values equal to or below 0.45, indicating lower efficiency (near to the theoretical short circuit model), the second group for values above 0.45 and below 0.50, indicating intermediate efficiency (close to the model of perfect mixing) and the third group equal to or higher than values of 0.50, the most efficient (tending to the theoretical model of displacement).

(good) $\varepsilon \geq 0.50$ (average) $0.45 < \varepsilon < 0.50$ (bad) $\varepsilon \leq 0.45$

RESULTS

By way of a summary we now transfer the numerical results to practical concepts that can be applied directly to architectural design. The following symbols represent three valuations of extraction in terms of the above mentioned efficiencies, their location in the perimeter of the bathroom indicating their position near the vertical and corresponding wall.

good location ☐ average location ◧ poor location ■

Figure 6, below, shows the most representative distributions together with the numerical and experimental results obtained corresponding to the first bathroom prototype 2.40 x 1.80 x 2.50m.

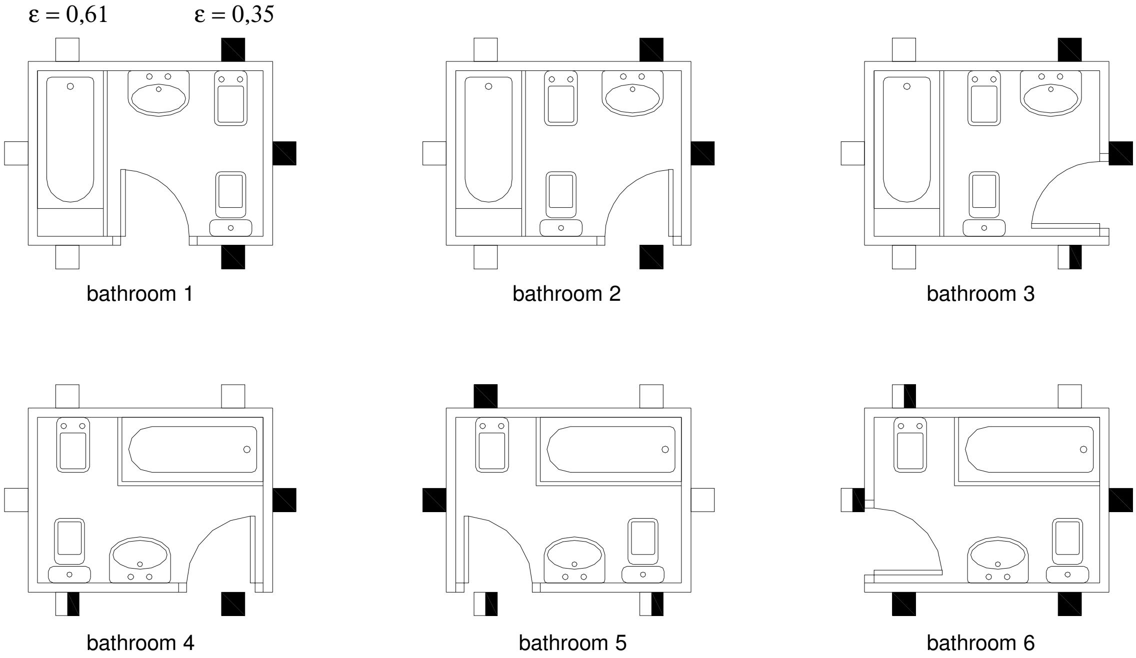

Figure 6. Bathrooms with the same surface and different locations for access doors, sanitary devices and ducts, and the corresponding efficiency for each solution

In two of the variants for the ducts in "*bathroom 1*" (Figure 6), values for efficiencies of 0.61 and 0.35 respectively are shown, coinciding with the maximum and minimum values of all the studied cases.

We also worked numerically with an alternative size of 2.00 x 2.00 x 2.50 m., obtaining a series of results, the most significant of which are shown in Figure 7.

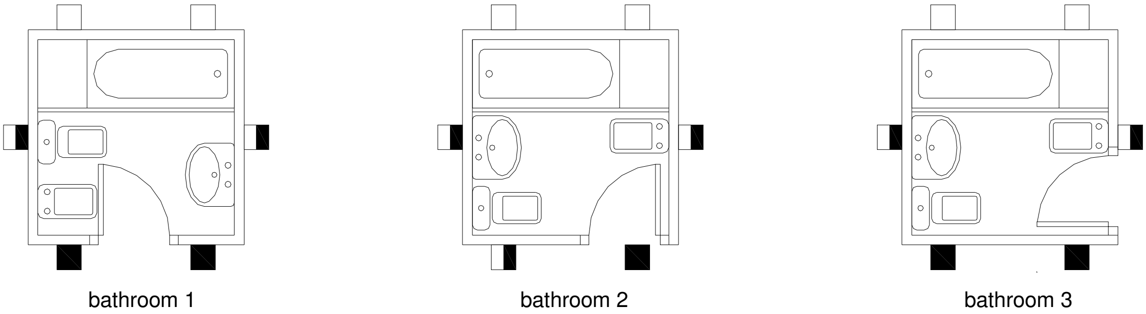

Figure 7. Results of another model of bathroom with other proportions

We even managed to study a conventional 2.00 x 1.50 x 2.50m model of toilet with multiple inlet and outlet variants. One significant example of this study is Figure 8, which offers a perspective of the room with the sanitary devices and a plan with the effects of the distribution.

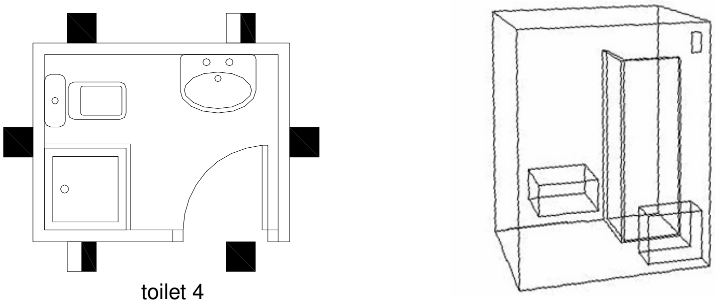

Figure 8. Sample of the results of the toilet

DISCUSSION

The following conclusions concerning architectural design may be drawn from the studies carried out and the results obtained:

- Closing shower spaces and bathtubs with screens, even if they do not reach the ceiling, seriously obstructs ventilation and significantly impacts ventilation efficiency in the toilet or bathroom.
- In general, locating the air outlet close to an entrance door which holds the inlet significantly reduces ventilation efficiency.
- In general, distancing both ducts (inlet and outlet) enhances efficiency, although the key factor is that the outlet should be located on the walls that delimit the shower or bathtub space if these have closed screens.

As pointers for further future research, other conclusions which may also be drawn are:

- The obtained efficiencies in general yielded low values (between 0.35 and 0.61). The induced flow was supposedly nearer to that of displacement, as a result of which theoretically these should have reached efficiency values above 0.7. This encourages further work aimed at improving them.

• In smaller rooms, we seem to have obtained consistently lower efficiencies, between 0.42 and 0.4. It appears that in these cases the location of the inlet and outlet had less impact.

ACKNOWLEDGEMENTS

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Infiltration simulation in a detached house – empirical model validation

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SUMMARY

This study discusses the empirical validation of a multi-zone infiltration model of an existing two-storey detached house in the cold Finnish climate. Empirical validation was performed by comparing simulated and measured pressure conditions of the building during a three-week test period in the heating season. The simulations were carried out using a dynamic simulation tool, IDA-ICE, which combines whole-building energy simulation and infiltration modelling. The initial data of the building model were obtained with extensive field measurements including the measurements of airtightness and leakage distribution of the envelope and performance of a ventilation system. According to the results, pressure conditions are slightly negative on the first floor and positive on the second due to the stack effect in the heating season. The empirical validation shows that the correspondence between the simulated and measured pressure conditions is good and the studied building model can be used for the infiltration and energy analyses.

INTRODUCTION

This study is part of the cooperation project between two Finnish universities; Tampere University of Technology and Helsinki University of Technology. In this project, called *Tightness, indoor air and energy efficiency of residential buildings* (AISE), extensive field measurements are being carried out in 159 dwellings with massive and lightweight structures consisting of flats and detached houses during the period 2005 – 2007. This study is concentrated on the infiltration modelling of one measured detached house. Infiltration, defined as uncontrolled air flows through a building envelope, depends on the air permeability of the structures and the pressure difference between indoor and outdoor air. The pressure difference is caused by wind, temperature difference over an envelope, and balance of the ventilation system. Wind conditions are strongly dependent on the building site and temperature-driven pressure difference is significant in a cold climate during the heating season. The objective of the study is to perform an empirical validation of the detailed multi-zone simulation model, including leakage distribution and other important parameters, with the intention of using the validated model for infiltration and energy analyses.

METHODS

Building description

The object of the study is a detached house comprising two floors (Figure 1). The house is situated in the metropolitan area of Helsinki and was built in 2000. The net floor area of the building is 172 m². The structures of the house are wood-frame construction provided with a plastic vapour barrier and the base floor of the house is a concrete slab on the ground. The level of thermal insulation of the house fulfils the requirements of the Finnish building code [1] and the house is equipped with a mechanical supply and exhaust ventilation system with heat recovery.

Figure 1 The object of the study is a typical Finnish detached house.

The method of construction, the ventilation system and the tightness of the modelling object corresponds to the typical detached house defined by the national project “Moisture-proof healthy detached house” [2], where 102 newly built timber-framed detached houses were measured in Finland during 2002-2004. Airtightness of the building envelope equals the mean level of the measured 102 detached houses.

Measurements

Initial data of the simulation model were collected by means of field measurements of the building. The following factors were measured:

- Ventilation
- Tightness of the envelope
- Leakage distribution
- Pressure and thermal conditions.

The measurements of the ventilation air flow rates, tightness and leakage distribution of the building were carried out using a single-shot measurement. Supply and return air flow rates were measured at the low (3/8) speed of the air-handling unit, which is the normal use of ventilation in the studied building during wintertime. The measured air change rate of the building was 0.3 ach and the ratio of the supply and return air flow rates was 0.93, affecting a slight negative pressure in the building. The measured air change rate is lower than the minimum requirement of the Finnish building code (0.5 ach) [3] and also slightly lower than the typical mean air change rate of the detached houses (0.41 ach) in wintertime when equipped with mechanical supply and exhaust ventilation system [4].

The tightness of the building was measured using a fan pressurization method. To measure the air leakage of the envelope, all the exterior openings – windows and doors – were closed and ventilation ducts and chimney were sealed; a leakage air change rate per hour at 50 Pa of pressure difference was measured. The leakage air flow rate was divided by the internal volume of the building to get the building leakage rate n_{50}; the resultant was 3.9 ach. The distribution of the leakage openings was studied using two-phase infrared-camera imaging of the envelope of the building [5]. The imaging was performed inside the building in a normal and 50 Pa under pressure conditions. The study was carried out during the heating season when the indoor and outdoor temperature difference was 25°C. According to an estimated vertical leakage distribution shown in Table 1, most of the leakage routes are at a junction of the roof and an intermediate floor; the routes are quite evenly distributed between the first and second floor.

Table 1 Vertical leakage distribution of the building based on the infrared camera imaging

Place of the leakage routes		Leakage distribution, %
2nd floor	Junction of roof	36
	Upper edge of window frame	4
	Lower edge of window frame	4
	Junction of intermediate floor	2
1st floor	Junction of intermediate floor	21
	Upper edge of window	0
	Lower edge of window frame	24
	Junction of base floor	10

Pressure and thermal conditions of the building were measured during a three-week period in the heating season. Occupants were living normally in the building during the follow-up measurements, which were carried out between 4th and 24th of March 2005. Pressure and temperature data were collected with loggers using a 5-minute time-step. The pressure difference over the envelope was measured on the first and second floor of the building. The measuring points were at one façade of the detached house; the principle of the measurement is shown in Figure 2.

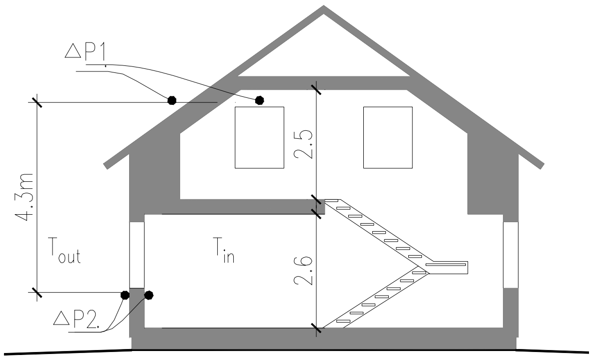


Figure 2 Measurement of the pressure conditions of the building.

Indoor air temperatures of the building were measured in a living room on the first floor and in a bedroom on the second floor. The outdoor air temperature was also measured next to the detached house during the follow-up measurements. The measured outdoor air temperature is shown in Figure 3; it is -7.1°C on average. The three-week period was very cold for Southern Finland, emphasizing the temperature-driven pressure difference over the building envelope.

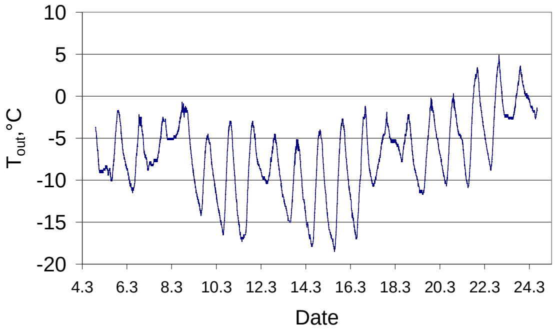
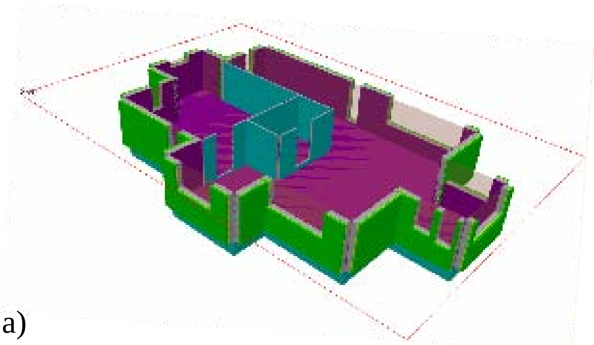


Figure 3 Measured outdoor temperature next to the detached house during follow-up measurements.

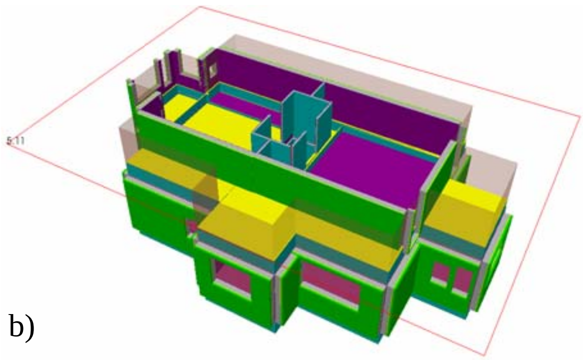
The dynamic simulation model

The building model was done using IDA indoor Climate and Energy 3.0 (IDA-ICE) building simulation software. This software allows the modelling of a multi-zone building, HVAC-systems, internal and solar loads, outdoor climate, etc. and provides simultaneous dynamic simulation of heat transfer and air flows. It is a suitable tool for the simulation of thermal comfort, indoor air quality, infiltration and energy consumption in complex buildings. A modular simulation application, IDA simulation environment and IDA-ICE, has originally been developed by the Division of Building Services Engineering, KTH, and the Swedish Institute of Applied Mathematics, ITM [6,7]. Today the application is commercial tool owned by EQUA AB. IDA ICE has reached high levels of penetration among practitioners and researchers in Sweden and Finland

The building model of the measured detached house comprises three different zones on the first and second floor (see Figure 4); the floors are connected by means of a staircase. Air flow between the zones, two floors and outdoors caused by the pressure differences is simulated by means of the principle of nodal network (see Figure 5.), where flow paths, cracks or openings between the zones or outdoors are described as flow resistances.



a)



b)

Figure 4 Profile of the 3-D plot showing the first floor (a) and the second (b) of the IDA-ICE building model. The figures are generated with the IDA-ICE 4.0 alpha version.

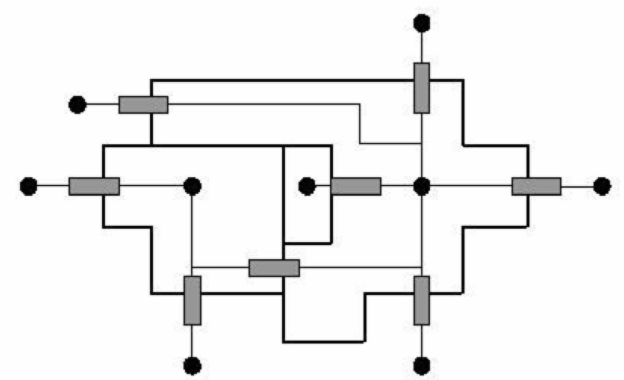


Figure 5 A simplification of the nodal network model of the first floor. Grey boxes describe the flow resistances of the flow paths; the pressures are calculated at the black nodes.

The building model was simulated using the hourly weather data of the three-week measurement period. The measured outdoor temperature data was used and the other required data, for example, wind velocity and direction and solar radiation properties, were taken from the weather station of Helsinki-Vantaa airport. Wind conditions of the environment were simulated using the wind-profile equation [8]

$$U(h) = U_m \cdot k \cdot \left(\frac{h}{h_m}\right)^a, \qquad (1)$$

where U(h) is wind speed at height h and U_m is wind speed measured in open country at the weather station, h is height from the surface of the ground, h_m is height of the measurement equipment and parameters k and a are terrain-dependent constants. The simulated building is in a typical Finnish suburban area with closely built houses where the height of adjacent buildings is approximately the same as the simulated one. The wind-profile equation was simulated with the values of the parameters (k =0.67 and a = 0.25). Wind-induced pressure conditions were simulated using constant wind-pressure coefficients defined at 45° intervals of a wind direction. The values of the wind-pressure coefficients used in the simulation were approximate values for typical detached houses with a height of up to three storeys [9]. Wind pressure outside the building façades was determined with the equation

$$P_w = \frac{1}{2}\rho \cdot c_w \cdot U^2, \qquad (2)$$

where ρ is outdoor air density and c_w is the wind pressure coefficient and U is the local wind velocity defined by Equation (1). The air infiltration is calculated for every façade; the connection between the indoor and the outdoor climate, as well as the connection between the zones, is simulated using bi-directional leakage openings, which are so-called flow resistances of the kind shown in Figure 5. The leakage openings were distributed over the building model according to the measured leakage distribution shown in Table 1. Air flow through the leakage opening is simulated in the building model with the widely used empirical power law equation [10]

$$Q = C \cdot \Delta P^n, \qquad (3)$$

where C is a flow coefficient that is related to the size of the opening and ΔP is the pressure difference across the opening, n is a flow exponent characterizing the flow regime. The flow exponent varies in value from 0.5 for fully turbulent flow to 1.0 for completely laminar flow. The total value of the flow coefficient and the mean value of the exponent for the whole building envelope were determined by the pressurization test.

RESULTS

The results of the three-week empirical validation period are discussed below. The simulated pressure conditions of the first and second floor of the detached house are compared against the measurement results. The simulated pressure differences over the envelope were logged from the building model at the same façade and the same height from the ground as in the measurements; the thermal conditions inside the building model correspond to the measured conditions.

According to the measurements and the simulation, the pressure difference is slightly negative on average on the first floor and positive on the second floor, see Figure 6 (a and b). The model gives some intermittent peaks of the negative pressure that clearly deviate from the measured pressure, but the correspondence is good on average. The pressure difference as a function of temperature difference between indoor and outdoor air is shown in Figure 6 (c and d). Almost completely parallel linear fits of the measured and simulated points show that the pressure difference decreases with an increasing temperature difference on the first floor and that the pressure difference is almost constant on the second floor. The different correlation between the pressure and the temperature differences on the first and second floor is a result of many factors, such as the stack effect, the ratio of the ventilation air flows, and the performance of air handling unit at the low outdoor air temperatures.

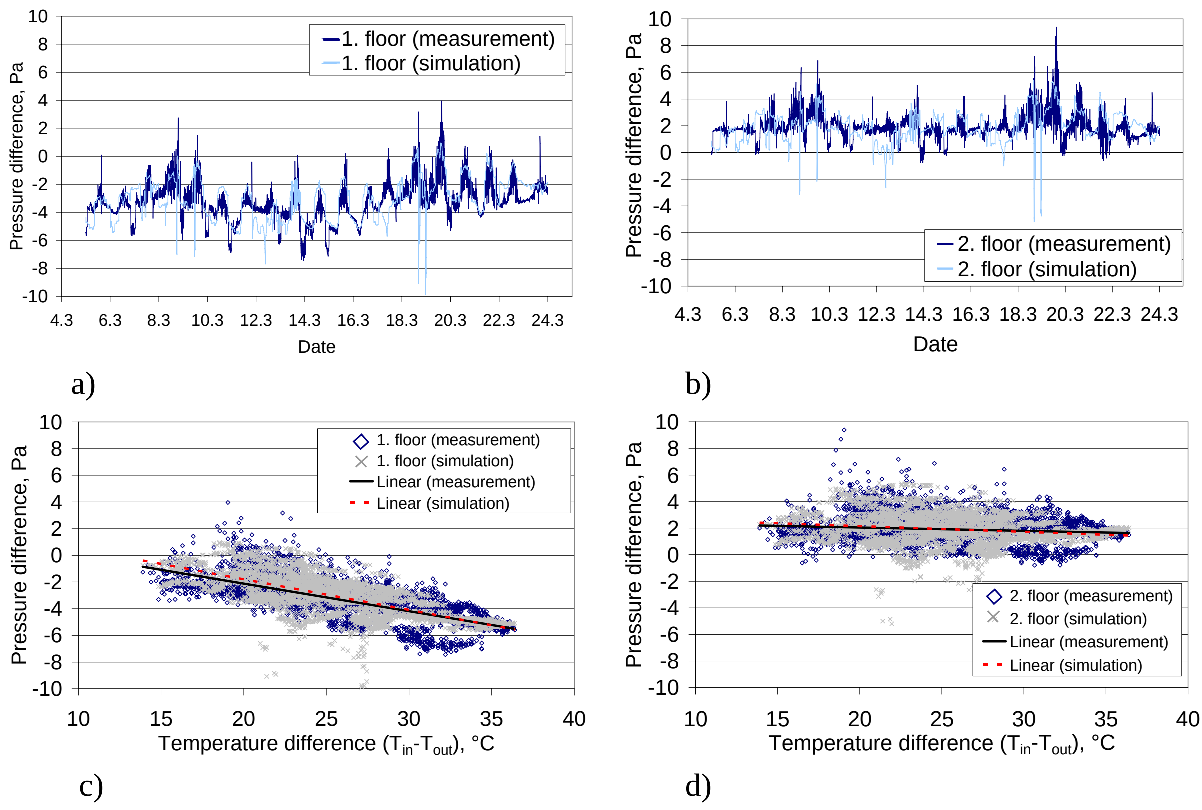


Figure 6 Measured and simulated pressure conditions of the detached house during the three-week measurement period from 4th to 24th of March 2005. The pressure difference over the envelope on the first and second floor are shown as a function of time (a and b) and as a function of temperature difference between indoors and outdoors (c and d).

The measured and simulated average indoor air temperatures and pressure differences of the validation period are almost equal (see Table 2). The set point of heating in the building model is chosen so that the average indoor air temperatures are as close to the measurement result as possible. Indoor air temperatures of the building are rather low, especially on the second floor, due to the occupant's way of life.

Table 2 Measured and simulated average indoor air temperatures and pressure differences across the envelope during the validation period.

Method	Indoor air temperature, °C		Pressure difference, Pa	
	1st floor	2nd floor	1st floor	2nd floor
Measurement	19.7	17.0	-3.3	1.9
Simulation	19.8	17.0	-3.1	1.9

DISCUSSION

The empirical validation of the studied building model shows that the simulated pressure conditions are in good agreement with the measurements. Because of this agreement, the multi-zone model can be used for the infiltration and energy analyses in the cold Finnish climate. Simulated pressure conditions of the building model seem to be realistic, even if the calculation of the wind-induced pressure conditions was simplified. The wind data used in the simulation was taken from the airport's weather station; the wind-pressure coefficients were approximate values for typical low-rise buildings in the sheltered environment and the values were not based on the measurements or CFD-calculations of the studied building. The empirical validation of the model was carried out for the cold period of the winter season when the buoyancy-driven pressure difference was emphasized and the studied building was situated in the sheltered suburban area where wind conditions were not strong. It is obvious that an accurate modelling of the leakage distribution is important, especially when buoyancy-driven pressure is significant.

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Air – tightness measurements of forty residential houses in Athens, Greece

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SUMMARY

Regular air tightness and infiltration measurements were performed in forty houses, in the area of Attica, Greece. Two measurement methods were used, the tracer gas decay method and the Blower Door tests method. Blower Door measurements were done in accordance with EN ISO 13829 [1]. Ambient conditions and temperature fluctuations inside the houses were measured as well.

A classification of houses examined, based on experiments' results was acted out in accordance with EN ISO 13790 [2]. The houses were classified into three air tightness categories, in regard to their air tightness in natural conditions and at a pressure difference of 50 Pa.

Furthermore, the total frame length was estimated for the whole housing stock, and a correlation between the air tightness measurements at a pressure difference of 50Pa and the total frame length was examined, for the sample of buildings and for each air tightness category.

A correlation between the airflow values, as they resulted from the fan pressurization method and the average infiltration rates, calculated by the tracer gas experiment results, has been extracted. Moreover, the effect of climate data including temperature and windiness and construction quality on the houses' infiltration characteristics has been investigated.

INTRODUCTION

Infiltration rates and therefore buildings' air tightness are important because they affect the energy use of the building and they impact the transport of pollutants so as the indoor air quality. From an energy standpoint alone it is almost always desirable to increase air tightness but indoor air quality may suffer [3].

Single-family dwellings in Greece are usually not equipped with mechanical ventilation system. When doors and windows are closed, air exchange occurs only by uncontrolled air leakage across the building envelope. The movement of air through leaks, cracks or other openings of the building envelope is known as air infiltration. High infiltration rates can cause excessive energy demand because of the need to condition the infiltrating air. At the other hand, insufficient air exchange can lead to high exposure to pollutants such as emissions from building materials, cooking, smoking or cleaning activities [4]. It is very important to estimate leakage characteristics in the housing stock in terms of construction's type and quality, buildings' age and climatic conditions, in order to control the energy demand and the pollutants' concentration.

Tens of thousands of fan pressurization measurements have been made in USA buildings. These data were collected and were analyzed in order to determine relevant leakage characteristics in the US housing stock in terms of construction type and quality, region and age [4], [5].
Very few information is available regarding infiltration of buildings in the Mediterranean area. In Italy, the issue building air tightness is still not widespread and ACH limits at 50 Pa pressure difference have not been introduced yet. However, many air tightness measurements were performed in residential buildings, last decade [6]. In France, air tightness measurement studies were performed too, in order to identify the major factors that affect the buildings' leakage area and improve the energy performance of buildings [7].
Tracer gas measurements were performed in order to evaluate the infiltration rates for each building. Blower door measurements were also performed but the data of this method could not be generally used to estimate airflows at natural conditions. The name comes from the fact that in the common utilization of the technology there is a fan mounted in a door [3]. Blower door data estimates airflows at a variety of pressures and mostly at a 50 Pa pressure difference. The advantage of this method is that their results are less affected by climatic conditions.

2. METHODS

2.1 Description of the residences

Regular infiltration and air-tightness measurements were performed in forty residences in the area of Attica, Greece. All forty residences are single-family buildings or double-family buildings and their entrance door is totally exposed to the exterior environment. Table 1 contains information about all the tested houses.

2.2 Measurements of ambient conditions and indoor temperature fluctuation

Ambient conditions were measured using the mobile meteorological station of Group of Environmental Studies of University of Athens. During experiments, ambient temperature, wind speed and wind direction were being traced, at 10m height and at building's average height above the ground. Outdoor dry bulb temperature at the level of the main entrance and indoor temperature in three different building zones were being traced.
Table 1 contains the average values of all parameters that were being traced over the time period of each experiment.

Table 1. Building characteristics ambient conditions and internal temperature fluctuation.

House	**Volume (m^3)**	**Average ambient temperature (ºK)**	**Average indoor temperature (ºK)**	**Wind speed at 10m height (m/sec)**	**House**	**Volume (m^3)**	**Average ambient temperature (ºK)**	**Average indoor temperature (ºK)**	**Wind speed at 10m height (m/sec)**
1	565	302.2	301.1	1.95	21	458	300.0	302.2	5.23
2	445	300.2	299.2	4.15	22	149	300.8	303.1	6.24
3	444	301.4	300.9	3.75	23	156	300.9	303.7	6.24
4	120	302.3	301.5	2.58	24	485	304.6	305.1	3.52
5	168	301.6	302.2	3.46	25	247	295.3	294.6	1.95
6	573	300.2	302.5	1.58	26	335	295.2	295.0	1.65
7	149	304.4	301.9	3.79	27	372	297.2	300.6	2.07

8	97	305.1	306.4	4.02	28	698	294.9	294.9	1.07
9	147	306.1	304.2	3.22	29	227	294.0	293.2	4.02
10	261	306.1	300.8	1.24	30	189	293.7	293.0	3.25
11	247	308.1	300.7	1.95	31	154	294.2	293.2	4.02
12	691	305.6	301.2	0.81	32	440	292.6	288.8	2.78
13	390	300.0	299.9	1.77	33	181	292.3	286.7	3.67
14	191	298.9	301.4	1.21	34	301	291.2	289.1	3.11
15	500	300.6	297.2	1.50	35	462	294.9	293.0	1.22
16	458	299.9	298.0	4.18	36	148	293.7	291.5	0.69
17	828	295.6	296.6	2.10	37	139	293.1	291.9	0.43
18	205	294.2	295.6	1.12	38	388	287.7	289.3	2.95
19	116	293.9	294.5	3.96	39	274	283.3	293.4	4.45
20	141	293.5	296.1	3.92	40	272	285.3	293.0	3.92

2.3 Infiltration measurements using the tracer gas "decay" method

Infiltration rate was determined by using the "concentration – decay" method. The tracer gas equipment consists of a central unit that controls the gas injection and sampling, an infrared radiation detector and a gas bottle. Mixing was ensured by the use of fans. The (inert) gas indicator that was used was N_2O.
Tracer gas was injected into each of the four building zones through four separate tubes - channels, while all house openings remained closed. As soon as the inert gas – indoor air mixing was complete and N_2O concentration target value was reached, the evolution of the gas concentration in each zone was measured.
The infiltration rate, in air changes per hour (ACH), for each of the four building zones that were tested as well as for the entire building (ACH_{tg}), was calculated by using the traced concentration values.
Table 2 contains the average infiltration rate of the whole building, for all studied houses.

2.4 Air tightness measurements using the fan pressurization method

Each building's air tightness was measured using a Blower Door in accordance with EN ISO 13829.
Blower door has a variable speed fan so that the pressure difference can be adjusted and an aluminum frame in order to seal the fan tightly into the doorjamb. The system is mounted in each house entrance door to measure the leakiness of the house. In order to measure the leakiness of the house, the blower door measures both the airflow through the fan and the pressure difference between the house inside and outside. Measurements are taken by increasing the speed of the fan until the pressure difference between the house and outside is at the desired level. Typically, testing is done between 20 and 70 Pascals (Pa). The airflow out of the house at that pressure is then recorded.
Measurements were carried out following method A – common building use – of EN ISO 13829 while also applying a pressure difference of 50Pa in order to fulfill the requirements of EN ISO 13790 [2] (former EN ISO 832).
All experiments' results are acceptable as they fulfill EN ISO 13829 criteria which require that the wind speed is lower than 6m/sec, the product of maximum building height (m) and temperature difference between outdoor and indoor dry bulb temperature is lower than 500m°K and the building's volume is lower than $4000m^3$.
Table 2 contains the calculated air changes per hour values for a 50Pa pressure differential (ACH50) and the criteria parameters values.

Table 2. Average values of infiltration rate (ACH_{tg}) and air tightness (ACH_{50}) of the buildings.

House	ACHav	ACH50	House	ACHav	ACH50
1	0.71	1.87	21	0.99	7.40
2	1.21	5.72	22	2.38	6.33
3	0.97	5.4	23	0.74	2.18
4	1.14	8.52	24	0.51	8.80
5	1.56	11.3	25	1.10	7.16
6	1.14	2.22	26	0.62	3.44
7	0.35	7.51	27	1.02	2.30
8	0.33	8.5	28	0.55	10.69
9	0.83	11.12	29	0.23	7.34
10	0.99	9.58	30	0.99	7.20
11	0.86	8.86	31	0.95	10.10
12	0.62	1.98	32	1.16	13.10
13	0.59	2.44	33	0.81	5.94
14	0.71	2.69	34	0.36	5.06
15	1.38	10.49	35	0.50	8.51
16	1.46	8.29	36	0.22	5.30
17	0.49	6.39	37	0.77	10.20
18	0.32	7.68	38	0.65	3.50
19	0.90	5.94	39	0.52	2.95
20	0.40	9.3	40	0.50	4.50

3. RESULTS

3.1 Classification of houses examined based on experiments' results

Tracer gas decay and pressurization test method were used in order to estimate the infiltration rate under natural conditions and the air tightness of the building respectively. According to EN ISO 13790 (former 832), house buildings can be classified into three categories in regard to their air tightness under natural conditions (infiltration) or at a pressure difference of 50Pa (ACH50) between indoor and outdoor air.
Table 3 shows the limit values (ACH_{tg}) of three air tightness levels for non shielded, naturally ventilated single family buildings with more than one exposed façade and their corresponding values when a 50 Pa pressure difference is applied between indoor and outdoor environment (ACH_{50}).

Table 3. Tightness levels for natural ventilated, non shielded single-family buildings

Air change rate (h^{-1}) at 50Pa	Ventilation rate ((h^{-1}) for naturally ventilated single family houses	Envelope tightness level
10	1.5	Low
4 – 10	0.8	Medium
4	0.5	High

Figure 1 and 2 contain the rating of each tested building, resulting from tracer gas decay and fan pressurization measurements.

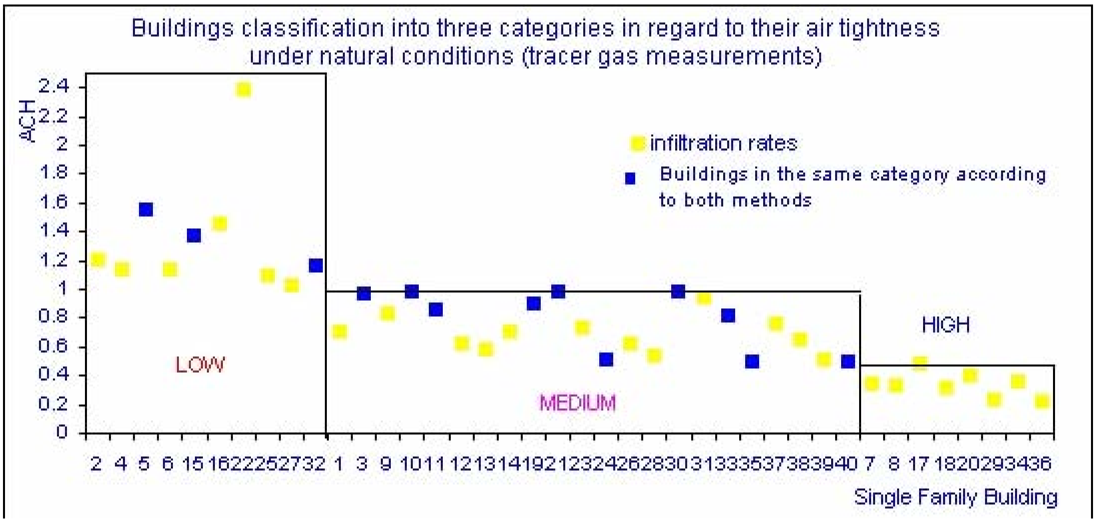


Figure 1. Building classification into three categories in regard to their infiltration under natural conditions (measurements with tracer gas “decay” method)

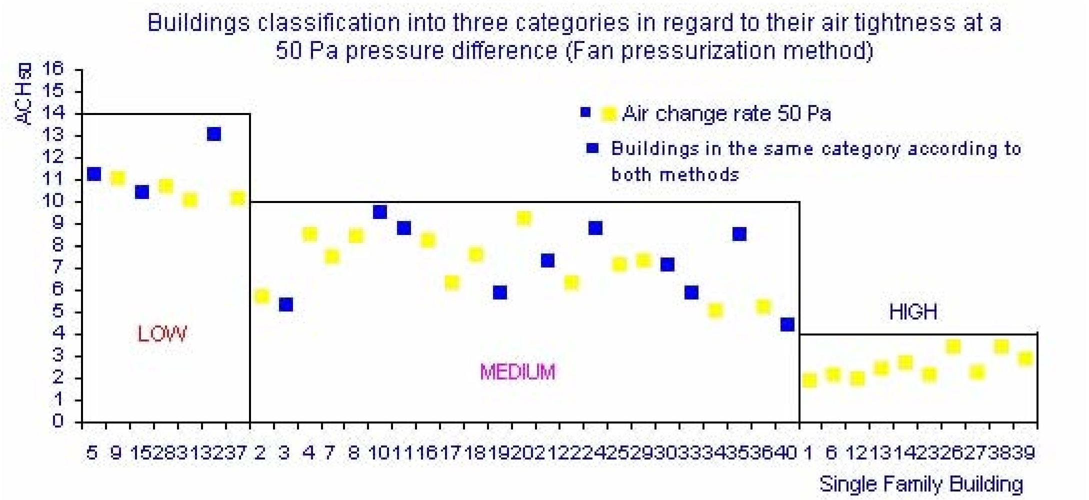


Figure 2. Building classification into three categories in regard to their air tightness at a 50 Pa pressure difference (Fan pressurization method)

3.2 Correlation between air tightness measurements and total frame length

One important factor associated with air tightness of a building is the total window frame length divided by the building’s net volume. The total window frame length was estimated for each building, it divided by the net volume and the “frame length factor (FLF)” was defined.

$$FLF = (TFL) / (NV), \qquad (1)$$

where *FLF* is the frame length factor, *TFL* is the total window frame length and *NV* is the net volume of the building.
Linear correlation between the air tightness measurements at a pressure difference of 50 Pa and the frame length factor FLF was performed for each one of three air tightness levels (high, medium, low).

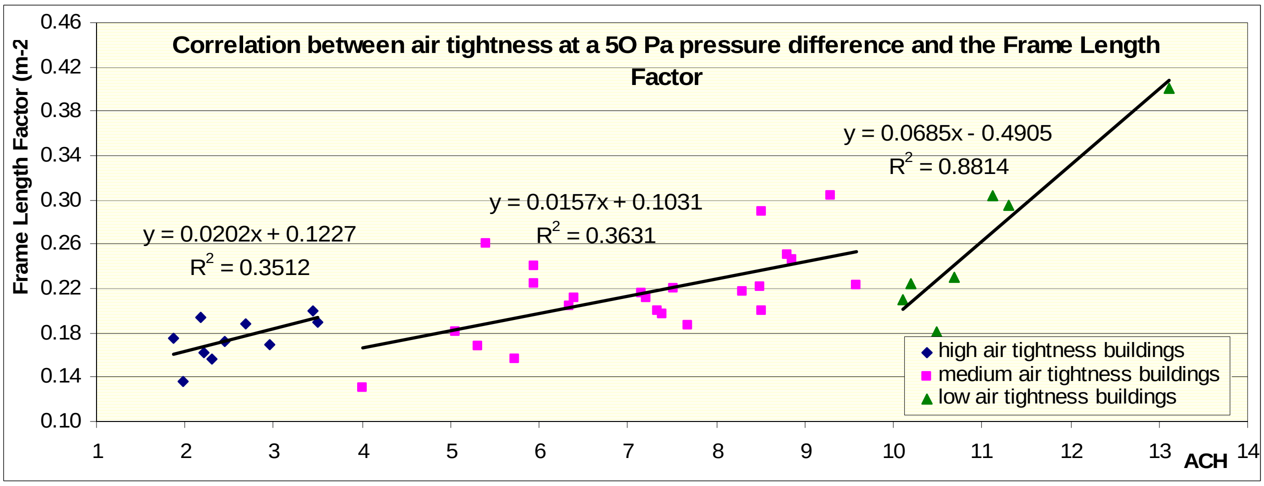


Figure 3. Linear correlation between air tightness at 50 Pa pressure difference and the total frame length factor for three air tightness categories

As it can be seen at Figure 3, the linear correlation for the “high” and the “medium” air tightness levels give lower R^2 values compared with high value of $R^2 = 0.88$ of “low” air tightness buildings. In case of “low air tightness” houses, the most important linear correlation between FLF and air tightness measurements is noticed, while this category is mostly affected by the total window frame length.

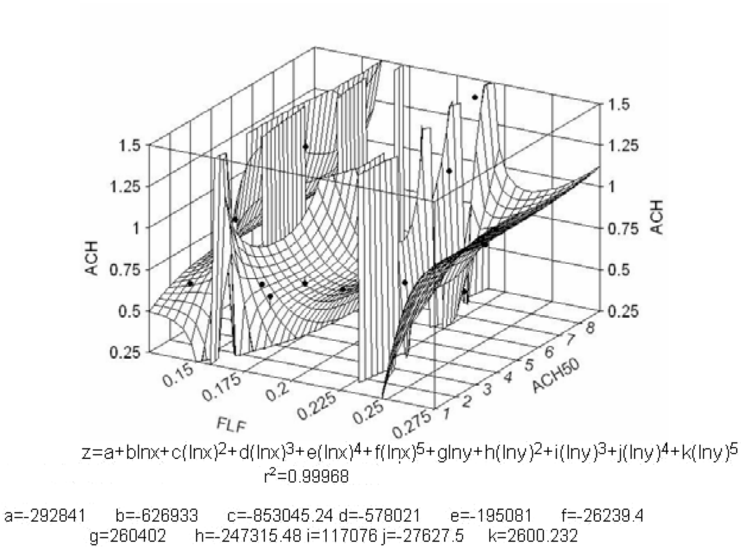
However, the sample of buildings is small and the correlation should be further checked in many cases.

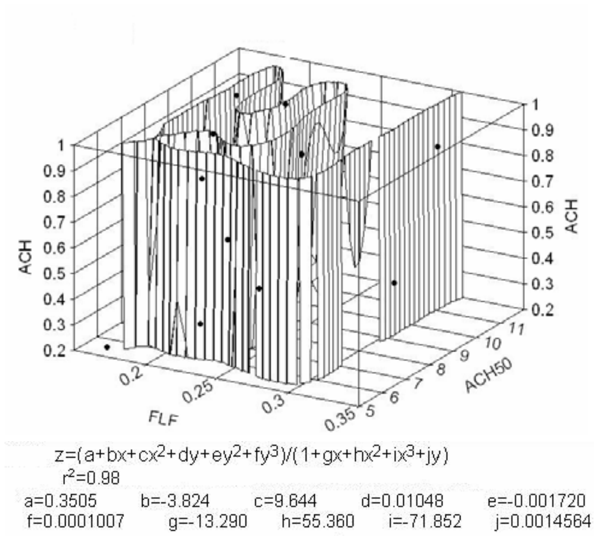
3.3 Correlation between the average infiltration rates, the “fan pressurization method” results and the Frame Length factor

A correlation between the airflow values, as they resulted from the fan pressurization method, the average infiltration rates, calculated by the tracer gas experiment results and the Frame Length Factor has been extracted.

All tested houses were classified in regard to their frame type. The most common frame types of the examined houses are those with wooden frame and those with aluminium sliding frames.

A correlation between the two method’s results was investigated for each “frame type” category. The correlation between three parameters gives the best results in case of buildings with wooden frame ($R^2 = 0.99$), and with aluminum frame ($R^2 = 0.98$) (Figure 4).





Buildings with wooden window frame Buildings with aluminum window frame

Figure 4. Correlation between infiltration rates (ACH), air tightness at 50 Pa pressure difference (ACH_{50}) and the total frame length factor (FLF) for each frame type category.

The effect of climatic data in air infiltration was investigated for each “window frame type” category. For the case of the wooden window frames (R^2=0.80) the effect of temperature difference and wind speed in air infiltration is bigger than in the case of sliding aluminium frames (R^2=0.75). However, the forty buildings have significant differences in their construction’s quality as well as in their age, so the climatic conditions’ effect is not clear. It would be desirable to measure the air infiltration for one specific building in different climatic conditions, so that we can have the appropriate results.

DISCUSSION

Air tightness of forty single-family buildings was calculated under natural conditions and under a 50 Pa pressure difference between indoor / outdoor environment. The buildings were rated according to their measured air tightness.

There is no correlation between infiltration measurements at natural conditions and the total window frame length. It is necessary to study the effect of climatic conditions on these infiltration measurements. A correlation between air tightness measurements at a 50 Pa pressure difference and the total window frame length was noticed, mainly at the “low air tightness” buildings. The total frame length affects the air leakage of each house and its air tightness. Also a classification of buildings according to their window frame type was realized, and a correlation between the airflow values, as they resulted from the fan pressurization method and the average infiltration rates, calculated by the tracer gas experiment results, has been extracted for each frame type category. However, the sample of buildings is very small and a large number of experiments are required.

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13 June 2007 at 10:00 - 11:30

A08
Indoor environment in schools

Indoor environmental quality in schools and academic performance of students: studies from 2004 to present

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A pilot study investigating classroom ventilation rates, and their association with reduced student performance was conducted within a school district in the USA in the spring term 2004. Data on classroom CO_2 concentrations (over a 4-5 hr time span within a typical school day) were recorded in 5th grade classrooms in fifty four elementary schools. In addition, investigators were able to work with the district to obtain standardized test scores (math and reading), and background data related to the students in the specific classroom studied in each school. Results of the pilot study have been presented by Shaughnessy et al. 2005, and Shaughnessy et al. 2006, demonstrating a modest association between class room ventilation rates and student performance in math standardized test scores, and also a need for further studies with larger sample size and more comprehensive assessment of indoor environmental quality (IEQ). A new school district in the USA with approximately 50 schools is currently undergoing a through assessment protocol and the preliminary results will be available by the end of 2006. In addition there is a new study in Finland starting in spring 2007 that will provide further knowledge on how and to what extend IEQ in schools affect academic performance of students.

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Ventilation Rates in Schools and Learning Performance

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Introduction

Previous studies underline the often inadequate ventilation rates in classrooms causing increased health risks among school children. Current UK ventilation guidelines specify a minimum of 3 L/s per person and recommend 8 L/s per person ventilation rate achievable in all teaching and learning spaces. Measured ventilation rates in UK primary schools are often below the minimum requirement. There is growing evidence of impairment of learning performance and increased absenteeism due to inadequate ventilation.

The purpose of the present research was to investigate the relationship between pupils' health, well-being and performance, and the indoor air quality in several primary schools in Southern England and to examine the suitability of current air quality guidelines.

Methods

Field surveys were carried out at eight different primary school buildings located in the proximity of Reading, UK. At each school two classrooms were selected for monitoring for three consecutive weeks. The first week was reserved to follow the classroom conditions without any intervention. During the second and third weeks a mobile ventilation system was installed in each classroom that could be set either to provide outdoor air or to re-circulate the classroom air. Carbon dioxide (CO_2) concentration, thermal conditions and other parameters were monitored in both classrooms and outdoors.

Computerised assessment tests and traditional paper-based tasks (addition/subtraction of numbers and reading comprehension) were used to evaluate pupils' performance on both the intervention weeks at low and improved ventilation conditions. Pupils' perceptions of the classroom environment, comfort, general mood and hunger were assessed on subjective scales following the performance tests.

Results

The current project is still in the phase of data collection hence preliminary results of the physical environment and performance tests conducted on paper from only one school are presented.

The fresh air supply in both classrooms was increased from 0.3-0.5 to 13-16 L/s per person due to the interventions. Average CO_2 levels in the classrooms during testing period were reduced from 1600-4000 ppm to 600-800 ppm as a result of increased ventilation conditions.

The performance of all pupils' increased by 5-6% in both the addition ($p<0.036$) and subtraction ($p<0.052$) tasks under improved ventilation compared to low ventilation conditions. These effects were even more significant for the pupils with higher math skills. They increased the number of error-free units by ~7% in both addition ($p<0.02$) and subtraction ($p<0.007$) tasks when working under the improved ventilation conditions.

Conclusions

- The present results strengthen the evidence of earlier findings that improved ventilation has beneficial effects on pupils' learning performance.
- Without intervention the existing ventilation rates in naturally ventilated school buildings remain below the minimum recommended levels unless thermal conditions make people open windows.
- Measures that allow a minimum supply of fresh air to the classrooms of naturally ventilated buildings are needed particularly if windows are not operated adequately by staff to control ventilation.

Study on Productivity in the Classroom (Part3)
Nationwide Questionnaire Survey on the Effects of IEQ on Learning Performance

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1 Introduction

This series of studies evaluate the effects of changes in the air quality and thermal environments on student performance in the classroom. This paper (Part 3) reports the result of nationwide field measurement based on subjective questionnaire surveys and objective test scores in a unified way. In this research, the relationship between the quality of the indoor environmental and student performance in college are linked quantitatively through this nationwide questionnaire survey and previous laboratory experiments. In addition, human psychology regarded to effect it most when learning performance is measured was evaluated "motivation for learning" using a questionnaire as self-assessment form.

2 Outline of questionnaire survey

It investigated 83 branches of a college, which are located in regions spread throughout Japan. The total number of subjects was about 4000. Objective learning performance was evaluated according to scores in standardized quizzes to measure the level of understanding of lectures. In addition to the objective evaluation using quiz scores, a subjective evaluation of learning performance was carried out using a questionnaire as a self-assessment form.

3 Results

Percentage dissatisfied with air environment did not have the correlation with quiz scores but had a significant correlation with a subjective evaluation of learning performance by a linear approximation ($p<0.006$) (Table 1).

Table 1 Objective & Subjective evaluation vs. Percentage Dissatisfied with the air environment

	Objective	Subjective	
	Quiz Score	Time Lost	Predicted rate of Improvement in Performance
Correlation Coefficient	0.23	0.87	0.82
Significance Level	0.50	0.003	0.006

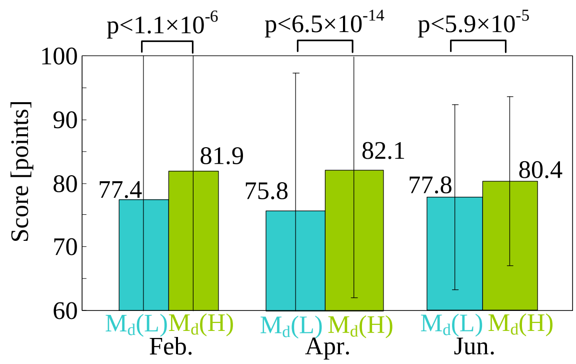

Figure 1 Result for the objective evaluation by motivation group

Figure 1 shows the results for objective learning performance by the motivation group. The quiz scores brought a significant high of 4.5 points ($p<1.1\times10^{-6}$) in February, 6.3 points ($p<6.5\times10^{-14}$) in April, and 2.6 points ($p<5.9\times10^{-5}$) in June by the M_d(H) (= Motivation during lecture is High) group. It became clear that the motivation to learn during the lectures significantly influencing the quiz scores, and the importance of evaluating the motivation to learn was shown. To measure the learning performance more exactly, it is thought necessary to incorporate motivation to learn as an evaluation axis.

4 Conclusions

(1) Objective and subjective learning performance improved with a decrease in the percentage dissatisfied with the air environment.

(2) Objective learning performance in the high motivation group was significantly better. To measure the learning performance more exactly, it is thought necessary to incorporate motivation to learn as an evaluation axis.

Indoor climate and improvement possibilities in educational premises

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1 Introduction

The purpose of the study was to investigate the indoor climate, mainly the carbon dioxide concentration and air change in gymnasiums of Tallinn.

There is evidence that many countries have significant and serious indoor environmental problems in schools.

From the educational standpoint, the indoor air quality and ventilation in school buildings may affect the health of the children and indirectly also learning performance.

In Estonia the indoor air quality has not been investigated sufficiently. In 2005 and 2006 a pilot research was done in a cold season. The main aim of investigation was the carbon dioxide concentration and the air change level determination in classrooms.

According to the regulations of the Estonian Ministry of Social Affairs the permitted maximum carbon dioxide level in classrooms is 1000 ppm, relative humidity 30...70%, indoor temperature 19...25°C.

In most school buildings in Tallinn ventilation systems have not been renovated, but windows have as rule been changed and external walls in single cases renovated.

Only natural ventilation and window airing of classrooms is used in these school buildings.

2 Methods

The research included 10 classrooms in 7 schools of Tallinn. The average area of classrooms is 59 m^2. Density in classrooms ranged from 1.6 to 3.3 m^2 per person. The students who studied in schools investigated were 8th to 12th grade students.

The occupation rate of classrooms was 44…92 %, the doors and windows were closed during the class. As rule the duration of the measuring of parameters in classrooms was 40-45 minutes. The instruments used for measurements were Testo series 400 and 435.

3 Results

The rise of CO_2 concentration in classrooms is quite different; it depends mostly on the occupation rate of the classroom, air change per student and the level of CO_2 at the beginning of the class. Air change is very modest: 0.3-3.9 l/s per student. With low air change and normal occupation rate the rise of CO_2 is remarkable– 900-1600 ppm. The level of CO_2 at the beginning of the class differs greatly: ranging from 590 to 2160 ppm.

The range of relative humidity in classrooms is from 24% to 54% and the indoor temperature is between 18 to 25.5 ^{o}C.

4 Discussions

With natural ventilation in classrooms and airing only during breaks it is practically impossible to keep the CO_2 concentration under the permitted limit, 1000 ppm [12], at the end of a class with full occupation rate. Calculations show that in classrooms with average occupation rate, with an approximate air change of 1 l/s per m^2 and with good airing during breaks it is possible to hold the maximum CO_2 concentration on a level up to1500 ppm. To keep the CO_2 concentration under the permitted limits in fully occupied classrooms mechanical ventilation is needed.

5 Conclusions

With high occupation rate in gymnasiums of Tallinn good indoor climate and permitted CO_2 concentration at the end of the class can achieved only with the use of balanced ventilation.

It is advisable to use in a classroom displacement or wall confluent ventilation, the latter increasing the efficiency of air change.

The necessary air change should be 7-10 l/s per student, depending on the method used for air change.

Air distribution and temperature control in classrooms

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1 Introduction
Experimental interventions have shown that increasing the outdoor air supply rate and reducing moderately elevated classroom temperatures significantly improved the classroom performance of many tasks. The results suggest that higher ventilation rates up to 10 L/s per person as well as more strict temperature control than commonly used can be recommended for classrooms.

In this study, air distribution solutions for classrooms aiming to lower air velocities and good room temperature control are studied by measurements in 6 schools and by temperature simulations.

2 Methods
Air velocity, room temperature and CO_2 measurements were done in 6 schools. All schools were either relatively new buildings or renovated buildings, all having modern mechanical supply and exhaust ventilation systems. Air velocity measurement points were selected with smoke test, so that the highest velocities could be measured.

Both constant air volume (CAV) or demand controlled (DCV) ventilation systems with constant or controlled supply air temperature were used in schools. Differences between DCV and CAV systems and supply air temperature control options were studied with temperature and ventilation simulations for typical classrooms with a varying heat load.

3 Results
The performance of wall, ceiling, duct and displacement diffusers was compared. The highest measured ventilation rate was 340 L/s in the classroom with two duct diffusers, Figure 1. The maximum air velocity in the occupied zone with supply air temperature by 6°C lower than room temperature was 0.19 m/s. Another duct diffuser with small nozzles showed similar performance. The third duct diffuser without nozzles (just a perforated duct) showed the maximum velocity of 0.29 m/s.

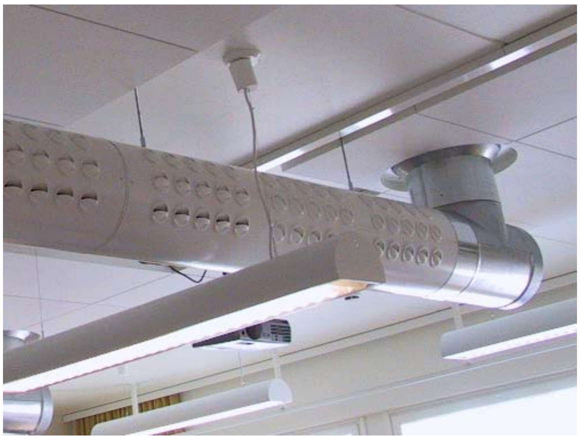

Figure 1. Duct diffuser with highest airflow rate

The wall diffusers were clearly not suitable for classrooms due to high velocities up to 0.43 m/s. Displacement ventilation diffusers were very sensitive to supply air temperature, as with a temperature difference of 3 °C velocities up to 0.28 m/s were measured.

4 Conclusions
Air velocity measurements showed good performance of duct and ceiling diffusers which provided maximum velocities less than 0.2 m/s and can be highly recommended for classrooms. The wall diffusers were clearly not suitable for classrooms due to high velocities which were also the problem of displacement ventilation diffusers at lower supply air temperatures.

Room temperature measurements showed a typical problem with temperature control as at the end of the heating season the temperatures up to 25 °C were measured. Parametric simulations showed that high supply air flow rates up to 10 L/s per person and cool supply air down to 14–15 °C were needed for room temperature control. This can be achieved with duct and ceiling diffusers measured.

Typologies of Hybrid Ventilation in Schools

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1 Introduction

In The Netherlands and other European countries the air change rate in schools increases in order to prevent health problems of students. The minimum air supply in The Netherlands is 7 l/s per student. Due to the high occupancy (generally 1 person at 2 m^2) prevention of draught needs much attention. Architects need ready-to-use ventilation design concepts for schools because of:

- A lack of knowledge of draught prevention,
- Individual architectural opinions,
- A low budget.

A research project is starting in The Netherlands to design options of natural, mechanical or hybrid ventilation. These systems for new or refurbished schools will be presented in a comprehensive way in a Dutch design guide and can be integrated in a REHVA-guide dealing with healthy schools in the near future.

2 Methods

The methods used and subjects discussed are:

a. Architectural research

- Systems of natural air supply, mechanical exhaust and several combinations with ceilings for three different façade typologies (positions of windows).
- A system of natural air exhaust via a double window and mechanical air supply for existing monuments.

b. Research of physical principles of air flows and heat transfer

- The mixing process of cold air with the surrounding warm air in order to prevent draught, with or without a false ceiling.
- An air outlet in a double window, the air resistance and dependency of wind pressures.
- The effects of different heating systems.

c. Airflow measurements and CFD-simulations

- The presented natural air supply and natural exhaust systems have been measured or simulated.

3 Results

Results are design options of:

- Self regulating natural air supply or exhaust systems.
- False ceilings, related to thermal and acoustic parameters.
- Different heating systems.

4 Conclusions

1. An integral approach of all the important “indoor” physical parameters is necessary even for common design problems.
2. For architects and other designers it is very important to have a compact database of images and knowledge to show comfort-problems and optional solutions.

Comparison between thermal comfort predictive models and subjective responses in Italian university classrooms

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1 Introduction

The Indoor Environmental Quality affects not only health and comfort, but also the occupants' productivity, so it strongly influences the general quality of working and educational environments. This study is focused on thermal comfort and aims at achieving a better knowledge about the subjective perception in naturally ventilated environments, in which the occupants have only some opportunities of behavioural adjustment. A particular, but significant, case is here analysed: naturally ventilated university classrooms. This study is part of a wider research started by the Building Physics and Indoor Environment Engineering Research Group (see http://www.polito.it/ftarch) of the Department of Energy (DENER) of the Politecnico di Torino, focused on environmental comfort in Italian school buildings. This paper focuses on the results from the thermal comfort field investigations in university classrooms.

2 Methods

The field study was conducted through physical observations and questionnaires, performed at the same time during the regular lesson time, in a period just before the start of the heating season. The approach consisted of administrating a questionnaire to a group of occupants while the investigator records certain microclimatic parameters. The object of this study, an extension of the previous study, was a university classroom of the Politecnico di Torino. The classroom was examined during the lesson time, in two periods outside the heating season, the first in September, with 103 students and the second in October, with 108 students. Heating through radiators and mechanical ventilation were off during the field campaigns. Two different comfort criteria, Fanger's PMV and an adaptive thermal comfort diagram, deriving from recent field studies, were applied, for the elaboration of the measured and observed data. The results were compared with the subjective votes of the students. Furthermore the votes of thermal preference were compared to the votes of thermal sensation.

3 Results

The two "objective" approaches lead approximately to the same results, in predicting the percentage of dissatisfied. The expected percentage of dissatisfied (< 10%) was compared to the percentage of dissatisfied from the questionnaires. It was found out a good agreement between the calculated PPD and the percentage of dissatisfied from the questionnaires when people voting (+2) on the seven points thermal sensation scale were considered satisfied. The judgments on the seven points thermal sensation scale were then correlated to acceptability and preference. The cumulate frequency distributions of ''wanting warmer'' and ''wanting colder'' versus the subjective judgment on thermal environment were plotted. The point corresponding to the minimum number of dissatisfied did not coincide with the thermal neutral condition (0 in the subjective vote) but was slightly shifted toward the positive values of the vote scale.

4 Conclusions

This study allowed a comparison between different predictive approaches for thermal comfort and a comparison between the predictions and the observed subjective responses. Furthermore it added new findings to previous researches conducted in Italian university and high school classrooms, during the heating season. The results of this study confirmed the trend outlined by the previous study, but with a minor intensity, suggesting a correlation between the thermal acceptability and preference and the outdoor temperature.

A comparative analysis of the indoor air quality and thermal comfort in schools with natural, hybrid and mechanical ventilation strategies

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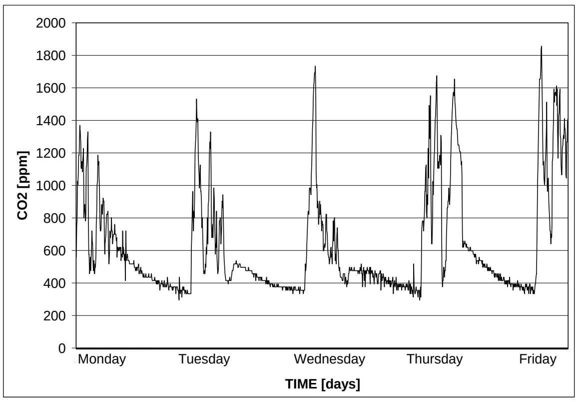
1 Introduction
The UK Government has committed to a massive programme of rebuilding and refurbishing schools in England and Wales in the next 10 to 15 years. To underpin this plan, the Department for Education and Skills has published design guidance - Building Bulletin 101 'Ventilation in School Buildings'. This performance standard document is cited as a means of compliance with the new Building Regulations Part F (Ventilation) in England and Wales. In this document CO_2 concentration has been chosen as the key performance indicator for the assessment of indoor air quality and ventilation performance in schools.

The work presented here forms part of a larger study investigating the ventilation performance of schools. That study, in addition to the requirements stated in Building Bulletin 101, is also investigating whether the comfort of occupants is compromised by the ventilation strategy in a particular classroom. This paper then focuses on a comparative analysis of indoor air quality and thermal comfort in three recently built schools with different ventilation strategies: natural, hybrid and mechanical.

2 Methods
Measurements were carried out in three schools in England during the heating season 2005-2006. All schools were built in compliance with Building Bulletin 93, which defines noise and acoustic criteria in relation to school design. The monitoring was carried out in two selected classrooms in each school over a period of five working days. Levels of CO_2 were monitored at five-minute intervals throughout the occupied day close to the occupied zone at seated head height to indicate the overall indoor air quality and provide a means of inferring the ventilation rate based on the number of occupants. In addition to CO_2 levels, thermal comfort parameters were also measured during the occupied periods in each of the selected classrooms.

3 Results
All the schools satisfied the recommended ventilation performance standards during the week that the measurements were undertaken – a record of CO_2 concentrations for one of the classrooms is given in figure 1.

Figure 1. CO_2 levels in a naturally ventilated classroom



With regards to comfort, not all of the classrooms met the relevant recommended levels.

4 Conclusions
The ventilation measurement results are apparently reassuring. However, this conclusion can be misleading unless one takes into account both the original design assumptions and then the actual occupancy of the classrooms and occupant behaviour in general. The work has raised concerns relating to the comfort provided in some of the classrooms.

Schools: All Problem Buildings?

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Seven schools underwent an energy audit, evaluating the existing situation through measurement and simulation and looking to possible retrofit measures and their economic feasibility with the energy performance tool (EPB) as an instrument. The results are troubling. The seven schools audited are all problem buildings: hardly any insulation, windows quite air leaky, central heating systems poorly designed and no usage of an on purpose installed ventilation system. As a consequence, IAQ is poor, with CO_2-levels passing 3000 ppm and a high percentage of dissatisfied with air freshness during teaching hours. Annual end energy use, however, is remarkably low compared to the EPB-prediction, showing that a potentially high energy demand evokes effective rebound behavior. The negative side of that is that hardly any energy conserving measure has enough impact to show a positive NPV at the end of the return period, let it be that the annuity on investment is lower than the annually avoided energy cost.

Ventilation of Dutch schools; an integral approach to improve design

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1 Introduction

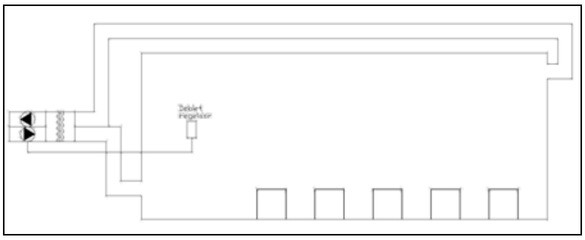
Indoor Air Quality and thermal climate in schools is very important as it has a direct relation to the health and performance of the pupils. The status quo in the Netherlands is presented (e.g. average CO2 levels in schools, quality of ventilation). The goal of a first study was to evaluate the performance of exhaust-only ventilation systems. The performance was rather disappointed there were a lot of problems and insufficient situations found.

Average and range of CO_2-concentrations for Dutch schools:

Study	**number of schools**	**CO_2, ppm** Average	Range
Joosten, 2004	5	1220	480-2400
van Dijken, 2004	11	1580	450-4700
van Bruchem, 2005	6	1355	550-3000

2 Contribution to the conference

During the next years different master students together with the staff of Technische Universiteit Eindhoven were researching different aspects of the problem and trying to find solutions. A new integrated approach to design adequate solutions for ventilation of school buildings was developed. First design results are described in the paper.



Schematic of displacement classroom ventilation

3 Conclusions

Many ongoing research efforts aim at introducing the used methodical design process into building design and architecture, and so linking architectural design with HVAC-design. In the Netherlands this has resulted in a stark interest in what is commonly named ‘integral design’.

Integral design is meant to overcome the difficulties of design team cooperation, by providing methods that make it possible to communicate the consequences of design moves on areas such as construction, costs, life cycle and indoor climate at early design stages, between the different disciplines.

Acknowledgement

Thanks goes to PIT (Stichting Promotie Installatietechnologie) for their financial support.

Particulate matter levels in Portugal (mainland and islands). A preliminary study for outdoor/indoor environment in basic schools.

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Abstract
This study deals with Particle Matter (PM) levels below 2.5 µm ($PM_{2.5}$) in Portugal and shows that US EPA (United States Environmental Protection Agency) directive is exceeded in a few places. $PM_{2.5}$ total mass concentration measured in several places located in Portugal mainland and islands and the outskirts are quite well correlated for a few sites. Results show that it is important to determine the elemental composition of $PM_{2.5}$, and to develop an epidemiological study in Portugal to find a possible association between $PM_{2.5}$ levels, sources and morbidity/mortality. However, the results imply that a source-oriented evaluation of PM health effects needs to take into account the uncertainty associated with the spatial representativity of the species measured at a few sampling stations. For that purpose the survey using biomonitors may contribute positively.

Indoor Thermal Environment in a Classroom Equipped with Air Forced System

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Summary
In this work the evaluation of the indoor thermal environment in a classroom equipped with air forced system will be made. In the classrooms' indoor thermal environment, with the air forced inlet in the door and the outlet located above the windows, two situations are analyzed: the inlet forced airflow made in the doors' lower and upper area.
This experimental study, made in a school building located in the South of Portugal (see figure 1), in Mediterranean climate, evaluates the thermal comfort (using the PMV index), local thermal discomfort (using the draught risk and uncomfortable air velocity fluctuations) and air quality levels (using the air renovation rate) that occupants are subjected.
The experimental test was made in Summer conditions in a warm September day using the inlet air flow coming from the corridor compartment. The corridor inlet air temperature, lower than the outdoor environment, is insufficient to guarantee comfortable conditions. Nevertheless, in general in this kind of warm days the students are in holidays.
The draught risk is highest in the occupied area near the door, mainly when the ventilator is placed in the door's upper area.
The equivalent air velocity fluctuations frequencies values are inside the uncomfortable interval, nevertheless the more uncomfortable air velocity fluctuations are verified near the lower members level in the desks placed in front to the window near the blackboard, when the ventilator is placed in the door's lower area, and near the head level in the desks placed in front to the door near the blackboard, when the ventilator is placed in the door's upper area.
The local airflow rate is lightly lower when the ventilator is placed in the door's lower area than when the ventilator is placed in the door's upper area. Nevertheless, in all situations the air renovation flow rate is insufficient.
In order to increase the air renovation rate, in future works, to use more than one extractor fan and to put them in the window upper area instead of the door area are recommended. This fact reduces substantially the influence of external wind direction, which was verified in the present work. Nevertheless the increase of the airflow rate, mainly for lowest air temperature levels, can promote local thermal discomforts levels, associated to the draught risk, mainly in desks located near the blackboard.

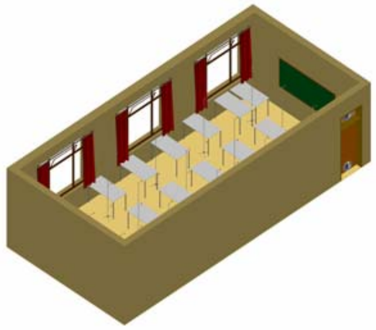

Figure 1. Scheme of the classroom used in the experimental setup.

A study on the effect of the airflow rate of the ceiling type air-conditioner on the ventilation performance

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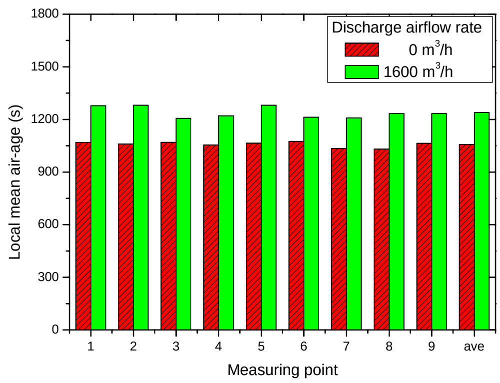
1 Introduction

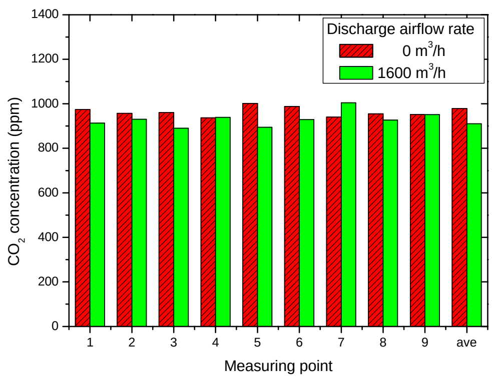
We performed the experiments and numerical simulations in a lecture room with a ceiling type air-conditioner and a ventilation system. With variations of the air discharge rate of the ceiling type air-conditioner and occupancy, CO_2 concentrations and local mean air-ages during class hours were investigated and the effect of the air discharge intensity of a ceiling type air-conditioner on ventilation effectiveness was analyzed.

2 Results

When the discharge airflow rate of the ceiling type air-conditioner, which is one of the indoor momentum sources, is increased, mean air age can be also increased. The reason is that the operation of the ceiling type air conditioner intervene the air current and makes the outdoor fresh air move a longer way. Therefore the indoor air quality may be deteriorated as the discharge airflow is increased. Unlike mean air age distributions, CO_2 concentration get to be somewhat decreased as the discharge airflow rate is increased. This reason is that the operation of the ceiling type air conditioner can make the mixing effect of indoor contaminants increased and the residual life time of indoor contaminants decreased.



(a) mean air age



(b) CO_2 concentration

Fig. 1 Experimental results for mean air age and CO_2 concentration at the measuring points with the different discharge airflow of 4-way cassette air-conditioner

3 Conclusions

The effect of the discharge airflow rate of the ceiling type air-conditioner on ventilation performance was studied. From the results, both of mean air age and residual life time must be considered for evaluating ventilation performance when the contaminants are generated indoors. And the increment of discharge airflow of the ceiling type air-conditioner can induce the piston effect and push the contaminants out of the occupied zone.

This shows that the ventilation performance can be increased when the momentum source like an air-conditioner, air cleaner, and fan is used in the room with mixing ventilation.

The Influence Of Mass Advance For Altitude Tropical Climate: Establishment Of Limit Conditions For Thermal Confort In Classrooms

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Summary
This article is part of a research in progress about comparative study methods for the Brazilian reality using among many other authors, Givoni (1969), Voght and Miller-Chagas (1970), Fanger-ISO(1970), ASHRAE (55-1992), Mahoney (1971), Humphreys (1978) and Olgyay (1962) methods. This research presents the principal concept to be evaluated by the Universal Fuzzy Controlled aiming to establish a reference to determine a possible interference of the acclimatization factor to determine thermal comfort. An experimental evaluation method was elaborated through a questionnaire applied to the population of students in classroom, monitoring the environmental variables inside and outside the classroom during the years 2004 and 2005. It shows their particularities for the altitude tropical climate of the city of São Paulo -Brazil. This article describes interference of mass advance at the responses of students in classroom comparing to internal and external environment variables. The data was gotten in parallel with classroom in full activity.

Key words: climate, experimental and simulations thermal performance, classroom and human thermal comfort behavior study case

Indoor habits of children aged 5 to 10 years learning at the public basic schools of Lisbon-city, Portugal

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Abstract
In this work, 36 basic schools of Lisbon city, Portugal followed a questionnaire of the ISAAC -International Study of Asthma and Allergies in Childhood Program. The questionnaire contains questions to identify children with respiratory diseases (wheeze, asthma and rhinitis) as well as their nutrition habits, ingested medication, environmental aspects, among others. The questioned children are 5 to 10 years old, and the answers are from June to December 2006. The results are from 995 children inquired who have shown 26.7% with wheezing symptoms, 9.2% with asthma, and 26.2% with rhinitis. The results obtained are compared with the results, interpretations and correlations obtained in the ISAAC 2002 program, which questioned 2484 6 to 7 years old children from the basic schools of Lisbon-city from November 2002 till March 2003.

Enhancing visual comfort in classrooms through daylight utilization

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1 Introduction

The present study is focused on the analysis and the evaluation of several daylight strategies aiming at the enhancement of visual comfort in classrooms.

2 Methodology

The study was conducted as a parametric analysis. On the basis of a typical classroom, the reference model (C_0) was created. A typical classroom is of rectangular shape, usually side lit with unilateral windows that account for 20% of floor area. The reference model is south orientated, side lit, with an unobstructed view to the sky vault. By changing the geometry, the position and the function of the transparent elements, 5 more models were created:

- Model C_1: glazed area: 20% of floor area. A light shelf is added, projecting towards both the interior and the exterior of the façade.
- Model C_2: glazed area: 25% of floor area. Daylighting is provided by lateral windows with light shelves (as in model C_1) and vertical south-orientated skylights, which are positioned on the roof at a distance of 3.5m parallel to the window wall.
- Model C_3: glazed area: 25% of floor area. Daylighting is provided by lateral windows with light shelves (as in model C_1) and vertical, south-orientated skylights, which are positioned on the roof at a distance of 4.6m parallel to the window wall.
- Model C_4: glazed area: 25% of floor area. Daylighting is provided by lateral windows with light shelves (as in model C_1) and vertical, north-orientated clerestories positioned on the opposite wall.
- Model C_5: glazed area: 25% of floor area. Daylighting is provided by lateral windows with light shelves (as in model C_1) and tilted, north-orientated skylights, positioned in two rows on the saw-tooth roof.

The visual environment prevailing in each case was estimated with the help of the simulation program ADELINE.

3 Results

In reference model C_0, the average levels of daylight are satisfactory, but the uniformity of daylight is not achieved. In order to moderate illuminance in areas near the windows and promote daylight penetration deep in the room, light shelves were incorporated. Simulations showed that the light shelves reduced the amount of light received in the interior of model C_1 relative to the conventional windows of model C_0, while the luminance distribution is better and the contrast between minimum and maximum values is limited. With the south skylights of models C_2 and C_3, daylight levels were increased without decreasing the uniformity of illuminance. In fact, it was found that the performance as regards daylighting is better when the skylights are located across the middle of the model's width. Bilateral daylighting with north orientated clerestories (model C_4) results to high levels of illumination, but the zone formed by low daylight factors is relatively wider, indicating a probable discomfortable visual environment. North orientated skylights (model C_5) have the best performance, since not only there is an overall increase of daylight factor on the working plane, but also the uniformity of its distribution is considerably intensified.

4 Conclusions

The above analysis showed that many alternatives regarding the location of openings exist for the enhancement of daylighting in classrooms. It is found that a combination of façade and roof apertures performs better than advanced façade systems (e.g.light shelves). From the examined models the north orientated skylights arranged in a saw-tooth roof combined with south windows appear to have the best performance. They bring more daylight in the rear areas, where illumination levels are low, daylight distribution becomes balanced and glare is avoided to a great extent as well.

Characterization of Indoor Air Quality (IAQ) in School using Biological Approach: A Case Study

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Abstract
Microorganisms play an important role in the development of the diseases associated with respiratory/lung function disorder in occupational environment. Biocontaminants forms a considerable part of the organic dust. It is a significant approach to characterize indoor air quality (IAQ) by measuring biological markers. IAQ of a school environment has been characterized by measurements of airborne biocontaminants (Bacteria and Fungi) of indoor dust. An innovative sampling strategy was designed and used to measure the concentration of airborne bacteria and fungi of viable or migrated, to be settled and settled dust. Three types of sampling techniques viz. Volumetric, Gravimetric and Swab sampling techniques were employed. Five sampling stations were installed to cover up maximum area of building. Sampling was carried twice in a month for eight months.

The results show that the airborne concentration of biocontaminants (Bacteria and Fungi) were varied from 10 to ≥3600 colony forming unit cfu). (Fungal concentration was ranged 10 to ≥1500 cfu, while bacterial concentration was ranged between ≥20 to ≤4000 cfu. Concentration of Gram Negative Bacteria (GNB) was found 25 to 30% higher than Gram Positive Bacteria (GPB). In conclusion, children and staff members working in such environment can be exposed to large concentrations of airborne biocontamints, which may increase a risk of work-related diseases. Study shows an impression of poor IAQ.

Study gives an idea about the use of array of sampling techniques concurrently. A combined use of such sampling techniques is simple, inexpensive approach in addition to no time consuming. It makes easy to sampling a large area as well as a large number of predictable samples can be collected at a time and sampling error can be avoid.

Key Words
Indoor Air Quality (IAQ), Biocontaminants (Bacteria and Fungi), Sampling Techniques.

Indoor environmental quality in schools in relation to academic performance of students: Observations of potential contributors to poor IEQ

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SUMMARY

Results of a pilot study published by Shaughnessy et al. 2006, demonstrated a modest association between class room ventilation rates and student performance in math standardized test scores, and also a need for further studies with larger sample size and more comprehensive assessment of indoor environmental quality (IEQ). A new school district in the USA with approximately 50 schools has undergone a thorough assessment protocol and the data is currently being analyzed. This paper focuses on these data, specifically on observations of potential contributors to poor IEQ. In addition there is a new on-going study in Finland that will provide further knowledge on how and to what extent IEQ in schools affect academic performance of students.

INTRODUCTION

It is assumed that poor indoor environmental conditions result in reduced learning potential and subsequent poor student academic performance (AP). However, there is limited data linking poor IEQ in the classrooms directly to student academic performance (Mendell and Heath 2005). Documentation of adverse effects of poor IEQ on AP is critical to motivate protective environmental guidelines in schools. One of the primary IEQ problems identified in earlier studies has been inadequate ventilation with outdoor air. For example, a recent study by Wargocki and Lyon (2006) found an association between improved performance (speed of tasks) and increased outdoor air supply rates.

A pilot study investigating classroom ventilation rates, and their association with reduced student performance was conducted within a school district in the USA in the spring term 2004. Data on classroom CO2 concentrations were recorded in 5th grade classrooms in fifty four elementary schools. In addition, investigators were able to work with the district to obtain standardized test scores (math and reading), and background data related to the students in the specific classroom studied in each school. Results of the pilot study demonstrated a modest association between class room ventilation rates and student performance in math, and also a need for further studies with larger sample size and more comprehensive assessment of IEQ (Shaughnessy et al. 2006).

A new school district in the USA with approximately 50 schools has undergone a thorough assessment protocol and the data is currently being analyzed. This paper will focus on these data, specifically on observations of potential contributors to poor IEQ. In addition there is a new study on-going in Finland (majority of the field work will be done in spring 2007) that will provide further knowledge on how and to what extent IEQ in schools affect academic performance of students. Specifically, the Finland study will explore the possible links between IEQ, AP, and students' health status, as it is assumed that poor IEQ can cause illness that require absence from school, and can cause health symptoms, hence decreasing performance (Daisey and Angell 1998).

This study program, as a whole, is specifically designed for a school setting and looks directly into the relationship between student AP and IEQ. The primary aim of the program is to study if student AP is significantly (in the statistical sense) related to IEQ in classrooms. The study also aims to further characterize the relationships, and identify critical determinants and links.

METHODS

Preliminary data is being acquired on a new school district in the USA with 50 elementary schools. The schools selected are representative of a consistent cross section of elementary schools. Specific emphasis was given to the classroom under investigation related to the students in the 5th grade study room. Schools in the district normally had more than one section of 5th grade students (multiple classrooms). The study room was selected (only 1 from each school) randomly, except for the provision that the HVAC system serving the classroom was restricted, when possible, to that classroom alone.

The new data focused again on the use of CO2 concentrations as a surrogate for ventilation rates. CO2 concentrations were logged over a two day period within typical school day occupancies under closed classroom conditions (i.e. windows and doors closed), and translated into ventilation rates per child, which were then adjusted based on the typical occupancy in the room throughout the school year.

Other indicators of the indoor environment related to the investigated classrooms were recorded by checklists incorporating a thorough assessment protocol. The visual observation data, based on current conditions and building characteristics included observations such as: general building and HVAC characteristics, evaluation of condition and performance of structural assemblies, use of materials, degree of clutter, general cleanliness of environment, sources of chemical and biological pollutants (animals, dampness/mold, VOCs, particles), and cleaning regimen. This paper focuses on building and HVAC characteristics in relation to ventilation rates. Thermal comfort parameters and airborne particle concentrations were also measured, but will be reported elsewhere.

In addition to the measurements and investigations, investigators were able to work with the district to obtain results from results of statewide-standardized test scores (math and reading), and background data related to the students in the specific classroom studied in each school. For the year of classroom IAQ sampling the percent of students scoring satisfactory or above in math and reading tests in each studied classroom was chosen as the primary metric of AP used in the study.

The data presented in this paper was input into a database and analyzed using SPSS statistical package version 14.0. In the initial screening, Pearson's correlation coefficient was used as a measure of linear association between ventilation rate and general building characteristics. Cross tabulation and chi-square test (categorical variables) or independent samples t-test (continuous variables) were used to study univariate associations between ventilation and building –related variables. Multivariate analysis was performed using logistic regression and step-wise method in selecting the independent variables.

RESULTS AND DISCUSSION

General information about the schools is shown in Table 1. Mean age of the buildings was approximately 50 years, and mean size was approximately 5450m2, leading to occupant density of approximately 0.086 occupants/m2.

Table 1. General information of the school buildings.

	mean (min-max)
Year when built	1958 (1907-2005)
Size of the building m^2	5450(3500-11950)
Number of occupants	
teachers	30(13-75)
other personnel	29(9-80)
students	381(162-1000)

Most of the school buildings were one story structures with slab-on-ground, with concrete frame and brick veneer exterior and flat roof (Table 2). All but three out of fifty schools relied on gas as the heating source.

Table 2. Predominant building characteristics.

	n(%)
One story construction	40(80)
Ground slab	45(88)
No basement	46(98)
Main frame structure material concrete/ concrete block	41(82)
External wall covering brick veneer	38(76)
Flat roof	36(72)
Bitumen roof membrane	37(74)
Type of heating source gas	47(94)

Predominant HVAC systems were roof top units and AHU’s (Table 3). Outdoor air was mechanically provided in 32(64%) schools and 17(34%) had mechanical exhaust. A total of 35(70%) class rooms could be ventilated by opening windows and in 13(26%) of the class rooms ventilation could be controlled mechanically. However, ventilation was perceived insufficient in most of the schools.

Table 3. HVAC characteristics

	n(%)
Predominant HVAC systems	
roof top units	15(30)
AHU’s	13(26)
heat only / window AC	10(20)
Predominant exhaust system	
natural (no mechanical exhaust)	33(66)
mechanical exhaust	17(34)

HVAC system provides outdoor air	32(64)
Ventilation be controlled in each class room by opening windows mechanically	 35(70) 13(26)
MERV rating of the filter 0-4 4-8	 14(28) 28(56)
Ventilation not perceived sufficient	30(60)

Ventilation rates ranged from 1.7 to 19.0 l/s per person. The highest ventilation rate observation appeared to be an outlier, and when excluded, mean ventilation rate was 4.6(1.7-11.4) l/s per person.

Ventilation rate did not correlate significantly with building age, size, occupancy, or with occupant density (data not shown). We then roughly divided classrooms into two groups based on whether class room had estimated ventilation rates 1) below or 2) above 4 l/s per person (close to 50th percentile), and cross tabulated the two-category variable with building and HVAC characteristics, as in Tables 2 and 3, which could be potentially associated with ventilation rate.

From building characteristics, a flat roof was significantly ($p \leq 0.05$) more common in the group of higher ventilation rates (above 4 l/s per person). All the class rooms above ground level fell into the group of higher ventilation rates. No significant differences were observed with respect to the other building characteristics (Table 2).

From HVAC characteristics, AHU's and roof top units were significantly more common in the group of higher ventilation rate, whereas five out of six schools with fan coil units fell into the lower ventilation rate category. Other characteristics that associated with higher ventilation rate were mechanical exhaust system, mechanical supply of outdoor air, and higher MERV rating of filter employed by the HVAC unit. Ventilation was perceived sufficient significantly more often in the higher ventilation rate category.

In multivariate analysis, two-category ventilation rate as the dependent variable, independent variables included in the model first round were roof type and HVAC system capacity in providing outdoor air. However, it was noticed that roof type associated with all HVAC characteristics (as in Table 3) in univariate level. It is not plausible that roof type as such would truly effect on the ventilation rate, thus the association seen is more likely to be caused by multicollinearity with other variables. Therefore, a second round was performed without roof type and the resulting two independent variables associated with ventilation rate variable were HVAC system capacity in providing outdoor air, and mechanically controllable ventilation.

Mean math and reading test scores in different ventilation rate categories are shown in Table 4. Although the mean math score was approximately 8% higher, and reading score was approximately 11% higher in the higher ventilation rate category (>4 l/s per person) than in the lower ventilation rate category (<4 l/s per person), this difference was not statistically significant. Further analysis will be made using more specific data on the ventilation rate and related variables, and also taking the confounding factors into account (in the earlier study, the effect of confounding factors was significant).

Table 4. Mean math and reading scores in class rooms with ventilation rates under or above 4 l/s per person.

	Ventilation rate	N	Mean	Std. Deviation
Math	< 4 l/s per person	21	69.7	14.7
	> 4 l/s per person	28	75.0	18.0
Reading	< 4 l/s per person	21	61.1	16.0
	> 4 l/s per person	28	68.4	20.4

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Ventilation Rates in Schools and Learning Performance

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SUMMARY

Associations between classroom ventilation and pupils' performance were investigated in primary schools in the United Kingdom. The concentration of carbon dioxide and other parameters were monitored for three weeks in two selected classrooms in each school. A direct air supply system through the windows was used to alter the ventilation rates in the classrooms. The system was set either to provide outdoor air or to re-circulate the classroom air while all other physical parameters were left unchanged. Computerised Assessment Tests and Paper-based Tasks were used to evaluate pupils' performance. Pupils' perceptions about the classroom environment, comfort, general mood and hunger were assessed on subjective scales. The present paper shows preliminary results obtained for one primary school out of eight being studied. Due to the intervention the fresh air supply increased from 0.3-05 to 13-16 L/s per person that increased pupils' work rate by ~7% in addition ($p<0.036$) and subtraction ($p<0.052$).

INTRODUCTION

Former reviews on the subject of school environments emphasised that ventilation is often inadequate in classrooms causing increased risk for asthma and other health-related symptoms among school children [1], [2]. Mendell & Heath [2] proposed that throughout the life of each existing and future school building immediate measures should be taken for the provision of adequate outdoor ventilation, control of moisture, and avoidance of indoor exposures to microbiologic and chemical substances considered likely to have adverse effects. The current ventilation standards and guidelines [3], [4] recommend a minimum fresh air supply rate of 8 litres/s per person for occupants in all teaching facilities. The recently published Building Bulletin 101 refers to proposed performance based standards limiting the level of carbon dioxide (CO_2) concentration to 1500 ppm over a full school day from 9:00 to 15:30 and specifies a minimum supply of external air at least 3 L/s per person in all teaching and learning spaces when they are occupied. Furthermore, a ventilation rate of 8 L/s per person for the normal number of occupants should be achievable under the control of occupants, although it may not be required at all times if occupancy level decreases. However, according to recent studies the average CO_2 levels in classrooms often exceed the above limit and ventilation rates are often below the minimum requirement of 3 L/s per person [5], [6].
The negative effects of poor ventilation rates on work performance in office buildings have been widely investigated [7]. Knowing the outcome of poor ventilation rates for the adult population it could be expected that not only the comfort and health, but also the learning performance of school children are affected by the poor environmental conditions in classrooms [2]. Following the earlier studies suggesting correlation between pupils' health

and work performance, [8], [9] there is growing evidence showing impairment of learning performance and increased absenteeism due to inadequate ventilation and unsuitable thermal conditions in classrooms [10], [11], [12], [13]. The main purpose of the present research was to investigate the relationship between pupils' health, well-being and performance, and the indoor air quality in several primary schools in Southern England. Another aim was to examine the suitability of the air quality guidelines in preventing the reported negative effects even when the recommended levels of fresh air to the occupants are met.

METHODS

Field surveys were carried out at primary school buildings located in the proximity of Reading during years 2006-2007. Up till now measurements have been done in eight different schools. The sample included schools that were built in the last 20-40 years. Except for one school, none of them had mechanical ventilation system and in most schools no control over the temperature was available to the staff. At each selected school investigations were carried out in two classrooms for at least three consecutive weeks. The first week was reserved to monitor the classroom conditions without modifying any of the indoor climatic parameters and to familiarise the children with the performance tests. During the second and third week a mobile ventilation system was installed in each classroom to control the ventilation rate and maintain the temperature within certain limits. The system was set either to provide outdoor air or to re-circulate the classroom air. Although the ventilation system was visible, the staff and the children were not informed of the ventilation conditions, i.e. whether it was providing fresh air or re-circulated air. The order of presentation of the fresh air/re-circulated conditions were made in a cross over repeated design for the two classrooms.

The ventilation system consisted of an exterior fan placed outdoors and simple ducting of diameter 200 mm led the air into the building through window openings, which were closed with Perspex plates (Figure 1, a). In the classrooms the air was distributed using Softflo air terminal units, which consist of a perforated duct with small nozzles creating confluent jets flow into the room [14]. The temperature of the supplied air was controlled by means of a duct heater (3kW) and a mobile air conditioning unit of 2.7kW built into the ventilation system. The capacity of the supply fan was selected to provide 200 L/s, matching the prescribed level of 8L/s per person in a classroom having on average 25 children. Silencers were also built into the system upstream and downstream of the fan to reduce the noise level propagating through the duct work into the classroom.

a)

b)

Figure 1. a) Exterior fan of the mobile ventilation system, b) Testing area with laptops and measuring trolley with the air terminal device in the background; the trolley was placed close to the testing area during performance tests.

Physical measurements: CO_2 concentration (0-5000 ppm), air temperature, globe temperature, relative humidity (RH), air velocity and light level were continuously monitored in each classroom and recorded with 3 minutes interval on a central logger using a wireless data transmission technique. These sensors were fixed on a trolley (Figure 1. b) and placed close to the testing area in the classrooms. In addition three thermistor type temperature probes were distributed on a vertical pole fixed to the trolley to record the temperature differences between pupils' head and feet levels. Separate units were placed outdoors and in the corridors to measure CO_2 concentration, temperature and RH. Mass concentration of airborne particles (PM2.5) and noise level were measured during the performance tests on pupils over a few hours. The amount of supplied air to the classrooms was measured with Venturi flow meters built into the duct system downstream of the fan.

Subjective evaluations: Simultaneous to the physical monitoring, measures of self-assessed environmental perception, comfort and health were obtained immediately after the performance tests were carried out. With some exceptions all pupils participated in the testing. The targeted age group of the children was between 9-10 years attending Year 5. This age group of pupils was selected because they remain in their classrooms most of the day and are therefore in the same environment throughout a school day. The pupils were asked to complete a simple questionnaire about the classroom environment, thermal sensation, mood, Sick Building Syndrome (SBS) symptoms and life style, such as hunger and quality of sleep over the previous night believed to affect their performance. The questionnaire about the classroom environment included questions about air stuffiness, dryness, perception of light and noise. The SBS questionnaire focused on symptoms of the mucous membrane and in upper respiratory tract, such as nose congestion, nose, mouth, throat and eye dryness, and neurobehavioral symptoms including headache, attention, dizziness, tiredness, sleepiness. Pupils were asked to rate the intensity of each symptom on Visual Analogue (VA) scales [15]. Thermal sensation was recorded using a 7-point PMV scale [16]. Furthermore, pupils were asked to rate the air movement around their body and inform whether it was acceptable or not.

Pupil's Performance Tests: Two different performance tests were administered to the pupils in each school. Traditional tests were carried out on paper for 40 minutes, including simple addition and subtraction of numbers (15 minutes each) and reading comprehension [17] (10 minutes) similar to that performed in a normal school day. New software (VISCOPE – Ventilation in Schools and Cognitive Performance) was developed that uses algorithms based on the work of Iregren *et al.* [18] to study changes of pupils' cognitive performance under different air quality conditions in classrooms. These tests were conducted on laptop computers set up in the classroom, similar to the method used by Coley and Beisteiner [19]. Both the traditional tests and the computer tests were given to pupils during their lessons preferably before the lunch break when the CO_2 concentrations had reached the maximum level of the morning's teaching session. The computer tests lasted for 20 minutes and were conducted in 3-4 consecutive groups, each group including 7-8 children. The tests, whether they were conducted on paper or computers, were carried out on each testing week on the same weekday and time period for each group of children.

Data analysis: Outdoor air supply rate was calculated based on the mass balance model of CO_2 on each testing day. The subjective and performance data were analysed using Wilcoxon matched-pairs test, using each subject as their own control. All p-values are 1-tailed of an effect in the expected direction.

RESULTS

The current project is still in the phase of data collection hence preliminary results of the physical environment and performance tests conducted on paper from only one school are presented. Detailed analysis of the performance results including those conducted on the computers will be published on the completion of the current investigations.

Figure 2 shows a typical CO_2 pattern in one of the classrooms during a weekday when performance tests were completed. The classroom of 156 m^3 was occupied by 23 children and a teacher at normal activity levels. The teaching schedule including lessons and break time can be clearly followed by looking at the changes in the CO_2 concentrations. The uncontrolled condition on Figure 2 shows the CO_2 level prior to any intervention in the classrooms. The CO_2 concentrations obtained during the week with the re-circulation ventilation are matching closely the uncontrolled levels seen during a normal school day. When the ventilation system was switched on to provide outdoor air the CO_2 concentrations were dramatically reduced and remained below 1000 ppm throughout the school day.

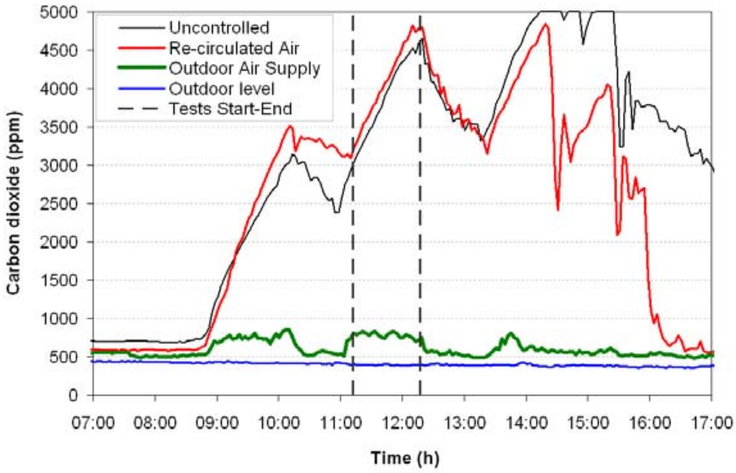

Figure 2. Typical pattern of the CO_2 level inside a classroom at different ventilation conditions on a testing day. The uncontrolled condition reflects CO_2 concentration during a normal school day without any intervention measures.

The average levels of the main physical parameters in the classrooms during the performance tests are presented in Table 1. At low ventilation rates the CO_2 concentrations during the performance tasks at a given day and classroom varied from 1600 ppm up to 4000 ppm depending on the occupancy level prior to testing. Temperature deviations between low and high ventilation rate conditions were within 1.7°C with one exception in classroom B where the difference in the operative temperature reached 2.7°C due to exceptional hot outdoor conditions during the tests conducted on paper. Relative humidity was generally higher at low ventilation rates due to moisture generation from people that is also reflected in the enthalpy of the classroom air. The air exchange rates in the re-circulation mode were not higher than 0.3 h^{-1} in both classrooms, showing an effective building tightness with closed windows that is responsible for the high levels of CO_2 and the extremely low outdoor air infiltration of not more than 0.55 L/s per person. The measured amount of fresh air supplied during improved ventilation was at180 - 190 L/s corresponding to 4.2 - 4.4 h^{-1} air exchange rates. The calculated air exchange based on the CO_2 mass balance model showed higher rates of up to 8 h^{-1} (12 - 16 L/h per person) which is most likely due to some windows being opened that enhanced cross ventilation. The average particle (PM 2.5) concentration during testing was in the range of 0.05 - 0.1 mg/m^3 and did not show major changes due to ventilation improvement. Classroom noise levels during testing were typically between 50 - 70 db(A) depending on the classroom activities. The background noise level originating from the ventilation system installed was less than 48 - 49 db(A).

Table 1. Average levels (± standard deviation) of main environmental parameters inside the classrooms and outdoors during performance testing; the ventilation rate calculation is based on the CO_2 mass balance model for each classroom; outdoor CO_2 level was at 380-420 ppm.

	class room	Testing on Computer		Pen & Paper testing	
		Re-circulated Air	Oudoor Air Supply	Re-circulated Air	Oudoor Air Supply
CO_2 level [ppm]	A	2876 ± 446	735 ± 58	1638 ± 364	709 ± 30
	B	4093 ± 509	783 ± 35	2086 ± 171	593 ± 7
Air temperature [°C]	A	20.5 ± 0.3	18.9 ± 0.2	20.9 ± 0.3	20.1 ± 0.2
	B	18.7 ± 0.4	20.3 ± 0.7	18.4 ± 0.4	21.5 ± 0.5
Relative Humidity [%]	A	67 ± 1	56 ± 2	69 ± 1	52 ± 1
	B	66 ± 1	55 ± 3	61 ± 1	64 ± 2
Operative temperature [°C]	A	20.5 ± 0.4	18.8 ± 0.3	21.1 ± 0.4	20.2 ± 0.2
	B	19.0 ± 0.5	20.2 ± 0.7	18.9 ± 0.4	21.6 ± 0.5
Outdoor temperature [°C]	A	16.9 ± 0.5	17.7 ± 0.5	24.9 ± 0.5	17.9 ± 0.1
	B	16.7 ± 4.5	21.2 ± 0.9	17.8 ± 0.2	25.0 ± 0.5
Enthalpy [kJ/kg]	A	46.41	38.41	48.28	39.64
	B	41.45	41.24	39.00	47.85
Ventilation Rate [L/s.person]	A	0.55	16.0	0.51	14.8
	B	0.36	13.9	0.20	12.2

Table 2. Results of a selection of subjective votes recorded following the performance tests; significance of statistical tests also appear next to each question; n.s.= not significant.

Perception / Symptom / Comfort	classroom	Testing on Computer			Pen & Paper testing		
		Re-circulated Air	Oudoor Air Supply	p <	Re-circulated Air	Oudoor Air Supply	p <
Air Stuffy (0) - Fresh (100)	A	48	81	0.01	34	72	0.01
	B	71	66	n.s.	66	52	n.s.
classroom Noisy (0) - Quiet (100)	A	62	91	0.01	66	87	0.01
	B	51	59	0.05	81	80	n.s.
Dreamy (0) - Attentive (100)	A	57	61	n.s.	52	63	0.08
	B	70	72	n.s.	59	61	n.s.
Tired (0) - Not Tired (100)	A	57	59	n.s.	39	50	0.10
	B	57	58	n.s.	61	57	n.s.
Sleepy (0) - Alert (100)	A	57	61	n.s.	40	63	0.01
	B	68	69	n.s.	63	60	n.s.
Feel like Working (0) - Do not feel like working (100)	A	46	56	n.s.	27	41	0.01
	B	68	68	n.s.	62	60	n.s.
Thermal Comfort (-3 = Cold, +3 = Hot)	A	1.1	0.3	0.02	1.8	0.2	0.01
	B	0.4	0.9	n.s.	0.1	0.9	0.01
classroom environment Bad (0) – Good (100)	A	60	77	0.04	56	69	0.03
	B	80	83	n.s.	78	81	n.s.

Selected results of subjective responses to the classroom environment immediately after testing are included in Table 2. The pupils in classroom A perceived the air as being fresher, the classroom less noisy and their general feeling about the classroom environment was significantly better in the condition with increased ventilation compared to that with re-circulation. There was a trend approaching significance towards higher alertness, better work mood and tendency for less tiredness and increased attention following the performance tests conducted on paper at the higher ventilation rates. The pupils' thermal sensation was in accordance with the existing temperature differences shown between the conditions. They were closer to neutral at operative temperatures between 18.8 and 20.2 °C and felt slightly warm at 20.5-21.6 °C.

Evaluation of the reading comprehension task was made according to the marking sheet provided with the tasks. Compared to a maximum mark of 16, the children in classroom A obtained an average mark of 9.4 in the condition with improved ventilation that showed a tendency of a higher rating ($p<0.09$) than 8.1 achieved in the other condition with low outdoor air supply rate. No significant change was found in the reading comprehension marks of children from classroom B between the two experimental conditions. Details of the maths based performance measures for all (40) children who completed the performance tasks in both experimental condition are shown in Table 3.

Table 3. Average speed, accuracy and overall performance (i.e. number of error-free units) of subjects achieved in the addition and subtraction tasks.

Class-room	Condition	Addition task			Subtraction task		
		Speed (units/h)	Error Rate (%)	Performance (units/h)	Speed (units/h)	Error Rate (%)	Performance (units/h)
A	Re-circulated Air	142.3	23%	111.2	149.1	43%	90.1
	Outdoor Air Supply	143.0	19%	118.7	142.7	37%	90.9
B	Re-circulated Air	139.2	15%	121.3	144.4	28%	103.4
	Outdoor Air Supply	144.4	13%	125.5	143.4	21%	114.3
A + B	Re-circulated Air	140.8	19%	116.0	146.9	36%	96.4
	Outdoor Air Supply	143.7	16%	121.9	143.0	30%	102.0

The children in classroom A tended to work more accurately in both addition ($p<0.07$) and subtraction ($p<0.07$) tasks and slight improvement in the overall performance of addition ($p<0.068$) was noticed at the higher ventilation rate compared to low ventilation. Similarly, the pupils in classroom B made significantly less errors ($p<0.01$) and achieved better performance ($p<0.029$) during subtraction at the higher ventilation rate. For classroom B the changes in the performance measures of addition did not reach significance levels. However, when the data for classroom A and B were pooled under the common hypothesis that the children work better under improved ventilation, significant or close to significant improvement was obtained in the overall performance of both addition ($p<0.036$) and subtraction ($p<0.052$). Separate analysis was carried out for children with higher math skills (25 pupils for both classrooms), i.e. excluding those who had a higher than 50% error rate in these tasks. This analysis resulted in similar but more significant effects than those for individual classrooms above. The children with higher math skills increased the number of error-free units in both addition ($p<0.02$) and subtraction ($p<0.007$) tasks when working under the improved ventilation conditions.

DISCUSSION

The CO_2 patterns over a school day are closely linked to the daily activities performed within or away from the classrooms and whether windows and doors are left open or not. The levels presented in Figure 2 reflect a situation when the classroom was occupied throughout a full school day and no windows were opened due to cool outdoor conditions without sunshine. Double glazed windows, installed at the majority of the schools studied, allow very little air infiltration. If windows are left closed in the absence of other means of providing a minimum amount of outdoor air CO_2 levels rise quickly (typically within 15-20 minutes) to 3000-4000 ppm under normal occupancy. Similar high levels in naturally ventilated classrooms have often been reported in UK schools [6], [10]. Adverse health effects associated with CO_2

exposure below 5000 ppm are difficult to evaluate since there are a number of other factors such as high pollution level from off gassing of building materials and elevated allergen concentration, appearing at low ventilation rates that also affect human wellbeing [20]. However, possible alteration in breathing and heart rate as well as loss of concentration and wellbeing due to CO_2 exposures in the range between 3000-5000 ppm may be expected [21]. Other adverse health effects due to CO_2 exposure such as dyspnea, headache, dizziness and lethargy were found mainly in medical investigations and short term exposures to CO_2 concentrations above 1% (10000 ppm) [22].
The thermal conditions during the first testing week were generally cooler both indoors and outdoors compared to the second week of testing. Therefore the average temperatures were somewhat higher under the re-circulated condition in classroom A and under improved ventilation in classroom B. Considering that the thermal environment may also affect work performance [11] the thermal conditions in classroom A would be in favour, and in classroom B would counteract the expected changes in performance due to improved ventilation. However, the alterations in temperature between the present experimental conditions were relatively small compared to those in which such effect were shown [11] and therefore these may be considered not to affect the present performance results. On the other hand the thermal conditions in classroom B have to some extent affected the pupils' perception in air freshness. Although the air quality conditions were improved the pupils did not perceive significant improvement in air freshness mostly due to the increased enthalpy of inhaled air at high ventilation [23]. The children in classroom A who effectively perceived a change in air freshness under improved ventilation also reported more positive effects in neurobehavioral symptoms (alertness, attention, tiredness) and work mood in contrast with children in classroom B.
A significant impact of the ventilation rate on the school work performance of pupils was observed in both classrooms. Summarizing the effects the overall performance of all children increased under improved ventilation by 5.1% and 5.8% for both addition and subtraction respectively. These effects were even stronger for the pupils with higher math skills. They increased their math performance by ~7% when working under the improved ventilation conditions. The magnitude of such effect is in the expected range that was seen in earlier studies investigating work performance due to improved ventilation rates [7], [11].

The present results strengthen the evidence of earlier findings that improved ventilation has beneficial effect on pupils' learning performance. Without intervention the existing ventilation rates in naturally ventilated school buildings remain below the minimum recommended levels if thermal conditions do not influence people to open windows. Measures that allow a minimum supply of fresh air to the classrooms of naturally ventilated buildings are needed particularly if windows are not operated adequately to control ventilation.

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Study on Productivity in the Classroom (Part 3)
Nationwide Questionnaire Survey on the Effects of IEQ on Learning Performance

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SUMMARY

Many research papers have been published on the potential effects of the quality of the indoor environment on productivity in classrooms and offices. This paper (Part 3) reports the result of nationwide field measurements based on subjective questionnaire surveys and objective test scores in a unified way.

It investigated 83 branches of a college, which are located in regions spread throughout Japan. The total number of subjects was about 4000. Objective learning performance was evaluated according to scores in standardized quizzes to measure the level of understanding of lectures. In addition to the objective evaluation using quiz scores, a subjective evaluation of learning performance was carried out using a questionnaire as a self-assessment form.

Percentage of dissatisfied of each schoolhouse did not have the correlation with quiz scores but had a significant correlation with a subjective evaluation of learning performance by a linear approximation ($p<0.006$).

INTRODUCTION

This series of studies evaluate the effects of changes in the air quality and thermal environments on student performance in the classroom. We have already reported the effect of indoor environmental quality on student performance based on field intervention surveys and laboratory experiments.

This paper (Part 3) reports the result of nationwide field measurements based on subjective questionnaire surveys and objective test scores in a unified way. In this research, the relationship between the quality of the indoor environment and student performance in college are linked quantitatively through this nationwide questionnaire survey and previous laboratory experiments. In addition, human psychology regarded to effect it most when learning performance is measured was evaluated “motivation for learning” using a questionnaire as self-assessment form.

NATIONWIDE QUESTIONNAIRE SURVEY METHODS

Questionnaire surveys were conducted to evaluate the relationship between satisfaction level and learning performance at branches of Nikken Gakuin College – i.e. the college used for the intervention surveys in the previous paper (Part 1) [Murakami, et.al, 2006].

Questionnaire surveys were carried out three times in total: in February 19, 2006 (winter) April 2, 2006 (spring), and June 11, 2006 (summer) at branches of the 83 Nikken Gakuin

College scattered over all regions of Japan. Two evaluations methods – an objective evaluation based on quiz score and a subjective evaluation based on a questionnaire on psychological factors – were used, and consistency was evaluated as well. In addition, human psychology regarded to effect it most when learning performance is measured was evaluated "motivation for learning" using a questionnaire as self-assessment form.

The lecture started at 9:00am. After the 180-minutes lecture ended (12 noon), the subjects took a 30-minute quiz, and then filled out the self-assessment form (questionnaire). Three five-minute breaks were provided during the 180-minute lecture.

The trial subjects were students taking a course for the qualifying examination for first-class architects. Those students were all highly motivated because nearly all of them were to take a qualifying examination scheduled for July. The total number of subjects was about 4000, most of whom were employed and in their twenties to forties. Since the students need to attend all of the lectures based on the curriculum provided by the college, the subject groups in individual measurement cases were almost the same.

Evaluation of learning performance

1) Evaluation of Objective Learning Performance

Objective learning performance was evaluated according to scores in standardizes quizzes to measure the level of understanding of the lectures. The purpose of the lectures was to prepare students to take the qualifying examination for first-class certified architects. Each standardized quiz consisted of 20 questions, each of which was answered by choosing one out of five options. Table 1 shows questions in a typical standardized quiz. To compare scores in quizzes for different lecture content, an adjestment was made to the scores based on data on the average scores in the examinations conducted by Nikken Gakuin College in fiscal 2006, and the difficulty levels of all examinations were standardized.

Table 1. Questions in Standardized Quizzes for the Objective Evaluation (in the field of architectural planning)

Question 10; Which of the following descriptions of different wiring methods used in office construction is the **most incorrect**?	
(1)	In the free-access floor wiring method a double floor is constructed, and the method uses the space between floors as the wiring space; this has the effect of reducing the design load on the floor.
(2)	The floor on the standard floor was made to the free-access floor of 6cm in height, and to correspond to the change in the layout of the office, considered in the office building.
(3)	In the under-the-carpet wiring method a thin cable is laid directly on the floor level, and a special floor finish is needed. However, it is possible to correspond to the change easily.
(4)	It wires a necessary place, and the bus baton wiring method is large the maximum, permissible current, and in the method to accommodate and to protect the conductor in this, is suitable for a mass power supply.
(5)	In general, the conductors used in the bus baton wiring method are made of copper or aluminum.

2) Evaluation of Subjective Learning Performance

In addition to the objective evaluation using quiz scores, a subjective evaluation of learning performance was carried out using a questionnaire as a self-assessment form. Table 2 shows the self-assessment form. Subjective learning performance is evaluated with Questions 4-(1) and 4-(2) shown in Table 2. The filing return made it answer by the answer sheet exam. Since a choice prepared only to ten pieces, the choice was set up based on the reply obtained by the previous paper (Part 1).

Table 2. Self-Assessment Form for the Subjective Evaluation

Question 1; **Yourself**
(1) Gender? 1.) Male 2.) Female
(2) Age? 1.) 10's 2.) 20's 3.) 30's 4.) 40's 5.) 50's 6.) 60's
Question 2; **Thermal Environment** (Heat and Cold)
(1) How does the temperature feel?
1.) Cold 2.) Slightly Cold 3.) Cool 4.) Moderate 5.) Warm 6.) Slightly Hot 7.) Hot
(2) Are you satisfied with the current thermal environment?
1.) Dissatisfied 2.) Slightly Dissatisfied 3.) Neutral 4.) Slightly Satisfied 5.) Satisfied
Question 3; **Air Environment** (Contamination and odor)
(1) Are you satisfied with the current air environment?
1.) Dissatisfied 2.) Slightly Dissatisfied 3.) Neutral 4.) Slightly Satisfied 5.) Satisfied
Question 4; **Understanding level** of lecture contents
(1) Please convert the effect and/or the frequency of effect of the indoor environment in the classroom on your understanding of today's lecture contents into lost time.
1.) 0 min 2.) 3 min 3.) 5 min 4.) 10 min 5.) 15 min
6.) 20 min 7.) 25 min 8.) 30 min 9.) 40 min 10.) More than 50 min
(2) How is the understanding level of the content of the lecture thought to improve if the factor of the indoor environment improves the understanding level of today's content of the lecture as 100%?
1.) 0% 2.) 5% 3.) 10% 4.) 15% 5.) 20%
6.) 25% 7.) 30% 8.) 40% 9.) 50% 10.) Over 60%
Question 5; **Motivation for Learning** (Motivation to absorb content of lecture)
(1) After entering your classroom, how high was your motivation for learning (Motivation to absorb content of lecture)?
1.) 0 -10 2.) 10-20 3.) 20-30 4.) 30-40 5.) 40-50
6.) 50-60 7.) 60-70 8.) 70-80 9.) 80-90 10.) 90-100

Evaluation of motivation for learning

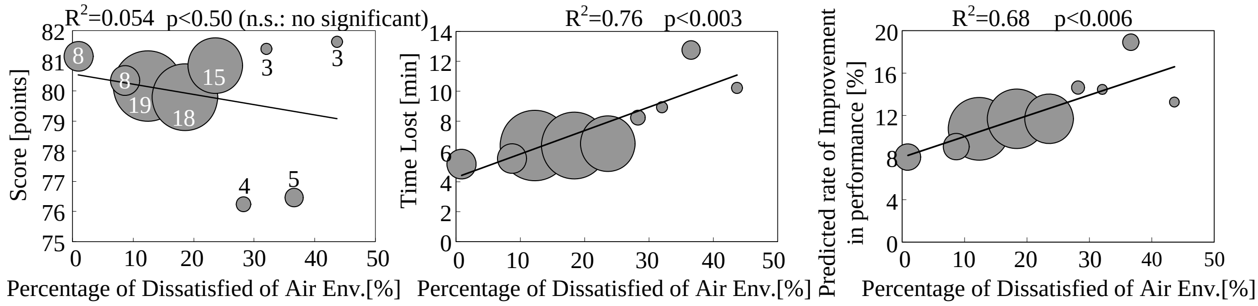
In this research, motivation for learning is defined as "motivation to absorb the content of the lecture." The investigation about motivation for learning tries evaluation of the motivation by which evaluation quantitative until now was not made almost, though it is a very important parameter in productivity research. Motivation to learn during the lecture was assessed using a questionnaire as a self-assessment form, too. Motivation for learning during lecture is evaluated Question 5 shown in Table 2.

The significance level was set at 5% and a corresponding t-test was used to compare the quiz results with varying environmental conditions. The Wilcoxon matched-pairs signed rank test was used as a corresponding rank scale in comparing the results of the self-assessment with varying environmental conditions.

RESULTS / LEARNING PERFORMANCE

Between learning performance and the percentage dissatisfied with the air environment

Figure 1 shows the relationship between the learning performance and the percentage dissatisfied with the air environment. It classified according to the unit 5% after calculating the rate of dissatisfied of air environment for every school building. Furthermore, the average value was calculated. The size of the plot in a figure expresses the number of school buildings in the point. The number of school buildings is shown during the plot. That is, the average value of the learning performance of the school building shown in the number under plot is expressed.



a) (Left) Quiz score vs. Dissatisfaction
b) (Center) Time lost vs. Dissatisfaction
c) (Right) Predicted rate of improvement in performance vs. Dissatisfaction

Figure 1 Learning Performance vs. Percentage Dissatisfied with the Air Environment

Figure 1a) shows the relationship between the results of the object evaluation (quiz score) and percentage dissatisfied with the air environment. Although there was a tendency for the quiz scores to improve as the percentage dissatisfaction decreased, there was no significant correlation between the two.

Figure 1b) shows the relationship between the results of the self-assessment of time lost due to the indoor environment and the percentage dissatisfied with the air environment. Time lost due to the indoor environment is based on the response to Question 4-(1). A linear relationship was observed between the subjective evaluation of time lost and the percentage dissatisfied with the air environment (R^2=0.76, $p<0.003$). In addition, the notation is omitting because the items of the number of school buildings are the same as that of the result with objective evaluation in figure 1a).

Figure 1c) shows the relationship between the results of the self-assessment "predicted rate of improvement in learning performance with an improvement in the environment" and the percentage dissatisfied with the air environment. The subjects were requested to report the learning performance improvement rates that they expected if the indoor environment was improved. Predicted rate of improvement in performance is based on the response to Question 4-(2). A linear relationship was observed between the subjective evaluation of the predicted rate of improvement in performance and the percentage dissatisfied with the air environment (R^2=0.68, $p<0.006$).
In addition, it got the same result the relationship between the results of thermal environment and learning performance.

Between objective evaluation and age

Figure 2 shows the relationship between the quiz scores and age. A linear relationship was observed between quiz scores and age (R^2 = 0.89, $p<0.06$ (n.s.: no significant)). It is thought that it was generated by accumulating experience, whenever it passes through age for preparation of the qualifying examination carried out every year. It was a result with the same said of April and June. In productivity research, in case working and study performance are measured, there is the necessity that experience must be taken into consideration.

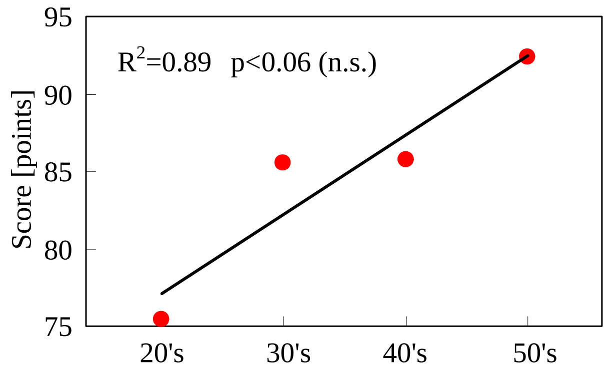


Figure 2 Object Evaluation vs. Age (February)

RESULTS / MOTIVATION TO LEARN

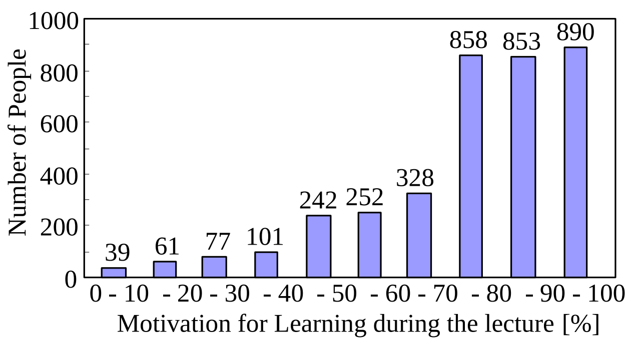
Distribution and classification of motivation to learn

Figure 3 shows the distribution of the motivation to learn during lectures in February. 70.3% of 2601 subjects among 3701 subjects reported themselves with 70% or more. It turns out that motivation for learning is kept high towards a qualifying examination. It was a result with the same said of April and June.

Then, the subjects were put into two groups: a higher motivation group (=M_d(H)) and a lower motivation group (=M_d(L)) to analyze the motivation during lectures on learning performance. Subjects in the higher motivation group were defined as those with 70% or more in the self-assessment.

Evaluation of learning performance by motivation group

Figure 4 shows the results for objective learning performance by the motivation group. The quiz scores brought a significant high of 4.5 points ($p<1.1\times10^{-6}$) in February, 6.3 points ($p<6.5\times10^{-14}$) in April, and 2.6 points ($p<5.9\times10^{-5}$) in June by the M_d(H) group. It became clear that the motivation to learn during the lectures significantly influencing the quiz scores, and the importance of evaluating the motivation to learn was shown. To measure the learning performance more exactly, it is thought necessary to incorporate motivation to learn as an evaluation axis.



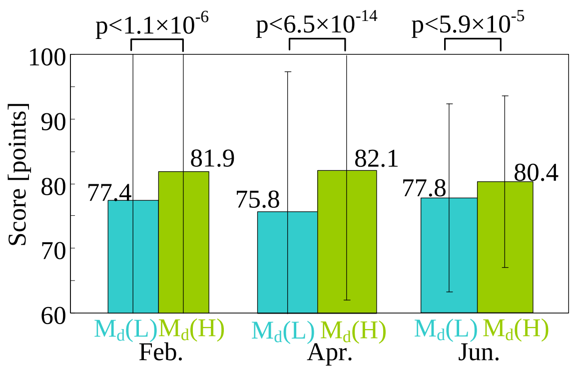


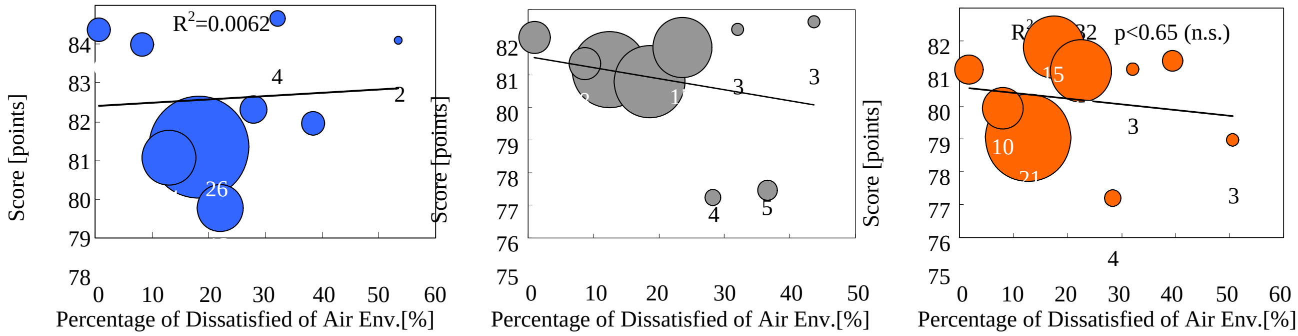
Figure 3 (Left) Distribution of the motivation to learn during lectures (February)

Figure 4 (Right) Results of the objective learning performance by the motivation group

DISCUSSION / DETAILED EXAMINATION OF THE LEARNING PERFORMANCE

Season

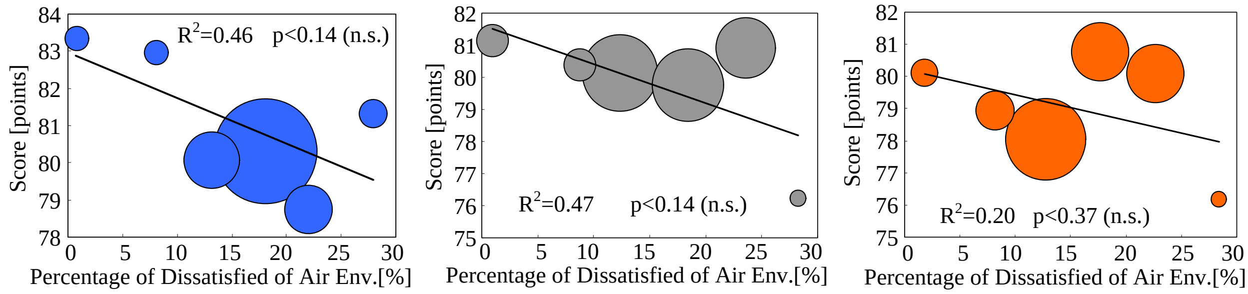
Figure 5 shows the relationship between the quiz scores and the percentage dissatisfied for air environments in every season. There was no significant correlation between the two in which season.



a) February b) April c) June

Figure 5 Object Evaluation vs. Percentage Dissatisfied with the Air Environment by Season

Next, Figure 6 shows the result of having excepted the school building whose rate of a dissatisfied of air environment was 30% or more. A fairly strong linear relationship was observed between the quiz scores and the percentage dissatisfied with the air environment, with R^2=0.46 in February, 0.47 in April, and 0.20 in June. In June, although the percentage dissatisfied with the air environment is high, a tendency with the sufficient quiz score can be observed. It is possible that students are carrying out the report by the side of nearby dissatisfaction more till because they become sensitive to environment then when the qualifying examination has approached. Moreover, even if dissatisfied, a possibility of doing themselves best can be considered.

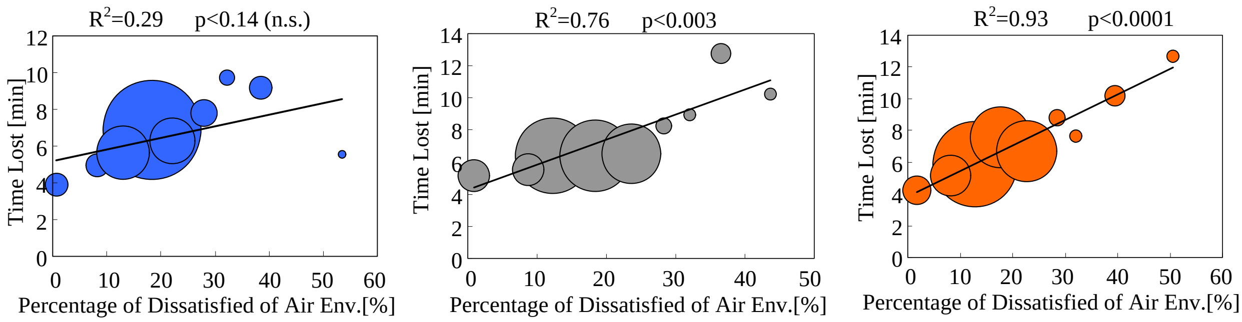


a) February b) April c) June

Figure 6 Quiz score vs. Dissatisfied (Excepted dissatisfied was 30% or more)

Figure 7 shows the relationship between the time lost due to the indoor environment and the percentage dissatisfied with the air environment in every season. A linear relationship was observed between time lost and percentage dissatisfied with the air environment, with R^2=0.29 in February, 0.76 ($p<0.003$) in April, and 0.93 ($p<0.0001$) in June. The correlation becomes stronger as the qualifying examination approaches. It was a result with the same said of predicted rate of improvement in learning performance with an improvement in the environment.

From the above result, it is possible that relation is the period to a target, and the objective and subjective evaluation.

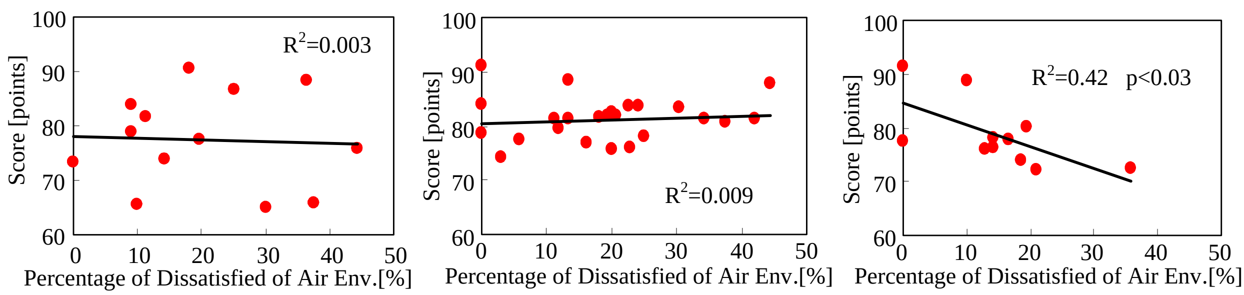


a) February b) April c) June

Figure 7 Subject Evaluation vs. Percentage Dissatisfied with the Air Environment by Season

Area

Figure 8 shows the relationship between the quiz scores and percentage dissatisfied with the air environment in every area for April. As candidates for the survey, Hokkaido and Tohoku were chosen as cold climates with 13 branches, Kanto as a moderate climate with 25 branches, and Kyushu as a hot climate with 11 branches. No relationship was observed between quiz scores and percentage dissatisfied with the air environment in cold and moderate climates. A linear relationship was observed with a significant correlation between the quiz scores and dissatisfaction in the hot climate (R^2=0.42, p<0.03). It was a result with the same said of February and June.



a) Hokkaido, Tohoku (Cold climate) b) Kanto (Moderate climate) c) Kyushu (Hot climate)

Figure 8 Object Evaluation vs. Percentage Dissatisfied with Air Environment by Area (April)

In addition, it got the same result the relationship between the results of thermal environment and learning performance.

CONCLUSIONS

(1) Objective learning performance improved with a decrease in the percentage dissatisfied with the air environment.
(2) Subjective learning performance improved significantly with a decrease in the percentage dissatisfied with the air environment.
(3) Objective learning performance improved with age.
(4) Objective learning performance in the high motivation group was significantly better. To measure the learning performance more exactly, it is thought necessary to incorporate motivation to learn as an evaluation axis.
(5) It is possible that relation is the period to a target, and the objective and subjective evaluation.

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Indoor climate and improvement possibilities in educational premises

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SUMMARY

The results of indoor climate investigation in 7 schools of Tallinn are analyzed in the paper. In classrooms with natural air change carbon dioxide concentration at the end of the class is very high, up to 1.5….3.5 times more than permitted level. With high occupation rate in gymnasiums of Tallinn the good indoor climate and permitted CO_2 concentration at the end of the class can be provided only with the use of balanced ventilation.

INTRODUCTION

Over the last few decades, considerable attention has been directed toward the problems of indoor air quality in schools. It has become increasingly clear that exposure to contaminated indoor air may not only be unpleasant, but can have serious adverse health effects.
Children of today spend increasingly more of their time in school environment.
Because children breathe a greater volume of air relative to their body weight compared to adults, they may be more sensitive to indoor air pollution [1, 2].
Indoor air quality in schools is influenced by many factors, like air temperature, relative humidity, air velocity, radiant temperature, textured surfaces, activities of the occupants in the building, the season, etc [3].
The available floor area per person in classrooms, around 1.5-2.0 m^2, is much less than that in office rooms, where 8-10 m^2 per person is normal. This implies a much larger ventilation demand in schools to remove contaminated indoor air.
Indoor climate in school buildings has been researched in many countries. One of the interesting questions is the carbon dioxide level in classrooms at the end of the class.
There are approximately 6000 school buildings in Finland, in which more than 700,000 people spend several hours of their time every day. Ventilation and indoor air quality of elementary and secondary schools were studied and in general the complaints about poor indoor air quality were more common in the schools with natural ventilation system than with mechanical ventilation systems, being least frequent with mechanical supply and exhaust air ventilation. Before the renovation process, the contents of carbon dioxide in the school buildings ranged between 1200...2400 ppm [4].
In 1996 and 1997 research was done about CO_2 concentrations for 96 classrooms in 38 Swedish randomly selected schools, 61% of them had mechanical supply and exhaust air systems while the reminder had natural ventilation. Concentrations average was 900 ppm CO_2 and the maximum was 2800 ppm [5].
The outcome of a field study in 11 primary schools in Eindhoven, the Netherlands in 2005 shows that average CO_2 concentration in half of schools is near to 2000 ppm and maximum CO_2 levels of 3000 or 4000 ppm in schools are no exception [6].
Indoor air quality testing in 28 classrooms in Warsaw (Poland) in 2000 shows that in most observations the maximum CO_2 level was from 2000 to 4000 ppm [7].

In 1991 CO_2 measurements in a non-random study of 9 U.S. non-complaint schools showed that concentrations ranged from about 400 to 5000 ppm (mean = 1480 ppm). CO_2 concentrations exceeded the 1000 ppm ASHRAE ventilation standard [8] in 74% of the rooms [5]. In Michigan schools CO_2 peak levels reached 2700 and 3300 ppm in a non-portable and portable classrooms [9].
There is evidence that many countries have significant and serious indoor environmental problems in schools. From the educational standpoint, the indoor air quality and ventilation in school buildings may affect the health of the children and indirectly affect learning performance.
In Estonia the indoor air quality in schools has not been investigated sufficiently. In 2005 a pilot research was done in a cold season. The main reason of investigation was carbon dioxide concentration and air change level determination in classrooms with natural ventilation [10]. In most school buildings in Tallinn (about 90%) ventilation systems are not renovated, but windows have as a rule been changed and external walls in single cases renovated. Only natural ventilation and window airing of classrooms is used in unrenovated school buildings. The results of measuring carbon dioxide concentration have shown that in classrooms with non-renovated ventilation, indoor air quality is very bad. At the end of the lesson the CO_2 concentrations reach the level of about 2000...3500 ppm, in single cases even up to 4400 ppm [10]. The cause of that is very low air change; 4–5 times lower than needed and insufficient airing during breaks or the lack of it.
Estonian Standard EVS839:2003 „Indoor climate"[11] permits carbon dioxide concentration 1000 ppm for category A, 1250 ppm for category B, 1500 ppm for category C and indoor temperature in classrooms 21…23°C for category A, 20…24°C for category B, 19…25°C for category C, relative humidity 25…45% and air velocity up to 0.2 m/s for cold season.
According to the regulation of the Estonian Ministry of Social Affairs [12] the permitted maximum carbon dioxide level in classrooms is 1000 ppm, relative humidity 30...70%, indoor temperature 19...25°C.
By prEN15251 recommended maximum values of carbon dioxide concentration in CO_2-controlled ventilation systems can be up to 800 ppm above the external air concentration for category C, 500 ppm for category B and 350 ppm for category A [13].

METHODS

Indoor air quality measurements were carried out in 10 classrooms of 7 typical gymnasiums in districts of Mustamae, Oismae and Center of Tallinn. The area of classrooms varied from 53 to72 m^2 and volume from 153 to 213 m^3, the age of students from 13 to 18 years, Table 1. Density in classrooms ranged from 1.6 to 3.3 m^2 per person.
The occupation rate in classrooms was from 44 to 92%, windows and doors were closed during the class. Information about the investigated classrooms and external air temperature and relative humidity are presented in Table 1. External air CO_2 level during the measurements was 410±30 ppm.
The duration of the measuring of parameters in classrooms was 40-45 minutes, except in the gymnasium of Sytiste, in which one class lasted 35 minutes. In same cases the change of indoor air parameters during breaks was measured. Dynamical registration of CO_2 concentrations make it possible to use CO_2 as tracer gas for air change measuring. Indoor air parameters were assessed at the breathing zone height of seated students, but about 1m away from the students. All classrooms were empty before the beginning of measurements. Initial indoor parameters are shown in Figures 1, 2 and 3.
Instruments used for measurements were Testo series 400 and 435 with probes for measuring

CO_2 concentration, relative humidity, indoor temperature and air velocity.
Natural ventilation and airing during breaks were used in the investigated schools, all of the windows in classrooms had been changed, in the gymnasium of Sytiste, in addition the envelope elements had been insulated.

Table 1. The characteristic data of investigated classrooms

School name	Area of class-room m^2	Vo-lume of class-room m^3	Num-ber of stu-dents	Age of stu-dents	No of class	Occu-pancy %	Exter-nal tem-pera-ture °C	Exter-nal relative humi-dity %
Russian gymn. of Oismae	55	167	17	16-17	2	0,47	5,0	95
Gymn. of Mooni 1	55	164	31	15-16	4	0,83	3,9	95
Technical Gymn. of Tallinn	56	167	34	14-15	2	0,92	7,0	99
Gymn. Liivalaia 1	59	194	17	14-15	6	0,44	0,3	98
Gymn. Liivalaia 2	54	179	25	13-14	7	0,80	0,3	98
Gymn. of Sytiste 1	72	206	27	16-17	2	0,65	-5,0	95
Gymn. of Sytiste 2	54	160	26	13-14	3	0,69	-5,0	93
Gymn. Arte 1	52	153	19	16-17	1	0,50	-18,0	83
Gymn. Arte 2	72	213	29	17-18	2	0,78	-18,0	83
Gymn. of Jarveotsa 1	55	152	28	13-14	4	0,75	-6,1	85

*The classroom in school investigated first
**The classroom in school investigated secondly

RESULTS

Main attention was paid to measuring carbon dioxide concentration and air change determination in a cold period: from November 2006 to January 2007. At the same time indoor temperature, relative humidity and air velocity were registered.

The dynamic change of CO_2 concentration during the class is presented in Figure 1. The rise of CO_2 concentration in different classrooms is from 550 to1600 ppm. Only in two classrooms the CO_2 level at the end of the class was less than 1500 ppm or near to it, but exceeded the permitted 1000 ppm [8, 12]. In different classes the rise of CO_2 concentration varies, depending mostly on the occupation rate of the classroom and the air change per student. The CO_2 concentration at the end of the class depends greatly on its initial level at the beginning of the class, the latter depending on how intensively the airing during the breaks took place.
Figure 2 shows the variable relative humidity level during the classes, ranging from 24% to 54%. As can be seen the relative humidity rises moderately and mainly stays within the permitted limits 30…70% [12]. Only in the Gymnasium of Arte (in the classroom investigated first) the relative humidity decreased, being caused by the low humidity content in external air at -18 °C, the low occupation rate of classroom and relatively high air change in classroom (when compared with other classrooms investigated). Permitted relative humidity in classrooms is achieved thanks to low air change.
Figure 3 shows the variation of indoor air temperature during the class. As can be seen the indoor air temperature ranges between 18 and 25.5 °C, which to some extent exceeds the permitted limits [12]. Only in the Gymnasium of Liivalaia the indoor temperature was too low 18-19°C.

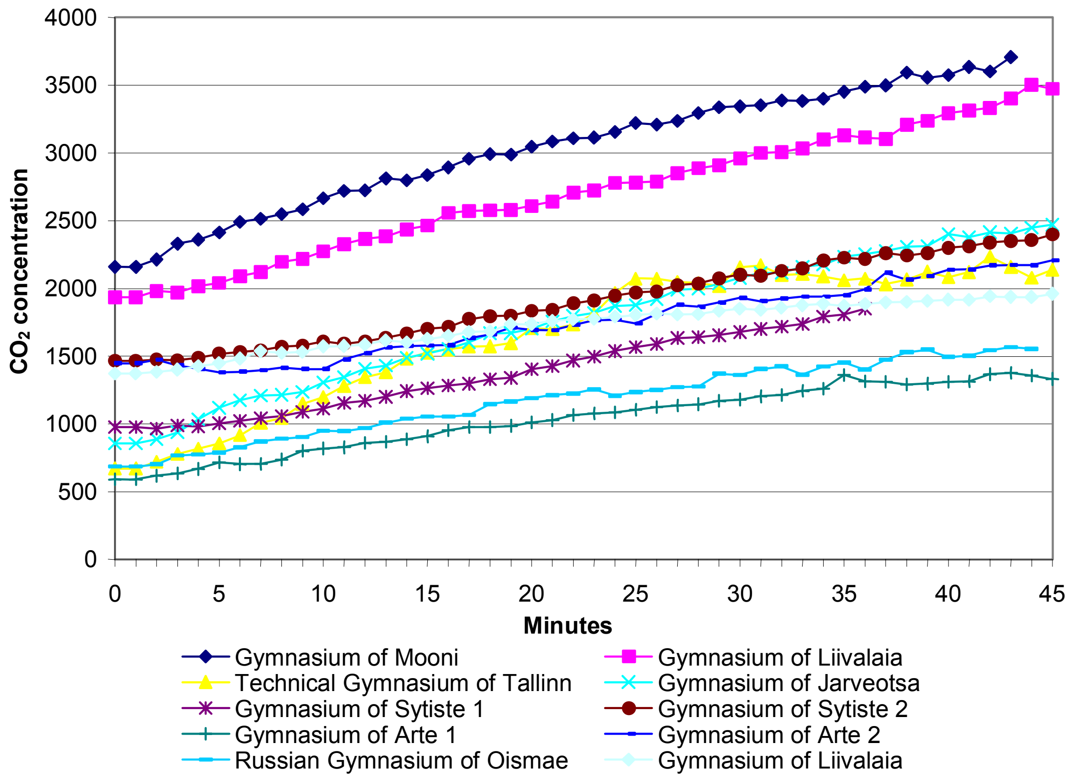

Fig.1. Variation of carbon dioxide in classrooms

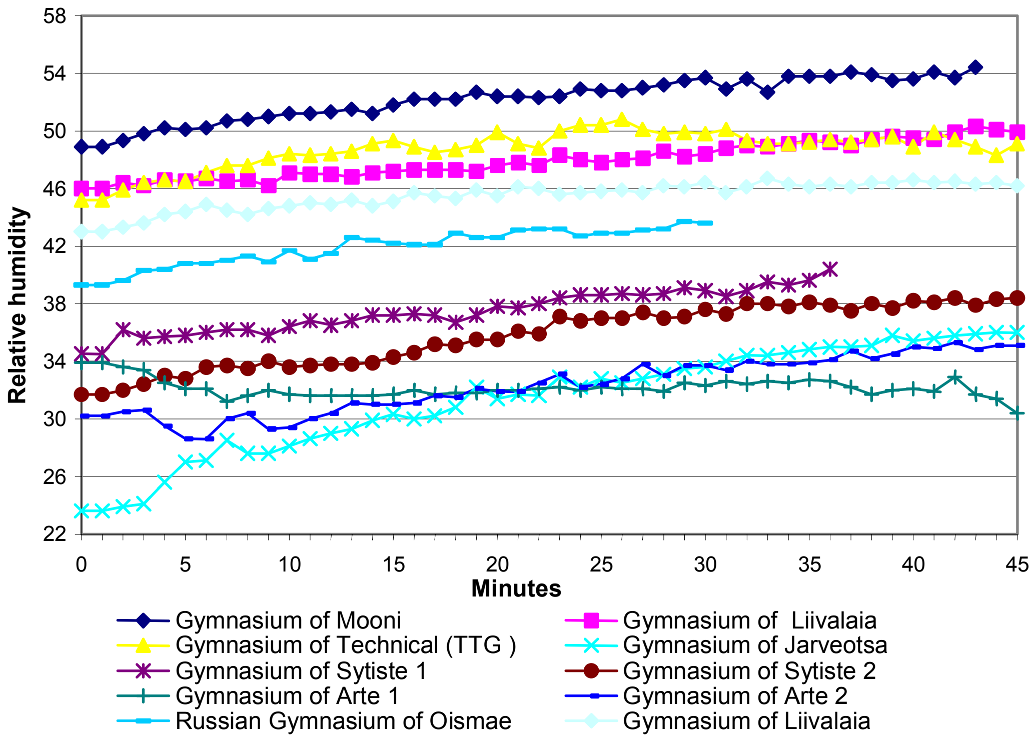

Fig.2. Variation of relative humidity in classrooms

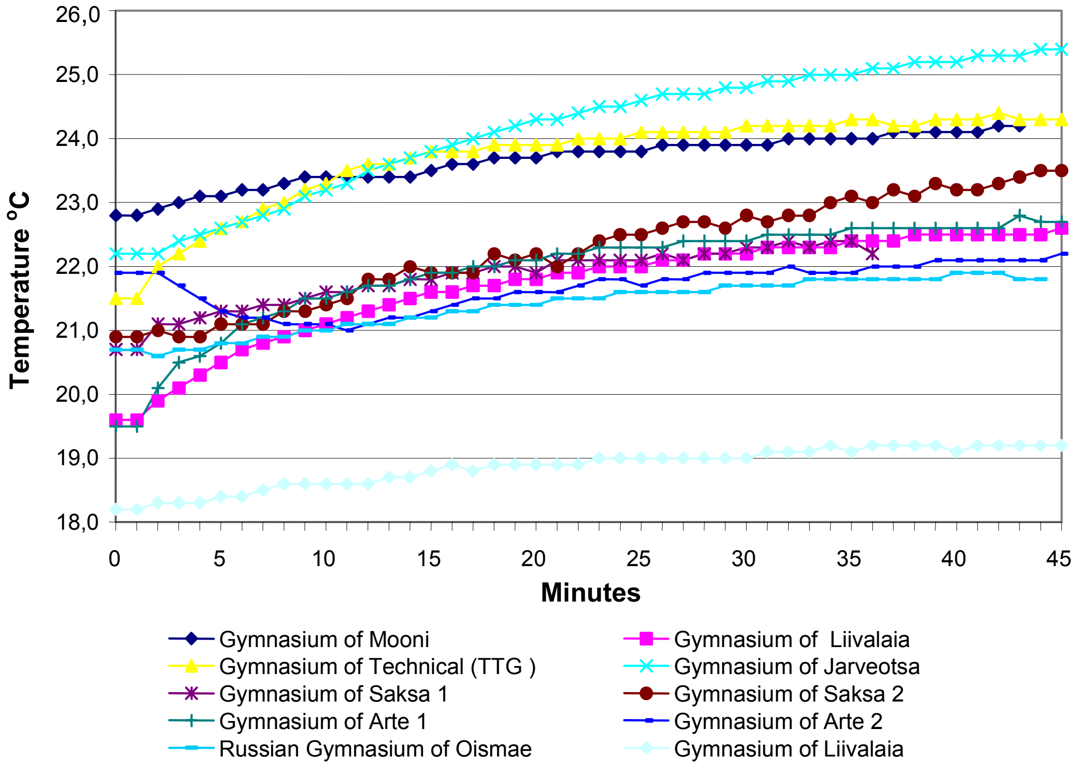

Fig.3. Variation of temperature in classrooms

Due to natural ventilation the air velocity in classrooms investigated was low: less than 0.1 m/s.

Dynamic registering of the carbon dioxide level enables us to determine air change in classrooms. To calculate the air change the following formula was used [10]:

$$\frac{L}{V}\cdot\tau = -\ln\frac{\frac{m}{L}+C_v-C}{\frac{m}{L}+C_v-C_o} \qquad (1)$$

where *m* is carbon dioxide generation in classroom, *L* is air change in classroom, *V* is volume of room, C_v is carbon dioxide concentration in external air (in supply air), *C* is carbon dioxide concentration in classroom air (in exhaust air), C_o is carbon dioxide concentration in the air of the classroom at the beginning of the class, τ is time.

The basic equation for carbon dioxide concentration C at time moment τ [10]:

$$C = C_v + \frac{m}{L} - \left(C_v + \frac{m}{L} - C_o\right)\cdot\left(e^{-\frac{L}{V}\cdot\tau}\right) \qquad (2)$$

The determined air change and other characteristic parameters of classrooms investigated are brought in Table 2. It shows that the air change is very modest: 0.3-3.9 l/s per student. With low air change and normal occupation rate the rise of CO_2 concentration is remarkable – 900-1600 ppm. Only in two classrooms the CO_2 concentration at the end of the class was up to 1500 ppm or near to it.

Table 2. Characteristic parameters of classrooms investigated

Name of school	Air change L/s	Air change L/s per pupil	Occupa-tion rate	CO_2 level at the beginning of class, ppm	Rise of CO_2 level, ppm
Gymnasium of Mooni	21	0.7	0.83	2160	1548
Gymnasium of Liivalaia (2)	7	0.3	0.8	1934	1540
Technical Gymnasium	60	1,8	0,92	670	1468
Gymnasium of Jarveotsa	60	2,2	0.75	887	1119
Gymnasium of Sytiste (1)	43	1,6	0,65	965	886
Gymnasium of Sytiste (2)	42	1,6	0,69	1467	932
Gymnasium of Arte (1)	59	4.0	0.5	591	785
Gymnasium of Arte (2)	62	2.1	0,78	1447	759
Russian Gymnasium of Oismae	61	3,6	0,47	686	868
Gymnasium of Liivalaia (1)	*30*	1,7	0,44	1373	587

DISCUSSION

With natural ventilation in classrooms and airing only during breaks it is practically impossible to keep the CO_2 concentration under the permitted limit, 1000 ppm [12], at the end of a class with full occupation rate.

With natural ventilation, the airing of classrooms during breaks is very important, without it the CO_2 level is usually over the permitted limit already at the beginning of the second class. In the classroom investigated the CO_2 concentration was 1.5 – 2 times over permitted value.

Calculations show (formula (2)) that in classrooms with average occupation rate, with an approximate air change of 1 l/s per m^2 and with good airing during breaks it is possible to hold the maximum CO_2 concentration on a level up to1500 ppm. This meets the carbon dioxide C class requirements by Standard of Indoor Climate [11], but does not correspond to the regulation of the Estonian Ministry of Social Affairs [12].

In practice, to gain the acceptable indoor climate, especially as regards CO_2, windows are kept opened also during classes.

To keep the CO_2 concentration under the permitted limits in fully occupied classrooms mechanical ventilation is needed. In our practice sometimes mechanical exhaust ventilation is used with fresh air supply through the fresh air valves, but the contemporary solution is balanced ventilation.

The investment of the first solution is quite low, but maintenance costs are high and sometimes it is difficult to assure the convenient indoor climate (the danger of draught). This kind of solution requires warming up the supply air and usually needs to renovate the heating system.

With high occupation rate in gymnasiums of Tallinn good indoor climate and permitted CO_2 concentration at the end of the class can be gained only with the use of balanced ventilation [10].

It is advisable to use in a classroom displacement or wall confluent [14] ventilation, the latter increasing the efficiency of air change and reducing energy consumption.

The necessary air change should be 7-10 l/s per student [10], depending on the method used for air change.

To achieve good indoor climate and energy savings it is necessary in addition to renovating ventilation also to renovate heating systems – automatic control of heat output of heating coils is necessary.

ACKNOLEDGEMENT

The authors are grateful to the schools where different field measurements were carried out.

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Air distribution and temperature control in classrooms

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SUMMARY

Air distribution solutions for classrooms aiming to lower air velocities and good temperature control are studied by measurements in 6 schools and temperature simulations. Air velocity measurements showed good performance of duct and ceiling diffusers which provided maximum velocities less than 0.2 m/s and can be highly recommended for classrooms. The wall diffusers were clearly not suitable for classrooms due to high velocities up to 0.43 m/s. Displacement ventilation diffusers were very sensitive to supply air temperature, as with the temperature difference of 3 °C velocities up to 0.28 m/s were measured. Room temperature measurement results showed a typical problem with temperature control as at the end of the heating season the temperatures up to 25 °C were measured. The parametric simulations showed that high supply air flow rates up to 10 L/s per person and cool supply air down to 14–15 °C were needed for room temperature control.

INTRODUCTION

Many studies have shown that poor indoor environmental quality (IEQ) in office buildings can reduce the performance of office work by adults [1,2]. While it is well documented that IEQ in schools is both inadequate and frequently much worse than in office buildings [3], there is little direct evidence that classroom performance is being negatively affected [4].

Experimental interventions [5,6] show that increasing the outdoor air supply rate and reducing moderately elevated classroom temperatures significantly improved the classroom performance of many tasks, mainly in terms of how quickly each pupil worked (speed). In these experiments the room temperature was reduced from 25°C to 20°C and the outdoor air supply rate was increased from 5.2 to 9.6 L/s per person. Thus the results suggest that higher ventilation rates up to 10 L/s per person as well as more strict temperature control than commonly used can be recommended for classrooms. This means new challenges in the design of classroom ventilation and temperature control. High airflow rates and low supply air temperatures, which are needed for temperature control, may easily cause draft. This may be avoided by careful selection of air distribution solutions and terminal devices.

In this study, air distribution solutions for classrooms aiming to lower air velocities and good room temperature control are studied by measurements in classrooms and temperature simulations. Performance of wall, ceiling, duct and displacement diffusers is compared at ventilation rates up to 340 L/s per classroom and supply air temperatures up to 6°C lower than room temperature. Based on the results, design guidelines of air distribution and temperature control solutions leading to healthy and productive indoor environment in classrooms can be given.

METHODS

Air velocity, room temperature and CO_2 measurements were done in 6 schools. All schools were either relatively new buildings or renovated buildings, all having modern mechanical supply and exhaust ventilation systems with ventilation rates, corresponding at least to Finnish minimum code requirements for ventilation of new buildings.

a)

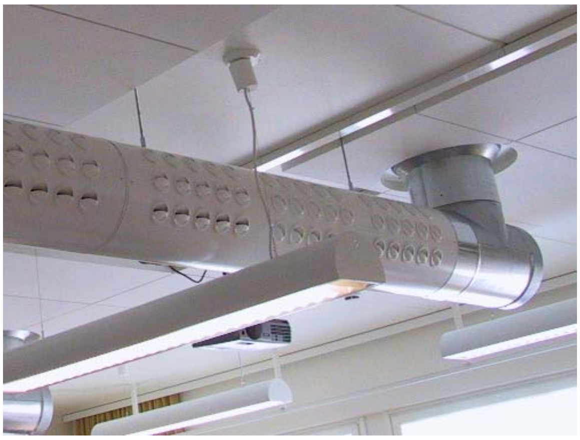
b)

Figure 1. School A with wall diffusers, a) and B with ceiling diffusers, b).

a)

b)

Figure 2. School C, a) and D, b) with duct diffusers.



a)

b)

Figure 3. School E with duct diffusers, a) and F with displacement diffusers, b).

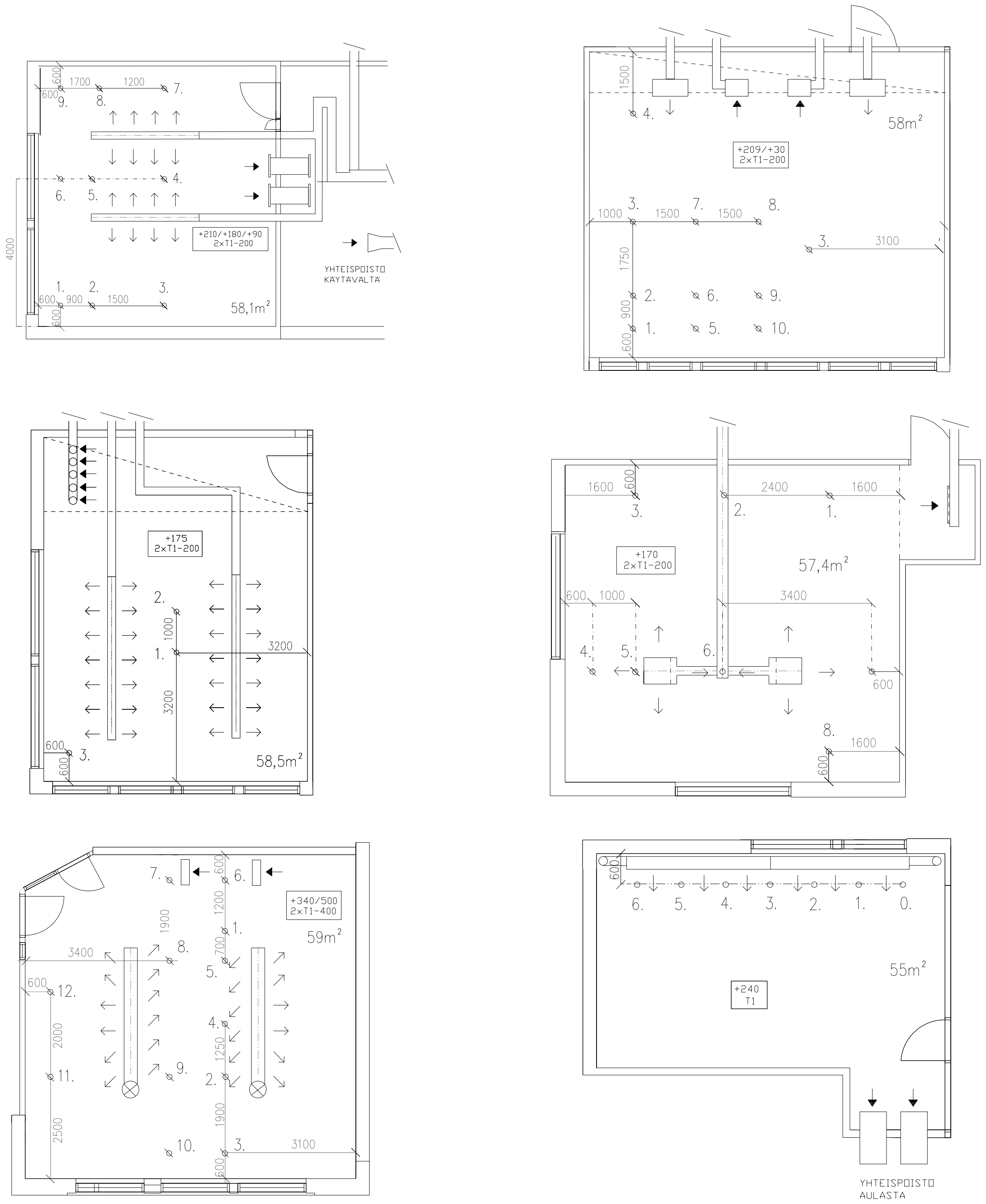


Figure 4. Air velocity measurement points in the classrooms (the classrooms are in the same order as in Figures 1–3).

School *A* had air distribution by ceiling diffusers (two diffusers per classroom) and school *B* by wall diffusers (two diffusers per classroom), which represents a low cost solution most commonly used, Figure 1. In three schools (*C*, *D*, *E*) duct diffusers (two ducts per classroom) and in school *F* displacement ventilation diffusers were used, Figures 2 and 3.

In schools from *A* to *D*, temperature and CO_2 concentrations were measured during a one week period in May 2006 from three classrooms in each school. Air velocity measurements were done in one classroom. In *E* velocity was measured in two classrooms and supply air temperature was set by 4 to 6 °C below the room temperature. In *F* data from previous measurements [7] was used to compare the performance of displacement ventilation to mixing ventilation. Air velocity measurement points were selected with smoke tests, so that the highest velocities could be measured. The measurement points are shown in Figure 4.

Both constant air volume (CAV) and demand controlled (DCV) ventilation systems with constant or controlled supply air temperature were used in schools. Differences between DCV and CAV systems and supply air temperature control options were studied with temperature and ventilation simulations for typical classrooms with a varying heat load.

RESULTS

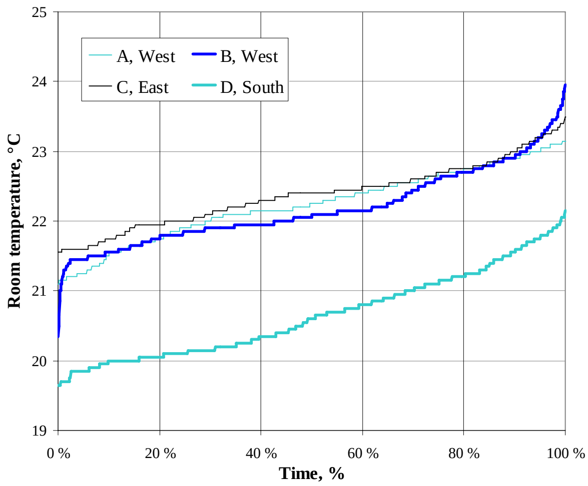
Measurement results in the schools

Supply air flow rates measured from terminal devices and typical occupancy in the classrooms is shown in Table 1. *E* was a new school having almost doubled supply airflow rate (design value of 12 l/s per person, 340 l/s in total) and also other target values of the highest indoor climate class. Schools *A* and *B* had CO_2- and CO_2&temperature controlled ventilation with 3 and 2 airflow steps respectively. In other schools constant air volume systems were used. *A* to *D* had a constant supply air temperature and in *E* and *F* supply air temperature was controlled according to exhaust air temperature.

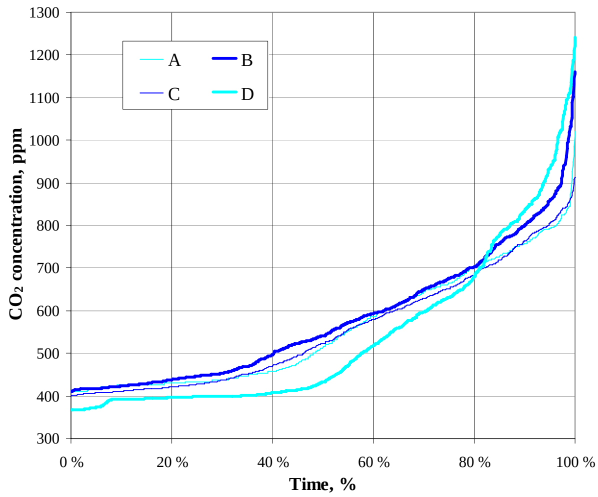
Table 1. Measured supply air flow rates and typical occupancy in the classrooms

School	Occupancy, pers.	Supply air flow rate, L/s per pers.	Supply air flow rate, L/s	Design supply air flow rate, L/s
A	20	7	138	210/150/90
B	27	7	186	210/30
C	22	6	136	175
D	20-25	6.8-8.2	168	170
E			348	340
F			180	180

Outdoor temperature during the measurement week was typical spring weather, between 9…12 °C. This represents the end of heating season and the results are compared to heating season target values. Room temperature and CO_2 results from classrooms where air velocity measurements were done are shown in Figure 5. Results are given only from the school time period which is from 8.00 to 15.00 on week-days. Temperatures measured from other classrooms in schools *A* to *D* (measured with loggers with a lower resolution compared to Figure 5) and previously measured temperatures from *G* are shown in Figure 6.

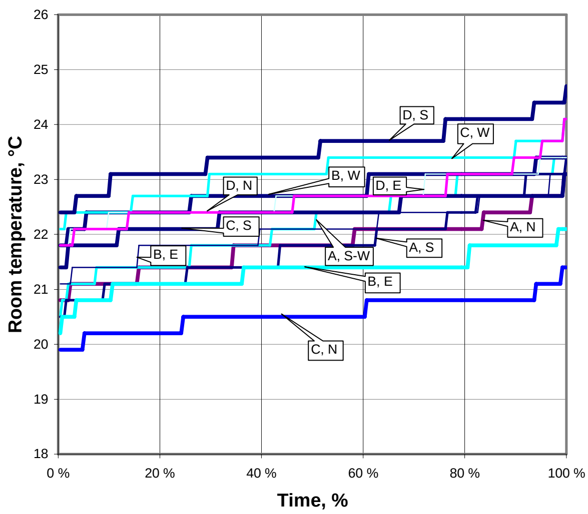


a)



b)

Figure 5. Duration curves of room temperature, a) and CO_2, b); data from 8.00 to 15.00 on week-days



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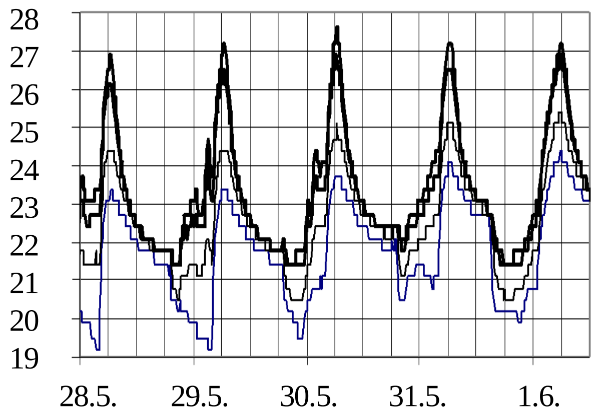


b)

Figure 6. Duration curves of temperatures in other classrooms in *A* to *D* from 8.00 to 15.00 on week-days, a) and temperatures from *E* during 28.5-1.6.2003, b)

Air velocity measurement results from the occupied zone are shown in Table 2. In *A* to *D* supply air temperature was not adjusted, but just measured. In *E* and *F* the effect of supply air temperature on draft was studied by changing the temperature of supply air and repeating the velocity measurements.

In *F* with displacement diffusers the results were very sensitive to the temperature difference. Air velocities at 0.1 m height, 0.6 m from diffusers, were 0.14…0.18 m/s when supply air temperature was equal to room temperature. Measurements with the temperature difference of 3 °C gave higher velocities, as shown in Table 2.

Table 2. Air velocity measurement results. Measurement locations are shown in Figure 4.

Meas. point	Air Velocity, m/s Measurement height 0.1m	1.10m	1.80m	Operative temperature, °C	Supply air temperature, °C	DT, room - supply air temperature, °C
A, duct diffusers, 138 L/s						
1	0.21	0.05	0.05	24.6	20.4	4.2
2	0.25	0.06	0.13	24.7	20.4	4.3
3	0.15	0.03	0.03	24.8	20.4	4.4
4	0.09	0.06	0.06	25.0	20.4	4.6
5	0.01	0.18	0.16	25.1	20.4	4.7
6	0.03	0.13	0.20	24.9	20.4	4.5
7	0.09	0.02	0.04	24.9	20.4	4.5
8	0.16	0.03	0.04	24.8	20.4	4.4
9	0.18	**0.29**	0.08	24.6	20.4	4.2
			Average, °C	24.8	Average, °C	4.4
B, wall diffusers, 186 L/s						
1	0.18	**0.43**	0.15	22.6	20.4	2.2
2	0.30	0.09	0.06	22.7	20.4	2.3
3	0.25	0.30	0.07	22.8	20.4	2.4
4	0.16	0.05	0.05	22.8	20.4	2.4
5	0.06	0.38	0.08	23.0	20.4	2.6
6	0.17	0.17	0.11	23.0	20.4	2.6
7	0.19	0.15	0.14	23.0	20.4	2.6
8	0.12	0.14	0.14	23.3	20.4	2.9
9	0.15	0.2	0.1	23.3	20.4	2.9
10	0.08	0.14	0.14	23.3	20.4	2.9
			Average, °C	23.0	Average, °C	2.6
C, duct diffusers, 136 L/s						
1	0.10	0.07	**0.16**	22.6	19.8	2.9
2	0.05	0.06	0.05	22.2	19.8	2.5
3	0.02	0.04	0.06	22.1	19.8	2.4
			Average, °C	22.3	Average, °C	2.6
D, ceiling diffusers, 168 L/s						
1	0.02	0.03	0.03	20.6	19.4	1.2
2	0.06	0.03	0.03	20.7	19.4	1.3
3	0.03	0.02	0.02	20.8	19.4	1.4
4	0.03	0.05	**0.14**	21.0	19.4	1.6
5	0.04	0.03	0.03	21.0	19.4	1.6
6	0.02	0.05	0.04	20.9	19.4	1.5
7	0.11	0.06	0.06	20.7	19.4	1.3
8	0.08	0.09	0.07	20.7	19.4	1.3
			Average, °C	20.8	Average, °C	1.4
E, duct diffusers 348 l/s						
2	0.11	0.09	0.09	21.70	17.50	4.2
3	**0.17**	0.09	0.09	21.70	17.35	4.4
4	0.09	0.07	0.08	21.60	17.20	4.4
5	0.15	0.10	0.06	21.50	17.20	4.3
6	0.08	0.09	0.12	21.40	17.15	4.3
7	0.07	0.11	0.13	21.40	17.30	4.1
8	0.16	0.11	0.11	21.50	17.35	4.2
9	0.06	0.05	0.04	22.00	17.35	4.7
10	0.05	0.04	0.05	22.20	17.40	4.8
11	0.08	0.06	0.07	22.00	17.85	4.2
12	0.10	0.08	0.08	21.80	16.75	5.1
			Average, °C	21.7	Average, °C	4.4
E, duct diffusers 348 l/s						
3	0.08	0.05	0.03	22.4	16.1	6.3
6	0.06	0.11	0.14	21.5	16.3	5.3
7	0.10	0.14	**0.19**	21.4	16.4	5.1
10	0.07	0.04	0.06	22.3	16.1	6.2
			Average, °C	21.9	Average, °C	5.7
F, dicplacement diffusers 180 l/s						
0	0.22					3
1	0.18					3
2	0.05					3
3	0.22					3
4	**0.28**					3
5	0.19					3
6	0.15					3

Room temperature simulations

Temperature control and ventilation rate options were simulated with IDA-ICE software for CAV and DCV systems without mechanical cooling in order to find good solutions for classroom ventilation and temperature control. The time period was chosen according to the typical use of schools: heating season from Jan 2 to May 14 and Oct 1 to Dec 23, and summer season from May 15 to May 31 and Aug 15 to Sept 30. Two classrooms were simulated as this configuration gave the same results as the configuration with six classrooms, Figure 7. A classroom with 30 students faced south and another with 20 students north. The occupancy profile used and the control curve of supply air temperature determined in the simulations are shown in Figure 8. For the south classroom, solar protection glasses were used.

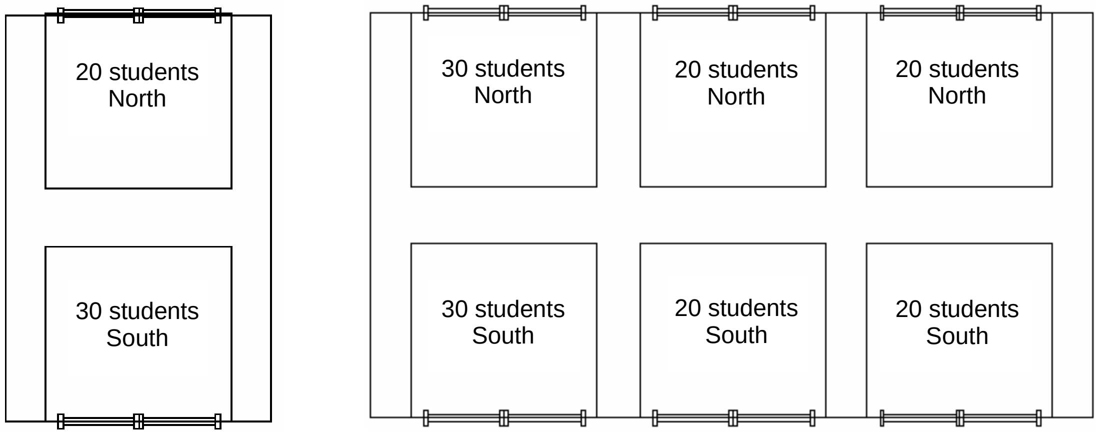


Figure 7. The configuration of 2-classrooms used in the simulations, a) The configuration of 6-classrooms used for the testing of 2-classroom configuration, b)

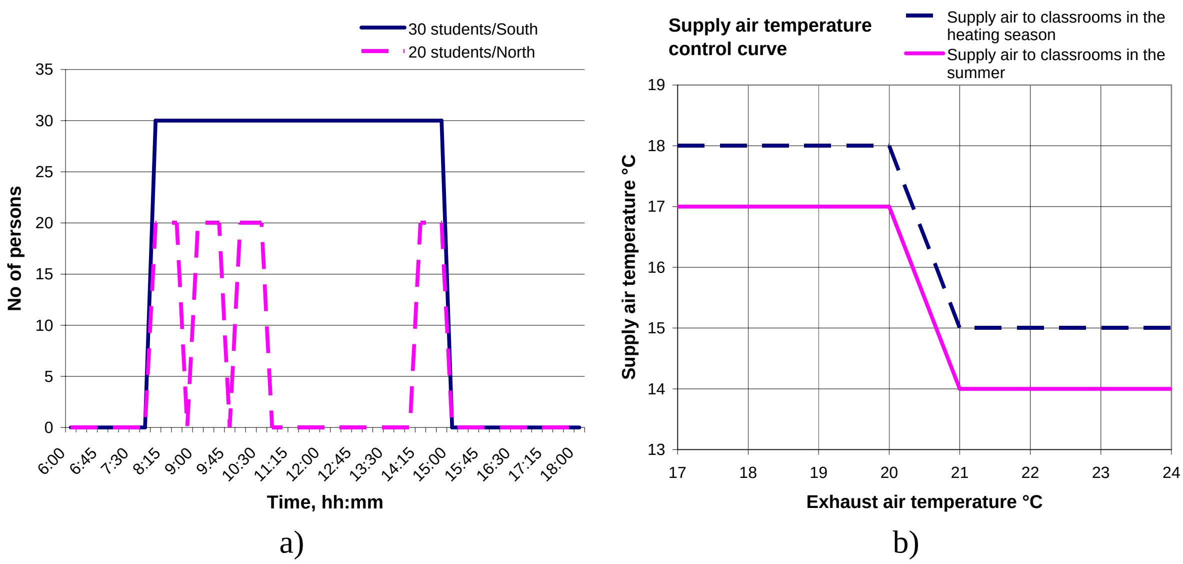
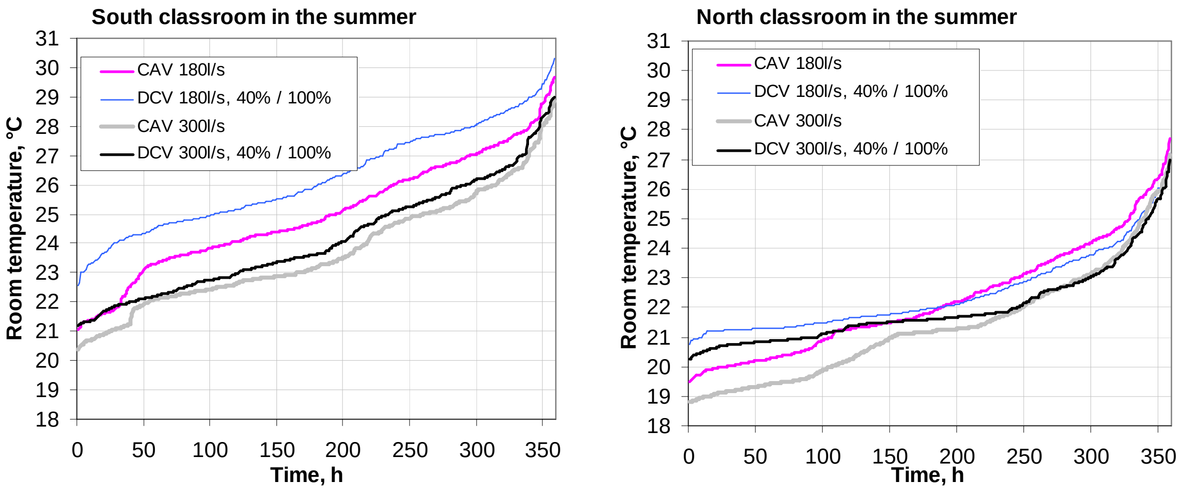


Figure 8. The occupancy profile of students, a) and determined supply air temperature control curve, b) (supply air temperature values can only be achieved if outdoor temperature is lower than supply air temperature, as there is no mechanical cooling in the system)

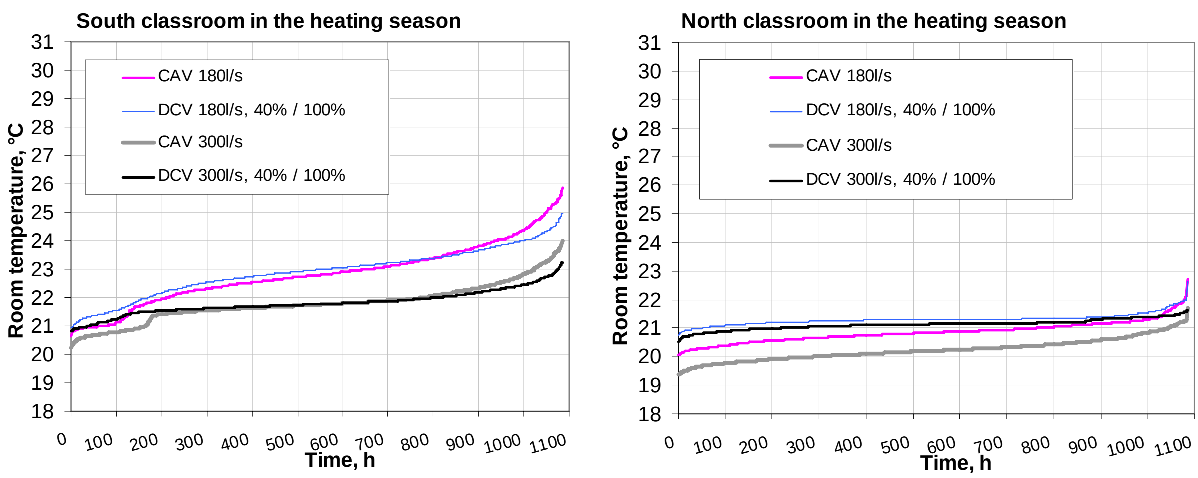
Two ventilation rates, 6 L/s per person, 180 L/s per classroom in total, and 10 L/s per person, 300 L/s per classroom in total, were simulated. For both rates CAV system and DCV system with CO_2 and temperature control was simulated. DCV system had two air flow steps, 100 %

and 40% of total airflow. The results are shown in Figure 9 for the summer period and in Figure 10 for the heating season. With night time ventilative cooling the summer period temperatures were possible to lower by about 1 °C from values shown in Figure 9.



a) b)

Figure 9. Summer period room temperature duration curves in the south classroom, a) and in the north classroom, b)



a) b)

Figure 10. Heating season room temperature duration curves in the south classroom, a) and in the north classroom, b)

DISCUSSION

Measured supply air flow rates were significantly lower than design values in three schools out of six. Due to relatively low occupancy the airflows per person were still sufficient, 6 L/s per person or more in all schools and CO_2 concentrations were less than 1200 ppm. Room temperature measurement results show a typical problem with temperature control as at the end of the heating season the temperatures up to 25 °C were measured. Simulated results show that this problem can be avoided without mechanical cooling if high enough ventilation rates and cool supply air is used. In the majority of the schools measured, ventilation rates were too low and supply air too warm for effective control of the room temperature.

Air velocity measurements showed remarkable differences between air distribution schemes. Duct and ceiling diffusers showed good performance with a maximum velocity less than 0.2 m/s and can be highly recommended for classrooms. The duct diffuser in *A* without nozzles (just a perforated duct) showed worse performance with a maximum velocity of 0.29 m/s. The wall diffusers were clearly not suitable for classrooms due to high velocities up to 0.43 m/s. Displacement ventilation diffusers were very sensitive to supply air temperature, as a temperature difference of 3 °C caused velocities up to 0.28 m/s.

The parametric simulations showed the effect of high internal gains: high supply air flow rates up to 10 L/s per person and cool supply air down to 14–15 °C was needed for room temperature control. However, mechanical cooling was not needed in the classrooms as schools are normally not used during summer holidays. DCV ventilation showed better performance in terms of thermal comfort when higher simulated airflow rate of 300 L/s per classroom was used. At lower airflow rate of 180 L/s the classrooms were less overheated with CAV ventilation. It was shown that a DCV system supplying 10 l/s per student and 300 L/s per classroom combined with night-time ventilative cooling ensured good temperature control in fully occupied as well as partly occupied classrooms. A similarly sized CAV system cooled down classrooms with low occupancy, thus, demanded additional heating of the supply air in classrooms. Air flow rate of the lower indoor climate category, 6 l/s per student and 180 L/s per classroom, resulted still tolerable but significantly higher temperatures, often above 23 °C in the heating season.

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Typologies of Hybrid Ventilation in Schools

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SUMMARY

This article is the result of a study on how to close the “gap” between architects and a building service consultant. Closing is necessary in order to be able to create both a better indoor climate and an interesting architectural environment. Especially for schools with natural air supply some basic physical principles of draught prevention should be checked in an early stage of the design or commissioning process. Air supply designs for the most common architectural problems should be easily available. Most of the presented design options have been evaluated with measurements and CFD-simulations. For schools with natural air exhaust (overpressure system) the results of an on-site test are presented. This system can be applied even in existing schools with single glass windows and a monumental façade.

INTRODUCTION

In The Netherlands and other European countries the air change rate in schools increases in order to prevent health problems of students. The minimum air supply in The Netherlands is 7 l/s (= ca. 25 m^3/h) per student. Due to the high occupancy (generally 1 person at 2 m^2 = 3.5 l/sm^2) prevention of draught needs much attention. Architects need ready-to-use ventilation design concepts for schools because of:

- A lack of knowledge of draught prevention,
- Individual architectural opinions,
- A low budget.

A research project is starting in The Netherlands to design options of natural, mechanical or hybrid ventilation. These systems for new or refurbished schools will be presented in a comprehensive way in a Dutch design guide and can be integrated in a REHVA-guide dealing with healthy schools in the near future. In this guide higher air change rates will be discussed as well: 35 m^3/h and 45 m^3/h per student = 4.9 - 6.3 l/sm^2.
Hybrid ventilation may have different meanings. Generally it comprises a low pressure natural ventilation system which can be supported by fans when necessary. However, a combination of natural air supply with mechanical exhaust or natural air exhaust with mechanical supply can be considered as hybrid as well.
Sometimes the building service consultant has to make design proposals for the architect. On the other hand architects can stimulate the creativity of a building service consultant to find a specific solution for a specific problem..

METHODS

The methods used during my study and subjects discussed with architects are:

a. Research of physical principles
b. Architectural research
c. Airflow measurements

a. Research of physical principles

The research provides a variety of options to deal with air flows and heat transfer:

- Mixing of cold air

Mixing of cold air with the surrounding warm air in order to prevent draught, without or with a false ceiling.

Research at the Delft Technological University [1] shows that it is possible to prevent draught (3.5 ls/m^2) by means of natural air supply without a ceiling. When the air is supplied just beneath the ceiling with an Archimedes-number of 0,001 (figure 1, II) the DR-value will be maximal 20%. However, it is possible to have an acceptable thermal comfort as well with other Archimedes numbers when the air will be supplied with a wide air inlet with the same size as the width of the room. When the velocity is low (figure 1, I) or very high (figure 1, III) the air velocity above the floor will increase. Concrete core and (floor) heating can - to a certain extend - compensate draught-risks by higher local operative temperatures near the floor.

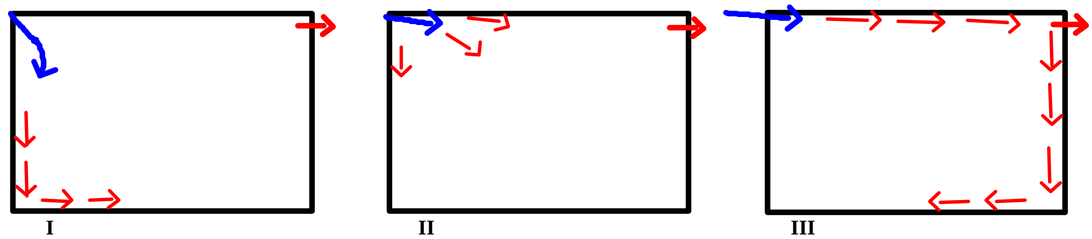


Figure 1: Air supply options without a false ceiling.

The integration of the air inlet at the ceiling needs much attention [2, 4], because of the fact that the narrow inlet-slit should be close under the ceiling.

In case of a false ceiling there are more possibilities of draught prevention and integration of an air inlet into the façade is more easy [4].

- An outlet via a double window

There is no information of the combined performance of an air outlet and a double window. The thermal qualities of the window and air flows are analyzed with CFD-simulations. The outlet system is tested in a police academy in Apeldoorn by measuring the air flow and pressure differences during the heating season.

- Heating systems

The most common heating systems are a radiator under windows, floor or concrete core heating and preheating of air of an air inlet. It is very important to understand the difference of the convection flow characteristic of radiator heating and floor heating. This has been evaluated with CFD-simulations (Phoenics, Flair).

b. Architectural research

Two systems are discussed:

1. Systems of natural air supply, mechanical exhaust and several combinations with ceilings for three different façade typologies (positions of windows).

After analysing the most general typologies of façades, design principles for architects are developed to support their design processes.

2. A system of natural air exhaust via a double window and mechanical air supply for existing monuments.

A design option has been developed for an existing building in which physical, aesthetic, practical and economic requirements are combined.

c. Airflow measurements and CFD-simulations

The presented natural air supply and natural exhaust systems have been measured or simulated.

The performance of the outlet system has been evaluated with the following equations:

The maximum air velocity through a vent with a very low air resistance is [1]:

$$U_{\max} = \sqrt{\frac{2 \cdot \Delta P}{\rho}} \quad (1)$$

Where U_{max} = the maximum air velocity (m/s), ΔP = the pressure difference in (Pa) and ρ is the volumetric mass of air (kg/m^3).

For instance, the maximum air velocity at 4 Pa and 20^{o}C will be 2.6 m/s. The measured air velocity of the system can be divided by the maximum air velocity. This is the airflow-efficiency, $\varepsilon_{airflow}$ (-):

$$\varepsilon_{airflow} = \frac{U_{measured}}{U_{\max}} \quad (2)$$

The air velocity is measured with a hot wire anemometer (Envic AFT-1D, 0-10 m/s, 3% tolerance) in the opening of the air exhaust system. The pressure difference is measured at both sides of the window (SETRA 267, +50 - -50 Pa, 1% tolerance). The measurements have a sample time of 1 minute.

RESULTS

a. Physical principles

In case of a false ceiling there are other options to prevent draught (figure 2). In this situation the warm air from the room en heath from the concrete ceiling is used to warm up the supplied cold air. The false ceiling has a large surface so there is more time for the cold air to mix with the warm air.

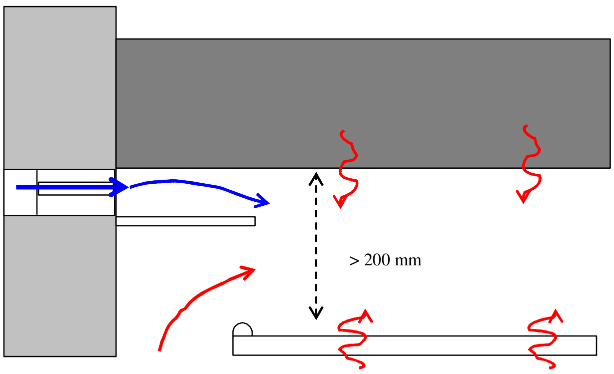


Figure 2. Presentation of some physical principles of draught prevention and promotion of mixing of cold and warm air flows, making use of a “common”-air inlet.

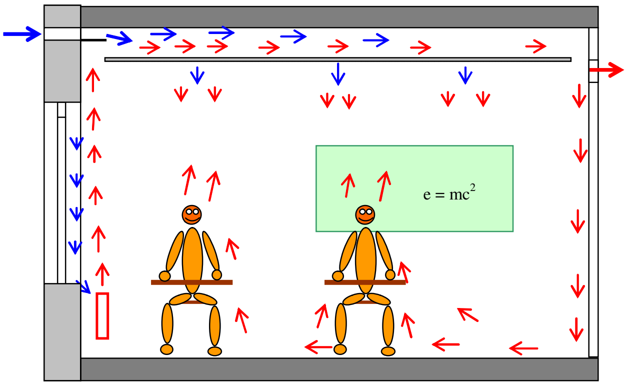


Figure 3. Presentation of the physical principles and comfort qualities of the system of figure 2. Draught is prevented by the false ceiling. Cold downward air flows from the air inlet are prevented by a “spoiler” connected to the air inlet.

Experience shows that a lot of discussions with architects are necessary to integrate this option in their specific design. Generally it is necessary to make draught problems visible for architects underline the draught-problem (figure 4).

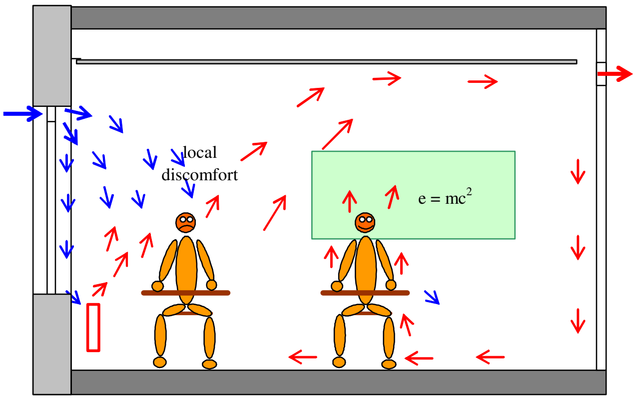


Figure 4. Presentation of local thermal discomfort of the system of figure 7b.

b. Architectural research

Some of the most common systems of natural air supply used by architects in The Netherlands are presented with the following figures:

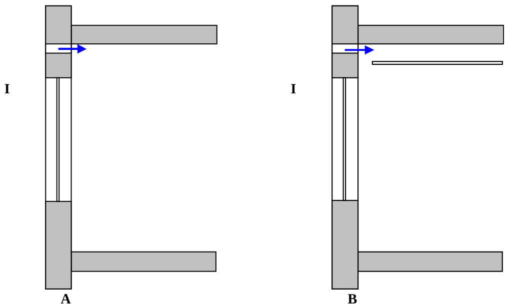


Figure 5. a) Air supply close to the ceiling, b) Air supply in combination with a false ceiling

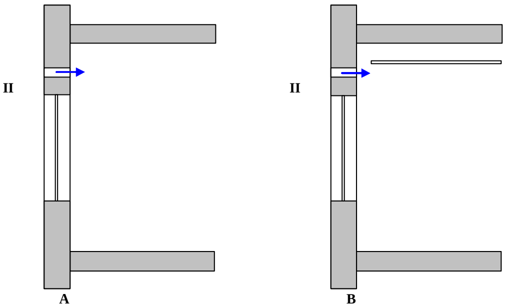


Figure 6. a) Air supply between the window and the ceiling, b) Combination with a false ceiling

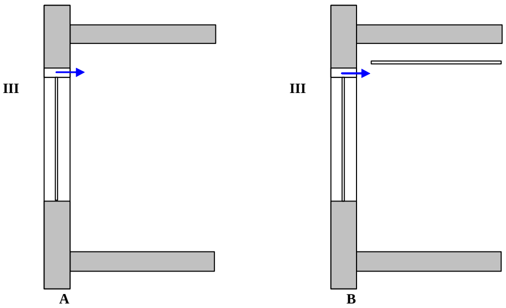


Figure 7. a) Air supply above a window, b) Combination with a false ceiling

An example of how architects try to integrate this knowledge in their design details is presented in figure 8:

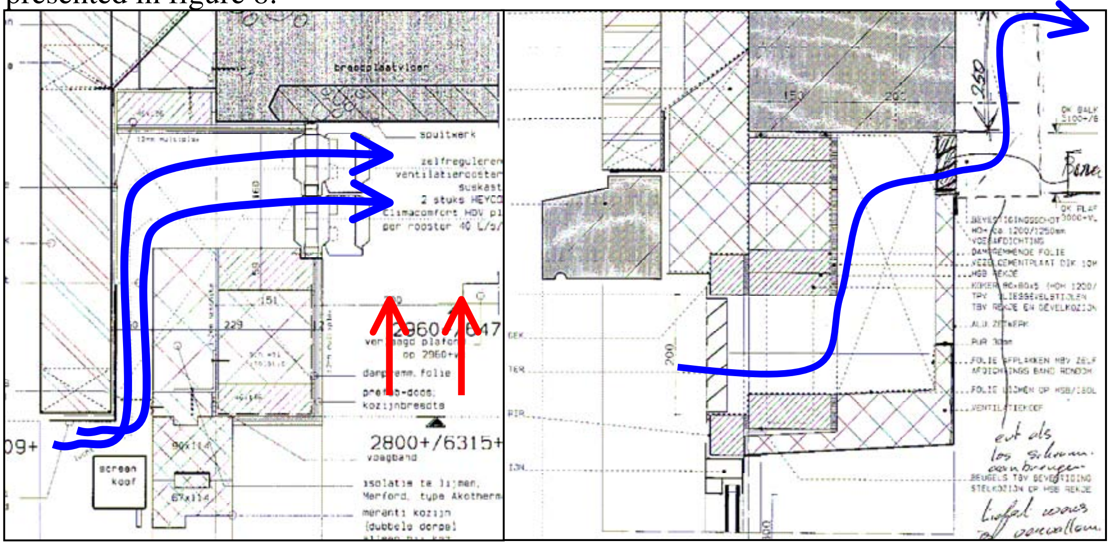


Figure 8: Two examples of air inlets details from different Dutch architects. The supplied air can be transported via the zone above a false ceiling (system III B, figure 7).

c. Airflow measurements and CFD-simulations

Results are design options of:

- A self regulating natural air exhausts system.
- False ceilings, related to thermal and acoustic parameters.
- Different heating systems: Concrete core or radiator heating and options of preheating supplied air.

An example of a self regulating natural air exhausts system is given below:

For a monumental building - the police academy in Apeldoorn - an air outlet system has been tested. In this case the air outlet of a self regulating vent is combined with a double window system (figure 9). Some advantages are:

- reduction of the amount of ducts in the building
- reduction of fan energy
- prevention of downdraught by increasing the temperature of the glass at the inner side
- reduction of infiltration of cold outdoor air

CFD-simulations show that a high position of the outlet is most favourable to prevent downdraught: the temperature of the window at the room side rises and the cold air between the windows cannot flow back into the room.

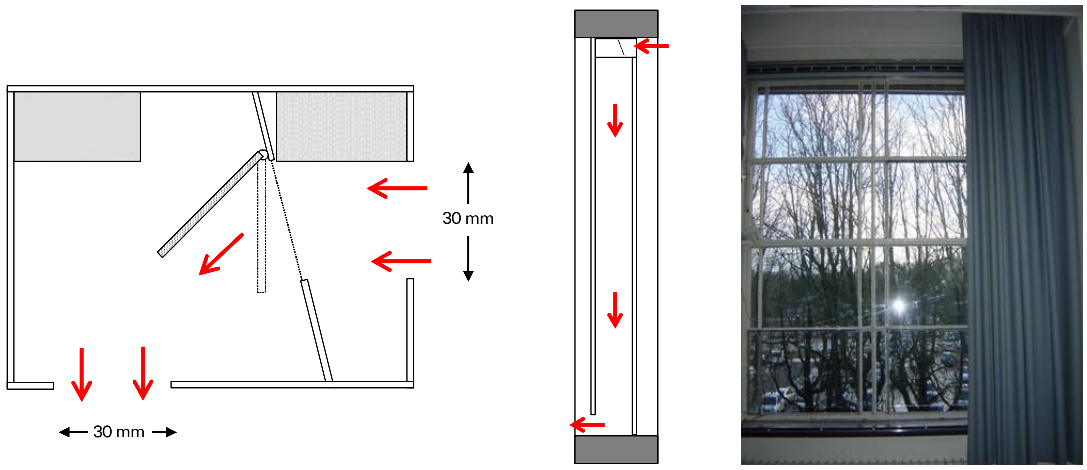


Figure 9: Presentation of the principle of a self-regulating air outlet (system 1, table 1) and a picture with the outlet at the upper side of the window.

The self regulating system prevents the wind to influence the airflow too much. Moreover, when the air supply fan is shut off, the heat loss due to ventilation is small (figure 10 and table 1).

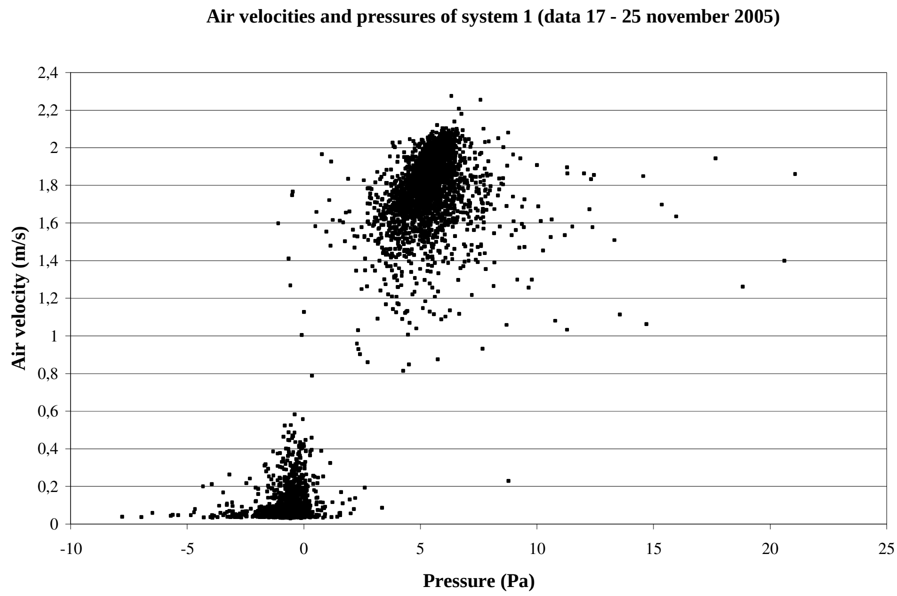


Figure 10: Measurement results of air outlet system 1

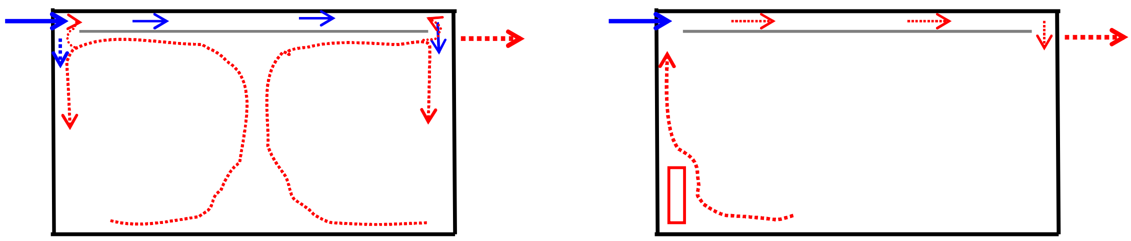
Table 1. Comparison of two natural air outlet-systems (1.72 m long, opening ca. 1.38 m long) with a repulse valve combined with a double window system. The measurements of system 1 and 2 are executed at the same time in a different window in the same classroom (Apeldoorn, 17-25 November 2005)

	System 1				System 2			
	Air velocity (m/s)	Pressure difference (Pa)	Flow (m^3/h)	Air flow efficiency (-) System	Air velocity (m/s)	Pressure difference (Pa)	Flow (m^3/h)	Air flow efficiency (-)
average	1,89	5,49	282	62 %	1,50	6,20	207	47 %
standard deviation	0,13	0,73	19		0,11	0,72	15	
maximum	2,28	21,07	340		1,86	27,91	256	
minimum	0,06	2,01	9		0,05	2,10	7	

Tab1e 1 shows a better efficiency of system 1 compared to system 2. The main reason is that the area of the smallest opening in system 1 is larger than of system 2 and the aerodynamic properties of system 1 are better than system 2. However, even the efficiency of system 1 can be improved avoiding sharp edges of the inlet, outlet and bends and creating a smooth channel without disturbances that can create eddies. In that case an air flow efficiency of more than 83% can be reached [5].

False ceilings, related to thermal and acoustic parameters:
To have the right reverberation time and good speech intelligibility it is necessary to create 25% absorption on the whole inner surface of a class room. For reasons of air quality the floors need a smooth surface and a large part of the walls and the ceiling should be available for acoustic materials. Acoustic elements can diffuse fresh air. They should not hinder the heat transfer and cleaning of these elements should be possible. At his moment there is a database of design options and a calculation model is developed to predict the acoustical properties of a classroom [6].

Floor or radiator heating and CFD-simulations:



Floor heating Radiator heating

Figure 11: Comparison of convection flows of natural air supply, combined with a false ceiling and different heating systems (derived from CFD-simulations).

Figure 11 shows that it is much easier for radiator heating than for floor heating to prevent downdraught from natural air supply and to promote the mixing of cold and warm air above a false ceiling. In case no radiator is available, heating of the ceiling - like concrete core heating - is necessary as well. However, at the opposite side of the façade there is always the risk of downdraught. This can be reduced by creating - for instance- a air permeable ceiling at the opposite side of the façade.

DISCUSSION

When natural air supply and natural exhaust-systems are compared both systems have different opportunities and can be energy effective. In table 2 some of the characteristics of both systems are compared:

Table 2: Comparison natural air supply and natural exhaust

	Air flow	Heat recovery	Heating and cooling	Fan energy
Natural air supply	Max. 3.5 l/sm^2	Heat pump on air exhaust	Heating and cooling with water	Very low
Natural exhaust	3.5 - 6.3 l/sm^2	Aquifer preheats and cools the supplied air	Heating with water and air, cooling with air	Low

This table shows that larger air flows can be supplied with a system of natural exhaust than with a system of natural air supply without creating a draught risk. This is possible by using the right air inlet-systems and using the corridor as an outlet as well.
By preheating natural supplied air, larger air flows might be possible, but this has not been tested up to now.

1. For architects and other designers it is very important to have a compact database of images and knowledge to show comfort-problems and optional solutions.
2. An integral approach of all the important "indoor" physical parameters is necessary even for common design problems.

ACKNOWLEDGEMENT

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Comparison between thermal comfort predictive models and subjective responses in Italian university classrooms

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SUMMARY

This work is focused on the evaluation of indoor thermal quality and shows some results of a wider field study in university classrooms. The field study was conducted through physical observations and questionnaires, performed at the same time during the regular lesson time, in a period just before the start of the heating season. The predictions of dissatisfied occupants, based both on Fanger's heat balance model and on an adaptive approach, were compared to each other. The subjective survey investigated the thermal acceptability, the thermal preference and the thermal sensation, asking students to assess their comfort on subjective scales. The calculated predicted votes were compared to the observed subjective responses. Moreover, the subjective mean votes were compared to the thermal environment perceptions in terms of acceptability and preference. The obtained results give a contribution to the enrichment of knowledge about thermal subjective responses in classrooms.

INTRODUCTION

Recently, CEN is working at the assessment of indoor environmental quality criteria, with the formulation of the prEN15251 European Standard Project [1]. The prEN15251 is strictly related to the European Directive 2002/91/EC [2], about energy efficiency of buildings, and deals with energy consumption problems; it proposes a method for the classification and certification of indoor environmental quality (IEQ), under the point of view of thermal comfort, visual comfort, acoustic comfort, indoor air quality and global comfort [3].
The IEQ affects not only health and comfort, but also the occupants' productivity, so it strongly influences the general quality of working and educational environments, having repercussions on production costs and social costs. In particular, schools are a category of buildings in which a high level of environmental quality may considerably improve occupants' attention, concentration, learning, hearing and performances [4].
This study is focused on thermal comfort and aims at achieving a better knowledge about the subjective perception in naturally ventilated environments, in which the occupants have only some opportunities of behavioural adjustment. A particular, but significant, case is here analysed: naturally ventilated university classrooms. The comparison between the subjective votes and the predicted votes, deriving from the objective monitoring of thermal parameters, allows the test in field of different existing criteria, based both on a deterministic approach and on an adaptive approach.
This study is part of a wider research started by the Building Physics and Indoor Environment Engineering Research Group (see http://www.polito.it/ftarch) of the Department of Energy (DENER) of the Politecnico di Torino, which was focused on environmental comfort in Italian school buildings [5,6]. This paper focuses on the results from the thermal comfort field

investigations in university classrooms. The field campaigns were performed during the lesson periods.
At present, two different approaches to the definition of thermal comfort coexist, each one with its potentialities and limits: the former is deterministic, the latter is adaptive.
Fanger's model [7] based on steady state heat transfer theory, has a deterministic approach and provides the basis of the main thermal comfort standards [8,9], for mechanically controlled environments.
Adaptive comfort models derive from field studies, having the purpose of analysing the real acceptability of thermal environment, which strongly depends on the context, on the behaviour of occupants and on their expectations. The analysis of "real-world" settings reveals that thermal preferences depend on the way people interact with their environment, modifying their own behaviour and adapting their expectations, to match the thermal environment [10].
Moreover, the recent studies on adaptive comfort approach qualify the thermal comfort not only by asking a judgment about the thermal sensation, but also by investigating the acceptability and preference of the indoor thermal condition with respect to conditions corresponding to thermal neutrality [11,12]. This tendency of preferring certain thermal environments was already argued by McIntryre [13]. In his studies, he found out that people of warm climates may prefer what they call a ''slightly cool'' environment and, on the contrary, people of cold climates may prefer what they call a ''slightly warm'' environment.
Furthermore recent field studies in classrooms confirm that people in naturally ventilated indoor environment are comfortable within a range of microclimate values that is larger than in a fully conditioned indoor environment [14].
Among the adaptive comfort diagrams, this study referred to the one developed in a recent EU-funded research project coordinated by the Oxford Brookes University (figure 1) [15]. This diagram is adopted into Standard prEN15251/2006 [1].

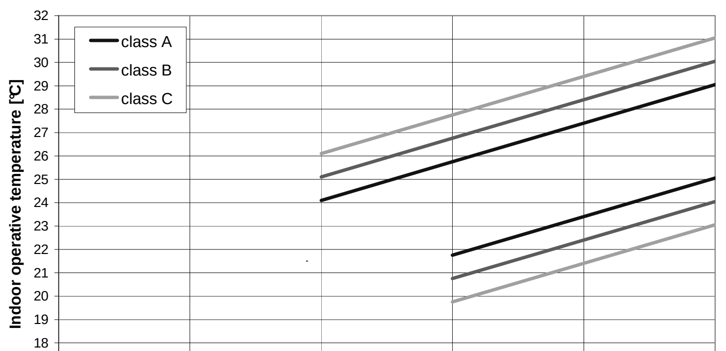

Figure 1. Adaptive thermal comfort diagram for the design of naturally ventilated environments, adopted into Standard prEN15251/2006

METHODS

Object of the study

In this study, a methodology based on both objective and subjective surveys for in field evaluation of thermal comfort was applied. The approach consists of administrating a

questionnaire to a group of occupants while the investigator records certain microclimatic parameters [6,11,14].
Such an investigation methodology is very important in classrooms, that are an example of indoor environment in which the adaptive opportunities are quite limited during the lessons period, but they are free during the hourly lesson breaks. In fact, students have to spend lots of time in listening and understanding lessons, remaining sitting at their desk. Moreover, the freedom of students in modifying and adjusting their activity level according to the thermal environment is, to a certain extent, limited during the lesson time, as well as the possibility to change the functioning parameters of the HVAC systems or to open/close the windows. But the same actions are free during the lessons breaks. The adaptive actions of the students to modify the microclimate parameters may include adding or removing layer of clothing, opening or closing windows, moving sun shading devices, etc. [10].
A previous study examined a representative sample of typical high school and university Italian classrooms; the study was performed during the heating period, that in Turin ranges from 15th October to 15th April; in particular, the investigations were carried out from the end of January to April 2002. The results are shown in [6].
The object of this study, an extension of the previous study, is a university classroom of the Politecnico di Torino, medium-sized and parallelepiped-shaped. The classroom is located at ‘‘Via Boggio’’ campus, a new building in a residential area at the border of Turin downtown, next to the headquarter of the Politecnico di Torino. The classroom has a floor area of 142 m^2, a height of 3,4 m and a volume of 483 m^3; the ratio of glassed area/floor area is W/F = 0,15; the exposition is E–SE. The classroom was examined during the lesson time, in two periods, the first in September and the second in October 2006, outside the heating season, with outdoor warm or mild climate conditions; the direct solar radiation into the classroom was important during the first survey, while the sky was cloudy during the second. Heating through radiators and mechanical ventilation were off during the field campaigns. The classroom contained 103 students at the time of the first survey and 108 in the second.
Two different comfort criteria, Fanger’s PMV [7], and the adaptive thermal comfort diagram (figure 1) [15], were applied to a naturally ventilated university classroom and the results were compared with the subjective votes of the students. Furthermore the votes of thermal preference were compared to the votes of thermal sensation.

Subjective approach

The subjective approach was basically aimed at finding out the judgement about the perception of the thermal environment in terms of acceptability and preference of colder or warmer environments. Moreover, a vote on the Fanger’s 7-points thermal sensation scale was asked to students.
Questionnaires were used to investigate the thermal perception. They were delivered and filled by the students while the measurements (see Section 3.3) were going on. The answers to the questions concerned the instantaneous assessment of microclimatic conditions. Students were uniformly distributed in their own classroom during their regular lecture hours.
The questionnaire was specifically set-up for the assessment of thermal quality in classrooms. It is a section of a more complete questionnaire, concerning the general environmental quality in classrooms, developed by an équipe of indoor environment engineers, mathematicians and physiologists. Its final shape belongs to the results of a number of tests aimed at verifying the reliability of the way in which the questions are proposed and the answers are done.
With regard to thermal comfort, among the general information, it was asked to the students to mark what they were wearing by means of a clothing check-list, in order to find out the actual clothing level (calculated using ISO 7730 [16]). This information was used in the evaluation of the PMV comfort index.

The questions in the section of thermal comfort concerned how the thermal environment was felt. In particular, students gave a judgement about its acceptability and preference, answering the following questions [6,11,14]:

- at this moment, do you consider the thermal environment acceptable or not?
- at this moment, would you prefer to feel warmer, cooler or no change?

Moreover, a judgement on the typical seven points thermal sensation scale (Fanger 7-points scale, ranging from -3 to +3, corresponding to very cold and very hot and 0 being the thermal neutral condition) was asked [7].

Objective approach

The indoor thermal environment was analysed by means of field measurement campaigns. The measured thermal parameters were:

- air temperature,
- plan radiant temperatures,
- air relative humidity,
- mean air velocity and standard deviation of air velocity.

The Indoor Climatic Analyser, type 1213 by Brüel&Kjær was used to perform the measurements of all the thermal parameters and 6 microdataloggers type 175-T1 by Testo, to measure air temperature. The measurements took at least 2 hours and were done only during the lesson time.

The parameters were measured in continuous at a height of 1.1m above the floor, according to the standard ISO 7726:1998 [16] for seated persons, for the whole classroom.

On the basis of ISO 7730 [9], two forms of local thermal discomfort were verified: from the mean air velocity and standard deviation of air velocity, the draft risk was evaluated and the local discomfort from radiant asymmetry was evaluated.

As far as the personal parameters are concerned, the metabolic rate was fixed at 1.2 met (sedentary activity) and the actual people clothing were obtained from the questionnaires.

The recorded data were elaborated in order to evaluate the thermal comfort Fanger's indices, PMV and PPD, according to ISO 7730 [9]; the clothing levels and the values of the thermal parameters were quite homogeneous, so the PMV was calculated from the medium values of the parameters.

In order to apply the adaptive diagram of figure 1, the "outdoor running mean temperature" was calculated from the outdoor daily mean temperatures of the days preceding the examined one (day n), with this formula [1,15]:

$$t_{ORM\,(n)} = (1-\alpha)\,(\,t_{OMD\,(n-1)} + \alpha\, t_{OMD\,(n-2)} + \alpha^2\, t_{OMD\,(n-3)} + \ldots), \qquad (1)$$

where $t_{ORM\,(n)}$ is the running mean temperature in the day n, $t_{ODM\,(n)}$ is the outdoor daily mean temperature in the day n and α is a constant between 0 and 1, defining the speed at which the running mean temperature responds to the outdoor temperature (a value of 0.8 implies that the characteristic time subjects take to fully adjust to a change in the outdoor temperature is around 5 days and corresponds to the highest correlation with comfort sensation).

The outdoor daily mean temperatures of the days preceding the examined one were obtained from hourly data, measured by the Meteorological Station of the Politecnico di Torino; the resulting values of the "outdoor running mean temperature" for the two examined days were respectively 19,8 °C and 20,1°C.

So the indoor operative temperature ranges were obtained from the adaptive thermal comfort diagram of figure 1, in function of the values of outdoor running mean temperatures previously found. The diagram of figure 1 shows three tolerance ranges for the indoor operative temperature, corresponding to three different expected percentages of satisfied people (90%, 80% and 65%).

RESULTS

As explained in the previous paragraph, the whole classroom was qualified "objectively", that is on the basis of measurements, observations and calculations, both through a deterministic and an adaptive approach, obtaining the following results for the global thermal comfort:

- a value of PMV and PPD for the whole classroom, according to the deterministic approach;
- an expected value of dissatisfied occupants, deriving from the comparison between the range of values assumed by the indoor operative temperature and the tolerance ranges obtained from the adaptive diagram for the examined day.

First of all, these two results were compared to each other, then these results were compared to the subjective votes deriving from the questionnaires.
As far as local thermal discomfort is concerned, two forms of local thermal discomfort (draft risk and radiant asymmetry) were evaluated, as explained in the previous paragraph. Both these evaluations didn't show any critical situation under the point of view of local discomfort, in both the surveys.
For both the examined days, the obtained value of PPD was < 10% (9% and 8% respectively). Considering that no particular forms of local thermal discomfort existed, in both the examined days, the expected percentage of dissatisfied was less than 10%.
Comparing the indoor operative temperatures to the tolerance ranges, with the adaptive approach, the expected percentage of satisfied people was at least 90% (indicated as PPD_adapt < 10%); in this case the use of the diagram would predict the effect of both global and local thermal comfort.
So the two "objective" approaches lead approximately to the same results, in predicting the percentage of dissatisfied. The coincidence of the two approaches might depend on the fact that, in the examined period, the outdoor climate is not extreme.
The expected percentage of dissatisfied (< 10%) was compared to the percentage of dissatisfied from the questionnaires, calculated as follows.
The dissatisfied people were first evaluated through the direct votes "non acceptable" expressed by the students for the thermal environment: this percentage is indicated as PD_direct in table 1.
A second way of evaluating the dissatisfied people was considering the ones voting (-3;-2) and (+2;+3) on the seven points thermal sensation scale, following the same approach used by Fanger in its experiments; this percentage was slightly corrected adding to it the percentage of students voting (-1,0,+1) but expressing discomfort for draft (indicated as PD in table 1).
Then, with a third approach, the dissatisfied people were considered the ones voting (-3;-2) and (+3) on the seven points thermal sensation scale; also in this case, this percentage was slightly corrected adding to it the percentage of students voting (-1,0,+1,+2) but expressing discomfort for draft (indicated as PD* in table 1).

Table 1. Results of the two field surveys performed in the same classroom

Survey	PPD	PPD_adapt	PD_direct	PD	PD*
September	9%	< 10%	24%	20%	6%
October	8%	< 10%	29%	24%	9%

The results presented in Table 1 show that both PD_direct and PD, resulting from questionnaires are always sensibly higher than the calculated PPD and PPD_dapt.
In particular, vote +2 (hot) is considered unacceptable by the Fanger approach. On the contrary, in the preformed study, environments voted as +2 seem to be partially accepted, as it is pointed out through the analysis of the subjective judges about ‘‘acceptability’’ and ‘‘preference’’ of a thermal environment. In fact, considering as dissatisfied people voting (-3;-2) and (+3) and as satisfied people voting (+2), the PD* agrees well with the measured PPD. This issue was also argued by Mayer [12]. A similar result emerged from the previous study [6], which was conducted only during the heating season, but it is important to note that in that period this trend was more evident, in fact environments voted as +2 seemed to be more accepted and also preferred.
The judgments on the seven points thermal sensation scale were then correlated to acceptability and preference, as shown in Figs. 2, 3 and 4.

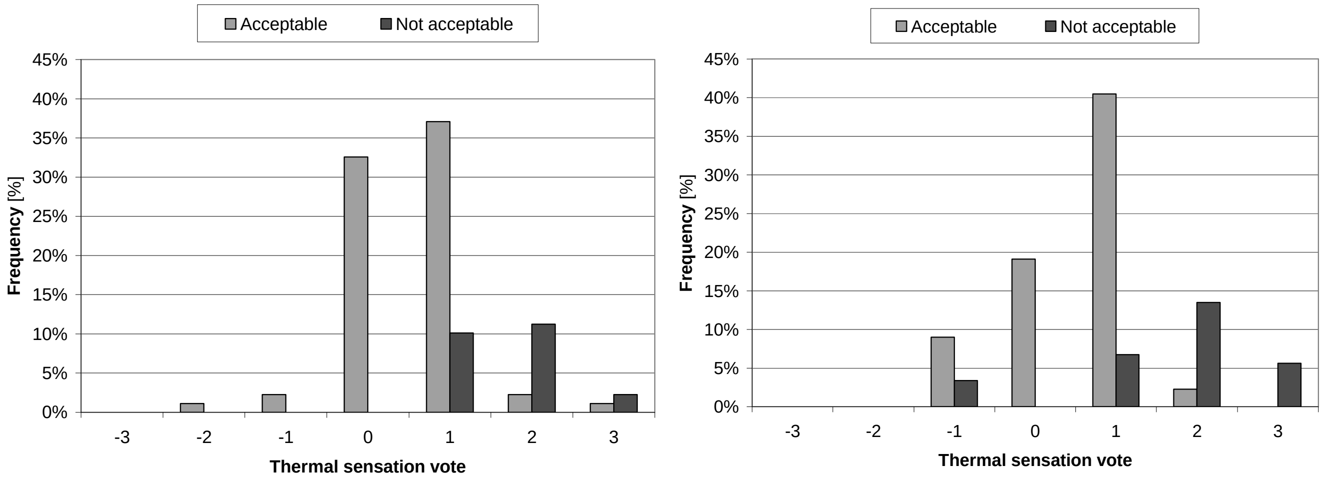

Figure 2. Frequency distribution of thermal acceptability votes versus thermal sensation votes

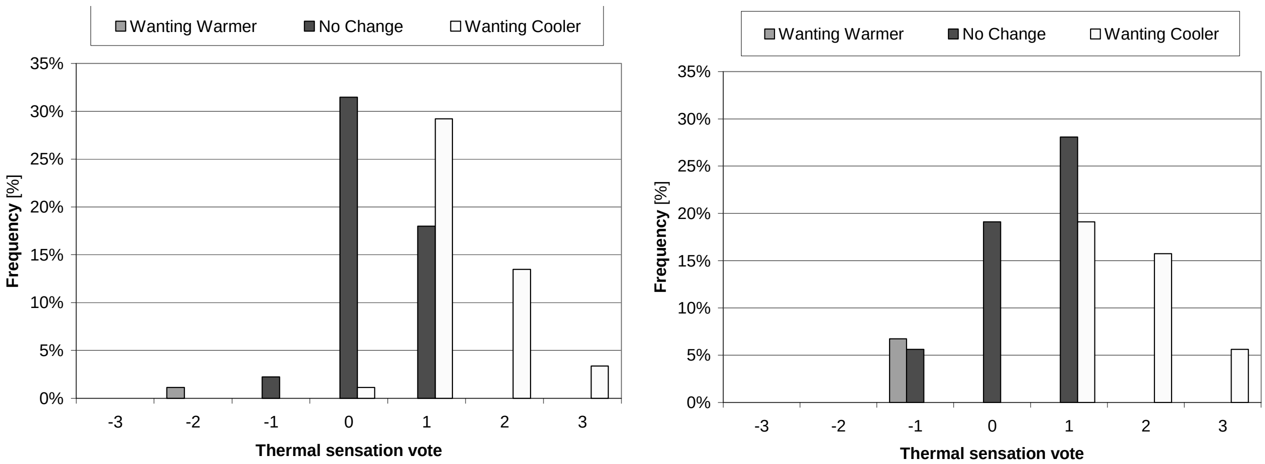

Figure 3. Frequency distribution of thermal preference votes versus thermal sensation votes

In particular, in Fig. 2 the frequency distribution of the acceptability is presented versus the subjective vote about the thermal environment.
Both the results of the surveys show that the distribution of ‘‘acceptable’’ is shifted toward the positive side of the vote scale; in particular, most of the people voting +1 considered ‘‘acceptable’’ the thermal environment. It is important to note that in the previous winter

study a similar trend was found for the examined classrooms, but with a more accentuated acceptability of warmer environments.
These statements are also verified by the diagrams in Fig. 3, plotting the frequency distribution of the preference versus the subjective vote about the thermal environment.
A very high percentage of votes ‘‘no change’’ corresponds to the vote +1, in particular for the second survey. The same phenomenon was very remarkable in the previous winter surveys.
Plotting the cumulate frequency distribution of ‘‘wanting warmer’’ and ‘‘wanting colder’’ versus the subjective judgment on thermal environment (Fig. 4), show the location of the point corresponding to the minimum number of dissatisfied. This point is at the intersection between the two cumulate curves. The figures show that such minimum does not coincide with the thermal neutral condition (0 in the subjective vote) but is slightly shifted toward the positive values of the vote scale.

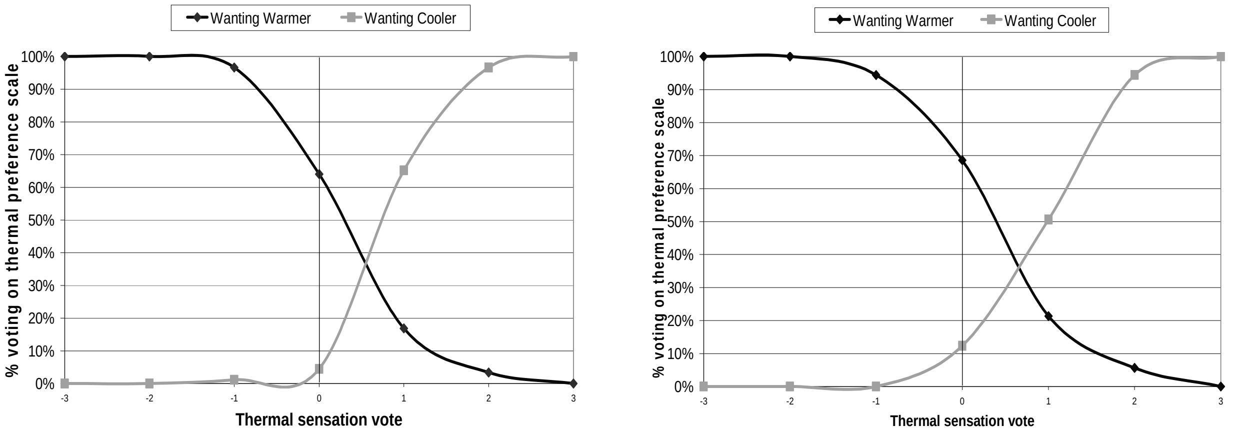

Figure 4.

DISCUSSION

In this work, a in field investigation methodology on thermal comfort was applied. The environmental parameters influencing thermal comfort were measured while, at the same time, the subjective judgements of the people about the thermal environment were expressed. Significant tendency and correlation were found out. The investigations were performed in the same university classroom in two different days, in September and October, before the beginning of the heating period.
This study allowed a comparison between different predictive approaches for thermal comfort and a comparison between the predictions and the observed subjective responses. Furthermore it added new findings to previous researches conducted in Italian university and high school classrooms, during the heating season. The results of this study confirmed the trend outlined by the previous study, but with a minor intensity, suggesting a correlation between the thermal acceptability and preference and the outdoor temperature.
The following results were found:

- it was found out a bad agreement between the calculated PPD and the percentage of dissatisfied from the questionnaires when people voting (-3;-2) and (+2;+3) on the seven points thermal sensation scale are considered all dissatisfied;
- it was found out a good agreement between the calculated PPD and the percentage of dissatisfied from the questionnaires when people voting (+2) on the seven points thermal sensation scale are considered satisfied.
- thermal environments which are judged neutral or slightly warm are accepted;
- thermal environments which are judged slightly warm might be preferred;

- in thermal environments which are judged as slightly cold, ‘‘wanting warmer’’ people are a high percentage and might prevail;
- in thermal environments which are judged as neutral, ‘‘no change’’ people prevail and there are no votes of ‘‘wanting warmer’’;
- in thermal environments which are judged as slightly warm, ‘‘no change’’ people are a high percentage and might prevail.

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A comparative analysis of the indoor air quality and thermal comfort in schools with natural, hybrid and mechanical ventilation strategies

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SUMMARY

Within the UK, the importance of providing adequate ventilation in schools has been recognised in a recently adopted document (Building Bulletin 101), which defines the set of performance criteria in relation to ventilation rates and indoor air quality in new school buildings. This paper describes a series of field measurements that investigated the ventilation rates and indoor air quality in three new secondary schools in England with respect to these new criteria. The study also analysed the overall performance of the integrated heating and ventilation systems with regards to comfort. All the schools satisfied the recommended ventilation performance standards during the week that the measurements were undertaken. However, this apparently reassuring message can be misleading unless one takes into account both the original design assumptions and then the actual occupancy of the classrooms and occupant behaviour in general. With regards to comfort, for many rooms the schools did not meet the relevant recommended levels.

INTRODUCTION

The UK Government has committed to a massive programme of rebuilding and refurbishing schools in England and Wales in the next 10 to 15 years [1]. To underpin this programme, entitled ‘Building Schools for the Future’, the Department for Education and Skills has published design guidance - Building Bulletin 101 ‘Ventilation in School Buildings’ [2]. This performance standard document is cited as a means of compliance with the new Building Regulations Part F (Ventilation) in England and Wales [3]. In this document CO_2 concentration has been chosen as the key performance indicator for the assessment of indoor air quality and ventilation performance in schools. The recommended ventilation performance standard can be summarised as follows:

1. the average concentration of CO_2 should not exceed 1500 ppm
2. the maximum concentration of CO_2 should not exceed 5000 ppm during the teaching day
3. at any occupied time the occupants should be able to lower the concentration of CO_2 to 1000 ppm
4. purpose provided ventilation in naturally ventilated buildings should provide external air supply to all teaching and learning spaces with (a) a minimum of 3 l/s per person, (b) a minimum daily average of 5 l/s per person and (c) a capability of achieving a minimum of 8 l/s per person at any time

5. purpose provided ventilation in mechanically ventilated buildings should provide external air supply to all teaching and learning spaces with a minimum of 5 l/s per person at all time, and (b) a capability of achieving a minimum of 8 l/s per person at any time

In addition to the requirements stated in Building Bulletin 101, the authors have investigated if the comfort of occupants was compromised by the ventilation strategy. A previous study has highlighted thermal comfort as an important aspect of providing winter time ventilation [4]. Taking these requirements into account, this paper focuses on a comparative analysis of indoor air quality and thermal comfort in three recently built schools in England with different ventilation strategies: natural, hybrid and mechanical.

METHODS

Measurements were carried out in three schools in England during the heating season 2005-2006. All schools were built in compliance with Building Bulletin 93 [5], which defines noise and acoustic criteria in relation to school design. The monitoring was carried out in two selected classrooms in each school over a period of five working days. Levels of CO_2 were monitored at five-minute intervals throughout the occupied day close to the occupied zone at seated head height to indicate the overall indoor air quality and provide a means of inferring the ventilation rate based on the number of occupants. Two Gascard II infra-red gas monitors ($MYCO_2$) (accuracy: 2% of the range - 0-5000 ppm) coupled with HOBO dataloggers were used for the indoor measurements. In addition, outdoor CO_2 was measured using a Telaire 7001 infra-red gas monitor (accuracy: 50 ppm or 5% of the reading, whichever is greater). Ventilation rates were also estimated over suitable intervals using Equation 1, a form of 'continuity equation' [6] [7] [8]:

$$C_{(t)} = C_{ex} + \frac{G}{Q} + \left(C_{in} - C_{ex} - \frac{G}{Q} \right) e^{-\frac{Q}{V}t} , \qquad (1)$$

where: $C_{(t)}$ - internal concentration of carbon dioxide at time t (ppm), C_{ex} - external concentration of carbon dioxide (ppm), G - generation rate of carbon dioxide in the space (cm^3/s), Q - internal-external exchange rate (m^3/s), C_{in} - initial concentration of carbon dioxide (ppm), V - room volume (m^3), and t - time (s).

In addition to CO_2 levels, the following thermal comfort parameters were measured during the occupied periods in each of the selected classrooms:

1. dry bulb temperature, measured via: a screened platinum resistance sensor to eliminate any thermal radiation effects, an air velocity compensation sensor and a relative humidity compensation sensor.
2. relative humidity was measured with a VAISALA capacitive sensor
3. globe temperature was measured with a platinum resistance sensor within a 30mm black sphere
4. air velocity was measured with a DANTEC heated thermocouple sensors

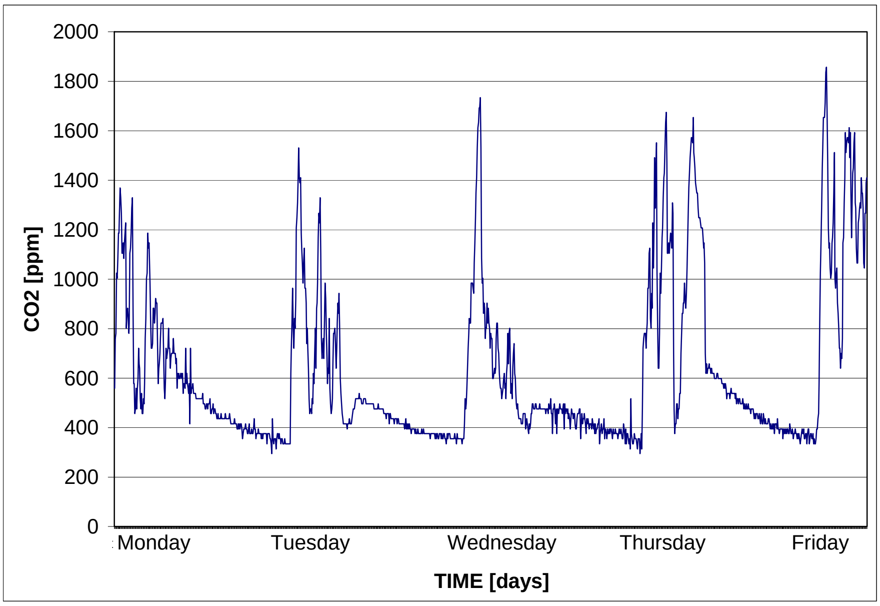
Measurements made every second were averaged over 2 minute intervals. The thermal comfort parameters were measured at two locations simultaneously, one being fixed at the normal work position of a pupil close to an openable window, while the second thermal comfort analyser was moved at different locations across the rooms. The procedure laid out in ISO Standard 7730-1995 was used to determine the PPD (percentage of people dissatisfied) and the DDR (draught dissatisfied rating) indices and specifications of the conditions for thermal comfort. A brief description of the ventilation strategy and findings for each school is given below.

RESULTS

School 1 - Natural Ventilation strategy: This school is located at an exposed rural site and represents a basic single sided naturally ventilated design with openable windows into deep classrooms. Each room had three top hung windows but each had a maximum opening angle of 30°. Some thought had been given to providing cross ventilation by providing a small grill into the suspended ceiling and a duct that led to the atrium space from which the classrooms were entered. No fan was found. A smoke test carried out suggested that the ducted extract contributed insignificantly to the ventilation strategy in the room. Heating was provided by low temperature hot water radiators located under the openable windows. Both classrooms (NV1 and NV2) have an identical ventilation strategy, but different use. Whilst room NV1 was used as a seminar room, room NV2 was used as a 'typical' whiteboard classroom. A number of small intervention studies were carried out in each classroom (windows opened/closed, etc.) to test the capabilities of the design to adequately ventilate the room. Note that the number of students during the 'observed' occupancy in the rooms was very low - usually 50% of the 'as designed' number of occupants. Therefore, two ventilation rates were calculated and reported. For example: in the case of the naturally ventilated room NV1 with three top hung windows opened the calculated ventilation rate for the 'observed' occupancy of 10 occupants was 11.7 l/sp. However, the ventilation rate in the case of the 'as designed' occupancy of 30 reduces to 3.9 l/sp.

Table 1: School 1 – summary of IAQ measurements

A typical CO_2 levels in a room with natural ventilation strategy (NV1)



CO_2 levels [ppm]*		
Classroom	NV1	NV2
average	960	1054
STD	331	397
maximum	1857	1725

Comments:
*note that the occupants kept the door to the atrium open most of the time
** calculated for 10 occupants (calculated for 30 occupants)
*** calculated for 30 occupants

Intervention	All windows opened (l/sp)	2/3 of all windows opened (l/sp)	1/3 of all windows opened (l/sp)	All windows closed (l/sp)	Cross ventilation (l/sp)
NV1**	11.7 (3.9)	n/a	6.0 (2.0)	2.8 (0.9)	20 (6.5)
NV2***	3.9	3.4	1.7	n/a	n/a

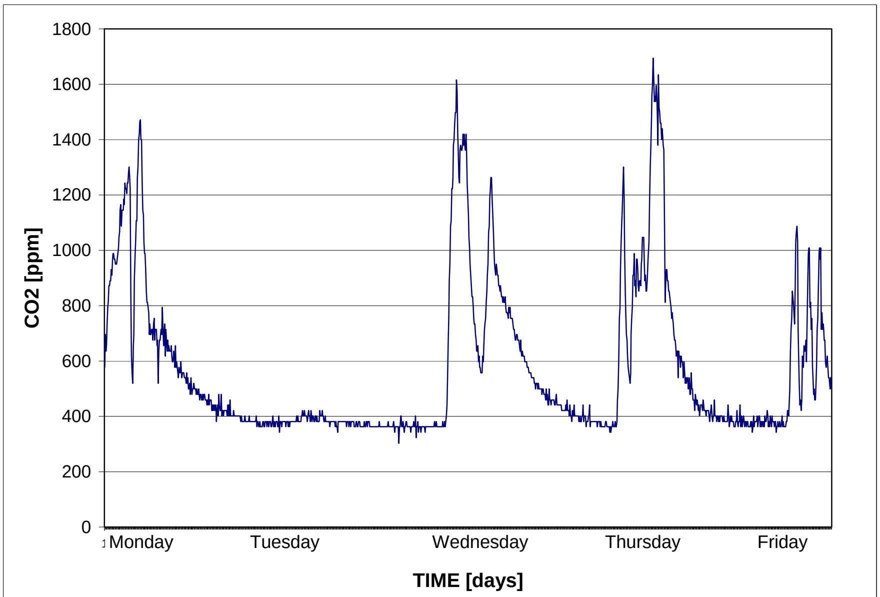
Table 2. School 1 – summary of thermal comfort measurements

Room	TC analyser	PPD [%]	DDR [%]	T [°C]
NV1	moving	11.4	15.8	22.4
NV1	stationary	16.9	30.4	25.8
NV2	stationary	12.6	13.4	21.5
NV2	moving	5.5	3	21.4

School 2 - Mixed Mode and Mechanical Ventilation strategy: This school has a combination of mixed mode ventilation and full mechanical ventilation systems. General ventilation is provided to the classrooms mostly by means of a packaged air handling unit (AHU) system. 100% fresh air is fed into the air handling unit where it is conditioned (tempered) - the system being controlled by a BMS. The tempered air from the AHU is conveyed to each classroom through externally buried concrete pipes (any mould growth in the pipes was not an issue that was investigated). Heating is by a combination of underfloor and trench heating. Suspended ceilings were not generally fitted so that the exposed thermal mass could provide some passive cooling. In the mixed mode classroom (HMV1) the designed ventilation strategy relies on both mechanical (variable speed fan) and natural ventilation (automatic windows). In the mechanically ventilated room the designed ventilation strategy relies on variable speed fan only.

Table 3. School 2 – summary of IAQ measurements

A typical CO_2 levels in a room with a mixed mode vent. strategy (HMV1)



CO_2 levels [ppm]*,**

Classroom	HMV1	HMV2
average	853	975 (1100)
STD	268	309 (320)
maximum	1472	1615 (1615)

Comments:
* attendance was low, usually 50% of the ‘as designed’ number of occupants;
** values in the brackets have been obtained for occupant numbers close to the ‘as designed’ number of occupants

Intervention	MV ON all windows opened (l/sp)	MV ON automatic windows opened (l/sp)	MV OFF all windows opened (l/sp)	MV OFF automatic windows opened (l/sp)
HMV1	10.6 (5.3)	6.7 (3.4)	n/a	3.0 (1.5)
HMV2	10.5	n/a	7.6	n/a

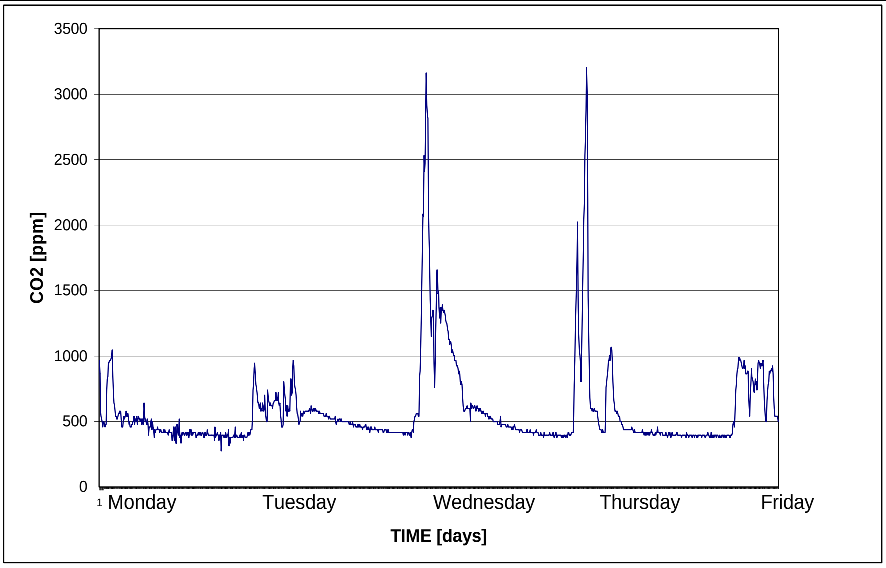
Table 4. School 2 – summary of thermal comfort measurements

Room	TC analyser	PPD [%]	DDR [%]	T [°C]
HMV1	moving	12.6	13.4	21.5
HMV1	stationary	5.5	3	21.4
HMV2	moving	17.4	9.8	20.3
HMV2	stationary	9.1	4.5	20.0

School 3 - Mechanical Ventilation strategy: The school is serviced by mechanical ventilation with under floor heating and manual windows. The teaching rooms (MV1 and MV2 are considered here) have full mechanical ventilation from a ceiling based supply and extract. The fresh air is tempered and conveyed to the room via a simple duct system mounted in a void above the suspended ceiling. There are two air inlets and only one extract within the classroom. The extract, which leads to the large void space above the suspended ceiling is not separately ducted but a small transfer hole connects this plenum void to the corridor. The corridors have additional extract fans located on the roof. This would suggest that mechanical ventilation provides positive pressure within the room. In addition to the mechanical ventilation system, the ventilation strategy in the room is underpinned by two smaller manually operated windows.

Table 5. School 3 – summary of IAQ measurements

A typical CO_2 levels in a room with a mechanical vent. strategy (MV1)



CO_2 levels [ppm]

Classroom	MV1	MV2
average	789	733
STD	171	142
maximum	1047*	880

Comments:
* note that during this period the number of occupants exceeded the maximum designed number of occupants (30+2)
** manually operated windows were used occasionally

Intervention	MV ON all windows closed (l/sp)	MV OFF all windows opened (l/sp)	MV OFF all windows closed (l/sp)
MV1	8.4	n/a	0.5
MV2	9.4	3.8	n/a

Table 6. School 3 – summary of thermal comfort measurements

Room	TC analyser	PPD [%]	DDR [%]	T [°C]
MV1	stationary	5.4	27.1	23.6
MV1	moving	5.9	18.5	24.1
MV2	stationary	5.8	9.9	22.2
MV2	moving	5.9	18.5	24.1

DISCUSSION

A. CO_2 levels

School 1: A typical diagram of CO_2 levels based on the recorded five minutes values is shown in Table 1 for the naturally ventilated room NV1. Generally, on each day, the levels of CO_2 increased from the start of the day reaching a peak at the end of the morning session and decreasing during the lunch time when the classrooms are generally unoccupied. The CO_2 levels start increasing again after the lunch break reaching the afternoon peak at the end of the last period. This is due to the following: a) the breaks between two classes are short (usually 5 minutes) not allowing the CO_2 concentration to equilibrate to the external level, b) in some cases pupils are allowed to stay in classrooms during the breaks contributing to an even more rapid build up of CO_2 levels, c) the lack of an effective ventilation strategy. Note that the standard deviations of CO_2 levels (STD) for the natural ventilation strategy applied here are very high (Table 1). In the case of these naturally ventilated rooms, the average CO_2 levels exceeded 1,000 ppm in room NV2 only. The lower values in room NV1 were partially due to low occupancy levels in the room. Despite low occupancy, the maximum levels recorded were still high.

School 2: A typical diagram of CO_2 levels for the mechanically ventilated rooms is shown in Table 5. The value of 1000 ppm was exceeded only once for a very short period of time. Note that during that short period, the number of the occupants in the room exceeded the designed number of occupants by 2. As expected, fluctuations of CO_2 levels are less significant than for School 1. The fluctuations in these rooms with constant flow rate fans were related either to occasional use of manual windows or to change in number of pupils in attendance. It should be noted that the use of fans with constant flow rate may result in over-ventilation and unnecessary energy consumption in schools with low attendance.

School 3: In the mixed mode classroom HMV1 the designed ventilation strategy relies on both mechanical and natural ventilation (automatic windows). During programmed occupancy periods the windows are set to remain open until the measured room space temperature falls below the set point by more than 2°C. However, due to security reasons (the classroom was located on the ground floor in a less secure area of city) the automatic control was overridden and the windows were kept shut. As a consequence, the role of the mechanical ventilation supply has shifted from a supplementary one to being the main ventilation provider. In order to test the ventilation strategy as it was supposed to be operated, the research team also opened the automatic windows manually (Table 3). Note that the standard deviation in this case of the room HMV1 was lower than in the case of naturally ventilated room. In this room the number of students during the ‘observed’ occupancy was very low, usually 50% of the ‘as designed’ number of occupants. Prior to this study, the volume of the flow entering the room HMV2 (fully mechanically ventilated) through the heater battery was too high causing discomfort to the students sitting near the trench. By reducing the air flow through the battery the problem of discomfort was addressed, but as a consequence of the reduced air flow the operational performance of the mechanical ventilation system was degraded. In this case it was not possible to change the furniture layout without incurring excessive costs (i.e. the furniture was fixed).

B. Estimated ventilation rates

The estimated ventilation rates have shown that the naturally ventilated classrooms were the least well ventilated in the normal style of usage but did not exceed any threshold values because of low occupancy levels under those conditions (Table 1, 3 and 5). The mechanical and mixed mode schools could exceed 8l/s per person but the greatest ventilation rate observed was in the naturally ventilated classroom when the door was opened to provide cross ventilation to the atrium. This mode was frequently used by the school when there were higher levels of occupancy in the classrooms.

C. Temperatures and comfort

With regard to the internal temperatures, CIBSE Guide A1 [9] suggests design criteria for educational buildings. For teaching spaces the specified winter temperature is 19-21°C. The average temperatures found in the schools varied depending on the school and room with NV1 and NV2 being the warmest between 24°C and 26°C and the rooms HMV1 and HMV2 being the coolest between 20°C and 22°C. For many rooms the schools did not meet CIBSE recommended levels for winter conditions and sometimes barely falling within the summer upper limit, indicating that there could be some discomfort among students due to the thermal environment. Note that the average external temperatures were app. 5°C with maximum temperatures exceeding 10°C for a few hours only.

Dissatisfaction due to air movement does not have a simple relationship with air speed; the draught index takes into account fluctuations in local air speeds and local temperatures in order to determine the percentage of people dissatisfied due to draughts. Note that the draught risk exceeded the generally accepted level of 15% a number of times. It should be noted that in mechanically ventilated rooms (Table 4) problems were experienced with draughts from the ventilation systems. In the rooms MV1 and MV2 the incoming ventilation air was 'dumped' from the air inlets causing discomfort to occupants. However, the highest DDR was related to a naturally ventilated room NV1 when all three top hung windows were fully opened (Table 1).

D. General

In this study all the schools satisfied the recommended performance ventilation standards leading to the conclusion that the implemented ventilation strategy was providing adequate ventilation. However, this is misleading unless one takes into account three important factors: a) the occupancy schedule for classrooms (i.e. the classrooms were not fully utilised during the 'normal' occupied hours, preventing CO_2 building up during the day), b) the occupancy level during classes (i.e. number of students attending classes) and c) occupant behaviour. As shown previously, these factors have had a significant effect on the performance of some classrooms.

The mechanical and mixed mode systems generally performed adequately but it was clear that these systems can provide a different set of difficulties for the occupants and operators. They tended to lack the flexibility of operation (for example: zonal control of underfloor heating system) offered by the natural ventilation system and also the reliance on control by a BMS required the school facilities staff to be familiar with the system and its use. The automatic windows used as part of the hybrid ventilation strategy had to be locked shut because of security concerns and this compromised the hybrid mode of operation.

ACKNOWLEDGEMENTS

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Schools: All problem buildings?

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ABSTRACT

Seven schools underwent an energy audit, evaluating the existing situation through measurement and simulation and looking to possible retrofit measures and their economic feasibility with the energy performance tool (EPB) as an instrument. The results are troubling. The seven schools audited are all problem buildings: hardly any insulation, windows quite air leaky, central heating systems poorly designed and no usage of an on purpose installed ventilation system. As a consequence, IAQ is poor, with CO_2-levels passing 3000 ppm and a high percentage of dissatisfied with air freshness during teaching hours. Annual end energy use, however, is remarkably low compared to the EPB-prediction, showing that a potentially high energy demand evokes effective rebound behavior. The negative side of that is that hardly any energy conserving measure has enough impact to show a positive NPV at the end of the return period, let it be that the annuity on investment is lower than the annually avoided energy cost.

Keywords: Schools, energy audit, energy consumption, building performance, indoor environment.

INTRODUCTION

As figure 1 shows, most school buildings in Flanders are quite aged. During the last decennia, retrofit and new construction lagged behind the needs. For that reason, the Flemish government agreed on a major program of retrofit, substitution and new construction, in collaboration with private investors. Investment projected over the years to come totals more than one billion euro. Among the requirements imposed figure energy efficiency and indoor environmental quality.

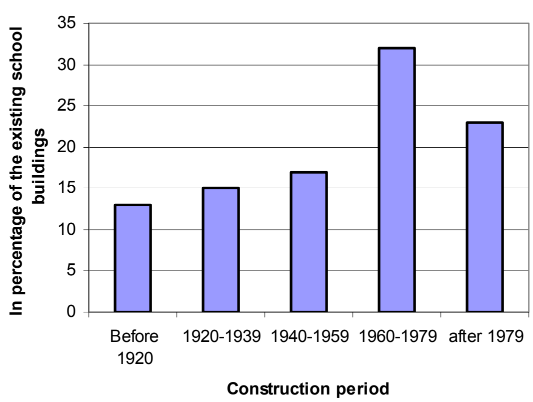

Figure 1 Age structure of school buildings in Flanders

It is thus of interest to find out how inefficient schools are and how successful energy retrofits, taking into account indoor environmental quality, could be. Such objective of course is not new. Some years ago, the IEA, Exco on Energy Conservation in Buildings and Community Systems, initiated an annex on Retrofitting of Educational Buildings (Anon, 2004). Gaitani et al, 2006, undertook a large scale energy analysis in Greek schools. They came up with five energy classes with the normalised annual consumption per square meter of floor area as indicator. Vermeir et al. 2003, reported on school acoustics, based on measurements in 47 classrooms. They found that 40% had a too high reverberation time for good speech transmission. Astolfi et al, 2003, analysed the overall environmental quality in four university classrooms. They saw discrepancies between the comfort indicators, based on measurements, and comfort satisfaction as expressed by the students. Wouters, 1989 and Hens et all, 2004, measured CO_2-concentrations in existing class rooms without on purpose installed ventilation system. The highest value noted was 5000 ppm at the end of a lesson. No class was at IDA3-level, the indoor air quality imposed by the actual Flemish energy performance decree (EPB).

SCHOOLS AUDITED

Seven schools have been audited in detail. Nine others are underway. However these audits are not finished yet. Table 1 lists the seven schools with their main characteristics.

Table 1 The seven schools

School	Type	Floor area m²	Volume m^3	Characteristics
1	Primary education (Machiels et al, 2006)	1083	4614	Oldest buildings from the fifties. Not insulated. First extension in the sixties. No insulation. Last extension in the nineties, insulated. No ventilation system. Hydronic central heating on gas
2	Special primary + secondary education, (Broers et al, 2005)	6738	23975	Built in the sixties. Not insulated. No ventilation system. Hydronic central heating on fuel
3	Special education (Defraeye et al, 2006)	3742	16935	Oldest building from the seventeen century. Gradually extended. No insulation. No ventilation system. Hydronic central heating on gas
4	Primary education (Coppens et al., 2006)	1553	5001	Built in the nineteen twenties. Part of the building retrofitted. Container class added. No to hardly any insulation. No ventilation system. Hydronic central heating on fuel
5	Primary and secondary education (part of a larger school) (Roelens et al, 2005)	7002	19070	Main building from the seventeen century. Reconstructed after world war 2. New part built in the eighties. Hardly ant insulation. No ventilation system. Hydronic central heating on gas
6	Secondary and technical education (part of a larger school) (Campforts et al, 2006)	12651	48453	School built in the nineteen eighties Roof insulated, walls not, mixture of single and double glazing. Ventilation system provided but not used. Hydronic central heating on gas
7	Primary education (Sanders et al, 2005)	2347	14496	School built between 1909 and 1911 Part of the building retrofitted. No insulation. No ventilation system. Hydronic central heating on gas renewed in 2002

BUILDING RELATED THERMAL PERFORMANCE

In Belgium, the thermal performance is expressed in terms of a level of thermal insulation:

$$\begin{array}{ll} C < 1 & K = 100U_m \\ 1 \leq C < 4 & K = 100U_m \Big/ \left(\frac{C}{3} + \frac{2}{3}\right) \\ C \geq 4 & K = 50U_m \end{array} \quad [1]$$

with U_m in W/(m².K) the thermal transmittance of the envelope and C in m the ratio between heated volume and enclosing surface, called compactness of the building. To calculate, the surfaces and layer sequence of all envelope elements plus their thicknesses have to be known. When the design documents fail, gathering those data demands time. It also introduces uncertainty, as even experienced auditors must guess the exact composition and layer thicknesses in roofs, floors and walls. As a consequence, calculated data are approximate, with an error bar of some 10%.
As table 2 shows, the seven schools audited perform poorly. To compare with, the EPB demands K45. The seven existing schools remain far above.

Table 2 Compactness, mean U-value and level of thermal insulation

School	What volume?	Compactness m	Average U-value W/(m².K)	Level of thermal insulation -
1	Whole	1.50	1.53 ±0.08	133 ±6.5
2	Part	1.58	1.98 ±0.10	167 ±8.5
3	Whole	2.33	1.24±0.06	86±4.3
4	Main building Second building 3 (container)	1.78 1.32 0.69	1.83 ±0.09 1.88 ±0.09 0.98 ±0.05	145 ±7 170 ±8.5 98 ±5
5	Old building (O) New building (N)	3.90 3.10	2.20 ±0.11 1.60 ±0.08	116 ±6 96 ±5
6	Whole	3.72	1.78 ±0.08	83 ±4
7	Whole	2.73	1.59 ±0.08	104 ±5

AIR TIGHTNESS

Per school, the air tightness of one to three classrooms was measured, using a Minneapolis blower door. The results are summarized in table 3 (V inside volume in m^3, n_{50} ventilation rate at 50 Pa, b air permeance exponent). The table reveals a large spread, from n_{50}=4 h^{-1} to n_{50}=12 h^{-1}. The real infiltration rate will be much lower, as it depends on the orientation of the exterior wall, the actual pressure difference between that wall and the corridor doors, the leakages in the separation wall with the corridor and the way the corridor acts as discharge. Hence, in none of the cases, infiltration may suffice to maintain correct ventilation during classes. The negative exponents also learn that in some classes important leaks close as the air pressure difference increases!

INDOOR ENVIRONMENTAL QUALITY DURING CLASSES

Indoor environmental quality was monitored by logging temperature, relative humidity and CO_2 in some of the schools during several class days in winter. An excerpt of the results is listed in table 4 and shown in figure 2. PD for air freshness should stay below 10%, while, as asked by the

EPD, in IDA 3 CO_2 should not pass 1350 ppm. Clearly, none of that is realized, making indoor environmental quality unacceptable.

Table 3 Air-tightness of separate classes (exterior walls only)

School	Class								
	1			2			3		
	V m^3	n_{50} h^{-1}	b -	V m^3	n_{50} h^{-1}	b -	V m^3	n_{50} h^{-1}	B -
1	288	6.9	0.20	260	6.0	-0.066	330	7.1	-0.058
2	128	4.0	0.67						
3	271	3.9	0.68	124	2.4	0.62	144	3.8	0.70
4	219	4.4	-0.027	218	8.2	-0.003			
5	137 (O)	10.5	0.67	120 (N)	4.3	0.77			
6	147	10.7	-0.024	147	4.6	0.002			
7	300	5.4	0.62	300	12.0	0.54			

Table 4 Indoor environmental quality

School	Temperature °C			Vapour pressure excess Pa			PD Air freshness	CO_2 ppm
	Mean	Min	Max	Mean	Min	Max	Max	Max
1	17.7	14.7	23.2	197	-138	722	40.9	1490
2	15.0	12.0	26.8	215	-155	800	No data	No data
3	20.3	14.2	25.2	198	-266	1059	57.6	2553
4	17.5	10.3	23.1	151	-169	609	30.6	3675
5	16.5	11.3	23.3	No data			No data	No data
6	17.1	11.1	25.0	No data			No data	No data
7	17.9	14.1	23.4	35.9	-403.8	495.6	18.4	No data

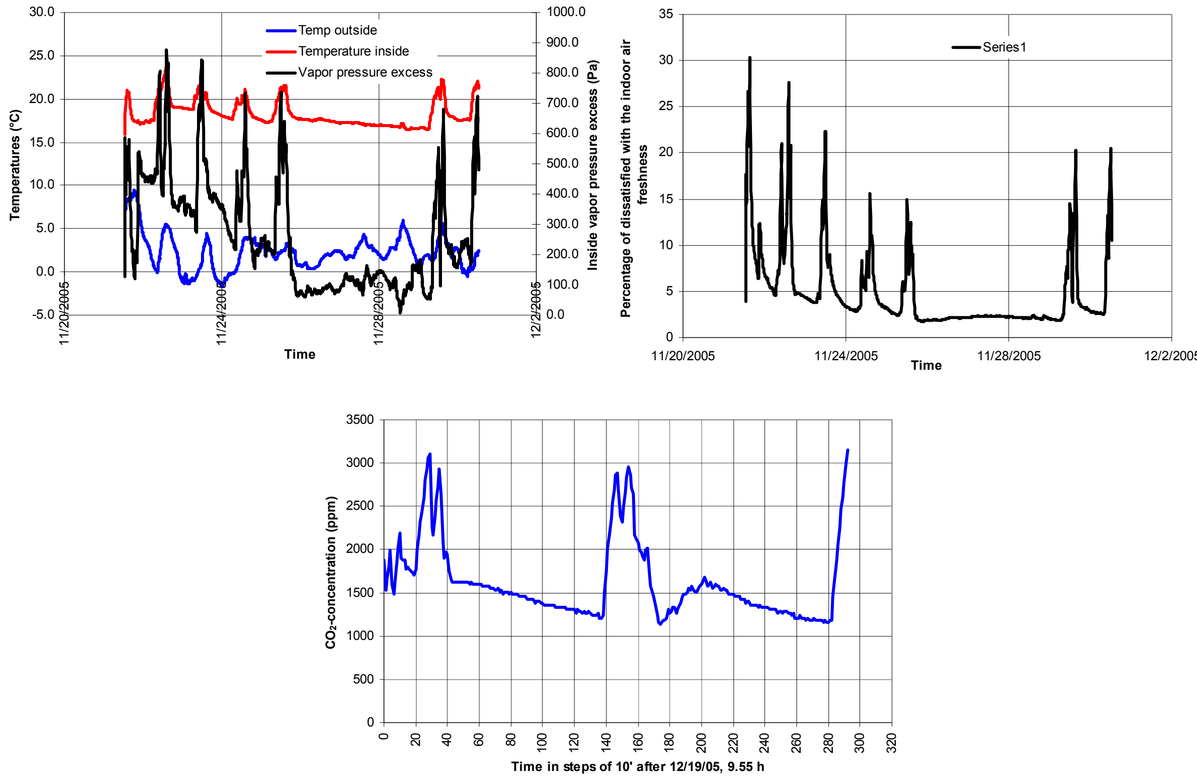

Figure 2 Indoor temperature, indoor vapour excess and percentage of dissatisfied with the freshness of the air in class 1 of school 1, CO_2-concentration in class 1 of school 4.

MEASURED AND PREDICTED ENERGY CONSUMPTION

For each school, energy consumption data were gathered by analysing the bills and measuring usage on a weekly basis. The results, transposed to the standard year of the EPB, together with the EPB-results, are listed in table 5. They show that, with the exception of school 5, the EPB overestimates energy consumption considerably. Several reasons explain that fact:

- The EPB assumes that ventilation guarantees an indoor environmental quality IDA3 during classes. To achieve that, the average on purpose ventilation flow should attain 29 m^3 per hour per pupil. In reality, during class hours, infiltration only does the job. In school 4 for example, CO_2 data gave an infiltration flow totalling 8.3 m^3/(h.pers). Happily, the hours, that on purpose ventilation is needed, are exaggerated in the EPB for schools. Not 2628, but 1332 hours is a better number. That moderates the extra energy needed for correct ventilation
- The EPB supposes that the whole school is continuously heated at 19°C. This is not the case. Corridors for example are not, while 17.5°C for the average temperature in the heated parts is a better guess.
- Internal gains are underestimated. The EPB takes 1 pupil per 4 m² of floor surface. A class of 20 should thus have a surface of 80 m². Classes are typically smaller, with surfaces closer to 55 m² for 20 to 30 pupils.

Table 5 End energy consumption, measured (reference year) versus calculated (EPU-tool)

School	Calculated according to the EPB MJ/a		Measured MJ/a		Boilers		Ratio calc./meas. Heating
	Heating ±10%	Electricity (light, auxill.)	Heating	Electricity (all)	Number	Mean $\eta_{p,flue}$	
1	2395000	191800	601000	156100	1	89.9	3.9
2	344 900 (part)	No data	3 240 000 (tot)	No data	2	89.8	-
3	4350000	No data	1517900	202830	5	94.9	2.9
4	1080000 (part)	88 459 (total)	308350 (part)	67600 (total)	1	90.3	3.5
5	2818000 (part)	No data	2677600 (part)	893400 (total)	3	87.0	1.05
6	8530000 (part)	1167850 (part)	11736000 (tot)	2990600 (total)	3	93.7	-
7	2950700	No data	1215600	No data	1	95.6	2.4

CONSEQUENCES FOR RETROFITTING

Judging retrofitting measures on cost effectiveness may be done by comparing the annuity on investment with the avoided annual energy cost or by evaluating the net present value over a stated period of time. The first possibility is very attractive. In fact, if the avoided energy cost passes the annuity then the measure effectively lowers the annual costs.

The problem yet is that, due to the overestimation of energy consumed, the EPB cannot be used as a basis for evaluation. One should use the measured data instead and multiply avoided energy, calculated by the EPB, with the ratio between measured and calculated energy use. Hence, when schools are retrofitted with inclusion of a correct IDA 3 ventilation system (which must be done), upgrading the insulation, increasing heating system efficiency and relighting may not compensate for the extra energy needed for ventilation. Take for example school 4. In the case being, infiltration and some adventitious ventilation increased energy demand for the main building with some 41000 MJ/a. A forced IDA3 ventilation system without heat recovery lifts that number with some 77000 to 191000 MJ/a. Compensating for that could be done by diminishing the demand with the same amount, for example by insulating the pitched roof, now with a U-value between 2.7 and 3.1 W/(m².K). Yet, even a U-value 0.2 W/(m².K) does not succeed in doing that. Following the EPB, it should as the demand may diminish with some 236000 MJ/a. In reality, decrease will be restricted to some 67000 MJ/a. Even less is possible, as part of that avoided demand is lost by an increase in inside temperature.

Or, the need for a better inside environmental quality, which means more ventilation, and the rebound observed in heating habits to compensate for bad thermal quality of the buildings, makes it extremely hard to realize economy when retrofitting schools with an eye on diminishing energy consumption and CO_2-emissions.

CONCLUSIONS

School buildings in Flanders are quite old, while investment in retrofit and new construction lagged behind. Energy audits in seven schools revealed that on the average the thermal quality of the older and even newer buildings is bad. Also, six out of the seven had no ventilation system and, in the one who had it, it was not in use anymore. Hence, ventilation is mainly infiltration linked, with some adventitious airing by opening a window between class hours. The consequence is bad indoor air quality, a fact which was already stated in previous studies and got confirmation by measurements in four of the schools.

Any school management tries to neutralize deficient thermal quality by persistent rebound behaviour. As a consequence, energy consumption, calculated with the official EPB-tool, typically overestimates energy use. That, and the requirement to guarantee an IDA 3 indoor environment during classes, jeopardizes most attempts to ameliorate energy efficiency during retrofit in an economic affordable way.

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Ventilation of Dutch schools; an integral approach to improve design

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SUMMARY

Indoor Air Quality and thermal climate in schools is very important as it has a direct relation to the health and performance of the pupils. The status quo in the Netherlands is presented (e.g. average CO_2 levels in schools, quality of ventilation). The goal of a first study was to evaluate the performance of exhaust-only ventilation systems. The performance was rather disappointed there were a lot of problems and insufficient situations found. During the next years different master students [1,2,3,4] together with the staff of Technische Universiteit Eindhoven were researching different aspects of the problem and trying to find solutions. In a following study, 6 schools with different ventilation systems were studied. Main conclusions from these studies were: IAQ in the evaluated schools did not meet the requirements and more ventilation was essential for better IAQ. A new integrated approach to design adequate solutions for ventilation of school buildings was developed. First design results are described in the paper.

INTRODUCTION

Indoor air quality has caught attention of the Netherlands Ministry of Housing, Spatial Planning and the Environment and a large campaign was started in 2005 to make the public aware of the dangers to health as result of poor ventilation in housing. Indoor pollutant levels are often greater than the outdoors, and since Dutch people spend nearly 90% time indoors, good indoor air quality is very important. Indoor Air Quality (IAQ) at schools is of special concern since children are extremely sensitive to results of poor air quality. IAQ in schools must reach the basic requirements and should be considered as a high priority because (Landrigan 1997): (1) Children more sensitive as they still developing physically and more likely to suffer from indoor pollutants, these growth processes are delicate and vulnerable to disruption, (2) Children are less well able than adults to metabolise and excrete most environmental toxins, (3) Children are relatively more heavily exposed to environmental toxins as they breathe higher volumes of air relative to their body weights. Good air quality in classrooms supports children’s learning ability. Poor IAQ in schools influences the performance and attendance of students, primarily through health effects from indoor pollutants (Mendell and Heath, 2005).

VENTILATION STANDARDS

There are numerous standards and guidelines covering indoor air quality (IAQ), recommended by international health associations, industry organizations and governments. Ventilation standards state either outdoor air supply requirements (volume per time per person), or outdoor air change-rate (h^{-1}), or both. Dutch schools have to meet the Dutch Building Code (Bouwbesluit), which requires a classroom ventilation rate of 2.8 $l/s{\cdot}m^2$ at an occupancy rate with 1.3 to 3.3 m^2 floor area per person. For a standard classroom of 50 m^2 and a maximum occupation of 32 students, this results in a ventilation rate of 4.2 l/s per person. Dutch Building Code refers also to guideline NEN 1089, which requires a ventilation rate of 5.5 l/s per person based on a level of 1000 ppm CO_2-concentration with a maximum of 1200 ppm. So depending on the situation the highest ventilation rate should be used.

The European legislature (CR1752) uses Fanger's model for dissatisfaction and specifies 3 categories of % occupants satisfied with perceived indoor air quality: high quality (at least 85%), medium quality (at least 80%), and moderate quality (at least 70%).

The guidelines for ventilation in North America, provided by ASHRAE (American Society of Heating, Refrigerating and Air-conditioning Engineers), Standard 62-1999, recommends a minimum ventilation rate of 8 l/s-person for classrooms. They pose an acceptable indoor air quality when the air is free of pollution in damaging concentrations and when a majority of satisfaction (80% or more) is found. Indoor air quality here is a combination of air contaminants and the response of occupants exposed to indoor air.

Carbon dioxide concentrations are often used as a substitute of the rate of outside supply air per occupant (Seppänen et.al 1999). IAQ in schools is primarily evaluated by CO_2-concentrations. ASHRAE Standard 62-1999 recommends an indoor CO_2-concentration of less than 700 ppm above the outdoor concentration (~1200 ppm) to satisfy comfort criteria with respect to human bio effluents. The Dutch standard NEN 1089 asks for a maximum CO_2-concentration of 1200 ppm in classrooms (van Dijken, 2004).

Literature on relationships (Shendell et.al 2004) between indoor air and environmental quality (IEQ) in class rooms and students health and academic performance has been reviewed by Heath and Mendell (2005) and Daisey et.al (2003). Relatively not many field studies were conducted on the performance of ventilation systems in schools. For European schools some results are shown in figure 2 (Daisey et.al 2003).

Study	**number of schools**	**CO_2, ppm**	
		Average	Range
Nielsen,1984	11	1000	500-1500
Norback et.al 1990	6	1290	950-1950
Norback,1995	6	1320	700-2700
Smedje,1997	96	990	425-2800

Figure 2. Average and range of CO_2-concentrations reported in the scientific literature for European schools (Daisey et.al. 2003)

In the Netherlands different investigations on indoor air quality were conducted and CO_2 levels measured. In 1987 the Agriculture University of Wageningen conducted research to indoor air quality. Measurements of CO_2 concentrations were done and in 8 out of 12 schools, CO_2 levels rose above the marginal value of 1200 ppm in more than 50% during school hours. (Sandt et al, 1987). In 1992 the local health department (GGD) in West-Brabant researched indoor air quality in secondary education buildings. Measurements of CO_2 levels were done and levels up to 4800 ppm were detected. (Leentvaart et al, 1992). In 1993 in Groningen different primary schools were inspected. In 3 out of 4 classrooms, CO_2 levels reached the marginal value of 1200 ppm. Maximums of 2400 ppm were detected (Meijer, 1993). In 1997 IAQ measurements were done at 4 primary schools in the region of East Noord-Brabant. Peak levels of 3500 ppm were detected. (Boske, 1997). Municipality Groningen did measurements in 16 classrooms and they found median levels of 919 to 1940 ppm (Wassing, 2003).

EXPERIMENTS

The goal of a first study was to evaluate the performance of exhaust-only ventilation systems. In 5 Dutch schools measurements were conducted in the heating season for a period of around 7 days. These measurements included: IAQ (CO_2), thermal comfort, airflow and outdoor conditions. A logbook and questionnaires obtained information about use of ventilation facilities and satisfaction of users. Results of the measurements showed that in 4 out of 5 evaluated classrooms the indoor air quality did not meet the requirements for good indoor air quality. CO_2-concentrations are too high indicating that ventilation is not adequate. Therefore, a first conclusion was that natural air supply in classrooms without any draught prevention is an unacceptable solution. Parallel to the research another study was done by Froukje van Dijken, she studied IAQ of 11 schools. Both results were used by the REHVA Taskforce 4 "Indoor Climate and Energy of School Buildings" in their primarily report.

In a following study, 6 schools with different ventilation systems were studied, to search for concepts, which had fewer problems. Main conclusions from this study were also: IAQ in the evaluated schools does not meet the requirements and more ventilation is essential for better IAQ. The capacity of ventilation systems has to be increased. However air supply by natural ventilation is limited to vents in the façade. In well insulated buildings the required heat supply is not sufficient to prevent draught due to the supply of cold outdoor air. Therefore, a more distributed way of supplying air is needed in these systems.

Study	**number of schools**	**CO_2, ppm**	
		Average	Range
Joosten, 2004	5	1220	480-2400
van Dijken, 2004	11	1580	450-4700
van Bruchem, 2005	6	1355	550-3000

Figure 3. Average and range of CO_2-concentrations for Dutch schools (Joosten 2004, v.Dijken 2004, van Bruchem 2005)

METHODOLOGY

The results of the measurements indicated that, based on the current ventilation standards, many classrooms are not adequately ventilated. As an object for further research is an improvement of the design process. (Mendell and Heath, 2005). A first line of defense against poor IAQ in classrooms is adequate ventilation and this should be a major focus of design efforts (Daisey et.al 2003).

The current design process for schools normally begins with selection of an architect. It then proceeds with programming and schematic design. Next the design engineer comes in and finally the BS (Building Services)-engineer completes the design team. But already many decisions were made in the early conceptual design that influences the development, contract documents, construction, commissioning and occupancy. Clear goals were not part of the often general programming and the design brief. The sooner goals, to ensure IAQ are brought into the design process, the easier and less costly they are to incorporate.

Integrated design promises major advantages. It draws from all disciplines involved in designing a building and reviews their recommendations as a whole from the early start of a project. Project team collaboration and integration of design choices should begin at the programming phase. Indoor air quality encompasses such factors as maintenance of acceptable temperature and relative humidity, control of airborne contaminants, and distribution of adequate ventilation air. It requires deliberate care and cooperation on the part of the entire project team. BS-engineers must design ventilation systems that dilute the by-products of occupant activities and, to the greatest extent possible, supply fresh air on demand in the right quantities, in the right locations.

Up to now the building design process is more or less sequential; first the building is designed and subsequently the heating/ cooling/ ventilating system. Communication between architect and building services consultants is based on abstraction, i.e., the exchange of abstract descriptions of a design. During design support, it's important to transfer the essentials of the applied structures and mechanisms without overloading the other party with unwanted details.

Design has normally a very dynamic nature, with a tendency to ad hoc actions, which should be supported by design aid systems. To develop the required model of design support an existing model has been extended: Methodical Design (van den Kroonenberg 1978). The methodical design process can be described on the conceptual level as a chain of activities which starts with an abstract problem and which results in a solution.

In order to survey solutions, engineers classify solutions based on various features. This classification provides means for decomposing complex design tasks into manageable size problems. An important decomposition is based on building component functions. The functional decomposition is carried out hierarchically so that the structure is partitioned into sets of functional subsystems. The decompositions are carried out till arrived at simple building functional components whose design is a relatively easy task, see figure 4.

The decomposed functions were put into an array. Left column were the functions listed, which were combined with a row of solutions for each function, see figure 5. The matrix of functions and their solutions are called a Morphological chart and were developed by Zwicky(Zwicky, 1969). Each combination of possible solutions for the functions was combined to overall solution for the design task. Selecting the most likely solution could be done with help of the Kesselring-diagram Kroonenberg, 1978). All the solutions were marked based on the criteria of the design brief. The design criteria were divided into criteria concerning functioning and realization. The relative score was presented in de S-diagram, see figure 6. By using Morphological charts communication between design team members becomes easier and there was a clear overview of all the possibilities discussed. The Kesselring method (Kesselring, 1954) made the decision process clear and understandable for the design team and all so for all people outside the team. The proposed method is used to design a ventilation system for a typical classroom. The solution that resulted is a balanced displacement ventilation system with heat recovery, as shown in figure 7.

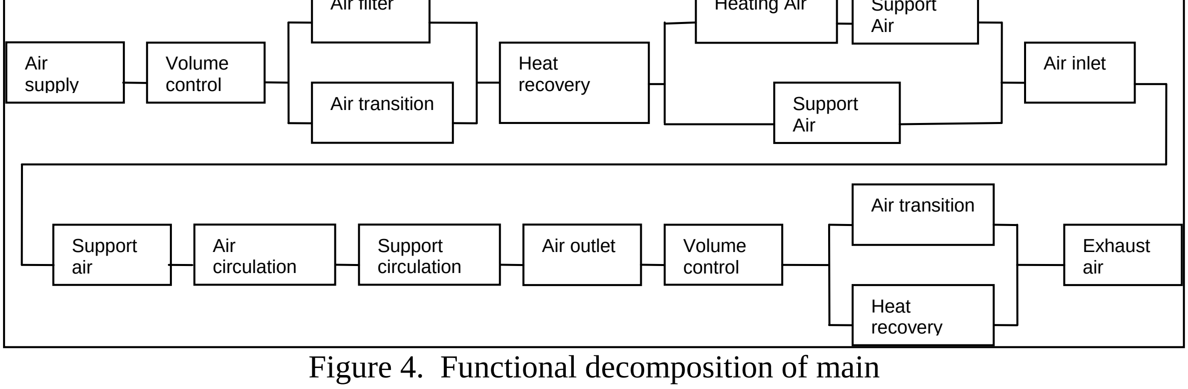

Figure 4. Functional decomposition of main

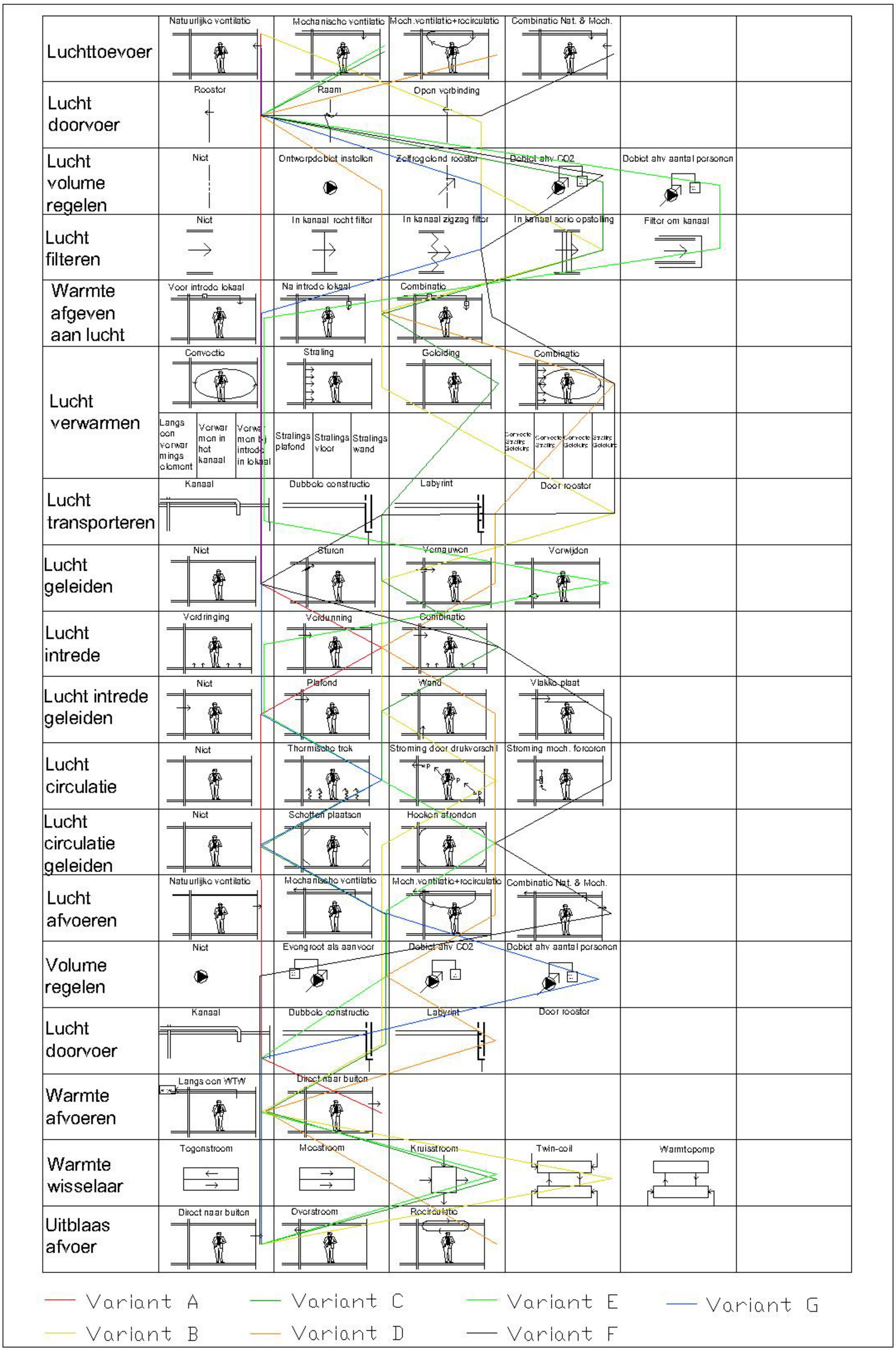

Figure 5. Morphological chart, 18 subfunctions with different solutions.

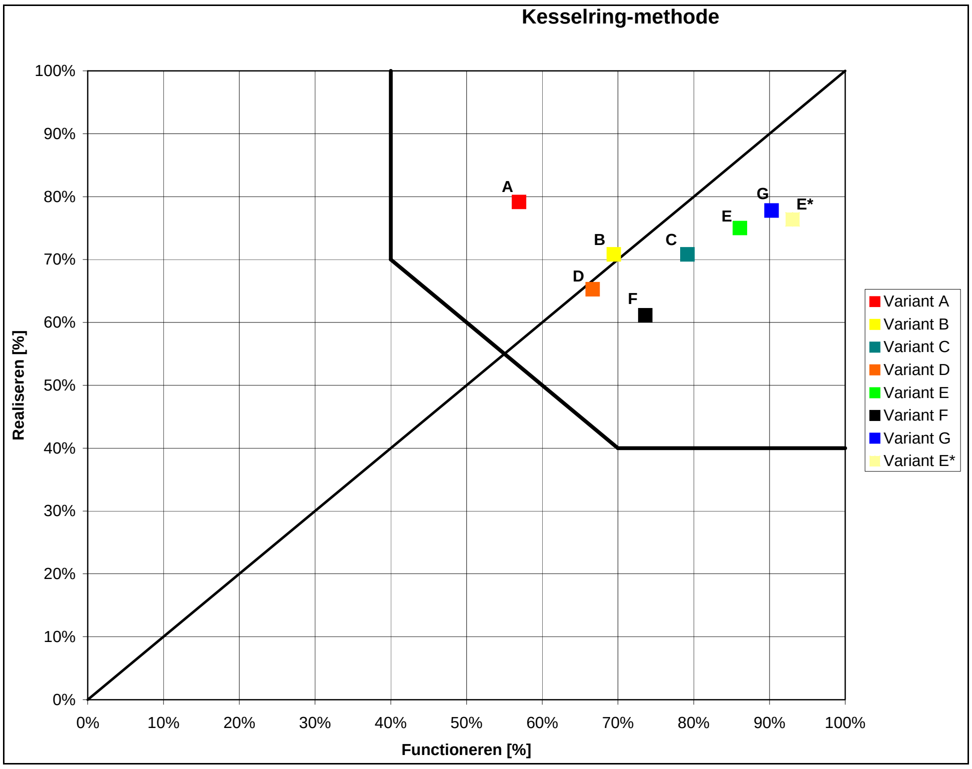

Figure 6. S-diagram Kesselring method

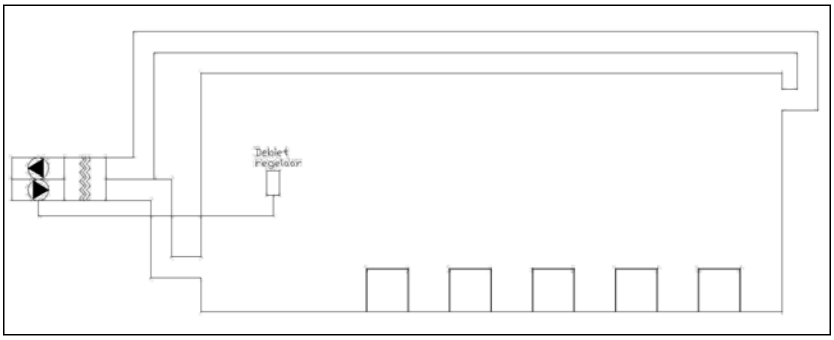

Figure 7. Schematic of displacement classroom ventilation

Instead of normal metal ductwork, textile air ducts are proposed; these can be removed easily and washed in a washing machine.

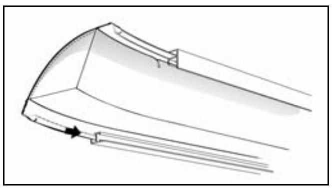

Figure 8. Textile air duct (Euro-Air, 2005)

Normally the flow pattern of heated air is problematic when displacement ventilation is used. In classrooms this is only the situation during start-up, as the pupils generate more than enough heat themselves once they are in the classroom. During the start-up the air distribution is like shown in figure 9. The walls and windows are primarily heated up and there is no good air distribution. But as there are still no pupils this is not a problem. When the room is used there is too much heat and the air through the displacement ventilation system has to be brought in with a slight under temperature. The expected flow pattern will become as shown in figure 9. By putting the textile air ducts nearly all around the floor of the classroom, a good distribution with a low air speed will be generated. In follow-up research, the solution will be simulated and a laboratory test will be done to verify the design.

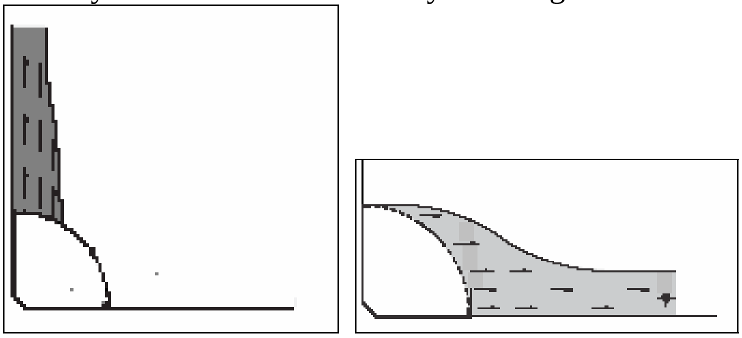

Figure 9. Flow pattern during start-up situation and during normal use

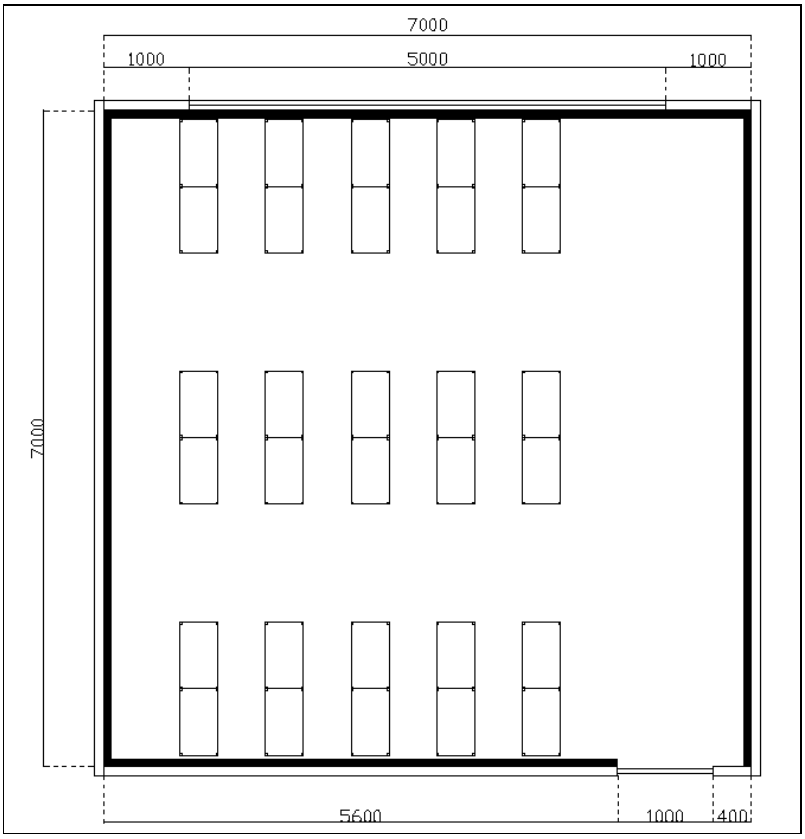

Figure 10. Lay-out displacement air-duct

Many ongoing research efforts aim at introducing the used methodical design process into building design and architecture, and so linking architectural design with HVAC-design. In the Netherlands this has resulted in a stark interest in what is commonly named 'integral design'.

Integral design is meant to overcome the difficulties of design team cooperation, by providing methods that make it possible to communicate the consequences of design moves on areas such as construction, costs, life cycle and indoor climate at early design stages, between the different disciplines.

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Particulate matter levels in Portugal (mainland and islands). A preliminary study for outdoor/indoor environment in basic schools.

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SUMMARY

This study deals with Particle Matter (PM) levels below 2.5 µm ($PM_{2.5}$) in Portugal and shows that US EPA (United States Environmental Protection Agency) directive is exceeded in a few places. $PM_{2.5}$ total mass concentration measured in several places located in Portugal mainland and islands and the outskirts are quite well correlated for a few sites. Results show that it is important to determine the elemental composition of $PM_{2.5}$, and to develop an epidemiological study in Portugal to find a possible association between $PM_{2.5}$ levels, sources and morbidity/mortality. However, the results imply that a source-oriented evaluation of PM health effects needs to take into account the uncertainty associated with the spatial representativity of the species measured at a few sampling stations. For that purpose the survey using biomonitors may contribute positively.

INTRODUCTION

Several reports revealed significant correlations between PM levels and increased respiratory and cardiovascular diseases, and mortality [1]. Understanding and controlling air pollution becomes then important but difficult, because the emission inventories and transport models are problematic in the evaluation of particulate atmospheric pollution.

In South European regions, such as Portugal, in addition to anthropogenic sources, the ambient aerosol has an important contribution from natural dust, due to local emissions from bare soil, and an influence of episodic African dust transport outbreaks [2]. Moreover, the Portuguese coastal areas have an important input of marine aerosol [3]. This is due to the geographic position of Portugal (on the extreme southwest of Europe) and the dominant western wind regime, influenced by the presence of the semi-permanent Azores high-pressure and the Icelandic low-pressure systems over the North Atlantic Ocean. These air currents bring anthropogenic influences from the three continents: America, Africa and Europe, having a large influence in the measured fine particles.

The present study uses the measurements of particulate air of aerodynamic diameter below 2.5 µm ($PM_{2.5}$) through the country, measured by the Portuguese Environment Institute (IA). The aim is to get a good knowledge of the $PM_{2.5}$ levels and their provenance, to help in interpreting the $PM_{2.5}$ values and their contents obtained by ITN at one site of Lisbon, under a project aiming to study the impact of the atmospheric aerosol in human health (POCI/AMB/55878/2004, financed by the Portuguese Foundation for Science and Technology[4,5].

METHODS

Air pollution data were obtained from the Portuguese Environment Institute (IA) Air Quality Monitoring Network. The latter consists on several stations distributed in Portugal providing hourly data of the main atmospheric pollutants such particulate matter, SO_2, NO, NO_2, CO, and O_3. In this work, data of particulate matter of aerodynamic diameter below 2.5 µm, designated by $PM_{2.5}$, registered every hour and averaged for 24 h, were selected for discussion.

Figure 1 presents the location of the sites where the data were registered: 1) Madeira archipelago: one site in Porto Santo island and three sites in Madeira island (Qta Magnólia, S. Gonçalo, S. João); 2) Portugal mainland – a) two sites in Faro (Afonso III and Joaquim Magalhães), two sites in Lisbon (Entrecampos and Olivais); b) one site in Alcoutim (Cerro), Santiago do Cacém (Monte Velho), Alandroal (Terena), Chamusca, Leiria (Ervedeira), Fundão (Salgueiro), Estarreja (Salgueiro), Viana do Castelo (Sra do Minho), Vila Real (Lamas de Olo), and Vermoim. They cover quite well the whole country and the archipelagoes, with exception of Azores archipelago.

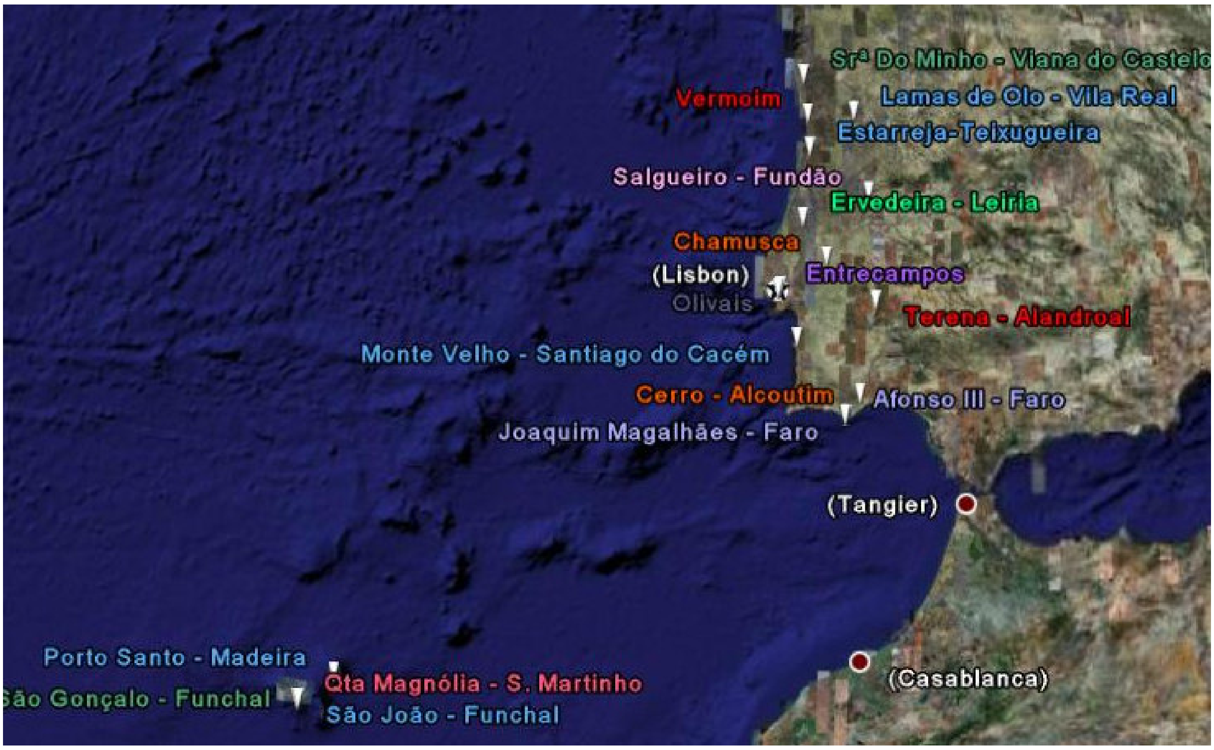

Figure 1. Sites where PM2.5 is being collected by the Portuguese Environment Institute network.

RESULTS

Table 1 presents, for the sites displayed in Fig. 1, the number of days with availability of $PM_{2.5}$ registrations, and some simple statistical data on the sets: mean, median, maximum, and minimum.

A few sites – Porto Santo, S. Gonçalo, Santiago do Cacém, Alandroal, Viana do Castelo, and Vila Real – have less than 75% of the expected annual results, and therefore should be restrictively interpreted. In the calculation of the mean and median, missing data were not filled up and were not considered as zero.

Table 1. Statistical data of $PM_{2.5}$ results.

$\mu g\ m^{-3}$	Porto Santo	Qta Magnólia Funchal	S. Gonçalo Funchal	S. João Funchal	Afonso III	Joaquim Magalhães
2005	Madeira Archipelago				Faro	
Nr. days	225	292	251	285	361	365
Mean	7.66	9.33	9.38	23.80	15.07	12.50
Median	7.62	7.44	9.46	22.75	14.31	11.63
Maximum	26.96	46.29	39.86	68.95	53.53	44.45
Minimum	0.04	0.22	1.00	5.57	1.00	2.28
	Cerro	Monte Velho	Terena	Entrecampos	Olivais	Chamusca
$\mu g\ m^{-3}$	Alcoutim	Santiago do Cacém	Alandroal	Lisbon		
Nr. days	362	271	256	365	354	352
Mean	7.99	12.38	10.53	22.37	15.26	14.85
Median	7.34	10.21	8.83	18.30	11.84	11.48
Maximum	44.21	78.00	86.86	168.99	124.83	181.58
Minimum	0.28	0.53	2.61	4.48	2.31	1.00
	Ervedeira	Salgueiro	Texugueira	Sra do Minho	Lamas de Olo	Vermoim
$\mu g\ m^{-3}$	Leiria	Fundão	Estarreja	Viana do Castelo	Vila Real	
Nr. days	360	360	365	150	251	352
Mean	33.79	9.90	26.32	14.52	13.91	27.33
Median	29.17	7.92	22.71	10.73	9.96	20.31
Maximum	323.79	61.13	129.33	133.17	97.21	121.83
Minimum	5.96	1.13	3.17	3.21	3.92	0.04

Mean and median values are very similar indicating normal distribution of the data. As of today the EU (European Union) has not put forward any $PM_{2.5}$ limit values, and taking this value as a reference one, the $PM_{2.5}$ annual average mass concentrations at S. João (Madeira), Entrecampos (Lisbon), Leiria, Estarreja, and Vermoim exceed it. As a result of a preliminary study carried out in the metropolitan area of Lisbon it was concluded that the EU directive is exceeded in a systematic way in the centre of Lisbon due to the traffic inside the city [6]. Maximum values are a few times too high to be believed, as it is the case for Chamusca, Leiria and Viana do Castelo. Figs. 2 and 3 represent the mean and maximum values of the sites, for a better visualization.

Pearson correlations were established between the sites and the results are shown in Table 2. The latter only shows the correlations for which the correlation coefficients are higher than 0.5. There are five correlations with correlation coefficients higher than 0.7, and they are displayed in bold.

The two sites of Faro are very well correlated, indicating the same emission source. The Faro's source is also influencing Alcoutim, situated at close distance at Faro's northeastern direction (see also Fig. 4). The source is most probably the international airport.

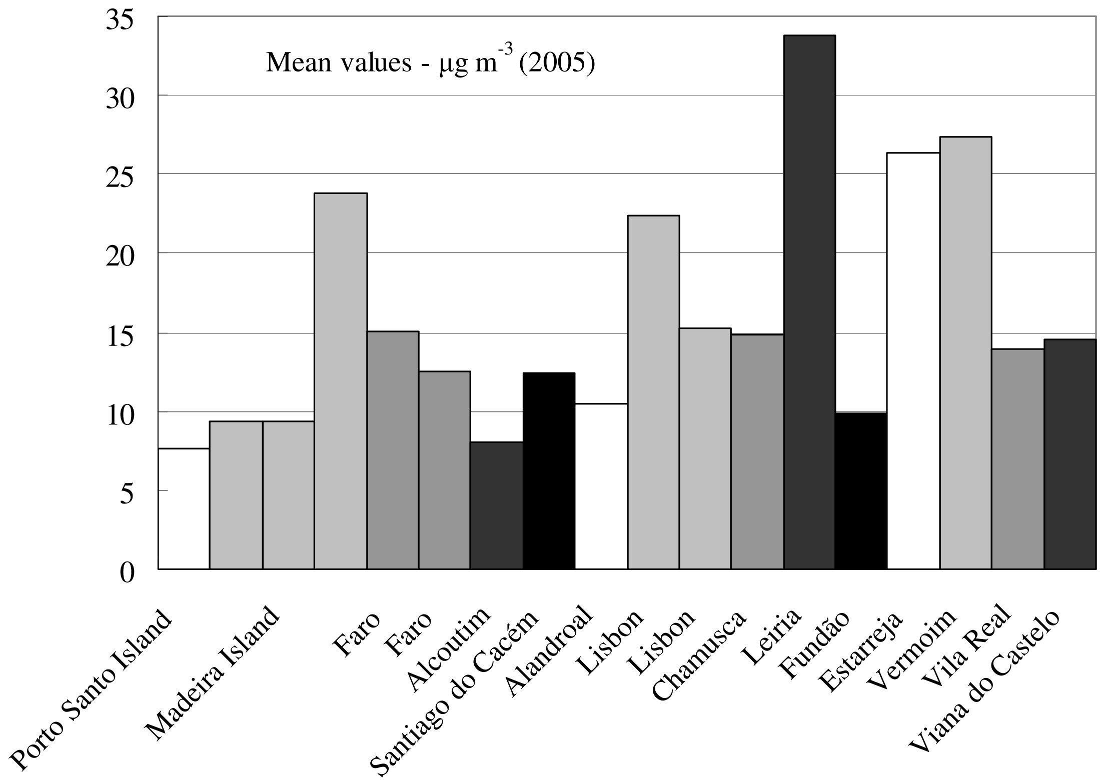

Figure 2. Mean values of PM2.5 registered at the sites monitored by the Portuguese Environmental Institute.

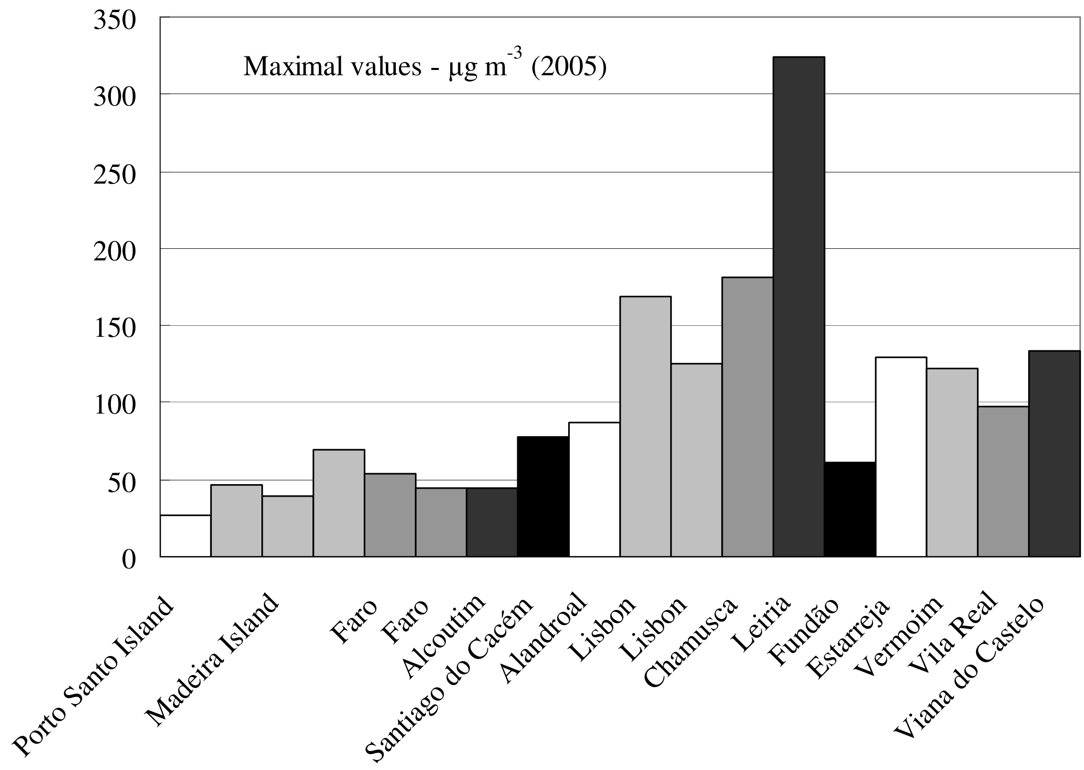

Figure 3. Maximum values of PM2.5 registered at the sites monitored by the Portuguese Environmental Institute.

Table 2. Correlation coefficients between pairs of sites. It is shown the cases with correlation coefficients larger than 0.50 only. Coefficients larger than 0.70 are in bold.

	C	F	G	H	I	J	K	L	M	N	O	P	Q	R
A				0.57				0.52						**0.71**
B	0.58													
E		**0.85**	**0.81**	0.66	0.68	0.51	0.51	0.51		0.50				0.64
F		1.00	0.67	0.67	0.65			0.51				0.51	0.50	0.65
G			1.00	0.66	0.64			0.55					0.53	0.69
H				1.00	0.66	0.63	0.67	**0.79**	0.52		0.52	0.65	0.52	**0.80**
I					1.00			0.66		0.59			0.55	0.66
J						1.00	**0.96**	0.55	0.68		0.67	0.57		
K							1.00	0.63	0.67		0.61	0.53		
L								1.00	0.65			0.56		**0.86**
M									1.00		0.58	0.59		0.59
N										1.00			0.68	
O											1.00	**0.76**		
P												1.00		0.60
Q													1.00	0.55
R														1.00

Note: A: Porto Santo; B: Qta Magnólia/Funchal; C: S. Gonçalo/Funchal; D: S. João/Funchal; E: Afonso III/Faro; F: Joaquim Magalhães/Faro; G: Cerro/Alcoutim; H: Monte Velho/Santiago do Cacém; I: Terena/Alandroal; J: Entrecampos/Lisbon; K: Olivais/Lisbon; L: Chamusca/Lisbon; M: Ervedeira/Leiria; N: Salgueiro/Fundão; O: Texugueira/Estarreja; P: Sra do Minho/Viana do Castelo; Q: Lamas de Olo/Vila Real; R: Vermoim.

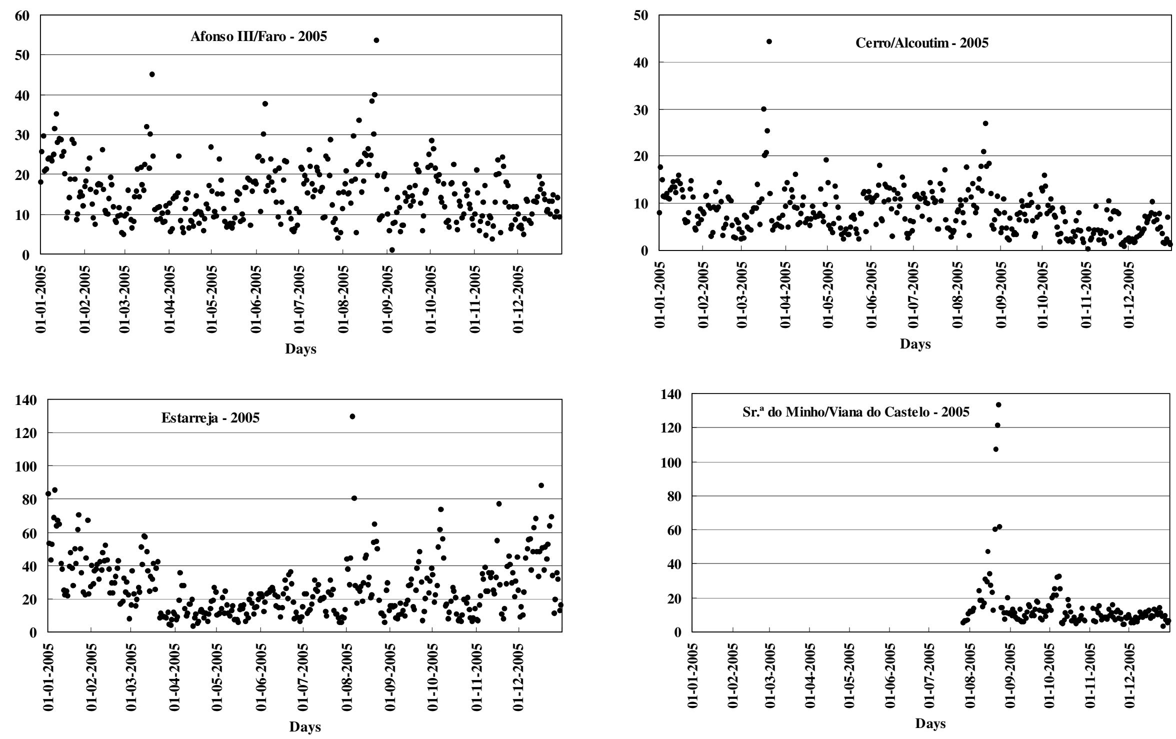

Figure 4. PM2.5 concentrations in μg m^{-3} for highly correlated sites.

Other high correlation concerns Viana do Castelo and Estarreja, the first with paper industry and the second with chemical industry, however Viana do Castelo has just half of the year registered (see also Fig. 4).

Vermoim, Chamusca and Santiago do Cacém, the three sites from north to south over the same direction line, are well correlated, appearing to indicate remote anthropogenic sources (see Fig. 5), originated from the predominant northern winds.

The two sites of Lisbon also correlate well as it has already being mentioned in [6].

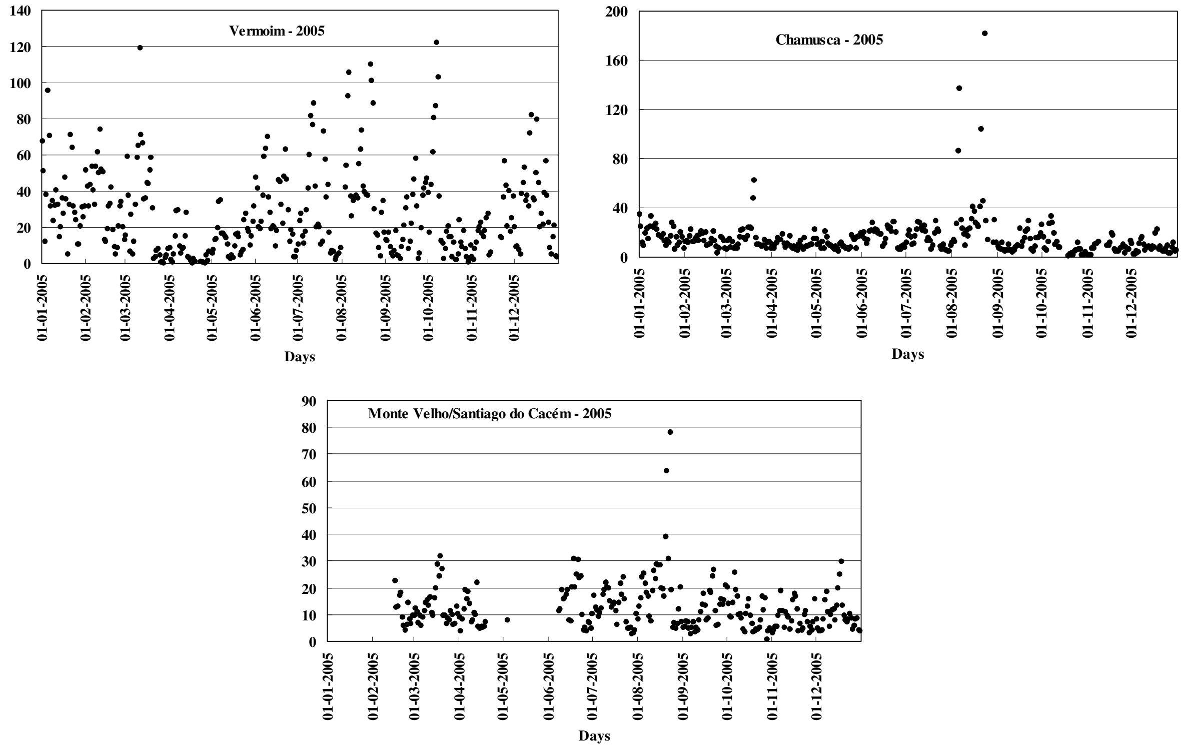

Figure 5. PM2.5 concentrations in μg m^{-3} for highly correlated sites.

At each site, the average per hour was calculated. Three main patterns were found: 1) two peaks of high concentrations, one around 9 h and other during the night (between 22 h and 24 h); 2) one peak during the night (21 h – 8 h); 3) constant emission without visible peaks. Situation 1) occurred for the cities localized sources as Faro and Lisbon, being concluded that it may be related to differentiation in intensity of traffic, either airplanes or cars; situation 3) occurred for the islands and Viana do Castelo, which may be related to sites more influenced by long-distance air mass transportation; situation 2) may be due to local industry . An example of each is shown in Fig. 6.

DISCUSSION

PM2.5 concentrations in Portugal are preoccupant; their values exceed very often the existing US EPA regulated values. There is not yet an EU and Portuguese legislation, however in a near future such regulations are expected. There is need to investigate from where these high values originate, this is, the emission sources. Some, out of the Portuguese borders, are helpless concerning the suggestion of reduction of emissions; others, emitted by Portuguese sources, may be reduced if the composition of PM2.5 is known to identify the sources. It is then suggested that the Portuguese Environmental Institute analyses or requests for analysis their PM2.5 filters, in order to get their composition, on basis of which anthropogenic sources may be recognized.

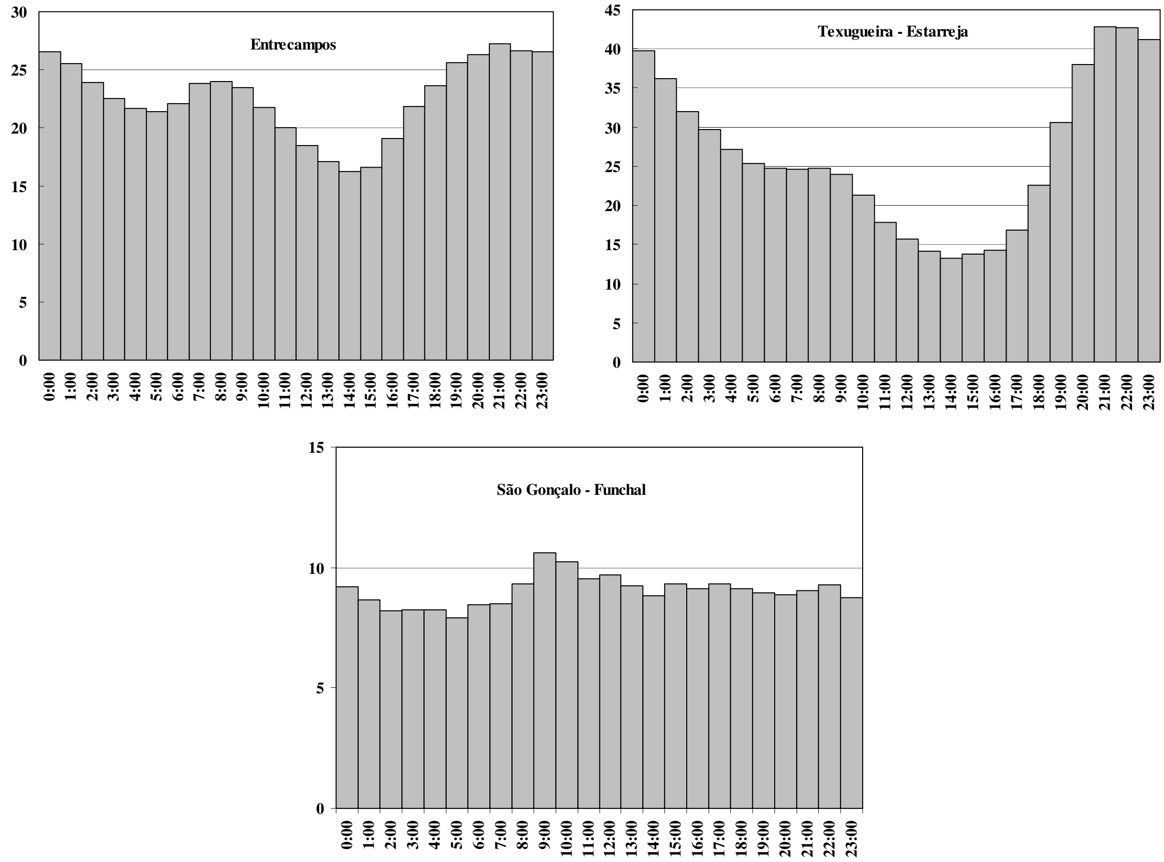

Figure 6. $PM_{2.5}$ concentrations in $\mu g\ m^{-3}$ for the hour average during 2005.

This study will be shortly followed by the recognition of the air mass trajectories which led to the high episodes observed, in order to understand the origin of such high masses.

In a recent crossing between lichen survey data held in 1993 in Portugal [7] with mortality data [8], nickel showed a statistically significant association with mortality from cancer in a simple linear regression without accounting for confounders. Bromine, iodine, lead, sulphur, antimony, and vanadium, were also associated with cancer mortality in this study.

I recent study [9], composition of PM2.5 was investigated in order to identify the main sources affecting an industrial/urban area with a special geographical position— 20km northeast of the southwest European coast, 10 km north of Lisbon, Portugal. Source apportionment results show that the major PM2.5 aerosol mass contributors include secondary aerosol (25%) and vehicle exhaust (22%). Anthropogenic sources affect mainly the fine fraction.

Measurement of PM2.5 is a better methodology to access the impact of anthropogenic activities on air quality. PM2.5 still contains important contributions from sea spray and dust natural emissions (of the order of 25%) but its main source contributions are from anthropogenic activities associated with thermal energy production and road traffic.

Similar speciation studies should be done in the samples of PM2.5 dealt with in this work.

ACKNOWLEDGEMENT

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Indoor Thermal Environment in a Classroom Equipped with Air Forced System

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SUMMARY

In this work the evaluation of the indoor thermal environment in a classroom equipped with air forced system will be made. In the classrooms' indoor thermal environment, with the air forced inlet in the door and the outlet located above the windows, two situations are analyzed: the inlet forced airflow made in the doors' lower and upper area.
This experimental study, made in a school building located in the South of Portugal, in Mediterranean climate, evaluates the thermal comfort (using the PMV index), local thermal discomfort (using the draught risk and uncomfortable air velocity fluctuations) and air quality levels (using the air renovation rate) that occupants are subjected.
In the thermal comfort level the air temperature, velocity and relative humidity and the radiative temperature, in the local thermal discomfort the air temperature and velocity fluctuations in different classroom interior points, while in the air quality level, using the decay tracer gases concentration, the carbon dioxide carbon concentration, are measured.

INTRODUCTION

The present philosophy of ventilation in the school building classrooms in the Algarve Region, in the South of Portugal, is the natural and/or crossed. The general idea is associated to the air inlet above the windows area, from the external environment, and the outlet above the door area, to the corridor, nevertheless, depending on the wind direction the airflow can also inlet above the door area, from the corridor, and the outlet above the windows area, to the external environment. In the absence of wind the occupants in the classroom frequently use the door and the windows open to increase the air quality level.
In order to obtain good thermal and air quality in indoor environments, used as educational spaces for young people, the inlet and outlet of clean air with acceptable temperature levels, through strategically placed points are necessary. The clean air when entering spaces should proportionate a pleasant micro-climate around the occupants, assure good air quality in the breathing area and extract the contaminants released by the occupants. Nevertheless, this ventilation system can proportionate local thermal discomforts.
In order to evaluate the thermal comfort level, in moderate environments equipped with air-conditioning systems, the PMV (Predicted Mean Vote) and the PPD (Predicted Percentage of Dissatisfied) indexes, developed in [1] are used. The PMV index is given as a value on the seven-point comfort scale and is based in four environmental parameters (air mean temperature, velocity and relative humidity and radiant mean temperature) and two personal factors (clothing and metabolic activity level). In accord to the main stream of the PMV and PPD indexes, the thermal neutrality of an individual is obtained when the body heat loss is equal to the body metabolic heat (PMV=0).

The extension of the PMV model was developed in [2] to be used in non-air-conditioned buildings in warm climates. This extension, used in warm environments, combines the “static” PMV model and the adaptive model. The idea is to use the traditional PMV model, that considers the human body thermal balance, and the expectations verified in the adaptive model. The extension of the PMV model, to be used in people non-subjected to air-conditioned environment, is based in an expectancy factor that will be multiplied by the “static” PMV value (mean thermal sensation vote). The idea is that people in non-air-conditioning environments in warm climates perceived the environment in a less severe way than the “static” PMV people.
Local thermal discomfort sensations in localized regions of the body may occur by incident airflows from ventilators, due to their intrinsic characteristics such as exit air velocity, airflow symmetry and their location in relation to the occupants. In accord to [3] the local thermal discomfort depends on the local air temperature, velocity and turbulence intensity.
According to [4] the local discomfort sensations, associated to the air velocity frequency fluctuations, are verified in frequencies between 0.3 and 0.5 Hz and according to [5] and [6], are verified to equivalent frequencies between 0.2 and 0.6 Hz.
The airflow rate inside an occupied compartment can be calculated using different recommendations and methodologies presented in national and international standards (also see [7]). Actually, for example, the ANSI/ASHRAE Standard [8] and the Portuguese standard [9] presented some airflow rates recommended values per person, while in [10] are presented some airflow rates based in the occupants comfort levels. In the last normalization is also necessary to consider the pollution load caused by the building itself (including furnishing, carpets and ventilation systems).
The ventilation topology used in the last works (see [11]), that will be used in the school buildings in this region, considers the air entering through the doors (in the first floor), passing through the corridors and atria (in the first and second floors), entering through the doors grids and leaving through the windows extraction fans to the external environment. In this ventilation philosophy the extraction fans in spaces with a long occupation period (classrooms, offices, library, auditorium, students’ and teachers’ rooms) are installed and in spaces with a short occupation period (corridors and atria), where the air is passing to the firsts spaces, will not be installed.
In the present study, using experimental means in real classrooms, in Summer conditions, some experimental tests using extractors, located in the door as well in the windows upper area, in order to increase the air quality level, reduce the local discomfort and increase the thermal comfort levels, will be made. In the present work some experimental results obtained in a classroom, equipped with extractors located in the lower and upper door area, will be presented. The idea is to guarantee a continuum airflow from the corridor to the classroom and after to the external environment, to make easier the indoors pollutants dispersal of the school building, instead of the present systems depending on the wind.

EXPERIMENTAL SETUP

The experimental tests were made in a real classroom with a floor area of 10.8×4.8 m^2 and with a height of 3.5 m (see figure 1). This classroom, inside a school building located in the South of Portugal, had three windows equipped with above dump trucks and one door equipped with air extractors placed in the door’s lower and upper area. The airflow comes into the corridor area, enters in the classroom through the extractors, upper or lower, and comes out to the external environment through the dump trucks located above the window

(also see figure 1). In each one of the experimental tests are calculated, sequentially and respectively, the global thermal comfort, local thermal discomfort and air quality levels.
In these experimental tests, being the objective to evaluate the indoor thermal environment in a classroom equipped with an air forced system, two situations are analyzed: the inlet forced airflow is made in the doors' lower area, while the second one is made in the doors' upper area. This experimental study evaluates the thermal comfort (using the PMV index), local thermal discomfort (using the draught risk and uncomfortable air velocity fluctuations) and air quality levels (using the air renovation rate) that occupants are subjected.
The air velocity and temperature fields are experimentally measured, in the occupied area, in 3 levels (0.5 m, 1.5 m and 2.5 m above the floor level) in a grid of 3×5 points (see figure 1). The other variables, like carbon dioxide concentration and air relative humidity, are measured in the classroom's central area 1.2 m above the floor level. The experimental tests were made in Summer conditions. In these measurements the following orthogonal axis directions were defined:

- The X direction, defined in the transversal classroom direction, with the origin in the door side;
- The Y direction, defined in the longitudinal classroom direction, with the origin in the blackboard side;
- The Z direction, defined in the height classroom direction, with the origin in the floor level.

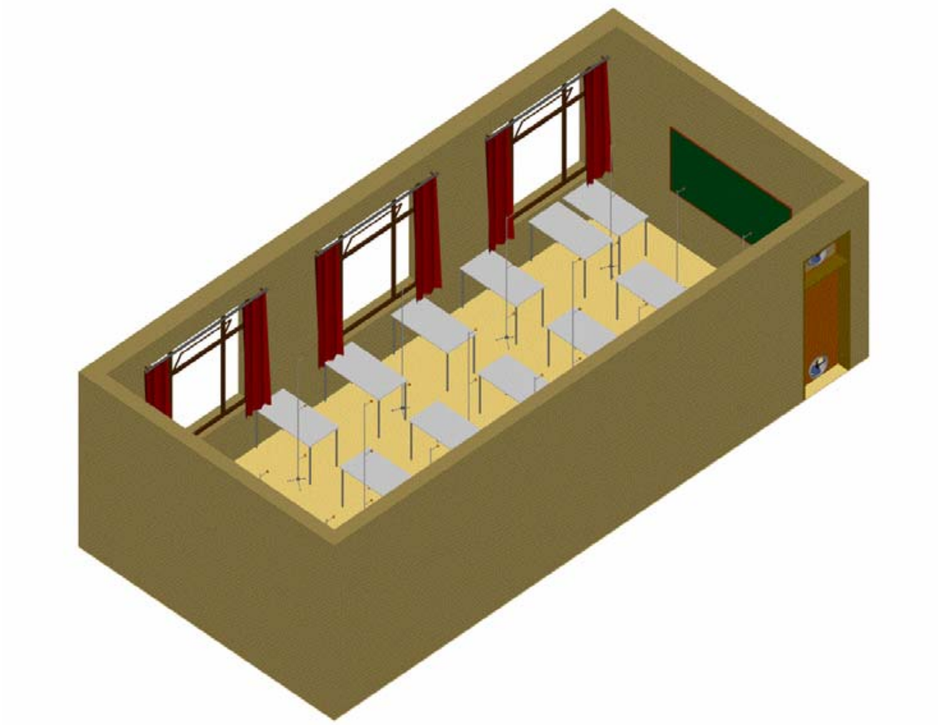


Figure 1. Scheme of the classroom used in the experimental setup. Location of the door, windows, dump trucks, extractors, desks and measuring points.

The air temperatures and velocities evolution are measured with three omni-directional probes, from Sensor, during 5 minutes at a rate of 6 samples per second, while the mean values of other environmental variables and the carbon dioxide concentration, during all test at a rate of 2 samples per minute, are measured by using Babuc-A, of a LSI.
The mean air temperature, velocity, relative humidity and the radiative temperature, with a pre-defined clothing and activity levels, are used to calculate the local thermal comfort level.

The measured air velocity and temperature fluctuations values are used in the calculation of the air velocity root mean square, the air turbulence intensity, the draught risk, the air velocity fluctuations frequencies and the air velocity fluctuations equivalent frequencies.
The carbon concentration decay, using the decay tracer gases method, is used to calculate the local air exchange rates.

RESULTS AND DISCUSSION

In this work, the evaluation of the indoor thermal environment in a classroom equipped with air forced system was made. This experimental study, made in a warm September day, evaluates the thermal comfort, local thermal discomfort and air quality levels that occupants are subjected.

Global thermal comfort level

In order to evaluate the global thermal comfort level, using the PMV index, in the environmental parameters are considered the air temperature (mean value of all measured data in the occupied area), the mean radiant temperature (equal to the air temperature), the air relative humidity and the air velocity value (mean value of all measured data in the occupied area), while in the personal parameters the clothing level (0.5 Clo as Summer clothing) and the activity level (1.2 Met) are considered. In the first tests, when the ventilator is placed in the door's lower area, the air mean temperature, mean radiant temperature, air mean relative humidity and air mean velocity are, respectively, 32,55 ºC, 32,55 ºC, 28 % and 0,21 m/s, while in the second test, when the ventilator is placed in the door's upper area, these values are 31.83 ºC, 31.83 ºC, 39 % and 0.25 m/s. The Predicted Percentage of Dissatisfied people (PPD), using the Fanger model (see [1]) with the correction presented for warm environments (see [2]), are in the first situation 28.98 % (PMV=1.07) and in the second situation 25.99 % (PMV=0.99). In accord to [10], the internal conditions are thermally uncomfortable.

Local thermal discomfort level

In order to evaluate the local thermal discomfort level the draught risk and uncomfortable air velocity fluctuations, that occupants are subjected, are considered. In the first one the percentage of dissatisfied people by draught risk is considered, while in the second one the air velocity fluctuation frequencies and the air velocity fluctuations equivalent frequencies are considered.
The measured air velocity fluctuations are used to calculate the air mean velocity, root mean square values and, consequently, turbulence intensity. This information, in combination with the local air temperature value, is used to calculate the draught risk (see [3]).
In tables 1 and 2 are presented, for the different locations, the air mean velocity (V_{air}), air mean temperature (T_{air}), air turbulent intensity (TI), draught risk (DR) and equivalent frequency (E_f) measured values. The results presented in the first table are associated to the air forced system localized in the door's lower area, while in the second table are associated to the air forced system localized in the door's upper area. The values presented in bold in these tables are associated to the uncomfortable situation, in accord to the present standards.
The air velocity fluctuation frequency, that can cause local discomfort sensation associated to the air velocity fluctuations in the occupants, in the occupied area, are presented in figures 2a and 2b. The results presented in the first figure are associated to the air forced system

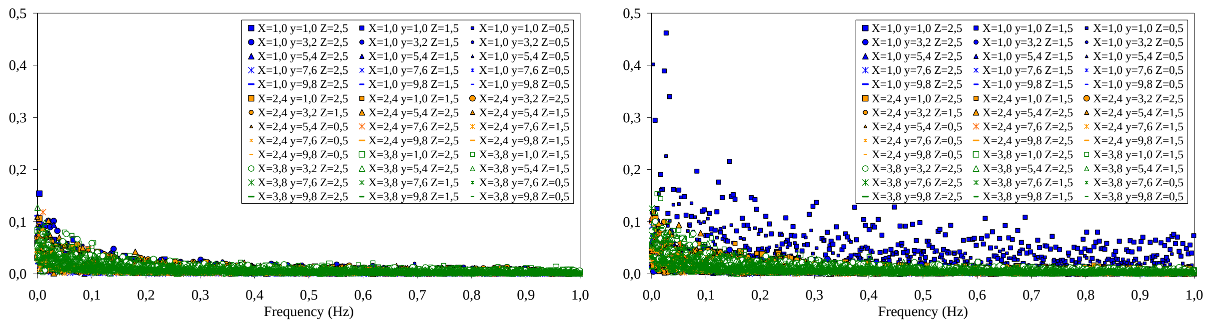
localized in the door's lower area, while in the second figure are associated to the air forced system localized in the door's upper area.

Table 1. Measured and calculated values, obtained in the experimental test with inlet forced air located in the door's lower area and with the outlet air located above the windows' area

Z=0,5	X=1,0					X=2,40					X=3,8				
	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8
v_{air} (m/s)	0,28	0,26	0,17	0,15	0,17	0,20	0,23	0,18	0,15	0,18	0,40	0,30	0,24	0,24	0,17
T_{air} (ºC)	31,62	31,83	32,12	33,38	33,06	31,87	31,88	32,67	33,53	33,54	31,70	31,76	33,07	32,55	35,23
TI (%)	14,66	21,68	20,97	15,11	19,78	22,28	25,67	21,11	14,68	24,57	33,96	33,14	23,61	21,94	21,28
DR (%)	4,37	4,33	2,23	0,59	1,11	3,17	3,91	1,68	0,43	0,62	9,64	6,50	1,76	2,58	0,00
E_f (Hz)	**0,48**	**0,46**	**0,22**	**0,26**	**0,26**	**0,44**	**0,49**	**0,29**	**0,27**	**0,30**	0,60	**0,47**	**0,53**	**0,37**	**0,23**

Z=1,5	X=1,0					X=2,40					X=3,8				
	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8
v_{air} (m/s)	0,23	0,24	0,18	0,16	0,19	0,20	0,21	0,19	0,15	0,17	0,24	0,27	0,23	0,18	0,20
T_{air} (ºC)	31,53	31,66	32,04	33,22	33,02	31,78	31,83	32,44	33,14	33,54	31,80	31,71	33,09	32,74	35,05
TI (%)	20,13	20,17	21,36	16,32	16,20	24,43	18,95	25,00	13,19	20,68	27,10	25,39	23,79	20,56	17,28
DR (%)	4,17	4,10	2,53	0,80	1,21	3,40	3,17	2,30	0,81	0,54	4,26	5,06	1,64	1,56	0,00
E_f (Hz)	**0,36**	**0,39**	**0,31**	**0,21**	**0,35**	**0,33**	**0,43**	**0,27**	**0,29**	**0,22**	**0,47**	**0,43**	**0,34**	**0,28**	**0,25**

Z=2,5	X=1,0					X=2,40					X=3,8				
	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8
v_{air} (m/s)	0,20	0,25	0,19	0,14	0,19	0,21	0,24	0,24	0,17	0,15	0,20	0,21	0,18	0,17	0,17
T_{air} (ºC)	31,62	31,63	31,96	33,16	33,07	31,75	31,65	32,07	32,93	33,45	31,82	31,77	32,93	32,69	33,73
TI (%)	31,88	29,00	29,07	14,12	26,38	27,32	23,61	27,52	28,40	19,68	25,77	27,43	29,74	22,43	17,24
DR (%)	3,92	5,03	3,09	0,72	1,33	3,77	4,39	3,77	1,44	0,57	3,34	3,84	1,56	1,62	0,30
E_f (Hz)	**0,30**	**0,35**	**0,30**	**0,25**	**0,27**	**0,35**	**0,40**	**0,45**	**0,30**	**0,30**	**0,45**	**0,34**	**0,34**	**0,42**	**0,29**



a) b)

Figure 2. Air velocity fluctuation frequencies, obtained in the experimental test with inlet forced air located in the door's lower a) and upper b) area and with the outlet air located above the window area.

In accord to the previous results is possible to conclude:

- The draught risk, in general in the two studied situations, is higher in the low body sections than the other bodies sections. Nevertheless, in general, all points are in accord to the Category A of [10];
- The highest draught risks are verified mainly in the desks located near the door, being inclusively in the second situation in the head level verified uncomfortable draught risks;

- The draught risk, in general, when the ventilator is placed in the door’s upper area are higher than when the ventilator is placed in the door’s lower area;
- The values verified in the air velocity equivalent frequencies are inside the uncomfortable interval;
- When the ventilator is placed in the door’s lower area the highest uncomfortable air velocity fluctuations are located near the lower members level, in the desks placed in front to the window and near the blackboard;
- When the ventilator is placed in the door’s upper area the highest uncomfortable air velocity fluctuations are located near the head level, in the desks placed in front to the door and near to the blackboard. Nevertheless, the air velocity fluctuations frequencies verified in this second test are more energetic than the verified in the first test.

Table 2. Measured and calculated values, obtained in the experimental test with inlet forced air located in the door’s upper area and with the outlet air located above the windows’ area.

Z=0,5	X=1,0					X=2,40					X=3,8				
	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8
v_{air} (m/s)	0,41	0,17	0,15	0,19	0,13	0,50	0,27	0,17	0,17	0,14	0,37	0,46	0,33	0,20	0,14
T_{air} (ºC)	31,12	31,78	32,26	31,97	32,08	31,10	32,04	32,21	31,96	32,24	31,37	31,89	32,04	31,97	32,91
TI (%)	45,50	20,63	15,87	22,09	13,45	36,62	30,63	24,12	20,21	14,86	40,78	20,40	22,75	20,31	13,47
DR (%)	15,51	2,69	1,71	2,75	1,55	**17,36**	4,72	2,22	2,45	1,56	11,43	7,92	5,32	2,86	0,91
E_f (Hz)	**0,50**	**0,35**	**0,30**	**0,25**	**0,29**	**0,58**	**0,49**	**0,30**	**0,22**	**0,28**	**0,55**	0,61	**0,43**	**0,40**	**0,24**

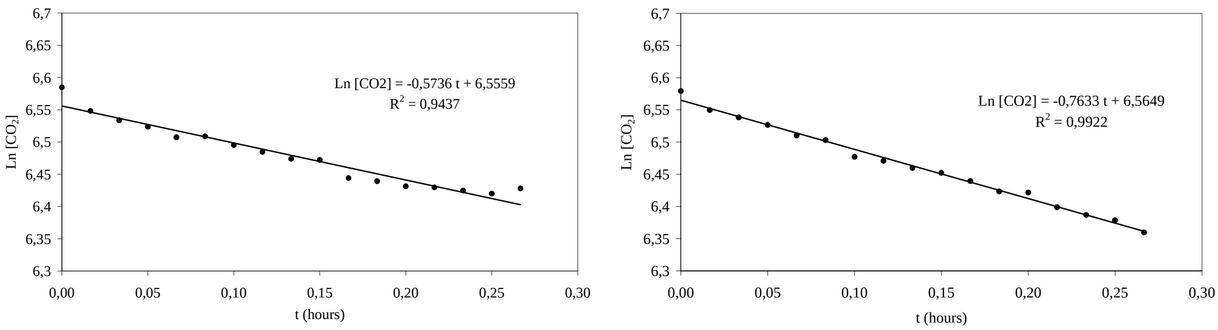
Z=1,5	X=1,0					X=2,40					X=3,8				
	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8
v_{air} (m/s)	0,89	0,24	0,18	0,19	0,16	0,25	0,29	0,20	0,18	0,17	0,24	0,27	0,30	0,18	0,16
T_{air} (ºC)	30,85	31,72	32,10	31,85	31,73	31,09	31,84	31,90	31,76	32,08	31,67	32,06	31,93	31,66	32,74
TI (%)	45,35	20,55	18,85	17,53	14,05	29,84	16,50	20,22	20,45	25,54	36,49	28,66	22,92	19,80	16,75
DR (%)	**51,32**	4,12	2,35	2,79	2,23	6,19	4,30	3,08	2,88	2,40	5,43	4,64	5,00	2,95	1,28
E_f (Hz)	0,66	**0,34**	**0,30**	**0,28**	**0,28**	**0,41**	**0,38**	**0,33**	**0,30**	**0,22**	**0,33**	**0,40**	**0,39**	**0,29**	**0,25**

Z=2,5	X=1,0					X=2,40					X=3,8				
	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8	Y=1	Y=3,2	Y=5,4	Y=7,6	Y=9,8
v_{air} (m/s)	0,34	0,29	0,21	0,20	0,15	0,22	0,22	0,24	0,20	0,15	0,23	0,28	0,21	0,21	0,18
T_{air} (ºC)	31,05	31,57	31,94	31,81	31,76	31,16	31,98	31,76	31,67	32,09	31,61	32,10	31,89	31,54	32,52
TI (%)	17,51	21,41	24,81	23,57	18,44	25,31	26,72	28,02	26,08	20,20	27,46	26,69	28,44	26,46	23,39
DR (%)	7,22	5,35	3,38	3,24	2,14	4,98	3,64	4,58	3,68	1,98	4,41	4,39	3,53	4,12	1,93
E_f (Hz)	**0,51**	**0,46**	**0,32**	**0,28**	**0,22**	**0,47**	**0,39**	**0,41**	**0,27**	**0,26**	**0,41**	**0,45**	**0,37**	**0,38**	**0,25**

Air quality level

The air quality level, evaluated by local classroom air renovation rate, are obtained in the central room area, 1.2 m above the floor level, in the respiration area, using the decay tracer gases method. In these experimental tests, before the beginning of the test, the carbon dioxide and the air were mixed using two mixing ventilators. In the experimental test, without mixing ventilators on, the local tracer gas is measured around 0.3 hours. In figure 3a) and 3b) are presented the carbon dioxide concentration logarithm decay in function to the time, in the central room point, respectively, when the ventilator is placed in the door’s lower and upper area.

In general the classroom, with a volume of 181.4 m^3, is frequently used by 24 students and one teacher, which are 25 occupants. For a classroom the ASHRAE Standard [8] recommend an air flow rate per person of 5 l/s (using outdoor air rate) and 6.7 l/s (using combined outdoor air rate), while the Portuguese Standard [9] recommend 30 m^3/h. Thus, the air renovation flow rate for the ASHRAE Standard [8] changes between 450 and 603 m^3/h, while for the Portuguese Standard [9] is 750 m^3/h.



a) b)

Figure 3. Evolution of carbon dioxide concentration decay, obtained in the experimental test with inlet forced air located in the door's lower a) and upper b) area and with the outlet air located above the window area.

In accord to the previous results is possible to conclude:

- The carbon dioxide concentration logarithm evolution, in accord to figures 3a) and 3b), shows as example a perfect logarithm decreasing. The experimental air renovation rates change between 0.57 h^{-1} when the ventilator is placed in the door's lower area and 0.76 h^{-1} when the ventilator is placed in the door's upper area;
- The local air renovation flow rate in the central respiration area changes between 103.4 m^3/h when the ventilator is placed in the door's lower area and 137.8 m^3/h when the ventilator is placed in the door's upper area;
- The recommendations defined by the ASHRAE Standard [8] and by the Portuguese Standard [9] are not guaranteed using this kind of ventilation system in this space.

CONCLUSIONS

In this work the evaluation of the indoor thermal environment in a classroom equipped with air forced system was made. This experimental study, made in a warm September day, evaluates the thermal comfort, local thermal discomfort and air quality levels that occupants are subjected, with an air forced system placed in the door's upper or lower area.

The experimental test was made in Summer conditions in a warm day using the inlet air flow coming from the corridor compartment. The corridor inlet air temperature, lower than the outdoor environment, is insufficient to guarantee comfortable conditions. Nevertheless, in general in this kind of warm days the students are in holidays.

The draught risk is highest in the occupied area near the door, mainly when the ventilator is placed in the door's upper area. The equivalent air velocity fluctuation frequencies values are inside the uncomfortable interval, nevertheless the more uncomfortable air velocity fluctuations are verified near the lower members level in the desks placed in front to the window near the blackboard, when the ventilator is placed in the door's lower area, and near the head level in the desks placed in front to the door near the blackboard, when the ventilator is placed in the door's upper area.

The local airflow rate is lightly lower when the ventilator is placed in the door's lower area than when the ventilator is placed in the door's upper area. Nevertheless, in all situations the air renovation flow rate is insufficient.

In order to increase the air renovation rate, in future works, to use more than one extractor fan and to put them in the windows upper area instead of the door area are recommended. This fact reduces substantially the influence of external wind direction, which was verified in the present work. Nevertheless the increase of the airflow rate, mainly for lowest air temperature levels, can promote local thermal discomforts levels, associated to the draught risk, mainly in desks located near the blackboard.

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A study on the effect of the airflow rate of the ceiling type air-conditioner on the ventilation performance

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SUMMARY

We performed a study on the effect of the discharge airflow rate of the ceiling type air-conditioner on ventilation performance in the lecture room with the mixing ventilation. The experiments and CFD were conducted for analyzing ventilation performance. The concept of mean air age and indoor CO_2 concentration were used for evaluating ventilation performance. We made the CO_2 generation model in the simulation and calculated a lot of cases with respect to the airflow rate of air conditioner and the ventilation flow rate. And the selected experimental measurements were performed in the lecture room of the same layout as the numerical one for verifying simulation results. Mean air age is gradually increased, but CO_2 concentration is oppositely decreased in the occupied zone with the increment of the discharge airflow rate of the ceiling type air-conditioner. This result shows that both of mean air age and residual life time must be considered for evaluating ventilation performance when the contaminants are generated indoors. And the increment of discharge airflow of the ceiling type air-conditioner can induce the piston effect and push the contaminants out of the occupied zone. From this result, it is found out that that ventilation performance can be increased when the momentum source like an air-conditioner is used in the room with the mixing ventilation.

INTRODUCTION

Recent studies [1-4] showed that many buildings have poor indoor air quality, and especially CO_2 concentrations measured in the room exceeded the present guideline [5]. In this case, the role of ventilation is very important as the strategy for more comfortably indoor environment because CO_2 cannot be removed by present air cleaning systems. Accordingly, various researches on the improvement of ventilation performance as well as ventilation itself have been carried out. In the view of IAQ, Lee and Chang [2] investigated the indoor and outdoor air quality of the five classrooms at five different schools in Hong Kong and pointed that PM_{10} and CO_2 concentration exceeded the HKIAQ limits due to the high outdoor PM_{10} concentration and inadequate ventilation respectively. Daisey et al. [3] reviewed the literature on IAQ, ventilation and health symptoms in schools and showed that ventilation was inadequate in many classrooms, possibly leading to health problems. Noh and Oh [4] performed the numerical study on the comparison of ventilation performance with variations of indoor momentum source such as air-conditioners, air cleaners and ceiling fan for displacement and mixing ventilations. They found out that ventilation performance is getting better or worse with variations of the intensity and the location of momentum source due to that the intensity and the location of momentum source affect the residence time of air in the room.

However, there are hardly ever studies on effect of the air discharge intensity of the personal air-conditioning systems such as indoor air-conditioners, air cleaners, and fans in school buildings on the ventilation effectiveness until now. Therefore we performed the experiments and numerical simulations in a lecture room with a ceiling type air-conditioner and a ventilation system. With variations of the air discharge rate of the ceiling type air-conditioner and occupancy, CO_2 concentrations and local mean air-ages during class hours were investigated and the effect of the air discharge intensity of a ceiling type air-conditioner on ventilation effectiveness was analyzed.

RESEARCH METHODS

Model descriptions

Fig. 1 shows the schematic design of a real lecture room which is presently used in one of universities in Korea. The dimensions of this room are 11.2m (L) x 6.65m (D) x 2.4m (H). This lecture room has 3 windows and 2 doors. The lecture room was generally occupied by 20 to 30 people including a lecturer and students when the experiments were carried out. In order to control the thermal loads in the lecture room, the ceiling type 4-way cassette air-conditioner. It is located at the center of this room. The discharge airflow of this air-conditioner is discretely varied from 0m^3/hour to 1600 m^3/hour, while the discharge angle is fixed to 30 degrees when the experiments were conducted. The ventilation system is composed of 4 supply diffusers, 4 exhaust ones, and a heat exchanger. Locations of supply and exhaust diffusers are determined to optimize the ventilation effectiveness.

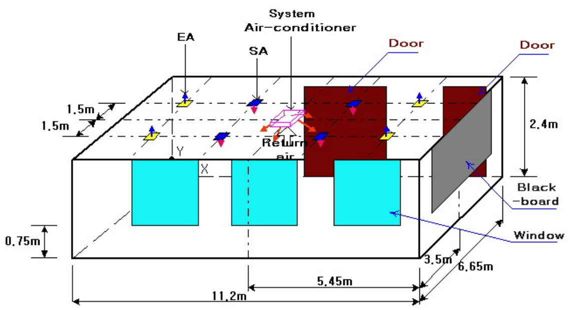


Fig. 1 Layout of the lecture room

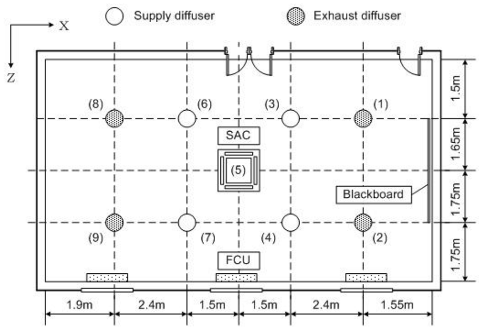


Fig. 2 Measuring points of the lecture room

Experiments

We measured CO_2 concentration and mean air age in the lecture room with respect to the discharge airflow rate of the ceiling type 4-way cassette air-conditioner. The discharge airflow rate was varied from 0 to 1600m^3/hour and ventilation rate was fixed to be 800m^3/hour when experiments were carried out. Based on the previous study [1], the discharge angle of the ceiling type 4-way cassette air-conditioner is fixed to be 30 degrees. As shown in Fig. 2, 9 measuring points were selected to analyze the local or the room-averaged performance for ventilation effectiveness. Sampling probes were placed at 1.1m high from the bottom. The experiments for each case were repeated over three times in the same case. All the measured values of indoor CO_2 were compensated according to the outdoor CO_2 variation. In the case of CO_2 concentrations, all experiments were conducted during the class hours and 20 people were asked not to move around. To examine the mean air age, all experiments were conducted after the class hours. SF_6 gases were released in the supply duct and were sampled at the same points like the case of CO_2 measurement. IAQ monitor (Graywolf, model IAQ-410; USA), Thermal comfort data logger (Innova, model 1221; Denmark), The multi gas monitor (Innova, model 1314; Denmark) were respectively used for CO_2 measurement, velocity and air temperature, and SF_6 measurement

CFD simulations

We adopted this numerical model with appropriate boundary conditions as a prediction tool for a couple of operating conditions. We considered the discharge airflow rate of 4-way cassette air-conditioner and the number of people as the control variables. All the occupants with hexagonal shapes sitting on were modelled as the rectangular objects with 1.2m height and the volume of $1.62m^3$. The CO_2 concentration emitted from occupants' mouth was selected to be $0.014m^3$/hour [5].

The flow in the lecture room was assumed to be 3D steady and incompressible [4]. The standard k-ε turbulence model was applied and the airflow transport can be described by the following time-averaged Navier-Stokes equation:

$$div(\rho \mathbf{V}\Phi - \Gamma_{\Phi,eff}\, grad\ \Phi) = S_\Phi \tag{1}$$

Where ρ is the air density (kg/m³), $\Gamma_{\Phi,eff}$ is the effective diffusion coefficient (kg/m·s), V is the air velocity vectors (m/s), S is the source term of the general flow property, and Φ is any one of the components shown in Table 1. When Φ=1, the general equation becomes the continuity equation. The effective diffusion coefficients and the source terms for different Φ are listed in Table 1. Also, Wall boundary conditions were treated as shown in Table 2.

Table 1 Terms, coefficients and constants in Eq. (1)

Equations	Φ	$\Gamma_{\varphi,eff}$	S_φ
Continuity	*1*	*0*	*0*
Momentum	U_i	μ_{eff}	$-\partial P/\partial x_i + g_i(\rho-\rho_0)$
Turbulence kinetic energy	k	μ_{eff}/σ_k	$P_k - \rho\varepsilon + G_k$
Turbulence kinetic energy dissipation rate	ε	$\mu_{eff}/\sigma_\varepsilon$	$\varepsilon(C_1P_k - C_2\varepsilon)/k + C_3G_k\varepsilon/k$
Concentration	C	μ_{eff}/σ_C	S_C

$P_k=\mu_t(U_{i,j} + U_{j,i})U_{i,j}$, $\mu_{eff} =\mu_t +\mu$, $\mu_t = C_\mu\rho k^2/\varepsilon$,
$(\sigma_k, \sigma_\varepsilon, \sigma_C, C_1, C_2, C_3) = (1.0, 1.314, 1.0, 1.44, 1.92, 1.0, 0.09)$

Table 2 Boundary Conditions for the numerical calculation

Inlet: $k_{in} = \frac{3}{2}(u_{in}\cdot I)^2,\ \varepsilon_{in} = C_\mu^{3/4}k^{3/2}l,\ I = 0.1,\ l = 0.5\times D_h,\ C_{in} = 500\,ppm$

Outlet: $\frac{\partial u}{\partial x} = 0,\ \frac{\partial k}{\partial x} = 0,\ \frac{\partial \varepsilon}{\partial x} = 0,\ \frac{\partial T}{\partial x} = 0,\ \frac{\partial C}{\partial x} = 0$

Wall: $u = v = w = 0,\ \frac{\partial k}{\partial x} = \frac{\partial k}{\partial y} = \frac{\partial k}{\partial z} = 0,\ \frac{\partial C}{\partial x} = \frac{\partial C}{\partial y} = \frac{\partial C}{\partial z} = 0$

Where, D_h is the width of inlet

The mean air age and boundary conditions were calculated by the following equation induced from equation (2) and (3) [6]:

$$div(\rho \mathbf{V} \tau - \Gamma_\tau \ grad \ \tau) = 1 \tag{2}$$

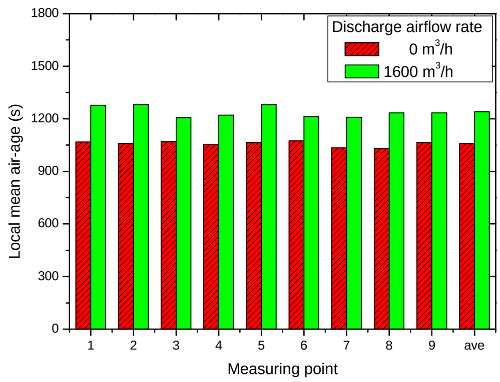
$$inlet: \tau = 0, \ \ outlet: \frac{\partial \tau}{\partial x_i} = 0 \tag{3}$$

Where, τ is the mean air age (s). x_i represent the 3D coordinates. The governing equations were solved by SIMPLE algorithm and 2nd order upwind scheme was used to discretize the convection term in equation (1). STAR-CD was employed to compute the airflow, temperature, and concentration distributions. The number of cells for numerical calculation is about 750,000 and the standard log-wall functions of Launder and Spalding were adopted for the next grid points to the surface.

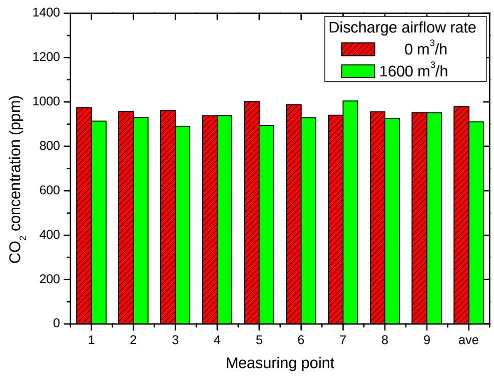
RESULTS

Experiments

Fig. 3(a) shows mean air age distribution for two different discharge airflows of 4-way cassette air-conditioner. When the 4-way cassette air-conditioner was not in operation, mean air age was measured in the range of 1030~1075 seconds. When the 4-way cassette air-conditioner was operated at the discharge airflow rate of 1600m^3/hour, the mean air ages were in the range of 1200~1280 seconds. These results shows that when the discharge airflow rate of the ceiling type air-conditioner, which is one of the indoor momentum sources, is increased, mean air age can be also increased. The reason is that the operation of the ceiling type air conditioner intervene the air current and makes the outdoor fresh air move a longer way. Therefore this indicates that the indoor air quality may be deteriorated as the discharge airflow is increased.



(a) mean air age



(b) CO_2 concentration

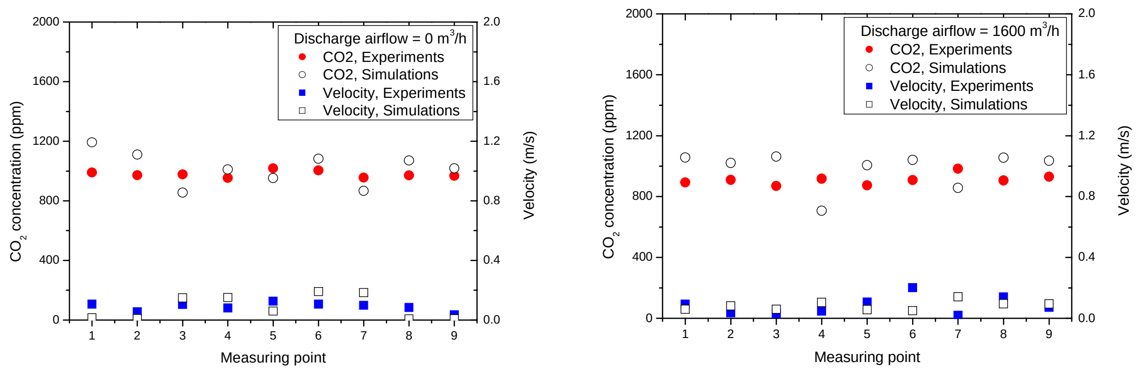
Fig. 3 Experimental results for mean air age and CO_2 concentration at the measuring points with the different discharge airflow of 4-way cassette air-conditioner

Fig. 3(b) shows the locally-averaged CO_2 distributions, which were measured at y=1.1m, with two different discharge airflow rates. When the 4-way cassette air-conditioner was not in operation, CO_2 concentrations were measured in the range of 950~1020ppm. When the 4-way cassette air-conditioner was operated at the discharge airflow rate of 1600m^3/hour, CO_2 concentrations were measured in the range of 870~980ppm. Unlike mean air age distributions as shown in Fig. 3(a), CO_2 concentration get to be somewhat decreased as the discharge airflow rate of the ceiling type air-conditioner is increased. This phenomenon had been

reported by the previous numerical and experimental studies [4]. This reason is that the operation of the ceiling type air conditioner can make the mixing effect of indoor contaminants increased and the residual life time of indoor contaminants decreased.
From these results it was revealed that both of the mean air age and the residual life time must be considered to evaluate the ventilation performance when the contaminants are generated from the indoor.

Validation of CFD model

A part of results from measurements were used for validating the following numerical calculations. Numerical simulations were designed to sufficiently meet the experimental work. Numerical simulations of flow patterns and CO_2 concentrations were compared with measurements at the section of 1.1m high above the bottom in a full-scale room.



(a) discharge airflow is $0m^3/h$ (b) discharge airflow is $1600m^3/h$

Fig. 4 Comparison of experimental results and numerical ones for CO_2 and velocity at the measuring points

Fig. 4 shows the comparison of experimental results and numerical ones for velocities and CO_2 concentrations at the measuring points when the discharge airflow rate is $0m^3/h$ and $1600m^3/h$. There were slight differences between experiments and simulations. In the case of CO_2 concentration, the locally-averaged concentrations were measured nearly the same as the room-averaged value in the experiments regardless of the discharge airflow rate. Their maximum differences of them were under 17% and the agreement was acceptable. In the case of velocity, all values were under 0.25m/s and satisfied an acceptable condition in the occupied zone. From these results we could make sure that the numerical model is sufficient to carry out the applications for evaluating the ventilation effectiveness with variations of the discharge airflow rate of the ceiling type air-conditioner.

Assessment of ventilation effectiveness

Fig. 5 shows the variation of indoor CO_2 concentration in the occupied zone with respect to the discharge airflow rate of the ceiling type air-conditioner, the occupancy and the ventilation rate. Regardless of the occupancy and ventilation rate, CO_2 concentrations is gradually decreased as the discharge airflow rate is increased from 0 to $800m^3/h$, while their values are rarely changed in the case that the discharge airflow rate is over $800m^3/h$. This result demonstrates that a minimum value exists with the increment of the discharge airflow rate. Fig. 6 shows CO_2 distributions at the plane of z=3.2m with respect to the discharge airflow rate of the ceiling type 4-way cassette air-conditioner. As the discharge airflow rate is larger, this deviation of indoor CO_2 concentration becomes smaller in the occupied zone. It is due to that increment of the discharge airflow rate will lead the indoor air to be more mixed as

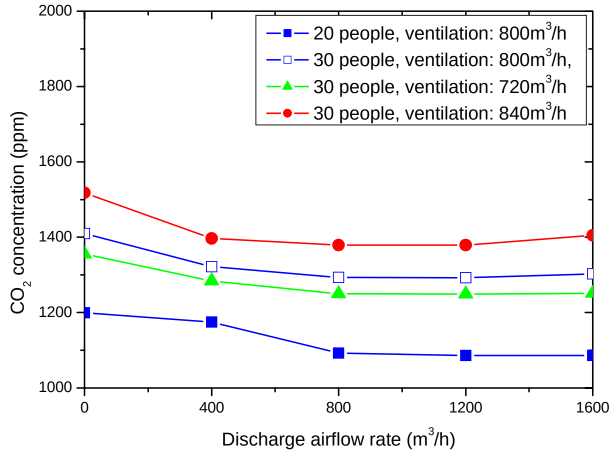


Fig. 5 The variation of indoor CO_2 concentration in the occupied zone with respect to the discharge airflow rate of the ceiling type air-conditioner, the occupancy and the ventilation rate

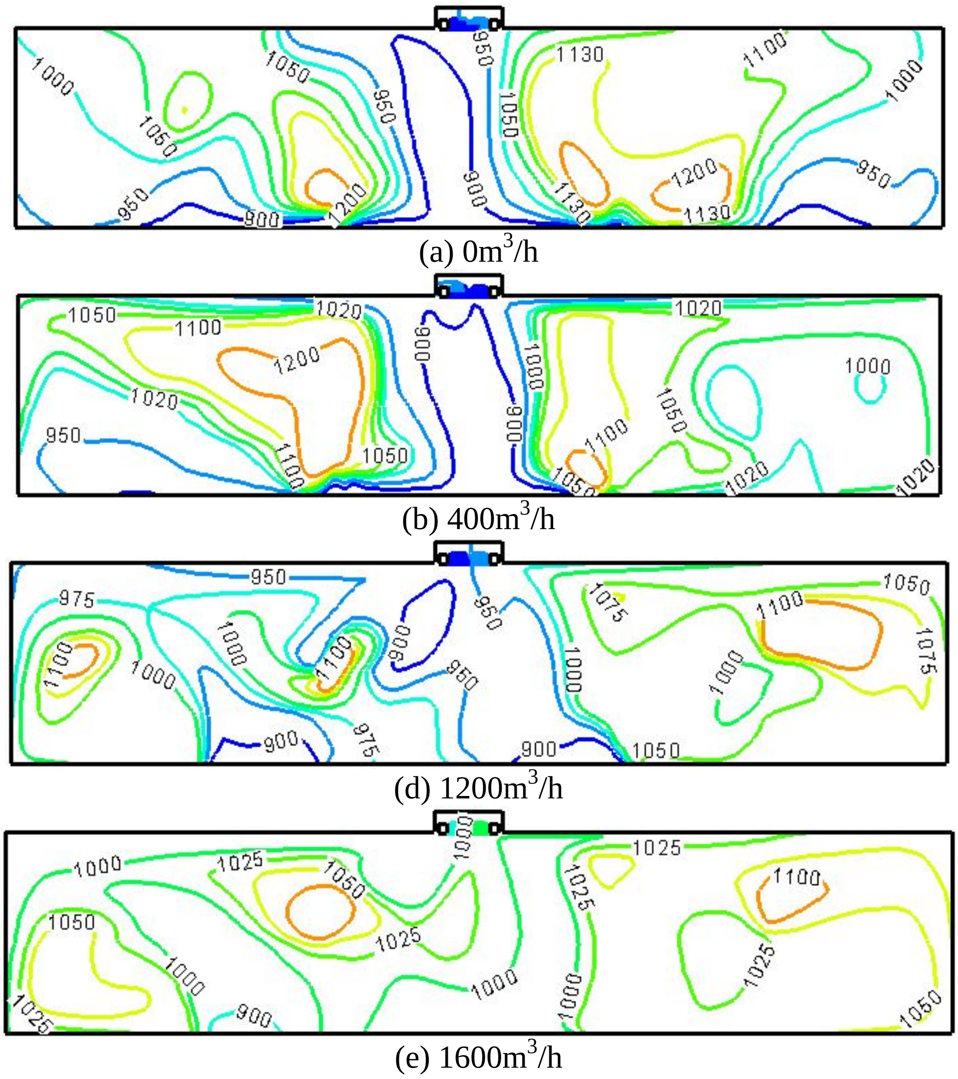


Fig. 6 CO_2 distributions at the plane of z=3.2m with respect to the discharge airflow rate of the ceiling type 4-way cassette air-conditioner mentioned above. Also, it is shown that the maximum area of indoor CO_2 concentration is gradually going up and their value is smaller as the discharge airflow rate is increased. This means that the discharge airflow of the ceiling type air-conditioner can induce the piston effect and push the contaminants out of the occupied zone. But, this effect is gradually diminished with the increment of the discharge airflow intensity as shown in Fig. 6.

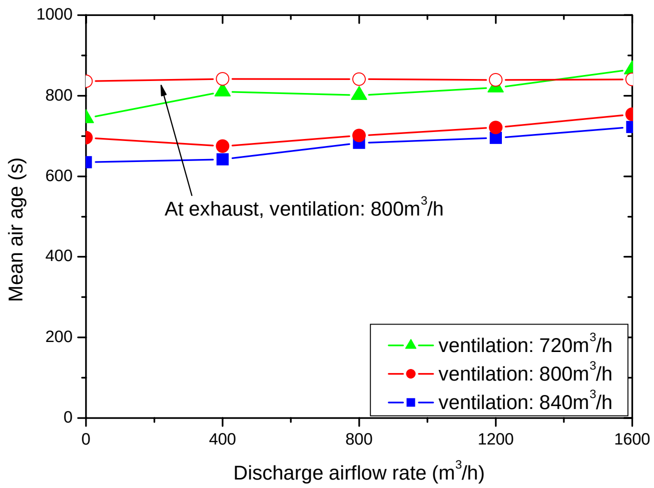


Fig. 7 The variation of mean air age in the occupied zone with respect to the discharge airflow rate of the ceiling type air-conditioner, the occupancy and the ventilation rate

Fig. 7 shows the variation of mean air age in the occupied zone and exhaust with respect to the discharge airflow rate of the ceiling type air-conditioner and the ventilation rate. Regardless of the ventilation rate, mean air age is gradually increased as the discharge airflow rate is increased. This calculation result shows qualitative similarity with the experiment. But, these numerical values of mean air age were not approximated to the experimental ones and there differences were about 200 seconds. It is due to that molecular weight of SF_6 is about 5 times heavier than one of air and the supply airflow rate was smaller than the exhaust one of $800m^3/h$. While, mean air age at exhaust is almost constant even though the discharge airflow rate is increased. This means that the residence time is not changed with the discharge airflow rate of the ceiling type air-conditioner. When the mean air age is increased and the residence time is constant like these cases, the residual life time should be decreased by the definition of air age. Therefore, the contaminant concentration like CO_2 emitted from indoor sources must be decreased with the decrement of the residual life time. These results show that the increment of mean air age does not always make indoor air quality deteriorated. Again, both of mean air age and residual life time are important parameters to be considered for evaluating ventilation performance when the contaminants are generated indoors.

CONCLUSIONS

The effect of the discharge airflow rate of the ceiling type air-conditioner on ventilation performance was studied. Mean air age is gradually increased, but CO_2 concentration is oppositely decreased in the occupied zone with the increment of the discharge airflow rate of the ceiling type air-conditioner. This result shows that both of mean air age and residual life time must be considered for evaluating ventilation performance when the contaminants are generated indoors. And the increment of discharge airflow of the ceiling type air-conditioner can induce the piston effect and push the contaminants out of the occupied zone. This shows that the ventilation performance can be increased when the momentum source like an air-conditioner, air cleaner, and fan is used in the room with mixing ventilation.

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The Influence Of "Ventilate Sill" Design As A Device To Improve Natural Ventilation Inside School Buildings In Hot And Humid Climate

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SUMMARY

This article is part of a research in progress about comparative study methods for the Brazilian reality using among many other authors, Givoni (1969), Voght and Miller-Chagas (1970), Fanger-ISO(1970), ASHRAE (55-1992), Mahoney (1971), Humphreys (1978) and Olgyay (1962) methods. This research presents the principal concept to be evaluated by the Universal Fuzzy Controlled aiming to establish a reference to determine a possible interference of the acclimatization factor to determine thermal comfort. An experimental evaluation method was elaborated through a questionnaire applied to the population of students in classroom, monitoring the environmental variables inside and outside the classroom during the years 2004 and 2005. It shows their particularities for the altitude tropical climate of the city of São Paulo - Brazil. This article describes interference of mass advance at the responses of students in classroom comparing to internal and external environment variables. The data was gotten in parallel with classroom in full activity.

Key words: thermal performance, building performance, thermal comfort, classroom thermal comfort, simulation, thermal simulation.

INTRODUCTION

This paper presents the experimental results collected in the years 2004 and 2005 about the limit condition of thermal comfort in classroom at altitude tropical climate.

The definition of thermal acceptable limits in edifications can have important implications on the thermal sensation on usuary, on project and on consumption of energy in the edification.

In order to obtain comfort of usuary in edifications there must be attention on limit conditions of thermal comfort – or the adaptation of man to climate. This adaptation is related to heat generated by digestion (produced by metabolism) and the work done by muscles, which must be dissipated in environment to keep the internal body temperature within this limits. Thus, disagreeable sensations as exaggerated heat loss by body, inequality of temperature among different parts of body, difficulty to eliminate excess heat that occur as much in function of environment condition as of acclimatization to this type of environment can be avoided.

It was observed that people living in hot regions and acclimatized to this kind of environment, preferred higher temperatures than those who came from colder climates, which justifies the necessity of different comfort standards for regions with different climatic conditions and economic development stages.

Brazil is a country formed by tropical regions and the occurrence of abrupt climatic alterations caused by air masses is very common – that is, the entrance of frontal systems which alter the perception of thermal comfort by the individual. The climate of the city of São Paulo where the data was collected can be classified as altitude humid tropical, with cold and dry winters, and hot and humid summers, having altitude of 630m, latitude of 21º 57'02"S and longitude of 47 º27' 50"W.

Groups of young attendants with homogeneous characteristics concerning physical vigour (ages), ways of dressing, level of expectations in relation to the termic environment, make up a specific universe of study. We can verify that there's a necessity of establishing specific rules and patterns to this kind of environment which consider its particularities.

The limits of human acclimatization in classrooms are related to their satisfaction with the thermal environment, which involves a certain degree of subjectivity. For that, it is necessary to verify the interference of environmental conditions in classrooms on thermal comfort, identifying the physical variables in the answer of its occupiers through questionnaires.

Afterwards this data is related with the measurements of external climate conditions; analyzing the influence of the advance of air masses in the answer on the comfort of the students in relation to the classroom by monitoring the external environmental conditions, and comparing them to the superior limits of temperature for the environment.

METHODS

The data was collected in classrooms with and without thermal conditioning. We respected the rules present in the ABNT NBR 15220, items 1 to 5 which establish the definitions, symbols and units; procedure for calculation of thermic properties of elements and components of buildings, Brazilian bioclimatic zone; measurement of the resistance and thermic conductivity by the principle of the protected hot plate and by the fluxmetric method related to the thermic performance of buildings.

We consulted the norms ASHRAE 55 -1981/1992/1995/2004, for conditions of thermic comfort to human occupation; ASHRAE 111-1988 to measurements, tests, adjustments, heat balance, ventilation and air conditioning and the norms ISO 7730(1994) to determine PMV and PPD and specification of the thermal comfort; ISO 7243 (1989) to hot environments, estimation of the thermic stress to workers; ISO (7933) 1989 to hot environments, analytical determination and interpretation of thermic stress using calculations of sweat rate. ISO (7726) 1985 to thermic environments, instruments and methods to measure physical values.

Read the article Plea 2004 Moraes, C.M.; *et al.*. and Clima 2005 Moraes, C.M.; *et al.*.

Classroom description

Room 2 of Mechanical Engineering building, without thermic conditioning, with capacity for 130 students and area of 200,68m2 composed of the 3 walls of masonry, a glass wall, ceiling with thickness of 10 cms and cold ground. The ceiling presents a lining below in double plates “eucatex” covered with glass wool of 50mm of thickness. Below this lining there are 24 lamps of 2 fluorescent lamps each and four ceiling fans.

Room “Asteróide” of Civil Engineering building with thermal conditioning and capacity for 36 students. Its area is of 60m2, L-shaped, with cold floor and structured by walls of “eucatex” with a wall of tempered glass and a partially open window. Its ceiling is 10 cm thick with a lining below in double plates of “eucatex” covered with glass wool of 50mm of thickness. Below this lining there are 23 luminaries of 2 fluorescent lamps each and an opening for air conditioning.

Classroom A1-02 located in region A of the building “Biênio” with evaporative cooling, with capacity for 100 students, and an area of 60m2 composed by concrete walls with superior openings of “vitraux” open partially, ceilings with thickness of 10 cm, ceiling composed by lining below with double plates of “eucatex” covered by glass wool with thickness of 50mm. Below this lining there are 23 luminaries for 2 fluorescent lamps each and openings for evaporative cooling. The floor presents carpet covering and the room is equipped with fixed and stuffed chairs.

The classroom S23 located in Civil Engineering building has air conditioning controlled by sensors with diffusers installed in the lining below and flagstone under with thickness of 10cm , having 3 walls constituted by dividers of “eucatex” and a glass wall. Its floor is of rubber and it is equipped with 25 working stations and personal computers for 48 students.

Equipments and measurements

Inside environmental conditions

The following instruments were used to measure the internal environmental variables (temperature, pressure and air humidity): - A datalogger, to which thermocouple type “T” and a humid and dry bulb by thermometers, were connected 1) Four thermocouple type “T” which measure the temperature of the air. Four points of the rooms were used, in the levels 0,10; 1,10 and 1,70 which are the recommended measures for evaluation of comfort for seated people. 2) Humid and dry bulb by thermometers measures the temperature of the dry and humid bulb. 3) Hot wire anemometer of manual hot wire (measures the temperature and air velocity of the internal environment, even being a low one). The revolving thermometer was used to check the measures of the anemometer. 4) Globe thermometer, used to measure the radiant average temperature.

External environment condition

The external environment was monitored by meteorological stations. It was obtained data on temperature, velocity, relative humidity and atmospheric air pressure and pollution; verifying the relation of CO_2, O_3, SO_4 between the increase in temperature and atmospheric pollution.

Based on the questionnaires for identification of thermal comfort sensation used in the studies of Fanger (1972), Humphreys (1970, 1972) and Leite (2002), and of post occupation assessment of Orstein (1992) a questionnaire divided in three parts was elaborated: 1) student profile: age, sex, height, weight, origin, nationality, course, ongoing year course, clothes description; 2) thermal conditions: temperature, preferences concerning the period; description of physiological sensations, pathologies; 3) discomfort factors: perception and preference about the movement of the air (air velocity), preference in relation to environment cooling, assessment of thermal comfort and preference concerning the location in the classroom.

The questionnaire was distributed when the classroom was in the maximum, medium and minimum limit of students per group, and they were done in periods, which did not disturb the students who attended classes during the semester. The criterion for the distribution of the questionnaires in the classrooms obeyed the verification of the weather forecast supplied by the Meteorological Station of the INPE which indicated the increase of the atmospheric temperature and its dependence in the advances of air masses or not. Later on, a comparison with data supplied by CPTEC/ DISME (INPE) which presents cross-country meteorological stations versus the time of the frontal systems to verify the periods of air mass arrival in order to observe whether they occurred on the date the questionnaire was filled in.

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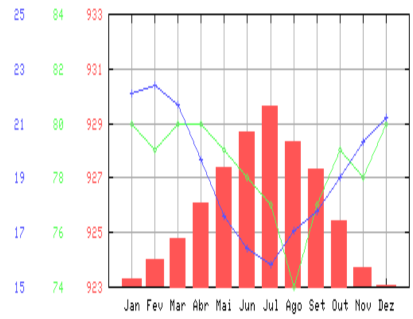
RESULTS

Climatological of the city of São Paulo

The industrialization of the city of São Paulo brought up alterations in temperature caused by pollution. Another significative factor in the alteration of the microweather of the city is the arrival of sea brise which causes violent summer storms when in contact with the hot air accumulated during the day. Figure a) below illustrates the climatological behaviour of the last 30 years in São Paulo, considering the variables of atmospheric pressure, average temperature and humidity of the air during the winter months (june, july and august). We observe that the highest pressures occur during those months while the temperature and humidity decrease.

Climatologic Graph



Pressure (hPa) São Paulo in the period 1961-1990
Humidity (%) São Paulo - period 1961-1990
Average Medium Temperature (ºC) in São Paulo - period 1961-1990

a)

Figure: a). Description of climate data of period 1961-1990.
Fonte: INMET - Instituto Nacional de Meteorologia (Data obtained on day 12 of december of 2005) (National Institute of Meteorology)

Climate Analysis of 2004 and 2005

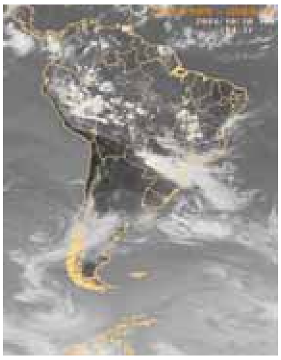
The climate analysis for the Southeast region in the months when the experiments occurred (March, April, May, August and November of 2004 and 2005) were gotten with CPTEC-INPE. (Figure 1).

In March of 2004 nucleus with positive anomalies of rain occurred and in 2005 the rains were associated to the entrance of frontal systems, on day 20 of this month occurred strong rain and hail causing danger to the local population. The occurrence of a greater number of front systems was observed in 2004 than in 2005. This indicates the intense temporal weather variation on the number of frontal systems at the same time of year. (Figure b,c).

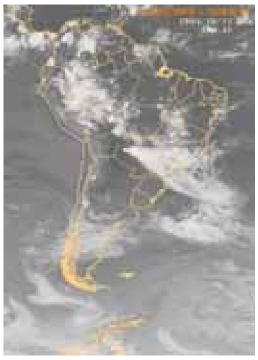
Methods of Thermal Comfort

The importance of application of thermal comfort methods in civil engineering is due to the necessity of revitalizing buildings. We notice that intervening within the possibilities in the climatic elements and in the architecture is important to enable a better adaptation of the human being to his environment in the classroom, work or home.

Figure d) summarizes the aspects of the human being as a homoeothermic organism, who reacts in order to preserve the balance when faced to variations in the temperature. In a short period of time exposed to a thermic differential, his body makes a metabolic effort through the hydrothermal regulation system, like the production of sweat or change in the breathing pace. As for a longer period, we can observe an alteration in the physiological behavior, when, for example, a person who is used to living in mild weather moves to a tropical region. Finally, when it comes to genetic adaptation, we can verify the different human phenotypes which are better acclimatized to determined thermal conditions, as skin color or amount of hair on the body.



b)



c)

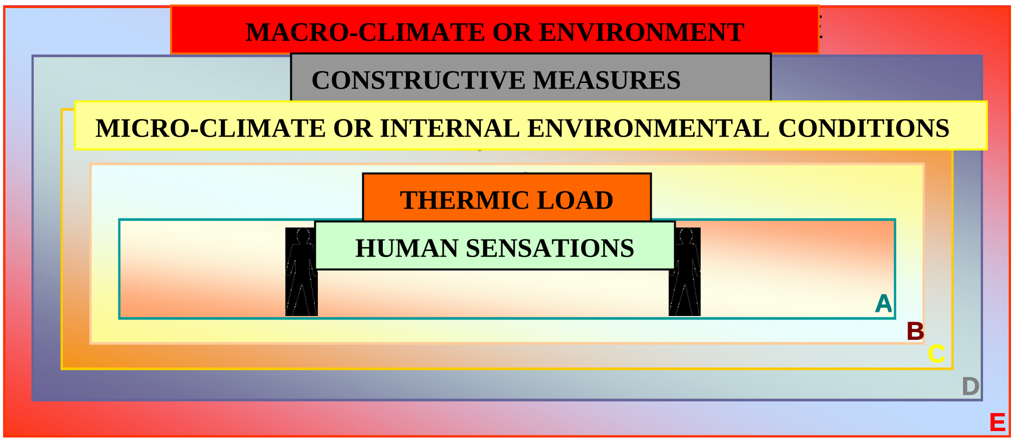
Figure a), b) Frontal System

b) October 10th. Beginning of alterations in weather conditions.

c) October 11th. Beginning of Polar mass domain.

In 1970, Fanger proposed comfort indexes from discomfort sensations.

In this sense, there is an analogy found in the methods proposed for the assessment of thermal comfort so for designed: the presence of two kinds of parameters, the personal and the environmental. Besides that, another observation may be made: all the methods present limitations either connected to the elements considered in the equations that leave aside important aspects, influencing the final result of the assessment or to the level to which a certain method may be used, as the diagram below shows:



d)

Figure d): Levels of reach diagram for the methods - effective levels

Such classification was made having as a rule the possibility of application or the objectivity of the method, that is, which practical application, within the levels above mentioned, of a certain method which determines values (results) for a certain case or construction. For example, there are methods which define values for a comfort zone to determine the best constructive conditions, then identified as “constructive measures”.

Of the mentioned methods, the present analysis takes up only the ones which concern the study of thermal control.
-In the scope of energy, can mention the noteworthy softwares: DOE-2 (1982), ARQUITOP (1990) e Ecotec (2002).
The research lines concerned with thermal comfort, like the works of: Dreyfus (1960); Olgyay & Olgyay (1962); Van Straaten (1967); Givoni (1969); Vogt & Miller-Chagas (1970); Croiset (1970); Mahoney (1973); Koenigsberger (1974); Evans (1980); Rivero (1985), which developed project guidelines, that is, focused on the evaluation of the thermal performance of the edification.

After that thermal comfort for environments was studied, that is, the evaluation of comfort sensation of the occupier, with the creation of methods and norms, that are Fanger (1972); Humphreys (1978) and the norm ASHRAE 55-1995, among other norms.

DISCUSSION

The first analysis allowed identifying and confirming what causes comfort and discomfort when the student is attending a class and it was important to test the proposed methodologies for this research. For all the rooms, the environment was monitored when empty, full or partially occupied. The experiment was valid for all the rooms, since it was possible to find out that all the students prefer cooling rather than heating, for in the winter the thermal balance can be reached by wearing heavier clothes. On the other hand, in summer, despite of wearing lighter clothes and drinking more water to control body temperature, it is hard to keep the concentration due to the heat, which indicates the insufficiency of these procedures. This was reported by the students during the year 2004 and they suggested that questions about concentration should be added to the questionnaire in 2005. Besides that, the majority of undergraduate students told that they organize their schedules in a way to have one or two free afternoons to study at their homes, in order to recover physically, creating conditions to reach their objectives since high temperatures cause a great sensation of tiredness.

Following, data about the population which occupies the monitored rooms is displayed.

The classroom in the Biênio building, with evaporative cooling, is occupied by first, second and third-year students, in the Civil, Mechanical and Environmental Engineering, totalizing 198 students in November, 2004. Their ages varied from 18 to 21 (20 in average), average weight was 60kg, average height 1,70m, 75% males and 25% female. The clothes more commonly used are jeans, shorts, t-shirts and sneakers. This room is not frequently used and its architectonic project is atypical in relation to the classroom pattern of the Polytechnic School of USP.

In the Asteróide room, which has air conditioned and is located in the Civil Engineering building, the attendance is of postgraduate students from the Department of Urban and Civil Construction Engineering, adding up to 20 students in May 2004. Their ages varied from 29 to 35 (28 in average), average weight was 64kg, average height 1,71m, 60% female and 40% male, generally wearing the following: jeans, social pants , shirts, shoes and sandals. This room is occupied in diverse hours, the amount of students varies a lot during a given day and its architectonic project is atypical, characterized by an “L” format.

Room S23 is also located in the Civil Engineering building, with capacity for 48 students. It was built according to a new tendency of architectonical project based on the Fanger’s methods for air-conditioning system installation. It has sensors which obtain the real temperature, which allows regulation for the adaptation to the ideal environment. The room is used by first-year students of the Civil Engineering course, being fully occupied during all the semesters. The questionnaires were answered in August, 2005 by 105 students whose ages varied from 17 to 23 (19 in average). Their average weight was 68,52kg, average height 1,75, 83% were male while 18% were female. They usually wore jeans, shorts, t-shirts and sneakers.

Classroom 2 located in the Mechanical Engineering building has ceiling fans and capacity for 130 students. It’s occupied by first, second and third-year students of the Civil, Mechanical, Naval and Mechatronics (Mechanics/Electronics) Engineering, totalizing 246 questionnaires answered in 2004 and 350 in 2005 by students whose ages varied from 18 to 27 (21 in average). These students had an average weight of 61,28 kgs, average height of 1,64m and 75% were male while 25 were female. Jeans, shorts, t-shirts and sneakers were the clothes usually worn.

The occupant’s characteristics presented similar metabolical rates and the rooms had a regular use. The greatest differences among the classrooms chosen for comparison are related to cooling solutions.

The data about external environment were collected at four Meteorological Stations.

1) Institute of Astronomy, Geophysics, and Atmospheric Sciences (IAG) – USP.
1a) State Park Meteorological Station 1b) University City Meteorological Station 2) Philosophy, Letters and Human Sciences College (FFLCH)– University City of San Paulo-Capital – Department of Geography - Laboratory of Climate and Biogeography 3) Department of Urban and Civil Construction Engineering (Endure Project). The differences

of atmospheric variables on the same day were verified. For instance, the 2nd of March of 2004 can be mentioned, characterized as entrance of pre-frontal, when it was observed the air temperature (°C) at the Meteorological Stations 1 a) 23,4; 1 b) 22,08; 2) 24,6; 3)24,9, the atmospheric pressure (Kpa) 1 a)922,7; 2)924,94; 3) 930, relative humidity (%) 1 a)70,9; 1 b) 84,26; 2)70,9; 3)63,59. This criterion was used for all the days of the field survey which allowed conclusions with the comparison of data from the 4 Meteorological Stations 1 b) and 3), installed on the flagstone of the buildings, coated with asphalt blanket, agree about the edges of atmospheric variables and oppose to the ones of Stations 2) and 1 a), installed in woody places, which explains these contradictions. In 2004 the air temperature for Station 2) agreed with the information from 1a). However, the relative humidity is characterized for the period of events of cold fronts.

Starting from the climatologic data of 2004 and 2005, the entrance of frontal, prefrontal and previous to prefrontal systems data could be verified. It was calculated the difference (ΔT) between the day of entrance of prefrontal temperature (Tp) with the day previous to prefrontal (Tap), relating it to the reply on thermal comfort given by the students. At a second moment it was verified the number of frontal system entrances for the months of the experiment and their passage by San Paulo, also presented in percentages during the years of 2004 and 2005. It could be verified that in 2004 there were more passages of frontal systems than in 2005 which indicates intense temporal variation of frontal systems at the same time of the year. An important datum observed in the student's answers is the ocular irritation reported, which motivated the verification of the conditions of atmospheric pollution. It could be evidenced that this irritation was directly related with the increase of air temperature that causes an increase on pollution. Based on these analyses, a relation was established between the results obtained about external and internal environment and the replies from the questionnaires.

As an example, April 17th 2004 can be mentioned as characterized as entrance of air mass at room 2 in the Mechanic Engineering building, without thermal conditioning. The first case of classroom study without thermal conditioning presents the data as frontal entrance data on April 17th where it was verified the variation of temperature (°C) 1a) 22,9 1b) 22,34 2) 23 and 3) 23,3 and relative humidity of air (%) 1a) 74,2 1b) 88,85 2) 80,8 and 3) 69,59, atmospheric pressure (kpa) 1a) 926,4 1b) 927,49 and 3) 934 from the four meteorological stations.

The day previous to the entrance of prefrontal system, on April 16th the air temperature (°C) variation was identified as 1a) 22,2 1b) 21,92 2) 22,2 and 3) 22,6 and relative humidity of air (%) 1a) 76,1 1b) 88,85 2) 82,9 and 3) 72,74, atmospheric pressure (kpa) 1a) 925,8 1b) 926,76 and 3) 933. With these data the deviation observed on external air temperature from day 17 to day 16 was of 0,6°C. The timetable analysis from the "datalogger" installed in the internal environment on day 17 of April shows that the variation of air temperature (°C) of the was TP1-23,72, TP2-23,78, TP3-22,45 and from the thermocouple that measured the wall temperature was TP4-25,80, relative humidity of air (%)82 and atmospheric pressure (Kpa) 926,40. On the other hand, the analysis from April 16th presents the variables of air temperature (°C) of thermocouple in TP1-23,3, TP2-23,49, TP3-22,08 and the thermocouple that measured the wall temperature TP4-25,63, relative humidity of air (%) 77 and atmospheric pressure (Kpa) 925,80.

The student's replies to the questionnaire about the sensation of comfort was 34% comfortable, 25% slightly cold and 25% cold, and it was observed that although there is a small variation of temperature in relation to the previous day there were no alteration on the questionnaire answers. It was observed that on the 17 of April between 12 and 17 o'clock, the thermocouples TP1, TP2 and TP3 indicated air temperatures very near to the external meteorological stations, that is, from 22 to 23°C. The thermocouple TP4 located near the wall presented an elevated air temperature, measuring 25,80°C, compared with the external temperature due to the influence of the edification envelope.

These analyses were elaborated for the classrooms with and without thermal conditioning and it permits us to conclude that the entrance of air masses significantly interfere on the students' answers

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Indoor habits of children aged 5 to 10 years learning at the public basic schools of Lisbon-city, Portugal

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SUMMARY

In this work, 36 basic schools of Lisbon city, Portugal followed a questionnaire of the ISAAC - International Study of Asthma and Allergies in Childhood Program. The questionnaire contains questions to identify children with respiratory diseases (wheeze, asthma and rhinitis) as well as their nutrition habits, ingested medication, environmental aspects, among others. The questioned children are 5 to 10 years old, and the answers are from June to December 2006. The results are from 995 children inquired who have shown 26.7% with wheezing symptoms, 9.2% with asthma, and 26.2% with rhinitis. The results obtained are compared with the results, interpretations and correlations obtained in the ISAAC 2002 program, which questioned 2484 6 to 7 years old children from the basic schools of Lisbon-city from November 2002 till March 2003.

INTRODUCTION

Children's exposure to air pollution is a very important issue mainly because children immune system is not fully developed, as well as their lungs. Consequently their response to pollution from traffic in urban areas, particles and other combustion sources are different to those observed by adults [1]. In addition, children spend more time outside, where the concentrations of our pollution are generally higher. The effects of air pollution in children with respiratory problems might be even higher than recognised.
Overall, evidence for effects of air pollution on children has been growing, and effects are seen at concentrations that are common today. Although many of these associations seem likely to be causal, others require additional investigation [1].

The aim of this work is to identify the children with respiratory problems with ages between 5 to 10 years, studying in the basic schools of the Lisbon city. It will be followed by other studies which will relate air pollution, with acute episodes of respiratory diseases in the same population. The identification was made by simple core questionnaires which followed other identification study – ISAAC - International Study of Asthma and Allergies in Childhood. ISAAC was formed in 1991 to facilitate research into asthma, allergic rhinitis and eczema by promoting a standardised methodology. ISAAC developed from a merging of two multinational collaborative projects each investigating variations in childhood asthma at the population level. It was a unique project which has attracted worldwide interest and unprecedented large scale participation. The aims of ISAAC were to describe the prevalence and severity of asthma, rhinitis and eczema in children living in different centres and to make comparisons within and between countries [2]. The study has included the city of Lisbon, and the results of this work will be compared with the results obtained in this 2002 international study.

The first part of the questionnaire identifies children with wheeze symptoms. Wheeze is a continuous, coarse, whistling sound produced in the respiratory airways during breathing. For wheezes to occur, some part of the respiratory tree must be narrowed or obstructed, or airflow velocity within the respiratory system is higher. Wheezing is commonly experienced by persons with a lung disease.
The aim of the second part of the questionnaire is to identify asthmatic children, a chronic disease of the respiratory system in which the airway occasionally constricts, becomes inflamed, and is lined with excessive amounts of mucus, often in response to one or more triggers. These acute episodes may be triggered by such things as exposure to an environmental stimulant (or allergen), cold air, exercise or exertion, or emotional stress. In children, the most common triggers are viral illnesses such as those that cause the common cold. This airway narrowing causes symptoms such as wheezing, shortness of breath, chest tightness, and coughing. Between episodes, most patients feel fine. The symptoms of asthma, which can range from mild to life threatening can usually be controlled with a combination of drugs and environmental changes.

The third part aims to identify children with rhinitis symptoms and hay fever. Rhinitis is the medical term describing irritation and inflammation of the nose. The primary symptom of rhinitis is a runny nose. It is caused by chronic or acute inflammation of the mucous membrane of the nose due to viruses, bacteria or irritants. The inflammation results in excessive mucous production producing a runny nose, nasal congestion and post-nasal drip. Hay fever is caused by pollens of specific seasonal plants in people who are allergic to these substances. When these symptoms are caused by pollens, the allergic rhinitis is commonly known as "hay fever", after the fact it is most prevalent during haying season. Allergies are caused by an oversensitive immune system, leading to a not direct immune response. Allergy occurs when the immune system reacts to substances (allergens) that are generally harmless and in most people do not cause an immune response. As noted above, hay fever involves an allergic reaction to pollen. A virtually identical reaction occurs with allergy to mould, animal dander, dust, and similar inhaled allergens. Particulate matter in polluted air and chemicals which can normally be tolerated can greatly aggravate the condition.

The ISAAC work concludes that there are several factors which might trigger acute episodes, or on other hand, be protective about respiratory systems diseases.
The last part of the questionnaire, follows this principle as described in the ISAAC studies, and gathers the information about these factors, so that later conclusions can be found.

METHODS

It has been established a contact with the ISAAC coordinator in Portugal – Dr. José Eduardo Rosado Pinto, prior to the study start. This contact lead to the elaboration of the questionnaires based on the ISAAC study.
The next step involved the contact with all the basic schools in the city of Lisbon, to present the project and to get authorization to deliver the questionnaires to the students. The questionnaires were delivered from June 2006 to the participating schools. At the same time oral presentation to the teachers and meetings with the school coordinators took place to explain the main targets to be achieved, and to motivate the filling of the questions and improve the number of students with answers. The distribution of the schools in the Lisbon city is shown in fig. 1

Figure 1 Distribution of the participating schools in the city of Lisbon, and number of answered questionnaires for each.

During this period and till February 2007, the data from the questionnaires were processed to make possible a further statistical treatment. As a first step the data treatment included the validation of the data base by comparing the results processed in two independent databases made by two persons.

RESULTS

Tables 1 to 10 resume the results obtained by processing the questionnaires as well as the results obtained in 2002-2003 on the ISAAC work for comparison. The non valid results were eliminated for each question, and was calculated the total number of answers, the percentages in each question in relation to the total number of answers or in some questions in relation to the number of the positive answers.

Table 1 Resume of schools inquiry

Inquiries Information		ISAAC
Number of schools contacted	96	-
Number of schools with answers	36	103
Number of inquiry's	995	2484
Answers from	Jun to Dec 2006	Nov 2002 to March 2003
Children with ages between	5 – 10 years	6 – 7 years

Table 2 Results related with wheezing symptoms

Questions in relation with wheezing symptoms	Results from this work			ISAAC RESULTS
	Nr	**% in relation to children with wheezing symptoms**	**% in relation to the total number of answers**	**% in relation to the total number of answers**
Number of children inquired with wheezing symptoms	266		26.7%	30.1 %
Number of children with wheezing symptoms in the last 12 months	113	42.5%	11.4%	14.1 %
Number of days with wheezing symptoms in the last 12 months	**Nr**	**% in relation to children with wheezing symptoms**	**% in relation to the total number of answers**	**ISAAC RESULTS**
1 to 3	80	30.1%	8.0%	not available
4 to 12	24	9.0%	2.4%	n. a.
more then 12	8	3.0%	0.8%	n. a.
Weakening because of wheezing symptoms				
Never	38	14.3%	3.8%	n. a.
1 night or less per week	49	18.4%	4.9%	n. a.
1 to 2 nights per week	26	9.8%	2.6%	n. a.
Children with talking limitation because of wheezing symptoms	23	6.8 %	2.3%	n. a.

Table 3 Results related to asthma

ASTHMA	Results from this work		ISAAC RESULTS
	Nr	*% in relation to the total number of answers*	
Children with asthma	92	9.2%	7.8 %

Table 4 Results related to Rhinitis

RHINITIS	Results from this work			ISAAC RESULTS
	Nr	% in relation with children with rhinitis symptoms	*% in relation to the total number of answers*	*% in relation to the total number of answers*
Children with sneezing crisis, runny nose or nasal congestion not associated with common cold	268		26.9%	31.2%
Children with sneezing crisis, runny nose or nasal congestion not associated with common cold in the last 12 months	232	86.6%	23.3%	26.2%

Table 5 Rhinitis symptoms along the year

Months with more nasal problems in the last year	Nr	% in relation to the total number of answers	Months with more nasal problems in the last year	Nr	% in relation to the total number of answers
Jan.	81	10%	Jul.	18	2%
Feb.	64	8%	Aug.	23	3%
Mar.	114	15%	Sep.	68	9%
Apr.	105	13%	Oct.	72	9%
May	91	12%	Nov.	53	7%
Jun.	44	6%	Dec.	49	6%

Table 6 Rhinitis influence in the daily activities

Influence of Rhinitis in the daily activities of the children	Results from this work		
	Nr	% in relation with children with rhinitis symptoms	% in relation to the total number of answers
Not affected	111	41.4%	11.2%
Low	77	28.7%	7.7%
Average	58	21.6%	5.8%
High	8	3.0%	0.8%

Table 7 Results related with Alimentary Habits

Eating habits	Results from this work		ISAAC RESULTS
	Nr	% in relation to the total number of answers	
a. Meat			
More than 3 times a week	850	85.4%	74.6 %
1 or 2 times a week	130	13.1%	21.2 %
Never or occasionally	11	1.1%	0.8 %
b. Fish	**Nr**		
More than 3 times a week	526	52.9%	52.8 %
1 or 2 times a week	409	41.1%	38.4 %
Never or occasionally	54	5.4%	4.2 %
d. Vegetables	**Nr**		
More than 3 times a week	570	57.3%	50.8 %
1 or 2 times a week	320	32.2%	33.0 %
Never or occasionally	86	8.6%	9.1 %
i. Butter	**Nr**		
More than 3 times a week	588	59.1%	58.5 %
1 or 2 times a week	297	29.8%	25.5 %
Never or occasionally	100	10.1%	10.1 %
j. Margarine	**Nr**		
More than 3 times a week	124	12.5%	20.0 %
1 or 2 times a week	293	29.4%	26.0 %
Never or occasionally	538	54.1%	42.9 %

Table 7(cont.) Results related with Alimentary Habits

Eating habits	Results from this work		ISAAC RESULTS
		% in relation to the total number of answers	
l. Potatoes	**Nr**		
More than 3 times a week	499	50.2%	63.5 %
1 or 2 times a week	459	46.1%	29.3 %
Never or occasionally	24	2.4%	2.2 %
m. Milk	**Nr**		
More than 3 times a week	940	94.5%	86.5 %
1 or 2 times a week	21	2.1%	7.2 %
Never or occasionally	21	2.1%	1.0 %
o. Fast food/hamburgers	**Nr**		
More than 3 times a week	40	4.0%	4.1 %
1 or 2 times a week	280	28.1%	20.4 %
Never or occasionally	661	66.4%	67.8 %

Table 8 Results about Breastfeeding

Breastfeeding	Results from this work		ISAAC RESULTS
	Nr	% in relation to the total number of answers	
Children that have been breastfeed	811	81.5%	82.2 %

Table 9 Results related with medicines and environmental aspects

	Results from this work		ISAAC RESULTS
	Nr	**% in relation to the total number of answers**	
Paracetamol Administration	**Nr**	**% in relation to the total number of answers**	
Paracetamol ministration in the first 12 months of life	239	24.0%	78.1
Paracetamol ministration in the last 12 months	**Nr**	**% in relation to the total number of answers**	
At least once a month	189	19.0%	17. 0 %
At least once a year	481	48.3%	73.9 %
Never	250	25.2%	5.1 %
Antibiotics administration	**Nr**	**% in relation to the total number of answers**	
Children with antibiotics ministration in the first 12 months of life	493	49.5%	53.6 %
Lorries traffic near home	**Nr**	**% in relation to the total number of answers**	
All day long	74	7.4%	6.4 %
Frequently	232	23.3%	23.9 %
Rarely	479	48.1%	50.2 %
Never	186	18.7 %	16.8 %

Table 10 Results related to parents smoking habits

Smoking habits		Results from this work		ISAAC RESULTS	
	Nr	% In relation to the total smokers	% In relation to the total number of answers	% In relation to the total smokers	% In relation to the total number of answers
Smoking mothers	348		35.0%		26.7 %
Less than 10 cigarettes/day	96	27.6%		14.7 %	
10 to15 cigarettes/day	76	21.8%		4.9 %	
15 to 20 cigarettes/day	124	35.6%		5.6 %	
More than 20 cigarettes/day	52	14.9%		1.0 %	
Smoking Fathers	394		39.6%		42.0 %
Less than 10 cigarettes/day	80	20.3%		11.0 %	
10 to15 cigarettes/day	63	16.0%		6.6 %	
15 to 20 cigarettes/day	128	32.5%		15.3 %	
More than 20 cigarettes/day	123	31.2%		6.0 %	
Smoking Mothers in the first year of age of the children	258		25.9%		17.8 %

Table 10(cont.) - Results related to parents smoking habits

Less than 10 cigarettes/day	58	36.8%			
10 to15 cigarettes/day	54	22.5%			
15 to 20 cigarettes/day	51	20.9%			
More than 20 cigarettes/day	258	19.8%			

As mentioned before, the ISAAC study found some risk and protective factors for the respiratory diseases studied. These factors are given in the tables 11 and 12, as well as the results obtained with our inquiries:

Table 11 Results relative to wheeze protective and risk factors

WHEEZE				
Eating habits				
Factor	**Protective**	**Risk**	**Results from this work**	**ISAAC RESULTS**
Fish		More than 3 times a week	52.9 %	52.8 %
Margarine	More than 3 times a week		12.5 %	20.0 %
Meat	More than 3 times a week		85.4 %	74.6 %
Milk	More than 3 times a week		94.5 %	86.5 %
Potatoes	More than 3 times a week		50.2 %	63.5 %
Fast Food		More than 3 times a week	4.0 %	4.1 %
Medicines				
Paracetamol		At least once a month (recently)	19.0 %	17.0 %
Antibiotics		In the first year of age	49.5 %	53.6 %
Environment				
Lorries Traffic		All day long	7.4 %	6.4 %
Smoking				
Smoking Mother	Now		35.0 %	26.7 %
Smoking Mother		First year of Age	25.9 %	17.8 %

Table 12 - Results relative to asthma protective and risk factors

ASTHMA				
Factor	**Protective**	**Risk**	**Results from This work**	**ISAAC RESULTS**
Eating Habits				
Milk	More than 3 times a week		94.5 %	86.5 %
Medicines				
Antibiotics		In the first year of age	49.5 %	53.6 %
Environment				
Lorries Traffic		All day long	7.4 %	6.4 %

DISCUSSION

The results obtained in the questionnaires give a good estimate about the respiratory diseases on the urban area of Lisbon, for the children with 5 to 10 years old. The number of inquiries was lower than the ISAAC study, but pretended to gather information about the population in the same area.

The results given are good information of the respiratory diseases for children in the urban city of Lisbon, for a population with a special concern due to their greater risk for these diseases. Although not exactly the same, the results obtained are comparable to the ISAAC study in 2002, which had a bigger population inquired. For this reason we consider our results acceptable and a good snapshot of the respiratory problems in the area studied.

We found 26.7 % (266) of children with wheezing symptoms from which 42.5 % (113) had the symptoms in the last 12 months and with about 30% (80) having 1 to 3 days of illness in the same period. Again for the children that have recognized having this respiratory disease, 18.4 % (53) have wakened up at night at least once in the last year with wheeze and 6.8% (23) had talking limitation again because of the symptoms. Comparing the results found with the ISAAC study for this respiratory disease, we find a very close percentage, although slightly minor in our Inquiry from 2006.

The results show 9.2 % (92) of asthmatic children with ages from 5 to 10 years in the city of Lisbon. This is a significant result for a chronic disease that might be trigged by an environmental stimulant. This fact reveals the importance of a more detailed study relating asthma (and other respiratory diseases) with atmospheric pollution. In addition the comparison with the ISAAC results from 2002/2003 show a slightly increase of the presence of this disease in 2006.

For rhinitis the results obtained are lower then in the ISAAC study, 26.9 % (268) and 31.2% respectively. The majority of these children had rhinitis in the last year, corresponding to 86.6% (238). According to the results obtained for rhinitis, the months where the disease is more evident are March, April and May, and in lower percentage January. This fact confirms the seasonal prevalence of this respiratory problem, more present in the spring months. In addition 21.6 % (59) of the children inquired with rhinitis had an average affection of the daily activities.

For hay fever, we found in our results the same percentage as in the ISAAC study: 2.6% (26). We recognize this percentage could be slightly higher because part of the population inquired could not identify this disease with the related symptoms and therefore did not answer the question.

The results obtained about the eating habits of the children are similar which is expectable as the eating habits should not change in a 5 year period. This is important when relating the risk and protective factors with the results obtained. Since the similarities of this study with the ISAAC 2002, we can conclude that the factors referred in table 11 and 12 may be applicable to the population studied.
The majority of the children of Lisbon have been breastfeed corresponding to 81.5 % (811). The result was very close to the result of the ISAAC questionnaire in 2002 – 82.2%.
For medicines given to children, our results show a big decrease of Paracetamol administration in the first 12 months of life, and in fewer amounts of antibiotics as well, in comparison with ISAAC results.

The children living with frequent Lorries traffic near home is significant: 23.3 % (232), and similar to what was obtained in 2002 (23.9%)
In terms of the smoking habits, it is important to refer the increase in the number of smoking mothers (26.7 in 2002/2003 to 35.0% in 2006), the increase in the number of smoking mothers in the first year of the children (17.8% in 2002/2003 to 25.9% in 2006) and a slightly decrease on the number of smoking fathers (42.0% in 2002/2003 to 39.4% in 2006).

As referred before, table 11 give the main protective and risk factors found in the ISAAC study for wheeze. For the results found in our work, we can consider important risk factor eating fish more than 3 times per week. This corresponds to 52.9% of children only more 0.1% than in the ISAAC study. We also consider important as a risk factor the ministration of antibiotics in the first year of age. It was found 49.5% of the children which is also close to the 53.6% in the ISAAC study.
In terms of protective factors, we found 85.4% of children eating meat more than 3 times a week (74.6% in the ISAAC study) and 94.5% drinking milk more than 3 times a week (86.5% in the ISAAC results). This can be considered the main protective factors for wheeze because of the higher percentage of answers and because the results are close to the results of the 2002/2003 study.
As we can observe in table 12 for asthma, the same considerations made for wheeze referring antibiotics in the first year of age as a risk factor and eating meat more then 3 times a week as a protective factor can also be taken for asthma.

The results found show the importance and the presence of common respiratory diseases in the city of Lisbon, in the same way ISAAC study in 2002 had shown. The percentages were lower than the ISAAC study in 2002 for wheeze and rhinitis which may show a decrease of these respiratory problems in the city. In opposite, there is an increase of the number of asthmatic children.

The data presented will be used in future studies and the main objective is to monitor episodes respiratory problems with atmospheric pollution.

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Enhancing visual comfort in classrooms through daylight utilization

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SUMMARY

Research has confirmed that the well-being and the school performance of pupils depend significantly on the quality of the luminous environment, which can be achieved through daylight utilization. This paper focuses on the impact of different fenestration systems on the visual comfort achieved in classrooms. Various window locations, clerestories, roof openings and light shelves were examined with regard to indoor daylight conditions. The study was conducted as a parametric analysis, in which models incorporating the above-mentioned systems were generated on the basis of a typical classroom. The outcomes of the analysis enabled the performance evaluation of the examined apertures. The main objective of this study was not only to derive results regarding the performance of each system separately, but also to highlight specific strategies, which would promote daylight admission without increasing the construction and operation costs in school buildings.

INTRODUCTION

Nowadays the merit of daylight is well acknowledged; research has revealed not only its significant impact on our visual system, but also its supportive role on physical and psychological health, improved productivity and performance of the occupants.

These positive effects are highly appreciated in schools, where daylight levels are directly related to student's performance and enable the creation of a pleasant atmosphere for learning and teaching activities. A research conducted in over 20,000 elementary students and 100 schools in three school districts of USA has provided evidence to this statement. It has been found that students progressed 26% faster in reading and 20% faster in math in classrooms with high levels of daylight. Similar findings were observed in classrooms with well-designed skylights and louvers controlling the illumination level [1]. Among the potential mechanisms that may have been responsible for the positive association between daylight and improved performance of students, improved visibility due to higher illumination levels and better lighting quality, mental stimulation, as well as improved mood and well-being are included.

This link between increased daylight levels and improved student performance has motivated the present study. It is focused on the analysis and the evaluation of several daylight strategies for the enhancement of visual comfort in classrooms.

METHODOLOGY

The study was conducted as a parametric analysis, in which models of classrooms incorporating various window locations, clerestories, roof openings and light shelves were generated on the basis of a typical classroom. The fenestration area in the cases under

consideration was assessed in order to comply with the standards provided by the State Organization responsible for the construction of School Buildings in Greece.
A typical classroom, presented in Figure 1, is of rectangular shape, usually side lit with unilateral windows that account for 20% of floor area. For the sufficient shading of the classroom, overhangs are incorporated in the design. The geometrical and optical properties of the classroom's transparent and opaque elements are displayed in Figure 1 and Table 1.

Sun protection of the openings is a crucial part of the school building design in Greece, where clear skies dominate. It aims not only in the reduction of excessive solar heat gains, but also in the limitation of glare caused by direct solar radiation and extreme differentiation of the luminance between internal surfaces. With respect to the orientation, sun protection is usually offered with overhangs, side fins or semi-transparent curtains and movable internal blinds.

On the basis of the typical classroom, the reference model (C_0) was created. It is assumed that the reference model is south orientated, side lit, with an unobstructed view to the sky vault. By changing the geometry, the position and the function of the transparent elements, 5 more models were created. The aim was to include fenestration arrangements offering a variety of ways to admit daylight, using the standard building design and construction. The transmittance of the glazing and the internal reflectances were kept unaltered. It must be mentioned that the decision concerning models' configuration was based on the initial simulation results and the evaluation of the daylight conditions prevailing in the reference case. A brief description of the models is presented below:

- Model C_1 (Figure 2): the area of the glazed surfaces remains unaltered (20% of floor area); a light shelf is added and divides each window into a view area below and a clerestory above, projecting towards both the interior and the exterior of the window façade. Light shelf plays a dual role; it enhances daylight in the space, while protecting it from direct sun radiation.

- Model C_2 (Figure 3): the glazed area is increased to 25% of the floor area. Daylighting is provided by the lateral windows with light shelves (following the configuration of model C_1) located on the classroom's façade, and the additional vertical south-orientated skylights, which are positioned on the roof at a distance of 3.5m.

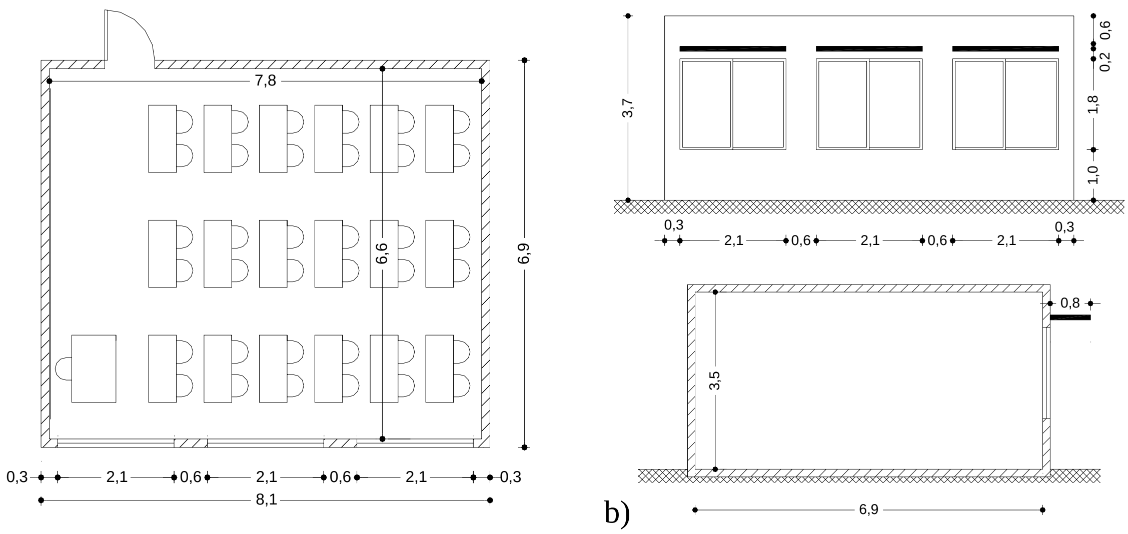


Figure 1. Plan (a), view and section (b) of a typical classroom -model C_0-.

- Model C_3 (Figure 4): the glazed area accounts for 25% of the floor area. Daylighting is provided by the lateral windows with light shelves (following the configuration of model C_1) located on the classroom's façade and vertical, south-orientated skylights, which are positioned on the roof at a distance of 4.6m (accounts for the 2/3 of the classroom's width) parallel to the window wall. In order to improve the daylight distribution, the roof extension beyond the skylight is inclined.

- Model C_4 (Figure 5): the glazed area accounts for 25% of the floor area. Daylighting is provided by the lateral windows with light shelves (following the configuration of model C_1) located on the classroom's façade and additional vertical, north-orientated clerestories positioned on the opposite wall. The roof is inclined across the classroom's width.

- Model C_5 (Figure 6): the glazed area accounts for 25% of the floor area. Daylighting is provided by the lateral windows with light shelves (following the configuration of model C_1) located on the classroom's façade and tilted, north-orientated skylights, which are positioned in two rows on the saw-tooth roof.

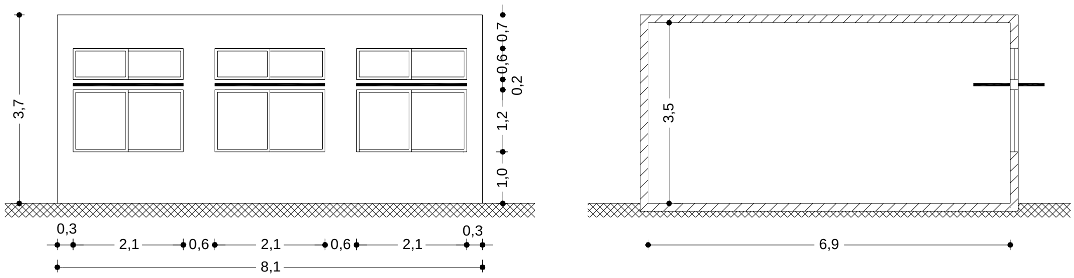


Figure 2. View and section plan of model C_1.

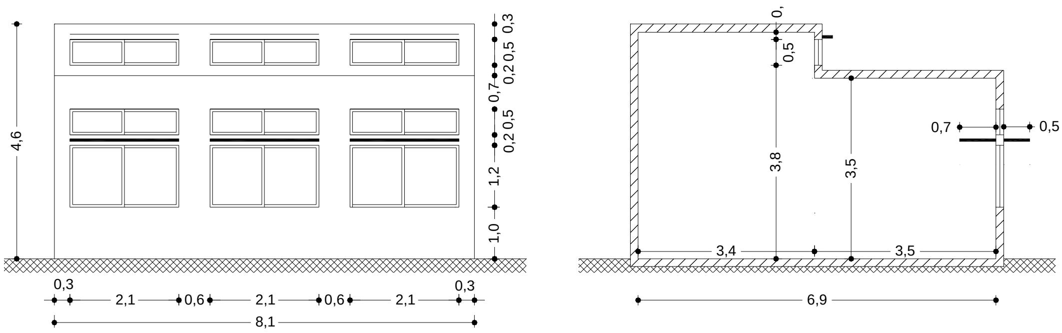


Figure 3. View and section plan of model C_2.

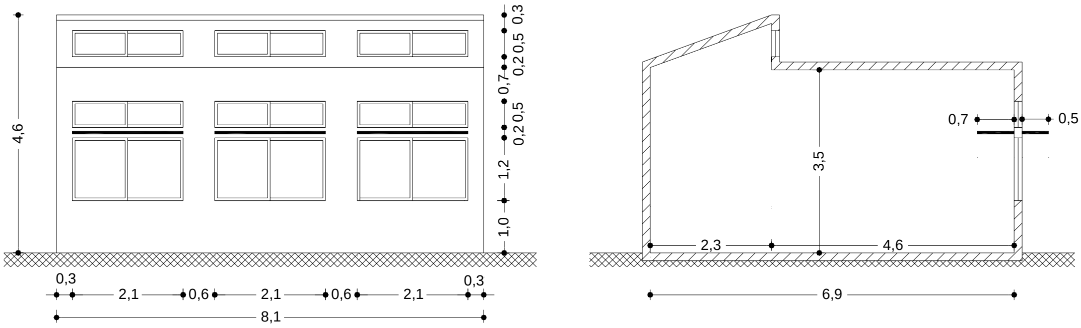


Figure 4. View and section plan of model C_3.

The visual environment prevailing in each case was estimated with the help of the simulation program ADELINE, an integrated lighting analysis tool for building design purposes, developed by the IEA Solar Heating and Cooling Task 12. It uses both radiosity and ray tracing techniques for the estimation and the visualization of daylighting conditions [2]. Details regarding the input data for the geometrical and optical characteristics of the transparent and opaque elements of the reference model C_0 and the models C_1 to C_5 are presented in Table 1. The working plane was regarded at a height of 0.8m above floor level. In order to validate the input data used for the simulations with regard to the local climate, in situ measurements were also conducted. A test cell with southern orientation, located in Thessaloniki, Greece (40^o N), was used for this purpose [3].

For the evaluation of the daylighting conditions prevailing in classrooms, where the adequacy of daylight is the main objective, the daylight factor was considered as the most appropriate parameter for indicating the quantity of admitted daylight and consequently the efficiency of the daylighting design. In bibliography, values of daylight factor ranging from 2% to 5% are reported as satisfactory. Furthermore, the homogeneity of the luminance distribution contributes to the rating of the ability of the daylighting system to attenuate glare. The quality of daylighting can be depicted by the visualization of each classroom’s interior, since the possibility of glare occurrence is associated to the excessive contrast of luminance between the various indoor surfaces. Illuminance ratios, such as minimum-to-maximum can be used in order to quantify lighting uniformity; i.e. ratios between 1:3 and 1:8 are acceptable [4].

RESULTS

The distribution of daylight factor on the working plane of the reference case C_0 and the models of the parametric study C_1 to C_5 are presented in Figures 7(a) to 12(a). The display of daylight factor contours provides a clear interpretation of daylight penetration in the classrooms. The visualization of daylight conditions is shown at figures 7(b) to 12(b). The appearance of excessive difference in luminance distribution is an indication of glare.

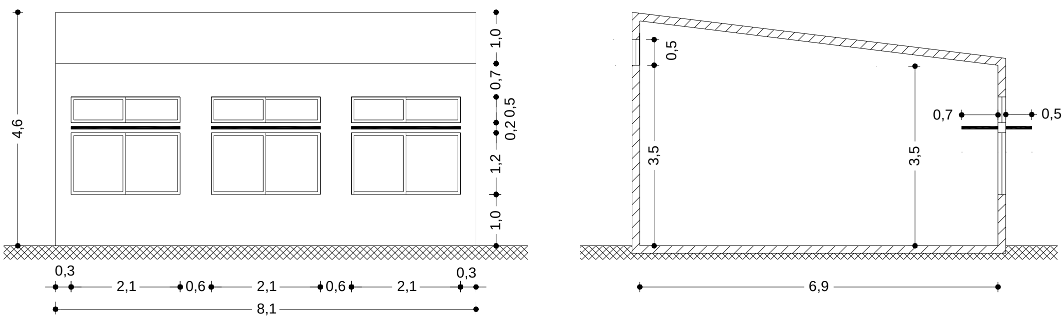


Figure 5. View and section plan of model C_4.

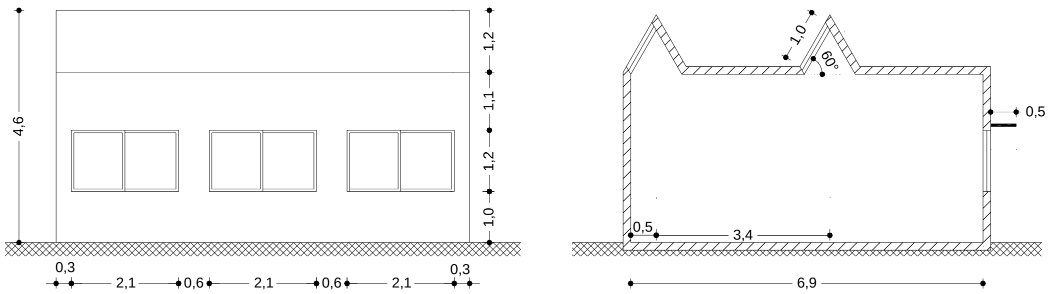


Figure 6. View and section plan of model C_5.

Table 1. The geometrical and optical characteristics of the transparent and opaque elements of the reference model C_0 and the models C_1 to C_5.

Model	C_0	C_1	C_2	C_3	C_4	C_5
Transparent elements						
Daylight technique	Side windows	Side wind., light shelf & skylights	Side wind., light shelf & skylights	Side wind., light shelf & skylights	Side wind., light shelf &clerest..	Side wind., & skylights
View windows						
No of windows	3.0	3.0	3.0	3.0	3.0	3.0
Length (m)	2.1	2.1	2.1	2.1	2.1	2.1
Height (m)	1.8	1.2	1.2	1.2	1.2	1.2
Position	Front wall	Front wall	Front wall	Front wall	Front wall	Front wall
Window sill height	1.0	1.0	1.0	1.0	1.0	1.0
Clerestories	-	3.0	3.0	3.0	3.0	-
Length (m)	-	2.1	2.1	2.1	2.1	-
Height (m)	-	0.6	0.6	0.6	0.6	-
Sun protection	Overhang	Light shelf	Light shelf	Light shelf	Light shelf	Overhang
Length	2.1	2.1	2.1	2.1	2.1	2.1
Ext. projection (m)	0.8	0.5	0.5	0.5	0.5	0.8
Int. projection (m)	-	0.7	0.7	0.7	0.7	-
Skylights & clerestor.	-	-	✓	✓	✓	✓
No of windows	-	-	3.0	3.0	3.0	6.0
Length (m)	-	-	2.1	2.1	2.1	1.0
Height (m)	-	-	0.5	0.5	0.5	0.5
Position	-	-	3.5m from front wall	4.6m from front wall	Back wall	Saw-tooth in 2 rows
Orientation	-	-	South	South	North	North
Inclination	-	-	90^o	90^o	90^o	60^o
Sun protection		-	Overhang	Overhang	-	-
Glazing, T_v	Double, 0.8	Double, 0.8	Double, 0.8	Double, 0.8	Double, 0.8	Double, 0.8
Opaque elements						
Reflectance						
Roof:	80%	80%	80%	80%	80%	80%
Floor:	30%	30%	30%	30%	30%	30%
Sidewalls:	50%	50%	50%	50%	50%	50%
Light shelves:	-	90%	90%	90%	90%	-

In reference model C_0, daylight reaches very high levels in areas near the lateral windows and is reduced across the classroom's length, ranging from 16.8% to 2.8%. The mean daylight factor is equal to 5.4%; however, on approximately 40% of the working surface daylight factor is lower than 3.0%. Moreover, the contrast between daylight levels in the interior of model C_0 often exceeds 1:9. It is therefore indicated that although the average levels of daylight are satisfactory, uniformity of daylight can not be achieved with conventional windows, since adequate daylight can not be admitted deeply in the room.

In order to enhance visual comfort, it is recommended to both moderate illuminance in areas near the windows and promote daylight penetration deep in the room. Towards this direction, light shelves were incorporated in the classroom's openings, in a way that a view window and a clerestory are created, without altering the total area of transparent elements. Their primary function is to redirect daylight to deep areas of the classroom's interior, while providing sunshade. Simulations showed that the light shelves of model C_1 reduced the amount of light received in the interior relative to the conventional windows of model C_0. Daylight factor varies from 15.3% in areas close to the windows to 1.8% in remote zones and is on average equal to 4.3%. Moreover, for almost half of the working surface, the daylight factor is lower than 3.0%. However, the luminance distribution is better and the contrast between minimum and maximum values is limited to 1:8 (fig. 8a & 8b). Such a behavior has also been observed in similar studies for light shelves [5].

With the intention of increasing daylight levels without decreasing uniformity of illuminance distribution, south-orientated vertical skylights were introduced in two models (C_2, C_3). The minimum and maximum values of daylight factor were found to be approximately equal in both models (2.4% and 16.5% respectively), but the average daylight factor was slightly higher in model C_2 (5.1%) than in model C_3 (4.9%). It is therefore obvious that the skylights of model C_2, located across the middle of the model's width, performs better as regards daylighting, when compared with those located at the 2/3 of the classroom width.

Since in Mediterranean climates, such as that of Greece, thermal performance is also very important especially during the hot months, north orientated clerestories placed on the back wall were also examined, given that south orientated apertures are associated with increased solar gains. The illumination in model C_4 ranges in high levels -the average daylight factor reaches 17%-, but the zone formed from values of daylight factor between 2.0% and 3.0%

Daylight factor (it is applied to the diagrams of Figure 7a to Figure 12a):

0,0-1,0 · 1,0-2,0 · 2,0-3,0 · 3,0-4,0 · 4,0-5,0 · 5,0-6,0 · 6,0-7,0 · 7,0-8,0 · 8,0-9,0 · 9,0-10,0 · 10,0-11,0 · 11,0-12,0 · 12,0-13,0 · 13,0-14,0 · 14,0-15,0 · 15,0-16,0 · 16,0-17,0 · 17,0-18,0

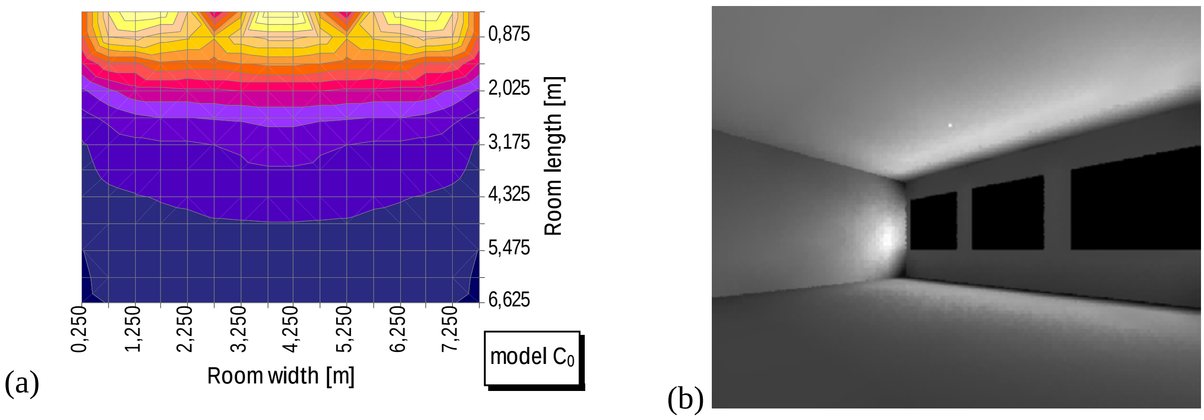


Figure 7. The distribution of daylight factor (a) and the visualization of indoor daylight conditions (b) prevailing in reference model C_0.

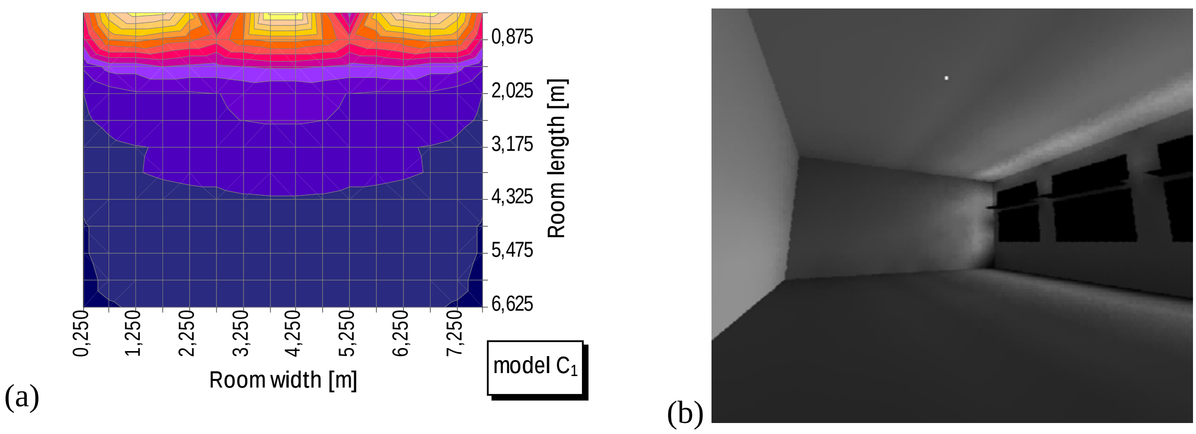
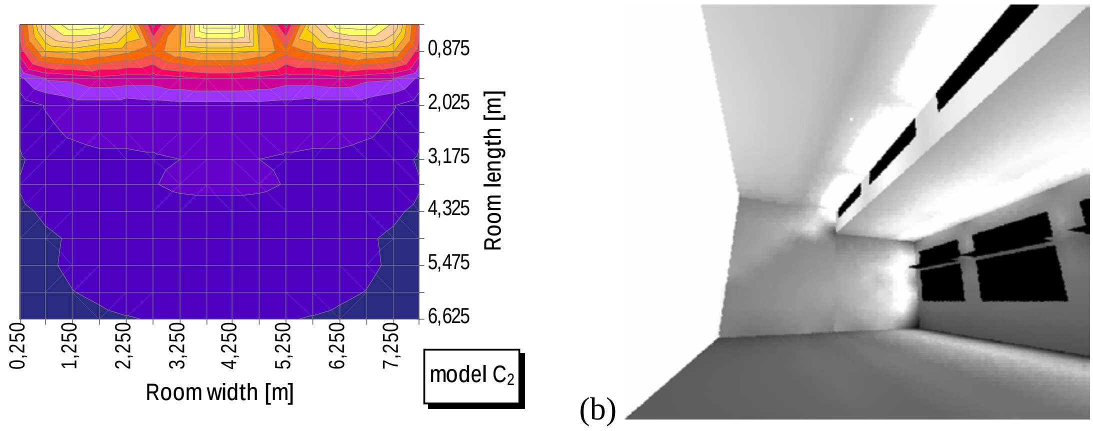


Figure 8. The distribution of daylight factor (a) and the visualization of indoor daylight conditions (b) prevailing in reference model C_1.

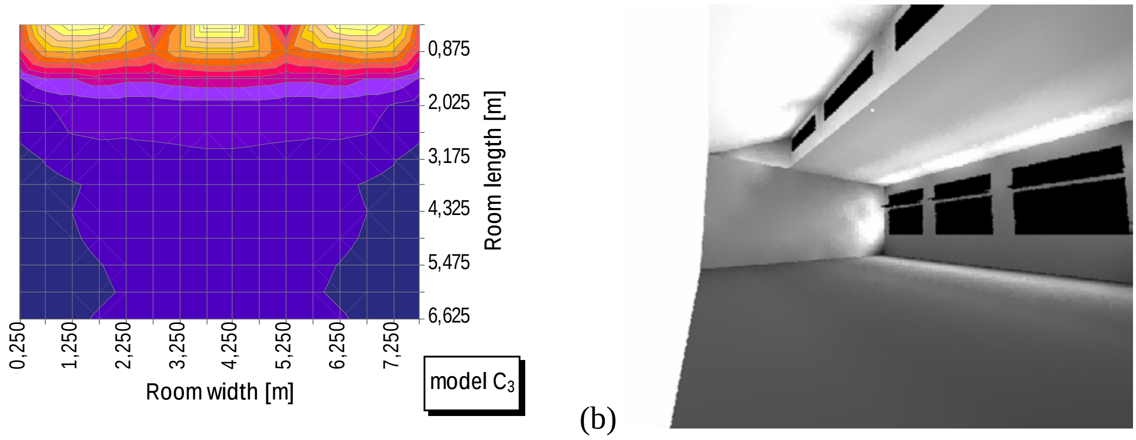
appears to be wider than the ones in models with south orientated skylights. The decrease of daylight factor in remote areas and the simultaneous increase of maximum values near the windows indicate a less uniform distribution of daylight on the working plane and consequently a probable discomfortable visual environment (Figures 11a and 11b).

On the other hand, when the north orientated apertures are tilted and arranged as skylights in the form of a saw-tooth roof, daylighting is significantly enhanced, despite of the absence of



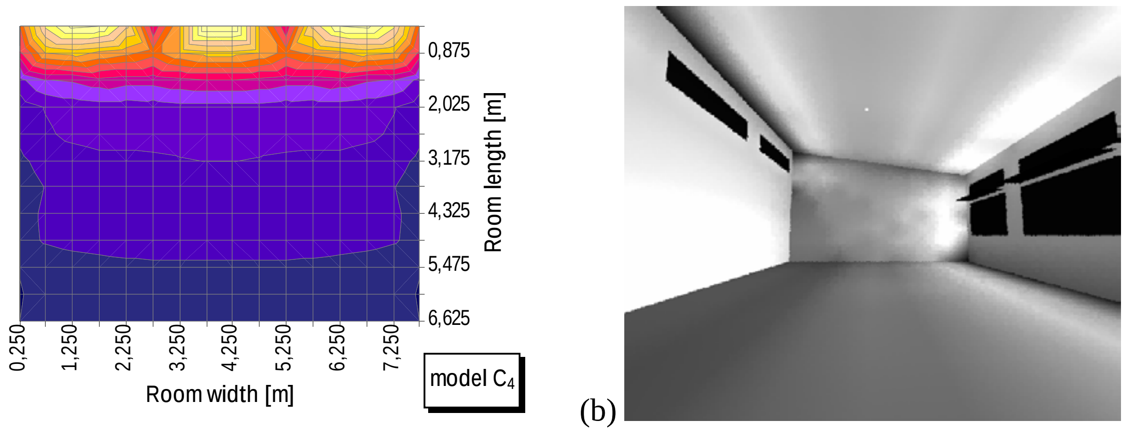
(a) (b)

Figure 9. The distribution of daylight factor (a) and the visualization of indoor daylight conditions (b) prevailing in reference model C_2.



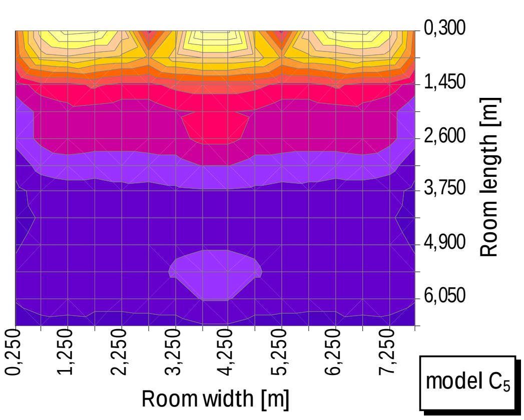
(a) (b)

Figure 10. The distribution of daylight factor (a) and the visualization of indoor daylight conditions (b) prevailing in reference model C_3.

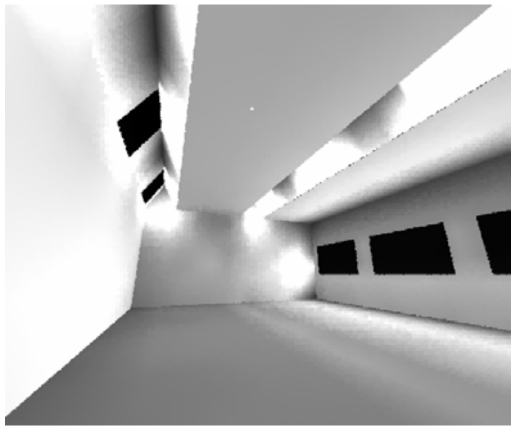


(a) (b)

Figure 11. The distribution of daylight factor (a) and the visualization of indoor daylight conditions (b) prevailing in reference model C_4.



(a)



(b)

Figure 12. The distribution of daylight factor (a) and the visualization of indoor daylight conditions (b) prevailing in reference model C_5.

the light shelves. Not only there is an overall increase of daylight factor on the working plane, but also the uniformity of its distribution is considerably intensified. The daylight factor ranges from 3.1% to 17.3%, representing a ratio of minimum to maximum equal to 1:6.

CONCLUSIONS

The above analysis has showed that many alternatives regarding the location of openings exist for the enhancement of daylighting in classrooms. It is found that a combination of façade and roof apertures performs better than advanced façade systems (e.g.light shelves). Also, skylights, clerestories and double side openings are better to be used instead of unilateral façade windows.

From the examined models the north orientated skylights arranged in a saw-tooth roof combined with south windows appear to have the best performance. They bring more daylight in the rear areas, where illumination levels are low, daylight distribution becomes balanced and glare is avoided to a great extent as well.

The construction of such a configuration is neither complicated nor expensive; on the contrary its benefits encompass visual comfort, sustainability and quality of life.

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13 June 2007 at 12:30 - 14:00

B06
Natural and hybrid ventilation systems

Thermal Comfort in a Naturally Ventilated Office Building in Karlsruhe, Germany – Results of a Survey

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SUMMARY

In order to compare measurements and subjective votes on thermal comfort in a non-conditioned indoor environment under German climate conditions, a field survey was carried out in an office and laboratory building in Karlsruhe during July 2005. Over a period of 4 weeks 50 subjects filled in questionnaires twice a day every Tuesday and Thursday and accompanying measurements were carried out at the workplaces. 90% of the votes on thermal sensation proofed the room temperatures to be "just right" or "slightly warm"; these votes cover ranges of more than 5 Kelvin of the operative temperature and also include 7.5% temperatures above 27°C. About 75% of all votes rated the indoor climate neutral or better although the room temperatures showed fluctuations in space (rooms of the building) and time (period of the study).

INTRODUCTION

A field study in a German office building was carried out to address the topic of thermal comfort and accepted indoor temperatures during summer, which is of great importance particularly in naturally ventilated and passively cooled buildings. Several studies (e.g. [1], [2], and [3]) showed that the subjective votes of occupants in naturally ventilated or passively cooled buildings correspond with a temperature band dependent on the outdoor temperature under transient summer conditions. With the introduction of the European "Energy Performance of Buildings Directive" an increasing number of commercial buildings without air-conditioning will be realized in the future. Therefore further knowledge is necessary about the perception and acceptance of a varying indoor thermal environment particularly during summer also taking into account future climate changes. Additionally, more experience with field surveys on thermal comfort has to be gained in comparison with climate chamber experiments.

DESCRIPTION OF THE BUILDING AND EXPERIMENTAL SETTINGS

The field study was carried out in an office and laboratory building situated on the campus of the "Forschungszentrum Karlsruhe", Germany. The building has a net area of approximately 5,300 m² and includes an older existing part and a new extension built in 2004, both accommodating (mostly smaller) offices as well as laboratories for chemical experiments. All offices in both building parts are ventilated naturally all year whereas the laboratories are ventilated mechanically due to the special requirements for these workspaces.

The new extension building was built as a low energy building comprising features like high heat insulation standards, a passive cooling concept for the offices as well as high daylight availability. Its thermal mass is discharged by night ventilation due to the stack effect in a central staircase, with cold outside air coming into the building through remote-controlled skylight windows in the offices. The older part of the building has suspended ceilings in the offices and less insulation of the building envelope. No passive cooling is used and it was therefore expected that the occupants' comfort perception would reflect the differences of the thermal behaviour of the two building parts.

During the study, which was carried out in July 2005 over a period of 4 weeks, short questionnaires had to be filled in by the participants twice a day every Tuesday and Thursday, resulting in 16 single surveys during the 4 weeks. In the questionnaire all aspects relevant to comfort, like room temperature, air velocity, humidity, air quality and light were addressed. Two slightly different questionnaires were used for the morning and the afternoon survey to gain some specific information related to the expectations about the indoor climate on entering the building and to changes of the indoor climate during the day. All questions had to be answered within a 5-point-scale by the participants. Sections for free comments were also provided.

A total number of 50 subjects who regularly work in the building participated in the whole study with half of them completing 9 or more single surveys (out of 16 in total). The surveys were accompanied by measurements of the relevant thermal comfort parameters during the time the questionnaires were filled in by the subjects. Additionally, the indoor air temperatures and relative humidity were recorded continuously throughout the 4 weeks in those rooms were the survey was carried out. Outdoor climate data for the site were available for the whole period. Further information on the experimental procedure is given in [4].

RESULTS OF QUESTIONNAIRES AND MEASUREMENTS

The study was carried out in a period which represented a typical but not very hot summer month for Karlsruhe with temperature maxima above 30°C on five days and distinct temperature differences between day and night on most of the days. The variations between single days and between shorter periods of similar climate conditions were strong enough to expect some affect on the subjects' votes. Radiation, temperature and humidity data are given in [4].

The resulting indoor temperatures for this period are given in figure 1. The room temperatures lie in an acceptable range for most of the time; only the temperatures in the rooms on the second floor of the old part of the building exceed 26°C for almost 50% of the whole period. The old part of the building shows a stratification of temperatures between the single floors. In the new part of the building, temperature differences between the floors are much smaller. All floors here show temperatures similar to the first floor of the old part. The effect of night ventilation is very different, both between the two parts of the building and between floors. Temperatures in the new extension did not often decrease below 23°C, even if the outdoor temperature was below 20°C at the same time. The second floor of the old building part without night ventilation hardly showed any cooling effect during the nights whereas the ground floor had the same characteristic as the new extension.

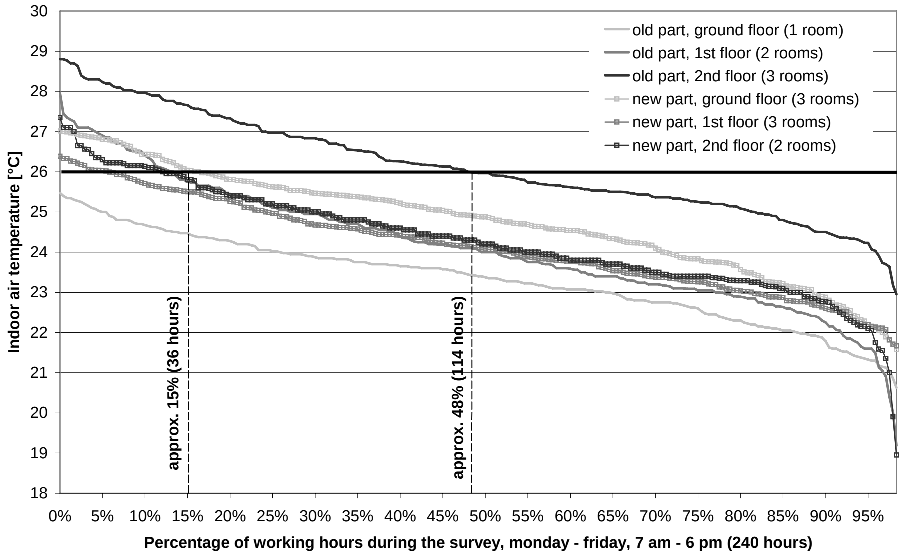

Figure 1. Indoor air temperatures in the rooms where the surveys were carried out. The bold black line at 26°C corresponds to the temperature limit of the German workplace regulations.

In figure 2 the votes for thermal sensation are given subject to the operative indoor temperature. The votes for "just right" and "slightly warm" represent 90% of all votes. They cover ranges in operative temperatures of more than 5 Kelvin and also include temperatures above 27°C. The votes for "very warm" (7% of all votes) cover a range from 25 to 30°C.

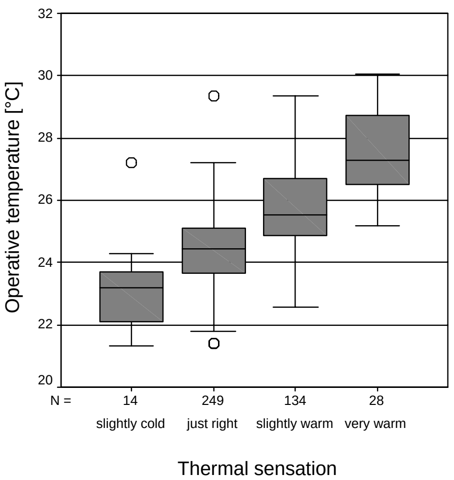

Figure 2. Box plot of votes of thermal sensation against operative temperature in the rooms. The lines in the boxes represent the median values, the grey boxes cover the mean 50% of the values and the thin lines show the whole range of all values. The small circles indicate outlines. The analysis of variance shows a significant correlation between operative temperature and votes of thermal sensation ($\alpha=0.05$, $p<0.001$, $N=425$).

Further analysis shows that the occupants accept higher temperatures in the upper floors – the median of the vote "just right" is at 24°C on the ground floor and 25°C on the second floor (see figure 3). This is also true for differences between the old and the new part of the building where the mean votes of "slightly warm" and "very warm" also differ significantly by up to 1.8 Kelvin ($\alpha=0.05$, $p=0.001$, $N=133$ for "slightly warm" and $N=27$ for "very warm").

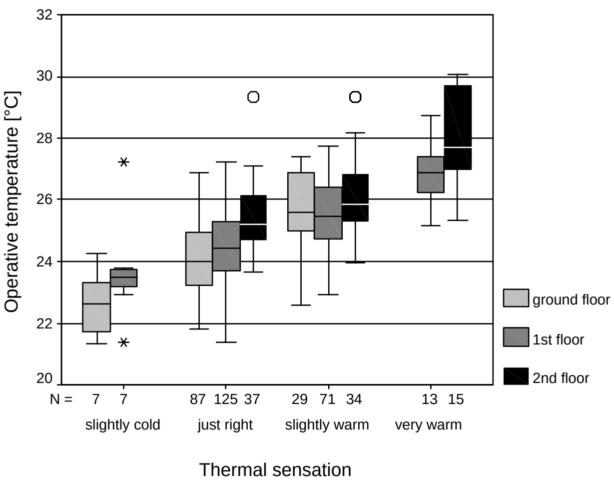

Figure 3. Box plot of votes of thermal sensation against operative temperature for the different floors of the building. The lines in the boxes represent the median values, the grey boxes cover the mean 50% of the values and the thin lines show the whole range of all values. The small circles indicate outlines. The analysis of variance shows a significant difference between the measured temperatures that are voted "just right" for the different floors ($\alpha=0.05$, $p<0.001$, $N=249$).

An increase of the indoor temperature during the day was perceived by approximately 66% of the participants which relates to the character of a free-floating building. Figure 4 shows that temperature ranges for thermal sensation votes are different in the mornings (8 a.m. to 10 p.m.) and in the afternoons (2 p.m. to 4 p.m.). In the afternoons, temperatures are judged "cooler". The median temperature of the vote "slightly warm" is 24.9°C in the mornings. This temperature is below the median value of "just right" in the afternoon (25.2°C). The median temperatures of the same votes are about 1.3°C higher in the afternoon.

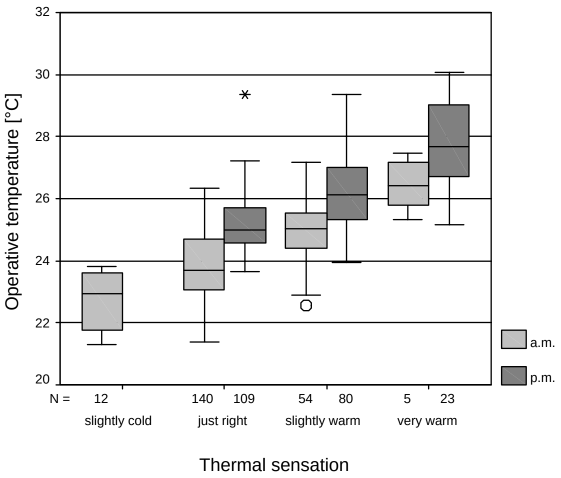

Figure 4. Box plot of votes of thermal sensation against operative temperature in the rooms in the morning and in the afternoon. The light grey boxes cover the mean 50% of the values in the mornings, and the dark grey boxes show the votes in the afternoons. The group "slightly cold" in the afternoon (n=2) has been excluded in the box plot. The analysis of variance shows a significant difference between votes in the mornings and in the afternoons ($\alpha=0.05$, $p<0.001$, $N=425$).

Analysis of the clo-values showed that most persons did not change their clothes on the days the surveys were carried out. The results also showed that the occupants used to open the windows more often in the morning (with cooler outdoor temperatures) compared to the afternoon but there is no indication from the measurements that the indoor temperature was influenced strongly. However, a rather large number of persons did not stay in the room between the two surveys but worked in a laboratory. They re-entered the office approximately 15 to 30 minutes before the afternoon survey.

Regarding the whole survey period the temperature range which is judged as "just right" varies significantly (α=0.05, p<0.001, N=249). Figure 5 shows that the ranges of July 5th, 7th and 21st equal each other, 12th and 19th are similar and July 14th and 28th show the highest temperature ranges voted as "just right". In the mornings, the lowest median temperature voted "just right" is 23.2 °C on July 21st; the highest median temperature is 25.2 °C on July 28th. In the afternoons, the lowest median value is 24.2 °C on July 7th and the highest value is 27.2 on July 28th. The maximum differences in median temperatures for the vote "just right" are 3 K in the mornings and 3 K in the afternoons with 2 K higher median values in the afternoons. It has to be taken into account that the clo-values change significantly throughout the period. People dress themselves according to the outdoor temperature, as can be seen in figure 5.

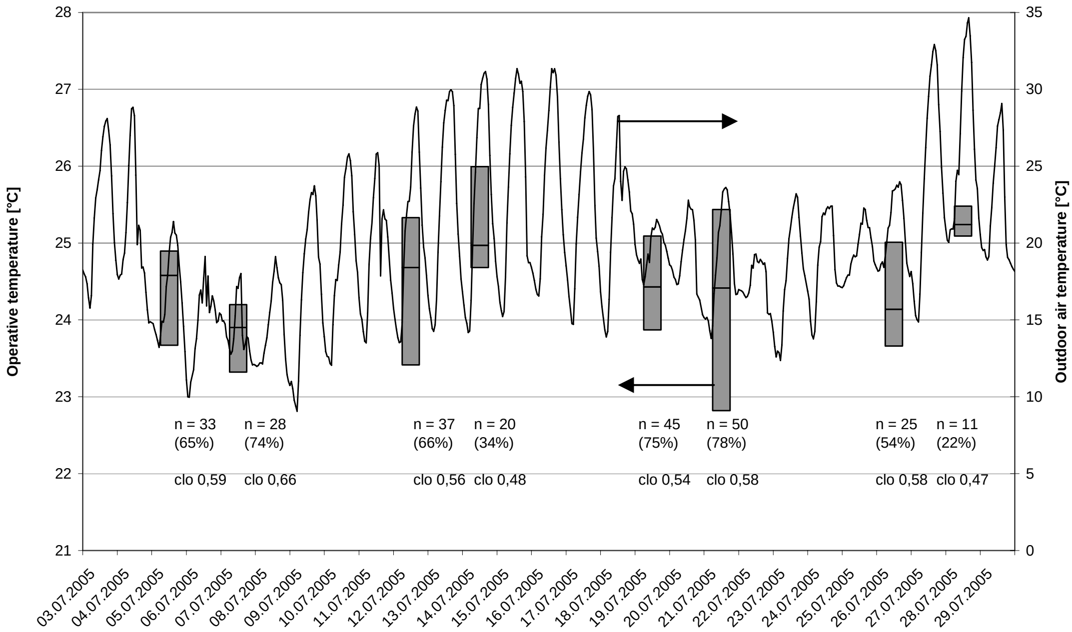

Figure 5. Outdoor air temperature during the whole study and operative indoor temperatures which were judged "just right"; the lines in the boxes represent the median values and the grey boxes cover the mean 50% of the values.

On all eight days most the participants (76%) expected the outdoor temperature to be as it was after they left their homes in the morning. No rules could be found for those votes where expectations were not fulfilled. The results for expectations concerning the indoor temperature before entering the workspace are similar: again, the majority (72%) expected the indoor temperature to be as it was on all days. If the expectations were not met the votes were mainly "slightly warmer" or "much warmer" (84%). Some of these votes can be explained by the cool outdoor temperature on that day or by unexpected changes in temperature. However, the number of votes / subjects is too small to obtain statistically significant correlations.

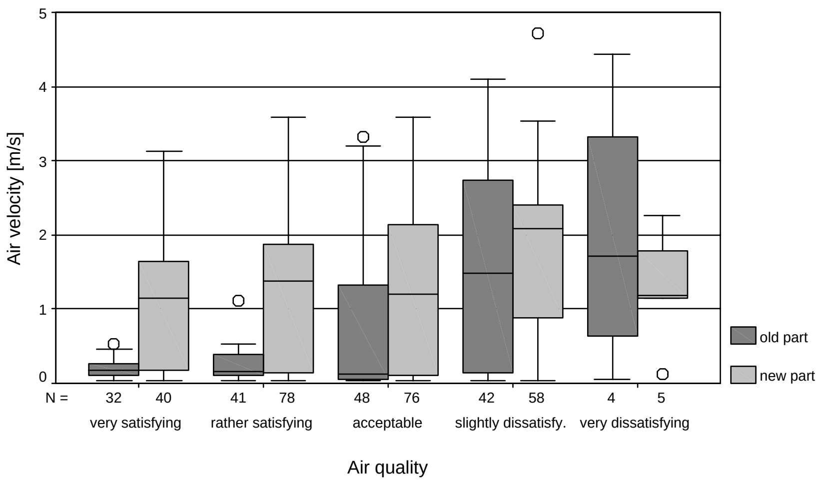

Figure 6. Box plot of votes of the perception of air quality against air velocity in both parts of the building. The dark grey boxes represent the values in the old part of the building, and the light grey boxes show the votes in the new part. The perception of air quality is not significantly different for both parts, air velocity differs significantly ($\alpha=0.05$, $p<0.001$, N=427). The correlation between air quality and air velocity is significant for the old part ($\alpha=0.05$, $p<0.001$, N=167) but not for the new part ($\alpha=0.05$, $p=0.006$, N=257).

With respect to perceived air movement the occupants wished to have higher air velocities with a thermal sensation of "slightly warm" or "warm". Subjects demanding stronger air movements felt no or only a slight movement (chi^2: $\alpha=0.05$, $p<0.001$, N=211). The evaluation of air quality also correlated significantly with the operative temperatures (analysis of variance: $\alpha=0.05$, $p<0.001$, N=424); negative votes were mostly "stuffy" and "sticky" coinciding with higher room temperatures. In figure 6 it can be seen that the air velocities increase with worse votes on air quality in the old part of the building. It seems that the occupants try to compensate the situation – bad air quality mostly as a synonym for high temperatures – by ventilating their offices.

The votes on thermal sensation, indoor air quality and overall indoor climate correlate with each other with a high level of significance. The self-reported productivity also corresponds significantly with these three parameters, and to the reported feeling (bad/well, tired/alert, hard/easy to concentrate on the work, depressed/in a positive mood). Figure 7 shows that only 9 votes out of 425 evaluated the (overall) indoor climate as "very unsatisfying" and 95 votes as "slightly unsatisfying". These votes correspond to a majority of votes of "very warm" and "slightly warm" for the thermal sensation. The neutral and positive votes on indoor climate coincide well with a large acceptance of the indoor temperature.

DISCUSSION OF THE RESULTS AND CONCLUSIONS

The methodology of the survey proofed to be practicable but very intensive for both the subjects and the researchers. After 4 weeks the motivation of the participants seemed to decrease which gives a hint for limiting extensive field studies to similar periods. The acceptance of the surveys was very high, probably because the participants were mostly scientists as well. The study in this particular building had two major shortcomings:

- the participants were not available for all surveys resulting in disparate samples for the single surveys;

- the participants did not work in their offices for the whole day and therefore experienced different room climates (particularly the climate in the mechanically ventilated laboratories).

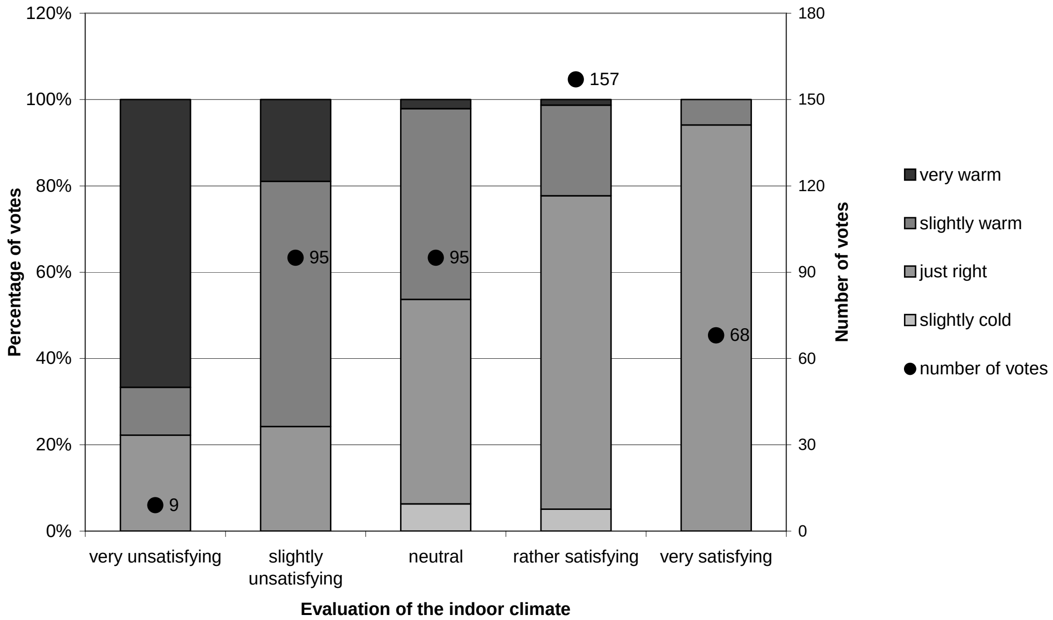

Figure 7. Relationship between votes of thermal sensation and overall satisfaction with the indoor climate

The results of the study show that a positive perception of thermal comfort is not limited by a sharp limit of the indoor temperature. Even the votes "just right" on the thermal sensation include operative temperatures higher than 27°C. About 75% of all votes rated the indoor climate neutral or better although the room temperatures showed fluctuations in space (rooms of the building) and time (period of the study).

On the other hand, the temperature levels in most monitored rooms were rather moderate with only 15% of the working hours of the whole period (240 hours) showing temperatures above 26°C. This proofs the good design of the building, particularly of the new part, in terms of passive cooling. The exception was the second floor of the old part of the building with 114 working hours above 26°C but below 29°C. In this part of the building, 50% of the indoor climate votes are negative, which is significantly above average (25%).

Regarding the fluctuations of temperatures and coinciding thermal sensations in different parts of the building it was found that occupants seem to accept higher temperatures in their offices. Their positive votes correlate to higher temperatures on the upper floors and in the old part of the building compared to other offices in the building. It was also found that the votes of thermal sensation correlate with the outdoor temperatures. The median temperature ranges of positive votes (e.g. "just right") are higher in the afternoon and on days with higher outdoor air temperatures. These findings are in agreement with other results on adaptive thermal comfort (e.g. [1], [2], and [3]).

The study showed that naturally ventilated and passively cooled buildings can be highly appreciated by occupants during summer if they are designed properly in terms of the indoor climate. Positive perceptions of thermal comfort do occur outside the temperature limits set in standards for air conditioned buildings. The study therefore confirms that adaptive comfort models are more suitable to predict the thermal sensation and thermal comfort of occupants, if periods with transient indoor (and outdoor) climate conditions are considered. This is mostly true for summer climate conditions, during which the study had been performed.

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Results of Monitoring a Naturally Ventilated and Passively Cooled Office Building in Frankfurt a.M., Germany

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SUMMARY

In this article the concept of a new energy-efficient office building and results of a 3-year monitoring are described. The monitoring was performed within the German funding programme EnOB [1]. In this building most of the offices are naturally ventilated and passively cooled. One aspect of the passive cooling is the nocturnal ventilation operated automatically by the building's control system. Another focus of the energy concept is on regenerative heating with wooden pellets.
Monitoring results show that the integrated planning enabled a very low consumption of energy for heating, ventilation, cooling and lighting. The monitoring could help to reveal operating problems and to lower the usage of energy. Though passively cooled, the rooms provided good thermal conditions. The given limits of room temperatures were exceeded only in an acceptable manner. Good air quality could be achieved just by natural ventilation.
In this article first the building and energy concept is described. Then results of the monitoring are shown, with a focus on the passive cooling concept and the achieved user comfort.

INTRODUCTION

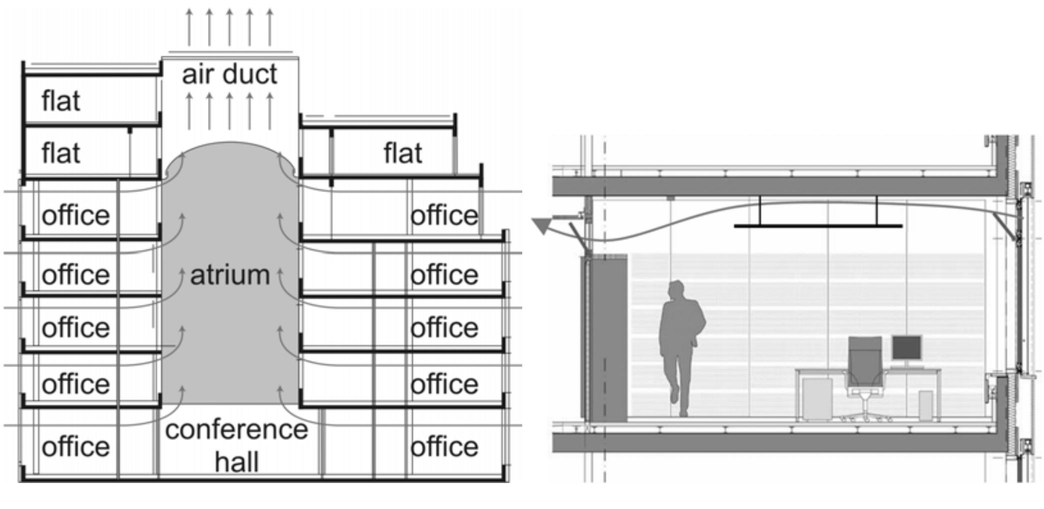
Air-conditioning and active cooling contribute significantly to the energy consumption of a lot of existing office buildings, particularly if primary energy is taken into account. Passive cooling strategies can diminish the energy consumption for cooling.
In new office building "East Arcade" of the KfW bank group (Figure 1) passive cooling and nocturnal ventilation are main aspects of the energy concept. The building is a demonstration project within a German funding programme by the Ministry of Economy and Technology. To apply for the programme the anticipated total annual primary energy use (heating, ventilation, cooling and lighting) had to be below 100 kWh per m² net floor. This was an ambitious aim but it could be achieved, as examination of 25 buildings has shown [2].

Building and Energy Concept

In order to achieve the strict given criteria an integrated architecture and energy concept had been designed. The new building was completed in 2002 and has space for 300 employees. Five of the seven floors are used as offices and the two top floors as apartments. The offices are arranged around an atrium which plays a central role in the passive cooling concept as it is used for night venting the building. It also increases the compactness of the building (area-volume ratio is 0.25 m^{-1}) and - combined with a good insulation - minimizes the heat loss through the façade. The energy for heating is provided by a pellet boiler and a condensation boiler handling peak loads. The pellets cover about 75% of heating energy throughout the year. The offices are heated by radiators mounted on the balustrade.

Figure 1. The south-elevation of the new KfW building

Moderately glazed areas with high selectivity and an efficient automatic external shading system help to reduce the ambient heat loads during warm weather conditions. Only rooms with high heat gains are conditioned by active cooling. For that purpose a heat exchanger derives part of the cooling energy from the fresh water supply.
However the majority of the offices is cooled passively by nocturnal ventilation and activating the building's thermal mass. Thereby the natural ventilation concept utilizes the stack effect in the central atrium (Figure 2a). For this purpose the skylights in the façade and towards the hallways are opened automatically during night. The concrete ceiling is exposed (without suspended ceiling) for optimal heat exchange with the air (Figure 2b).
In addition to the night venting option the rooms offer the opportunity for day-time (single-sided) natural ventilation by manually operable windows and skylights. Here the user can determine the opening times of the skylights himself or choose a short-term ventilation on the control panel, lasting for 3 minutes.



a) b)

Figure 2: a) Cross section of the KfW building with the central atrium, showing air flow pathway during night ventilation. b) Section through an office.

Two main objectives were defined. According to the EnOB-programme the total primary energy use should be lower than 100 kWh/m².
Furthermore the temperatures in the offices should not exceed a defined upper temperature limit during more than 60 hours a year, which is 60 hours with the attendance of the user. According to the decision of a German regional court in 2003 [3] the temperature in offices should not rise over 26 degrees until ambient reaches 32 degrees. Over 32°C outside there should be a minimum difference of 6 Kelvin between ambient and inside temperature.

METHODS

In this project energy consumption, operation of the technical equipment and the occupants' comfort conditions have been monitored since May 2003, using about 300 sensors. One main focus was to check whether the real measured values would match the planned and predicted energy consumption of the building. Therefore the circuits for heat and cold supply were equipped with heat meters. Facilities using much electric energy were stocked with electric meters. For examining the user's comfort room air temperature and CO_2-concentration in some representing offices have been recorded. Additionally the state of the sun blinds, the windows and the window skylights were stored to provide information about the user's behaviour and the function of the automatic building control. Also the efficiency of the passive cooling concept was observed, completed by temperature sensors mounted into the ceiling. Continuous data were recorded in 10-minute-intervals, event data like the opening of a window were recorded in real-time. Every night the data was exported by the control system into different text files on a computer on site and transferred to a database on a server at university.

RESULTS

Energy Consumption

Regarding the final energy usage of the building the energy for heating was detected as the major part. To this point constantly high supply temperatures for the heating circuits of up to 90°C were measured even during summer. The reason was that two supplemented hot water boilers needed permanently high temperatures. The high supply temperatures and active heating circuits lead to heat losses even during summer.
High exhaust temperatures of the pellet boiler revealed a further problem. An expert report indicated that the efficiency of the boiler was too small which led to a higher consumption of the wooden pellets.
Regarding primary energy heating has less influence, as wooden biomass is regarded with a primary energy ratio of only 0.2 whereas electric power is multiplied by 3.0.
The mechanical cooling for the IT and technical service rooms used a big amount of primary energy. The cooling machine runs with electrical power, its COP (coefficient of performance) is only 2.6. In the beginning in a lot of IT rooms set temperatures of 20°C made the cooling run even 24 hours a day. Changing the operation rules and rising the set temperatures to 26°C helped significantly to save both electrical power for ventilators and energy for cooling.
This is just an example of how monitoring can help to reveal problems and reduce and optimise the energy consumption of a building. Thereby the main problem weren't necessarily conceptional flaws but an improper operation of the technical equipment.

Figure 3 shows the total primary energy consumption of the building for 3 measured years and expected values for the 4th year.

During the three years of monitoring not only problems could be found but it also was shown that the building had a very low energy consumption compared to usual office buildings. The performance of ventilation and lighting turned out to be very good and user comfort could be achieved without active cooling.

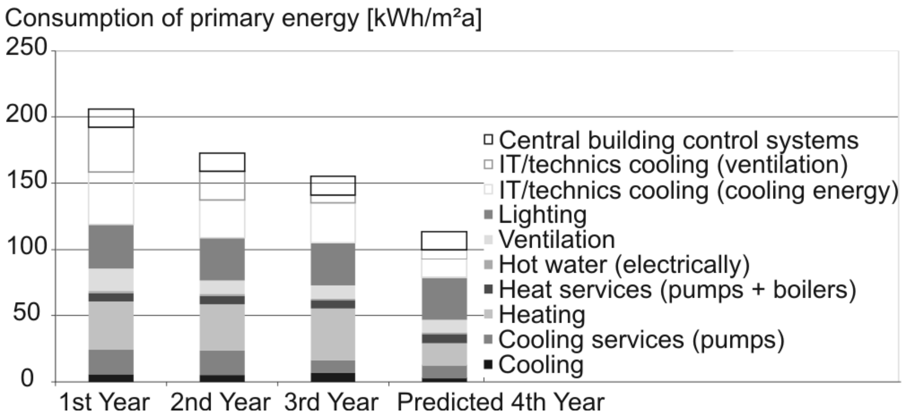


Figure 3: Primary energy consumption during the 3 years of monitoring including a prediction for the fourth year, all according to criteria of the funding programme. Energy for building control and IT-cooling is marked differently as it is not included in each of the compared buildings from the programme.

Passive Cooling

In addition to a glazing of low energy transmission and reduced internal heat loads an automatic shading system supports the passive cooling concept during summer days. For every room an integrated shading calendar of the control system calculates whether the sun hits the façade and whether the blinds have to be closed. The user is able to override the automatic regulation by the control panel in his room.

Within the monitoring project the positions of blinds have been recorded in 30 different rooms on all sides of the building. Looking at the average time that a blind is closed one can see that during summer the rooms are shaded up to 6 hours a day (Figure 4). The blinds are regarded as closed if more than 50% of the window is covered. From November until the end of February the average closing time is less than one hour a day.

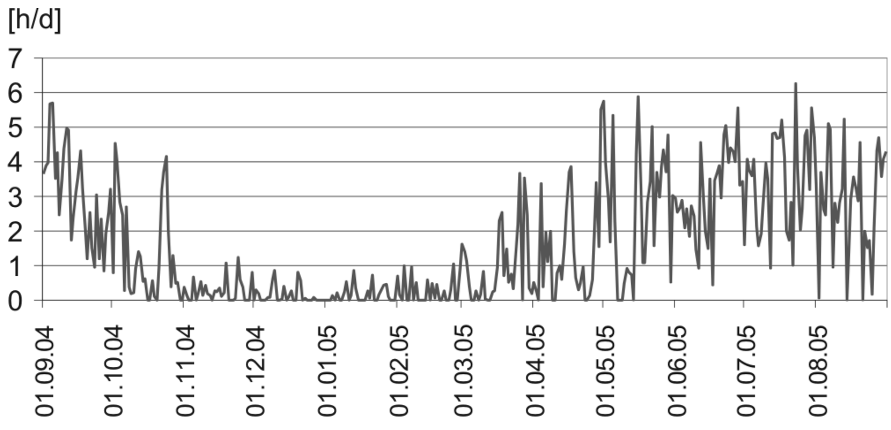


Figure 4: Average closing time of blinds during the day in all 30 measured offices

If you consider the percentage of rooms with closed blinds during a day separated by the orientation you can find out if the shading calendar is working properly. In Figure 5 it is evident that blinds on the eastern façade move down much earlier than the southern or western ones. Shading of each orientation is stopped automatically at a fixed time when the sun cannot hit the façade anymore. Beside a small number of western offices being regulated like southern rooms the function of the shading calendar does not reveal any remarkable failures. During winter it is only the southern façade with blinds being closed.

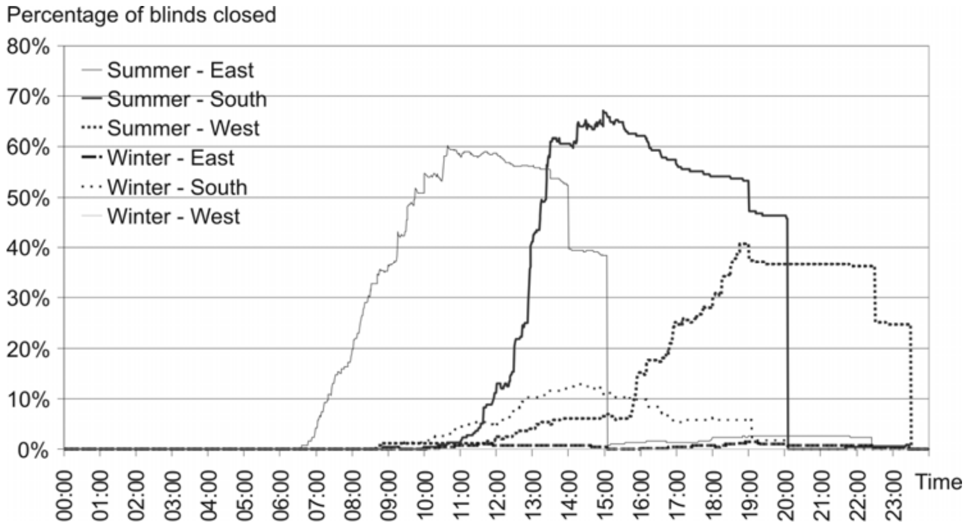


Figure 5: Average percentage of rooms with the blinds closed during a typical day in summer and winter

In order to obtain information on the effect of room air temperatures on the thermal behaviour of the concrete ceilings during night ventilation the temperatures of the ceiling (on the surface and in a depth of 4 cm) were recorded in ten offices.

In rooms featuring exposed concrete ceilings there was a pronounced effect of the room air temperature on the ceiling temperatures. Figure 6 shows how the cooler air temperature reduced the ceiling temperatures. On the other hand, some offices had suspended ceilings in which case there was no correlation between the room air and the ceiling temperatures.

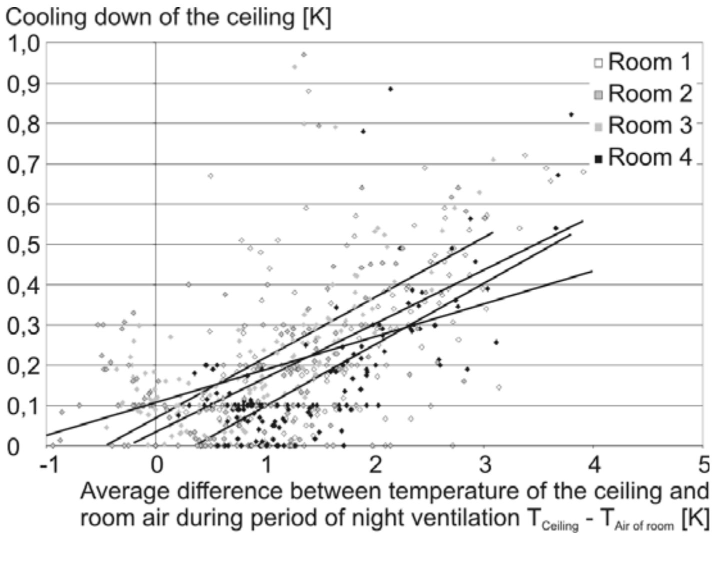


Figure 6: Cooling of the ceiling during the night ventilation from 21 p.m. until 6 a.m. depending on the average difference between the ceiling and the air temperature.

Thermal Comfort

In order to evaluate the thermal comfort conditions in the offices, air temperatures and the attendance were recorded in 10 rooms (out of 200) since 2003 and in 20 further rooms from autumn 2004 on.
The room temperature limit for acceptable conditions had been defined to 26°C for ambient temperatures up to 32°C and with a minimum temperature difference of 6 Kelvin between the indoor and ambient air temperature when the ambient temperature exceeded 32°C (referred to as 32/6-limit). Even during the hot summer of 2003 the room temperatures of frequently used and passively cooled rooms in average surpassed this limit by only 2.1% (32 hours) of the overall attendance time [4]. The maximum temperature in these offices reached 29.6°C with the outdoor air temperatures reaching 40°C (Figure 7).

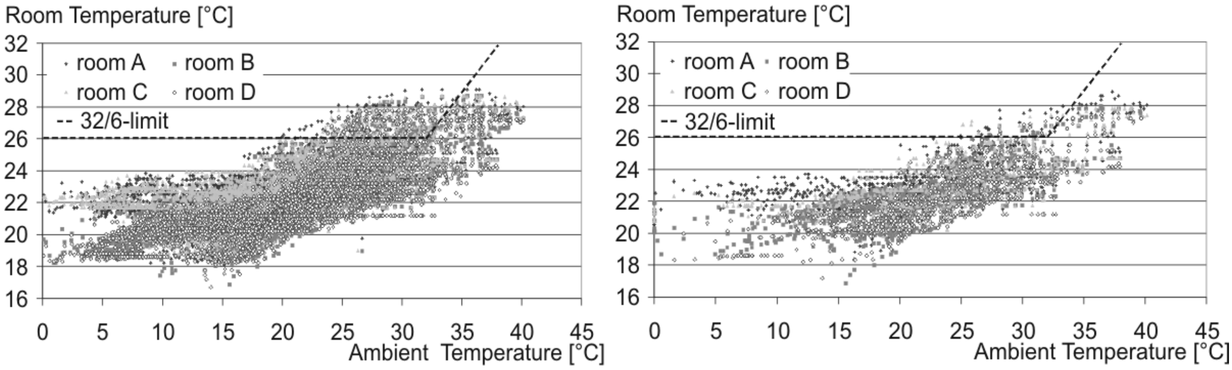


Figure 7: Average hourly air temperature of four passively cooled offices from May 2003 to October 2003 plotted over the actual ambient air temperature. Data including all recorded hours (left) indicate a number of hours when the comfort limit was exceeded. When considering the attendance times (right) there were rarely any temperatures above the limit. The left picture also shows the lower limit of 18°C for the night cooling.

In 2004 the average time above the limit conditions was 0.4% (9 hours), in 2005 2.1% (38 hours) of the total attendance time (Figure 8). The maximum temperature recorded in these offices was 27.2°C during the summer of 2004 and 28.2°C in the summer of 2005.

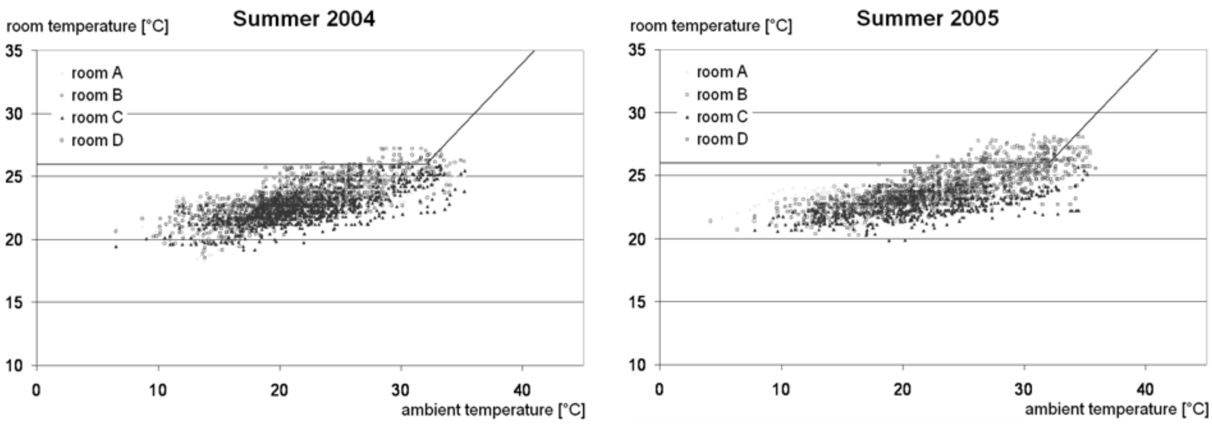


Figure 8: The average exceeding hours for the same four rooms are 9 in summer 2004 and 38 in summer 2005

In summer 2006 however the 60 hour limit was exceeded in different rooms. The reasons were malfunctions of the atrium lid and therefore a failure of the night ventilation. In combination with the very warm nights during June there had been one week with temperatures up to 30°C in the offices. This shows how important the proper operation of the building is if there is no active cooling system for backup.
Though being just a spot sample, the results of the four rooms showed that passive cooling is possible even with extreme weather like in 2003 if the building is operated in the right way.

Air quality

In 5 rooms CO_2-concentrations have been measured additionally for checking if sufficient air quality was achieved just by natural ventilation. Two of these offices were ventilated by air conditioning, the other three by natural ventilation. The results of more than two years monitoring showed that the limit of 2000 ppm was never reached, the limit of 1500 ppm was exceeded just for a few hours - independent of the type of ventilation [5].
On average, the CO_2-concentration was below 1500 ppm for 99.8% and less than 1000 ppm for 98.6% of attendance time. Figure 9 confirms that in every room the concentration was below 800 ppm at least in 90% of the attendance time. There was no significant difference between naturally and mechanically ventilated spaces
The analysis showed that the employees' manual use of the skylights did not depend on the ambient temperature. As an example, on average the skylights were open as often in January 2005 as in June 2005. In contrast, the manual opening of windows was dependent on the outside climate. In June 2005 occupants used the windows more than three times as often for ventilation as in January 2005. With a decreasing frequency of opening the windows in winter, the chance to measure higher CO_2-concentrations rose (Figure 10).

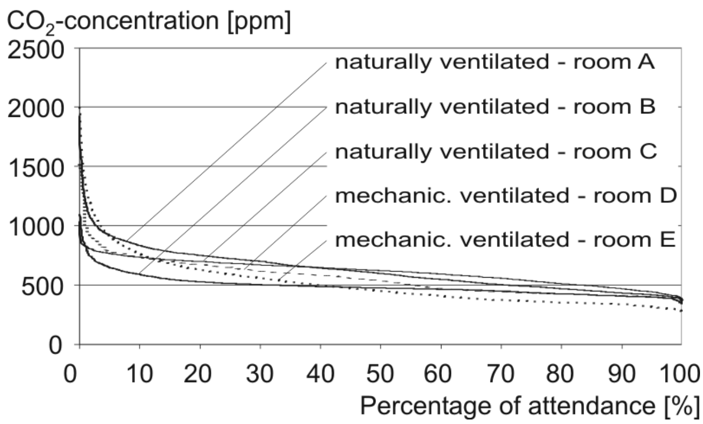


Figure 9: Cumulative frequency distributions for hourly averaged CO_2-concentration of air in 3 naturally and 2 mechanically ventilated offices. Data for one year, only hours with attendance of the occupant

DISCUSSION

The 3-year monitoring of the KfW bank building showed that the ambitious target of a primary energy consumption of 100 kWh m^{-2} a^{-1} was almost reached after the third year with a prediction to fall below this value in the future. Compared to conventional office buildings in Germany this building shows an outstanding energy performance.
Not conceptual errors but failures in operating the building's facilities caused the higher energy consumption in the beginning. This is in accordance to the results of the other

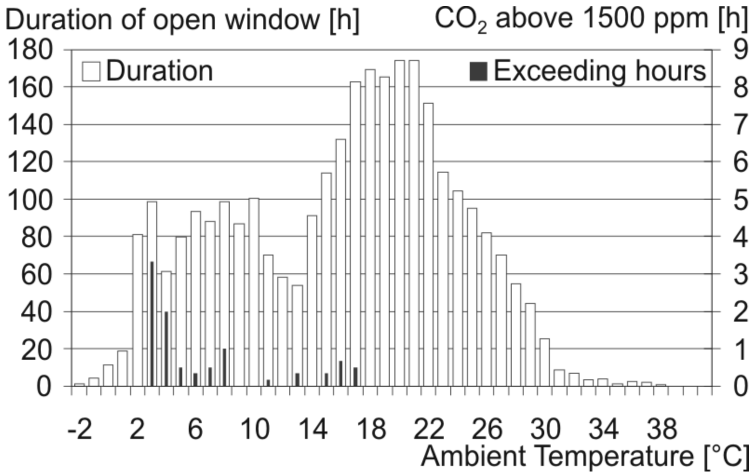


Figure 10: Total time of window opened with CO_2-concentration exceeding 1500 ppm for an office over one year, classified in steps of ambient temperature

demonstration buildings of the programme and underlines the importance of a continuous monitoring of a building. Important preconditions for this are appropriate data management and visualisation as well as experienced facility managers who are familiar with energy performance issues.

It could also be proved that even under extreme climate conditions – like summer 2003 – acceptable thermal conditions in the passively cooled offices could be achieved. The proper operation of the shading system and the night cooling are important to keep the room temperatures within the required limits.

Examinations on the CO_2-concentration in sample rooms have shown that without mechanical ventilation good air quality was obtained.

ACKNOWLEDGEMENT

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Thermal Comfort Measurements in a Hybrid Ventilated Office Room

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SUMMARY

Global climatic warming trends are becoming more and more obvious. They are characterized by longer and more intense heat wave periods as witnessed in extreme by the summers 2003 and 2006. This situation asks for mechanical cooling applications in office buildings. Is it possible to reach good indoor environmental conditions using hybrid ventilation strategies only? This question has been investigated by measurements in the *Forum Chriesbach*, a new low energy office building located in Dübendorf, Switzerland. The ventilation concept of this building is based on an earth-to-air heat exchanger system for hygienic air supply during day-time and a passive night cooling strategy by stack ventilation through window openings in the façade and towards a large atrium with roof outlets. Outdoor and indoor climatic conditions as well as internal heat loads have been measured in an office room over the summer heat wave period 2006. The measured data shows, that the hybrid ventilation system is able to hold the operative room temperatures during the heat wave period 2006 in the acceptable comfort range of ±0.5 PMV and to meet the requested indoor air quality limits (CO2 < 1000 ppm). It thus confirms the thermal simulation results made by the project design team.

INTRODUCTION

Hybrid ventilation, the combination of natural and mechanical ventilation, is an encouraging strategy to achieve energy-efficient office buildings with good indoor air quality and thermal comfort. International studies carried out within IEA Annex 35 “Hybrid Ventilation in New and Retrofitted Office Buildings”[1] have shown, that there exist hybrid ventilation solutions which together with other sustainable technologies such as daylighting, passive solar heating and passive cooling strategies may be able to provide good indoor environmental conditions and low energy use in office buildings in moderate climates like those in central Europe. For this kind of buildings an integral approach in the design of the energy, indoor climate, fire safety and security concept is required. Since low-energy buildings interact strongly with solar radiation and indoor load conditions an appropriate overall robust control strategy has to be applied which includes a particularly effective sun protection system. In addition a higher user satisfaction concerning the thermal comfort can be attained when an individual control of the windows is possible.
Air cooling in summer and preheating in winter with buried pipe systems acting as earth-to-air heat exchangers are becoming popular in Switzerland. Studies made at the University of Geneva [2] have shown good energy saving potentials for office buildings at competitive costs. Passive cooling by night ventilation is mainly used to improve the thermal comfort situation in summer and has been applied with success in high mass office buildings. Detailed thermal analysis and simplified design tools have been developed and validated by monitoring the new building of the Fraunhofer Institute for Solar Energy Systems ISE in Freiburg [3]. In a German demonstration and monitoring programme on the energy performance of 22 advanced office buildings without mechanical cooling systems it has been shown, that the yearly primary energy use can be kept within a limit of 100 kWh per m^2 net heated floor area [4] The

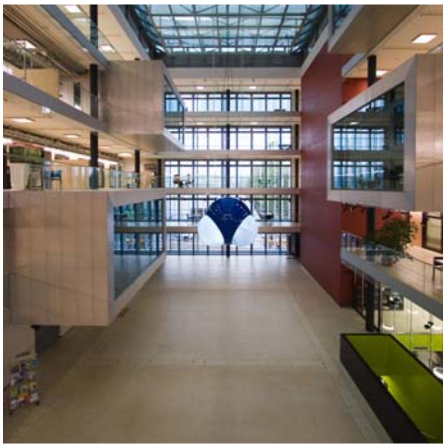
Forum Chriesbach is designed to get along with about 14 kWh/(m^2 a) end energy use respectively 36 kWh/(m^2 a) primary energy use for heat, cold and electricity.

METHODS

Building description

Forum Chriesbach, the new main headquarters of the Swiss Federal Institute of Aquatic Science and Technology (Eawag), is an innovative low energy office building [5]. As winner of an architectural competition, the project team has chosen a holistic approach for the design of a sustainable house of the future, which meets the client requirements concerning healthy indoor environmental conditions and substituting primary energy by renewable energy sources and optimizing the careful use of land, materials, water and finances.
The key elements of the 6-story building with a total volume of 38’615 m^3, a ground surface area of 1’886 m^2 and a total floor surface area of 8’533 m^2 are the highly insulated building envelope, the large atrium buffer zone with a glazed double roof, massive concrete floor constructions and fire escape balconies on the façade with moveable blue glass panels for solar control as shown in Figure 1. The smart energy concept consist of 50 m^2 vacuum solar collectors with a heat storage water tank of 12 m^3, 459 m^2 photovoltaic panels on the roof and a mechanical ventilation system using an earth-to-air heat exchanger for preconditioning the air including a heat recovery system (see Figure 2) with a design ventilation rate of 8.3 l/s per person. In summer an additional passive cooling strategy by night stack ventilation through the windows via the large atrium in the centre of the building (see Figure 3) is used.

a)



b)

Figure 1. View of the office building with a) glass lamellae for solar protection and b) atrium

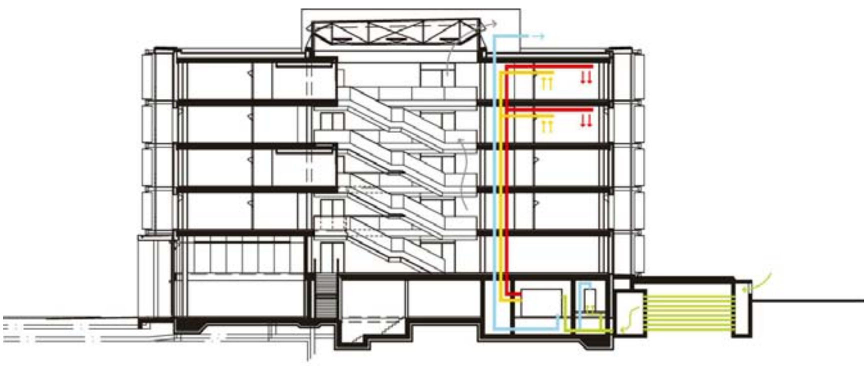


Figure 2. Day time mechanical ventilation via earth-to-air heat exchanger with buried pipes

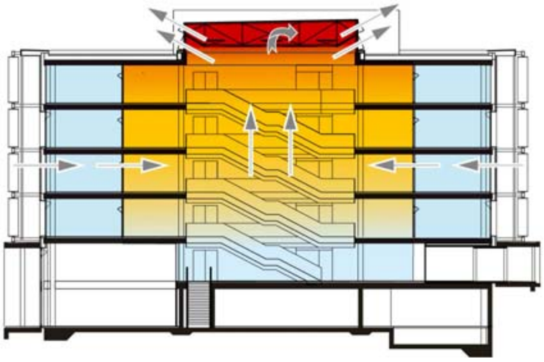


Figure 3. Night time passive cooling by stack ventilation via the atrium to the roof outlets

Measurement layout

The measurement of the indoor environmental conditions has been made in the office room FC-D41 with 30 m^2 floor area in the south-west corner of the 4th level (see Figure 4). The room has 4 working places and was occupied by 1-3 persons (1.2 met, 0.5 clo). The data acquisition system used consists of a comfort analyzer B&K 1213 complying with the requirements given in ISO 7726 [6] and two separate data loggers for the indoor air quality and the electric power measurements. The specifications of the sensors used are given in Table 1. The goal of the measurement was to check if the comfort and indoor air quality criteria given in ISO 7730 [7] and EN 15251 [8] are met during the heat wave period in summer 2006.

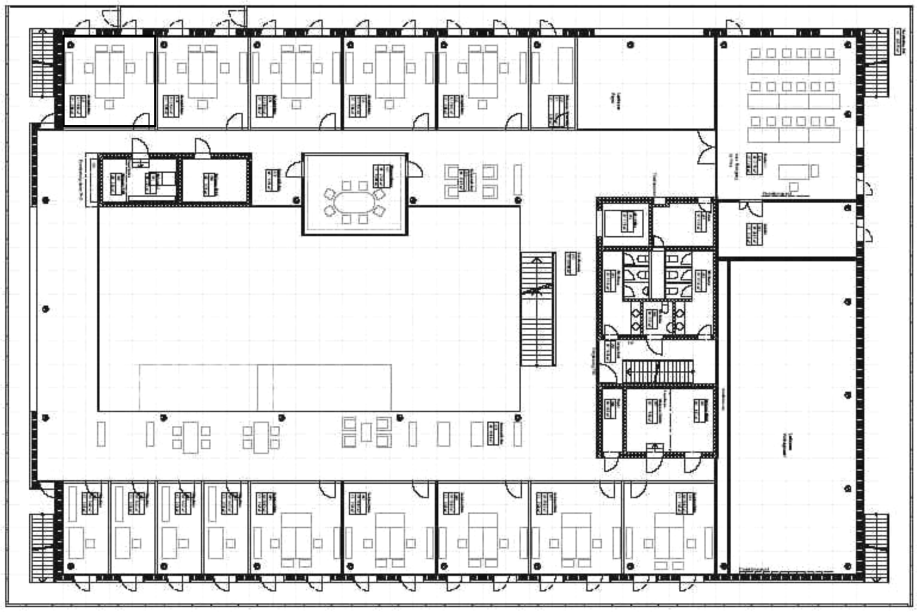


Figure 4. Floor level D with the measured office room D41 in the south-west corner (yellow)

Table 1: Measurement sensors with specifications

Quantity	Sensor	Transducer	Unit	Accuracy
Air temperature	Pt100	MM0034	°C	± 0.3 K
Operative temperature	Pt100	MM0060	°C	± 0.3 K
Air velocity – anemometer	Hot wire	MM0038	m/s	± 0.05 m/s
Supply air temperature	Pt100	Hygroclip	°C	± 0.1 K
Relative humidity	Capacitive	Hygroclip	%	± 3 %
CO2 concentration	Vaisala NDIR	GMW 22D	ppm	± 5 %
Global solar radiation	Kipp & Zonen	CM21	W/m^2	± 2 %

RESULTS

The measurements in room FC-D41 have been started on July 15th and included a heat wave period of 17 sunny days with maximum outside air temperatures up to 35°C, followed by a colder period characterized by a temperature drop of 15 K (see Figure 5 and 6). During this heat wave period, the cooling effect of the earth-to-air heat exchanger is demonstrated in Figure 7: During daytime the supply air temperatures lay 7-10 K lower than the outdoor air temperatures and hardly exceeded the upper comfort limit temperature of 26°C. The operative temperatures are in the requested comfort range and reach at the end of the heat wave period the upper limit of 26 °C (see Figure 8). The relative humidity lies during daytime between 40-60% and during night between 60-80% due to the temperature drops caused by the night cross ventilation (see Figure 9). The same picture can be observed with the air velocity in the office

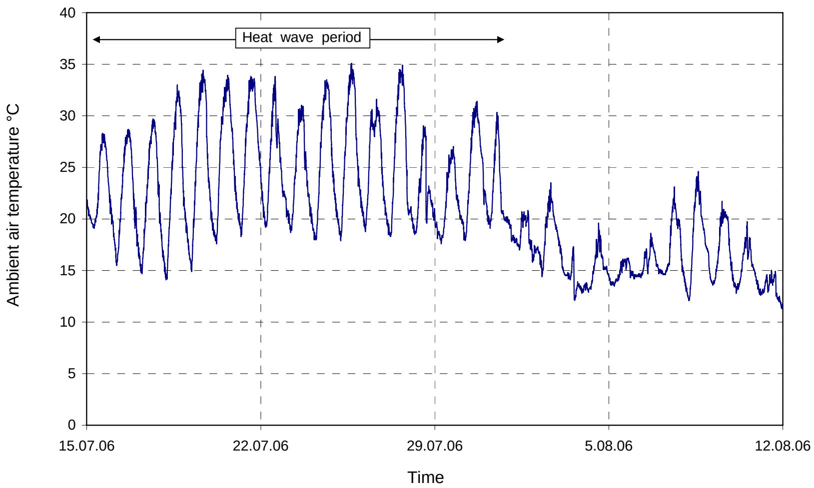


Figure 5: Air temperature in Dübendorf, period 15. July – 12. August 2006 [§]

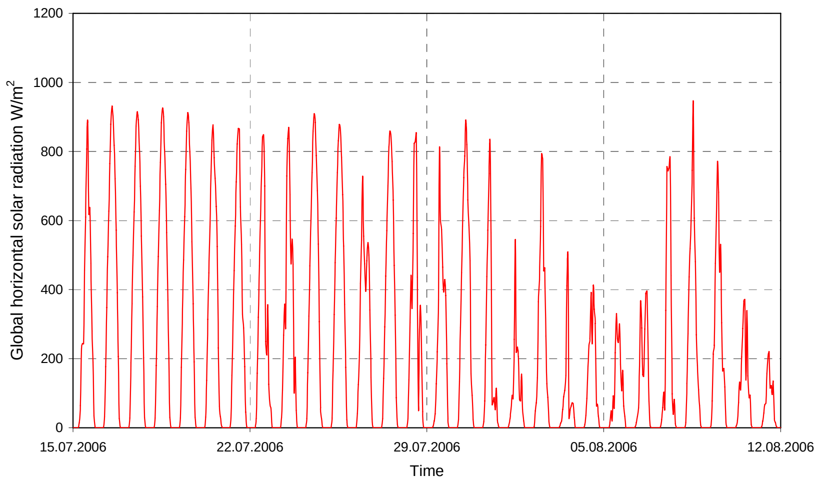


Figure 6: Solar radiation in Dübendorf, period 15. July – 12. August 2006

[§] Measured by the NABEL monitoring network station in Dübendorf (BAFU / Empa) room (Figure 10), which shows some peaks due to controlled window openings during night and some openings in the morning by the occupants.

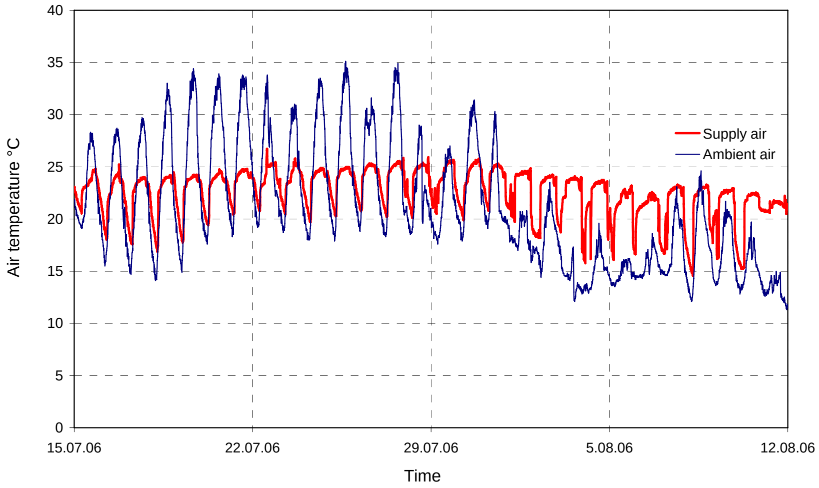


Figure 7: Supply air temperature versus ambient air temperature

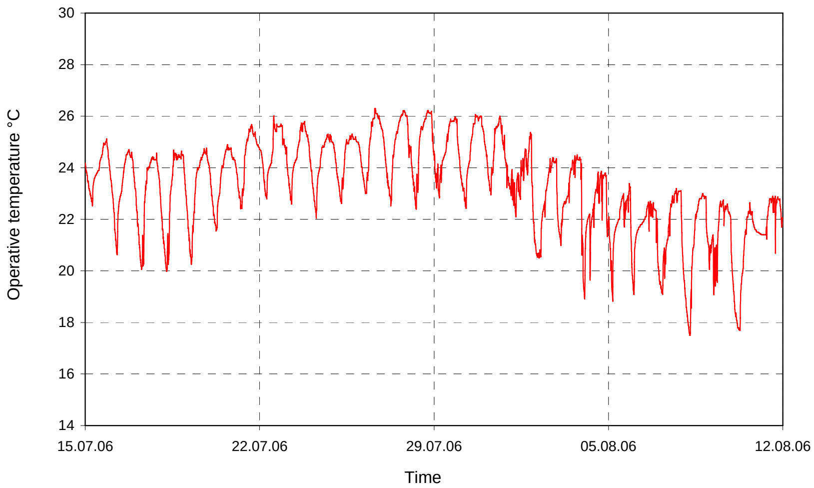


Figure 8: Operative temperature in the office room FC-D41

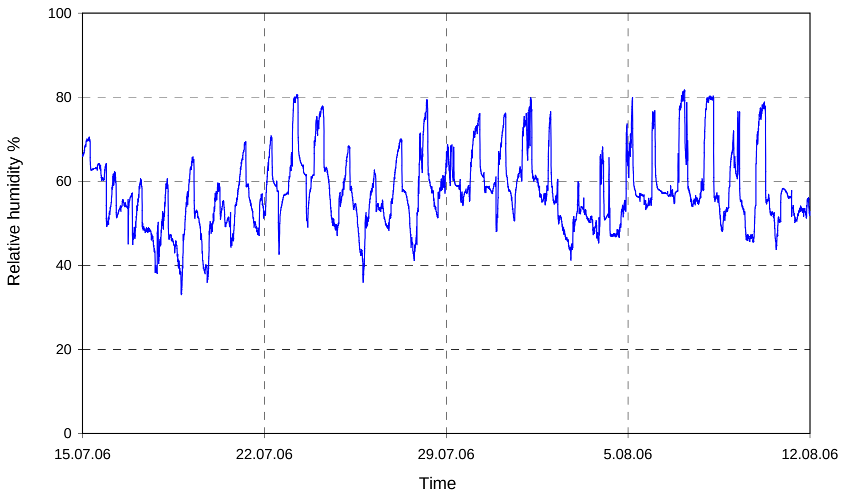


Figure 9: Relative humidity of the supply air in the office room FC-D41

The CO2 concentrations shown in Figure 11 are in an acceptable range, the chosen design supply air flow rate of 17 l/s for the room is obviously sufficient. The total internal heat loads from people, equipment and lightning are given in Figure 12.

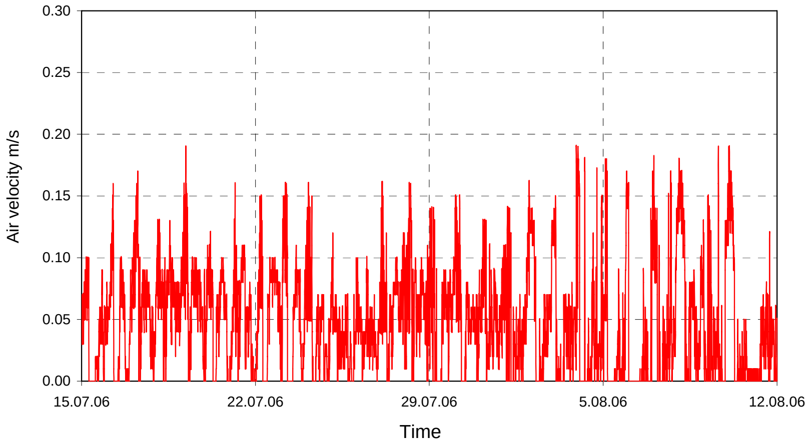


Figure 10: Air velocity in the office room FC-D41

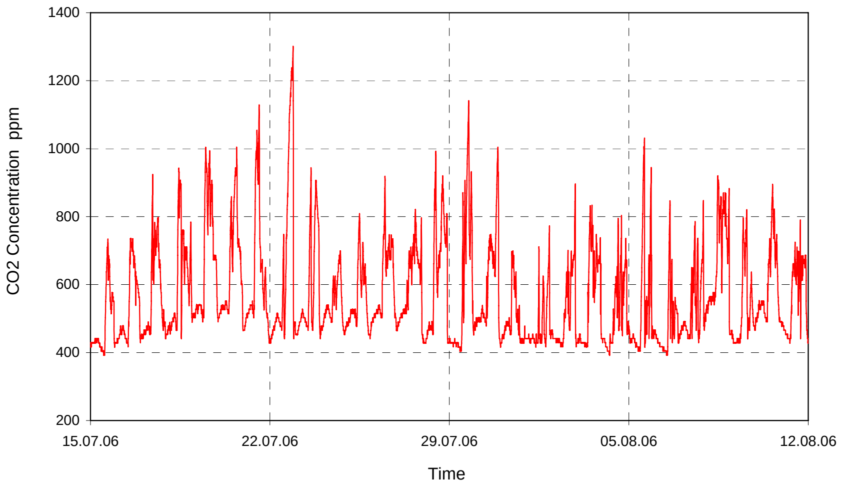


Figure 11: CO2 concentration in the office room FC-D41

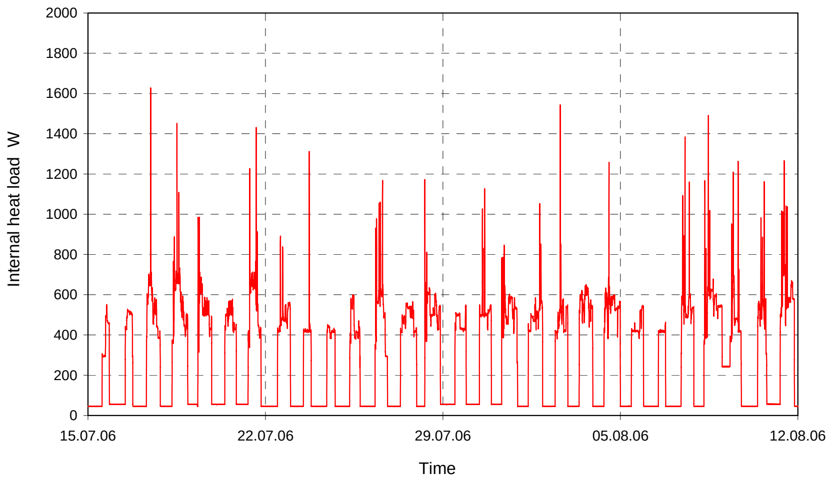


Figure 12: Total internal heat loads in the office room FC-D41

All measurement results are summarized in Table 2. In Figure 13 and 14, the indoor environmental conditions are classified according to EN 15251 for the 17 heat wave days, occupied from 8 am to 9 pm. The operative temperatures lie during 86% of the hours in the category I band (predicted mean vote PMV ± 0.2) and the indoor air quality criteria is met during 91% of the hours in the best category I ($CO2 \leq 750$ ppm).

Table 2: Measured data over the summer heat wave period 15. July – 1. August 2006

Parameter	Mean value	Maximum value
Outdoor environemt		
- Ambient air temperature (10 minutes data)	24,3 °C	35,1 °C
- Relative humidity of the air (10 minutes data)	57,7 %	96,1 %
- Global horizontal solar radiation (hourly data)	271 W/m^2	931 W/m2
Indoor environment		
- Daily internal heat loads (people, equipment, lighting)**	18,4 W/m^2	20,5 W/m^2
- Daily hours of occupancy and operation	11,5 h	15,5 h
- Daily internal heat gains (people, equipment, lighting)	211 Wh/m^2	292 Wh/m^2
- Room air temperature (6-minutes data)	24,1 °C	27,0 °C
- Operative room temperature (6-minutes data)	24,2 °C	26,3 °C
- Rel. humidity of the room air (6-minutes data)	57,1 %	80,6 %
- Air velocity (6-minutes data)	0,053 m/s	0,190 m/s
- CO2 concentration (6-minutes data)	585 ppm	1301 ppm

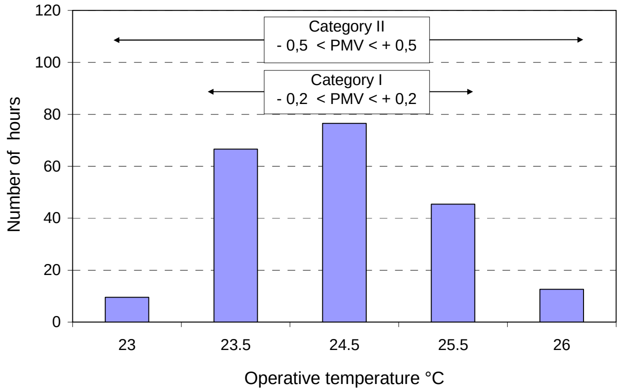


Figure 13: Operative temperature distribution over the period 15. July – 1. August 2006 **

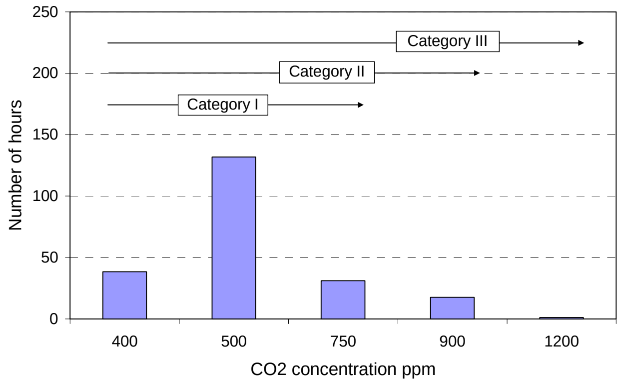


Figure 14: CO2 distribution over the heat wave period 15. July – 1. August 2006 **

** during hours of occupancy from 8 am to 9 pm all days

DISCUSSION

Low energy office buildings like the *Forum Chriesbach* tend to be strongly dominated by internal heat gains from people and electrical equipment. They are very welcome in winter to cover the heating demand, but can be a problem in summer leading to an unwanted cooling demand. The use of a hybrid ventilation concept is one possibility to overcome this summer problem. In the case of the *Forum Chriesbach* building, an earth-to-air heat exchanger system is used to satisfy the hygienic ventilation during occupancy time together with a passive night ventilation to use the high heat storage capacity of the massive concrete building structure. The extensive heat wave period in summer 2006 has been used to check if the *Forum Chriesbach* building is able to achieve the requested indoor climatic conditions. The measurement of the thermal comfort and the indoor air quality parameters has shown that the best comfort category I defined in the new European standard EN 15251 has been met during 86% of the occupied time and that the category II, which complies with the Swiss standard is fulfilled all the time. This result is remarkable since normally one would expect that with passive cooling principles some cut backs in the thermal comfort level have to be accepted. The measurement also demonstrated that the internal heat gain situation was higher than expected due to higher numbers of working hours and that the large ambient temperature drop after the heat wave period was not taken into account by the night ventilation scheme. Demand controlled ventilation becomes therefore a more important topic in order to react to this kind of unforeseen situations.

ACKNOWLEDGEMENT

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Natural Ventilation System for a School Building Combined with Solar Chimney and Underground Pit

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SUMMARY

The specific features of natural ventilation system combined with solar chimney and underground pit installed in a new school building and the measured results for four years are described. In summer and intermediate seasons outside air is introduced into the building of four stories through underground pit from the intake provided in the north end, brought into the different occupied rooms from which the room air is discharged to outside through the solar chimneys of 8 meters high above the roof by chimney action or pulling out action by wind. The measured results for four years after opening the school showed that the energy performance has improved year after year owing to the natural ventilation system. Some consideration on the improvements in the system performance is given for future design of natural ventilation with solar chimney.

1. INTRODUCTION

As can be seen in the minarets of vernacular buildings in hot countries, solar chimney is like the one to give rise to buoyant force by solar energy for natural ventilation. This effect could be enhanced by coupling with the underground pit where outside air is cooled down, thus bringing cool air into the occupied spaces.

The natural ventilation system combined with solar chimney and underground pit was introduced in the buildings of Kitakyushu University in the west island of Japan as shown in Figure 1. The performance of natural ventilation with supply and exhaust air volume and cooling effects, the relationship among the weather factors, indoor/outdoor temperature difference and the exhaust volume out of solar chimney for four years is summarized.

Figure 1. Outside view of main building from southwest

Figure 2. Solar chimney on the roof

2. OUTLINE OF BUILDING AND SYSTEM

The main building consists of two long buildings: north wing with office block attached and south wing as shown in Figure 3. The north wing is subdivided into four blocks of identical size. Field measurement was performed with the Block-1 of the north wing building at 10 minutes interval for a long term.

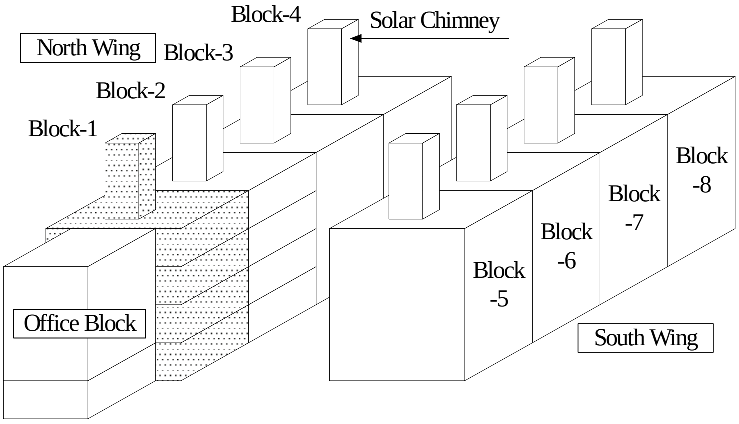

Figure 3. Measured block of north wing

Figure 4 shows the air flow routes during the period of natural ventilation from underground pit to solar chimney on the right side and the air flow routes of auxiliary air conditioning system on the left. Natural ventilation takes place by opening the grilles of outside air intake. During the heating season natural ventilation does not work in order to prevent heat loss, while outside air is introduced through the underground pit warmer than outside. The scheduled operation started since the second year.

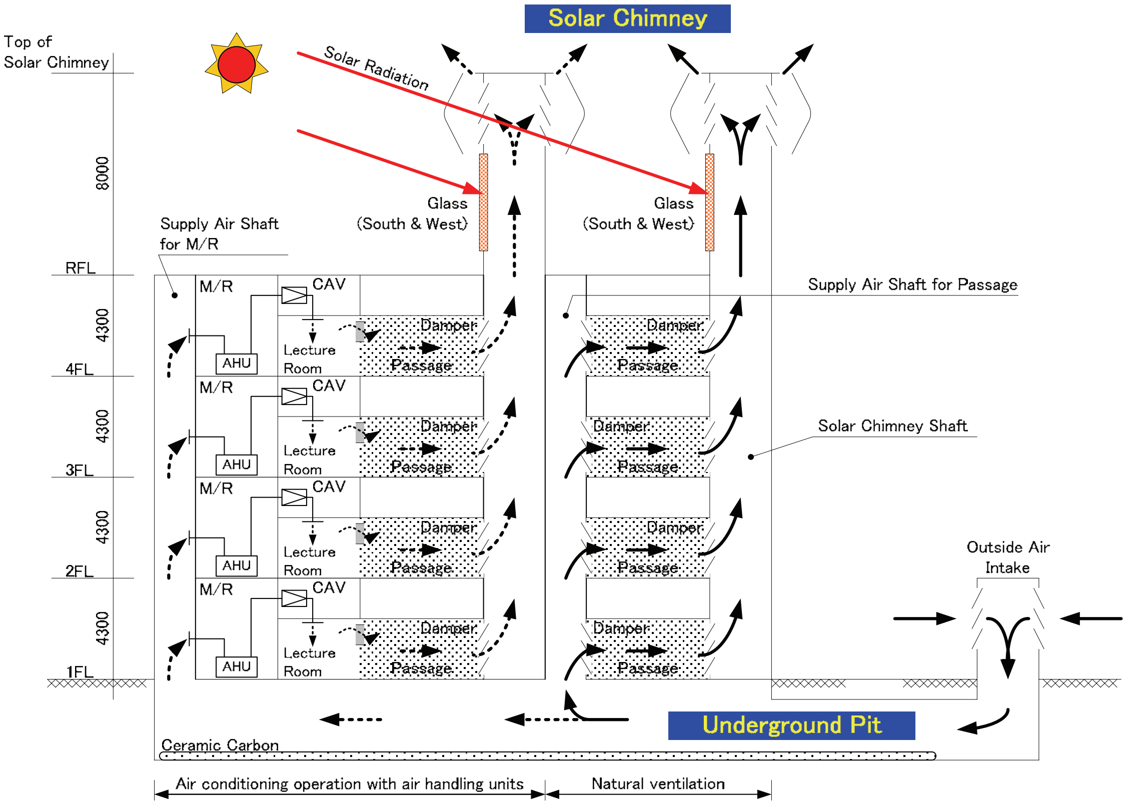

Figure 4. Air flow routes of natural ventilation system (right) and air conditioning system (left)

3. PERFORMANCE OF NATURAL VENTILATION

3.1 Cooling and Heating Periods

Table 1 shows the number of days of natural ventilation during cooling and heating periods. It was found that the cooling period for natural ventilation turned to be greater year after year to occupy 57% of annual total days in the third and fourth years, while in heating period decreased to 43% in the last two years.

Table 1. Number of days of cooling and heating periods

YEAR	1st	2nd	3rd	4th
PERIOD	365	365	366	365
COOLING	185	170	209	211
HEATING	180	195	157	154

3.2 Operation period of Natural Ventilation

Figure 5 shows the monthly number of hours of natural ventilation estimated from the opening hours of the grilles of supply air for the corridor space. The operating hours of natural ventilation in the third year turned out 3,058 hours, while in the fourth year 1,702 hours. This decrease may be attributed to the cool summer in the third year and hot summer in the fourth year. The number of students increased in the fourth year by 25% and this must have affected this difference.

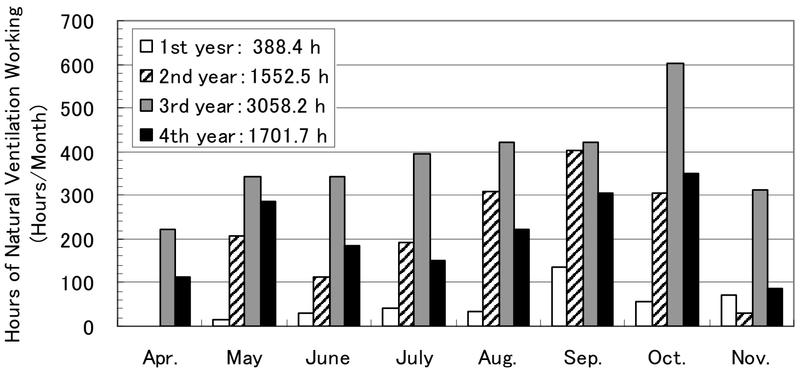

Figure 5. Monthly hours of natural ventilation

Figure 6 shows the percentage of monthly hours of natural ventilation and that of air conditioning operation with air handling units (AHU) in the third year. Except for the month of April and November when switching of heating and cooling operation was made, the percentage of natural ventilation operation during the cooling period was 45-60% in May through September and 80% in October.

Most of the time of natural ventilation took place during the night 18:00 – 8:00, which covered 80% of the total.

3.3 Air flow volume during the period of natural ventilation

In fact quite a large size of exhaust fan is installed on the roof for the toilets of four floors, which affects the performance of natural ventilation. The air flow volume of natural ventilation from the inlet to the underground pit and that from solar chimney to outside are different depending on whether the toilet exhaust fan is operating or not.

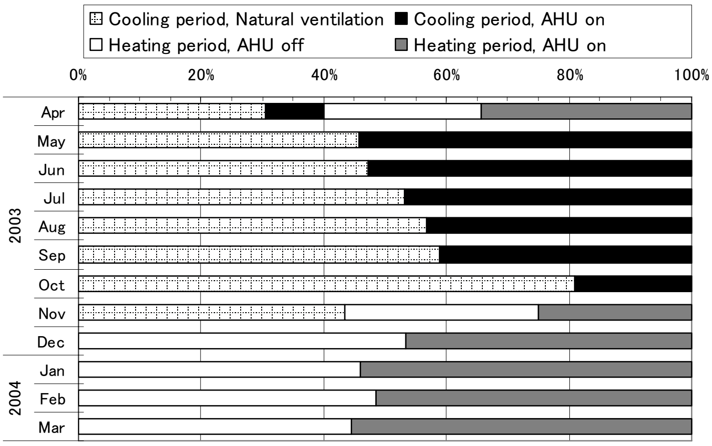

Figure 6. Percentage of hours of natural ventilation and AHU (air handling unit) in cooling and heating periods in the 3rd year

It can be recognized that the supply air volume from the underground pit and the exhaust air volume out of solar chimney tend to increase year after year owing to the improvements of occupants behavior.

The supply air volume from the underground pit of 6,100m^3/h and the exhaust air volume out of solar chimney of 4,100m^3/h are found quite stable for four years.

3.4 Cooling effects under natural ventilation

There are two types of natural ventilation: the first one is the effects of cooling outside air by the underground pit. The second one is the effects of cooling the building skeleton by the outside air after flowing from the underground pit until being exhausted out of the solar chimney.

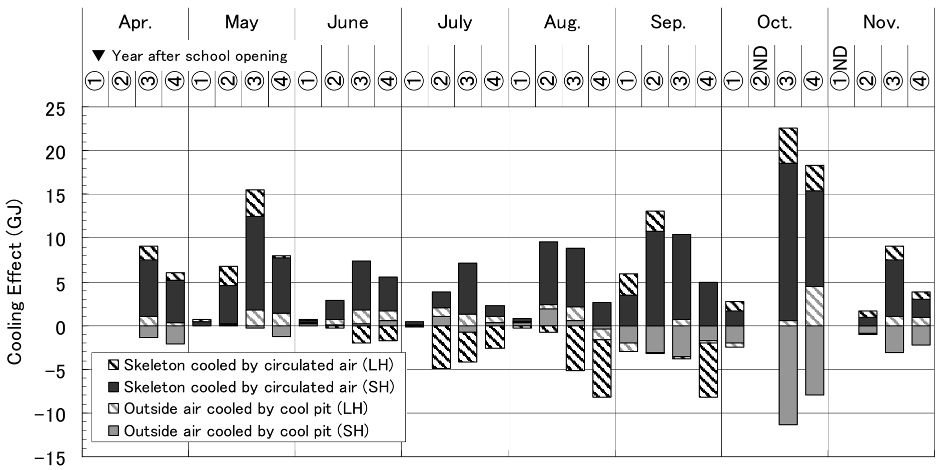

Figure 7. Monthly cooling effects by natural ventilation

Figure 7 and Table 2 show the values of the cooling effect under natural ventilation. It can be found that the sensible cooling effect in the underground pit in spring and summer is negative.

Table 2. Cooling Effects by underground pit

Unit: GJ

Year after school opening	Outside air cooled by underground pit	Skeleton cooled by circulated air	Total
1st	-4.74	10.14	5.40
2nd	1.03	26.59	27.63
3rd	-9.91	68.80	58.89
4th	-7.50	24.80	17.30

This may be attributed to the fact that the underground temperature delays by one month against the outside air temperature in their peak values as shown in Figure 8 and that the cool air introduced from outside during the night is warmed slightly within the underground pit by natural ventilation. However, the slightly warmed outside air is still cool and eventually enters into the building to cool the building structure. When it enters into occupied spaces in winter, the warmed air in the underground pit would bring a natural heating effect.

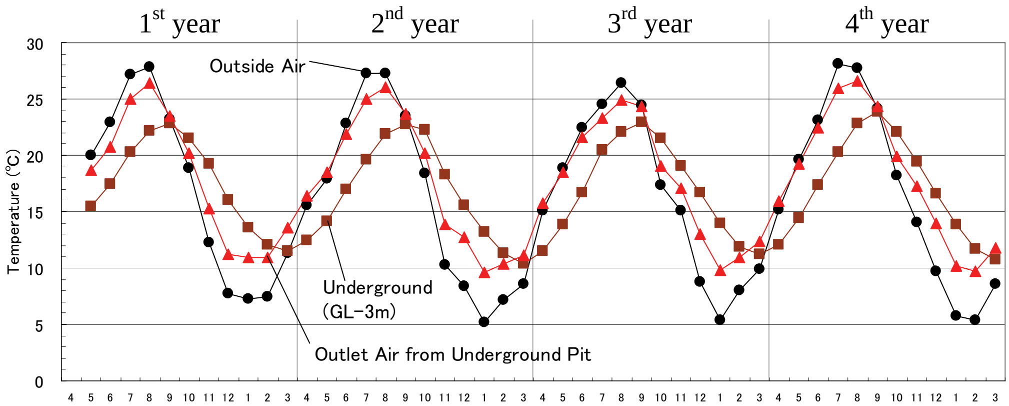

Figure 8. Annual temperature variations of outside air, underground, and outlet air from underground pit

4. AMOUNT OF HEAT EXCHANGE WITHIN THE UNDERGROUND PIT

4.1 Operation hours of AHU

Figure 9 shows the annual variation of air handling unit (AHU) operation hours. It was found that the AHU operation hours in the fourth year during cooling and heating periods increased by 69% and 18% respectively. This was due to the increase in the number of students and hotter summer.

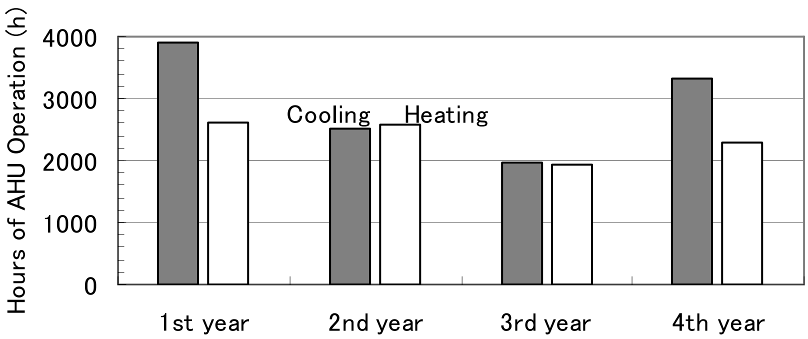

Figure 9. Trend of AHU operation hours

4.2 Amount of heat exchange within the underground pit

The amount of heat exchange within the underground pit in the fourth year was 34.9GJ during cooling period and 20.4GJ during heating period against 39.2GJ and 38.5GJ respectively in the third year.

Table 3 shows the monthly amount of heat exchange within underground pit within the month the amount of heat exchange turned out maximum. The highest value in cooling period of the fourth year may be due to the decrease in air flow speed within the underground pit below 0.5m/s with higher inlet temperature.

Table 3. Monthly maximum amount of heat exchange within underground pit

Period	1st year	2nd year	3rd year	4th year
Cooling	July-01 2.99kJ/m3	July-02 3.00kJ/m3	August-03 4.75kJ/m3	July-04 5.58kJ/m3
Heating	December-01 3.87kJ/m3	January-03 2.98kJ/m3	January-04 4.06kJ/m3	February-05 2.97kJ/m3

The greatest monthly amount of the heat exchange of the cooling period occurred in August with 5.58kJ/m^3 and that of the heating period in January with 2.97kJ/m^3 in terms of total heat exchanged against the air volume through the underground pit in monthly cumulative values.

This suggests that the performance of heat exchange within the underground pit has not been degraded in comparison with that in the past three years.

4.3 Percentage of reduction in energy for air conditioning

Table 4 shows the annual variation of energy for air conditioning recorded in the central monitoring system for the building of north wing and office block. Energy for air conditioning was 2,201GJ in cooling period and 4,077GJ in heating period in the fourth year. The number of students has increased every year after school opening.

Table 4. Trend for energy for air conditioning in north wing and office block

The year after school opening	Period of data	Energy for air conditioning			Number of registered persons			Energy for air conditioning per registered person	Energy for air conditioning per floor area (1～4F) *
		Cooling period (GJ)	Heating period (GJ)	Total (GJ/Year)	Office block (Persons)	North wing (Persons)	Total (Persons)	(GJ／Person·Year)	(GJ／m^2·Year)
1st	May/2001 ～Apr./2002	1,919	3,630	5,549	48	329	377	14.72	0.275
2nd	May/2002 ～Apr./2003	1,734	3,456	5,190	41	598	639	8.12	0.257
3rd	Apr./2003 ～Mar./2004	1,278	3,389	4,667	46	920	966	4.83	0.231
4th	Apr./2004 ～Mar./2005	2,201	4,077	6,278	47	1207	1254	5.01	0.311

*Floor area of measurement : Office block = 2659.75m^2、North wing = 17543.75m^2、Total 20203.50m^2

Energy used for air conditioning in the third and fourth years was 4.83GJ/person/year and 5.01GJ/person/year respectively or 0.231GJ and 0.311GJ/m^2 year, which were found smaller than the preceding two years.

5. DISCUSSION

Some consideration on the improvements in the system performance is given for future design of natural ventilation with solar chimney.

1. In the design of the solar chimney, it is primarily important to reduce friction loss for the sake of obtaining higher driving force of solar chimney. The solar chimney of this building had bent portions in the course of air flow due to some problems against planning scheme.

2. The simple method to find an optimum shape of solar chimney in terms of height and horizontal dimensions should be proposed.

3. The optimum relationship between the size of solar chimney and the length of underground pit should be identified. The section and length of the underground pit of this building could have been a little smaller.

4. Proper combination of natural ventilation system and auxiliary air conditioning system should be investigated. The system in this building seemed a little too complicated.

5. Behaviour of occupants should be taken into account in the system design. For example mosquito screen should have been provided for the windows to be opened.

6. CONCLUSION

1. The hours of natural ventilation have increased year after year. From the third year to the fourth natural ventilation was performed for 61% of the cooling period of 5,016 hours and the cooling effects during the hours of natural ventilation was 58.9GJ. This is regarded as instruction to the occupants to realize not to switch AHU on whenever natural ventilation could be possible.

2. The total amount of heat exchange within the underground pit while AHU operating in the third year after school opening was found 71.7GJ. The monthly cumulative value of the heat exchanged within the underground pit was 4.75kJ/m^3 in cooling period and 4.06kJ/m^3 in heating period.

3. Energy used for air conditioning in the north wing and office block in the third year after school opening was proved to be 4.83GJ/ person/year or 0.231GJ/ m^2/year, which was lower than that of the previous year because of an increase in the number of students.

4. The effect of reducing the fresh air load by the use of underground pit under AHU operation in the third year after school opening could be recognized as 30.7% in cooling period and 11.3% in heating period. Percentage of reduction on the energy used for air conditioning of the north wing and office block turned out 12.3% in cooling period and 3.8% in heating period.

ACKNOWLEDGEMENT

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Performance Estimation of Window-Mounted Solar Heat Driven Ventilation System by Numerical Analysis

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SUMMARY

This research proposes a simple device added to a window. The device improves indoor ventilation using solar heat. It made from shoji paper, which is the traditional paper of Japan. The structure of the device is five air layers from five partitions of shoji paper. The device provides insulation. In summer, it is used for ventilation. In winter, it is used for insulation at night. Experiments were conducted with the device, and the ventilation performance and thermal insulation were confirmed. This paper reports the results of a numerical analysis of the device installed in a building model. The difference of climate regions and direction of the building were considered in the numerical analysis. The amount of ventilation and the room temperature of the building in six regions were confirmed.

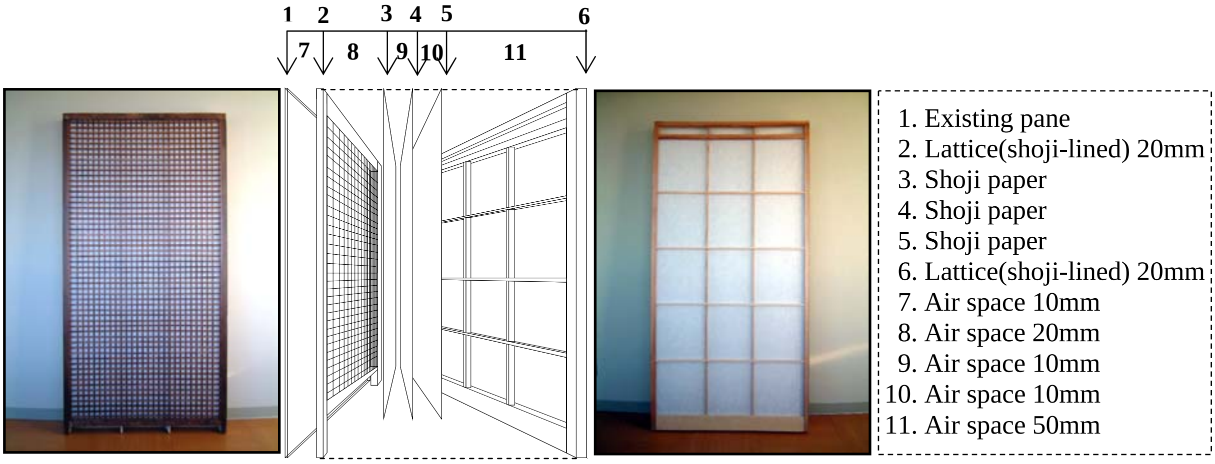
INTRODUCTION

In recent years, various window systems such as Double-Skin and air ventilation windows have been developed with the intent to provide a comfortable indoor environment. These indoor environments have been designed with devices that block the outside environment from entering. As a result, rooms are airtight, highly insulated, and maintain a constant environment. However, one problem is that the devices are complicated. Here, a simple device using solar heat as the driving power of ventilation is proposed. It maintains indoor comfort using changes in the outside environment, as is similar to traditional Japanese construction.

METHODS

Outline of device

Figure 1 shows the designed fitting. It is made from shoji paper, which is the traditional paper of Japan. The structure of the device is five air layers created by the five shoji paper partitions. The multiple air flow of the five air layers due to the shoji paper partitions is installed.
Figure 2 shows the principle of the device. Buoyancy is created by a decrease in the density in the air layers because of the solar heat gain from the outside. This chimney effect is used as the driving power of the ventilation. Indoor air is drawn by the chimney effect, and ventilation is created. The heat generated on the surface of the device is extracted before it reaches indoors. In summer, it is used for ventilation.
Moreover, the device provides insulation when the air layers of the device are sealed up. The midair layers between the shojis do not transmit heat. Therefore, it can be used for insulation at nighttime in winter.



a) b) c)

Figure 1. View of device. a) Outside. b) Detailed cross section. c) Inside.

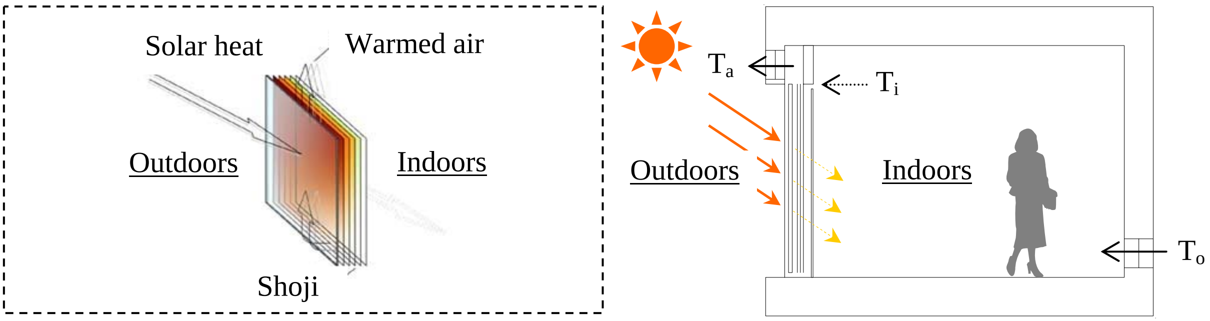


Figure 2. Chimney effect of solar heat.

Performance evaluation experiment of device

The performance of the device was confirmed by experiment. This experiment result was previously reported[1)]. Table 1 a) shows the amount of ventilation to quantity of solar radiation and b) shows various overall heat transfer coefficient. A ventilation experiment was performed to verify the ventilation performance by measuring insolation and circulation. The amount of ventilation obtained by this device was 93 m^3/h. The calculation value was 80 m^3/h. Compared to the calculated value, the experimental value reached a higher value due to the external wind effect. The insulation efficiency of the device was examined to determine whether the performance was adequate for a house. The heat loss experiment was conducted according to the JIS[2)] standard. The overall heat transfer coefficient reached a low value of 1.63 W/m^2K. Moreover, it was a sligthtly smaller value than the value of general fittings. This may indicate that the number of shoji layers should be five in the production fitting to increase the thermal resistance of the air spaces. Therefore, it is thought that this device can be adjusted to most houses because the adiabaticity of the production fitting is more excellent than that of general fittings.

Table 1. a) Amount of ventilation. b) Various overall heat transfer coefficient.

a)

Quantity of solar radiation [W]	Amount of ventilation [m^3/h]	
	Calculated value	Experiment value
733	80	93

b)

Heat street rate [W/m^2K]		
Single window 3mm	Double glass window 12mm	Production fitting
6.00	3.40	1.63

Outline of simulation

The external wind effect was large in the ventilation experiment. The performance of the building model with the installed device was also estimated by a simulation. The performance of the building model was compared for different climate and azimuth conditions. Furthermore, the building models of various conditions were forecasted and the indoor thermal environment of the building models was evaluated.
Figure 2 is a cross section of the building model in the simulation. The floor, wall, and roof are completely insulated. The influence of the insulation heat and the outside temperature is given from the window. Here, the window side of the building model is the azimuth of the building model. The building model was turned to four azimuths facing the south, the north, the west, and the east for the analysis. The building model was analyzed for the weather of six typical areas. Japan is long and has different geographical features in the north and south. Therefore, the climate is greatly different according to the region (Figure 3).
Figure 4 shows the thermal and airflow network model. In the numerical analysis, the thermal and airflow network model simulation program NETS[3)] was used.

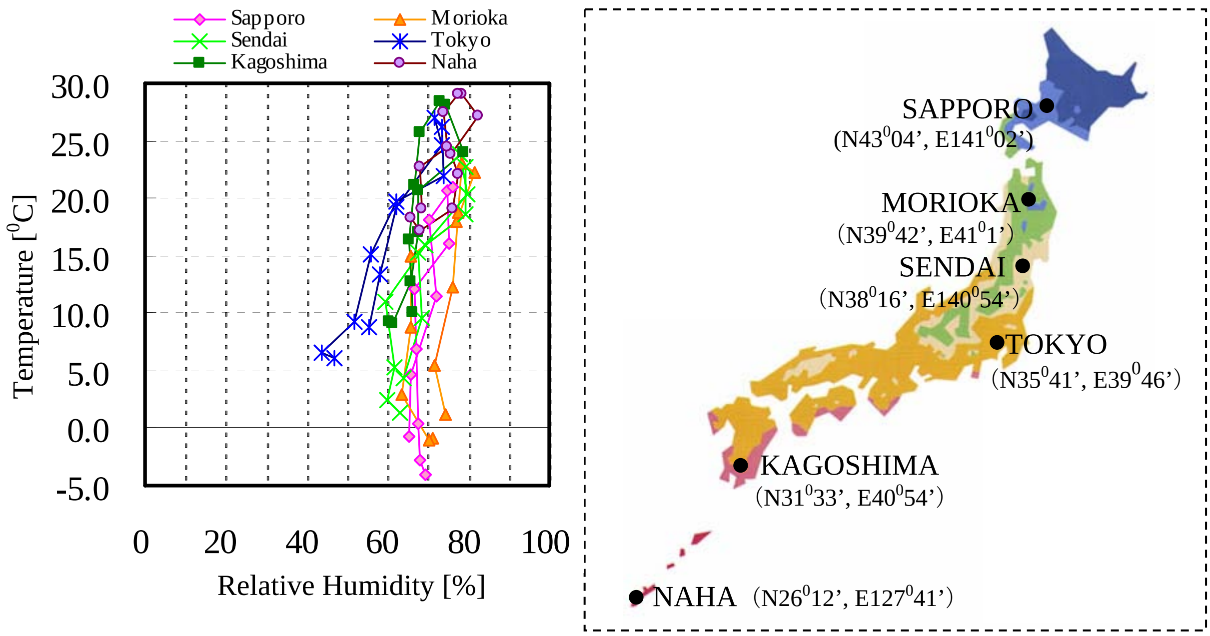


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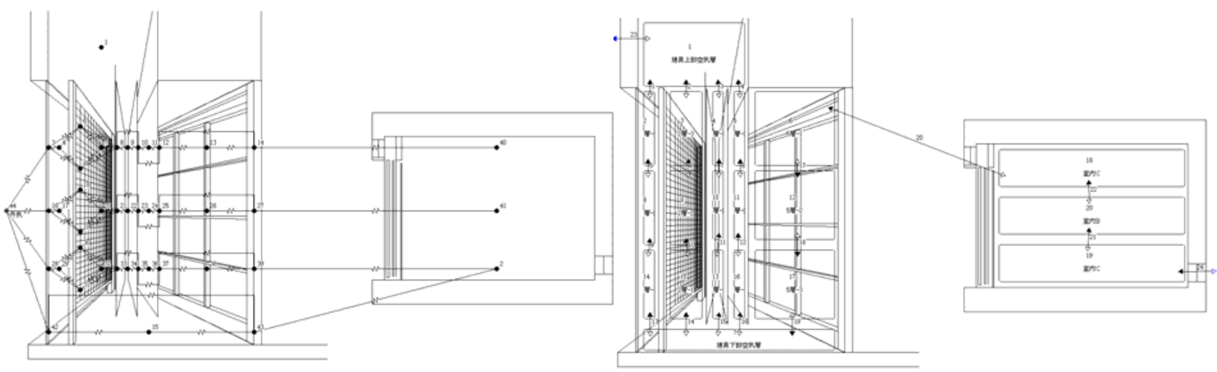
Figure 3. a) Climograph. b) Analytical regions.



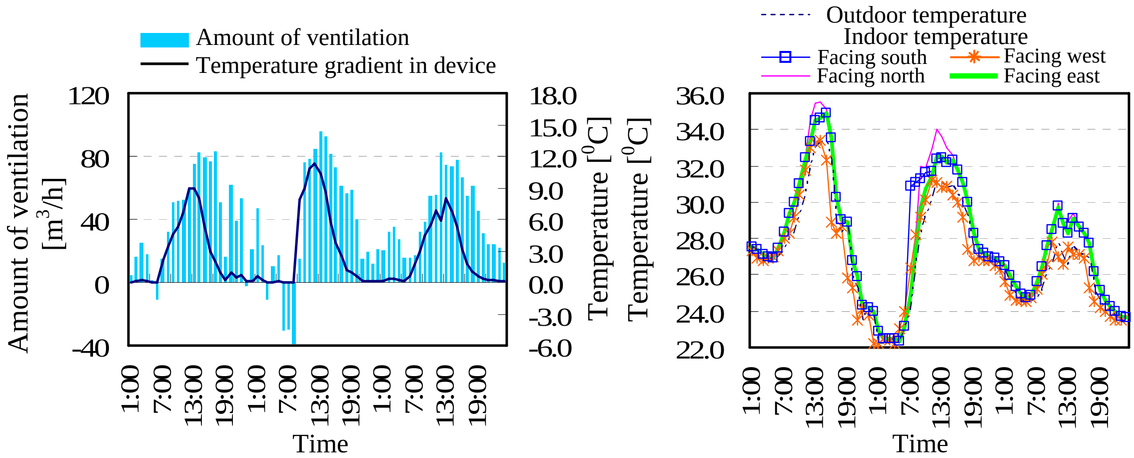
a) b)

Figure 4. a) Thermal network model. b) Airflow network model.

RESULTS

Natural ventilation and thermal environment

Figure 5 a) shows the vertical temperature gradient in device and amount of ventilation. This figure shows the value of the building model facing the south in August in Tokyo. The amount of ventilation follows the temperature gradient. Therefore, ventilation due to the temperature gradient was confirmed. Figure 5 b) shows the outdoor and indoor temperature on the building model for all four azimuths. This figure shows the value of the building model in August in Tokyo. The room temperature of the model facing the east is the lowest, when there is a maximum amount of ventilation.



a) b)

Figure 5. a) Temperature gradient and ventilation performance. b) Outdoor and indoor temperature.

Amount of waste heat

Figure 6 is the scatter chart of the amount of waste heat. This figure shows the value of the building model facing the south and the north in August in Tokyo. There are correlations in the waste calories and the quantity of solar radiation. Ventilation performance is improved when the solar radiation is high, because the amount of ventilation depends on the heat acquisition of the device. However, the amount of waste heat of the building model facing the north was calculated as 50 % of that of the building model facing the south.

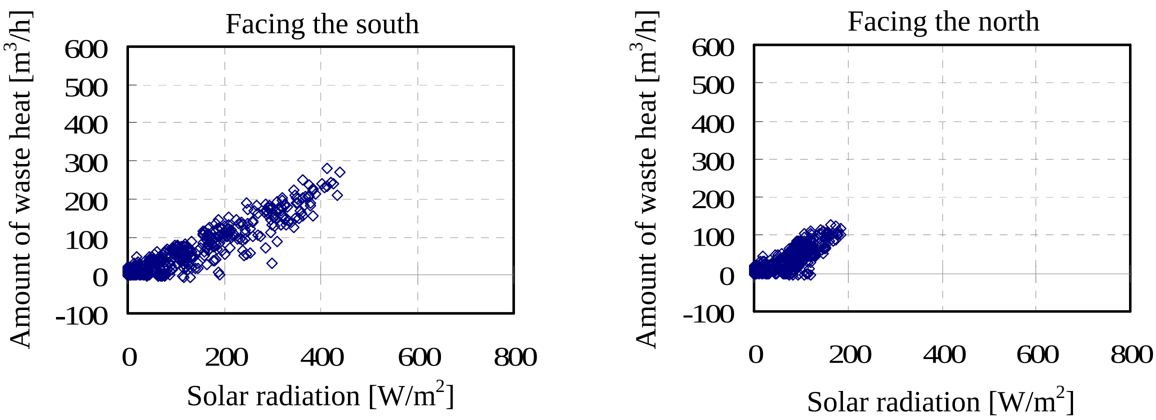
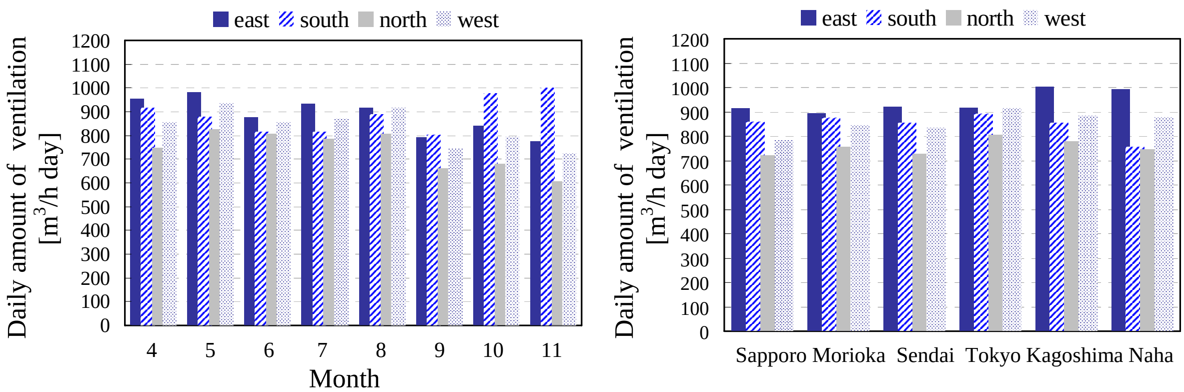


Figure 6. Amount of waste heat to quantity of solar radiation.

Amount of ventilation

Figure 7 a) shows the daily amount of total ventilation from April to November in Tokyo. The building model facing the north has a calculated amount of ventilation from 60% to 90% of that of the other azimuths models in each period. Figure 7 b) shows the daily amount of total ventilation in the six regions of Japan. The model building facing the north has a calculated amount of ventilation from 84% to 98% of that of the other azimuths models in the various regions. The difference of the amount of ventilation by the region of the building model was slight.

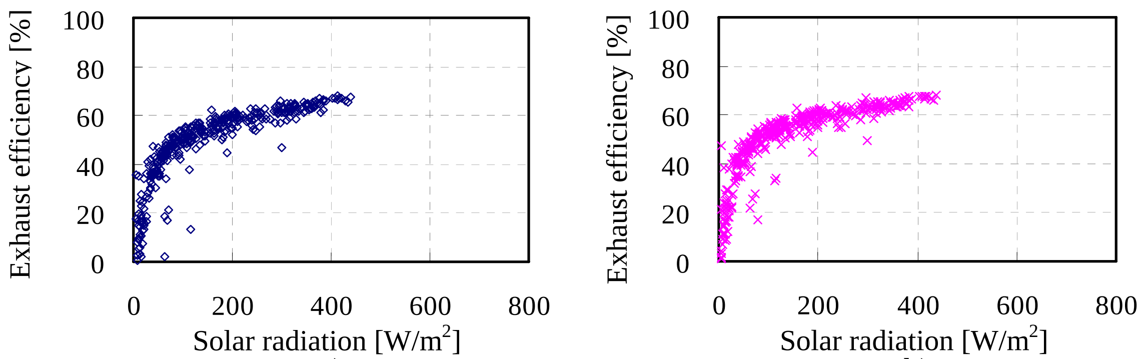


a) b)

Figure 7. Daily amount of ventilation. a) From April to November. b) In six regions in August.

Exhaust efficiency of device

Figure 8 shows the relation of the exhaust efficiency to quantity of solar radiation in Tokyo. It is thought that the width of each air layer in device influenced the amount of ventilation from the result of amount of ventilation. Therefore, performance of the structure which made double of the width of second, third and fourth air layer was confirmed. The exhaust efficiency was calculated from the temperatures fluctuations of T_a and T_i and the temperatures fluctuate of T_a and T_o (Figure 2, (1)). The difference of the exhaust efficiency was slight, when the width of air layers is made wide. When the quantity of solar radiation exceeds 200 W/m^2, the inclination of the scatter chart becomes small. The exhaust efficiency reaches a maximum of 70% as the quantity of solar radiation increases.



a) b)

Figure 8. Relation of exhaust efficiency. a) Before structural modification. b) After structural modification.

$$\text{Exhaust efficiency [\%]} = \left(\frac{T_a - T_i}{T_a - T_o}\right) \times 100\,, \qquad (1)$$

DISCUSSION

This paper next examines the efficiency of the ventilation system on the basis of the simulation results.

Performance of device

There are correlations in the amount of ventilation and the quantity of solar radiation. It was shown that the solar heat helped to drive the power of the ventilation. The amount of ventilation follows the temperature gradient. Therefore, natural ventilation was confirmed. In addition, the heat generated on the surface of the device was extracted by ventilation before it came indoors. Thus, the thermal insulation performance of the device was confirmed. However, the exhaust efficiency by ventilation reaches a maximum when the quantity of solar radiation exceeds 200 W/m^2. Therefore, the difference of the amount of ventilation in the six regions and the four azimuths was slight.

Design of device

The width of each air layer greatly influenced the amount of ventilation. Thermal resistance in the device is large. Therefore, it is necessary to review the design of the device. However, experimental results of the insulation performance may indicate that the number of shoji layers should be five in the production device to increase the thermal resistance of the air spaces. Therefore, the device is thought to have adjustability to various conventional houses.

This research proposes a simple device added to a window with function of ventilation and thermal insulation. The functions demanded from a window are light, view and ventilation. The proposed device must provide a secure, continuous structure from outside to inside. Therefore, the device that provides both ventilation and insulation efficiency was produced. The ventilation performance of the device using a natural resource and the insulation performance of the device using a natural material were confirmed. People will feel the positive effect from connecting to nature by using this device. The technology proposed here can be combined with traditional skills to create a new architectural style for future generations.

ACKNOWLEDGMENT

NETS is thermal and airflow network model simulation program developed by Dr. Okuyama H of Shimizu Corporation Institute of Technology. The author greatly appreciates the offer of NETS.

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Whole year simulation of natural and hybrid ventilation performance and estimation indoor air quality for modernized school building

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SUMMARY

Within the Polish - Norwegian project SUREBILD (Sustainable Redevelopment of Buildings) special attention was paid to find the economically feasible technologies that might improve current poor indoor air quality in Polish schools. Detailed analysis has been performed for selected primary school including different scenarios of modernization of ineffective passive stack ventilation system. The results indicated that hybrid ventilation was the most interesting option from energy point of view. The paper describes whole year simulation of natural and hybrid ventilation performance and estimation of indoor air quality. Detailed simulation with 1 min time step was carried out using CONTAMW simulation environment. The comparisons take into account airflows, CO_2 concentrations and perceived air quality expressed in decipol units. Because of huge amount of data the results are partly aggregated by statistical methods.

INTRODUCTION

In order to support theoretical education by practical application, school constructed nowadays usually incorporate elements of sustainable development. However, as 70% of school buildings in Poland are older than 20 years, existing buildings' modernization is the crucial problem. Within the project SUREBILD many potential technologies for sustainable modernization of these schools have been analyzed. Special attention was paid to find the economically feasible technologies that might improve current poor indoor air quality in Polish schools.

Measurements indicate that ventilation rate is typically within the range 2-6 m^3/(h pupil) while required value is 20 m^3/(h pupil) [1]. Inefficient ventilation systems result in high CO_2 concentrations (up to 4200 ppm) in the classrooms [2]. The project team studied number of different ideas that can improve this situation [3]. A general recommendation for modernization of Polish schools is that more attention should be paid to selection of low polluting finishing materials. In addition, natural ventilation should be supported by adapted mechanical systems in order to assure energy efficiency and comfort irrespective of weather conditions and occupants behaviour. Although the installation of fans and pumps will induce an additional electric energy demand, the possibilities for efficient heat recovery in mechanical systems can reduce the need for heat considerably. Demand control of the ventilation can further improve the energy performance without compromising indoor air quality and comfort.

Although relevant measurements were not carried out in schools in Zgierz, the site inspections, discussions with users and indirect indicators like lower than estimated for standardized conditions fuel consumption indicate that similar situation is present in this school.

METHODS

Current ventilation system

At the moment, the school in Zgierz is equipped with ineffective natural ventilation. Fresh air is supplied through leaky windows and then heated within the classrooms by the space heating system. Exhaust air from the classrooms passes into the school hall via transfer grills. Several stacks located in the hall allow the air to leave the building. The cross sections of the transfer grills and the grills mounted in the stacks are strongly reduced by the operating staff partly for energy conservation reasons and partly due to noise transfer.

Indoor air quality is occasionally improved by opening windows in the classrooms (single side ventilation). In winter this procedure is performed during the breaks between lectures. During warm days in spring and autumn windows are also opened during lectures. Because of the climatic conditions in Zgierz (design range of ambient temperature –20 ÷ +30 ºC), a system based purely on natural ventilation without heat recovery has very poor energy performance during the heating season. However, natural ventilation can be a very good passive mean to provide natural cooling and comfort in the cooling season.

Concepts for ventilation system modernization

The initial architectural concept for rehabilitation of the school involves the introduction of a new atrium that would increase the useful floor area in the building [4]. Experiences with energy performance in hybrid ventilated schools in Norway [5] indicate that in order to find the optimal ventilation concept, combined solutions involving both natural and mechanical ventilation should be considered. In the case of Zgierz school, three basic relevant concepts have been analyzed:

- stack based hybrid ventilation with run around heat recovery,
- decentralised mechanical ventilation with high-efficiency heat recovery,
- mixed mode ventilation.

The energy consumption was estimated for these options (taking into consideration additional variants). Basic assumptions for simulations are presented in table 1.

Table 1. Summary of basic assumptions for different ventilation options.

Option	Air volume [m^3/(h pupil)]	Heat recovery efficiency [%]	Specific fan power [kW/(m^3/s)]
Current situation	20	-	-
Hybrid ventilation	30	~ 60	0.5
Decentralized mechanical system	30	>80	2.5
Mixed mode hybrid ventilation and decentralized mechanical system	30	>80	1.5

The results of the simulations presented in table 2 show that the total primary energy demand (560.4 kWh/m^2 in the base case) can be significantly (~63%) reduced by thermo-modernisation (improved thermal characteristics of the building envelope components, replacement of old coal fired boilers - average efficiency ~50% - by new energy efficient gas boilers; average efficiency ~95% and reduction of thermal loses in the heating network) and by improving building air tightness (additional 4%). Implementation of hybrid ventilation or mechanical ventilation can also strongly contribute to energy conservation respectively up to 76% and 73.3% respectively. Primary energy calculation were performed using primary resource energy factors: 0 for renewable energies, 0.5 for biomass, 2.5 for electrical power and 1.0 for all other fuels.

Table 2. Energy consumption simulations extended to whole year [6].

Case	Heating demand [kWh/m^2]	Reduced heating demand relative to base case [%]	Lighting energy demand [kWh/m^2]	Fan energy demand [kWh/m^2]	Total energy demand [kWh/m^2]	Total primary energy demand [kWh/m^2]	Reduced primary demand relative to base case [%]
Base case	209.2	0.0	15	0	224.2	560.4	0.0
Building rehabilitation	152.9	26.9	15	0	167.9	207.4	63.0
Infiltration red. from 0.4 to 0.2 ach	133.1	36.4	15	0	148.1	185.4	66.9
Hybrid not utilizing culvert	81.1	61.2	15	3.1	99.2	135.4	75.8
Hybrid utilizing culvert	80.4	61.6	15	3.1	98.5	134.6	76.0
Mechanical ventilation	65.4	68.7	15	15.7	96.1	149.4	73.3
Mixed mode ventilation (MV/HV)	65.6	68.6	15	9.4	90.0	133.9	76.1
MV/HV with 0.1 ach infiltration	58.3	72.1	15	9.4	82.7	125.8	77.6

As hybrid ventilation turned out to be very interesting solution, this system has been investigated more detailed. The concept of stack based hybrid ventilation with run around heat recovery applies one or a few central air intakes. Outdoor air passes through refurbished, ground coupled culverts in the basement (figure 1). Axial help fans are placed at the culvert entrances to provide the necessary airflow when natural driving forces are insufficient.

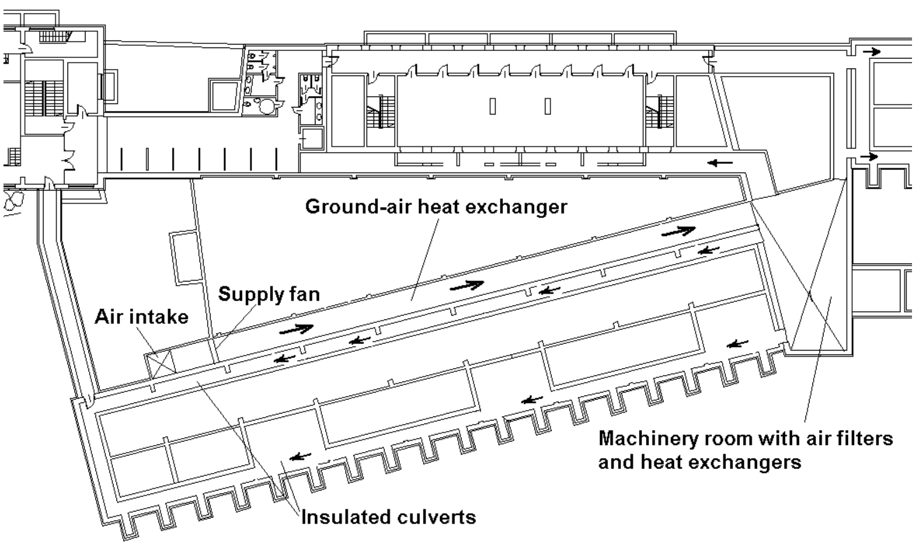

Figure 1. The plan of the basement in primary school in Zgierz after modernization with description of hybrid ventilation – one of the analysed versions [4].

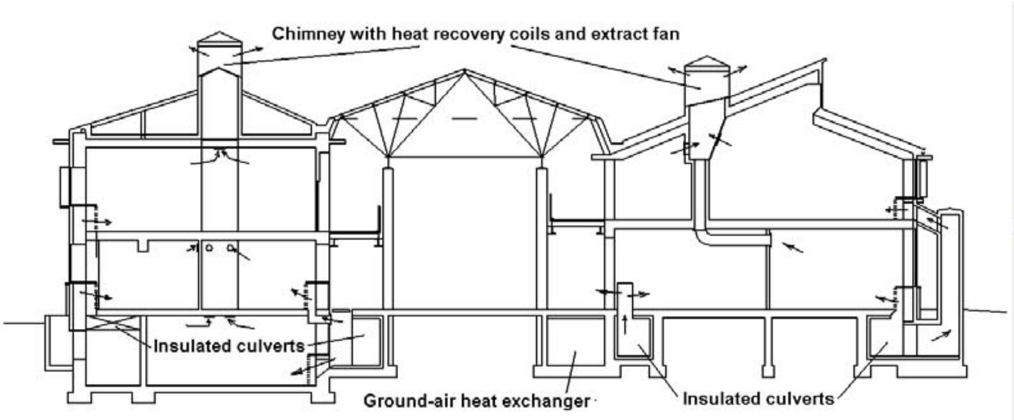

Figure 2. The cross section through two school buildings and the atrium with description of hybrid ventilation – one of the analysed versions [4].

The fans also prevent thermal stratification in the culverts and increase turbulence, resulting in increased convective heat transfer between the supply air and the culvert walls. The air is then filtered before preheating with filters placed at the end of the culverts.

After filtering, the air passes through a first set of heat exchangers connected to similar heat exchangers at roof level via pipes circulating water with some antifreeze solution. The heat exchangers at roof level recover heat from the exhaust ventilation air with an assumed efficiency of 60 %. The supply air then passes through a second set of heat exchangers connected to the central (water) heating system in the building, providing the additional heat needed to reach the desired supply air temperature. The air is distributed to the various rooms in the building via horizontal insulated ducts placed in the intake culverts connected to vertical ducts leading up to the various classrooms. The air is distributed into the classrooms via diffusers for displacement ventilation, and is extracted at ceiling level via bypass openings leading to the corridor.

Central air outlets are connected to the first floor corridors. The ventilation air that enters the ground floor corridors enters the first floor corridors via staircases or other bypass openings (figure 2). In addition to the exchangers for heat recovery, the central air outlets are equipped with demand controlled axial help fans. The control of the supply and extract fans are co-ordinated so that the system operates close to balanced mode.

The design of the exhaust tower/chimney should be given special attention in order to utilize wind as a driving force for ventilation, and prevent cold outdoor air from entering and coming in contact with the recovery heat exchanger, which otherwise would reduce the heat recovery efficiency.

It is also essential that the filter and the heat exchangers have large cross sections to ensure that the required airflow rates can be ensured with minimal pressure drop.
A similar concept has been successfully applied in many Scandinavian school buildings. However, many of these pilot study projects have unnecessary weaknesses that should easily have been avoided. It is therefore important to take advantage from the knowledge gained from these projects if this concept is to be applied at the Zgierz school.

Indoor air quality assessment

As indoor air quality is generally recognized as the key problem in Polish schools, the assessment of contaminant concentration in modernized school was essential for sustainable redevelopment of Zgierz school. As detailed information about properties of finishing materials that could be used during modernization are not available, carbon dioxide generated during room occupation was selected as the characteristic contaminant. The assessment was performed using computer program CONTAMW ver. 2.4 [7]. CONTAMW is a simulation tool designed to determine:

(a) airflows: infiltration, exfiltration, and room-to-room airflows in building systems driven by mechanical means, wind pressures acting on the exterior of the building, and buoyancy effects induced by the indoor and outdoor air temperature difference.

(b) contaminant concentrations: the dispersal of airborne contaminants transported by these airflows; transformed by a variety of processes including chemical and radio-chemical transformation, adsorption and desorption to building materials, filtration, and deposition to building surfaces, etc.; and generated by a variety of source mechanisms, and/or

personal exposure: the predictions of exposure of occupants to airborne contaminants for eventual risk assessment.

Figure 3 presents the sketchpad of CONTAMW program with zonal representation of Zgierz school. Simulations performed with 10 min time step covered all rooms in modernized school [8].

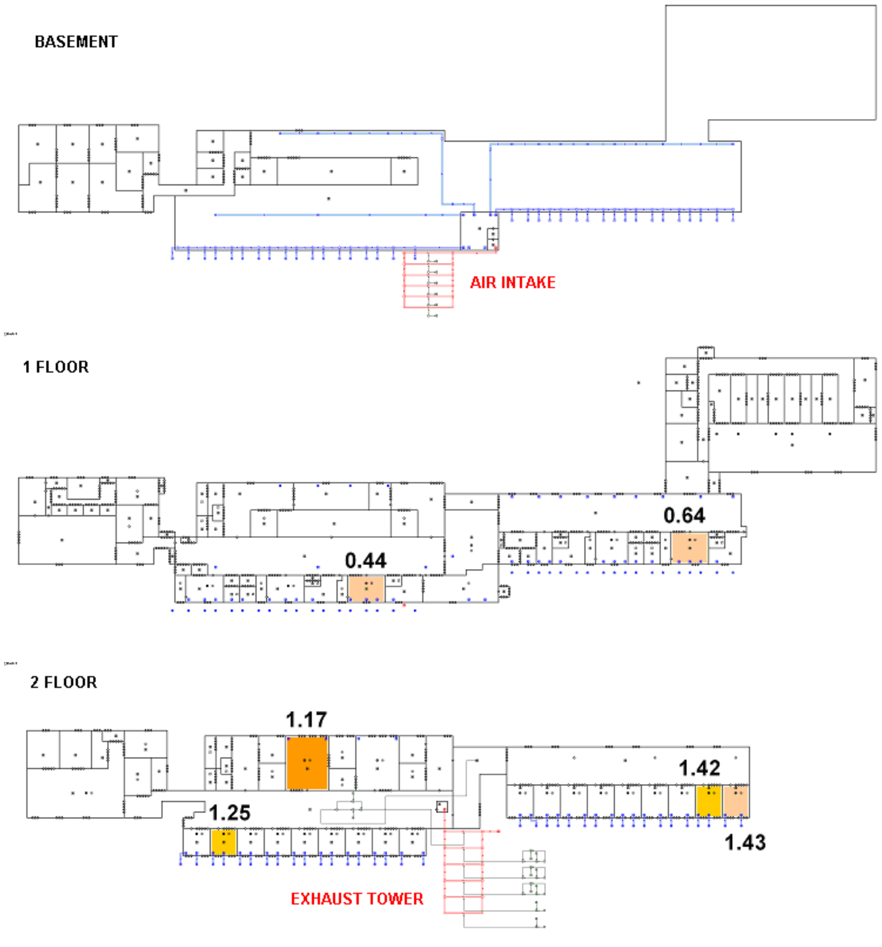

Figure 3. Zonal representation of Zgierz school (centralized hybrid ventilation option and control system connections); sketchpad of CONTAMW program [8].

RESULTS

The results obtained for selected room at 1 floor (room 0.44) are presented at figures 7 while those obtained for room 1.25 (representative of rooms located at 2 floor) are presented at figure 4. Performed estimations indicated that passive stack ventilation should not be recommended any more for modernization of Polish schools. Even though the classrooms was equipped with 12 vents (at 10 Pa) ventilation intensity is too small and resulting levels of CO_2 are high. Such conditions would result in increased concentrations of other pollutants emitted e.g. from finishing materials. Moreover in case of passive stack ventilation indoor air quality is strongly associated with weather conditions. In both presented classrooms concentrations of pollutants are the highest in June (hot period), and the lowest in January (cold period). It is worth to point out that rooms located at 2 floor due to lower stack heights have generally worse level of indoor air quality.

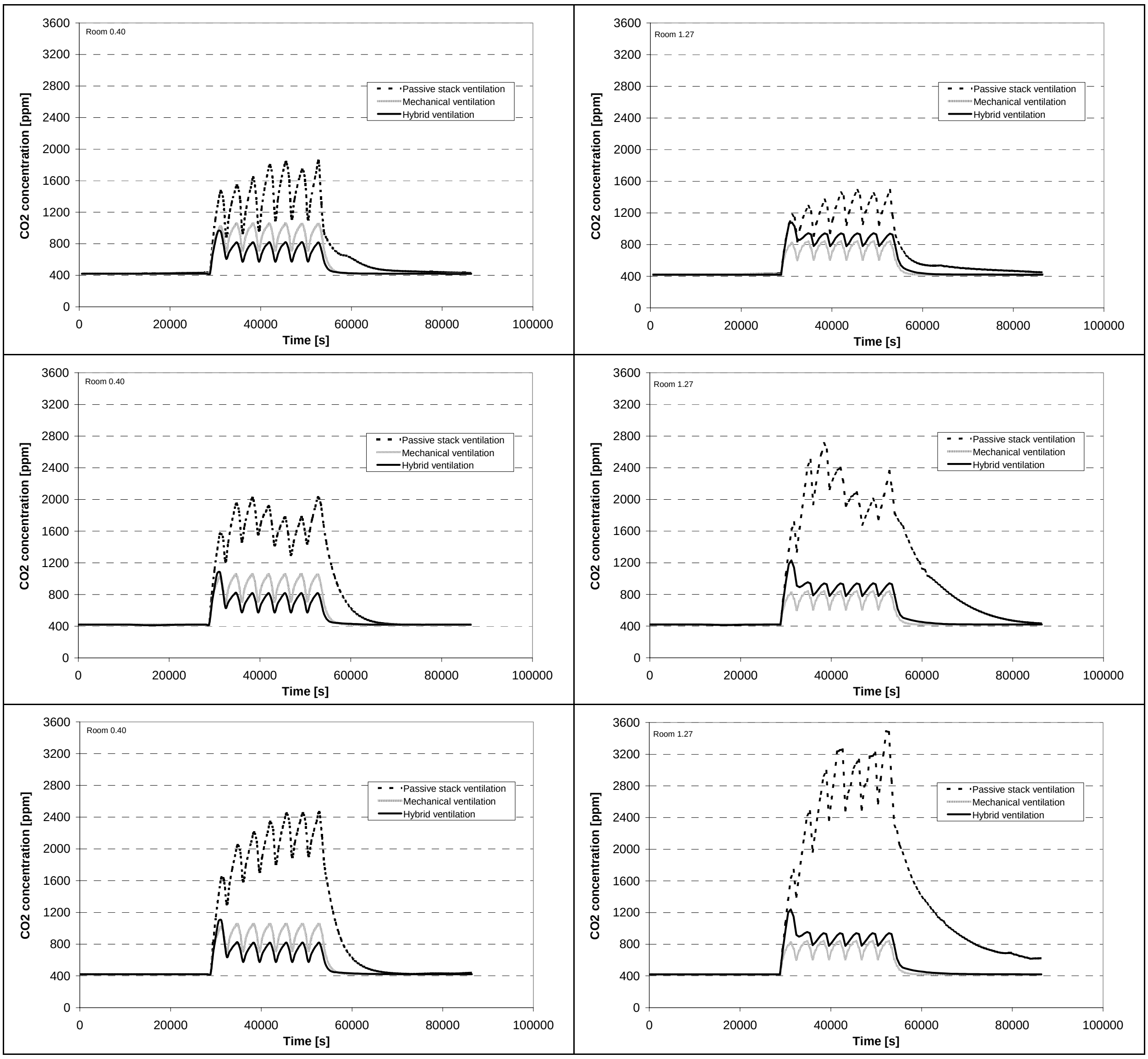

Figure 4. Comparison of CO_2 concentration levels in one of classrooms at 1 floor (room 0.40 - left) and in one of classrooms at 2 floor (room 1.27 right) selected day in January (top); Selected day in April (middle) and selected day in June (bottom) [8].

On the other hand both mechanical and hybrid ventilation are capable to maintain required air quality within classrooms irrespectively of weather conditions.

Mechanical ventilation is based on constant air volume concept (30 m^3/h person) while concept of hybrid ventilation assumes application of control system based on CO_2 concentration. Therefore in some cases the ventilation rates can be lower than minimum 20 m^3/h required for every pupil. However these situations are observed only in the morning after start up of the system when air in classrooms is still clean. The tuning of control algorithm in real building could easily reduce number of hours with ventilation rates below minimum rewired values. On the other hand new proposals of EU standards in case of CO_2 based demand controlled ventilation propose the requirements just fore CO_2 concentration not for ventilation rate.

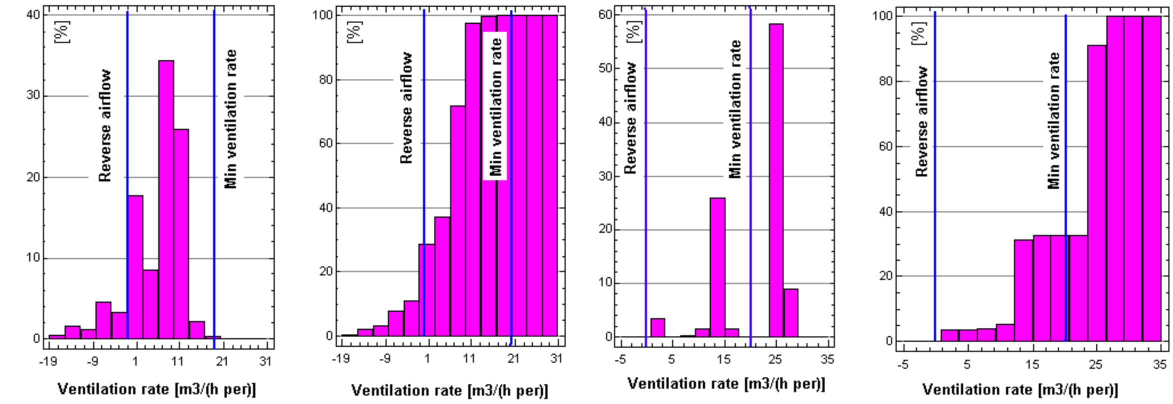

Figure 5. Comparison of frequency and cumulative frequency of ventilation rates in one of classrooms at 2 floor (room 1.43) natural ventilation with air vents (two graphs on left) and demand controlled (CO_2) (two graphs on right) [8].

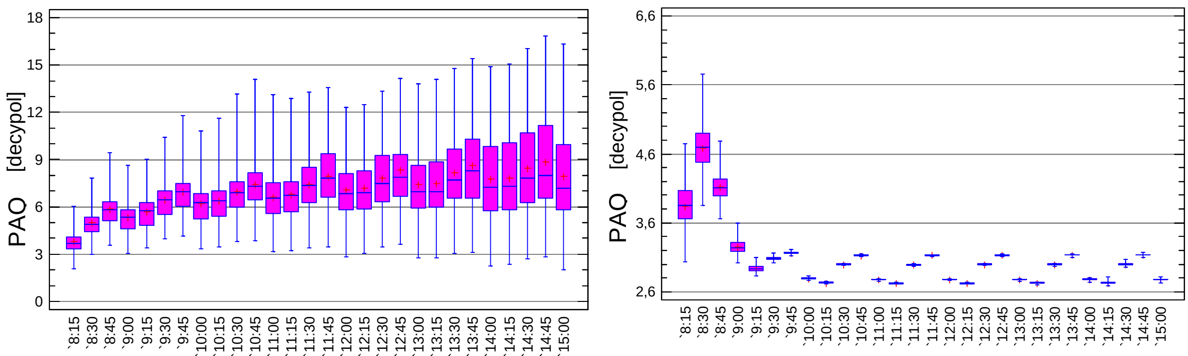

Figure 6. Comparison of average profiles of Perceived Air Quality PAQ in room 1.43 for winter conditions: natural ventilation with air vents (left) and demand controlled (CO_2) (right) [8].

The comparison of distribution of ventilation rates in case of natural ventilation with vents and for hybrid ventilation is presented at figure 5 (for mechanical ventilation air flow rates are constant). These figures represents data obtained for room 1.43 at 2 floor. The graphs once again indicates that natural ventilation does not provide enough fresh air to the classrooms. In analyzed classroom minimum ventilation rates per person are never reached, and moreover during almost 30 % of time air flows has reversed direction (air supply through transfer grills and exhaust through vents in windows. This phenomenon is nor observed in case of hybrid ventilation. It should be pointed out that very low ventilation rates are observed just after pupils enter the classroom. When sensors measure increased concentration of CO_2 ventilation rates are increased.

Figure 6 presents the example of simulations of perceived air quality (assumed emission from finishing materials 0.1 olf/m^2) . It can be observed that PAQ can be lowered from ~8 decipol to 3 decipol that could lead to reduction of percentage of dissatisfied visitors from 57 to 33%.

DISCUSSION

The IAQ analysis indicated that both solutions mechanical ventilation and hybrid ventilation can be regarded as very close to optimal option of ventilation for Zgierz school. Differences between mechanical and hybrid ventilation are the result of applied CO_2 based demand controlled strategy for ventilation intensity and assumed levels switching between speed of fans. The great improvement in IAQ can be obtained together with significant reduction of primary energy use of the school. The results are based on analysis of just one school. However, this object is representative for hundreds of schools erected in sixties in XX century. Similar analysis of these schools would thus probably lead to resembling conclusions.

ACKOWLEGEMENT

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The importance of accurate wind pressures for natural ventilation design

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SUMMARY

Many naturally ventilated building designs rely on buoyancy (or "stack effect") and night cooling of the structure. It is well known that for an exposed building, even mild winds can produce pressures well above those due to stack effect. It is also common for efficient night purging to rely on wind-driven flows. While tools such as CFD can reliably predict airflow rates through the building, without accurate pressure boundary conditions at the openings to the outdoors, these simulations have little value. A common mistake is to underestimate the complexity of the turbulent flow around buildings in the atmospheric boundary layer. Most of the important flow features that control the pressures on the building surface are intermittent and vary significantly with wind direction. In addition, in most locales, pressures for all wind directions need to be considered, since there is no true predominant wind direction. A specific case-study of a natural ventilation design is used to demonstrate these issues.

COMPARING STACK EFFECT TO WIND PRESSURES

It is well known that the forces driving air through a building during natural ventilation are buoyancy (often referred to as "stack effect") and wind pressure. See for example [1], where the possibility of these forces opposing one another is discussed. Stack effect depends upon a temperature difference between the indoor and outdoors, according to the equation

$$\Delta P = \rho_o \cdot g \cdot \Delta h \left(\frac{\Delta T}{T_o + \Delta T} \right) \qquad (1)$$

where ΔP is the pressure difference, ΔT is the Temperature difference between indoors and outdoors, T_o and ρ_o are the air temperature and density outdoors, g is the acceleration of gravity (9.8 m/s^2), and Δh is the height difference between the inlet and outlet.

Wind pressure across a building is often reported as a fraction of the stagnation pressure of the wind,

$$\Delta P = \tfrac{1}{2} \rho_o U^2 \cdot \Delta Cp \qquad (2)$$

where U is the mean wind velocity at some reference height, and ΔCp is the difference in the pressure coefficient between the inlet and outlet locations. The reference height is an important consideration when calculating the wind pressure. Wind data from airports is often recorded at a height of 10 m, yet pressure coefficients are commonly referenced to velocities at or well above building height. All pressure coefficients discussed in this study are referenced to a height of 150 m, where wind speeds will depend upon whether the site in open country (~1.5 times faster than at 10 m) or an urban area (~2.5 times faster).

For a 150m reference height, mean pressure coefficients of between -1.0 and 1.0 are common. If we assume that the windward side of a building has a mean pressure coefficient of +0.6, and that the leeward side is -0.4, the ΔCp = 1.0. By combining Eqs. (1) and (2), we can predict the mean wind speed at which the wind force will exceed the buoyancy force in this case. Results are shown in Figure 1:

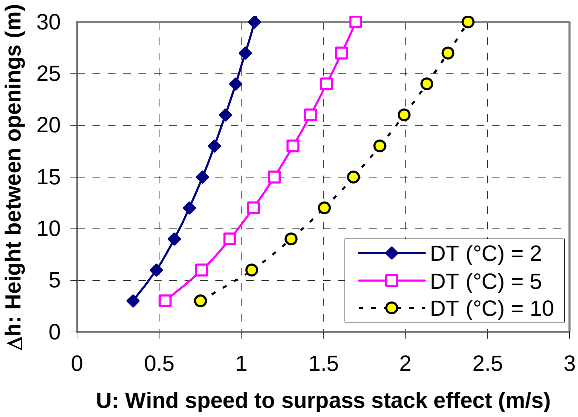


Figure 1. 10 m wind speed above which wind effects surpass buoyancy for ΔCp = 1.0.

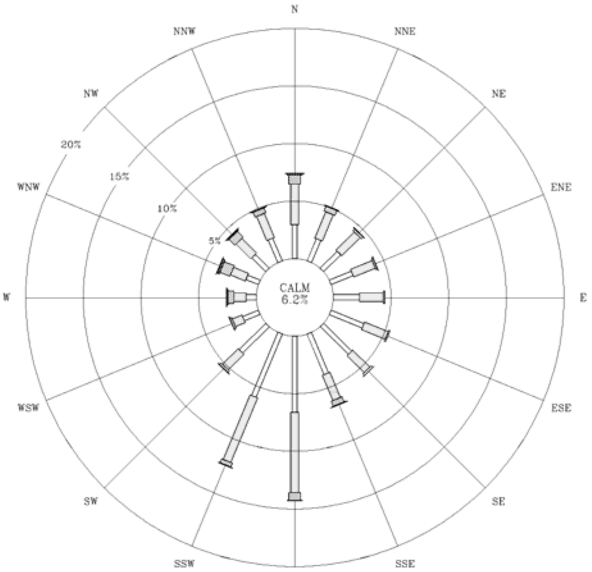
It is clear for most buildings that wind pressures will begin to dominate once wind speeds rise above 1.5- 2.0 m/s, which is a majority of the time in most places. For example, wind data for two cities is presented later in this paper. At these locations, winds exceed 2 m/s 84% of the time in Denver, Colorado, USA and 62% (95% in the afternoons) of the time in San Juan, Puerto Rico.

THE IMPORTANCE OF CONSIDERING ALL WIND DIRECTIONS

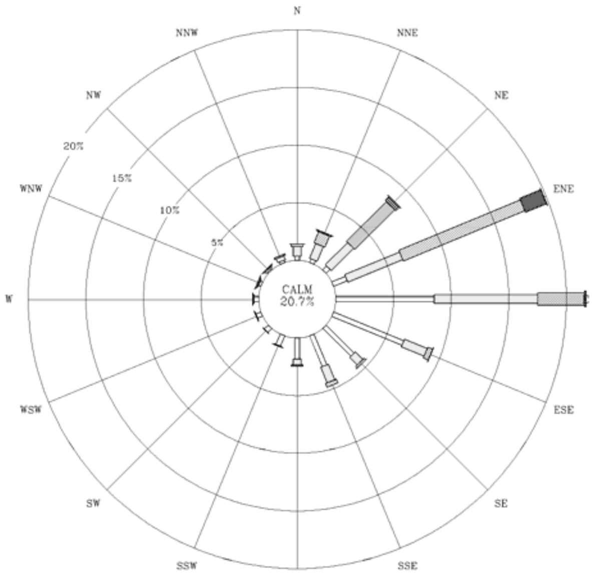
Many design guides for natural ventilation use the predominant wind direction as a primary consideration in the design [2]. However, many locations do not have a true dominant wind direction. Figure 1 shows wind roses from two cities: Denver, near the center of the USA, and San Juan, Puerto Rico, located in the Caribbean Sea. While south winds are the most common in Denver, the combination of SW-SSE winds accounts for only 38% of the winds. Winds come from other directions nearly twice as often, so a design relying too heavily on these directions may not provide adequate ventilation most of the time. For a design to justifiably focus on only a few wind directions, a truly dominant wind direction is needed, in which case the wind rose should resemble that of San Juan. The tropical trade winds, coming from the NE through ESE, are blowing nearly 60% of the time (and account for 75% of the winds, since it is calm 20% of the time).

It is also important to examine when the winds occur, both seasonally and diurnally. Figure 3 illustrates contours of wind frequency versus direction and time of day for Denver and San Juan in the autumn. Denver's south winds are clearly seen to be a nocturnal phenomenon – useful for night purging, but not for cooling during the heat of the day. Conversely, San Juan's ENE trade winds are absent overnight, but dominate completely in the late afternoon.

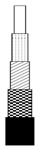
A subtle difference in the Denver data between Figure 2 and Figure 3 can be detected. The dominant overnight winds in the contour plot are from SW (225°), rather than south, as in the wind rose. This is because the anemometer was moved roughly 12 miles when a new airport was opened in 1995. The local valleys at the airports are not steep, but they nonetheless



a) Denver, Colorado, 1964-1995.

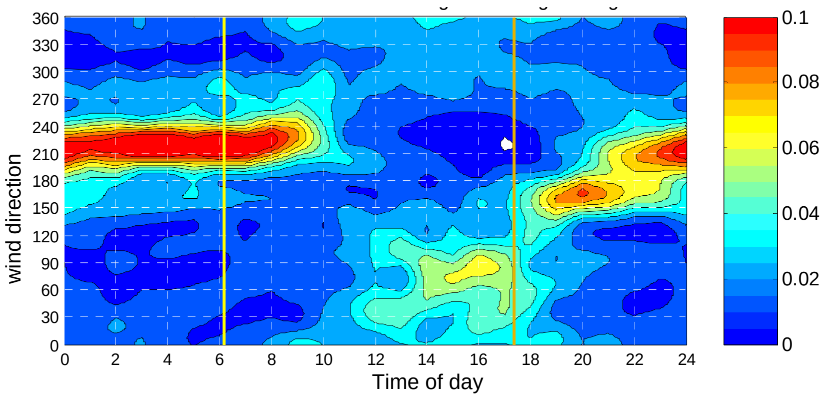


b) San Juan, Puerto Rico, 1996-2003

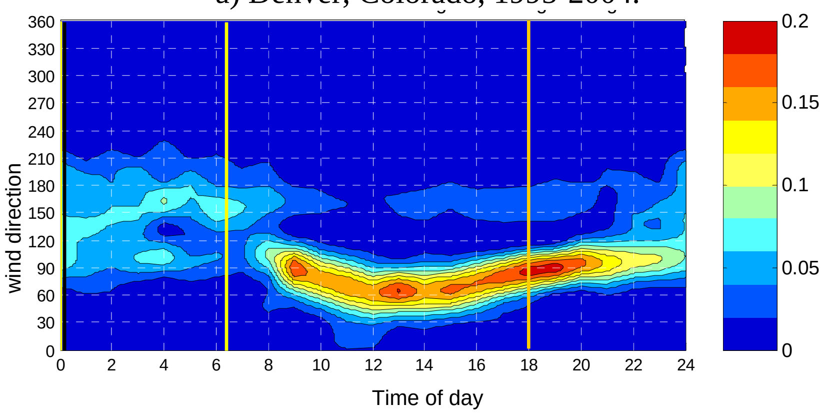


< 4 m/s	< 3.5 m/s
4 – 7 m/s	3.5 – 5.5 m/s
7 – 11 m/s	5.5 – 8.5 m/s
11 – 14 m/s	8.5 – 11 m/s
> 14 m/s	> 11 m/s

Figure 2. Wind roses for two airports



a) Denver, Colorado, 1995-2004.



b) San Juan, Puerto Rico, 1996-2003.

Figure 3. Fraction of the time that winds come from a given 10 degree range, as a function of time of day. Vertical yellow lines indicate sunrise and sunset. Data is for October.

channel the flow differently. It is always worth considering the possibility that winds at the site of a new building will not match those recorded at the airport, particularly where proximity to the ocean or significant topography are involved.

THE IMPORTANCE OF FLUCTUATING PRESSURES

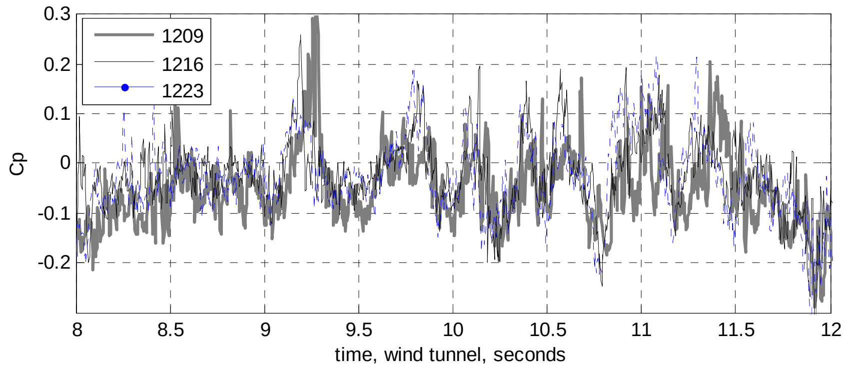
Pressure fluctuations on a building’s envelope are considerable. Peak pressure coefficients most locations on a building vary from -3 to +2, with Cp values below -5 in corner regions if vortices form. These wind pressures will very quickly propagate through the entire internal space. A formula for predicting this response time can be found in [3], but for relatively large indoor spaces such as atria, the time constant is less than one second.

A sample case is examined to demonstrate the importance of pressure fluctuations. Figure 4 is a photo of the pressure model of a 250 m wide, 150 m long 4-level parking garage. Note the absence of any significant nearby structures.

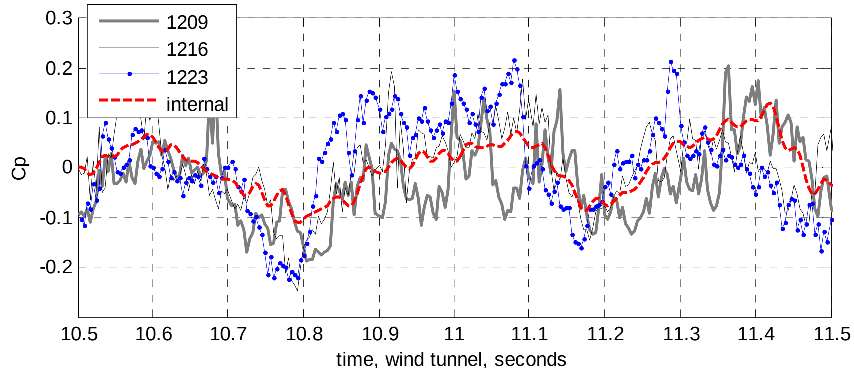
Figure 4. Photo of parking garage pressure model in the boundary layer wind tunnel.

Figure 5 shows a time series of wind pressure coefficients on the northwest face. The pressures shown are all on level 3. The wind is from the southwest in this case. The mean pressure on this face is very close to 0. If the average pressures were used to predict the wind driven flow, there would be none. However, it is clear that air both enters and exits the openings of this face as the pressures rise and fall.

In Figure 5a, the pressures across the entire face appear to follow fairly a common pattern. For example, between 9.0 and 9.5 seconds all three locations experience high positive pressures. Upon closer inspection of the time series in Figure 5b, we see that the pressures do diverge from time to time. This indicates the possibility of air entering at one place on the face, while simultaneously exiting at another.



a)



b)

Figure 5: Pressure Coefficient time series, U_{ref} measured at 150m. a) 4 seconds of data b) 1 second.

The duration of these gusts is an issue – if they are too brief, they will not ventilate as deeply into the garage. The time scale in the wind tunnel can be converted to full scale using Equation (3):

$$t_f = t_{m.} \frac{U_m}{U_f} R \qquad (3)$$

where t is time, the subscripts *m* and *f* indicate model scale and full scale respectively, and R is the model scale length ratio, which is 400 in this case. If wind speeds at full scale match those during the testing (roughly 4-5 m/s at a height of 10 m), then each second in the tunnel is 400 seconds, or nearly 7 minutes at full scale.

Figure 5b also includes a time history of the estimated internal pressure in the garage. (The internal pressure is not measured in the wind tunnel because the time-scaling similarity of Equation (3) does not apply to internal pressures unless the internal volume is modified. [3].) Instead, it is estimated using a flow balancing approach [4]. It is the difference between this internal pressure and the external pressures measured at the openings around the building that will control the ventilation rate. For example, just before the 11 second mark, air would enter at location 1223 (at the west end) and exit at 1209 (at the east end), and shortly after 11 seconds, this pattern reverses itself. In both cases, there is very little flow at location 1216 near the center of the face.

These are the fluctuating pressures that drive single-sided ventilation.

The origin of the pressure fluctuations is twofold – in the approach flow turbulence and local flow structure intability. The approach flow contains turbulence generated by flow over upstream surface roughness. The turbulence includes larger eddies that completely envelope the building, and mid-scale turbulence that produces local high pressures on only one part of the structure. The effects of these scales can be roughly approximated using a quasi-steady approach [5]- for example, the effects of low-frequency lateral turbulence can be approximated by examining the average pressures for a range of wind directions.

The pressures beneath the flow separation at a building's corner is not well predicted by quasi-steady theory, however [6]. This is in part because the separated flow zones are inherently unsteady, with flow structures such as vortices forming, moving, and disintegrating. Pressures in these areas are linked both to wind speed fluctuations and to these flow structures [7]. The behaviour of the flow separation is in turn linked to very small scale turbulence in the incident flow, which is thought to interact with the shear layer and cause earlier reattachment and the ensuing higher suction pressures [8].

Note that while it is not as prevalent in this study as in many urban locations, nearby buildings and terrain present a third source of pressure fluctuations on a building, as they shed vortices, channel flow, and shelter parts of neighbouring buildings with their wakes.

CFD SIMULATIONS

The reality of these pressure fluctuations has considerable implications for CFD modelling of airflow through a building. It is not uncommon for a CFD practitioner to recognize the deficiency of using mean pressure boundary conditions for an isolated square building from a handbook when simulating a building in a more complex environment. Given the flexibility of the tool, it is not uncommon for the CFD practitioner at this stage to simply extend the bounds of his or her simulation a few metres outside of the building.

Unfortunately, there is no way to adequately model the flow around the building without including a good boundary layer approach profile and all significant nearby buildings and terrain features. Essentially, if something was necessary to accurately simulate the external flow in the wind tunnel, then it will be necessary to do so in the CFD simulation as well. The computational requirements for such a simulation are the topic of considerable research, but it is clear that to create a realistic, time dependent, turbulent boundary layer and resolve all of the unsteady shear layers on and near the building would require far more computational resources than the internal flow model.

The pressure data from the wind tunnel tests were used as boundary conditions for an unsteady 3-D CFD simulation of flow through the garage. The simulation used over 4 million cells, with RANS turbulence modelling (realizable k-ε). Simulations were performed both with and without rows of cars in place. In the absence of cars, good agreement was seen when compared to a 2-D simulation. Sample output images from the simulations are shown in Figure 6.

The wind is coming directly from the left of the images in these simulations (from 230°). The top of the image is project north. Around the edge of the domain, staircases are shown in grey, column lines in green, and a floor-to-ceiling walls on the west side is shown in black. The four white blocks in the centre represent floor-to-ceiling obstructions (elevator shafts and service rooms). The two round portions at the bottom of the images are circular access ramps.

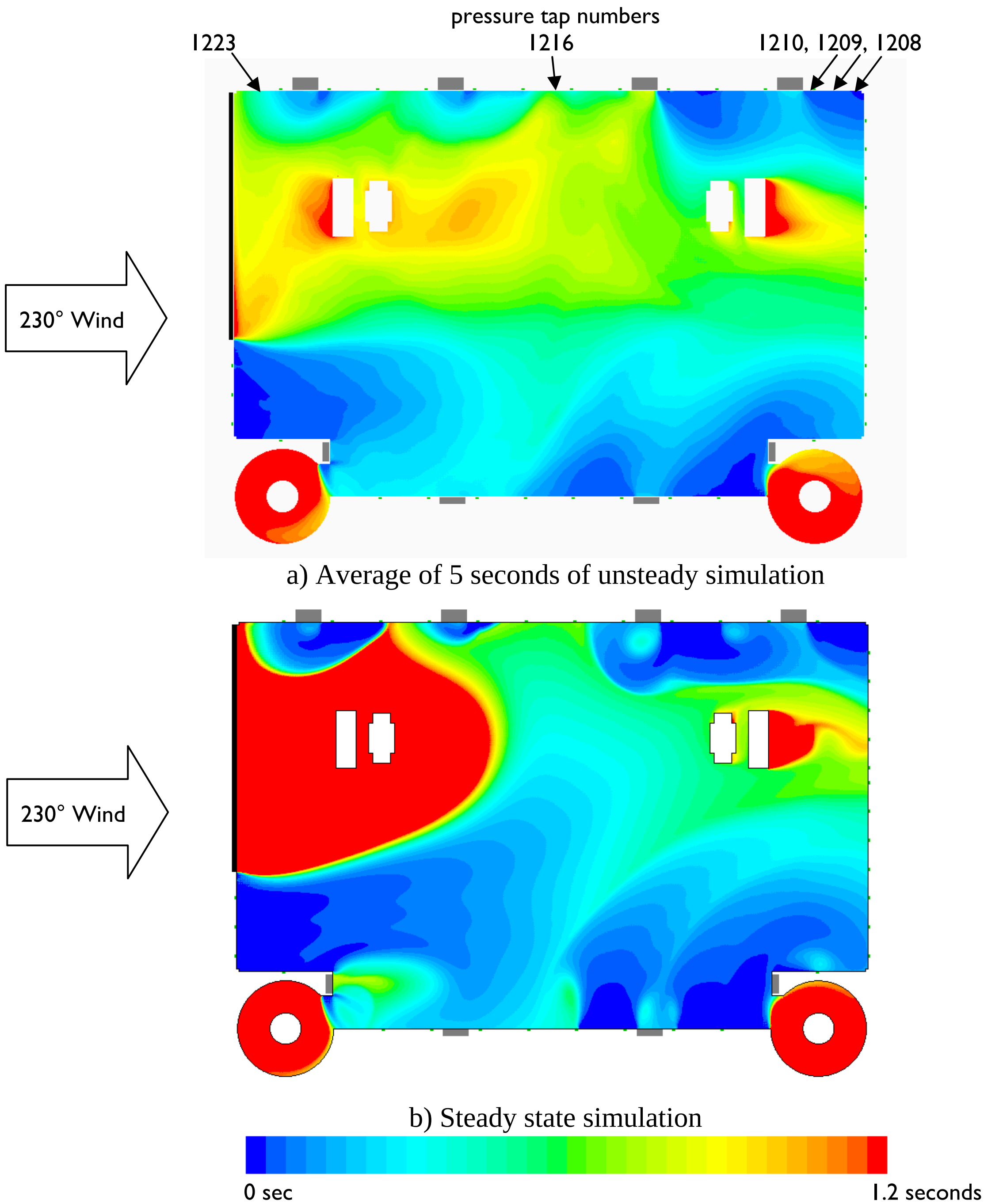


Figure 6. Top view of the garage showing mean age of air (MAA) contours.

In the steady state simulation result of (b), a large, stable recirculation zone forms directly behind the west wall. Some of the air within this zone is over 10 seconds old in the simulation. In the unsteady simulation (a), this area is occasionally flushed clean when tap 1223 sees positive pressures, or exhausted when this location experiences negative pressures. Consequently, the steady state simulation significantly under predicts the natural ventilation.

Note that while this type of flow reversal is beneficial for a parking garage, as it keeps emissions from cars from building up, this is not always the case. For a smoke removal system using natural ventilation, it would be very detrimental to have the exhaust flow reversed by gusts of wind.

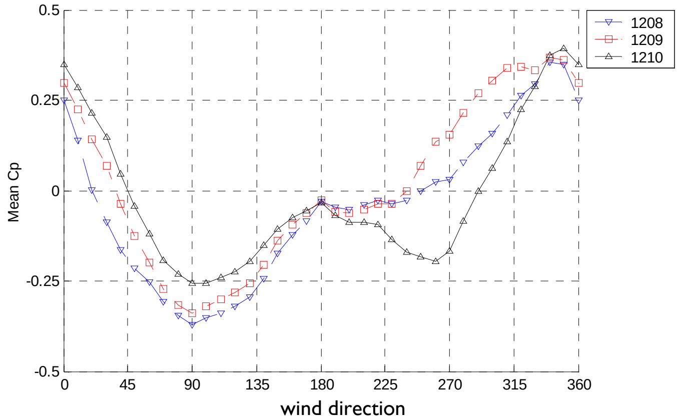


Figure 7. Cp vs. wind direction for taps in NW corner

The flow patterns along the north side of the garage are worth examining more closely. We see that single sided ventilation is enhanced by the two west side stairwells, which create local zones of positive pressure on their upwind sides, and negative pressure downwind, increasing ventilation locally. This pattern is illustrated in Figure 7, showing mean pressure vs. wind direction for three adjacent taps between the northwest stairwell and the corner (see Figure 6). Tap 1210 exhibits negative pressures associated with the separation region behind the stairwell for wind directions between 225° and 290°. It is also important to note how quickly the pressures and the relationship between the pressures changes with wind direction. For this reason, running simulations using only a limited set of wind directions can be misleading. A wind direction shift of only 20° can cause a flow reversal which would not be recognized in CFD simulations run only at every 45 or 90 degrees.

CONCLUSIONS

- Wind pressures will frequently dominate natural ventilation flow rates and paths.
- Winds from all directions should be considered in most designs.
- Pressure fluctuations play a crucial role in ventilation patterns, so steady state simulations are of limited value and should be used with caution.

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Exterior Climate and Building Ventilation

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SUMMARY

The article shows methods for wind influence on building ventilation systems analysis. Solution has been carried out through the experiment in wind tunnel as well as by CFD modelling. Mean pressures and pressure variations have been measured on the model surface with respect of the ventilation intake and exhaust usual situating, with height of the roof parapets modification. The experimental results was presented by pressure coefficients-output is possible convert to different boundary conditions. The wind effect was analysed in different wind direction. The measurement and simulation is focused to the top part of the building façade and to the roof.

INTRODUCTION

Natural ventilation is an air flow inward and outward interior and it is caused by a pressure difference between building interior and exterior. Natural ventilation can create the healthy indoor environment and can save energy but the prediction of ventilation can be difficult. The air is driven in and out due to pressure differences produced by wind or buoyancy forces. Intake air is not usually controlled.

Natural ventilation is typical permanent ventilation, with limited regulation. Energy efficiency depends on the local condition insight and outsight the building. It induces that the same system operates differently from the others in the same conditions. The main influence is caused by the wind.

METHODS

Wind influence on buildings

Wind influence on the building is investigated in the context with static load to the building bearing system. Air flows around the building and the influence on building systems was analysed in the last few years. Wind influence on pedestrians and ventilation systems is significant now.
Natural ventilation systems work with temperature differences or with the wind influence. A typical system works with temperature difference and air intake is realised through the building envelope and air output is situated in the roof. Exterior conditions have influence on an air input (input items) and air flow on the building is fluctuating. There is an impact on the

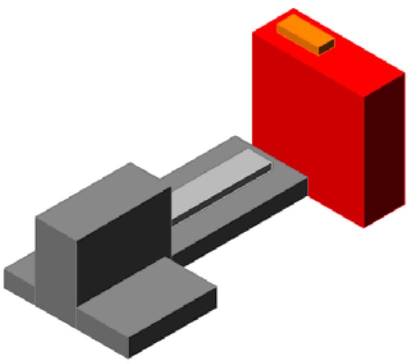
air output on the roof too. An air output is influenced by positive or negative pressure, which is induced by an aerodynamic effect (a building shape, ventilation systems, a parapet, etc.) Ventilation systems induced by the wind are systems where the wind influence is dominant and the density change by temperature difference is small. A typical system is a one side ventilation or a cross ventilation. But ventilation efficiency is fluctuating.

Exterior and building

There are many ways how to analyse the wind influence on the building. These methods were first of all evaluated for a bearing system design. Maximal pressure influence is needed for this purpose. Maximal wind velocity was statistically analysed in different directions. But ventilation systems work with real wind velocity and wind direction. We need typically average values and local extremes (low and high velocities and pressures). High wind pressure exceeds the ventilation function and regulation is needed, low wind pressure stops ventilation.

Measurement in a real building

Design conditions and the cause of systems failure were already measured in previous studies. Wind velocity measurement was realised in the area where the building was built. Data set was use for a local condition analysis. But data set analysis is really difficult and a correct interpretation is problematic [1].
Real measurement is typically realised with plate or membrane pressure indicators. Plate indicators are put to the building façade and wind pressure pressing the plate. Membrane bending is measured in membrane indicators. These indicators are convenient for extreme pressure measurement in particular.



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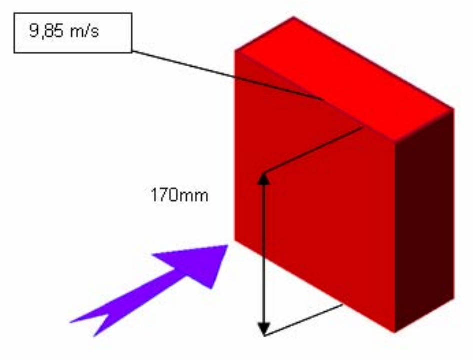

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Figure 1. a) Model scheme of the Faculty and connected buildings, b) Faculty of Civil Engineering model with wind velocity in the roof layer (wind angle 0 - perpendicular to the wind direction)

Computer simulation

Computational fluid dynamics (CFD) provides an alternative approach to calculate ventilation and airflow distributions around buildings. Computer simulation substitute real measurement and save time in this field of study. The most problematic situations can be defined and modelled later in a wind tunnel with better accuracy. CFD models comparison with experimental measurement was compared in previous studies [2]. Experimental 2D model with one side ventilation was compared as the example.
The most common CFD method is Reynolds averaged Navier–Stokes (RANS) method. The method is relatively fast but there can be a problem with a turbulent region around the building especially around the roof. The better method is Large-eddy simulation (LES) which

separates flows into large eddies and small eddies and computes it separately with different methods.

Measurement on building model

Measurement on building model in the wind tunnel is more precise than real measurements. Air flow around the building, pressure on the building façade, flow trajectories can be measured with different wind velocities and azimuth.

The Aeronautical Research and Test Institute (VZLU) boundary layer wind tunnel was used for this measurement. Urban, suburban or plain terrain can be simulated and vertical wind profile can be modified.

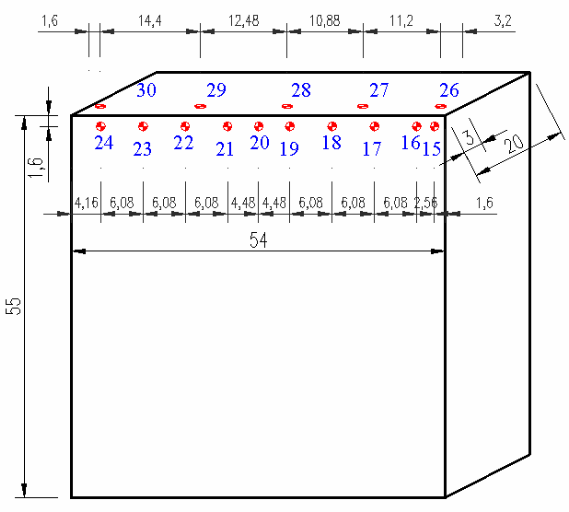


Figure 2. a) Faculty of Civil Engineering, b) The building model with measuring points (dimensions on real building)

A wind effect is determined by a wind velocity and wind direction, time dependence is according to a boundary layer scale. Meteorological effects in large scales are disturbing. That is reason why experimental measurement in the full scale is problematic (dates, intervals). Experimental measurement in the laboratory is time constant and repeatable. Models with complicated geometry and surrounding conditions can be measured too.

RESULTS

Study of the ventilation system performance needs detailed airflow information around and inside the building. The velocity and pressure distributions can be used to determine the ventilation rate.

The measurement was done in boundary layer wind tunnel in Prague. The model was in the scale 1:320 and in different configuration with changeable parapet high on the flat roof. This scale was convenient to model changes (surrounding building), the boundary layer wind tunnel dimension and profile. The real building is a rectangular 15-floor building. Building substructure is connected to surrounding buildings. The wind direction was changed with step 30°.

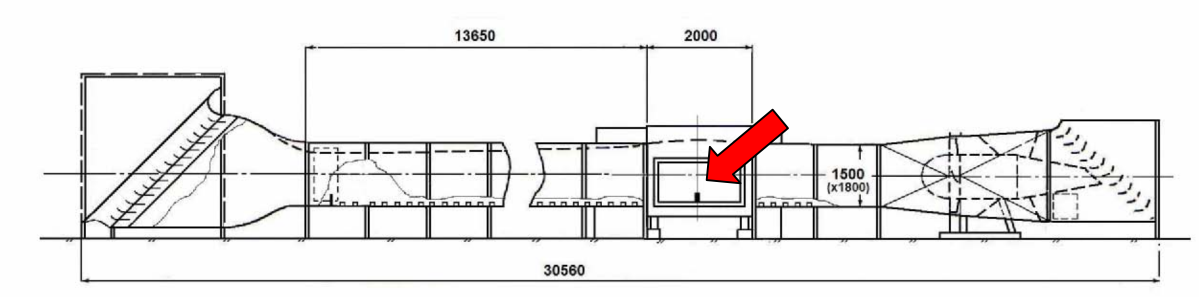


Figure 3. a) The Boundary layer wind tunnel (The Aeronautical Research and Test Institute in Prague) with marked model position

The principle of measurement uses an analogy between a model and a building with given boundary conditions.

Table 1. Measuring points and data set example (angle 0)

Measuring points	15	16	17	18	19	20
Average pressure [Pa]	30,9	42,9	43,4	41,78	41,68	42,27
Standard deviation [Pa]	23,5	22,5	22,3	21,72	19,74	21,75
Pressure coefficient c_{pi}	0,34	0,48	0,48	0,46	0,46	0,47

The measurement was done in the wind tunnel with three different wind velocities in 64 points on the model surface. Wind pressure in the building envelope was measured. The wind vertical profile corresponds to a sub-urban terrain (roughness length is 0,65mm, a suburban roughness). Dynamic pressure is 56,9 Pa in the model roof layer. The pressure was measured by Prandtl probe.
These pressure differences corresponds to the method of unit-less pressure coefficients c_{pi}.

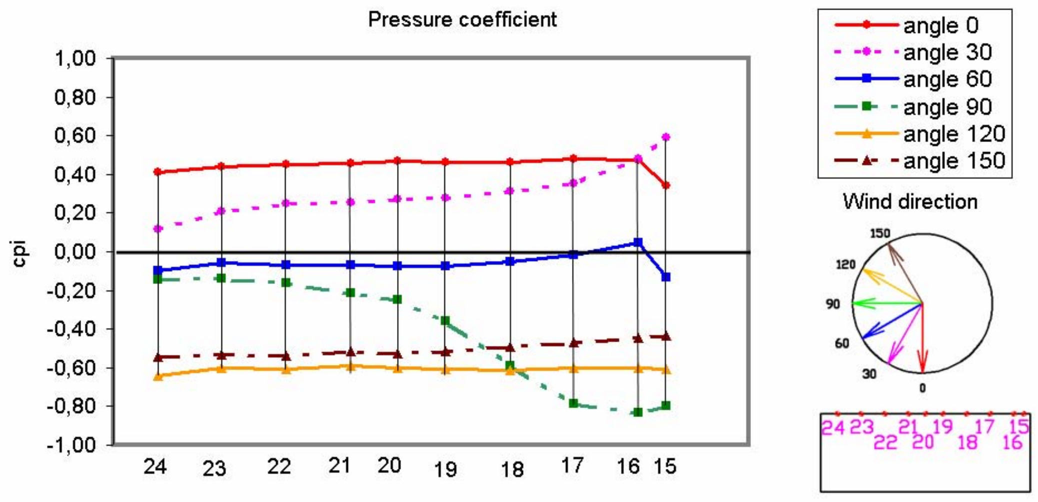


Figure 4. Pressure coefficients (without parapet) 1,6m below the roof layer

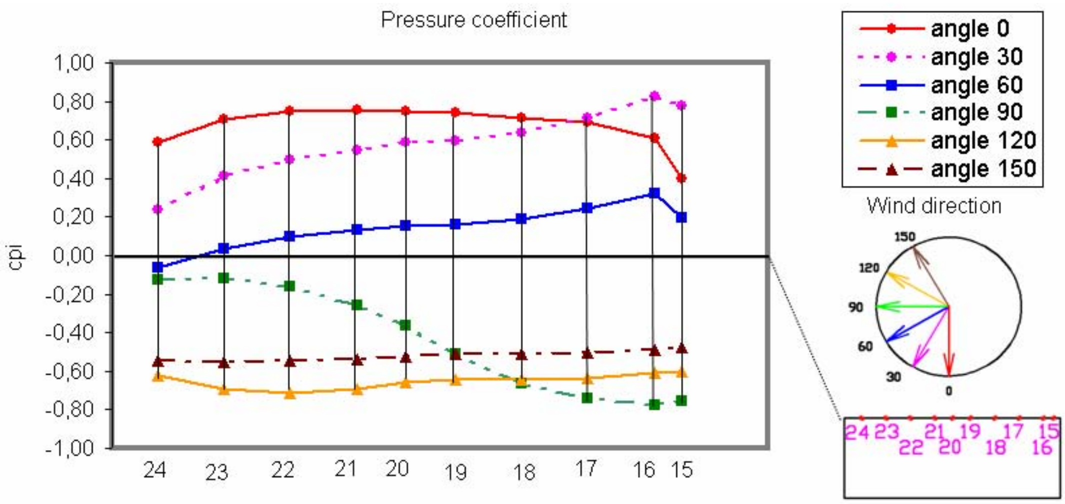


Figure 5. Pressure coefficients (parapet 3,5m) 1,6m below the roof layer

Results from the top part of windward façade (points 15-24) and the roof (points 26-30) are presented.

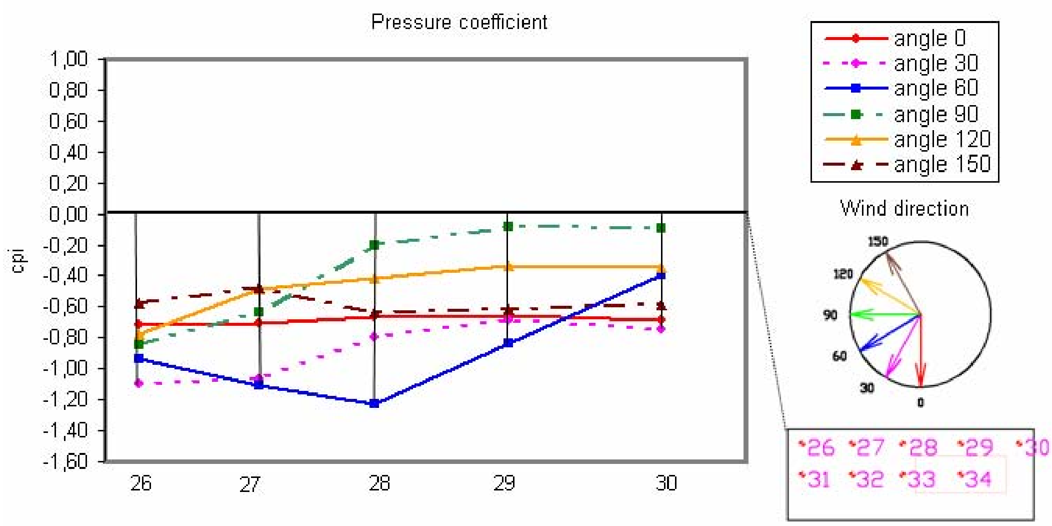


Figure 6. Pressure coefficients (without parapet) on the roof

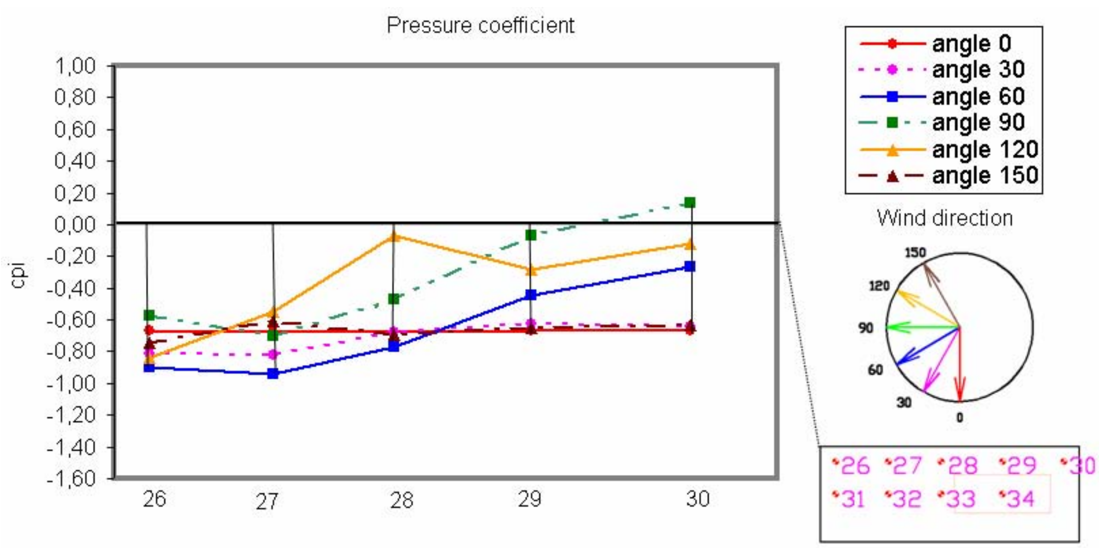


Figure 7. Pressure coefficients (parapet 3,5m) on the roof

The data set was recorded with frequency 10000 s^{-1} and was compared statistically. Average pressures, standard deviations and pressure coefficients cp were recalculated.
The coefficient c_{pi} may be recalculated to different conditions than those used during measurement.

$$c_{pi} = \frac{p_i - p_{ref}}{\frac{1}{2} \cdot r \cdot U_{ref}^2} \qquad (1)$$

Where p_i is pressure [Pa], p_{ref} is reference pressure [Pa] (roof layer), ρ is density [$kg \cdot m^{-3}$] and U_{ref} is reference velocity [$m.s^{-1}$]-air velocity in the roof layer.

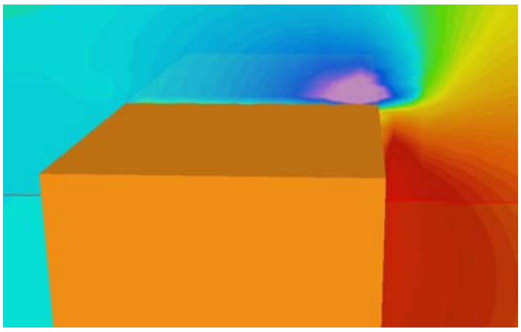
CFD simulation

ENVI-MET models

Software ENVI-MET is specialised software for external condition calculations. Surface parameters can be specified, different terrain properties may be applied. Building shapes are limited by rectangles. The exacting character of calculation demands time; the other disadvantage is interface. Boundary conditions are relatively simple.

FLOVENT models

Commercial CFD software is convenient for complicated situations. The wind profile can be specified (or set up) by vertical separation to layers with constant parameters. But the wind profile was not disturbed by a built up area.



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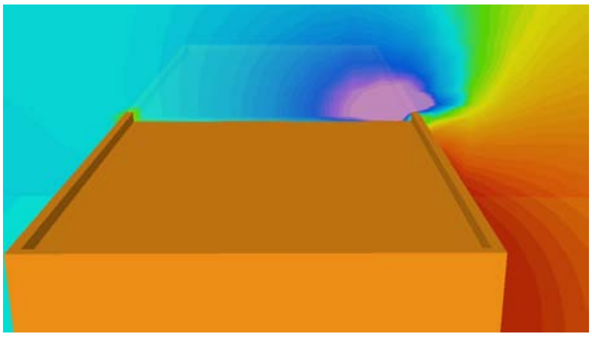

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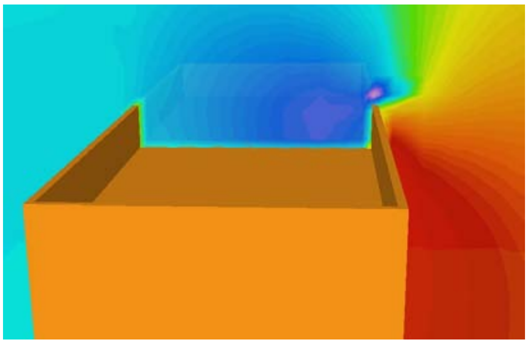

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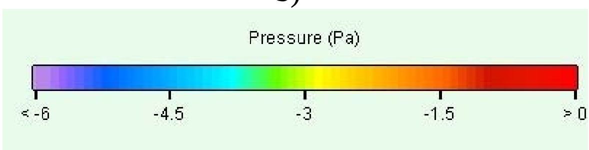
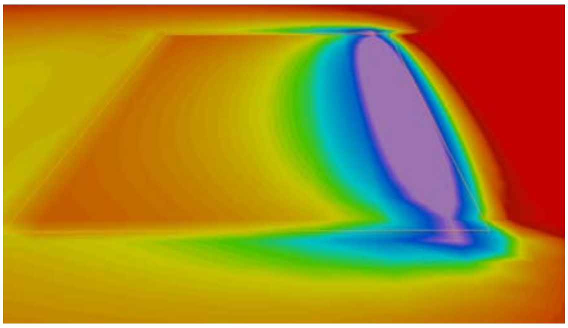


Figure 8. Pressure field in vertical section (wind angle 0) a) Building without parapet b) Building with parapet 0,6 m c) Building with parapet 3,5 m Flovent 6.1



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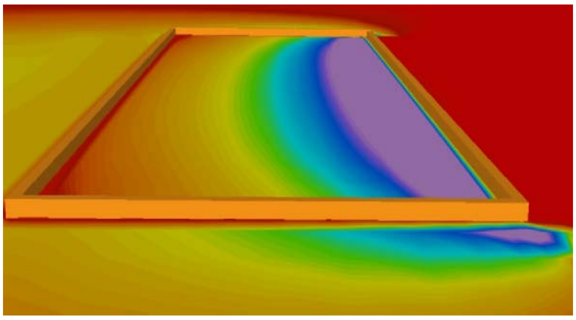

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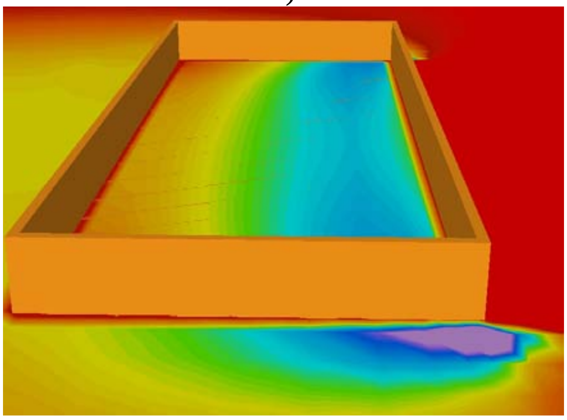

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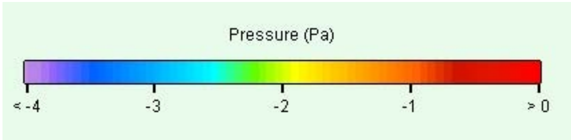
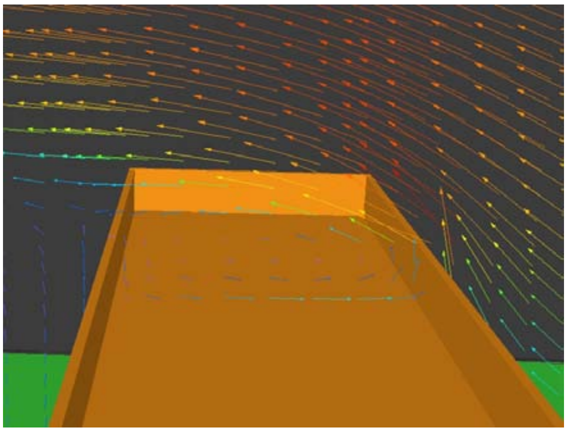


Figure 9. Pressure field in horizontal section – roof surface (wind angle 0) a) Building without parapet b) Building with parapet 0,6 m
c) Building with parapet 3,5 m
Flovent 6.1

Characteristic airflow around the building roof was assessed (Fig. 9, 10). The reverse air flow arises in case of building with high parapet above the roof area (Fig.11).



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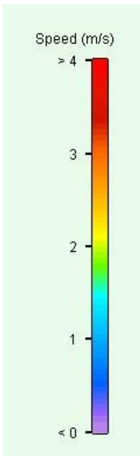

Figure 10. Velocity vectors around the building parapet (3,5m) (wind angle 0) a) Flovent 6.1

DISCUSSION

The air flow around a building surface and the influence on a ventilation system is relatively complicated. Even though simple flow can be determined by CFD simulation, the experimental measurement in the boundary layer wind tunnel must be done in case of detail investigation or complicated situation. The investigation results show that the higher parapet on the model induces higher pressure in the top part of building windward façade (the highest floor level).

The pressure influence on the roof surface with height of the roof parapets modification shows that smaller parapet increases negative pressure on the windward roof area. Higher parapet clams the reverse air flow close the roof surface. The reverse air flow can cause ventilation system failure especially if the input and exhaust is situated on the building roof. The air mixing risk is in this space. Building parapet with lower height (0,5-1,5m) is convenient for natural or hybrid ventilation exhausts – the wind support is the highest in windward part of the roof.

ACKNOWLEDGEMENT

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Ventilation Potential: Examining the Effects of Growing Densification in the Tropics

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SUMMARY
The density of Dhaka is increasing exponentially, and with it the demand for residential accommodation. Rapid densification is increasing hard surfaces and changing the urban texture, thereby escalating heat absorption. This intensifies dependency on valuable conventional energy resources, for thermal comfort.
Research shows that regional climatic data differs markedly from localised microclimatic effects within the built environment. In built-up residential areas of the city there is the lack of natural wind flow, one of the most important ingredients for comfort during the warm monsoons. Electricity is growing costlier day by day and its supply is erratic, while load shedding to balance the supply and demand, is very common.
This paper presents the findings of an ongoing research, to investigate the effects of growing densification in the city on the potential for natural ventilation. The work is based on the premise that the effect of variation in density will be reflected in the comparative differences in qualities of the thermal environment (indicated by the temperature and wind speed measurements) in the different residential areas of Dhaka. Higher temperatures in the denser parts of the city may be attributed to larger daytime absorptions and slower night-time cooling, and lack of breeze to distribute the unevenness. In conclusion some passive measures are suggested to rectify the negative aspects of the environment.

Keywords: thermal environment, natural ventilation, urban density

INTRODUCTION
Dhaka is a tropical city lying on the edge of the Tropic of Cancer, with a composite monsoon climate having a rather long warm-humid season [1]. In the past four decades since it became the capital of independent Bangladesh, the city has grown exponentially and is projected to become the second largest city in the World with respect to population growth by the year 2015 [2]. The influx of population increases demand for residential accommodation, which raises the density of the built environment.

Research shows that the main ingredients for comfort [3] during the warm monsoons is the presence of air movement. The built environment modifies values of solar radiation, wind flow and with it, the temperature, humidity, and related climate data [4]. Rapid densification, increases obstacles to wind flow, and also increases hard mass or hard surfaces, affecting the thermal environment by changing the urban texture and escalating heat absorption. There is therefore a wide difference in regional wind availability and microclimate found on urban sites and in the respective temperature data. Previous research also shows that air velocities within building interiors falls dramatically from outdoor regional averages, further lowering the ventilation potential of available wind in typical residential areas of Dhaka [5].

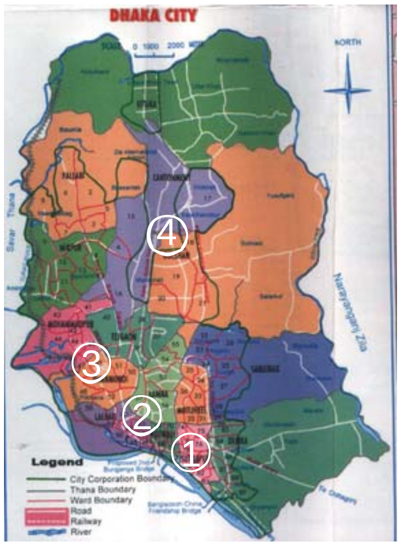

Figure 1: Map of Dhaka with survey Spots indicated by numbers
Source: www.bangladeshdotnet

In the absence of naturally induced air movement within building interiors, residents of Dhaka resort to the use of electricity, to accelerate evaporative cooling, allowing a semblance of thermal comfort, despite the high humidity. But active energy is growing costlier day by day, with the latest price hike scheduled for March 2007 [6], while its supply is very erratic with load shedding a common phenomenon [7]. These black-outs are employed by the authorities, to keep equity between the supply and demand of electricity.

Addressing these issues, this paper presents the findings of an ongoing research [8] conducted by the authors, where the potential for natural ventilation is examined from an investigation into the effects of growing densification in the city.

The basic hypothesis of the work is that the effect of density variation will be reflected in the comparative differences in the temperature and wind speed measurements of the different fabric patterns in the formal and informal residential areas of the city. Higher temperatures found in the denser parts of the city may be directly attributed to larger absorptions during the day and slower night-time cooling due to changed wind speeds and convection patterns.

One of the main objectives of this work was to increase awareness of the wind regime in different residential areas of the city and to suggest passive measures to rectify the negative aspects of the environment.

THE INVESTIGATION

Setting the Context

The study was conducted on the residential areas listed below, representative of differing levels of development and density in Dhaka metropolitan area, including formal and informal sectors (indicated in Map by numbers in Fig.1).

- Informal sector residential development where growth pattern is incremental and dense, and no definitive planning control is followed, e.g. Sutrapur (1) in old part of the city.
- Institutional housing where the development is far more controlled, eg Teachers' Quarters of BUET (2).
- Formal high-density residential area, e.g. Dhanmondi Residential Area (3), where many of the original one-bigha plots (generally in grid-iron layout) have been sub-divided into smaller plots for descendents, and the remaining original one unit houses per plot are now being replaced by six-storied apartment buildings.
- Formal medium-density residential area, e.g. Banani Residential Area (4), with large plot sizes, most still occupied by single families (sometimes joint or extended family) in one unit houses.

Physical survey of these residential districts revealed density of building mass and ratio of open and built areas along with surface characteristics and vegetal cover. These data were then correlated with measured thermal variables of temperature, humidity and airflow using portable instrumentation.

Data collection at survey spots and analysis

Suitable buildings were selected in each of these spots for data collection. For actual data collection, transition points were identified within the buildings, defined as transitional areas either between indoor and outdoor, or the immediate outdoors of dwelling units. These points form the boundary conditions for the building fabric. Recognising that transitional areas form buffers between the external and internal environments, understanding of the thermal environment in these spaces can make passive climatic control easier. Moreover, dependency on active control is largely reduced if the transitional point environment does not get extreme.

Table 1 Comparing the Density and physical qualities of the four residential districts

Location	Sutrapur	Dhanmondi	BUET T. Q.	Banani
Existing avg. Floor Area Ratio (FAR)	4 to 5	3.5 to 4.5	1.1 to 1.3	1.5 to 2.5
Existing avg. land coverage by building structures on plots	85 to 95%	70 to 80%	25 to 30%	55 to 65%
Existing avg. ratio of hard : soft surface (exc. road area approx.)	95 : 05	80 : 20	30 : 70	60 : 40
Population density in persons per acre approx.	357	310	162	180

For Air-flow data, measurements were taken at three different heights at every survey spot using three weather meters[*] simultaneously. Temperature and Relative Humidity data were measured with the help of two additional data loggers[†], while two maxima-minima thermometers were used for simultaneous maximum and minimum temperature data. Prior to the field data collection, necessary field calibration was made by the recommended calibration kit according to the specification supplied with these instruments. The regional averages were also noted from weather stations and published sources for the survey days to note micro-climatic deviation at surveyed spots. For solar radiation data it was assumed that the values would be similar for all sites as they are located within a few kilometres of each other and there is no noticeable difference in air quality/atmospheric condition.

Field data was collected within a limited time, during May, when conditions are representative of high thermal discomfort. April is the hottest month in terms of mean maximum temperature [9] [10], and the effects of its interaction with built form is expected to set in after a few days, i.e. by the middle of May.

A comparative analysis of climatic data collected from the different residential areas was then made to identify variations in thermal environments and to ascertain the extent and causes of variations. This analysis is expected to help formulate correlations between the thermal environment, climate and density of building fabric.

RESULTS AND ANALYSIS

The Context

Physical survey of the four residential districts revealed characteristics given in Table 1. The first row gives the existing *floor area ratio* (FAR) of each of the survey areas, calculated from the volume of built spaces per unit of ground surface area. This ratio is directly proportional to density of development and is found highest in Sutrapur, closely followed by Dhanmondi.

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[*] Kestrel 3000 Pocket Weather Meter; Nielson-Kellerman; USA

[†] Programmable data logger HOBO H08-007-02. Logger is initiated by software BoxCar Pro 4 (BCP4.0-ON) supplied with the logger. Measure:RH/Temp/2x External.

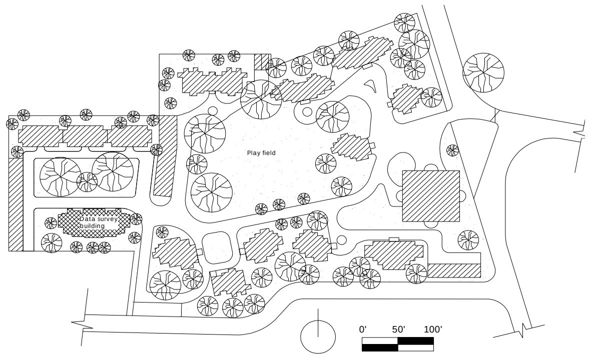

Figure 2a: Low density area at BUET

A new set of construction rules has just been implemented in Dhaka [11], but its effects will not be felt until construction in accordance with it has taken place. Therefore the following discussion is based on the previous Building Construction Act of 1996 [12]. With only 1.25m side setback and 1.5m front setback, residential districts therefore are very compact and dense. Where plots are small, setbacks are further reduced. Owners therefore tend to build upto the limit, leaving almost no open spaces, e.g. in the Sutrapur area. There is also mass scale violation of building construction rules and unauthorized changes in design after formal approval. All this leads to high land coverage and very high density, as seen in the second row of Table 1, resulting in reduced gap between adjacent buildings, poor provision for natural lighting and ventilation.

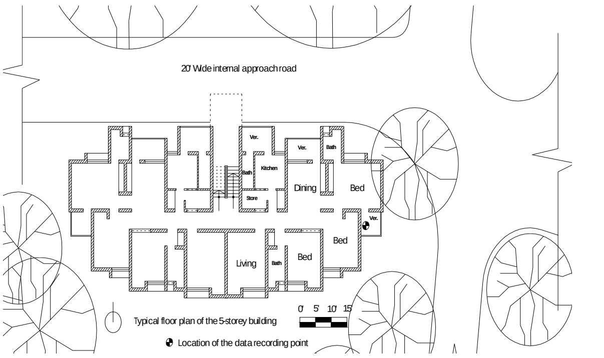

Figure 2b: Plan of Surveyed building at BUET

In the Sutrapur area, a good mixture of buildings of variable heights were found, which created a rough urban texture. In Dhanmondi, most buildings tend to be six storied, and in BUET the norm is five storey walk-ups, with only one eleven storey tower building. The Banani area also has a mix of building heights, many only two storeys high.

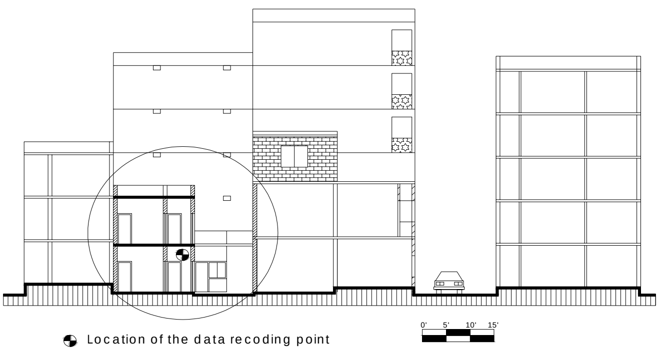

Figure 3a: Lack of setback around buildings, Sutrapur.

Figure 3b: Typical Dense Settlement, Sutrapur

Only in Banani and BUET (Fig. 2a) one-third or more of the plot areas were kept open for vegetation and green development. In areas where the hard to soft ratio is low, evaporation potential increases, while solar absorption is also much reduced.

Population density was found highest in Sutrapur (Figs 3a, 3b), while many professional activities were discovered going on in and around the residences. In Dhanmondi (Figs 4a, 4b) also various commercial activities are encroaching onto the residential environment, which has increased the activity level of this area. Such activities and densities are bound to contribute to higher temperatures.

Figure 4a: Survey Spot: Dhanmondi

Building spacing is seen to vary despite the same setback rules being applicable in all the settings. In Sutrapur, the buildings are constructed very compactly without leaving setback spaces on all four sides. In most cases, buildings in adjacent plots were built without keeping any space in between and no window opening on that side was possible (Fig. 3a). Window openings in the ground floor were only found possible on the road side. Windows on sides are only possible on upper floors, where the adjacent building is lower. These upper storied openings are liable to remain closed permanently, once surrounding adjacent structures are extended vertically. So the provision for natural light and ventilation in the living spaces in this residential area is extremely restricted.

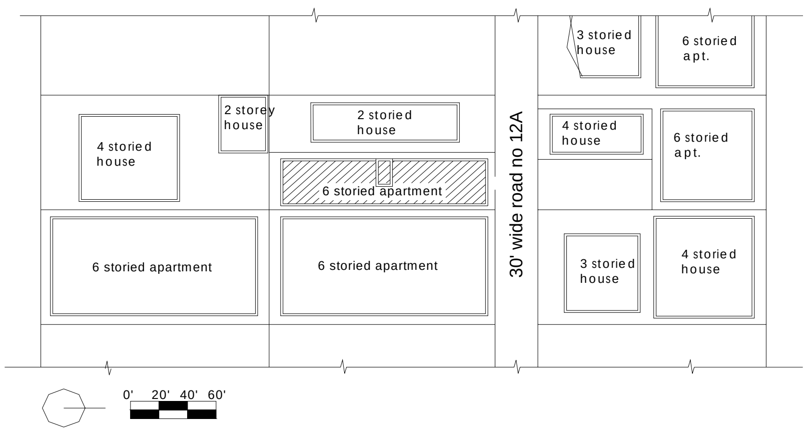

Figure 4b: Plans of surveyed buildings in Dhanmondi

This compact situation acts like a big porous insulating layer between the sky and the earth. The heat that penetrates inside the buildings during day time can not be easily released after sunset. As summer goes on, the cumulative effect could raise the overall indoor temperature posing threat to thermal comfort.

Figure 4c: Section showing transitional space in Survey spot at Dhanmondi.

Because the gap between two buildings was found big enough to facilitate ample light and ventilation through the different vertical layers of the buildings, Banani and BUET area show a different pattern (Fig 2a). During summer, nocturnal cooling can be expedited by this ventilation and by avoiding daytime ventilation, comfortable situations can be retained for a long time.

Temperature and Wind Velocity Data Comparison

Preliminary readings revealed that maximum outdoor air-temperature is found between 1.00 to 3.00 pm. For this reason, temperature at 2.00 pm. taken for fifteen consecutive days at transition points in the four survey spots (Fig 5) shows that generally, Dhanmondi attained highest and Banani the lowest temperatures compared to the other areas.

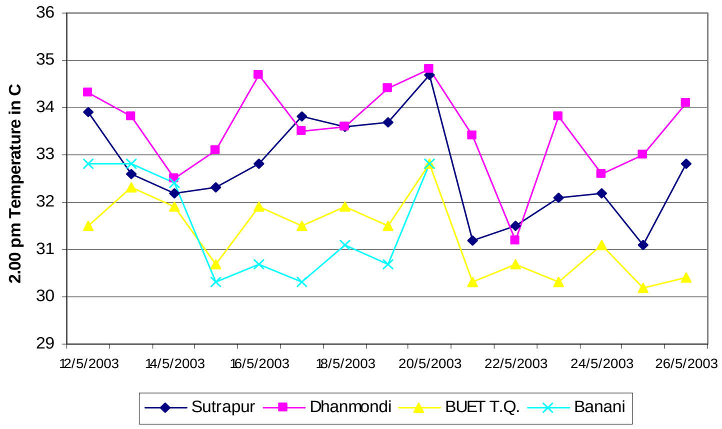

Figure 5:Comparative 2.00pm temperatures in the surveyed spots.

The results also show that the temperature figures are closely related in the Sutrapur and Dhanmondi region (both high density areas) on the one hand, and in the BUET and Banani region (both of lower densities) on the other. Mostly variations between spots were significant, being on an average, about 2°C, reaching a maximum of 4°C. A relationship between the temperature readings and surface quality can also be established from these results. Thus, where the FAR value and proportion of hard surfaces is higher, higher temperature values were obtained.

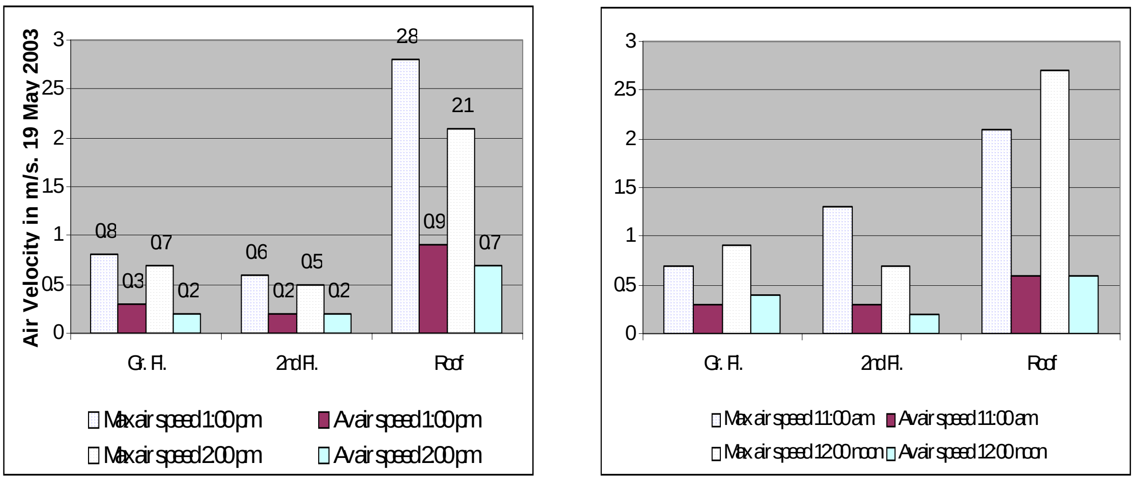

Figure 6: Measured air velocity at different levels of Dhanmondi (left) and BUET (right)

Air velocity readings were also taken at all the survey spots. Fig 6 gives readings for the average and maximum air speeds in high density Dhanmondi and low density BUET. The readings show how the velocity rises with building height for both the low density and high density situations. It is also clear from the readings that there is very little dependence on density when ground level and top level flows are concerned, i.e. almost the same velocity is measured in high and low density areas at these levels. However, at mid-levels, the high density area (Dhanmondi) displays a fall in the wind gradient in comparison with the ground level flow. For the low density area (BUET) on the other hand it is seen that there is a steady increase in wind gradient with height. This is expected because in the former case there is a lot of obstruction at lower levels which results in turbulent and inconsistent wind patterns, while in the case of BUET the wind hits the buildings without its laminar flow being disturbed, as building to building distance is considerable. Therefore there is a steady increase in wind flow with height. This supports research findings that wind velocity reductions at lower levels are more pronounced with increased urban roughness [13] due to turbulence. In the Dhanmondi area, the reason for the air speed at ground level being higher compared to mid levels, could be because the ground floor is kept free for car parking.

DISCUSSION

The following points emerge from an examination of the data:

- Temperature is higher in denser areas. These areas also have a higher percentage of hard surfaces, which may be a contributing factor.
- Wind speed was not found to be directly affected by density of built form, either at ground levels, where there is maximum interruption to airflow, or at roof top levels, where the flow is uninterrupted. But at mid levels, density plays a significant role, possibly due to the higher turbulence in dense spaces, where building heights are also variable.
- It is possible that the higher temperatures measured in denser areas may be caused due to lower wind speeds in these spaces. Therefore the accumulated hot air does not find easy escape and the heat remains trapped.
- As the global horizontal solar radiation data is unlikely to vary in spaces so close to each other, it is possible that the radiation is absorbed by the greater percentage of hard surfaces in the denser setting. This may cause the elevated temperature in the denser regions.

The hypothesis, that a dense area can create a situation similar to a big porous insulating layer between the sky and the earth, trapping heat, seems to be justified from the findings. The effect of density variation is therefore reflected in the comparative differences in the temperature and wind speed measurements.

Recommendations

One of the objectives of the study was to be able to suggest ways in which the situation can be made more comfortable, i.e. where wind speeds can be increased at the lower levels and temperature levels can be kept down. Whereas it is understood that any recommendations should be tested before being considered as having a strong basis, nevertheless, some pointers towards improving the situation, have been tentatively presented below.

1. More openings and height variations at mid levels would undoubtedly increase wind flow at those levels, thus increasing comfort under warm humid conditions. This could be done by introducing terraces interspersed within the built fabric, bringing variation in the building volume.
2. Increasing soft surfaces could bring temperatures down, thus reducing dependency on wind flow to impart thermal comfort. To achieve this, vegetation can be introduced at different levels, cutting down on solar radiation gains.
3. Creating some open areas within an otherwise dense situation would undoubtedly alleviate a lot of the wind and temperature stagnation. Such an open area could draw winds out from the surrounding dense areas, thus improving the overall situation.

Building activity is a continuing activity in Bangladesh, as in much of the developing World. It is imperative to be conscious of the effects of urbanisation and building activity on comfort potential and energy consumption, both of which can be brought to acceptable levels if adequate ventilation can be targeted. This paper is an attempt to make concerned professionals aware of these facets of Architecture and the environment.

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Ventilation Design in High-Rise Residential Buildings and Infectious Disease Spread

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SUMMARY

In the SARS epidemics in 2003, cluster of cases occurred in high-rise residential(HRR) building blocks, especially in Hong Kong, which gave rise to the concern of the possible roles of air flow. In this paper, the multiple parallel airborne transmission routes are discussed. In particular we closely investigated one of the most likely virus-spread paths, which is related to single-side ventilation air flow through open windows caused by buoyancy effects. Both tracer gas and CFD (computational fluid dynamics) techniques have been employed, and tt was found that the upper floor air can contain up to 7% of exhaust air directly from the lower floor. The results can well explain the RNA fragments of Corono-Virus (CoV) found within the sampled deposits on the window sills of the upper floors of the two index patients' flats during the SARS outbreak. Implication for ventilation design and infection control in HRR will be discussed.

INTRODUCTION

Natural ventilation plays two essential roles traditionally in our buildings. On one hand, ventilation can modulate the indoor temperature for better thermal comfort in summer since outdoor air is usually lower than indoors; on the other hand, it also helps to dilute indoor air pollutants and improve indoor air quality in most cases. The required quantity and quality are actually different for these two purposes. The roles of natural ventilation in modern high rise residential buildings are more complicated. Especially, in the rapid developing Asian regions, a majority of the population live in high-rise, multiple-households apartment buildings. The climatic conditions also render air-conditioning widely applied, with the window-type and split-unit type air-conditioners dominating in the residential buildings. Ventilation is provided via open windows, with exhaust fans provided in the toilets and kitchens. The exhaust air is typically not centrally stacked, but left to drift around the building blocks. This can potentially have many unwanted consequences for multi-households, multi-story high-rise buildings (Niu, 2004). On the ordinary days, it's not uncommon for the residents to detect from the smell in his own unit what the neighbor is cooking – a sign of cross-contamination of the ventilation air. With respect to infectious disease control, this inter-flat or inter-zonal flow becomes a serious issue.

When airborne infectious disease spread is concerned, there are also a number of other parallel transmission routes from person to person. Interior corridors are typically poorly ventilated though the residents tend to spend little time there; lift tend to be crowded, and lift lobbies are usually poorly ventilated as well. Depending on the floor plan, direct, horizontal air flow from one unit to the adjacent unit can occur. In particular, to maximize the land-use efficiency, high plot-ratio, defined as the total floor area over the land area, can reach to 10 in urban development. This tends to result in both horizontally-close units and high-rise designs (Figure 1), which increase the risk of air-borne infection spread (Yu et al 2004). It appears that infection risks associated with these routes have not been well studied, and many infectious diseases like common cold, influenza and tuberculosis may be transmitted through these routes.

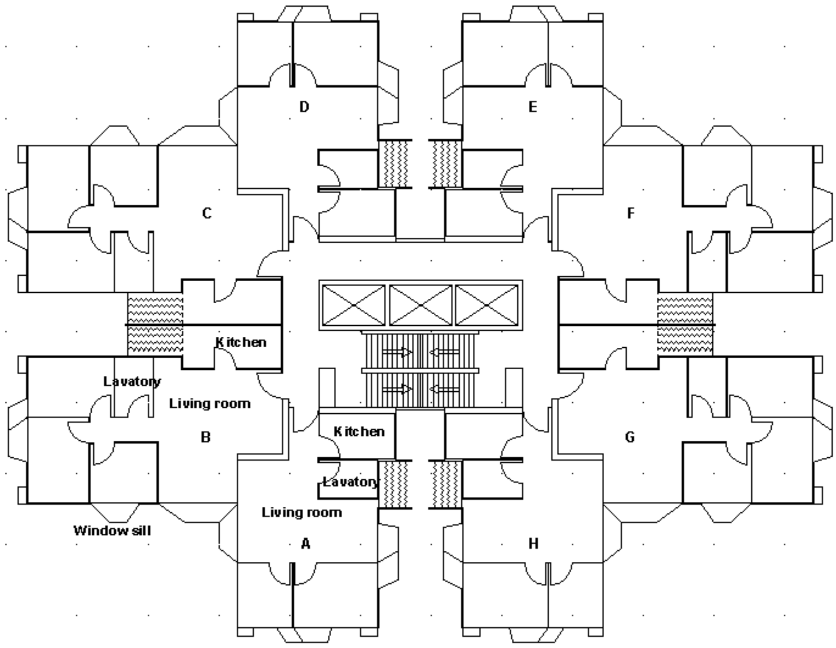

Figure 1. Typical floor plan for high-rise residential apartment buildings, typically around 30 stories high. In this plan, 8 units on each floor share interior corridors and the three lifts.

This investigation is focused on the vertical transmission pattern in high-rise residential buildings observed during the SARS outbreak in the spring of 2003 in Hong Kong. In apartment building A, which is about 30 story high, and where five households were affected with eleven SARS infection cases, RNA fragments of SARS virus were detected within the sampled deposits on the window sills of the upper floors of the two index patients' flats; and in one of these two upper floor households there were no infected SARS cases. In another high-rise building B, all the 6 infection cases in 3 households occurred along one vertical block. In the widely-publicized outbreak in another residential estate, which includes five blocks of 30 story apartment buildings, and where a total number of 321 infections occurred, only one case occurred below the 8th floor. While the transmission paths of such large scale outbreaks remain a myth up to the present moment (Yu et al 2004), it appears that there was a vertical spread pattern.

Our hypothesis is that, on windless days, the outflow from a window on the lower floor will re-enter the adjacent upper floor, and therefore bring the virus-laden droplets generated by an index patient upstairs. For high-rise buildings, there may exist an upward "cascade" air flow, so that the disease can be transmitted from a lower resident all the way to the top floors. The objective of this project is to reveal the mechanisms of this airflow by actually quantifying the fraction of the lower-floor exhaust air within the intake air of the adjacent upper floor. This paper reports the preliminary finding of this study.

SINGLE SIDE VENTILATION

The driving forces of natural ventilation can be classified into two categories: one is the wind pressure, and the other is the buoyancy/stack effect caused by temperature differences. Single ventilation refers to the purposely provision of vent openings on the single side of a room space, so that no cross-ventilation is possible. The original intention is to minimize the effect of wind pressure, and maximize the stack effect due to indoor/outdoor temperature differences. Typically, indoor temperatures are higher than outdoors under natural conditions, outdoor cold air tends to enter the room at the lower apart of an open window and at the same time warm air tends to leave the room at the upper part of the window. In buildings with multiple story and multiple flats on each floor, the overall air flow path through the multiple openings can be highly variable depending on the wind direction and wind speed. With the floor plan as illustrated in Figure 1, residents tend to close their doors when entering their home, so that essentially single side ventilation situation arises. Under high wind conditions, cross ventilations may still be significant via door gaps, but this often causes undesirable horizontal air contaminant transport from one unit to the other. In fact it so happened in an extreme case that VOC(volatile organic compounds) emissions from paint spray during a decoration was suspected to be the cause of an infant death of the neighboring unit. Here it should be emphasized that under low wind conditions, single-side ventilation will be the dominant air flow pattern, as illustrated in Figure 2 and 3. From the point of view of air pollutant accumulation, the worst-case occurs on windless days. In the Hong Kong climate, the windless days can occur in March and April, which happened to be the period during which SARS case clusters. According to our previous survey, residents in Hong Kong may only switch on the exhaust fans in their kitchen or toilets when these two rooms are actually used. In this case, buoyancy due to indoor-outdoor temperature difference will be the only driving force. In multi-story high-rise buildings, it is suspected that this single side air flow may cause vertical transmission of air-borne infectious diseases. In this study, we attempt to quantify the percentage of air from the lower floor present in the upper floor via this route.

INVESTIGATION METHODS AND RESULTS

Onsite tracer gas measurements

This investigation is performed using experimental tracer-gas and flow visualization technique. The building investigated has single-side natural ventilation via open windows. Two rooms located at immediate upper and lower floors, which have windows flush with the façade (Figure 2), were selected for site measurements. The floor to ceiling height is about 2.7 meters, and the room depth is about 5 meters, and the width is about 3.5 meters. This façade feature is similar with buildings A and B as described in the introduction, where clusters of SARS case occurred and vertical transmission patterns were observed. The measurements were continuously conducted form one month in February and March respectively in 2004 and 2005. The experiment was conducted in the spring time because it was the season of SARS outbreak, and is also the high season of common-cold and influenza occurrences in Hong Kong. During the periods of measurement, the outdoor temperatures varied from 10.6 to 24.0^{o}C with the mean of 17.3^{o}C. The indoor temperatures of the two rooms varied 12.8 to 23.3^{o}C (mean = 18.4^{o}C) and 14.1 to 23.1^{o}C (mean = 18.7^{o}C), respectively.

In the on site experimental studies, SF6 was released at a constant release rate in the center of the single room flat in the lower floor, while the SF6 concentration levels at six points were monitored simultaneously using a B&K1302 multi–gas monitor (Figure 3). An example of the monitored concentrations upstairs and downstairs are shown in Figure 4. At the same time, a smoke generating machine released smokes into the lower flat, and air movement of the exhaust air from the lower floor, and possible direct re-entry into the upper floor can be visualized by the smoke movement. Since we cannot control the weather, the flow phenomena we observed would be the combined results of both the wind pressure and buoyancy effects.

Figure 2. Two rooms located in adjacent upper and lower floors in a student dormitory were selected for tracer gas measurements

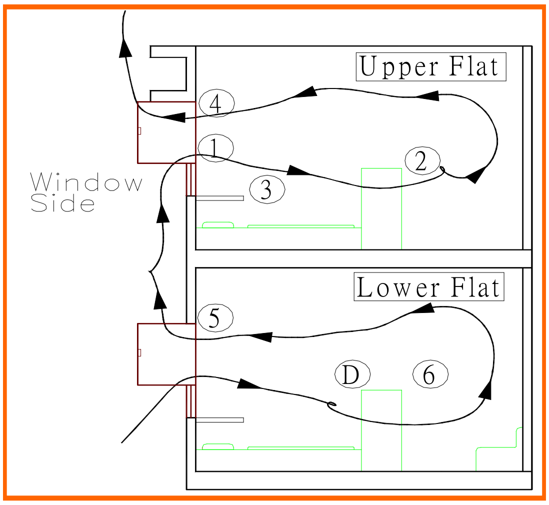

Figure 3. Possible air flow path associated with single side ventilation and locations of the tracer gas dosing point (D), and sampling point (1 to 6)

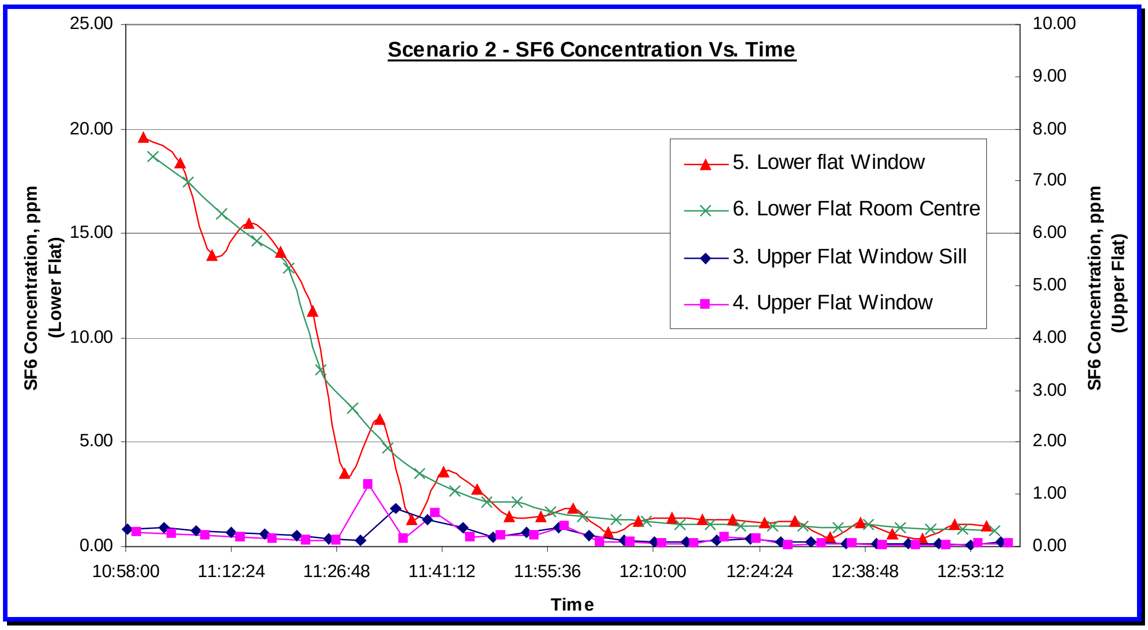

Figure 4. Monitored tracer gas concentrations in the two flats when tracer gas was released in the low floor

Based upon these monitored tracer-gas concentrations, an index, called mass fraction *M*, can be worked out. For instance, M_{6-3} can be calculated based upon the monitored tracer gas concentration at point 3 and 6, which would indicate the fraction of the air originating from point 6 but present at point 3. Using the one month data obtained in 2005, such an index is plotted as Figure 5, it can be seen that the maximum value of the mass fraction is 0.07, which means 7% of the air in the upper floor directly comes from the lower floor. It appears that higher wind decreases the re-entrance slightly.

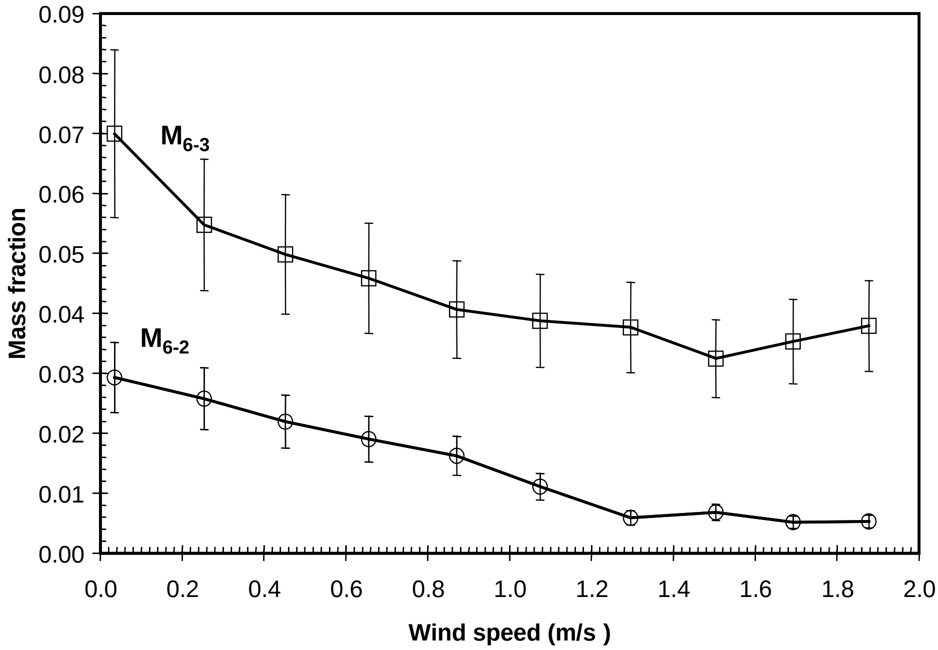

Figure 5 The mass fractions indicating the presence percentages of air coming from the lower floor at two points in the upper floor.

CFD simulation

CFD(computational fluid dynamics) simulation technique (Launder and Spalding 1974; Niu and van der Kooi 1992a, 1992b) was also employed. Both transient and steady-state simulation results have been obtained using the standard k-ε turbulence model. The reliability of the numerical simulation has been first checked with good quality laboratory experimental results obtained from Denmark (Heiselburg 2003). In Heiselburg's laboratory experiments, the transient air exchange via a single-side ventilation window was measured using fast-response anemometry system, and the indoor surface and outdoor air temperature differences were closed monitored.

Presented in Figure 6, is the simulated air flow path-lines and the velocity magnitude through two open windows of two adjacent vertical floors. In general, the velocity caused by buoyancy alone is rather low, with the maximum around 0.5 m/s, as indicated by the colored zones in Figure 6, 7, and 8. On a windless day of outdoor air temperature of 20 ºC and indoor room surface temperature of 28ºC, it can be seen in Figure 8 that the outdoor air enters the lower room through the lower part of the window, and leaves the room at the upper part of the window. At the same time, it can be clearly seen in Figure 7 that a portion of the outflow from the low floor re-enters the upper room through the lower part of the upper window. It looks as if the warm exhaust plume from the lower floor functions as an air curtain, with which ambient air has to mix first to enter the room. Based upon the calculated velocity across the two window openings, air change rates through the two window openings have also been calculated, and tracer gas have also been added into the lower room, and similar results as for the onsite measurements have been obtained.

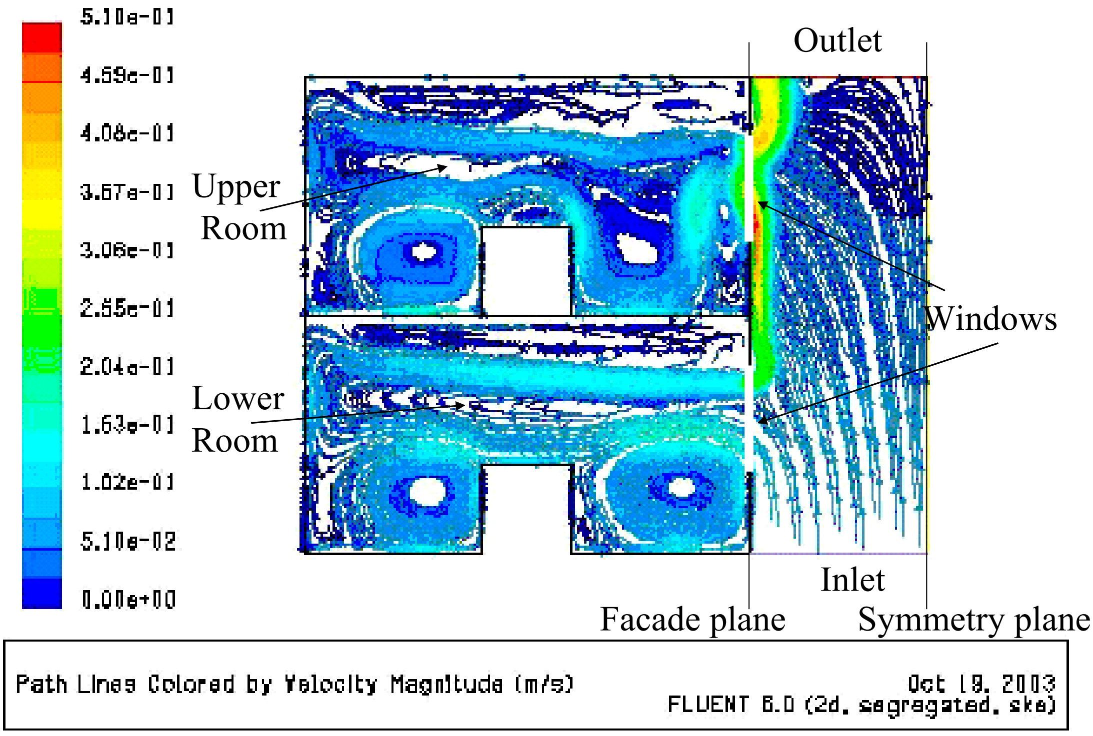

Figure 6. Air flow patterns indicated by flow path lines

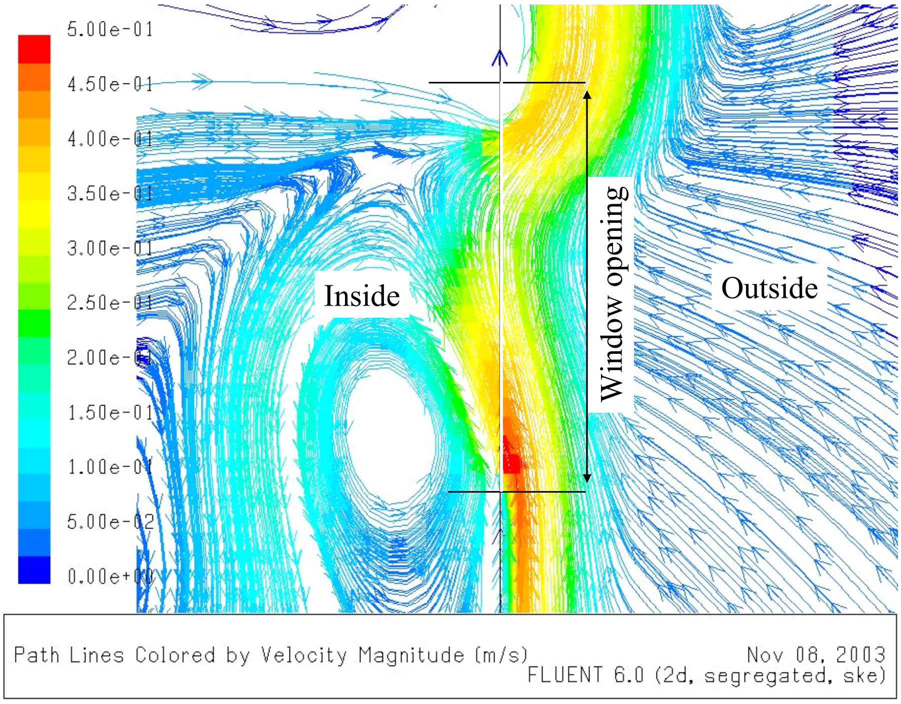

Figure 7. Airflow path lines across the window at the upper floor

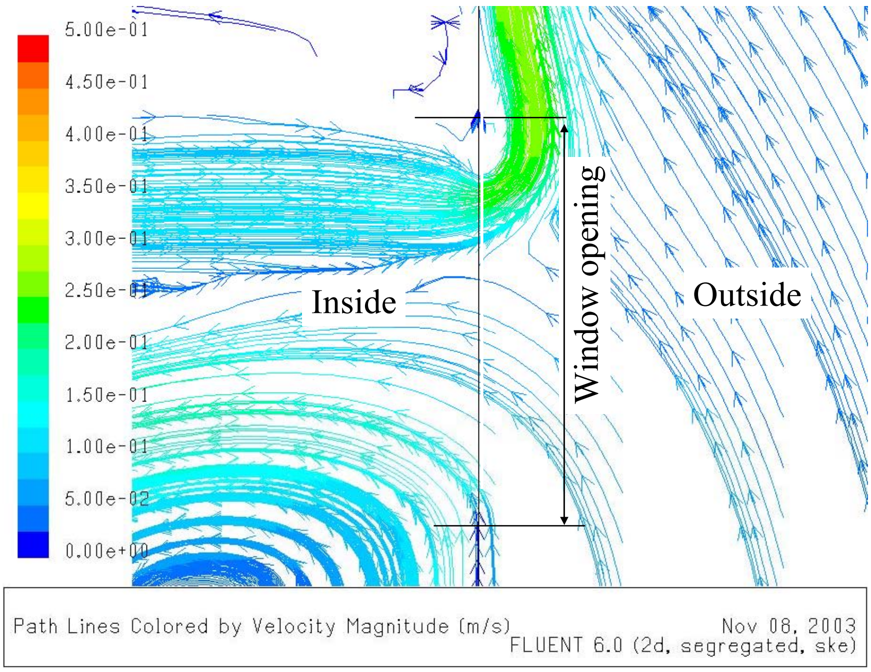

Figure 8. Air flow path lines across the window at the lower floor

CONCLUSION AND IMPLICATIONS

During the SARS outbreak, the presence of SARS virus RNA strings were found in the window deposits, and this study revealed the virus could have been transported up by the single side ventilation air flow. It should be noted that, depending on the window opening habits of the residents, this could be a continuous transmission route. More epidemiological data on other aerosol transmitted diseases would also be helpful to further confirm this significant transmission route. The confirmation of the transmission route can have a number of implications to both building design and infectious disease control. With respect to the former, windows flush with façade should be avoided, and ledges added above the window can reduce the transmission to a certain extent. It appears that complete avoidance of this cross contamination through ventilation control is costly and difficult. With respect to the latter, more targeted and sooner intervention can be implemented in case of any highly-infectious disease outbreaks, which may be more cost-effective.

ACKNOWLEDGEMENTS

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Numerical Study on a Hybrid System by Using a Portable Air Conditioner

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SUMMARY

This paper presents a new design concept of hybrid air conditioning system by using a portable air conditioner to achieve a better personal control of indoor environment without sacrificing the responsibility to a sustainable environment. The results of the detailed computational flow dynamics (CFD) simulation in this study, through a systematic analysis on the parameters of wall opening, outdoor wind speed and temperature, show that with an appropriate design of wall openings, this hybrid system can sustain a comfortable environment until affected by strong wind blast. At that moment, a suitable control of the sizes of these openings is suggested to extend the proper functioning of this hybrid system to achieve the dual benefits of comfort and energy conservation.

INTRODUCTION

In recent years, personalized cooling systems are under research and development to reduce CO_2 emission and achieve energy conservation. Most of the research focused on commercial HVAC systems to improve the energy performance, air quality and productivity of indoor environment. Akimoto et al. [1] and Sasaki et al. [2] developed a task air conditioning system and conducted a subjective field study on the thermal comfort and productivity. Furthermore, Chiang et al. [3, 4, 5] designed a new partition-type fan-coil unit (PFCU) system to provide the users with greater flexibility and convenience for dedicated local environment control. According to their results, as compared with a conventional air-conditioning system using ceiling diffusers, this personalized cooling system can reduce about 45% of cooling energy consumption.

The same concept can also be applied to the residential dwelling or small office buildings. However, at these places, portable air conditioner is more flexible to be implemented than the task or PFCU air-conditioning systems. Actually it has been used as a spot cooler for a long time to provide occupants cooling without wasting energy to cool the entire room. The cooling capacity can be largely reduced to as little as 500 W in comparison to several kWs for the one using a conventional window or split-type room air conditioner. However, the need of a hose to ventilate the hot air discharged from the condenser to the outdoor often prohibits this appliance from being widely used. In this regard, this paper intends to develop a hybrid air-conditioning system to create an environment with natural but comfortable room air circulation that combines the buoyancy-induced airflow formed by the direct hot air discharged from this machine and the ventilation air flowing through some wall openings.

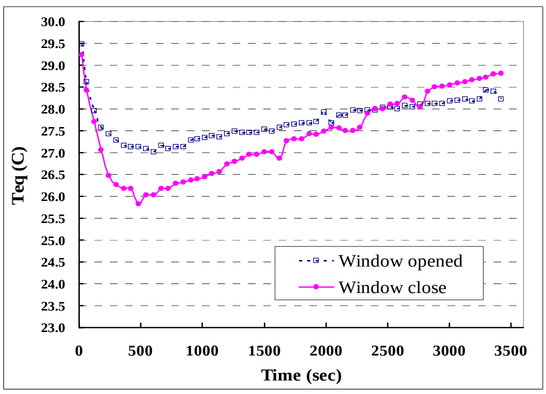
Aiming at the hybrid air-conditioning system in an office building, Chang et al. [6] studied a natural ventilation system for a model office building where air flows in through the opening on one wall, removes the heat and contaminants generated at the task zone and exhausts to the outdoor through another opening on the opposite wall adjacent to a courtyard. Mechanical cooling system with underfloor air distribution is used as assisted cooling only when the natural ventilation is unachievable to meet the demand. Their work and Axley et al. [7] both indicate that the outdoor conditions and the designs of wall openings are the crucial parameters which affect the performance of the hybrid air-conditioning system. For residential buildings, the circumstances may be different because of the limitations of building structures and the use of a portable air conditioner. Therefore, the objective of this study is to achieve the following conditions so that the hybrid ventilation system can perform its ideal design functions in a residential building as well,

1. The hot air discharged from the condenser must rise as a thermal plume [8] to the upper portion in the room and, in the process, avoid any mixing with room air.
2. Thermal stratification must be established such that hot air can be exhausted from the upper opening on the wall.
3. Sufficient outdoor air must be introduced to make up the air being exhausted.

METHODS

The concept of designing this hybrid system was first proved by conducting a simple experiment. As shown in Figure 1(a), a spot cooler supplied cool air to a test room and a manikin sat inside the room to measure the comfort level. Initially, the temperature of the test room was set at 28^{o}C and the outdoor chamber was controlled at 30^{o}C. After the machine was turned on, the equivalent temperature perceived by the manikin dropped to the lowest values in 10 minutes and then gradually increased because of the accumulation of heat discharged by the condenser in the room. The variations of equivalent temperature were compared for two test conditions: one with a window opened on the wall and the other with the window close. For both cases, the equivalent temperatures eventually rose to an uncomfortable circumstance because heat was not efficiently removed from the space. However, when the window was opened, the indoor temperature increased in a slower rate. But, due to a large amount of hot air continually moved into the room from outdoor without any control mechanism, the perceived temperature of the manikin was finally not in the acceptable range. This experiment implies that we can avoid the accumulation of heat in the room by having an opening but, to achieve this hybrid system correctly functioning, a proper design of the opening is the prerequisite.

(a) Pictures of the experiment room



(b) Equivalent temperature perceived by a thermal manikin

Figure 1. Experimental set-up for testing a portable air conditioner and its test results

Although it is not difficult to study the proper wall openings in a climatic chamber, simulating the outdoor wind effect is not easy at this moment. Therefore, this paper uses computational flow dynamics (CFD) method to complete a systematic analysis on the influential parameters such as wall opening, outdoor wind speed and temperature. Moreover, the use of CFD method can reveal detailed phenomena of flow fields and temperature distributions in such a closely interacting system.

In this study, a model room, as shown in Figure 2, has the size of 4m x 3m x 5m. The heat gains from the ceiling and through the floor are assumed to be 12 W/m^2 and 18 W/m^2, respectively. The influence of outdoor temperatures is studied by adopting two temperatures of 28^{o}C and 25 ^{o}C, corresponding to a severe and a mild outdoor condition. The impact of outdoor wind blowing toward this building is analyzed under the conditions from the static wind (namely, completely calm) to a maximum wind speed of 0.75 m/s. Inside the model room, a portable air conditioner is placed on a short stool of 0.6m height and about 2m away from the left wall. An occupant in sitting position faces toward the portable air conditioner and is subject to the cool air supplied by this machine. Cool air leaving the evaporator with the air flow volume of 100CFM (air speed 0.84m/s) and temperature of 20^{o}C. The hot air discharged by the condenser has the air volume of 80CFM (air speed 0.63m/s) and temperature of 40^{o}C. It should be noted that we have slightly increased the hot air discharge ports on the machine so that the air speed is reduced so as to be able to form a thermal plume flow rising from the machine. This modification avoids the mixing of the discharged hot air with the room air and ensures an acceptable thermal stratification in the room.

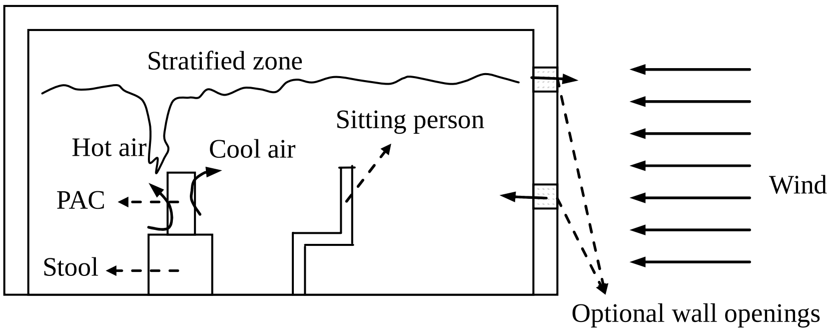
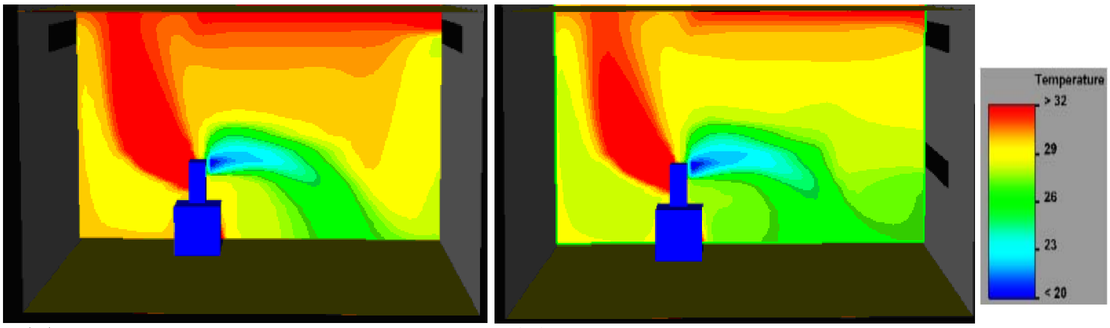


Figure 2. Sketch of the model room for CFD simulation

RESULTS

From the preliminary experimental test results, as demonstrated in Figure 1, we have found that although hot air discharged from the portable air conditioner can be exhausted to outdoor through the opened window. However, proper design of the openings is very important to form proper natural ventilation, including the processes of introducing cooler air into the room, removing interior heat loads and exhausting hot air outside. Chang et al. [6] investigated the hybrid ventilation in the intermediate seasons for an office room with large depth. There are openings on the opposite two side-walls in their model room. Low temperature outdoor air flows in through the opening on the exterior wall and rejects from another opening on the wall adjacent to a courtyard. The air flow in their model room is in a unilateral direction. The implementation of the similar design in this study, however, is found inferior to the design with two opening on the same wall. As depicted in Figure 3(a), under the circumstance that the wind speeds is zero, high temperature air tends to accumulate in the upper portion of the room and air movement is almost blocked. In contrast, for the design with two openings on the same wall, as indicated in Figure 3(b), the outdoor air can easily flow through the lower opening to the room and discharge from the upper opening. This hybrid system demonstrates a smooth air circulation and a favored thermal stratification.

Hence, the follow-up discussion of this paper will focus on this kind of openings deign to explore the effect of opening size, outdoor temperature and wind speed.



（a） openings on two opposite walls （b） openings on the same wall

Figure 3. Influence of openings design on the room temperature distribution

After adopting the design of two openings on the same wall, openings with the sizes of 2m x 0.3m and 2m x 0.15m were investigated. Because achieving a comfortable local environment surrounding the occupant is the main goal of this study, a mean perceived temperature is defined to justify the comfort level of the sitting occupant by averaging the air temperatures at the selected positions of 1.1m, 0.6m, 0.1m heights, corresponding to the temperatures experienced by the head, waist and legs of the occupant. The perceived temperature calculated from the CFD simulation results are shown in Figure 4 for different openings and outdoor conditions. It is found that when the outdoor wind speed is below 0.4 m/s, the perceived temperature can be kept at less than 26.5^{o}C, which satisfies the general requirement of a comfortable environment. Also with larger openings, the perceived temperatures are comparatively lower than those of the small openings. It must be due to the fact that the discharged hot air can be more smoothly vented outdoor. However, a reversed situation occurred as the outdoor wind speed is increased to over 0.5 m/s. The occupant might feel a sharp increase of the perceived temperature. To improve this drawback, using a small opening (*eq.* 2m x 0.15m) can completely suppress this problem.

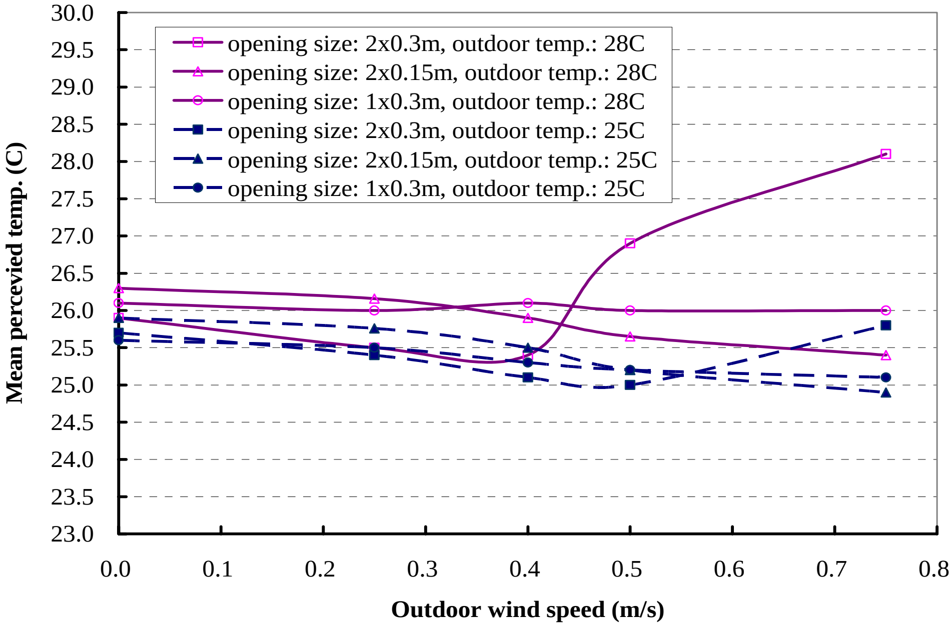
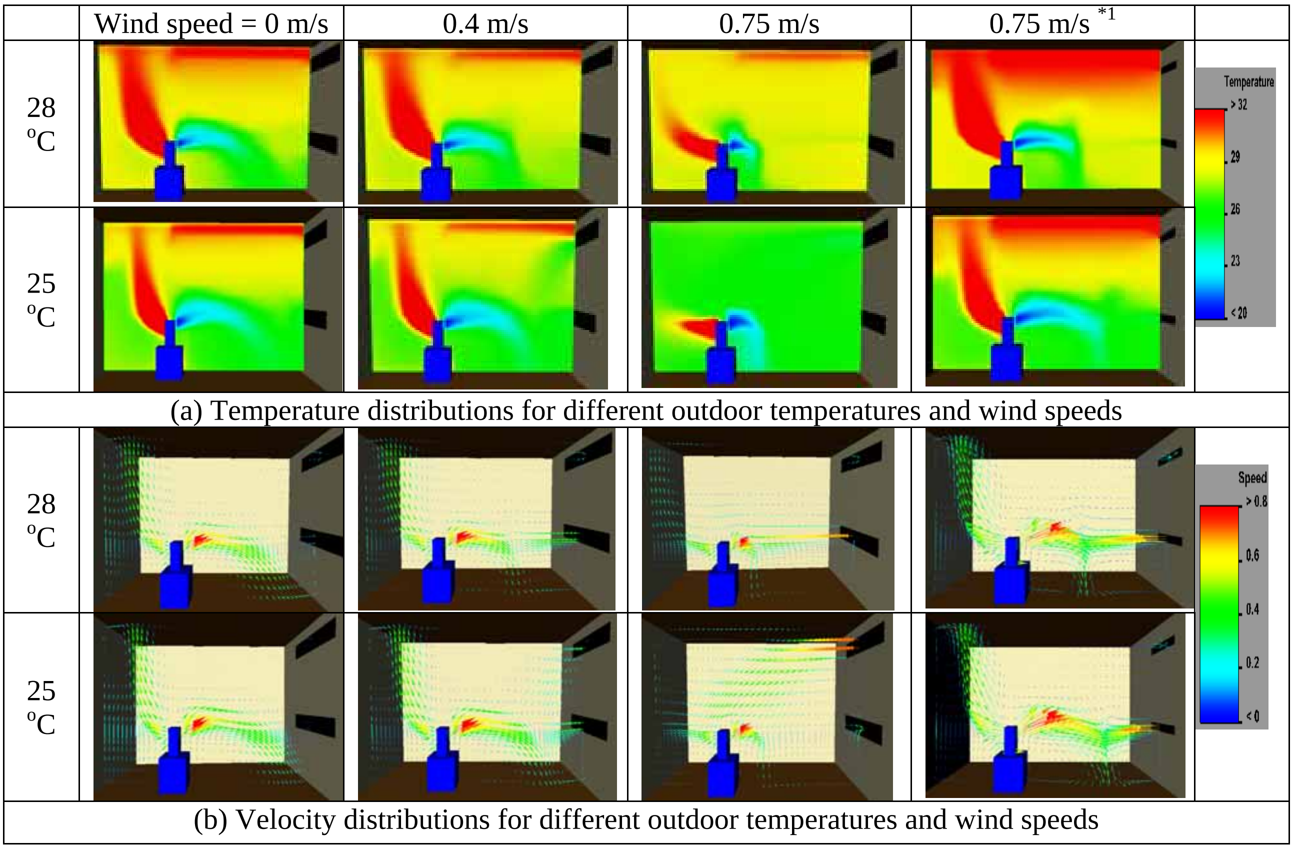


Figure 4. Variations of outdoor conditions vs. mean perceived temperature

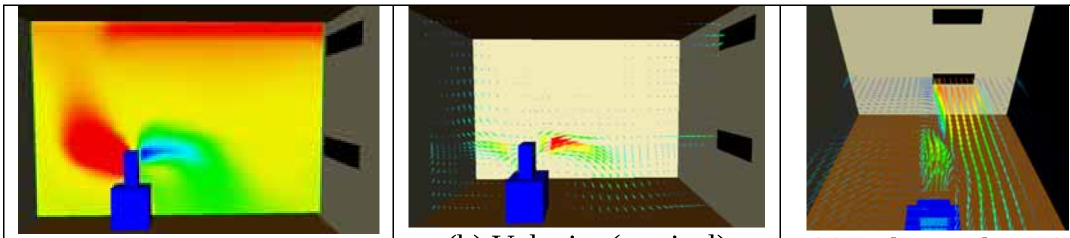
Figure 5 summarizes the detailed air flow patterns created by the interactions of the portable air conditioner and the natural ventilation system. For the high outdoor air temperature (*i.e.* T_o=28°C) cases, the ventilation air always flows in from the lower opening and flows out from the upper opening. As outdoor wind speed is slow, the stratification of temperature distribution is well established in the room. With the movability of this air conditioner, comfortable cooling can be provided to any portion of the space with less energy consumption. The hot air discharged by the machine moves quickly upward without disturbing the occupied zone. Nevertheless, ventilation air flow starts to hold back the cool air supplied from the machine by the increasing outdoor wind speed. The cool air is not able to reach the occupant when the wind speed is higher than 0.75 m/s. In this circumstance, the occupant is subject to complete outdoor conditions. With a reduced opening (*i.e.* 2m x 0.15m), the machine can regain its control of the living environment, but the upper stratification layer dramatically increases. For the low outdoor air temperature (*i.e.* T_o=25°C) cases, the flow patterns present an interesting different feature. Without the assisting of the strong buoyancy force, the cold ventilation air is found to flow in from both openings as the outdoor wind speed increases.



Note:*1 flow fields with reduced openings (*i.e.* 2m x 0.15m)

Figure 5. Air flow distributions of the hybrid system formed by the portable air conditioner and natural ventilation system

Although, reducing the opening sizes from 2m x 0.3m to 2m x 0.15m can mediate the disturbance caused by the high speed air flows through the lower opening, a large stratified zone containing a lot of hot air is formed in the upper portion of the model room. To overcome this deficiency, an improved design by shifting the openings to one corner in the room can successfully avoid the confrontation of two air streams formed by the portable air conditioner and the natural ventilation. As shown in Figure 6, although the inflow through lower opening is of high velocity, the air stream can keeps off the cool air supplied from the machine. Hence, the comfortable level of the occupant can be held in a better condition. This finding implies that if these openings mounted with some adjustable louvers to divert the air flow direction, the portable air conditioner can always perform very well with natural ventilation.



(a)Temperature (b) Velocity (vertical) (c) Velocity (horiz.)

Figure 6. Flow fields for the cases where the openings are shifted to the right corner (opening size: 1m x 0.3m, wind speed: 0.7 m/s, T_o:28^{o}C)

DISCUSSION

A thorough research on a new hybrid air-conditioning system is accomplished. With the room air circulation that combines the buoyancy-induced airflow formed by the direct hot air discharged from a portable air conditioner and the ventilation air flowing through some wall openings, a comfortable environment is achieved while consuming much less cooling energy. The detailed CFD simulation results reveal the detailed flow fields influenced by the major parameters of wall opening, outdoor wind speed and temperature. As manipulating the wall openings design, this hybrid system can sustain a comfortable environment until affected by strong outdoor wind blast. To avoid this circumstance, two additional designs, one with narrow openings and the other with swapped openings, are found to be able to extend the proper functioning of this hybrid system. In practices, mounting these openings with adjustable louvers to divert air flow direction, the portable air conditioner can always perform very well with natural ventilation. Expected flow phenomena at the beginning of this study are all satisfied including the formation of a thermal plume and a thermal stratification in the room and the induced natural ventilation flow through the openings on the walls.

ACKNOWLEDGEMENT

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Application of Hybrid Ventilation System using the Balcony Space in Apartment Housing

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SUMMARY

Recently, the ventilation for the removal of pollutants has received much attention in response to the increased needs for the health and better comforts of the occupants in apartment housing. To provide constant ventilation air flow, constant airflow mechanical ventilation system is predicted to be installed, and the installation of such mechanical system would lead to many problems.

The purpose of this study is to evaluate application of hybrid ventilation system using the balcony space in apartment housing. As a result through research method, this paper begins with a conceptive explanation of the hybrid ventilation system. Also, through the analysis of thermal condition of balcony space and seasonal characteristics of ventilation in Seoul, the design condition of hybrid ventilation system is investigated. The simulation on hybrid ventilation mode showed that natural ventilation is possible 50.96% of time yearly. CFD simulation result for the analysis of visible flow testing showed that the optimal ventilation mode was maintained smoothly during the intermediate period everywhere except in the room next to the kitchen. For the ventilation in summer and winter, mechanical ventilation was determined to be adopted for the more ventilation airflow.

INTRODUCTION

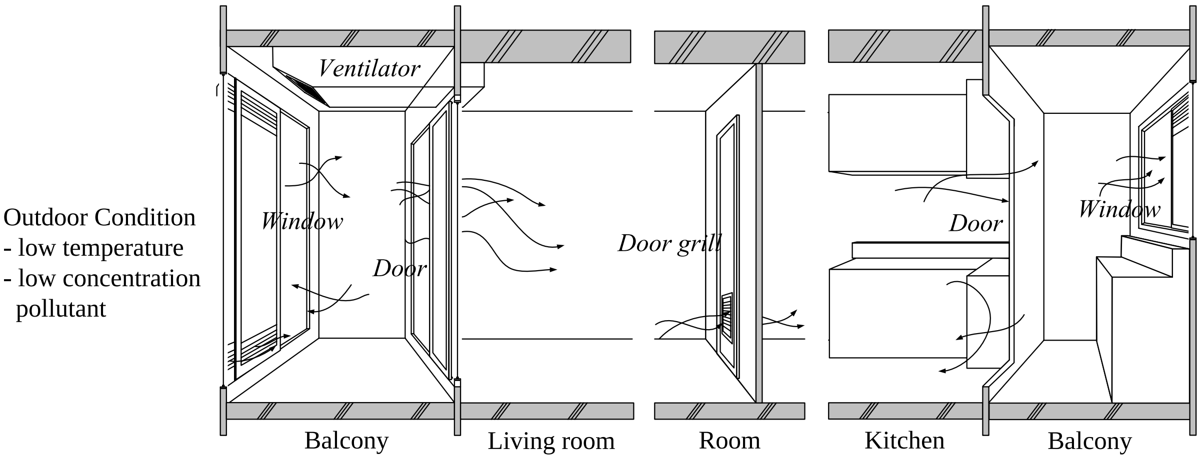
Recently, the quality of indoor air has received much attention as people try to secure a comforting and healthy living environment. Also, the improved construction techniques and energy conservation ideas have made ventilation an essential part of living [1].

Moreover, having to notify the people of indoor air condition of apartments before the moving in under the Code of Indoor Air Quality has raised people's concern about their living environments in apartments [2].

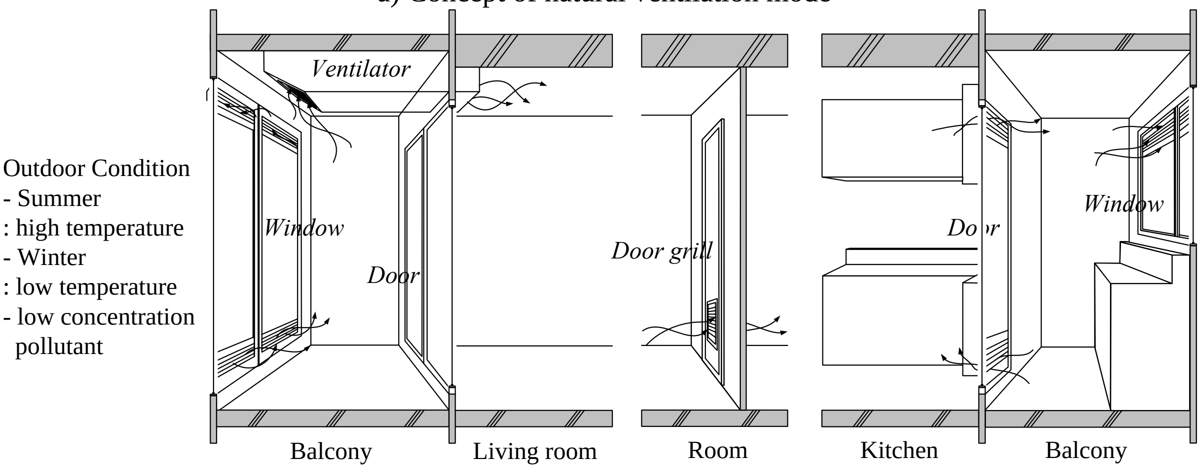
Consequently, the constant airflow mechanical ventilation systems that aim to remove any indoor pollutants are actively being used as the necessity of ventilation in living environments increases. These systems, however, are expected to increase energy consumption because most of them heavily rely on machineries. In addition, Outdoor air induction is expected to influence thermal comfort during the heating and cooling periods

Therefore, the performance of the hybrid ventilation system that jointly uses natural ventilation and mechanical ventilation using the balcony space in apartment housings is evaluated as a proposal in this research.

Apartment housing was selected as the target model of this research. TRNSYS, TRNFLOW and CFD were used to verify the applicability and evaluate the performance of the hybrid ventilation system.



a) Concept of natural ventilation mode



b) Concept of mechanical ventilation mode

Figure 1. Concept of the hybrid ventilation using balcony space proposed in this research

THE CONCEPT OF HYBRID VENTILATION

The hybrid ventilation system uses natural ventilation to maintain the indoor environment status in the spring and fall and the amount of direct solar is reduced by installing sun blinds in the balcony in the summer while providing minimum required ventilation through mechanical ventilation to decrease ventilation load and to remove any indoor pollutants.
The balcony space is minimally ventilated in the winter to use the preheating effect and it is maximally ventilated in the summer in order to maintain a condition similar to outside. Ventilation load is reduced by using a fan coil unit that chills the air coming in from the balcony. Figure 2 illustrates the structure of hybrid ventilation system attached fan coil unit

The concept of the natural ventilation mode

The natural ventilation mode of the hybrid ventilation system can balance and dilute the lower pollutant concentration and the temperature of indoor than outdoor by bringing in outside air. A cross ventilation occurs when opening all the indoor and balcony windows during natural ventilation as shown in Figure 1(a). A pressure difference between inside and outside is induced by operating the ventilation fans in the kitchen and the bathrooms to support the ventilation when there isn't enough natural ventilation taking place.

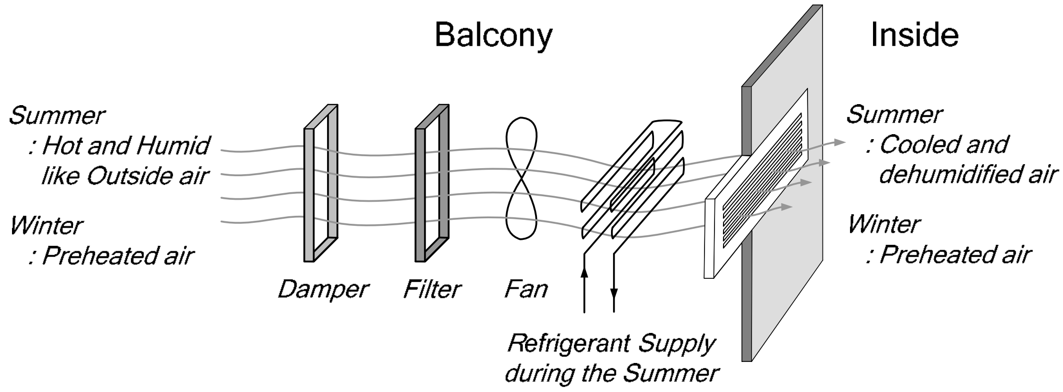


Figure 2. Structure of ventilation system attached fan coil unit

The concept of the mechanical ventilation mode (Figure 1(b))
The mechanical ventilation mode of the hybrid ventilation system secures the required ventilation amount to maintain the indoor air quality. The mechanical ventilation mode is applied when natural ventilation is not applicable due to the season and it chills the air in the balcony before inflow it indoors in the summer. In the winter, it sends in air preheated in the balcony space. The supply air dilutes the indoor pollutant concentration and flows out through the exhaust fan in the kitchen.

SIMULATION

Selecting the target building

The following conditions were considered in selecting the target for evaluating the performance of the hybrid ventilation system in apartment housings.
(1) An RC-structured apartment in Seoul
(2) A new apartment ready for moving-in within the next year
(3) An average sized apartment with high rate of residency for applications in the future
An 85m^2 apartment facing south that was completed in 2005 was selected according to the above conditions. It has been the most common size for the recent 5 years (1998~2002) and its floor plan is shown in Figure 3.

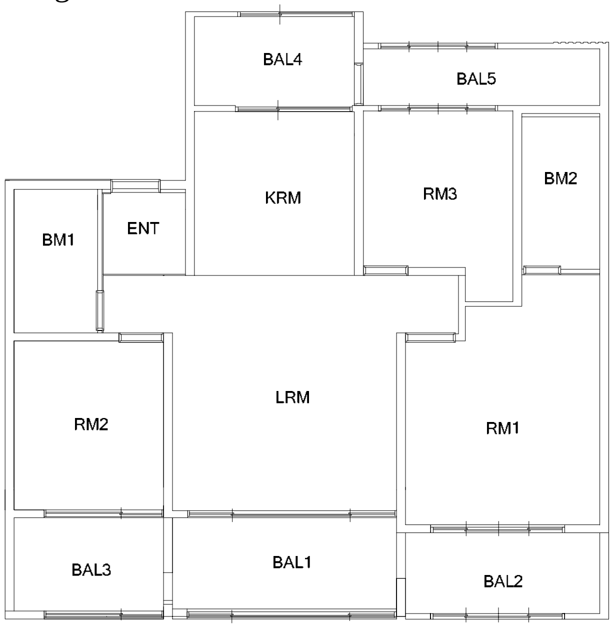


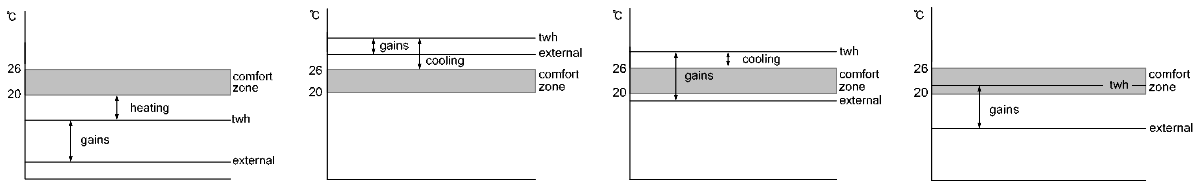
Figure 3. Target evaluation model and zoning

The evaluation model of this research was divided into 8 zones according to their spatial features as shown in Figure 3. Of these zones, heating and cooling are applied to the master bedroom (RM 1), bedroom 1(RM 2), bedroom 2(RM 3), the living room and the kitchen, and the entrance. Natural ventilation window was set up in the window of balcony considering the thermal properties of the balcony.

Simulation Conditions

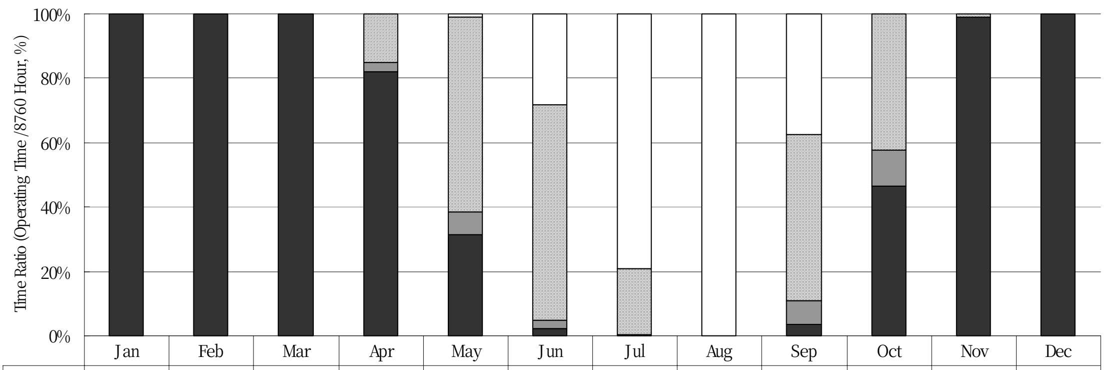
The 33-P type (surface area 85.8 m^2) evaluation model in Seoul is an apartment housing facing south with RC-structure as shown in Figure 3. The ceiling is 2.3m high with envelope's overall heat transmission rate at 0.483 W/m^2 K. Seoul Standard Weather Data (SAREK, 1995) was applied as the weather data.

The experiment assumed that there were 4 family members with weekdays and weekends strictly distinguished. It was assumed that the parents used the master bedroom (RM 1) and each child used each of the two rooms (RM2, RM3). Regular times were set for going to and returning from school (8AM - 6PM), going to and returning from work (7AM - 8PM), meals (7AM, 1PM, 8PM), and sleeping (11PM - 6AM) and the room activities were scheduled accordingly. The air tightness was set to the average apartment housing at 0.22 ACH [3].



MODE 1	MODE 2	MODE 3	MODE 4
Min. vent. (winter)	Adequate vent.(inter-season)	Max. vent.(inter-season)	Min. vent.(summer)
Tout ≤ 8.0℃ (Tin < 20℃)	8.0℃ < Tout ≤ 14.0℃	14.0℃ < Tout ≤ 26.0℃	Tout > 26.0℃ (Tin > 20℃)

(a) Ventilation mode configuration settings



	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
MODE 4	0.00	0.00	0.00	0.00	0.85	28.17	79.17	100.00	37.57	0.00	0.00	0.00
MODE 3	0.00	0.00	0.00	15.21	60.69	67.15	20.63	0.00	51.46	42.23	0.81	0.00
MODE 2	0.00	0.00	0.00	2.64	6.92	2.50	0.00	0.00	7.50	11.31	0.00	0.00
MODE 1	100.00	100.00	100.00	82.15	31.54	2.18	0.20	0.00	3.47	46.46	99.19	100.00

(b) Monthly time ratio of ventilation modes

Figure 4. Configuration settlings of ventilation modes and monthly time ratio

Simulation Results

(1) Ventilation mode settings

The ventilation mode setting is as shown in Figure 4(a) to maintain a comfortable indoor thermal atmosphere. The ventilation method suitable to this country is classified according to

the ranges of the outside temperature and the room temperature and the conditions were configured based on the standard comfortable room temperature (20℃~26℃).
Figure 4(b) shows the monthly ventilation mode usage. Minimal ventilation (MODE 1) is applied in January - May, September - December, when low outside temperatures are brought inside and heating is therefore required to reach a comfortable room temperature. The rate of use of MODE 2 and MODE 3 during these periods was 22.08%. This 22.08% indicates the amount of time required for the outside temperature to be preheated in the balcony before flowing indoors.
MODE 4 is applied from June to August, when the outside temperature is higher and uses the minimal ventilation in order to prevent from any heating overloading. The ratio of the use of MODE2 and MODE 3 was 30.09% during this period.

(2) Optimal operation plans by ventilation modes
The operation condition of the hybrid ventilation system is detailed based on simulation results. It derives the surface area of the opening appropriate for optimal operation through sensitivity analysis.
The operation concept of the hybrid ventilation mode is divided into 4 ventilation modes and the seasonal operation conditions and methods are as shown in Table 1.

EVALUATION

Analysis Plans

In order to evaluate the applicability of the hybrid ventilation system, the airflows of the 4 seasonal modes were examined through the CFD (Computational Fluid Dynamics) simulation analysis and Star-CD was used as the CFD program.
The size of the openings of each ventilation modes were set according to the previous ventilation operation methods, the outside wind speed was set to the average value in Seoul at 2.5m/s, and the wind speed was set to 0.5m/s in the case where exhaust fan has to be operated due to the lack of the natural ventilation.

Analysis Results

(1) Minimum ventilation in the winter
Table 2(a) (1) shows the air flow on the surface of the door louver installed 0.4m above the floor. The outside air in the balcony from the inflow is distributed to room 1, room 2, and the living room through mechanical ventilation (1 ACH) and the air on the inside flows out through the door louvers to the kitchen and the rooms. Most of the air flowing out from the rooms flows into the living room and room 3 and it circulates with the walls of room 3. Therefore the overall air flow seemed moderate except for the bathrooms.
(2) Adequate ventilation in the inter-season
Table 2(b) (1) shows the air flow on the surface at 1.5m above the floor (breathing height). The air velocity when it enters was 2.0 m/s and the overall air velocity was 1.0 m/s. It can be seen that the air is naturally ventilated to the balcony in the kitchen.
(3) When lacking natural ventilation in the inter-season (using ventilation fan)
When lacking natural ventilation in the inter-season, the surface airflow pattern [Table2(c) (1)] is similar to the airflow pattern [Table2 (b) (1)] in adequate natural ventilation but it was 0.5m/s lower overall. It can be seen that air is forcibly ventilated from the kitchen and the bathrooms, airflow is ventilated from the kitchen area to the balcony and through the ventilation fan in the kitchen [Table (2(c) (2) (3)].
(4) Maximum ventilation in the inter-season

An airflow pattern similar to the adequate natural ventilation can be seen on the surface as shown in Table2 (d) (1). Furthermore, the airflow occurs over a wider range than the adequate natural ventilation around spots with faster airflow, and the stagnancy has also decreased.
(5) Minimal ventilation in the summer
Table2 (e) (1) shows the airflow on the surface of a door louver installed 0.4m above the floor. Although it shows airflow similar to that of the winter, the indoor airflow was the lowest in the summer.

CONCLUSION

The hybrid ventilation system was evaluated considering indoor air quality and thermal comfort of a 33-P type (surface area 85.8 ㎡) apartment.
The research can be concluded as following.
(1) A comfortable environment can be maintained with the balcony-using hybrid ventilation system, operating the natural ventilation for 50.96% of the year.
(2) As for the operation strategies of the hybrid ventilation system, the balcony windows were closed, only vent grill (1.9ACH) was used for the ventilation in the winter, the balcony window was opened 1.5 ㎡ for the adequate ventilation in the inter-season (maintain 38 ACH), the inside window was opened 0.5 ㎡ (22 ACH, supply and return air control), the balcony window was opened 3.0 ㎡ during the maximum ventilation in the inter-season (maintain 74 ACH), the inside window was opened 1.0 ㎡ (42 ACH), the balcony window was opened 3.0 ㎡ during the minimum ventilation in the summer (maintain 74 ACH), and the operation strategies by outside temperature is shown in Table 1.
(3) According to the results from CFD simulations of hybrid ventilation modes, the adequate and the maximum ventilation in the inter-season both showed satisfactory air flow. However, room 3 was anticipated to have higher pollutant concentration because it is used as an exhaust air path as well as showing indoor air circulation. The minimum ventilation used the mechanical ventilation system (1 ACH) in the summer and winter showed low airflow as well as frequent stagnations, therefore need for more ventilation and a more fluent airflow.

ACKNOWLEDGEMENT

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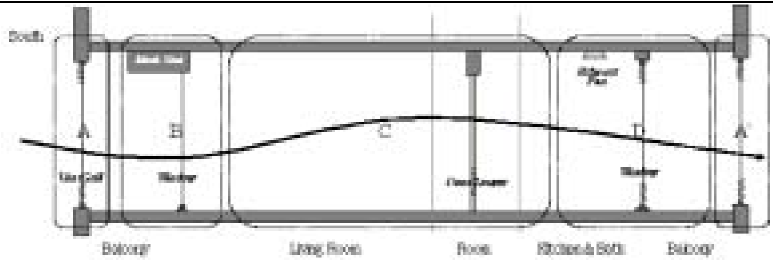
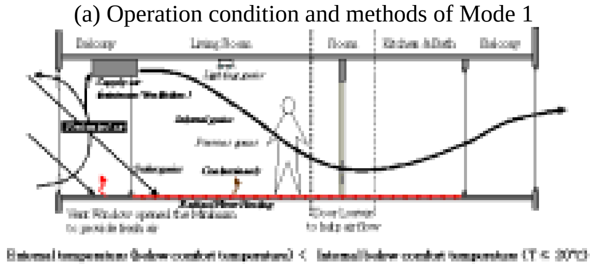
Table1. Ventilation modes and the seasonal operation conditions and methods



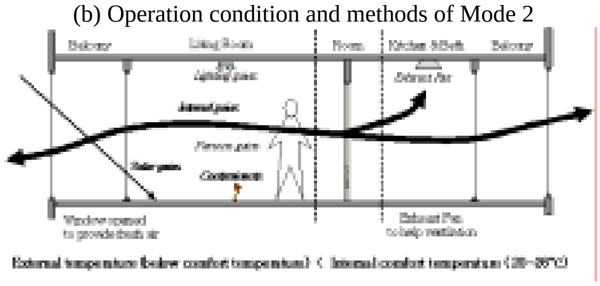
	Ventilation control (A, A')	Supply air control (B)	Heating & Cooling	Exhaust air control (D)
Mode 1; Winter Min. Vent.	Window vent. grill (1.9 ACH)	Mechanical vent.	Radiant floor heating	Window vent. grill (Natural vent.)
Mode 2; Inter-Season Adequate Vent.	1.5 m²; window open (38 ACH)	0.5 m²; window open (22 ACH)	-	0.5 m²; window open (+ Exhaust fan)
Mode 3; Inter-Season Max. Vent.	3.0 m²; window open (74 ACH)	1.0 m²; window open (42 ACH)	-	1.0 m²; window open (42 ACH)
Mode 4;Summer Min. Vent.	3.0 m²; window open (74 ACH)	Mechanical vent. with Fan coil unit	Package air-conditioning	Window vent. grill (Natural vent.)

(a) Operation condition and methods of Mode 1



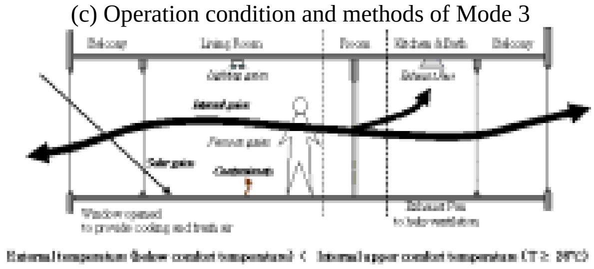
Season	Outdoor air temp.	ACH	Operation
Winter	Min. vent. of balcony Tout 〈8°C	Min. 0.5	△ × (Radiant Floor Heating)
		Medium 0.7	△ × (Radiant Floor Heating)
		Max. 1.0	△ × (Radiant Floor Heating)

(b) Operation condition and methods of Mode 2



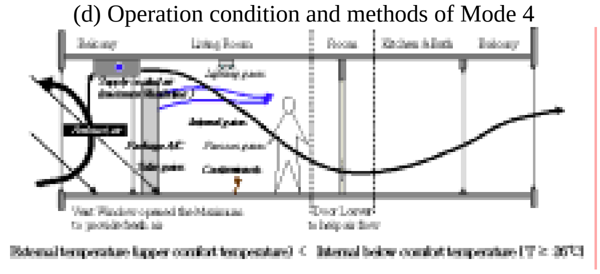
Season	Outdoor air temp.	ACH	Operation
Inter-Season	8 < Tout ≤14℃	Adequate 0.5~2.5	○ (window vent grill)
		Too low ~0.25	○ □ (K:630CMH, B:60CMH)
		Too high 2.5~	○ (Min. vent.)

(c) Operation condition and methods of Mode 3



Season	Outdoor air temp.	ACH	Operation
Inter-Season	14<Tout≤26°C	Adequate 0.5~2.5	○ □
		Too low ~0.25	○ □ (K:630CMH, B:60CMH)
		Too high 2.5~	○(Adequate Vent.)

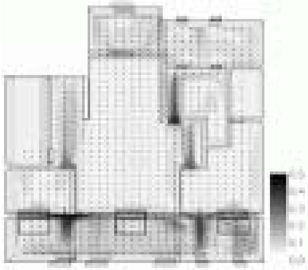
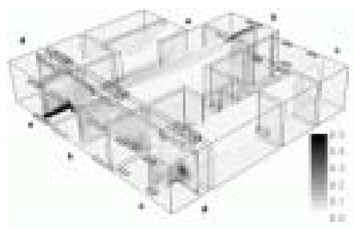
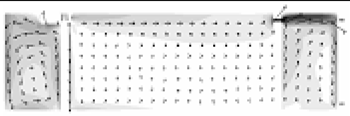
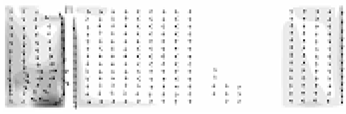
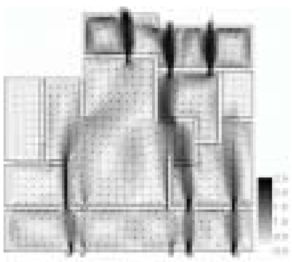
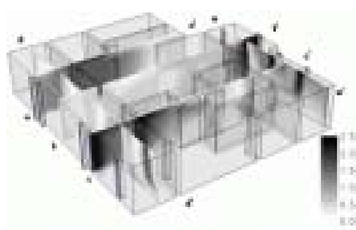
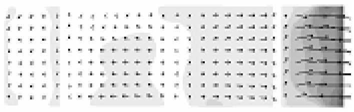
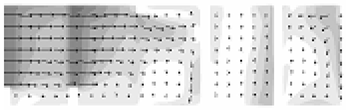
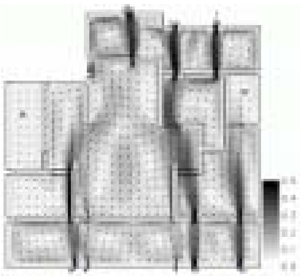
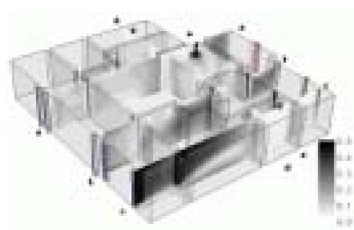
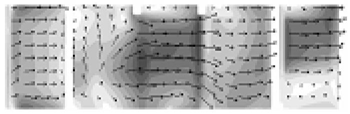
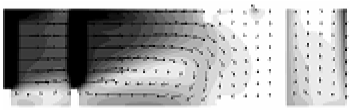
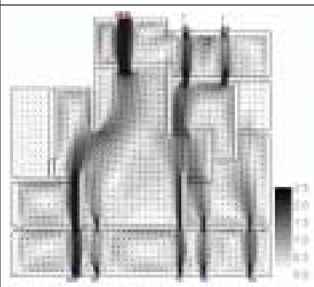
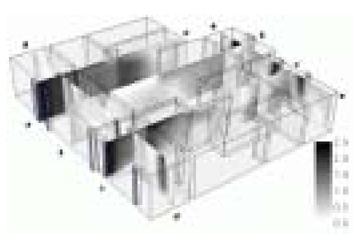
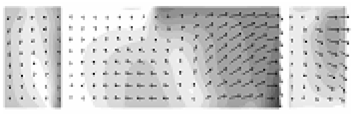
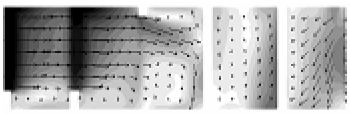
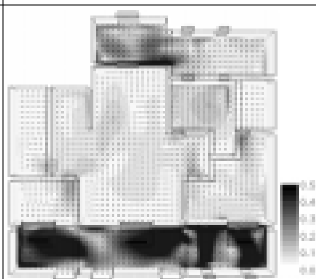
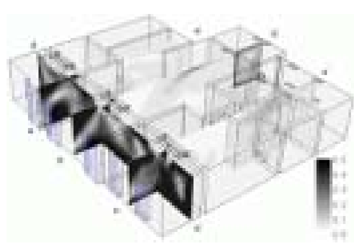
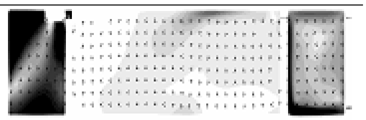
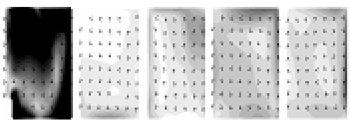
(d) Operation condition and methods of Mode 4



Season	Outdoor air temp.	ACH	Operation
Summer	Max. vent. of balcony Tout ≥ 26°C	Min. 0.5	△ × (Package air-conditioning, F.C.U)
		Medium 0.7	△ × (Package air-conditioning, F.C.U)
		Max. 1.0	△ × (Package air-conditioning, F.C.U)

○ Natural Ventilation, □ Exhaust Air, △ Mechanical Ventilation, × Auxiliary, * K: Kitchen, B; Bath room

Table 2. CFD analysis results by ventilation methods

Ventilation Mode	(1) Surface Airflow Pattern	(2) Spatial Airflow Pattern	(3) Cross-Section Airflow Pattern
(a)Min. ventilation in the winter MODE1			 b-b'  c-c'
(b)Adequate ventilation in the inter-season MODE2			 b-b'  c-c'
(c)Using ventilation fan when lacking natural ventilation in the inter-season MODE2'			 b-b'  c-c'
(d)Max. ventilation in the summer MODE3			 b-b'  c-c'
(e)Min. ventilation in the summer MODE4			 b-b'  c-c'

Hybrid Controlled Trickle Ventilators

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SUMMARY

The performance of "hybrid controlled" trickle ventilators, that is background ventilators whose opening area depends upon both the pressure difference across the vent and the relative humidity of the room air, are investigated by means of computer simulation and compared to that of conventional fixed ventilators for two internal moisture production rates. The results show the hybrid ventilator performs better than the fixed ventilator, resulting in lower ventilation heat loss while maintaining lower relative humidity.

INTRODUCTION

In UK dwellings "controlled" background ventilation is often provide by the installation of trickle ventilators. These units are controllable to the extent that they can be opened or closed by residents. They do not adjust automatically to environmental conditions. The relationship between the opening area of such vents and indoor air quality has recently been investigated [1]. It was noted that the ventilation provided by such ventilators is highly dependent upon weather conditions, leading to the possibility of both under and over ventilation unless appropriately controlled by occupants. Automatic ventilators however have the potential ability to react appropriately to environmental conditions. Two types of automatic ventilator, pressure and relative humidity, RH, controlled, are currently commercially available. Pressure controlled ventilators reduce their opening areas when exposed to higher pressure differences, reopening at lower pressures. Such ventilators could in theory reduce over-ventilation due to gusts of wind, leading to improved comfort and energy efficiency. Relative humidity controlled ventilators open when internal relative humidity is high and partially close when relative humidity is low, potentially leading to improved indoor air quality, energy efficiency and comfort.

The possible benefits of automatic ventilation control of trickle ventilators, in dwellings have been recently investigated [2]. Results confirmed that active vents could be of use in eliminating over ventilation and hence reducing ventilation heat loss whilst at the same time as ensuring adequate air quality. An improved type of automatic ventilator however, would be controlled by *both* RH and pressure, closing in times of high pressure only if internal RH was below a given threshold. This current work then focuses on the potential performance of such a 'hybrid' trickle ventilator i.e. a *combined* RH and pressure device. An extensive series of relevant simulations have been performed and this paper reports on the performance of such a device. The simulations explore the potential benefits of the hybrid ventilator, compared to fixed ventilators, specifically in relation to different internal moisture productions rates. When internal moisture production is low, the hybrid vent should react by reducing the ventilation rate and hence ventilation heat loss, reducing over ventilation. When the moisture production rate is higher, the hybrid ventilator should increase the ventilation rate and hence reduce the occurrence of high internal RH.

Hybrid ventilators in low moisture producing houses should have the potential to reduce ventilation heat loss, but not improve indoor air quality. In dwellings with high moisture production rates hybrid vents will increase ventilation heat loss in order to improve indoor air quality. The relationship between the equivalent opening area of the ventilators, pressure, moisture production rate and internal relative humidity can be represented in a simplified manner mathematically, in the following equations. The internal vapour pressure excess, V_x (Pa), can be expressed in the following form:

$$V_x = G.R_v.(T_i+T_e)/(2n.Vol) \qquad (1)$$

where: G is the rate of moisture production, (kg/hr),$R_{v,}$ water vapour gas constant 462 Pam3/kgK,$T_{i,}$ internal air temperature, (K), $T_{e,}$ external air temperature, (K), n, air change rate, (ach^{-1}), Vol, Volume of dwelling, (m^3).

Note, this equation assumes zero hygroscopicity of the building and furnishings. The air flow through the ventilator is related to the open area of the ventilator A_i:

$$A_i = 1272.5(p)^{-0.5}.q_v \qquad (2)$$

Assuming n, the whole house ventilation rate, is the sum of the ventilation through ventilator n_v, and background ventilation, n_b, through cracks, and if we make the simplifying assumption that air enters through half of the ventilators and exits through the other half, and that the flow through all ventilators is equal in magnitude, then:

$$q_v = n_v Vol/N.1.8 \text{ (l/s)} \qquad (3)$$

$$n = n_v + n_b \qquad (4)$$

If

$$Z = G.R_v.(T_i+T_e)/2.Vol.((SVP_i.RH_i)-V_e) \qquad (5)$$

then

$$Z - n_b = (1.8N.A(p)^{0.5})/1272.5.Vol \qquad (6)$$

given

$$n_b = A_b(p)^{0.5}/1272.5.Vol \qquad (7)$$

then

$$1272.5.Vol\ Z = [1.8NA + A_b].(p)^{0.5} \qquad (8)$$

Given the internal and external temperatures, the external relative humidity, the volume, external façade area and air permeability of the dwelling, and the number of ventilators, then the equivalent opening area of each ventilator needed to maintain a given indoor RH as a function of pressure and moisture production rate can be calculated and graphed, figure 1. In this example the following has been assumed: T_i =293 K, T_e = 278 K, RH_e= 0.80, house volume of 200m^3, an external façade area of 170m^3, with a measured air permeability of 5m^3/hr/m^2, 10 ventilators. When pressure across the ventilator is high and moisture production is low, only a small equivalent opening area is necessary to maintain acceptable room RH. With moisture production rate of 8kg per day and a pressure of 6Pa, an opening area of 1000mm^2 is sufficient to result in RH of less than 65%. At a higher moisture production rate of 12 kg/day and lower pressure differences, of only 1Pa an equivalent opening area of 4000mm^2 is required to maintain RH below 70%.

METHOD

In order to assess the theoretical benefits of automatically controlled trickle ventilators, annual simulations of a test house using EnergyPlus were carried out. Energy Plus [3], was chosen as the computer simulation model as it allows the thermal, energy, ventilation and moisture performance of the house to be modelled simultaneously in one program. In order to simulate automatic control of the ventilators, a visual basic routine was used to set the ventilator opening schedules, based on room RH and pressure across the ventilator, and a series of iterative simulations run. The ventilators are modelled in a standard dwelling previously detailed by the authors [1] used to test the performance of standard trickle ventilators. A schematic layout of the house showing positioning of the trickle ventilators is given in Figure 2. The air infiltration of 5m^3/hr/m^2 at 50Pa is distributed evenly among the three external walls. The external climate used for the modelling was the Kew, (London, UK), TRY.

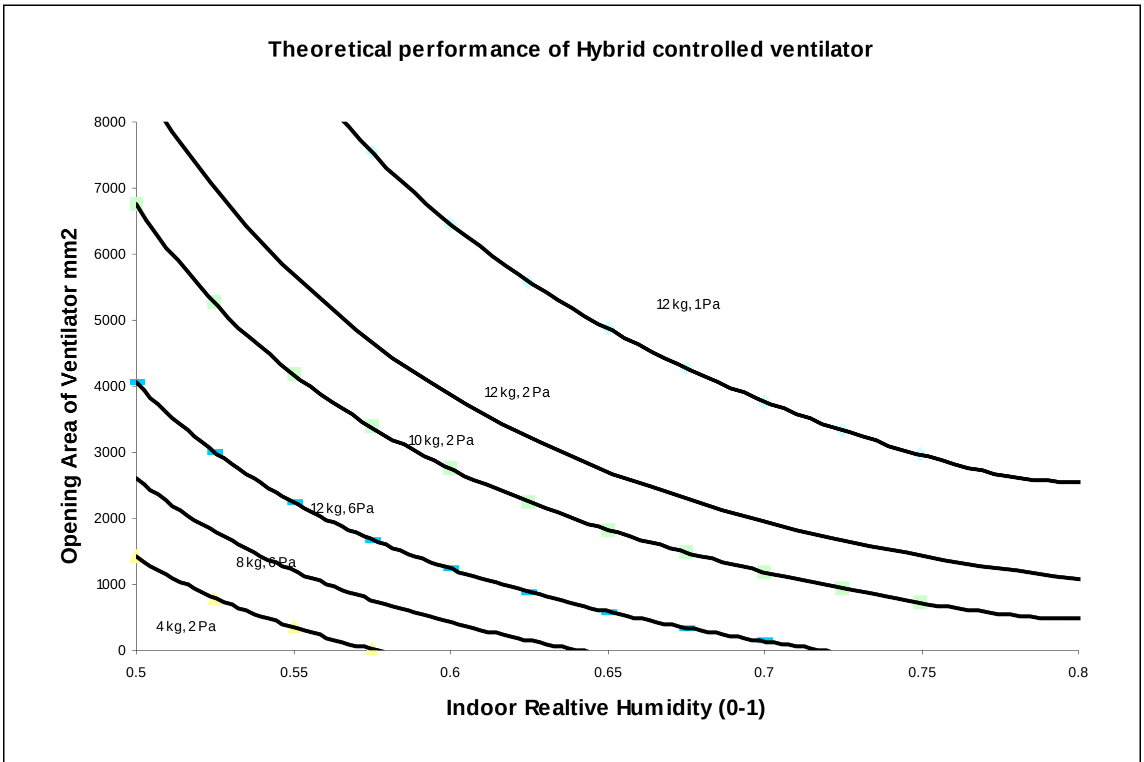

Figure 1. Theoretical performance of a hybrid controlled ventilator

Figure 2. Schematic layout of dwelling. House Volume 200m^3

Two ventilators were modelled were:

1) A standard fixed vent with a fixed open area of 4000mm^2

2) A hybrid (pressure and RH) controlled ventilator with an opening area of 1000 mm^2 if room RH<60% **OR** pressure>6Pa, and 6000mm^2 for room RH> 60% **OR** pressure <6Pa

Two moisture production rates are modelled 6kg/day and 12 kg/day.

RESULTS

The results of annual simulations comparing the performance of the 2 ventilators are summarised in Table 1, for both the low and high moisture production rates. The average heating season ventilation rate, ventilation heat loss and average RH are presented. The heating season was assumed to be October to April inclusive.

Table1. Performance of ventilator during the heating seasons

Ventilator (moisture production)	**Mean background ventilation rate (ach^{-1})**	**Mean Vent Heat Loss (KWh)**	**Mean RH (%)**	**Hours RH >70 (hrs)**
Fixed (lower)	0.51	1760	49.9	20
Fixed (higher)	0.51	1760	58	496
Hybrid(lower)	0.32	1093	50.6	2
Hybrid(higher)	0.43	1425	58.9	394

DISCUSSION AND CONCLUSIONS

The results show the hybrid ventilator performs better than the fixed ventilator, which cannot react to the moisture load. At the lower moisture load the hybrid ventilator reduces ventilation heat loss by 38% compared to the fixed ventilator without incurring an increased number of hours when average internal RH is above 70%. At the higher internal moisture production the hybrid ventilator still reduces the ventilation heat loss by 19% at the same time as reduces the number of hours RH>70% by 21% compared to the fixed ventilator. It is noted that the method of simulating the performance of hybrid trickle ventilator is a simplification, considering only two discrete states, (opening area), and does not model the ventilator continually adjusting to changing conditions. The absolute values of the improvements in ventilation heat loss and relative humidity are of course subject to many uncertainties and assumptions. Further work is underway to develop the model and these results should only be treated as a proof of concept, indicating that hybrid controlled ventilators appear to be of benefit when compared to fixed ventilators. Moisture is considered as the only pollutant, the presence of other indoor contaminants may require greater ventilation rates than those provided by a ventilator designed purely to control on internal relative humidity and pressure differences.

ACKNOWLEDGEMENT

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A07
Advanced components for ventilation and AC

Improved Motor Technology for Small Fans – Impact of Efficiency Gains on System Design

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SUMMARY

Small energy efficient fans have become commercially available for use in HVAC-systems; this open up new possibilities for alternative design of air distribution systems. In this paper the effect of integrating small fans in duct systems is investigated. Modelling and measurements shows that this design has the potential to save energy. When integrating fans in a duct system it is important to consider the system as a whole to find the most energy efficient system solution.

INTRODUCTION

With increased demands for energy efficient and flexible ventilation variable air volume (VAV) systems is an attractive solution. The conventional approach to VAV-systems in Sweden is to use a central fan that pressurise the whole distribution system and use dampers to control and divide the airflows. By replacing control dampers with small, variable speed fans each zone or room of the building can have an individually controlled airflow without any unnecessary fan work. Even for constant air volume (CAV) systems, small integrated fans can have advantages such as reduced fan energy usage and better flexibility.

To reduce the energy consumption, the efficiencies of these fans have to be good. Traditionally small fans have had poor efficiencies compared to larger fans. Now, however, permanent magnet motors (PM-motors) are commercially available for small fans and the efficiencies are much better. Driving the fans with these motors have many advantages such as higher torque or output power per volume, better dynamic performance, low maintenance [1] and built in variable speed drive. Thus the benefit of improved efficiency over a wide working range and reliability is maintained at negligible cost [2].

In conventional distribution systems a centralised fan pressurises the whole duct system and dampers control the airflows. The dampers control the airflow by introducing a flow resistance into the duct system. In VAV systems all branches are equipped with control dampers to adjust the airflows. The requisite pressure for a distribution system is what is needed for the function of the distribution system. Damper authority, diffuser function, stability and flexibility decide how large this requisite pressure is.

There are two main advantages of this new system design compared with conventional distribution systems regarding required fan power: lowered requisite static pressure in the distribution system and less fan work needed to transport the airflow. The latter advantage is present in both CAV and VAV systems whereas the first will only occur for VAV systems.

In conventional systems the fan must provide enough static pressure to overcome the pressure drop in the duct branch with the highest flow resistance. This means that in all other branches there will be a surplus pressure that does not provide any useful work. With the new system design this unnecessary fan work can be avoided since each fan just provides the work needed for that particular branch. Both CAV and VAV have this saving potential. In VAV systems the dampers that control and regulate the airflows requires a certain available static pressure to function properly. The new system design does not include dampers for airflow control so the requisite pressure will be lower than for conventional system and will therefore be a part of the savings potential.

METHODS

This paper is based on a laboratory study of a distribution system and theoretical modelling. The laboratory study includes both measurements and calculations. In the theoretical model a hypothetical distribution system was used for simulations.

Laboratory Study

The measurements were performed on a small-scale distribution system normally used for demonstration purposes, see figure 1. Measurements were performed for two system design cases; 1) on a conventional system set-up, with a VAV-box in each branch, and 2) with the VAV-boxes replaced by variable speed fans. The airflow in each branch and in the main duct was calculated from measurement of the pressure drop across a measuring flange. The static pressure was measured with nipples placed at best possible positions to minimise disturbances of the flow inside the duct.

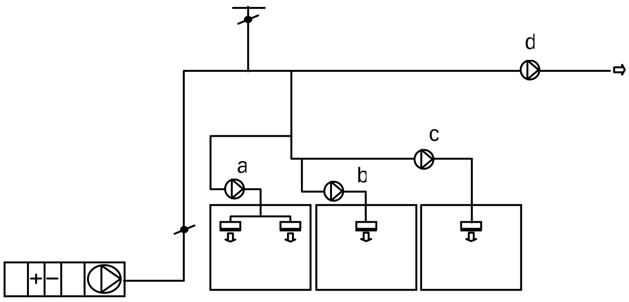

Figure 1. The distribution system for the laboratory study.

The branches are divided into three parts: duct, VAV-box and diffuser. Flow resistance coefficients, k_i, are used to describe these three parts. The relation between pressure drop and flow in each branch is given by:

$$\Delta p = (k_{duct} + k_{VAV-box} + k_{diffuser}) \cdot \dot{V}^2 , \quad (1)$$

where Δp is the pressure drop and $\dot{V}$ is the volume airflow. In case 2, where fans replace the VAV-boxes, the $k_{VAV\text{-}box}$ is set to zero. Power demand for case 2 is calculated according to equation (2).

$$\dot{W} = \frac{\Delta p_{CentralFan} \cdot \dot{V}_{CentralFan}}{\eta_{CentralFan}} + \sum_i \frac{1}{\eta_i} \Delta p_i \dot{V}_i , \quad (2)$$

where $\dot{W}$ is fan power and η is the fan efficiency. For simplification of the modelling and to make it possible to compare results at different operation the efficiencies are assumed to be constant. Due to the non-optimised distribution system design, both the central fan and integrated fans worked at low efficiencies during measurement. The efficiencies used in the calculations and the theoretical model are therefore set to 75% for the central fan and 40% for the small fans, when not stated otherwise.

Measurement Uncertainties – Two types of manometers were used, Furness Control Limited, type FCO332 and Halstrup, type EMA 84. The instrument uncertainties are at worst 0.20% and 1% respectively. Since the objectives are to evaluate potential energy savings and practical limitations the measurements give a satisfactory accuracy.

Theoretical Modelling

A hypothetical model of a medium sized distribution system with seven branches was used to investigate limitations and potentials of the new system design. The pressure gain was divided between the air-handling unit and the distribution system and the total maximum airflow was 1,4 m^3/s. The required fan power was calculated with equation (2).

The flow resistance of the air-handling unit is given as a pressure drop and the pressure drop of the distribution system is a function of the length of the duct branch. There is a requisite pressure necessary for the function of the system, this is divided into two parts: requisite pressure for the diffuser and requisite pressure for the VAV-box. For the new system design the requisite pressure only consists of the diffuser part since there are no VAV-boxes. The requisite pressure is set as constant.

For airflows lower than 1,4 m^3/s the pressure drop for the distribution system and air-handling unit varies with airflow according to equation (3), where index max indicates the highest airflow and pressure drop for the system in the model.

$$\Delta p = k \cdot \dot{V}^2 = \frac{\Delta p_{Max}}{\dot{V}^2_{Max}} \cdot \dot{V}^2 , \qquad (3)$$

RESULTS

Laboratory Study

Measurements were performed for several operating conditions for both the conventional and the new system design. For an example of results for the conventional system see table 1. In this case the static pressure after the air-handling unit was 190 Pa and the total volume airflow roughly 400 l/s. Using data from the measurements the potential reduction of required fan power for the distribution system was calculated and presented in figure 1.

Table 1. Airflows and static pressures, conventional system.

Branch	Static pressure [Pa]		Airflow [l/s]
	Before VAV-box	After VAV-box	
A	116	15	58
B	67	46	61
C	101	53	38
D	-	-	248

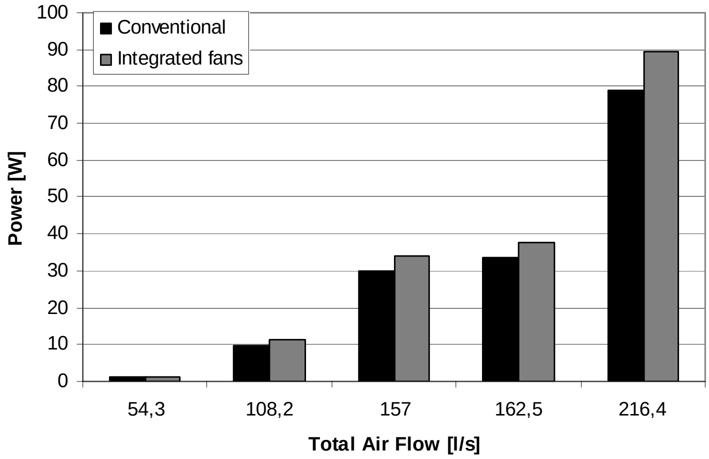

Figure 2. Total fan power demand for the studied distribution system at different air flows, both with conventional design and with integrated fans.

Figure 2 shows that for this distribution system a change from conventional to new system design will lead to increased fan power demand. There are several reasons for this, for example non-optimised duct system design and that the reduced flow resistance by removing the VAV-box is not enough to compensate for fan efficiency decrease when using integrated fans instead of a centralised fan.

Theoretical Modelling

The results from the simulations are presented as the new system designs performance relative to a corresponding conventional system, so the value 1 means equal to the conventional system. Figure 3 shows the effect on the total required fan power of the system as a function of the pressurisation provided by the central fan.

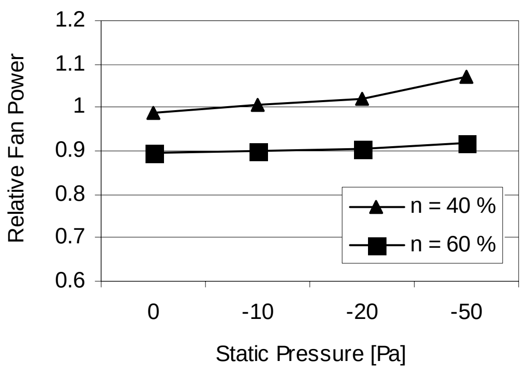

Figure 3. Relative fan power as a function of static pressure before the first fork for two efficiencies for the integrated fans.

As can be seen in figure 3 the system with integrated fans goes from being more energy efficient to less with small decreases in static pressure at the point where all the duct branches join. During measurements the same effect could be observed. Figure 3 also shows that with higher efficiency of the integrated fans this trend is less pronounced but will still be present as long as the efficiency of the integrated fan is lower than that of the central fan. This limits the extent to which the pressure gain from the central fan can be reduced from an energy efficiency point of view.

Pressurisation from the central fan is an important aspect and is closely linked with the duct system design. To show the effect of duct system design several simulations were performed and some results are presented in figure 4. For all cases the total airflow and the necessary pressure difference for transporting the airflow through the longest duct branch is the same. There are three cases presented here: 1) with symmetrical duct system where all the branches are of equal length and have equal airflow, 2) with an asymmetrical duct system (different duct length in the branches) where one branch is dominant and 80 % of the total airflow passes, and 3) with the same asymmetrical duct system but where all branches transport equally large airflows.

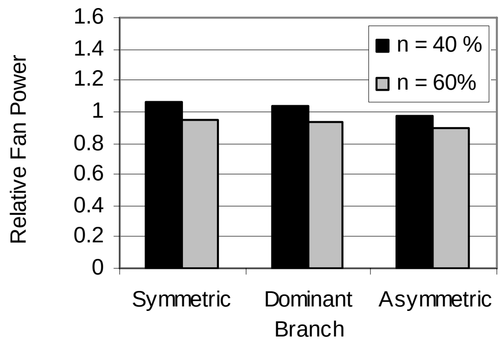

Figure 4. Relative fan power for different duct system designs and efficiencies of the integrated fans.

As can be seen in figure 4 the saving potential with the new system design for CAV systems and today's fan efficiencies is limited and is highly dependant on duct system design and distribution of airflows. But for increased fan efficiency the new system design will require less fan power regardless of duct system design and for the asymmetrical case the potential energy savings will be as large as 10 %.

Simulations showing the improvement of system performance as a function of increased integrated fan efficiency were made for both VAV and CAV, see figure 5. The airflow for the VAV system was equal to half the CAV airflow. For VAV systems the new system design can require less fan power even with a perfectly symmetrical duct system but, as for CAV, some degree of asymmetry leads to an increased savings potential.

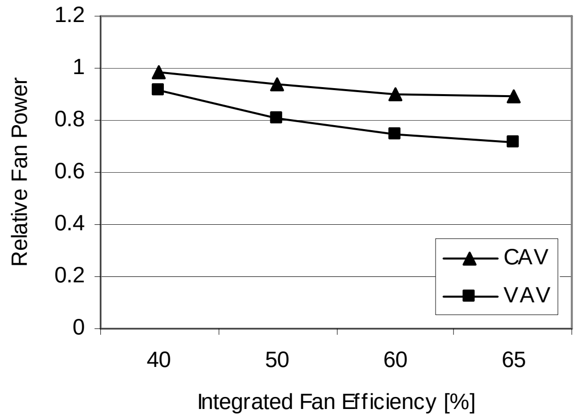

Figure 5. Relative fan power as a function of integrated fan efficiency for different system designs.

To show the potential advantages of the new system design the results for a suitable duct system was simulated. These results show that the potential savings with the new system design is about 10 % with an integrated fan efficiency of 40 %. As figure 5 shows the new system design can have an even bigger saving potential for VAV than for CAV. With increased fan efficiency the energy savings for this system can be up to 25-30 %.

DISCUSSION

Requirements

To make this system design feasible some requirements must be fulfilled. Here only those requirements that separate the new system design from conventional systems are discussed. First and foremost there must be available fans with suitable working range, size and performance. The fans must be able to provide a high enough pressure gain to overcome the flow resistance of the distribution system even at low airflows. The measurements performed showed that the fans used in this study fulfilled these requirements. However, the efficiency of fans and motors of this size is still not very high, around 35 to 40 %.

Savings potential

As mentioned above the savings potential is due to the unnecessary fan work that results from differences in flow resistance between the duct branches. Therefore the design of the duct system and distribution of airflows will determine the potential savings. For a perfectly symmetrical duct system the savings potential will be limited, in fact such a system can be less efficient than a conventional system. There should not be an airflow distribution where one duct branch transports the main part of the total airflow. Such a system can be even less suitable for the new system design than perfect symmetry. Worth mentioning here is that since energy for fans is a big part of total energy usage for buildings even small savings or increases in system efficiency can be profitable. In VAV systems the saving potential also depends on how the airflows vary in different parts of the duct system. The new system design is especially suited for systems where there is a varied demand for airflows.

Limitations

As shown above there is definitely a potential to save energy by replacing conventional systems with this new system design, however there are of course limitations to the applicability of the new system design and its savings. The measurements did not show any saving potential. Figure 2 shows that it is less effective with integrated fans for this system. This is due to limiting factors that will be discussed below.

Control of the Central Fan - Since the central fan has a higher efficiency than the integrated fans it should provide as large part of the work as possible. Ideally that means that the central fan should provide the work necessary to transport the airflow through the air-handling unit and pressurise the distribution system so that no additional fan work is necessary for the branch with the *least* flow resistance. For all other branches the integrated fans will add the extra pressure gain necessary to overcome the pressure drop of that branch. In a CAV system it is straightforward to pick out the branch with least resistance but in a VAV system this can vary. A more simple approach would therefore be to control the work of the central fan so that it keeps a constant static pressure at the first fork in the duct system. Since the integrated fans work parallel to each other a reduction of this static pressure will result in an increase in the necessary pressure gain for *all* the integrated fans and thereby an increased energy usage.

Duct system design - In the section about Requirements and Savings Potential the effect of unsuitable duct system design is shown and as mentioned above this design will also affect the control of the central fan. For a duct system to be suitable for this new system approach it should have the following attributes:

- System size (several duct branches)
- Airflow in each duct branch of the same magnitude (no single dominant branch)
- Preferably varied length of the duct branches
- No small ducts branching off, a collected system

The measurements for this study were performed on a very small duct system with just three branches and small airflows. For this system no energy savings were measured, even if this system was modelled and simulated it was not big enough to give any energy savings, see figure 1. This was due to several factors. Branch *b* has the highest flow resistance and in addition the flow through this branch is high. The pressure gain by removing the VAV-box is not enough to compensate for the decreased efficiency when driving this branch with the smaller fan. As a result of this the power demand increases. However, in branch *a*, the power demand is almost halved whereas in branch *c* a slight increase is seen.

The saving potential with integrated fans is dependent on the surplus pressure and following fan work for all branches except the one with the highest flow resistance, as described above. But by reducing the pressure gain from the central fan the necessary work is instead performed by the smaller integrated fans that have lower efficiency than the central fan. For this new system design to be profitable the savings due to surplus pressure and fan work must outweigh the loss in fan efficiency and this requires a system with several duct branches

The varied duct branch length and similarly sized airflows are necessary to provide the desired difference in surplus pressure and fan work. In conventional systems it is common with small ducts, supplying one single diffuser branching off from the main duct. Since the central fan pressurises the whole duct system this works fine. The same approach will lead to an impossible amount of integrated fans with the new system design.

Practical Aspects

Apart from the above discussion there are also several practical aspects that will affect this new system design. In this section some of these will be discussed.

Placement of Fans - By placing the fans in the duct system, the inhabitants of the building will be affected by maintenance since this will take place right among them. Another effect of fan placement is noise.

Economy - This system approach requires a number of fans, each with sensors for airflow control. This study has not included any economical analyses but the expected high cost of this system must be weighed against the energy savings and other advantages such as flexibility.

Pressurisation of the Duct System - In conventional duct systems there is a positive static pressure compared to the pressure inside the building. That means that any leakage will be directed out from the ducts. With the new system design parts of the duct system will have a negative pressure and leakage airflow could enter the duct system and thereby spread to other parts of the building.

Reversed Airflow - Negative pressure in parts of the duct system require careful design and a control system to prevent reversed airflows. During the measurements in this study one integrated fan was turned off while the two others were turned on to give maximum airflow. This resulted in a reversed airflow through the branch with the turned off fan. A simple solution to prevent this would be to place a static pressure sensor downstream of the fan. The fan would then be programmed to always maintain a certain positive static pressure even if no airflow is necessary.

Conclusions

There is definitely a potential to save energy by changing the design of ventilation systems by replacing the conventional approach with one centralised fan with many decentralised fans integrated in the duct system. The saving potential is not universal for all duct system but depends on several factors such as symmetry, pressure drop and flow for each branch etc. It requires careful design and there are many practical aspects that need to be considered. This approach will also put demands on engineers to think in different ways in planning their duct systems, choosing fans, placement of sensors and so forth.

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SPECIFIC FAN POWER – a tool for better performance of air handling systems

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SUMMARY

The electrical energy needed for ventilation fans and air handling units (AHU) plays an increasing role in the energy demand for buildings. Recent studies show that the electrical energy consumption can rather easily be reduced from the "traditional" level (between 5 and 10 kW/m^{3}s) into a modern level (2 to 2,5 kW/m^{3}s) with proper design and installation. Even lower levels are technically possible, but not yet widely economically feasible.

Several countries have already set, either as requirements or as recommendations, maximum target values for SFP. This has been the first important step towards energy efficient air handling systems. However, in real practice other measures are needed, including simple but reliable tools for designers for SFP calculation, and measuring guidance for installers. To check and assess the SFP should be easy and clear also for the building owner, the end user, the inspectors etc. The need for measurements should be taken into account in the AHU construction to enable easy but reliable measuring, and guidance is needed for system design as well as for those who perform the measurements.

INTRODUCTION

The electrical energy needed for ventilation fans and air handling units (AHU) plays an increasing role in the energy demand for buildings. Recent studies show that the electrical energy consumption can rather easily be reduced from the traditional level with proper design and installation.

The "Specific Fan Power" (SFP) value, expressed in kW/m^{3}s, indicates the demand on power efficiency of all supply air and extract air fans in a building. Several countries have already set, either as requirements or as recommendations, maximum target values for SFP. This has been the first important step towards energy efficient air handling systems. However, in real practice other measures are needed, including simple but reliable tools for designers for SFP calculation, and measuring guidance for installers. To check and assess the SFP should be easy and clear also for the building owner, the end user, the inspectors etc.

The present regulatory values vary typically between 2 and 3 kW/m^{3}s. The newly revised European Standard EN 13779 [1] gives a classification of SFP values, which also takes into account the fact that in some special cases (requiring e.g. HEPA filters, humidification or high-efficiency heat recovery) a higher electrical energy consumption is unavoidable. In the new EN 13779, the whole range is from 1 up to 9 kW/m^{3}s, and the recommended maximum excesses of the classified SFP values (e.g. due to high-efficiency filtration) are also defined.

WHAT IS SPECIFIC FAN POWER?

The new EN 13779 [1] includes an Annex which gives more detailed guidance about how to express, specify and also validate the SFP value in practice.

Target value for the **Specific Fan Power, SFP**, indicates the demand on power efficiency of all supply air and extract air fans in a building. This value should be defined during the early design stage for determining the useful power demand and so the energy consumption required for transporting air throughout an entire building.

A supplementary factor SFP_E makes it possible to assess how efficiently **individual** air handling units or fans utilize electric power. The definition of the SFP_E is different for heat recovery air handling units with supply air and extract air, and for separate supply air or extract air handling units and individual fans.

During the design process the SFP value for the entire building, defined as the weighted average of the SFP_E values of individual units and fans (see item "Example"), shall be compared to the target value and checked in case any changes in the individual SFP_E values appear.

Another useful specific fan power is SFP_V for validation. The intention with this value is to have a factor which is simple to specify and check. The difference between SFP_E and SFP_V is the load condition, **design for SFP_E and validation for SFP_V**. It is recommended that both SFP_E and SFP_V values are calculated (using manufacturer's software, for example).

The SFP value for the whole building is defined as follows: "The combined amount of electric power consumed by all the fans in the air distribution system divided by the total airflow rate through the building under design load conditions:

$$SFP = \frac{P_{sf} + P_{ef}}{q_{max}} \qquad (1)$$

where

SFP is specific fan power demand in kW x m^{-3} x s

P_{sf} is the total fan power of the supply air fans at the design air flow rate in kW

P_{ef} is the total fan power of the extract air fans at the design air flow rate in kW

q_{max} is the design airflow rate through the building, generally the extract air flow in m^3 x s^{-1}

In terms of SFP **for the whole building**, any fan powered terminals shall be included when they are connected to the main air supply system.

For individual air handling units or fans, to enable the designers of building projects to quickly determine whether a given air handling unit will positively or negatively meet the overall demands on power efficiency, a SFP_E for the individual fan or AHU has been defined. In a constant air volume flow system, the demands shall be met at the design air flow and design external pressure drop (pressure drop in the ducting). In a variable air volume flow system, the demands made on the SFP_E shall be met at the partial air flow and the related external pressure drop, specified for each air handling unit specification or at another point in the reference documents of the project. Therefore data at design maximum air flow and design maximum external pressure drop shall be specified, as well as the partial flow and the related external pressure drop. If the data concerning partial air flow and related external pressure drop is not specified, the following figures can be used as default values for determining the SFP_E (background and more details are presented in EUROVENT 6/8 [2]):

Partial air flow (default value): 65 % of the design maximum air flow
Partial external pressure drop (default value): 65 % of the design maximum external pressure drop.

.The **specific fan power** ***SFP*****E – ** for **supply and extract air units** (normally also equipped with heat recovery), is the total amount of electric power, in kW, supplied to the fans in the AHU, divided by the largest of supply air or extract air flow rates (i.e. not the outdoor air or the exhaust air flow rates) expressed in m^3/s under design load conditions.

$$SFP_E = \frac{P_{sfm} + P_{efm}}{q_{\max}} \qquad (2)$$

where
SFP_E is the specific fan power of a heat recovery air handling unit in kW x m^{-3} x s
P_{sfm} is the power supplied to the supply air fan in kW
P_{efm} is the power supplied to the extract air fan in kW
q_{max} is the largest supply air or extract air flow through the air handling unit in m^3 x s^{-1}

Air handling units with liquid-coupled coil heat exchangers and separate supply air and extract air sections also belong to this category of air handling units.

For separate supply air or extract air handling units and individual fans, the **specific fan power,** ***SFP*****E** is the electric power, in kW, supplied to a fan divided by the air flow expressed in m^3/s under design load conditions.

$$SFP_E = \frac{P_{mains}}{q} \qquad (3)$$

where
SFP_E is the specific fan power of the air handling unit/fan in kW x m^{-3} x s
P_{mains} is power supplied to the fans in the air handling unit/fan in kW
q is air flow through the air handling unit/fan in m^3 x s^{-1}

The intention with the SFP_V value is to have a factor which is simple to specify during building design and straightforward to validate when commissioning and controlling the ventilation system. The ***SFP*****v** is the electric power, in kW, supplied to a fan divided by the air flow expressed in m^3/s **under validation load conditions.** When defining a ventilation system specification it is convenient to specify the highest permissible SFP_V as this will help to influence the choice of air handling units or fans towards those of a desired power efficiency.

EXAMPLE

In a typical commercial or public building there are a few air handling units and also some separate supply or extract air fans to serve different purposes. There may also be a wide variety of activities in the building, some of which can be served by a simple fan (e.g. exhaust from toilets) and others requiring high-level air treatment and distribution (auditoria, exhibition facilities etc.). For this reason, the SFP values for individual units and fans can vary within a wide range especially in multi-purpose buildings. The following example presents also how to calculate the SFP for the whole building as a weighted average of all individual SFP's, as the total power consumption of all fans altogether, divided by the total supply **or** extract air flow, whichever the greater. EN 13779 [1] actually defines several ways to determine the SFP values, due to the still remaining national differences in the definitions. It is therefore important also to include in the design and commissioning documentation, which calculation procedure has been applied in each individual case.

AHU equipped with both supply and extract air units

Supply air fan	**Air flow m^3/s**	**Ductwork pressure Pa**	**Power supplied to the fan 1) kW**	**Extract air fan**	**Air flow m^3/s**	**Ductwork pressure Pa**	**Power supplied to the fan 1) kW**	**SFP_E of this AHU $kW/(m^3/s)$**
S-1	0,5	300	0,98	E-1	0,5	250	0,85	3,66
S-2	2,5	250	3,36	E-2	2,8	250	3,93	2,60
S-3	6,9	300	9,17	E-3	7,2	300	8,71	2,28
S-4	3,3	250	4,33	E-4	3,6	250	4,83	2,54
Total	13,2		17,8		14,1		18,3	

Separate supply air units or fans

Supply air Fan	**Air flow m^3/s**	**Ductwork pressure Pa**	**Power supplied to the fan 1) kW**	**SFP_E of this fan $kW/(m^3/s)$**
S-5	0,4	300	0,66	1,65
S-6	1,2	220	1,44	1,20
Total	1,6		2,1	

Separate extract air units or fans

Extract air fan	**Air flow m^3/s**	**Ductwork pressure 2) Pa**	**Power supplied to the fan 1) kW**	**SFP_E of this fan $kW/(m^3/s)$**
EF-1	0,1	160	0,06	0,60
EF-2	0,2	220	0,17	0,85
EF-3	0,5	350	0,35	0,70
EF-4	1,0	220	0,67	0,67
Total	1,8		1,25	

Total supply air flow	13,2+1,6	14,8 m^3/s
Total extract air flow	14,1+1,8	15,9 m^3/s
Total electrical power	17,8+18,3+2,1+1,25	39,4 kW
SFP =	**39,4/15,9**	**2,48 $kW/(m^3/s)$**

1) Power supplied to the fan
This means the power supplied to the fan at design air flow and given pressure loss of the ductwork. This value can be calculated for example using the manufacturer's dimensioning software. This figure is used as input data for calculation of the SFP for the entire system. This figure includes the efficiency of fan, motor, belt drive and frequency converter. This is also the power, which should be verified by measurements in the completed installation after balancing and final adjustment of air flows.
2) Ductwork pressure, in case of separate extract air fan

APPLICATIONS, AND TASKS FOR DIFFERENT TARGET GROUPS

Guidance for design, measuring and documentation has been developed during the recent years in Finland in order to support practical implementation of the relevant clauses of the National Building Code [3], which gives a maximum target value for SFP (**2,5 kW/m^{3}s for ordinary systems;** for special applications a higher value is allowed) and also requires validation by measurements in commissioning.

An unofficial guideline was published in 2004 and revised in 2005, freely available in www. A simple excel calculation tool for designers was included in the second edition. The feedback from designers and inspectors indicated that the information was not very widely known.

In order to have the SFP calculation and validation widely adopted also in practice, a short summary was added in the information package in 2006. It points out the importance of the issue to the "key players" as follows:

-the designer will make the system design, including: specifying the target value for SFP, selecting the AHU's, calculating the exact SFP value (for each unit and fan, and finally the overall SFP as the weighted average as described in the example above), and dimensioning of the air distribution and diffusion system accordingly. The designer also makes the design documentation including the SFP calculation sheet

-the installer will install the system and take care that the system is built according to the designer's specifications – not only physically, but also functionally. The installer takes also responsibility of the SFP measurements at the commissioning stage, including the relevant measurement report

-the building inspector will check the design documentation and the measurement report and confirms their conformity to the building regulations

It has to be pointed out that the use of SFP as guidance for system design and construction is a relatively new issue everywhere. There is no scientific data yet available about the SFP level and how much it has reduced during the recent years. However, the measurements done by some manufacturers and inspectors indicate that the most common regulatory values, between 2 and 2.5 kW/m^{3}s, represent the state-of-art in modern applications. Lower regulatory values do not yet seem to be very realistic, taking also into account the fact that some 15 years ago a much higher power consumption (between 5 and 10 kW/m^{3}s) was the normal practice. Levels down to 1 – 1,5 kW/m^{3}s are technically possible, but not yet widely economically feasible.

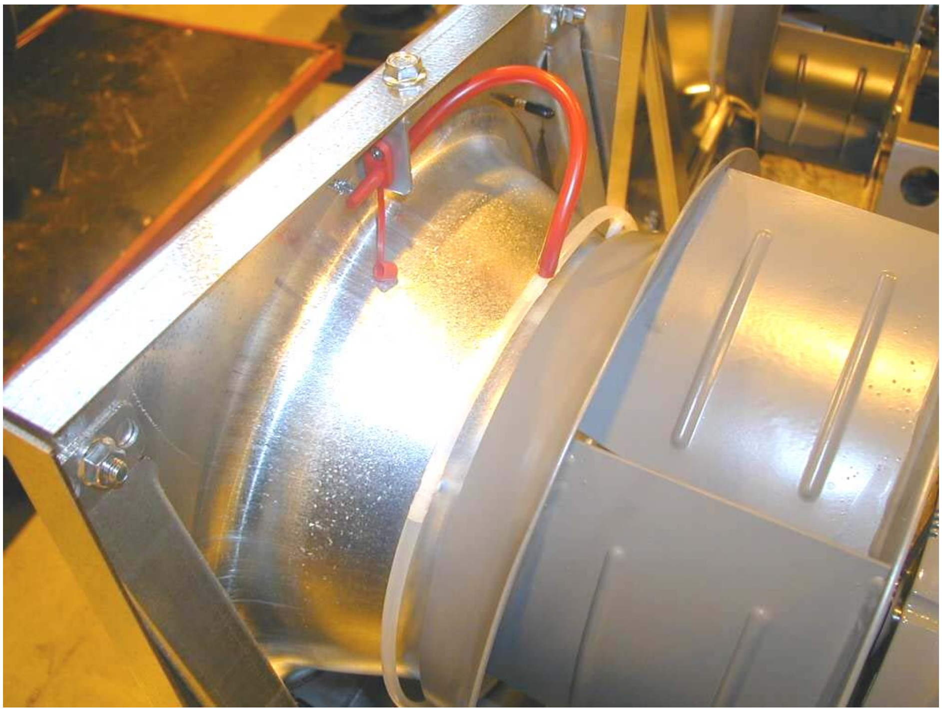
DISCUSSION

Recent studies show that the electrical energy consumption can rather easily be reduced from the "traditional" level (between 5 and 10 kW/m^{3}s) into a modern level (2 to 2,5 kW/m^{3}s) with proper design and installation. Even lower levels are technically possible, but not yet widely economically feasible.

In addition to put some target values for the SFP number, it is also important to give tools for practitioners so that the target values can really be achieved and maintained in real practice.

The SFP value can be easily measured in practice, if the measurability is taken into full consideration in design and construction of the AHU. Normally the power consumption can be easily measured in modern AHU's, see the figure on the left.

Also the air flow has to be measured to determine the SFP value. To do this in an easy but reliable way, the AHU or fan should be equipped with a built-in air flow measurement arrangement. One example is presented in the figure below.



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2. Recommendations for calculations of energy consumption for air handling units. EUROVENT 6/8, Eurovent/Cecomaf, 2005.
3. Indoor Climate and Ventilation of Buildings. National Building Code of Finland, Part D2, 2003. http://www.ymparisto.fi/default.asp?contentid=127529&lan=fi&clan=en

The impact of thermal loads on indoor air flow

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SUMMARY

The potential for draught discomfort and high air velocities in the occupied zone are often studied with only cooling design in mind. During the transition season, however, downward flows with high air velocities may occur in the occupied zone due to cold window surfaces. Airflow generated by supply air terminal dervices may further enhence the velocity in the occupied zone. Furthermore, convection flows caused by thermal loads may significantly affect the air flow conditions in the room as a whole and assist the occurance of high velocity near occupants. Analyses of results from full-scale measurements with chilled beams presented in this paper reveal that installations with possibilities of convection flow oposing the supplied flow should be avoided. Generally speaking, convection flows have less impact on air distribution in rooms with chilled beams installed in lengthwise direction than when installed crosswise in rooms.

INTRODUCTION

The local air velocity, temperatures of the room air and air jets, and fluctuations in air velocity are the key factors that determine the risk of a draught. In designing an indoor environment, it is important to evaluate the impact of the various solutions on the air flows in the room. It is particularly important to analyse air velocities in office buildings and on other business premises with relatively high requirements for cooling capacity. The maximum allowed speed in the occupied zone is specified in the present standards for the cooling and the heating seasons. No recommendations have been set for the transition season in the standards, which, in practice, means that designers are free to use the recommendations for either the winter or the summer season.

Convection flows, for example caused by a cold window, may, however, be sufficiently strong during the transition season and may generate high velocity in the occupied zoned and cause draught discomfort. When chilled beams are used the air velocity in the occupied zone is affected by the window's surface temperature, the direction of the supplied air flow relative to the convection flow from the cold window, and the convection flow from operating radiator (if any) below the window.

The effect of the convection flows on the air distribution in rooms is usually ignored and the assessment focuses only on the flow supplied from air terminal devices. This article addresses the effect of convection flows on the room air distribution. In particular the interaction of convection flows from heat sources with different strength and location and from warm window with the airflow supplied by chilled beams is studied and reported.

EFFECT OF CONVECTION FLOW ON JET DETACHMENT

The effect of a convection flow created by a thermal load on the detachment point of the supply air jet as shown in Figure 1 was studied [1]. The measurements aimed at examining the variation occurring in the detachment point of the jet when the thermal load and the chilled beam's supply air flow rate were altered. The test room (4.0 x 2.8 x 2.8 m (H)) had a room-wide unidirectional beam and a 1.2-metre-wide band-shaped reheat coil (Fig. 1). In the measurements, the output capacities of the convector was set to 200 W (71 W/m), 300 W (107 W/m), 400 W (143 W/m), and 500 W (179 W/m), while the primary flow rate supplied from the chilled beam was 6, 8, and 10 l/s per m.

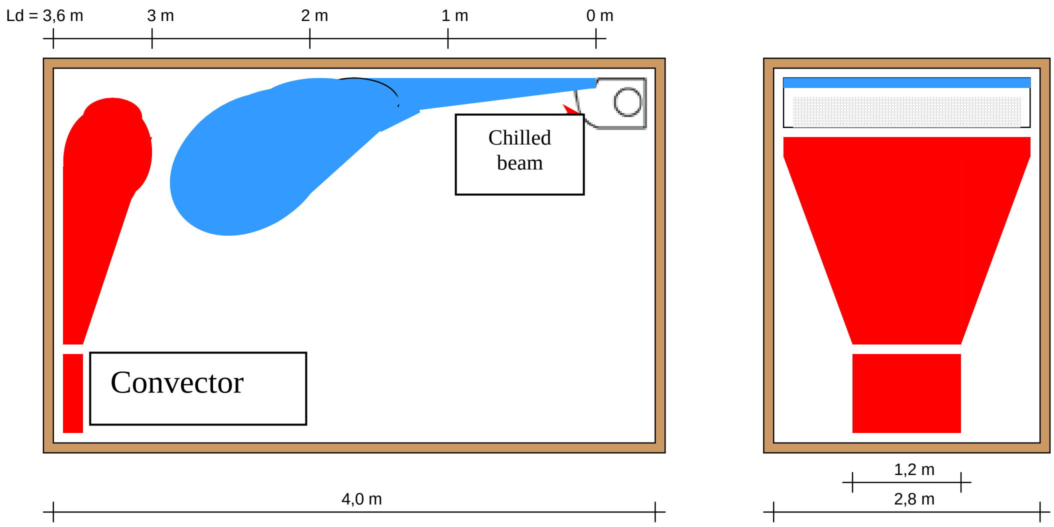

Figure 1. Test arrangement used to study the effect of convection flow on the detachment point of an opposing supply air jet of a room-wide unidirectional chilled beam.

Figure 2 presents the distance of the detachment point of the supplied air jet as a function of the thermal load's capacity is shown. The supply air flow rate from the unidirectional chilled beam is a parameter. The results indicate that the convective flow corresponding to the cooling effect of a 150 W/m room device is sufficiently great to cause the detachment and jet droped in the occupied zone with all air flows studied. With the typical airflow rate of 8 l/s per m (2 l/s per m^2) for a unidirectional beam, the momentum flux caused by a thermal load of 50–75 W/m is sufficient to release the jet before it reaches the opposite wall. In general, instalation of unidirectional chilled beams with supplied jets opposing convection flow from heat sources should be avoided.

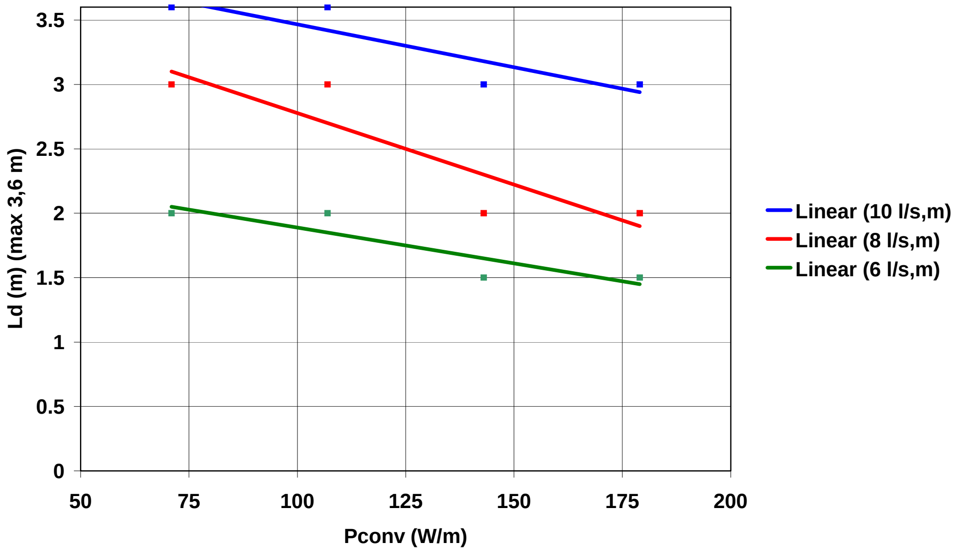

Figure 2. The detachment point of the jet as a function of the convection load of a band-shaped heat source (W/m) and the supplied primary air flow (l/s/m) from a unidirectional chilled beam.

SIGNIFICANCE OF THE POSITIONING OF THE AIR SUPPLY DEVICE

The impact of the positioning of the air supply device on the flow distribution in the room during cooling (summer) and mid-season situations was studied in a test room (2.4 x 4.6 x 2.8 m (H)). A 300 mm wide chilled beam with a total length of 2,100 mm (1,900 mm effective length) was installed in both the longitudinal and latitudinal direction in the room as shown in Fig. 3. The air velocities in different points of the occupied zone were measured and compared. Figure 4 presents the air velocities measured in the cooling season (60 W/m^2), while Figure 5 for the transition season (25 W/m^2). In the case of 60 W/m^2, the heat loads were: a computer (100 W), a dummy (60 W), light fittings (144 W) and warm window (350 W). Heat loads in the transition case as before, except the window was not heated.

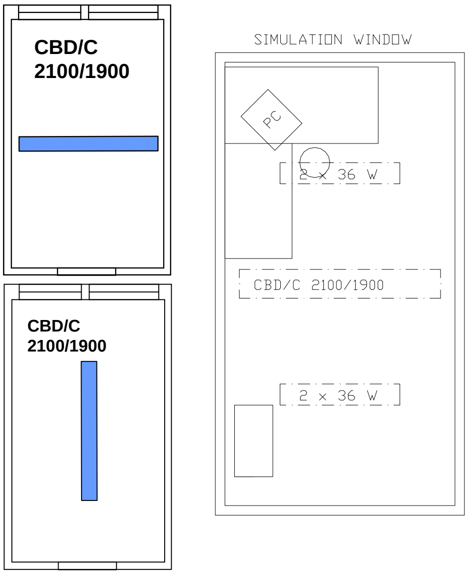

Figure 3. Test room arrangements with a longitudinal and latitudinal active chilled beam.

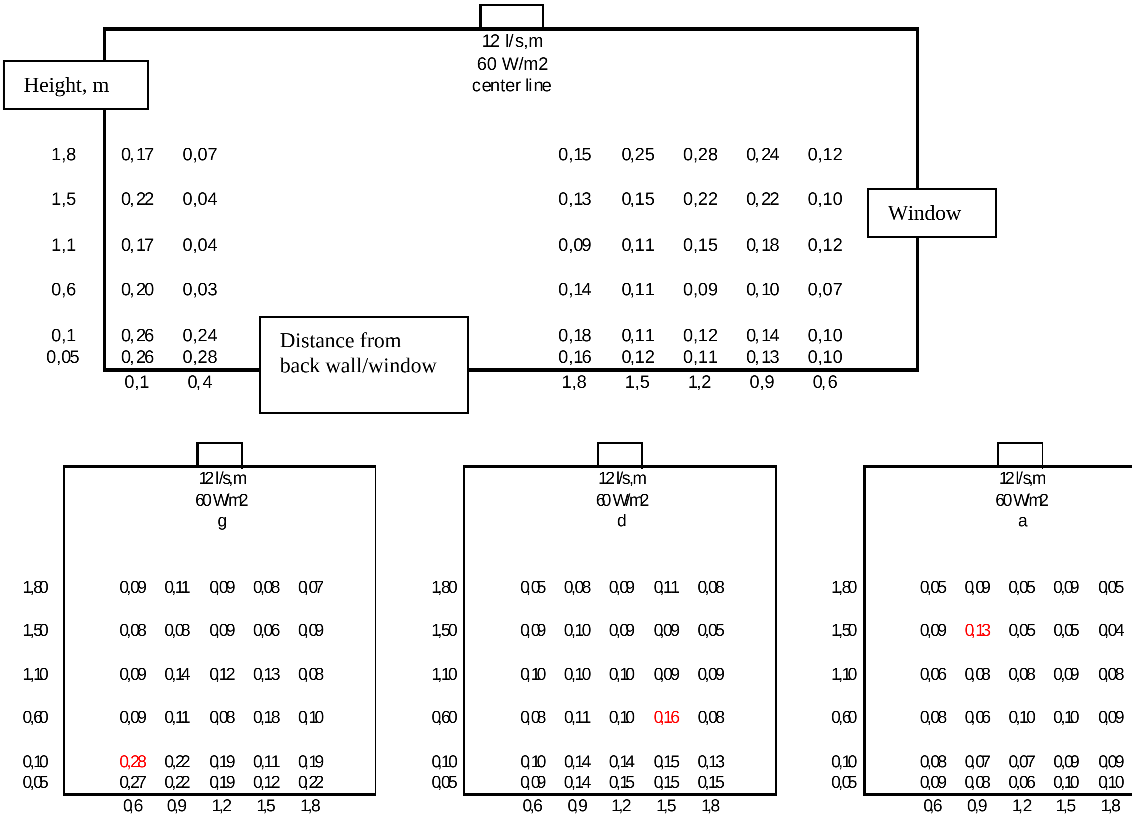

Figure 4. Air velocities (m/s) in the latitudinal (top cross-section) and longitudinal installation (bottom cross-sections: a = 0.6 m, d = 1.5 m, and g = 2.4 m from the window) measured with chilled beam (2100 mm length) at cooling capacity of 60 W/m^2. Cooling case (summer season).

In the cooling situation (Fig. 4), the convection flow from heat sorces concentrated at the window wall (a warm window, human occupant, and computer) reverses the air flow from the latitudinal beam and the maximum velocities occur close to the corridor-side wall. In the longitudinal installation (Fig. 5), the effect of the convection flows is smaller, and the air velocities are lower, than in the longitudinal installation.

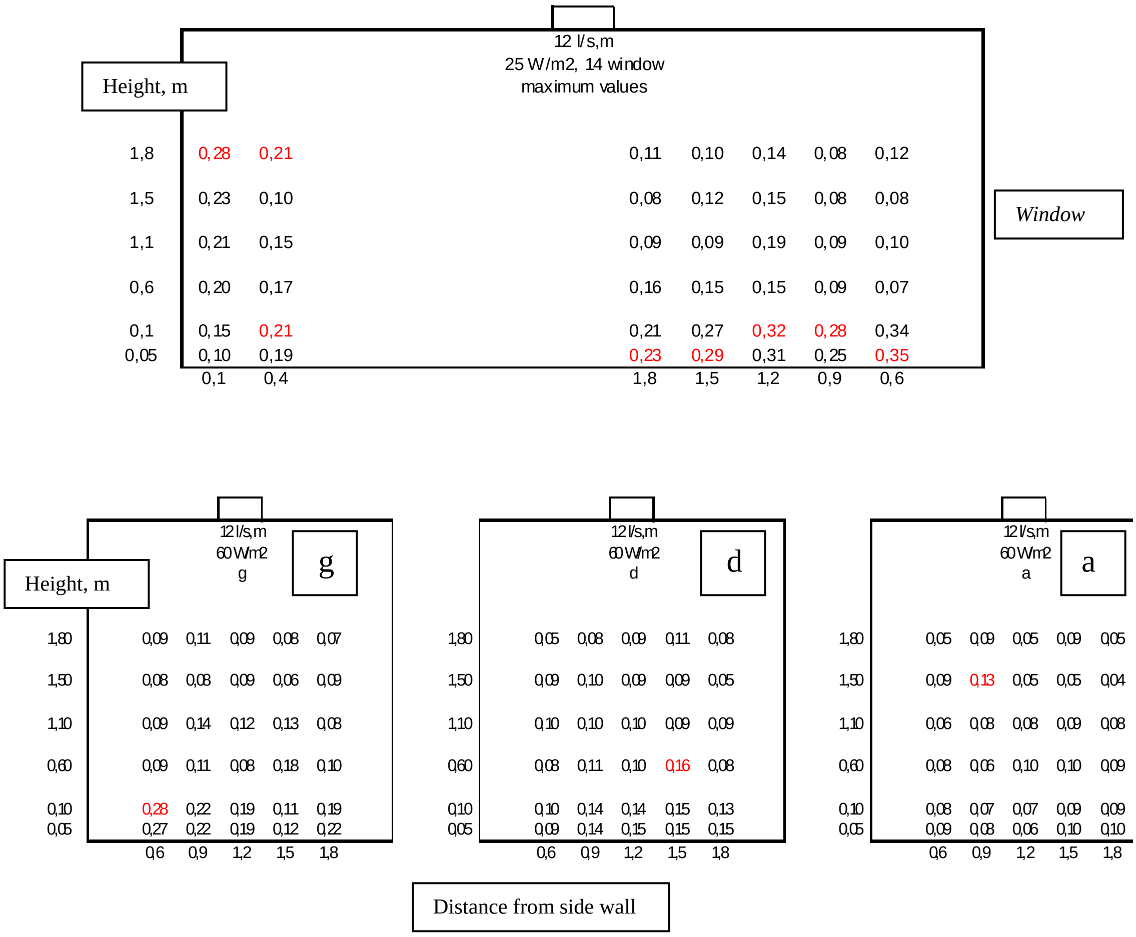

Figure 5. Air velocities (m/s) in the latitudinal (top cross-section) and longitudinal (bottom cross-sections: a = 0.6 m, d = 1.5 m, and g = 2.4 m from the window) installation of a same-size beam with a cooling effect of 25 W/m^2 and a window surface temperature of 14 ^{o}C.

During the transition season (Fig. 5), the supply air jet of a beam installed in the latitudinal direction emphasises the convection flow of a cold window, and high speeds (0.23–0.32 m/s) are found at floor level (5–10 cm from the floor) at a distance of 0.6–1.8 m from the window. The velocities are lower in a longitudinal installation than in a latitudinal one. However, rather high velocities occur at certain points, primarily in the centre of the room. Generally speaking, it should also be noted that a radiator installed below a window can eliminate the convection flow that increases the risk of a draught caused by a cold window. In practice, this means that, to minimise the draught risk, the radiator must be turned on during the transition season – regardless of the cooling requirement.

THE EFFECT OF THE LOCATION OF THE THERMAL LOAD ON ROOM AIR DISTRIBUTION

The effect of the location of the thermal load on air velocities in the occupied zone was studied in a test room (6.5 x 4.0 x 2.8 m (H)) with an exposed active chilled beam. The chilled beam with a total length of 5 m was installed in longitudinal direction in the centre of the room. The beam's distance from the ceiling was 250 mm. The total cooling load was 85 W/m^2 when the primary (outdoor) supplied air flow rate was 2 l/s per m^2. In the measurement, the heat loads were: three computers (3x 123 W),three dummies (3x 120 W), light fittings integrated into the chilled beam (140 W), heat gain from floor panel (300 W) and warm window (350 W). The size of the window was 1.3 (W) x 1.9 (H) m. The area of the heated floor was 5.5 m². The velocity field was measured at three different thermal load distributions: 1) normal office room with thermal load on the window and side walls, 2) 50% on the window wall and 50% on the corridor wall, and 3) all load on the window wall (Fig. 6).

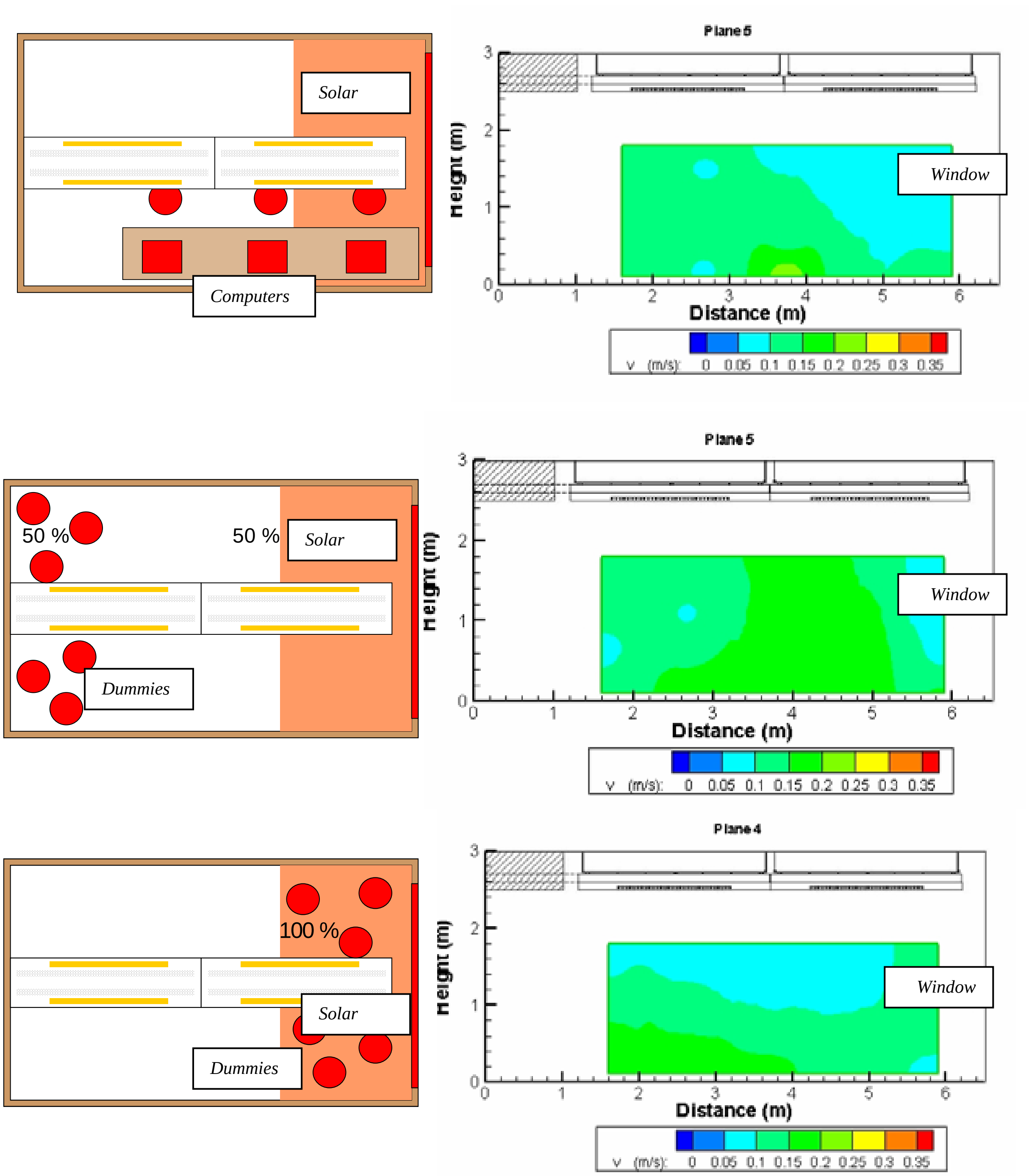

Figure 6. The location of thermal load and its impact on air velocities in the occupied zone. Top: normal office load. Middle: 50% on the window wall and 50% on the corridor wall. Bottom: 100% of the load on the window wall.

In all cases studied, the air velocities were relatively low (less than 0.22 m/s). The highest air velocities occurred in cases where the load had been altered from that of a normal office. The results shown in Fig. 6 by colored field of constant speed reveal that locating all heat sources close to the window wall generates air distribution with a maximum speed in the proximity of the corridor wall, while, when the heat sources were evenly distributed between the window and corridor walls, the maximum speed occurred in the middle of the room. Overall, the location of the thermal load did not significantly affect the operation of the chilled beam for the conditions of the performed measurements.

DISCUSSION AND CONCLUSIONS

Draught discomfort is one of the most often reported compaints in rooms. During design, the draught risk should be considered not only for the conditions which require maximum cooling (summer season) but also for the transition season between summer and winter, e.g. spring and authumn. Depending on the location of the air supply device and the convection flows generated in the room , the transition season may feature air velocities comparable or even greater than those of the maximum cooling sonditions. In general, the air velocities in rooms with latitudinal-installation of chilled beams are greater than than those in rooms with longitudinal-installation of the chilled beams.

The area of occurrence of the maximum speed in the occupied zone depends on the streangth and the location of the heat sources, its momentum flux and on the supplied flow pattern with respect to the convection flow. Generally speaking, convection flows have less impact when the chilled beam is installed in the longitudinal direction than when it is installed in the latitudinal direction.

The results of this study identify a complex interaction of lows which takes in rooms ventilated with chilled beams. Simple analytic modelling of the jets and thermal flows is not sufficent to identify the airflow distribution in rooms. Therefore, it is possible to accurately analyse the various load situations only by full-scale measurements and/or CFD predictions. The design tools currently available enable studying the distribution of air supplied from chilled beams (and in general from other air supply terminal devices) however without taking account of the effect of the convection flows from heat sources. It should be noted that there are no standardised methods for presenting air velocities in rooms, and each manufacturer of air terminal devices has its way of presentation.

ACKNOWLEDGEMENT

The writers should like to thank TEKES (Technology Agency Finland) for funding this project. Special thanks go to Mika Komulainen (Lappeenranta University of Technology) for measuring the thermal sources and supply air jets and to Lyuben Bozhkov and Boryana Yordanova (International Centre for Indoor Environment and Energy, Technical University of Denmark) for the measurements related to the effect of the heat sourse location.

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Experimental study of space cooling using ceiling panels equipped with capillary mats

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SUMMARY

The purpose of this paper is to investigate the thermal performances and the effect on the thermal comfort of a cooling ceiling installed in an experimental test room called Minibat. With a 9.6m^2 surface and controlling the temperatures on all the exterior walls of the cell, Minibat was the perfect environment for this experimental study. The studied ceiling panels were equipped with capillary mats using polypropylene as material. During the experiment we have analyzed different cases where the ceiling surface temperature varied between 15°C and 19°C. Different parameters like air temperature, humidity or surface temperature were measured during the experiment. To evaluate the thermal comfort we have calculated the PMV value for different chilled ceiling surface temperatures. The results indicate that the cooling ceiling could assure the indoor thermal comfort and with good thermal performances in terms of specific cooling rate or vertical temperature asymmetry.

INTRODUCTION

The majority of air-conditioning devices function on the principle of pulsated air, where the hot air of the room is recycled, cooled and returned into the room. The increase of the thermal loads in the buildings, mainly due on arrival of data processing and of office computers, the installation of air-conditioning systems was necessary to neutralize these loads and to create a good indoor thermal comfort. Air conditioning systems, which consume large quantities of energy, have become a necessity for almost all the buildings [1] to provide a comfortable indoor environment.

Currently the evacuation of these quantities of latent and sensible heats is done mainly with air treated introduced by air diffusers. To maintain comfort under these conditions, a greater volume of cooled air must be provided to the working area. Disadvantages such as noise, cold-drafts, air temperature differences between the human head and foot or energy wasting in certain cases, show that a new cooling system needs to be proposed.

The use of water as a coolant to cool surfaces of buildings (ceilings or walls) is consequently a tempting alternative solution. It is even more appealing as water cooling requires much lower flow rates and thus much smaller areas of piping. The chilled ceiling radiant panels are room cooling systems for placement in the ceiling zone.

Their cooling surfaces are connected with closed circuit heat conducting pipework containing flowing chilled water. With a cooled ceiling, the temperatures of a room's surfaces are lower than with air-conditioning solutions and the same is true when other partition surfaces are cooled. The principles of cooling ceilings are not very different from those of a radiator with tubes or hotplates.

The hot air arriving in contact with the cooled surface is cooled below the average temperature of the room and therefore descends at low speed into the occupation zone [2, 3]. The main difference between cooling ceilings and air-conditioning systems is the mechanism of heat transportation. Air-conditioning using air employs convection only, while cooled ceilings employ a combination of radiation and convection. With cold ceilings, the transfer of radiant heat occurs by a clear emission of electromagnetic waves from the hot occupants and their environments to the radiant ceiling.

The sensation of comfort produced by a cold ceiling can be compared to that felt during a summer night where one feels the freshness of the sky even if the ambient air temperature can be higher than 26°C so when using a chilled ceiling it is possible to get the same conditions of comfort with more elevated air temperatures than a classical air-conditioning system [5]. With a cooling ceiling the temperature of water or its flow rate can be varied so that the surface temperature is adapted to the desired conditions.

However the many advantages on thermal comfort [4] or energy reduction [5], the cooling ceiling has not reached its full potential because of the condensation risk on the chilled surface. To prevent the risk of condensation on the chilled water pipes or radiant surfaces, the water flow temperature should not be below 16°C. The dew point temperature of the indoor air must always be lower than the surface temperature of the cooling ceiling, this being a reliable way to avoid condensation.

For a more precise protection dew point detectors can be installed at the coldest point on the flow pipe of the cooling ceiling installation. They indicate the beginning of condensation at an early stage and trigger an increase in supply water temperature or a chilled water supply shutoff. To reduce the dew point temperature and assure the hygienic quality of the indoor air it is necessary to have a ventilation system to bring fresh air in the occupancy area.

This article purpose is to present an experimental study on a chilled ceiling installed in a test cell called Minibat. The main aims were to analyze the vertical temperature asymmetry and the thermal comfort. By taking measurements of air temperature, relative humidity or surface temperature we have been able to give relevant and realistic conclusions on this subject.

THE EXPERIMENTAL SET-UP

The experimental cell Minibat, presented in Figure 1, is composed of two identical parts which dimensions are 3.10m x 3.10m x 2.50m respectively to (x, y, z). In our study we have used the test cell number 2 where we have installed the radiant cooling ceiling panels. The glazed façade separates Minibat from a climatic chamber whose temperature is controlled and can vary between -10°C and 40°C. A thermal guard allows keeping the five others faces at a constant temperature, which can vary between 5°C and 30°C. During our tests, the thermal guard temperature was kept at 26°C. A battery of 12 spotlights, of 1000W each one, not used in this study, makes it possible to simulate an artificial sunning (gas-discharge lamps with metal halides which spectrum is similar to the sun one). All the faces temperatures are measured using thermocouples of resolution ±0.4°C, and each face with 9 thermocouples. The temperatures of the climatic chamber and the various parts of the thermal guard are measured using Pt100 probes which resolution is of ±0.3°C.

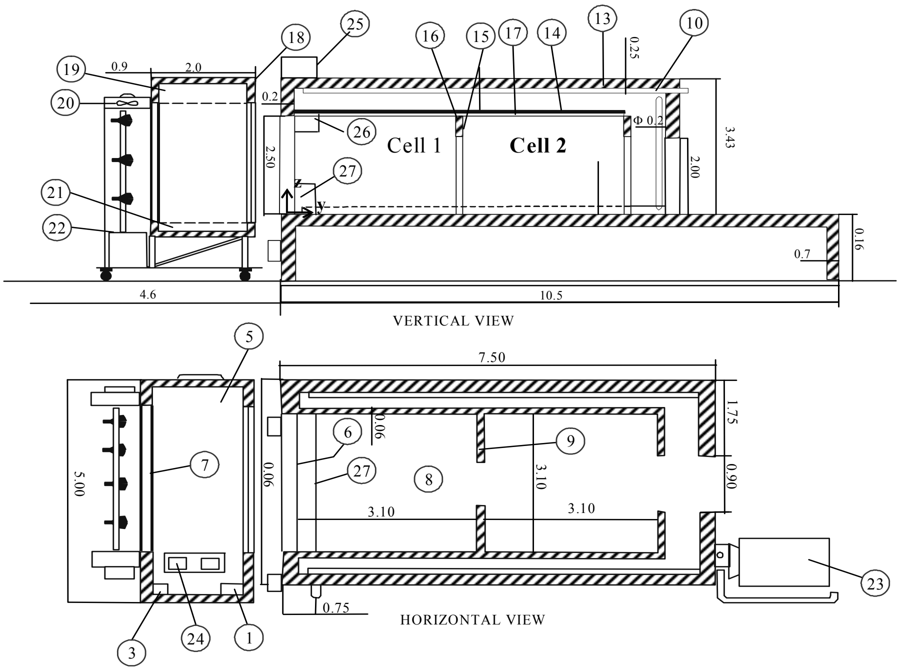


Figure 1. Experimental test cell Minibat

Table 1 presents the structure of the wall, ceiling and floor of the experimental test chamber. The cooling ceiling panels are metallic and are insulated with 8 cm foam to reduce the heat losses. These one have been added on the lower face of the existing ceiling.

Table 1. Composition of the test cell Minibat (before adding the cooling ceiling)

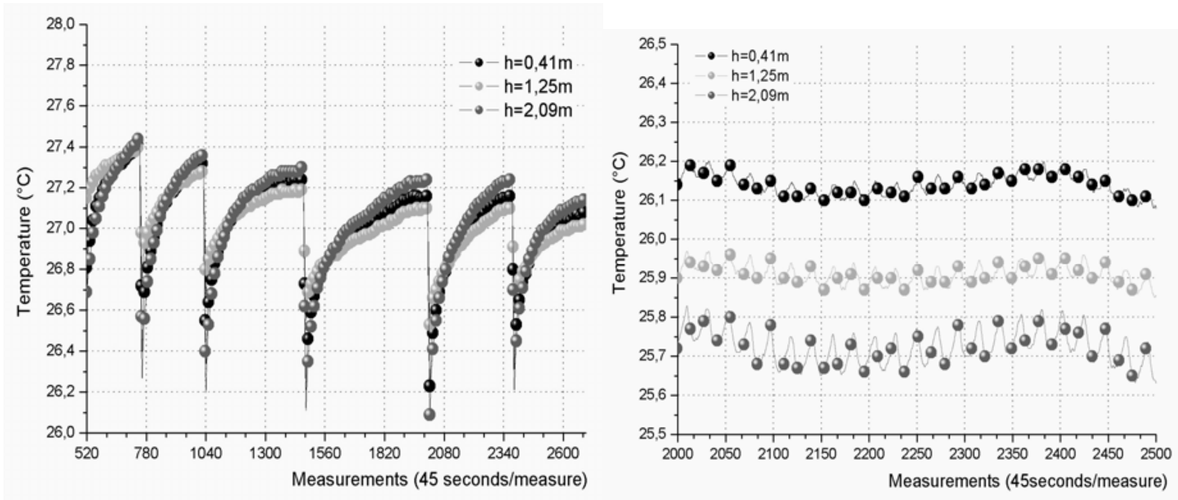
Wall	**Material**	**λ [W/m°C]**	**ρ [kg/m³]**	**Thickness [mm]**
Floor	concrete	0.16	400	200
Wall	plaster plate	0.35	817	10
	insulated material	0.06	200	50
	plaster plate	0.35	817	10
	wood plate	0.136	544	50
Ceiling	plaster plate	0.35	817	10
	wood plate	0.136	544	8
	insulated material	0.06	200	55
	wood	0.136	544	25

For the data acquisition we have used a computer and a multifunction board with more than 100 connections. To measure the indoor humidity we have used a humidity sensor placed at the centre of the room with a precision of ±2.5% for measurement of relative humidity between 10 and 90%. A globe temperature is also measured at the center of the room. Concerning the cooling ceiling, we have used 9 panels equipped with capillary mats and being placed only in cell 2. The exterior diameter of the tubes is 3mm and they are spaced by 1.5cm. An indoor heat gain of 250W was installed in the test cell which has a surface of 9.61m^2. To measure the vertical asymmetry of temperature we have taken measurements for six points of height.

Several tests were done by modifying the surface temperature of the cooling ceiling in order to see the effect on the thermal comfort or the vertical asymmetry.

RESULTS

The temperature of the thermal guard was set to 26°C during the whole experimentation. It was observed that there were variations but with a maximum of 1°C due to the system controller but which we have considered totally acceptable. Figure 2 presents the temperature for different heights on the exterior and interior walls surfaces of the test cell.



Exterior walls (thermal guard) Interior walls (cooling ceiling activated)

Figure 2. Temperature variation against the walls during the experiment

Being surrounded by surfaces that have large temperature differences (cooling ceiling and the walls in our case) may be a cause of discomfort, even when the air temperature is considered in the comfort zone. These conditions of discomfort are frequently caused by cold windows, un-insulated walls or direct sunlight. In general, people are more sensitive to asymmetric radiation caused by a warm ceiling than that caused by warm vertical surfaces [7].

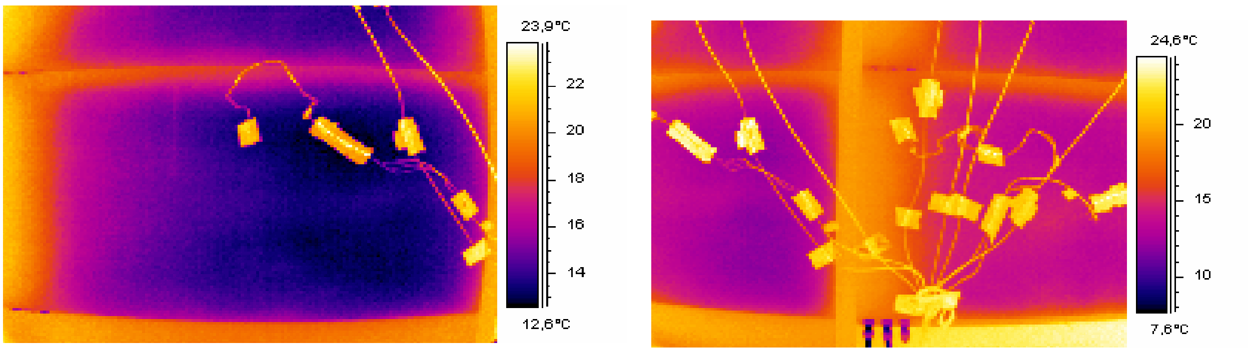


Figure 3. Thermal screenshots of the cooling ceiling during the experiment

The temperature on the surface of the chilled ceiling is relative constant on each panel, except the connection between them (see Figure 3). A reason that temperature on the surface is not perfectly uniform could be that a gap of air was formed between the tubes and the metallic panels and acting like an insulation on some areas.

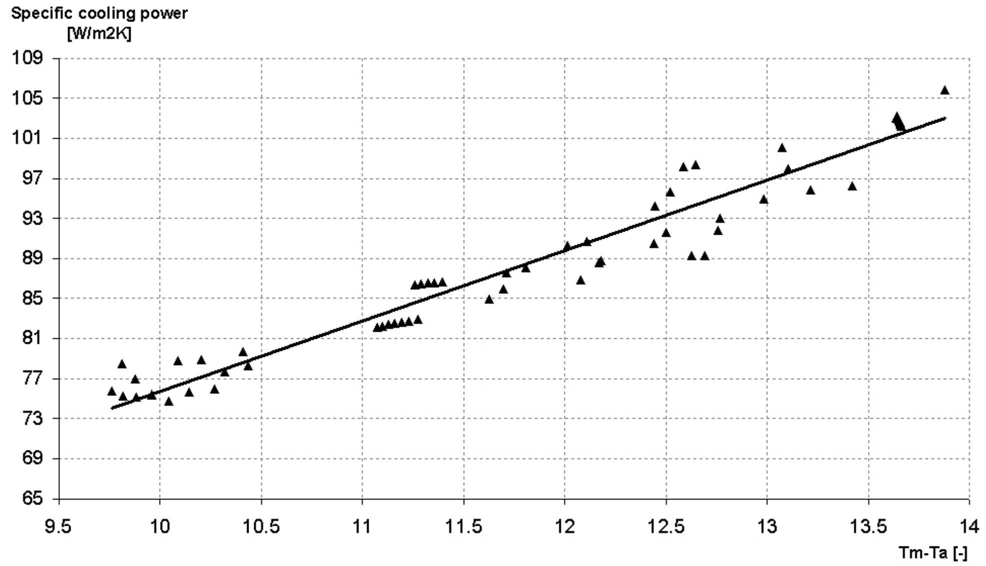


Figure 4. Specific cooling power of the cooling ceiling in function of the difference between the mean water temperature and the air temperature

The cooling power of a radiant ceiling (RC) system is a function of the heat transfer between the room and the cooled ceiling. This heat transfer has two components: radiation and convection which can be calculated. In Figure 4 is presented the specific cooling power of the chilled ceiling depending on the difference of the mean water temperature which was measured during the experiment and the air temperature.

The specific cooling power of a cooled ceiling can be expressed by the following empirical equation:

$$q = 8.92\ (t_{air}-t_{cold\ surface})^{1.1}$$

where q is the sum of the convective and radiant heat transfer [W/m^2]
t_{air} is the air temperature of the room [°C]
$t_{coldsurface}$ is the temperature of the cooling ceiling [°C]

A large vertical air temperature difference between the head and ankles may be also a discomfort cause. Based on criteria of 5% dissatisfied, the allowable temperature difference is 3°C, which applies for situations where the temperature increases with height from the floor (i.e., the head is warmer than the feet). With a cooling ceiling the temperature decreases when we are approaching the chilled ceiling height and when the temperature of the ceiling is low enough we can even sense the refreshment on the body skin.

The results indicate that in the centre of the test cell, the stratification is acceptable and is smaller than the 3°C limit imposed by standards. In Figure 5 it can be observed the vertical temperature asymmetry for three different cooling ceiling temperatures. In all the cases the difference between the ankle and the head is about 1°C. The data presented in Figure 4 was obtained by measuring the temperature in six points of height, from 0.4m to 2.3m. The time step of one measure was of 45 seconds which correspond to the minimum imposed by the acquisition system to pass throw all the measurements including air and surface temperatures, water temperature, relative humidity or globe temperature. We can conclude on this part that the cooling ceiling don’t create discomfort in the working area but the need of taking measurements also near the walls seems necessary to be done. All the temperatures were measured by thermocouples type K with a precision of ±0.3°C.

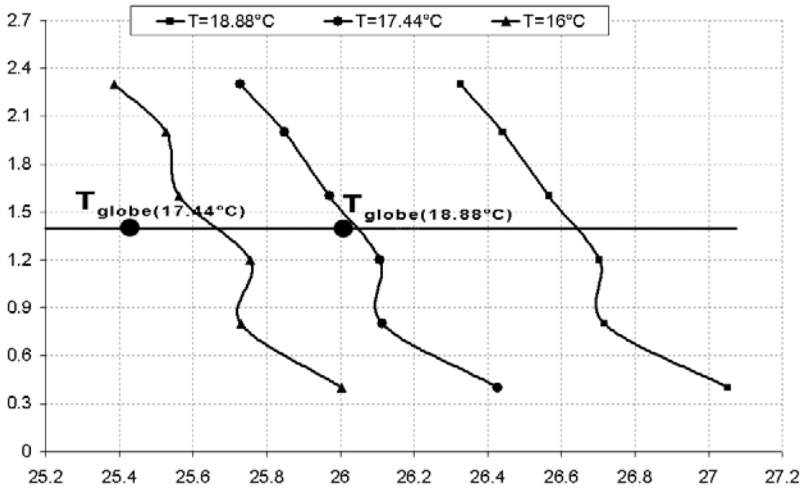


Figure 5. Vertical temperature asymmetry in the centre of the test room for different ceiling temperatures

Thermal comfort is very difficult to define because we need to take into account a range of environmental and personal factors when deciding on the temperatures or the ventilation that will make a comfortable indoor space. The best that we can realistically hope to achieve is a thermal environment which satisfies the majority of people or the so called ”reasonable comfort”. In order to analyze the thermal comfort in a room cooled by a chilled ceiling we have taken measurements of air temperature, globe temperature, relative humidity and surface temperatures (9 thermocouples per internal surface of the room). By simulating the test cell using the computational fluid dynamics (CFD) we have estimated the air velocity inside the room. The PMV (Predicted Mean Vote) [6] was calculated by using the data obtained after the tests. The PMV index establishes a thermal strain based on steady-state heat transfer between the body and the environment and assigns a comfort vote to that amount of strain.

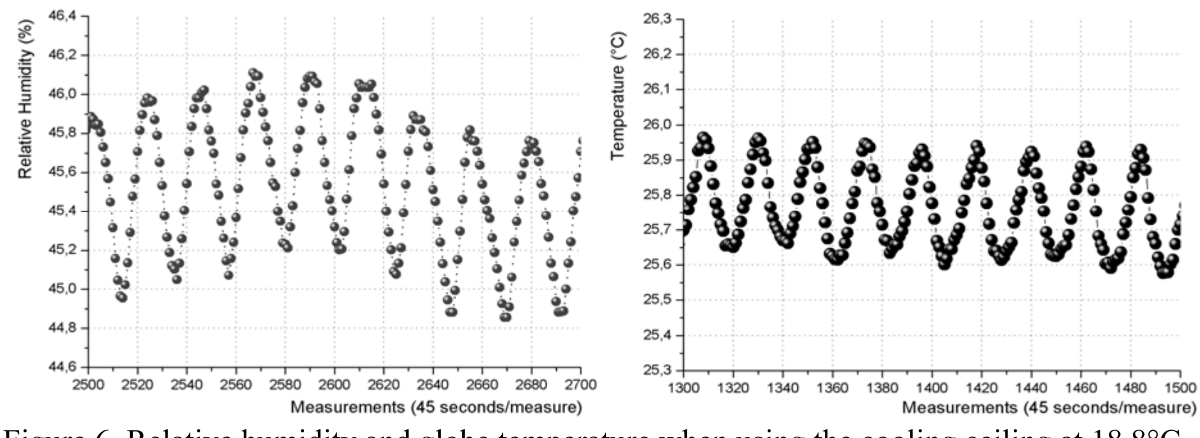


Figure 6. Relative humidity and globe temperature when using the cooling ceiling at 18.8°C

In Figure 6 are presented the evolutions of relative humidity and the globe temperature during more than 12 hours of experimentation. We can observed that the relative humidity was stabilized at a value of 43.5% and the globe temperature at 25°C when using the chilled ceiling at 17.4°C. The PMV equation for thermal comfort is a steady-state model. It is an empirical equation for predicting the mean vote on an ordinal category rating scale of thermal comfort of a population of people. The air velocity was considered after the CFD simulations of 0.15 m/s, the rest of parameters influencing the thermal comfort being measured. In Figure 7 it can be observed that the PMV index for different clothing insulations when using the cooling ceiling surface at different temperatures. We have only presented two cases of ceiling surface temperature which were the more relevant, knowing that in general the cooling ceiling temperature must be kept at higher values than the dew point temperature of the indoor air in order to avoid the condensation risk.

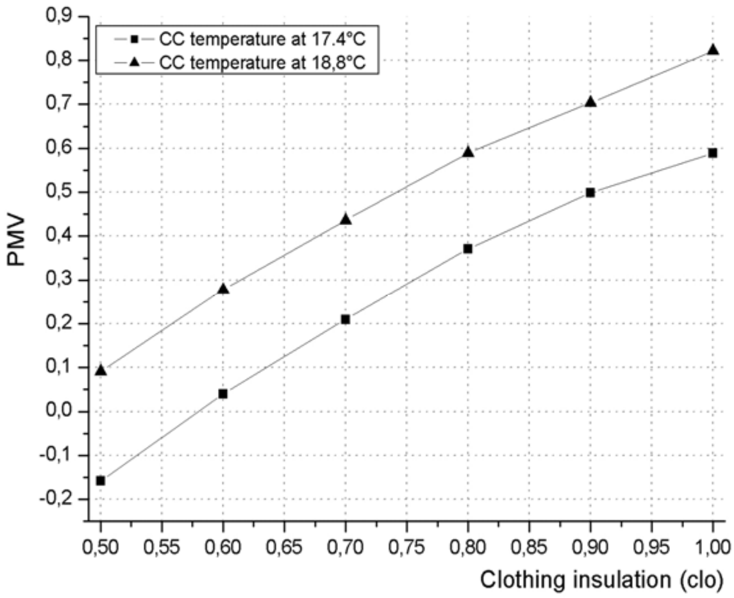


Figure 7. PMV for different clothing insulation when using the cooling ceiling at different temperatures

The values presented in Figure 7 show that the cooling is a very attractive solution for the thermal comfort. It can be observed that when the cooling ceiling temperature is at 17.4°C and when the metabolism is 1.2 met (70W/m^2) and the clothing insulation is 0.9 clo the obtained value of PMV is in the limit proposed by the norms.

DISCUSSION

The results obtained after the experiment showed that with a cooling ceiling the vertical temperature asymmetry is less than 1.1°C, acceptable value which not create a discomfort in the occupancy zone. Another important aspect of the test was the effect of the chilled ceiling on the thermal comfort. The temperature of the ceiling was changed in order to see the effect on the thermal comfort, the results being very satisfying. The PMV was calculated for different clothing insulation, its values fulfilling the limits imposed by the standards. These discussions prove that ceiling radiant cooling systems can be more comfortable than conventional air cooling systems, due to less air movement, minimal vertical air temperature difference and with good results on the thermal comfort even for higher values of the ceiling temperature. However the big amount of data obtained more extensive experimental test research are sorely needed especially on the condensation risk problems or the effect of hygroscopic materials on the thermal performances.

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Calculation Method For Summer Cooling With Radiant Panels

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SUMMARY

Radiant panels achieved a growing success in the last years as a cooling system, because they guarantee improvement in comfort and energy saving.
The aim of the present work is to propose a method to evaluate the thermal behavior of radiant panels used for summer cooling. Experimental observations lead to notice that part of the radiant load hitting panels surface is directly removed by the refrigerating fluid (Direct Water Load), never becoming a room load. In order to evaluate the role of this fraction of load on thermal balance, it has been developed a specific procedure deriving from the Room transfer functions method.
Through a lighting analysis tool it has been furthermore evaluated the Direct Water Load due to solar radiation, the most important radiative load for typical office rooms, in function of geometrical configuration and surfaces reflection factors (α, ρ). The right understanding of the role played by Direct Water Load on thermal behavior of radiant panels may lead to a better design of this kind of cooling systems.

INTRODUCTION

The use of radiant panels as a cooling system is quite new in a lot of countries, but it is developing extremely fast, conquering each year relevant market slices.
The success of this kind of HVAC system is bound to design, comfort and energetic reasons. The use of radiant panels leads to have more free usable net floor space, to reduce draft risk, typical of cooling air systems, to achieve quite uniform temperatures into the rooms, to obtain significant energy saving.
For comfort reasons, anyway, the surface temperature of panels must not be lower than 19°C and in order to avoid moist condensation it must not be lower than the dew point temperature of the room. Since cooling panels can control only sensible loads and not control air humidity and quality, they are always coupled with primary air, which can provide any air control. Sometimes primary air may even help the radiative system to balance peak loads, if they grow up beyond panels thermal efficiency.
Radiative systems use wide exchange surfaces, and since human body exchanges heat through radiation approximately for the 50% of the whole balance (in cooling period), it is clear that panels temperature can be quite near to project air temperature maintaining optimal comfort condition. At the same time, since the most of heat exchanged between the body and the cooling system is through radiation, air temperature can have little higher values, without creating comfort problems, since operative temperature doesn't change so much, so the thermal sensation of people remain the same. Higher values of air temperature bring anyway to a reduction of the cooling load, for the reduction of heat dispersions through walls and air infiltration.

The most important element which can bring to significant energy saving is anyway the opportunity of using energy transfer mediums, usually water, with moderate temperature (near to 16°C) [1]. This is why these systems are often called as "low exergy", because they can work using energy which has a very limited convertibility potential, such as heat close to room air temperature and this means that radiant panels can be fed by solar collectors (solar cooling) or heat pumps [2, 3].

METHODS

Cooling Load Evaluation

In order to study a cooling system, it is essential to use a multi-step analysis which considers the effect of loads distribution during the whole day. Heat gains and cooling loads to be removed by the cooling system are not the same during different hours, because the thermal mass of the building accumulates heat (especially due to solar radiation) and after a time lag re-emits this into the room. So a specific heat gain at the time T_θ, may have its real effect on the cooling system only at the time $T_{\theta+n}$.
The most common calculation methods, as the Room transfer functions method, use therefore the concepts of Heat Gain and Cooling Load, as explain after:

- The *Heat Gain* is the amount of heat gendered or introduced into the room at a specific time T_θ
- The *Cooling Load* is the amount of heat that must be taken away from the room at the same time T_θ, in order to maintain the project conditions of temperature and humidity
- The *Heat Extraction Rate* is the real heat rate taken away from the room by the plant at the time T_θ (it is linked to the plant inertia)

The thermal inertia of the room hence correlates Heat Gains and Cooling Loads. It is possible to break Heat Gains between convective gains and radiative gains, the fist ones, working directly on air, are not involved in thermal mass absorption phenomena, the radiative gains are, on the contrary, partly absorbed by the elements of the room having relevant thermal mass. After a certain time, only a reduced part of the absorbed radiation is re-emitted into the room through infrared wave and convection. So thermal mass operates on radiative loads delaying and dimming their effect, while convective and latent loads directly become cooling loads (fig.1).
Heat Extraction Rate is mostly important for estimating energy use over time, but it is not relevant to calculate design peak cooling load, for this reason we don't consider it in our analysis.

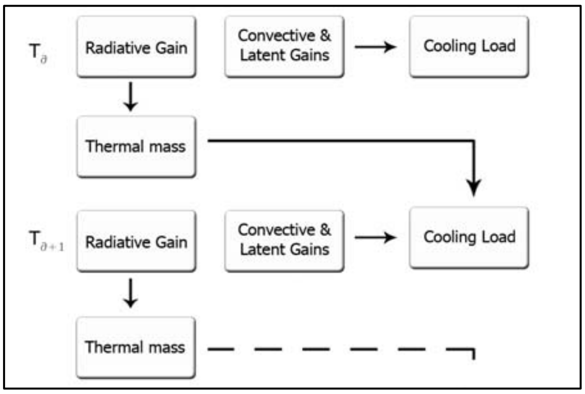

Figure 1. Heat loads scheme inside a room

Cooling Load Evaluation For Radiant Panels

Cooling load evaluation for radiant panels needs a further level of analysis, due to specific characteristics of this kind of cooling systems [4].

Radiant panels are light elements, usually in metallic alloy or plasterboard, without relevant values of thermal mass; they have on their backside a coil system for the refrigerating fluid flow (typically cold water). If a radiative flux (solar radiation, artificial light, far IR radiation…) hits the surface of the panel, the energy transfer medium directly removes it. So that radiative gain will never become a Room Load, because it will never be re-emitted through convection into the room. It is therefore possible to compare the behavior of radiant panels to the one of a well with a radiative power foisted by the temperature of the refrigerating fluid. The scheme about Heat Gain and Cooling Load previously shown (fig.1), must consequently be corrected when radiant panels are used, considering the rate of the radiative gains hitting the panels surface and directly removed by water.

Each Radiative Heat Gain at time θ must be divided in the rate F directly removed by water, called Direct Water Load (DWL) and in the remaining part (1-F) which will be influenced by the room thermal inertia (fig.2). If we consider the Room transfer functions method for predicting cooling loads, the ratio of radiative Heat Gain absorbed by thermal mass at time θ (1-F), will become a Room Load (RL) at time θ+n.

In the most general situation, in which both panels and air are used to removed sensible loads, the RL at time θ will be partly (X) contrasted by the radiative system through the Surface Load (SL), and partly (1-X) by the air, through the Cooling Load air (CLa). The sum of Surface Load at time θ, together with the Direct Water Load at time θ, represent the Cooling Load water (CLw), so the amount of heat removed from the room by the radiative system at time θ (fig.3).

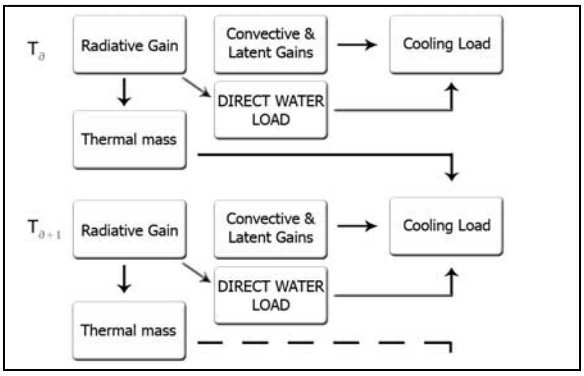

Figure 2. Heat loads scheme inside a room with radiant cooling panels

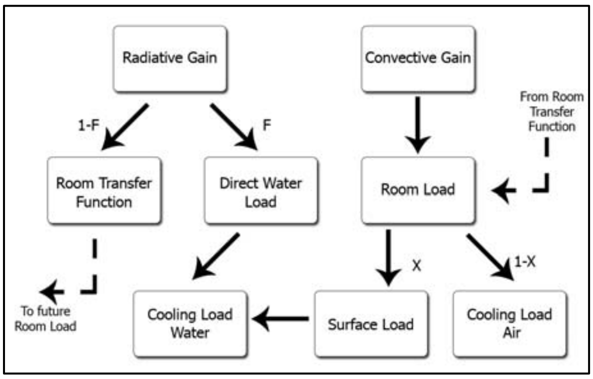

Figure 3. Cooling Load generation with radiant cooling panels

It is very important to underline the different origin of the two component of cooling load burdening at time θ on the radiative panels: the Direct Water Load directly derives from Heat Gains at time θ while the Surface Load derives from Room Loads and so its value is influenced by delay effect due to room thermal inertia and expressed by room transfer functions. This analysis is confirmed by the empirical approach adopted by some constructors, which suggest to subtract a certain ratio of the radiative gains, from the cooling load evaluated with traditional methods, before determining the efficiency of panels.

Distinguish between DWL and SL is not just a theoretical exercise, it has indeed practical design consequences [5, 6]. When it exists a high Direct Water Load, the Room Load is strongly reduced, and so the Surface Load and/or the Cooling Load Air. Since the Surface Load is limited by the panels surface temperature, when the Room Load is little, panels can anyway succeed in removing the whole sensible load or even a lower radiant surface may be

needed in order to contrast the load. It is possible otherwise to reduce the Cooling Load air, if primary air is anyway needed to control indoor climate.
It must be however underlined, that even if Surface Load or Cooling Load air are lowered by using this analytical approach, the whole load burdening on the plant is higher, because the Direct Water Load is an instantaneous load, so it is not dimmed by room thermal mass effect.
Another interesting aspect depends on the different origin of DWL and SL: if the temperature difference between air and panel surface is zero, the Surface Load is zero too, but if there is at the same time a radiative flux hitting the panel, the Cooling Load is not zero because there is a Direct Water Load. The fundamental equation for cooling load evaluation is the following:

$$\dot{Q}_{CLw} = \dot{m}_w \cdot c_p \cdot \Delta T_w, \quad (1)$$

where ΔT_w is the difference between inlet and outlet temperature of the refrigerating fluid.
So the cooling efficiency of the radiative system can be expressed as follow:

$$\dot{q}_{CLw} = \left(\dot{Q}_{CLw} / A\right) = \left(\dot{m}_w \cdot c_p \cdot \Delta T_w / A\right), \quad (2)$$

And the surface cooling efficiency:

$$\dot{q}_{SL} = \left(\dot{Q}_{SL} / A\right), \quad (3)$$

The two characteristic efficiencies are linked together through the following relation, which considers the DWL:

$$\dot{q}_{CLw} = \dot{q}_{SL} + \left(\dot{Q}_{DWL} / A\right) = \dot{q}_{SL} + \dot{q}_{DWL}, \quad (4)$$

In order to evaluate the Cooling load, DWL and SL must therefore be studied separately.

Heat Removed By Panels Surface – Surface Load
Concerning the calculation method shown before, the amount of load removed by the panels surface at each time step, depend on:

- Convective heat exchanges between room air and panels surface
- Radiative heat exchanges among panels surface and walls surfaces

The cooling power of the panels is therefore strictly bound to panels surface temperature, which is however strongly limited by moist condensation problems and comfort needs. The value of this temperature is the most critical point of the whole system thermal behavior.
The heat power removed by the panels surface can be evaluate through the follow equation:

$$\dot{Q}_{SL} = h \cdot A \cdot \left(T_{sp} - T_{op,eq}\right), \quad (5)$$

where h is the surface conductance, A is the panels surface, T_{sp} is the temperature of the panels surface, and $T_{op,eq}$ is the operative equivalent temperature of the room, expressed as follow:

$$T_{op,eq} = \frac{h_r \cdot T_{mr} + h_c \cdot T_{air}}{h_r + h_c}, \quad (6)$$

T_{mr} represents the mean radiant temperature of the room, calculated excluding the panels surface temperature. Since the most common value used in design stage is the air temperature, and since the T_{mr} is a difficult parameter to be obtained, it is commonly used a simplified equation to evaluate the heat power removed by the panels surface:

$$\dot{Q}_{SL} = h \cdot A \cdot (T_{sp} - T_{air}), \quad (7)$$

Direct Water Load Evaluation

The most difficult element to evaluate is the Direct Water Load, because its value depends not only on the characteristics of the radiative source and on the radiative properties of the surfaces of the room, but even on the geometrical shape of the room.
Generally it can be expressed as:

$$\dot{Q}_{DWL} = F \cdot \dot{Q}_{rad}, \quad (8)$$

where F represent the ratio of the radiative flux Q_{rad} which is directly removed by the heat transfer medium.
The F value has been studied with the help of two simulations tools, the first one is a simple calculation sheet, based on the Reflection method, which lets to follow the path of the radiation (solar, human body, appliances) within the room. Through the Angle Factor source-wall, the radiation is spread over the floor, the walls and the ceiling. After this, through the Shape Factor, part of the incident radiation is reflected from the walls surfaces all around again [7]. Following all the reflections it is possible to evaluate the amount of each radiative flux removed by the radiative panels.
The other one is instead a lighting analysis tool, called Lighscape and so it lets to study the F value just for solar radiation and some artificial light source, but not for fully IR radiation sources. Anyway solar radiation is the most relevant radiative flux, especially in highly glassed buildings. This second tool is able to simulate both direct and diffuse radiation, drawing maps of light distribution over the different walls, and this is extremely important to improve the accuracy of the analysis, since the first model simulates just perfectly diffusing sources and materials.
It is interesting to point out that, for solar radiation, even do not considering the UV and IR fractions, lighting tools produce just little errors, whom bring to little underestimations of the Direct Water Load. Solar spectrum is divided in four main parts, which at ground level assume the following values: ultraviolet (6%), visible (55%), near infrared (39%) and far infrared (0%) [8]. Since all kind of glass used in the construction field are opaque to far infrared radiation, it can’t enter into the room, so it is not considered for cooling load evaluation. Anyway it is just a principle statement, since solar far infrared radiation at ground level is practically inexistent, because filtered by atmosphere.
To obtain the most general considerations, in the following analysis we will consider a clear glass with the common thickness of 6 mm, having the following standard characteristics:

Table 1. Solar radiation mean transmission factors, for UV, Visible and IR wavelengths

Global	UV	Visible	near IR	far IR
τ_e	τ_{UV}	τ_v	τ_{IRn}	τ_{IRf}
81%	56%	89%	74%	0%

So the solar radiation entering into the room is composed by:

2. Percentage of UV, Visible and IR entering into the room through a clear 6 mm glass

Φ_{IN}	UV	Visible	near IR	far IR
100%	4%	60%	36%	0%

The elements into the room are anyway generally considered as gray bodies, so with an emissivity and an absorption factor to infrared radiation of the 90% ($\varepsilon=\alpha=0,9$) [9]. It means that just the 10% of the IR radiation entering into the room is involved in DWL generation, the remaining 90% is absorbed by opaque elements, then re-emitted after a time lag into the room and so it is involved in SL generation. The 10% of the 36% represent just the 4% of the whole entering radiation, so using a lighting tool to evaluate the F factor, which doesn't simulate UV and IR radiation, it is committed a global error of the 8%, acceptable for engineering applications.
Furthermore, UV radiation has a little contribution to cooling load generation, its energy is principally involved in surface material aging and degradation, and just a little portion of it generates heat [10].
It is obvious that all F values obtained with the lighting tool must be multiplied for a reduction factor (0,6 for a clear glass with thickness of 6mm), in order to be referred to the global radiation, since solar loads values are expressed for the whole solar radiation.
It is interesting to notice that DWL is practically generated only by visible radiation since IR contribute just for the 4% of the entering radiation, when it is considered a clear glass with thickness of 6mm. With other kind of commercial glass this consideration is anyway correct, since the ratio τ_{IRn}/τ_v usually reduces or, at least stays the same.
Within a room with cooling radiant panels, therefore, light reflection factors of surfaces play a fundamental role in DWL, SL and CL generation. A highly reflective room, for instance, will have high Direct Water Loads, and lower Surface Loads. Furthermore peak values of Cooling Load will occur in correspondence with solar gain peak values.
For this particular condition, it is possible to state that part of the thermal inertia of a room is constantly operating with IR radiation, while another slice of it works, or not, in function of the surfaces light absorption and reflection factors. In a room with cooling radiant panels, the effective activation of the room thermal inertia depends therefore not only on the surface thermal resistance of walls, but even on surfaces light reflection factor.
On the contrary it is possible to observe that if room surfaces had lower emissivity, some more IR radiation would be involve in Direct Water Load generation and less in Surface Load generation.

RESULTS

F Value For Solar Radiation

The two analysis tools described before have been used to evaluate the F value for a radiant ceiling, as a function of different light reflection factors of the room floor, and of different room dimensions. All the other room surfaces has been considered with a constant light reflection factor of 70%, corresponding to light paint colours and a metallic ceiling.
In the following graphs, the ratio *L/h* is used as an adimensional indicator of the room geometry. It grows with the growing of the floor surface, in fact *h* is the height of the room, while *L* is the side of the square floor.
The following simulations consider solar radiation for a latitude of 45° North, a full glassed building, and rooms with only one surface facing outside, therefore completely glassed. This is the condition in which it is possible to reach the highest level of DWL, and furthermore it has become very common in office buildings.
The commercial lighting tool can give different F values as a function of the window orientation, because it considers both direct and diffuse radiation. It could be possible to have a F value for each orientation, each hour and day of the year, but it would require huge amounts of simulation time. The reflections method, instead, considers the window like a diffusing light source, so the relative position of the room respect to the sun is not evaluated.

In order to have a useful design instrument, the values obtained with the commercial lighting tool at different hours, have been used to obtain mean values for summer design day (21 July) and winter design day (21 Janurary). Buildings with fully glassed envelope, usually office buildings, may indeed need cooling even during winter time, because solar heat gains and internal gains can be very high; this is why even the winter design day has been investigated.

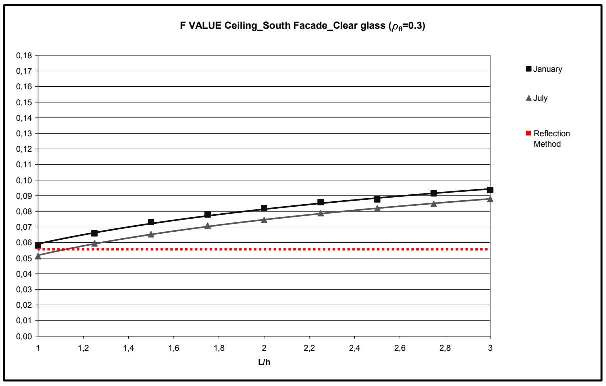

Figure 4. F values for a low reflecting floor

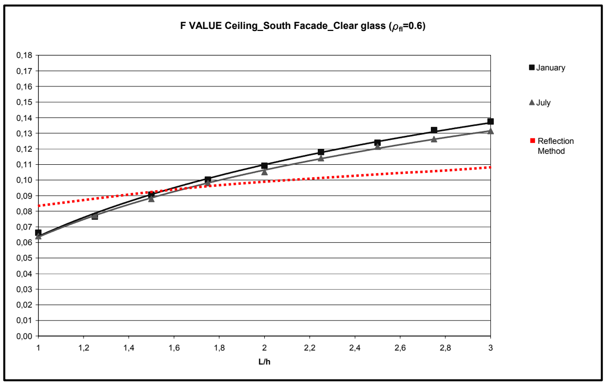

Figure 5. F values for a high reflecting floor

As shown in Figure 4 and 5, curves obtained with the reflection method are generally lower than the ones obtained with the commercial lighting tool, because the first method doesn't consider the real path of light inside the room.

Calculation Example

In order to evaluate the effect on design choices deriving form the use of the different curves proposed, a calculation example has been developed.

It is considered a room with a completely glass façade (L/h=2,5), oriented toward south at latitude 45° North and realize with a single clear glass. Just solar radiation is consider to overheat the building. Internal summer condition of project are established: T_{air} = 26°C and U.R. = 50%.

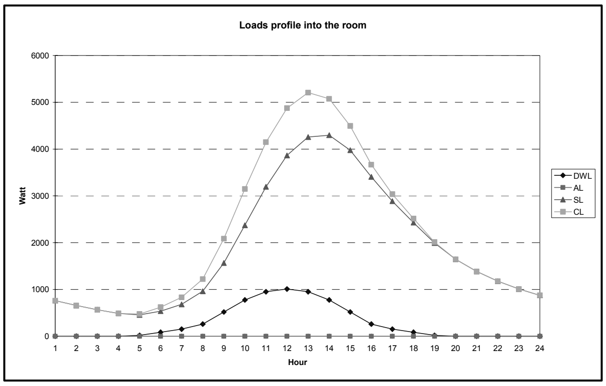

Figure 6. Load profiles with lighting tool

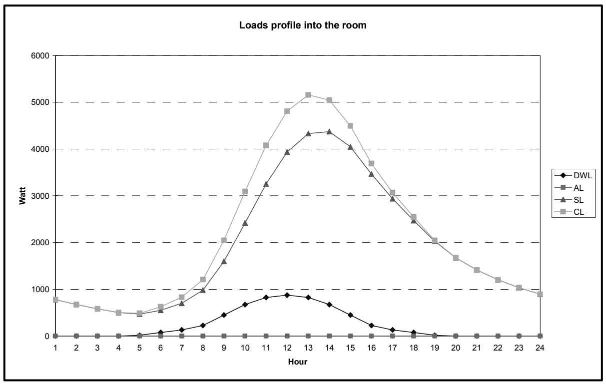

Figure 7. Load profiles with reflection method

Figure 6 shows Direct Water Load, Surface Load and Cooling Load hourly profiles inside the room, obtained using F value calculated with the commercial lighting tool. Figure 7 shows the same Loads evaluated with F value obtained using reflection method.

It is easy to see that in the second graph DWL is lower, while SL increases, furthermore in the second graph the load not carried by the radiant system is higher than in the first evaluation, and this means that primary air must works strongly when DWL is low.

The peak value of Cooling Load is anyway higher when DWL is higher, and this because this is an instantaneous load, that is not dimmed by room thermal mass.

Surface Load peak value, using F values evaluated with the reflection method, is 100 W higher than peak value calculated using the commercial lighting tool as reference. This means that in design phase lower temperatures for the heat transfer medium or wider radiative surfaces, will be considered.

DISCUSSION

The calculation method proposed in this paper may help the designers to better understand thermal behavior of cooling radiant panels. It has been developed using the Room transfer functions method, but it can be easily and worthily adopted even with the Radiant time series method [11].
The effect of Direct Water Load, is extremely important for design aspects, because it strongly influences Cooling Load and Surface Load peak values. As seen before, higher Direct Water Loads means lower Surface Loads but higher Cooling Loads, and even the delay of Cooling Loads may be reduced if Direct Water Loads are sensibly high.
Two different tools have been used in this analysis in order to evaluate DWL. For little rooms, the tools give similar results, while for larger rooms the values tend to be different, anyway values can be considered quite closed. F values obtained with the commercial lighting tool are generally higher, because this instrument considers even the direct solar radiation. The reflection method, instead, considering just diffusing surfaces, tends to underestimate the DWL, producing more cautious solutions in the design phase.
In order to confirm these reflections it would be extremely important to obtain real values, or values deriving from physical tests into a climatic chamber, to deal with the ones obtained with simulation tools.

ACKNOWLEDGEMENT

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Human response to thermal environment in rooms with chilled beams

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SUMMARY

The importance of heat load and airflow pattern control for occupants' thermal comfort is studied in a full-scale test room ventilated with chilled beams. The room was furnished with two desks, computers and table lights. Solar irradiation on window and part of the floor was simulated as well. The room temperature was maintained at 24ºC. Thirty subjects (15 female and 15 male subjects) participated in the experiments. The increase of the heat load from 30 W/m^2 to 70W/m^2 increased substantially non-uniformity of the thermal environment in the room. The percent of subjects dissatisfied due to their thermal comfort conditions and due to draught increased with the increase of the heat load. The provided induction control of the chilled beams changed the air flow distribution in the occupied zone and decreased the reported draught discomfort. The induction control is efficient way for improving occupants' thermal comfort but it should be used carefully in practice.

INTRODUCTION

The air-conditioning of office buildings is maintained mainly with systems based on the mixing ventilation principle. During the recent years the use of active chilled beams for ventilation of buildings has increased due to their ability for removing higher heat loads in more energy efficient way than other systems for office ventilation [1]. Furthermore chilled beams, allow for control of the induced amount of room air and thus for change of airflow pattern in rooms when uncomfortably high velocity is reported by occupants.

Air distribution in rooms with active chilled beams is result of complex interaction of ventilation flow from the beams with the convection flow generated by heat sources, occupants, warm/cold window surfaces, office equipment. The airflow distribution depends on several factors, including arrangement of chilled beams, supplied airflow rate and momentum flux, lay out of workplaces, strength and location of heat sources, etc. Only limited studies on the air distribution in rooms with chilled beams is reported in the literature [2, 3, 4]. The impact of the airflow interaction in rooms with active chilled beams on occupants' comfort is needed. The efficiency of the airflow pattern control for achieving comfortable thermal conditions for occupants is not documented.

This paper presents results on human response to thermal environment generated by active chilled beams. The impact of heat load and airflow pattern control on thermal comfort is studied with human subjects under realistically simulated laboratory conditions. Only part of the collected and analyzed results are presented and discussed in this paper.

EXPERIMENTAL DESIGN

Experimental facilities

A full-scale test room (L *x* W *x* H = 5.4m *x* 4.2m *x* 2.5m) was used to simulate a real office room. The room was furnished with two desks each with a real computer to simulate the heat loads from office equipment. Artificial windows with surface temperature control were used to simulate the impact of the outdoor environment during summer season. Several heating panels were placed on the floor and used in some of the experiments to generate heat and simulate solar irradiation (it is difficult to simulate properly the solar radiation). The work place near the window is referred as WP2 and this on the opposite side as WP1 (Figure 1).

Three chilled beams were mounted crosswise installation as shown in Figure 1. Previous physical measurements identified this lay out as most critical in regard to occupants' thermal comfort [3]. The amount of room air inducted into the beams was regulated by shutters placed in the beams. When the shutter is closed (right side of the chilled beam shown in Figure 1) maximum cross section area for the flow is ensured and thus maximum amount of room air is induced through the beam). This setting is referred in this paper as "Induction 1 (I1)". When the shutter is open (left side of the chilled beam in Figure 1) the cross section area decreases and thus the air supplied to the room decreases (the amount of induced room air also decreases). This setting is referred as "Induction 2 (I2)".

The surface temperature of the artificial windows, the air temperature, etc. were measured and controlled.

Experimental conditions

Three experiments referred in Table 1 as Case 1, Case 2 and Case 3 were performed. In cases 1 and 2 the heat load was kept constant but the induction of room air was changed, Induction 1 (I1) and induction 2 (I2). The low induction (Case 2) was activated for the side of the chilled beams facing WP1. The cooling capacity of the chilled beams was maintained the same by lowering of the temperature of the inlet water for the heat exchanger. In Cases 1 and 3 the induction was kept constant (I1) but the heat load was changed.

Table 1. Experimental conditions.

Case number	Induction control	Temperature in the room, °C	Heat loads, W/m²	Surface temperature of windows, °C
Case 1	High induction (induction 1)	24.0± 0.2	70	40.4
Case2	Low induction (induction 2)	24.1 ± 0.1	70	41.5
Case 3	High induction (induction 1)	24.0 ± 0.2	30	36.1

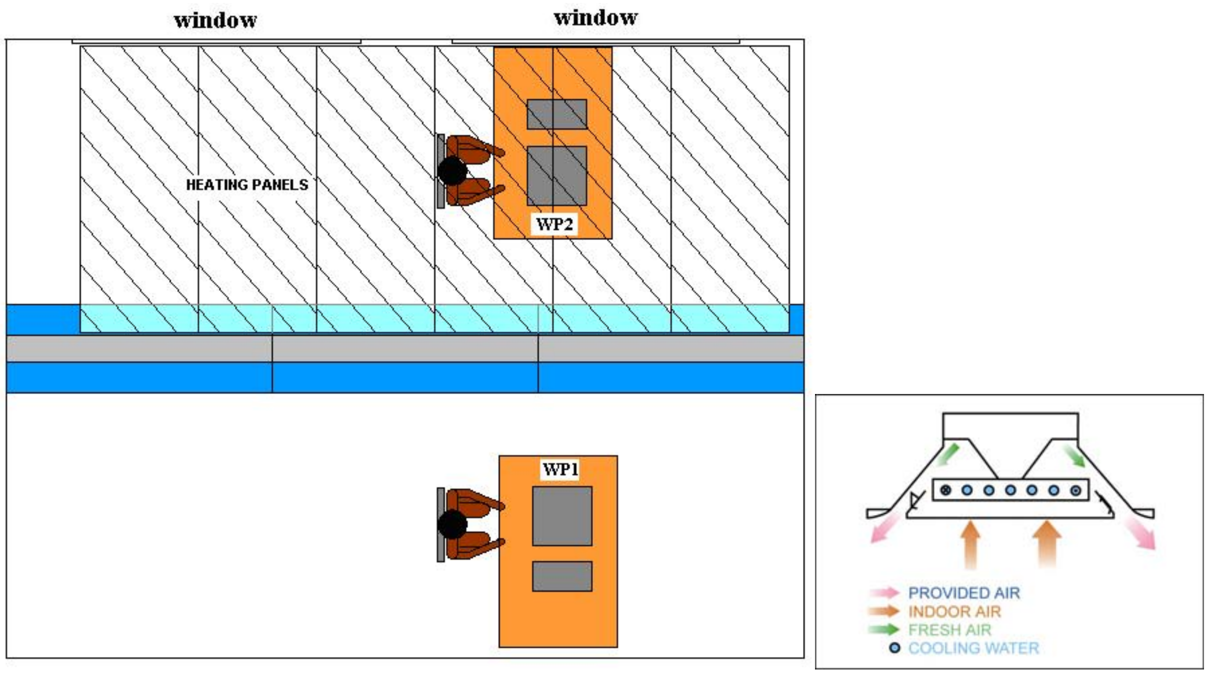

Figure 1. Experimental set-up in the room and sketch of cross section of the chilled beam with right shutter closed and the left shutter open is shown as example.

During the experiments the flow rate of the primary air was 1.5l/s/m². The experiments represented summer conditions and the room temperature was maintained to be 24ºC.

The distribution of the heat loads in the test room for each case is listed in Table 2. The window surface temperature of 40.4 ºC is high but possible in practice.

Table 2. Distribution of the heat loads

Case number	Heat gain from windows, W	Heat gain from PCs, W	Heat gain from people, W	Heat gains from floor panels, W	Heat gain from lightning, W
Case1	2x325	2x80	2x75	520	116
Case2	2x325	2x80	2x75	520	116
Case3	2x220	-	2x75	-	116

Subjects

Thirty subjects, 15 males and 15 females, participated in the experiment. They were at age from 20 to 45 years (average 24.3 ± 5 years).

Experimental procedure

Each subject participated in three experiments (Cases 1, 2 and 3 in Table 1). At their arrival, before entering the experimental office, the subjects relax for 20 minutes from previous activities. During that time they were explained the experimental procedure and how to fill in the questionnaires. They were not informed about the experimental conditions, e.g. the temperature in the room, the ventilation system, the flow rate, etc. After 20 minutes the subjects entered the test room two at a time and sat at the workplaces. They were asked to fill in the questionnaires every 5 minutes staring from the fifth minute. The subjects were allowed to write or read while seated on the desk. The subjects stayed in the office 25 minutes then they went out for 10 minutes break. After the break they returned to the office and stayed there for another 25 min. This time they changed the workplaces. They completed questionnaires as during the first 25 min. During the whole experiment the subjects were encouraged to adjust their clothing according to their preferences. At the end of the experiment for the day they filled in a questionnaire about their clothing. This procedure was identical for the three experiments the subjects participated.

Questionnaires

The subjective response to the thermal environment was collected by means of questionnaires. They were presented to the subjects on a paper. The questions related to whole body and local thermal sensation - 7-point ASHRAE scale [6], acceptability of the thermal environment (ranging from clearly unacceptable to just unacceptable and then from just acceptable to clearly acceptable, air movement sensation, body part where the movement was felt, whether it was acceptable or not, and the last question was on preference for more, less or unchanged air movement.

Data analyses

The data obtained from the questionnaires was analyzed statistically using the program Statistica 5. Each of the samples was tested for normality of the distribution using the Shapiro-Wilk test. Tests for significant differences of the samples were performed with Wilcokson match pair test.

RESULTS AND DISCUSSION

The individual whole-body thermal sensation votes as reported by the subjects were used to define the average thermal sensation vote. Figure 2 compares the obtained results for each of the three experiments. The average thermal sensation of the occupants in all of the cases was within the comfortable range of ±0.5 (from slightly cool to slightly warm) recommended in the present standards [6, 7]. Only at WP2 in Case 2- 70W/m²- I2 the thermal sensation was above slightly warm, 0.7. In this case the thermal sensation of the occupants was influenced by the thermal radiation from the warm windows which were located 0.9 m from WP2. The thermal sensation was just below neutral only at WP1 in the experiment with 30W/m² (Case 3).

The results were statistically checked for significance. Significant differences were found in the average thermal sensation between WP1 and WP2 in Case 2-70W/m²- I2 ($p=0.006$), at WP2 between Case 1- 70W/m²-I1 and Case 2- 70W/m²-I2 ($p=0.0262$) and at WP2 between Case1-70W/m²-I1 and Case3-30W/m² ($p=0.039$). The subjects felt warmer the whole body with the conditions at WP2 in Case 2-70W/m²- I2.

The local thermal sensation for the different body parts as reported by the subjects was analyzed. The local thermal discomfort due to warmth was defined for the body parts with local thermal sensation “slightly warm”, “warm” or “hot” which was felt as uncomfortable. Figure 3 shows the results of these analyses.

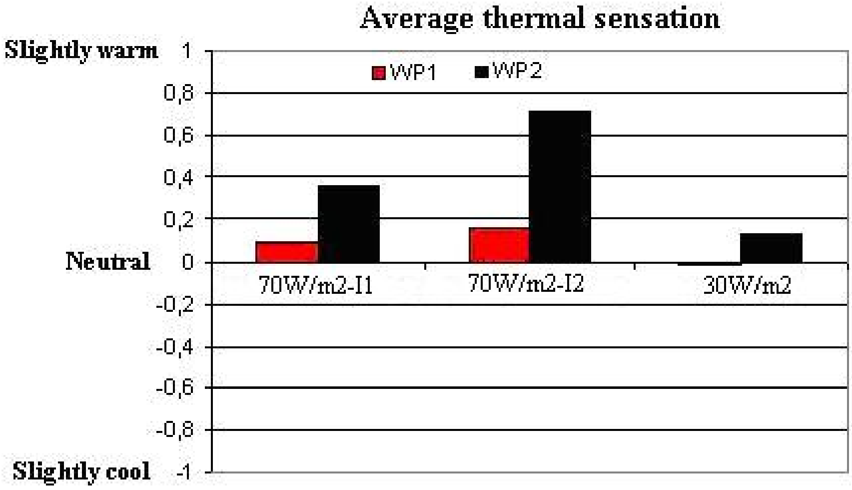

Figure 2. Average thermal sensation of the subjects at workplace 1 (WP1) and workplace 2 (WP2).

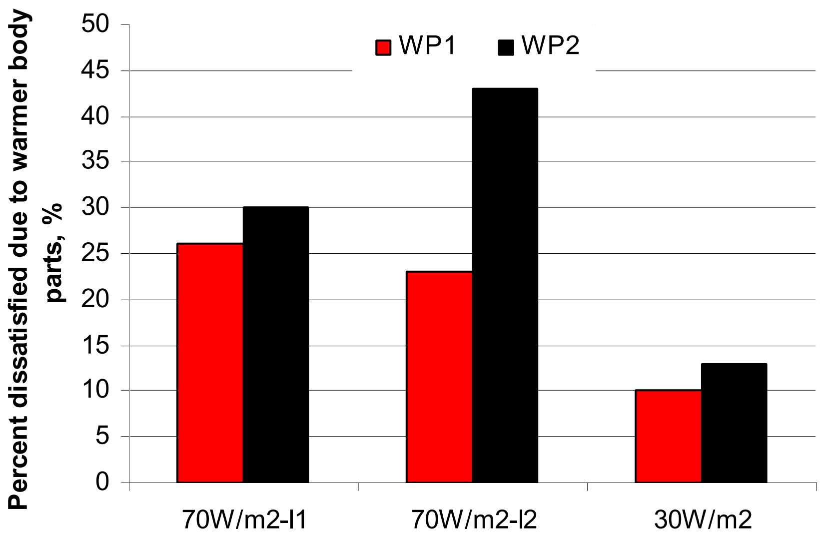

Figure 3. Percent dissatisfied due to warmer body parts

The highest percent of subjects who felt local warm discomfort for one or more body parts was for Case 2 -70W/m²- I2. The conditions at WP2 in Case 2 -70W/m²- I2 were identified as the worst; almost half of the assessment panel (43%) felt discomfort at that workplace located close to the heated window. The local warmth discomfort decreased when the heat load in the room decreased (Case 3- 30W/m²).

During the experiments the subjects were asked to assess whether they felt air movement at any of the specified body parts and if so, if the air movement was comfortable or not. The subjects’ responses were defined as draught discomfort if: 1) the person felt particular body part cooler than neutral and 2) if the person felt the local cooling uncomfortable; and 3) if the person felt air movement and voted that the feeling is uncomfortable and 4) the subject requested for “less air movement”. Figure 4 represents percentage dissatisfied due to draught at any body part.

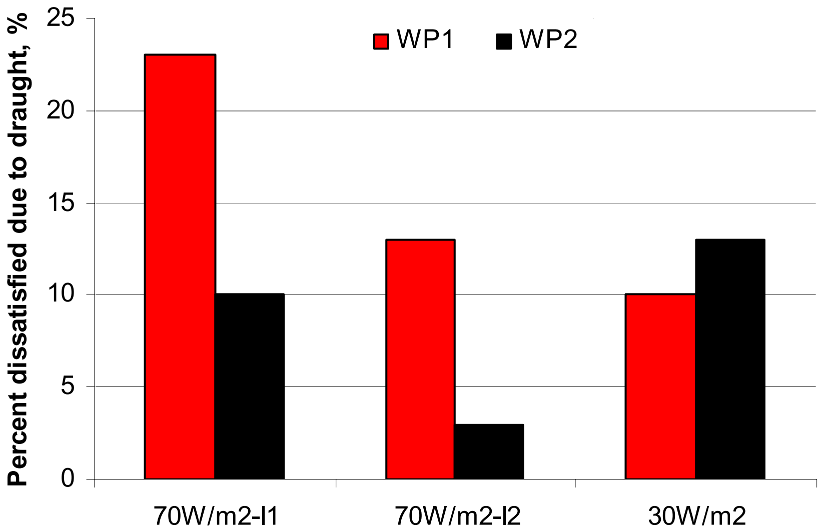

Figure 4. Percent dissatisfied due to draught based on all four criteria

The highest percent of subjects who felt draught was reported in Case 1 - 70W/m²- I1 at WP1. When the induction control was used, Case 2-70W/m²-I2, the percent dissatisfied due to draught subjects at WP1 was lowered substantially; in this case only one person reported on draught. The results show that with the lowering of the strength of the heat sources (Case 3 - 30W/m²) the total percent of draught complaints decreased.

DISCUSSION

Acceptable thermal environment for the whole body was achieved under all the tested conditions. The average thermal sensation of the subjects achieved with the chilled beams at the two workplaces was between neutral and slightly warm. As expected due to the radiant temperature asymmetry generated by the warm window the subjects felt warmer at the WP2 than at WP1. The flow interaction as discussed in the following left the subjects at WP2 with less air movement for cooling their body. However this did not affect subjects’ general thermal sensation.

The general thermal sensation, the local thermal sensation and the draught discomfort reported by the subjects was result of the interaction of the thermal flows generated by heat sources and the ventilation flow supplied from the chilled beams. The convection flows from the window, the solar irradiation simulated on the floor, the person and the PC concentrated on one side of the window (WP2) in Case 1 (70W/m²- I1) were powerful enough to deflect the ventilation flow toward the opposite side of the room as shown schematically on Figure 5. This resulted in large number of draught complains at WP1 (Figure 4). The use the induction control of the chilled beams on the side of the WP1 (Case 2-70W/m²-I2), decreased the velocity which resulted in decreased draught complaints at WP1 as well as at WP2. The local thermal discomfort decreased as well (Figure 3). Thus the efficient use of the induction control for decrease of draught discomfort in rooms with chilled beams was validated.

The activated induction control decreased the draught complaints but the number of subjects reporting warmth discomfort at WP2 increased from 10 to 13, i.e. near the window. Due to the airflow interaction less cooling air reached the subjects at WP2 and this resulted in warmer thermal sensation. Thus the warmth discomfort should be carefully considered together with draught discomfort in practice in rooms with chilled beams when induction control is provided.

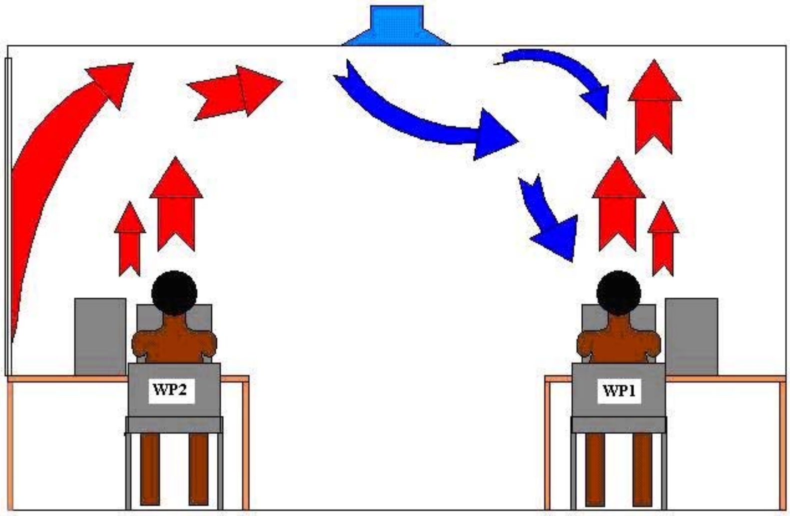

Figure 5. Air distribution in Case 1 (70W/m²- I2)

The importance of the airflow interaction was demonstrated in the experiments with different heat load as well. Non-uniformity of the thermal environment decreased when the heat load was decreased from 70W/m² (Case 1) to 30W/m² (Case 3) and this affected occupants' thermal comfort. Draught discomfort decreased (Figure 4), the whole body thermal sensation improved (Figure 2) and the reports on uncomfortable local warmth discomfort decreased (Figure 3).

The results reported in this paper were supported by results from comprehensive physical measurements on air distribution and thermal environment in rooms with chilled beams [3, 4]. They confirm the importance of careful consideration of flow interaction in rooms in general. In future buildings occupants and solar irradiation will remain the major heat sources. The increase of the outdoor temperature due to global warming and the use of glass facades will affect the airflow interaction in spaces resulting in increased non-uniformity of the thermal environment. At the same time the recognized impact of the indoor environment on occupants' performance [9] will lead to requirements for higher quality of indoor environment in the future standards than the requirements specified in the present indoor climate standards. Therefore the need for of individually controlled environment in buildings which will be able to compensate for the non-uniformity of the indoor environment and for the large individual differences existing between people in regard to the preferred environment will increase.

CONCLUSIONS

For the conditions provided by the chilled beams the average thermal sensation of the subjects was within the requirements of the existing standards and guidelines (ISO 7730, CEN 1752, REHVA Guidebook).

The flow interaction in the room under the studied conditions had major impact of occupants draught sensation. Higher draught discomfort was discovered for the conditions with higher heat load in the room. The induction control proved to be efficient way for diminishing draught discomfort.

ACKNOWLEDGEMENT

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Consideration for minimising draught discomfort in Rooms with Active Chilled Beams

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SUMMARY

Active chilled beams are an air conditioning solution for ventilation, cooling and heating. By careful design the solution can result in a comfortable indoor environment with a high degree of individual control at a low operational cost. During the design process draught risks should be analyzed since many factors can influence the resulting airflow pattern from chilled beams. The paper focuses on the factors affecting the thermal comfort in a room ventilated by chilled beams. Both experimental results as well as theoretical considerations are included in the analysis. Practical guidelines for minimizing draught risk are given and features of active chilled beams are described in the paper.

INTRODUCTION

Active chilled beams is the complete indoor environmental solution. Maintaining both a comfortable thermal as well as atmospheric indoor climate by handling both ventilation, cooling and heating using only a minimum of energy it is a solution which should be considered for all new or refurbished buildings. The solution offers high temperature cooling and low temperature heating resulting in high energy performance as well as only the amount of fresh needed for maintaining a comfortable atmospheric indoor climate. Additionally the solution offers small space requirements for ducting and good individual control of the indoor environment.
The working principle of an active chilled beam is shown in figure 1.

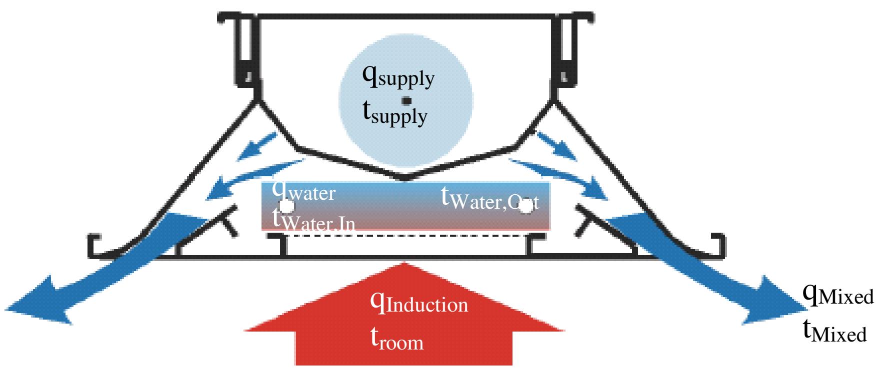

Figure 1 Working principle of active chilled beam.

The supply air is dispersed at a high velocity through a number of nozzles placed inside the chilled beam. The jets from these nozzles creates a withdrawal of the air in the vicinity of these small jets and subsequently creating an airflow through the coil of the chilled beam ($q_{induction}$). This induced air flow is cooled when passing the chilled water coil and supplied to the room mixed together with the primary (outdoor) air. The temperature of the air supplied to the room is dependent on the room air temperature, the primary air temperature and the cooling of the coil. The cooling output is controlled by adjustment of the water flow through the coil. This beam works similarly in heating mode.

A chilled beam system delivers only the necessary amount of primary (outdoor) air in order to obtain a good atmospheric indoor climate. The thermal indoor environment is maintained as described above.

Since the first active chilled beams were introduced in the 1980s the development has been extensive – increasing cooling outputs and higher supply air flow rates, but also additional features such as heating, adjustment of velocity as well as build-in service features have been added subsequently up until today. The increased output of cause increase the risk of draught subsequently increasing the need for good design of systems.

The paper focuses on the factors affecting the thermal comfort in a room ventilated by chilled beams.
As illustrated in the following figure 2 a number of different factors influence the flow pattern as well as the resulting thermal comfort in a room ventilated and cooled or heated using chilled beams. The dimensions of the room, the location of the chilled beams, the number, size and location of heat sources, the primary and secondary airflow rates from the chilled beam, but also possible ceiling mounted obstacles can influence the airflow from the chilled beam.

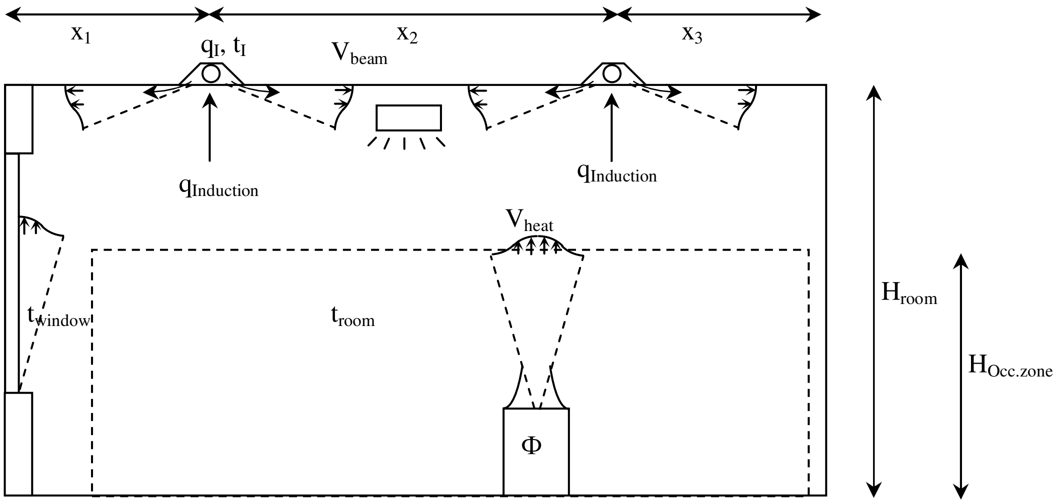

Figure 2 Factors influencing flow pattern and thermal comfort in room ventilated by chilled beams.

The present paper focuses on the practical design of active chilled beam systems and states certain guidelines in connection to the design of chilled beam systems. Experimental data are used for analyzes of theoretical relationships describing the airflow and for outlining practical considerations.

METHODS

All measurements were conducted using active chilled beams integrated in a suspended ceiling placed in a full scale office room. The chilled beams used were 1800 mm long with a coil length of 1500 mm. The measurements were conducted in a room representing an office room. The room had dimensions 5.4 x 4.2 x 2.5 m. The chilled beams were all equipped with a device (*Halton Velocity Control*) which can limit the slot opening of the beams and hereby reduce the induced airflow rate.

Different factors influential for the flow interaction in the room were analyzed through the measurements including placement of chilled beams in rooms (3 or 4 chilled beams), different strengths of heat loads and locations (heated surfaces of window and floor simulating solar radiation, heat generated by occupants (1 or 2 sitting persons in front of computer were simulated), primary air flow rate and induced airflow rate. Both temperatures and velocities were measured in a vast number of points in the room in the experiments. Further information concerning these measurements can be found in [1].

RESULTS

When designing chilled beam systems it is of great importance to optimize the thermal comfort in the room not only by matching the needed cooling capacity, but also to avoid possible draught risks that otherwise could occur due to the airflow from the chilled beams. It is however not easy to predict the complete picture, since the location of heat sources may change many times during the lifetime of the building as well as location and effect of flow diverting obstacles may be difficult to estimate. However there are a number of factors which can be accounted for and which should always be taken into consideration when designing chilled beams systems.

The height of the room will have an effect on the flow pattern in the room since the air from the chilled beam has a lower temperature than the ambient, and the air acceleration due to the temperature difference can cause draught risks. The recommended maximum height of rooms equipped with chilled beams is approximately 3.5 meters, above this value the thermal forces can cause draught risk in the occupied zone if the cooling need is high.

Due to the air flow from active chilled beams being dispersed from slotlike diffusors the velocity decay can be described as a plane wall jet (when attached to ceiling and also when diverted down a wall). The velocity decay for a plane wall jet can be shown theoretically to being inverse proportional to the square root of the distance [2]. This means that the velocity is maintained much better than a normal radial diffuser ($v \sim 1/x$), and that the temperature decay of the jet is small and can cause draught risks when the jet is directed downwards, either due to meeting a jet generated by a neighboring chilled beam or due to a wall or an obstacle.

Despite the fact that the primary supply air flow rate is kept at a minimum the total recirculated airflow in the room may be quite large. This is also an important matter limiting the installation height due to the undercooled air from the chilled beams.

The following figure shows velocity and temperature distribution measured in the test room with chilled beams installed in a lengthwise positioning compared to the window façade. The depictions show a measured example of a bad positioning of the chilled beams which clearly results in some flow interaction between the heat sources and the flow from the chilled beams. The flow from the chilled beam is diverted away from the ceiling immediately after leaving the chilled beam and subsequently dropping straight into the occupied zone. It can be seen from the figure that this also results in a very uneven temperature distribution.

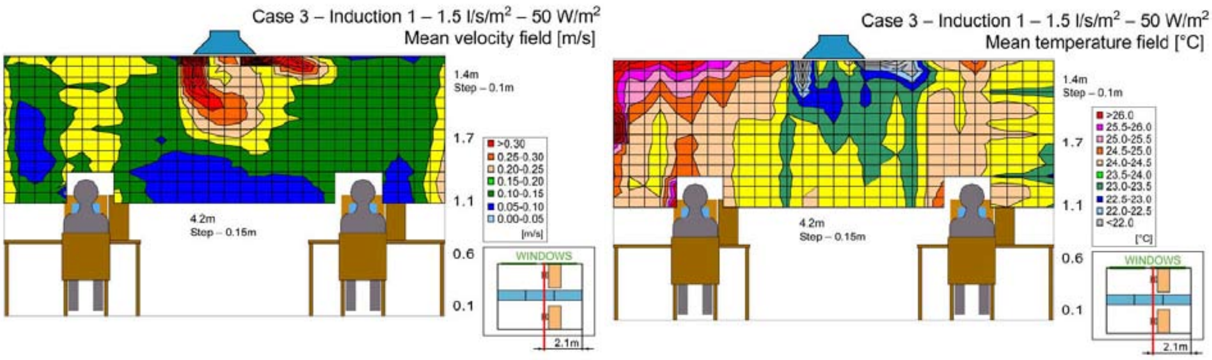

Figure 3 Measured velocity (left) and temperature distribution (right) in an office room with two occupants and a lengthwise chilled beam installation. Heat load strength and distribution and the primary airflow rate are defined. Induction 1 means maximum induced room air. . The measurement plan in shown with red line on the sketch in the figure [1].

Possible flow interaction should be taking into account when designing chilled beam systems (to the extent possible) and the depictions above gives a clear indication that chilled beams installed in a lengthwise position can cause unnecessary draught risks.

As already discussed the distance between chilled beams and the distance to the nearest wall are also matters of great importance for the velocity in the occupied zone, i.e. for the draught discomfort. Opposing beams can have impinging jet flows which have a resultant flow directed downwards and beams close to walls will have a resulting flow downwards. Here the earlier mentioned velocity decay can be used to estimate the necessary distance between the beams and the distance to the nearest wall, it is however necessary to know the constant (K value) describing the velocity decay:

$$\frac{v_x}{v_0} = K_P \sqrt{\frac{h_0}{x}} \qquad (1)$$

The K value is dependent on the specific slot of the beam, and the internal nozzles in the beam.
Assuming that the thermal forces are small compared to the initial forces the equation must then be solved for $v_x = 0.2$ m/s or similar demand and the necessary distance x can then be determined as from half the distance between the beams (or the distance to the wall) plus the vertical distance from the ceiling to the occupied zone. This assumption can be checked if horizontal distance here is larger than the penetration length which can be estimated by the following expression [3]:

$$x_s = 1{,}58 \cdot K_P^2 \cdot \left(\frac{v_0^2}{\Delta T_0}\right)^{2/3} \cdot h_0^{1/3} \qquad (2)$$

The following figure shows the velocity distribution of two opposing beams which clearly shows the two airflows being joined and directed downwards.

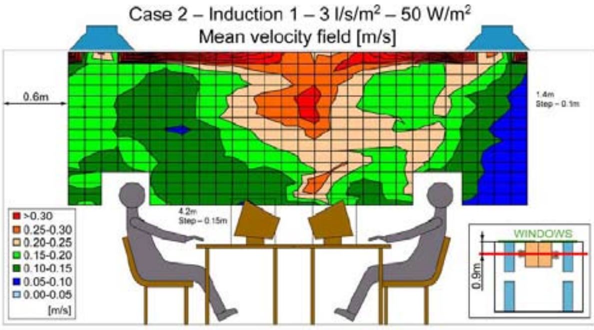

Figure 4 Measured velocity distribution of two opposing beams with joining airflows at 3 l/s/m² (primary air) and 50 W/m². Induction 1 means maximum induced room air. The measurement plan is shown with red line on the sketch in the figure [1].

The importance of the heat load strength for the velocity and temperature distribution in rooms with chilled beams is shown in Figure 5. The depictions show the same general airflow distribution with high velocities near the beams and close to the concentrated heat sources. However the general velocity level around the persons is higher when the cooling is 80 W/m² compared to 50 W/m². The higher velocity level is caused by the larger difference in the temperature level between the airflow from the beams and the ambient as well as the larger heat load from the floor.

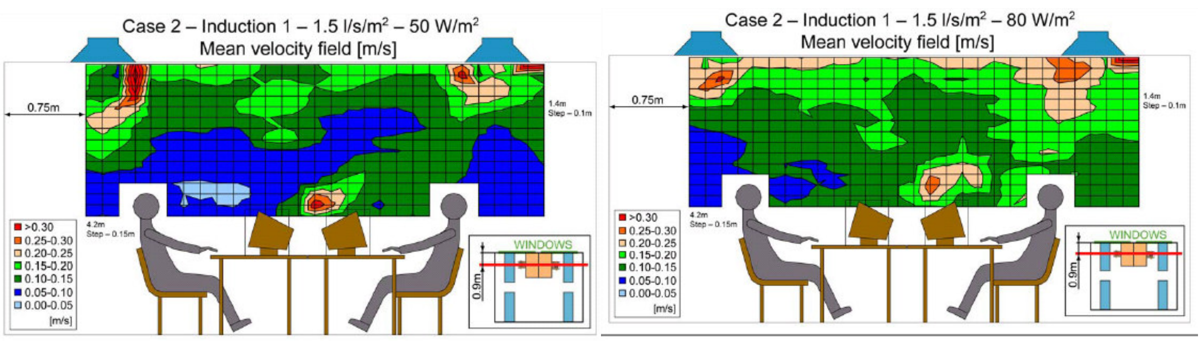

Figure 5 Measured velocity distribution at two different heat loads. Heat load strength and distribution and the primary airflow rate are defined. Induction 1 means maximum induced room air. The measurement plan is shown with red line on the sketch in the figure [1].

Comparing the left depiction in Figure 5 with the measurements shown in Figure 4 it is obvious that a higher air flow rate causes a higher general velocity level and thereby a higher draught risk.

Another way of changing the velocity distribution or limiting the velocity level in the room is using a device that limits the slot opening which in turn limits the induced air flow.
The following Figure 6 shows the measured velocity distribution for two different positions of the velocity control; fully open shown in the left depiction and limited in the right. The figure shows a higher velocity level when the velocity control is fully open, due to the higher total amount of recirculated air. On the other hand the figure also shows that the airflow detach from the ceiling when the slot opening is limited. This device which has 3 positions has

proven very useful in many office buildings since it is possible to change the local velocity level when unforeseen matters such as local heat sources and flow diverting obstacles change the flow from the chilled beams. Designing the chilled beam systems when using the middle position of the velocity control gives the possibility of adjusting the local draught risks.

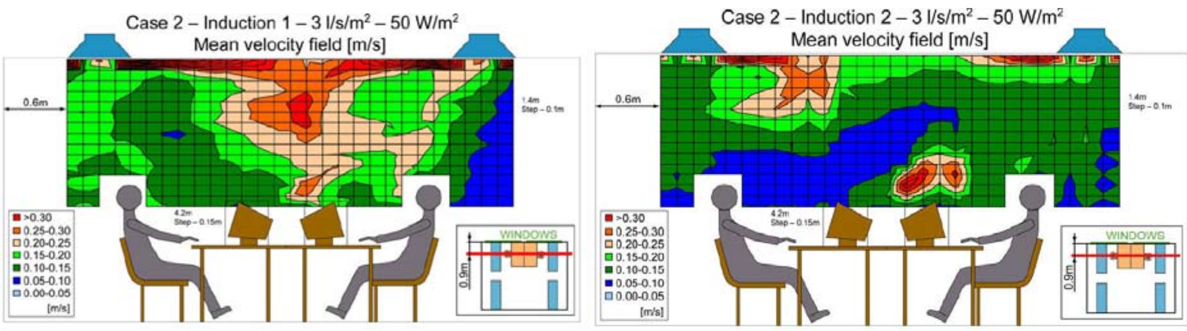

Figure 6 Measured velocity distribution at two different positions of the velocity control (device reducing the slot opening). Induction 1 means maximum induced room air, while Induction 2 means reduced induction of room air. Heat load strength and distribution and the primary airflow rate are defined. The measurement plan is shown with red line on the sketch in the figure [1].

DISCUSSION

Active chilled beams offer good indoor climate conditions and high level of flexibility in the office. It is a solution that can offer a comfortable indoor environment with a high degree of individual control at a low operational cost. However active chilled beam systems requires careful design since many factors influence the airflow in the room and thereby the possible draught risks.

Chilled beams should be installed perpendicular to the window façade thus avoiding flow interaction between the heated surface and the flow from the chilled beams. If other considerable heat sources are known these should also be taken into account.
Chilled beams should not be used in rooms with a room heights above 3.5 m since this will increase the draught risk in the occupied zone significantly.
The higher the necessary cooling capacity or the higher the outdoor airflow rate the higher the general velocity level which may result in unnecessary draught risks.

Further guidelines for design of chilled beam systems can be found in [4] as well as documentation provided by manufacturers.

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High Quality Thermal Environment by Chilled Ceiling in Office Building

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SUMMARY

An air-condition system shall be designed to provide the required indoor environment under specified condition. According CEN Report 1752 it could be distinguish three categories for indoor environment. Considering higher level of comfort there is a rising cost of HVAC systems. In hereby published research main parameters of thermal comfort and IAQ in offices have been measured such as PMV and PPD indexes, indoor air velocity (w), percentage of turbulence intensity (TU), DR index, and carbon-dioxide concentration.

Results are seem to be proved that in order to provide the highest comfort category there is a need of more expensive HVAC systems.

INTRODUCTION

The examined office building, located in downtown Budapest, is equipped by BARCOL CBA chilled ceiling system (cc. 4000 m^2), supported by radiators to provide the required comfort level in heating season.

During working hours the building was occupied by 687 office workers. A central ventilation systems is cover the required amount of fresh air for them (68,3 m^3/h,person). 75-85% of total area of chilled ceiling is made up suspended ceiling panels in size 600x600 mm. In some area with significant heat loads it is additionally placed further passive cooling coils above the suspended ceiling. Otherwise rooms with slightly significant heat load, 45-55% of the ceiling area is covered by cooling panels. Designed cooling inlet/outlet water temperature level is 15/17°C. However higher water temperature level (17/19^{o}C) have found sufficient to provide proper thermal comfort in summer. In this case condensation could not be an issue.

In the building windows are free to open, although sensors are installed to close the cooled water valves in case of airing.

METHODS

In field studies following parameters have been investigated: air temperature (t_a), wet globe temperature (t_w) mean radiant temperature (t_{mr}), air velocity (w), PMV and PPD indexes, carbon-dioxide concentration (c_{CO2}), cooled water inlet (t_{wi}) and outlet (t_{wo}) temperature. Schematic diagram of chilled ceiling systems, measuring equipment and self –control system can be seen in Figure 1.

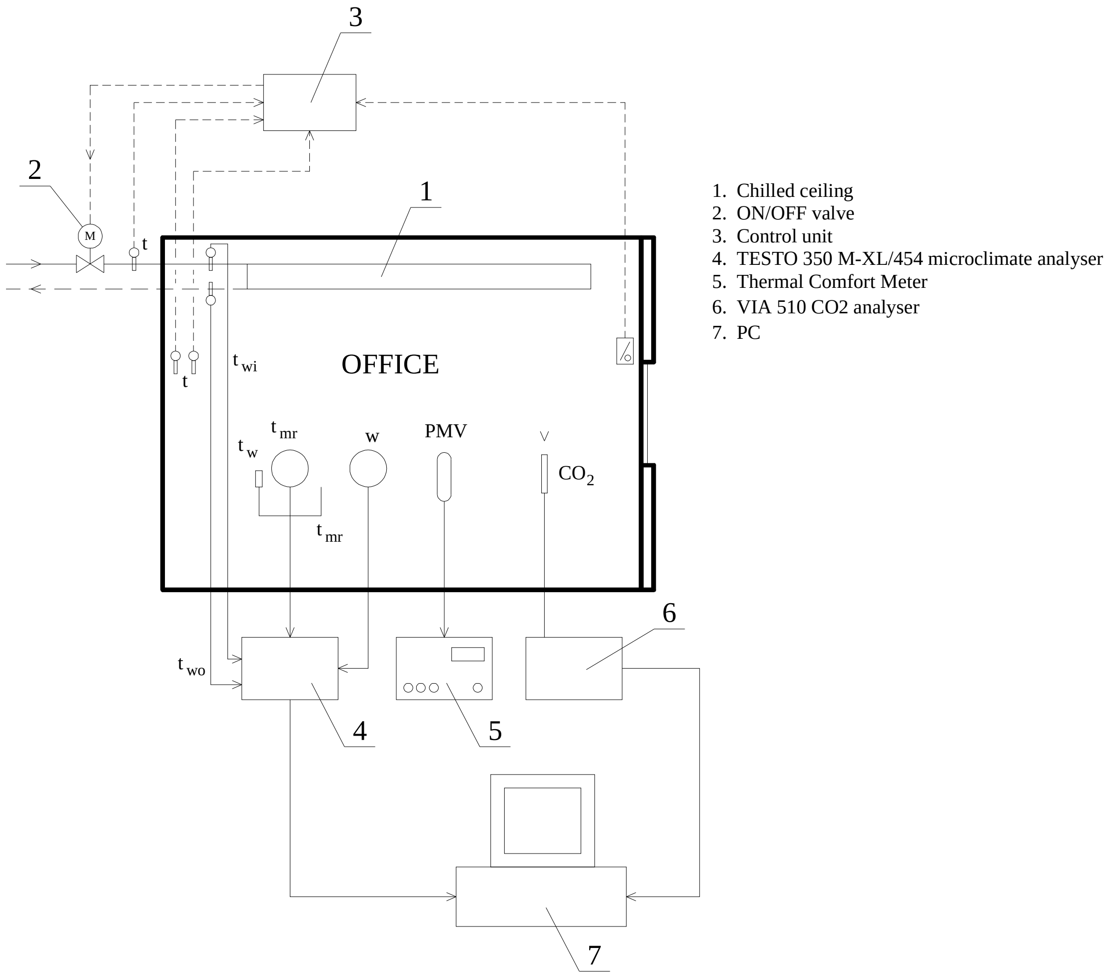


Figure 1. Schematic diagram of chilled ceiling systems, measuring equipment and self – control system

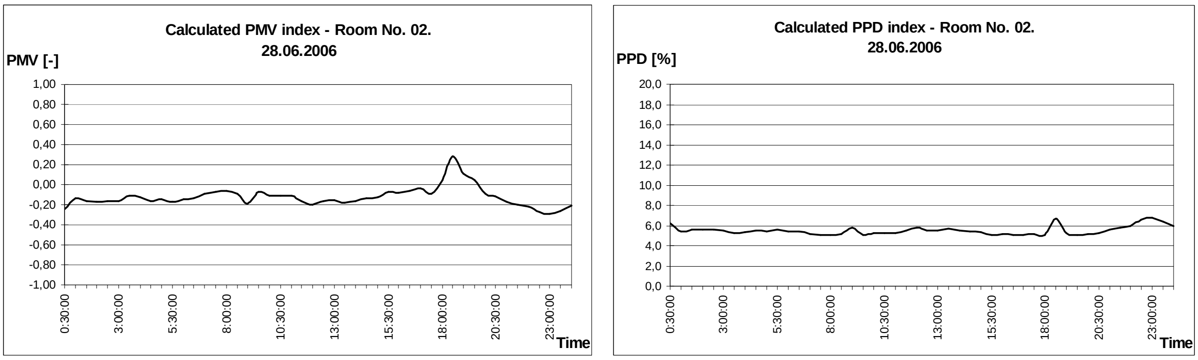
Air velocity values have been collected in every 12 second; other parameters have been measured in 5 minutes. Due to automatic measuring system two week long continuous examination and data collection have been carried out in summer and intermediate season as well.

By evaluating thermal comfort following parameters have been taken into account: activity level 1 met (light sedentary office work); thermal capacity of clothing 1 clo.

RESULTS

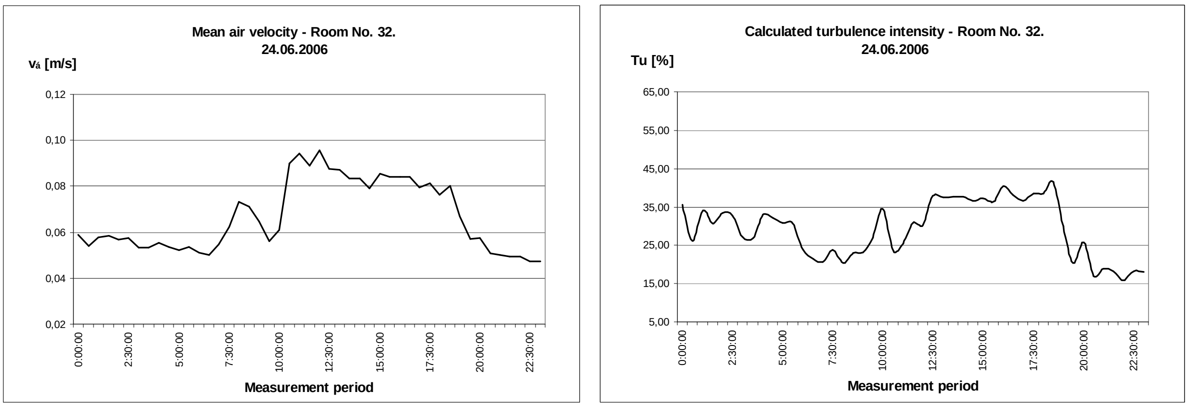
Results have been collected and saved using computer software. PMV index has been measured by Thermal Comfort Meter and also it has been calculated based on additionally measured thermal comfort parameters. Air velocity data has been as a base at the calculation of turbulence intensity and DR index, using computer software, developed by our research team. Considered measuring period to averaging has been 3 minutes.

In Figure 2. show variation of PMV and PPD indexes in an average summer day. Calculated mean air velocity and turbulence intensity can be seen in Figure 3. Percentage of dissatisfaction due to draught can be seen in Figure 4.



a) b)

Figure 2. Variation of a) PMV and b) PPD indexes in an average summer day



a) b)

Figure 3. Calculated a) mean air velocity and b) turbulence intensity

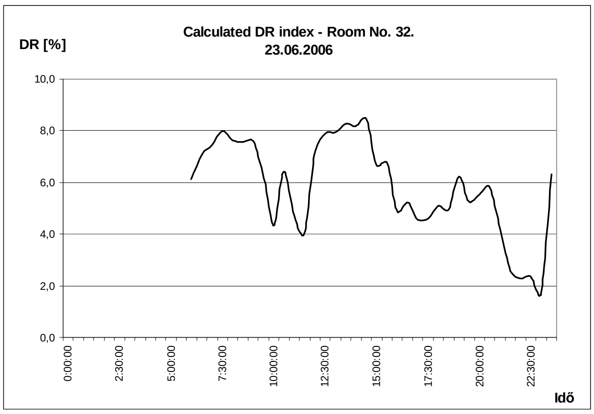
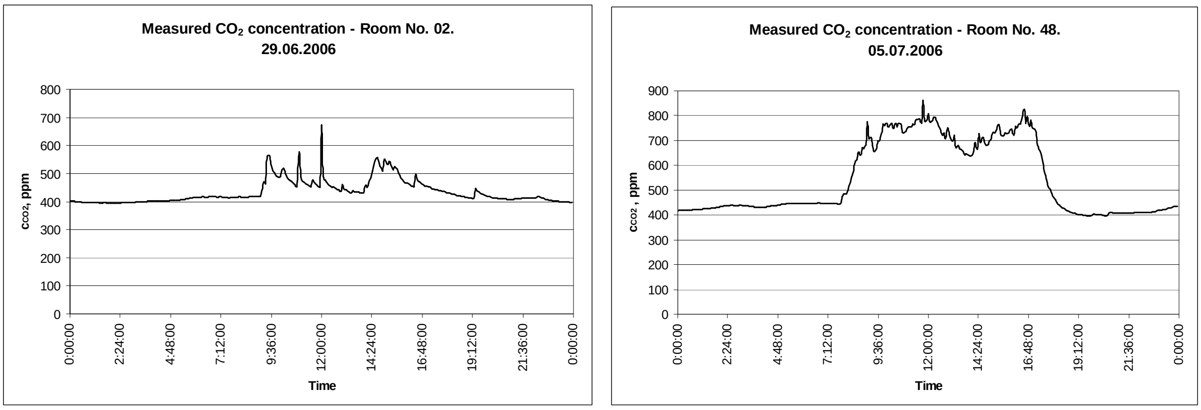


Figure 4. Percentage of dissatisfaction due to draught

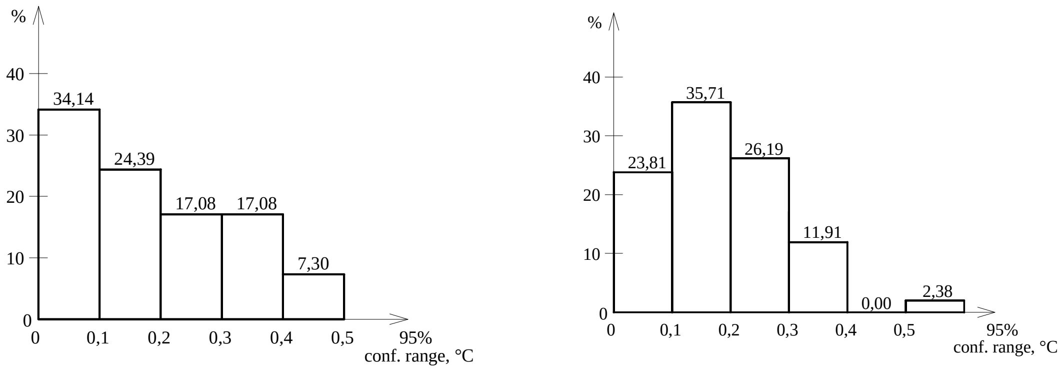
Carbon-dioxide concentration has been also measured in singular and landscape offices too. Results can be seen in Figures 5.



a) b)

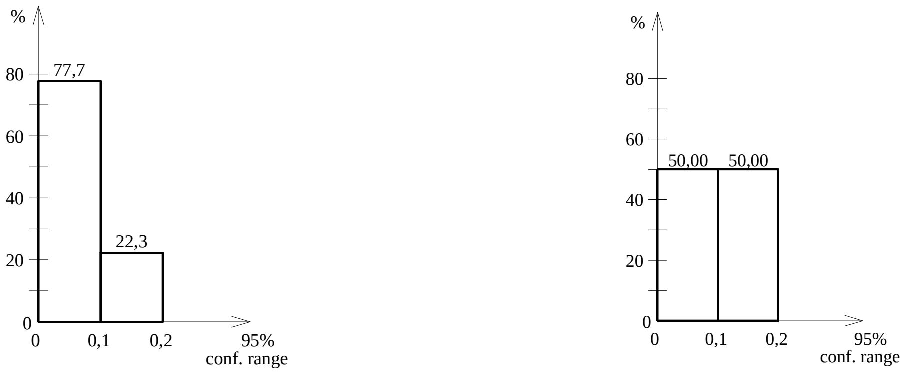
Figure 5. Carbon-dioxide concentration in a) singular and b) landscape office

Mathematical evaluation of results has been carried out according principles of probability theory. Thus distributions of measured and calculated data are presented graphically as well. Histograms are shown in Figure 6. (air temperature) and in Figure 7. (PMV index, for intermediate season and in case of designed outdoor temperature in summer).



a) b)

Figure 6. Histograms of air temperature in a) intermediate season and b) case of designed outdoor temperature in summer



a) b)

Figure 7. Histograms of PMV index in a) intermediate season and b) case of designed outdoor temperature in summer

DISCUSSION

To draw conclusions guideline CR 1752 has been taken into consideration. These are as follows:

- In intermediate season maximum air temperature difference is lower than 0,5 K. (Category “A”: Δt_{max} = 1K)
  Under design condition (summer) maximum air temperature difference is lower than 0,6 K as well. (Category “A”: Δt_{max} = 1K)

- Variation of PMV is lower than 0,3 in intermediate and summer season as well. (Category “A”: ΔPMV_{max} = 0,4)

- Percentage of dissatisfaction caused by draught (DR index) is lower than recommended value according category “A”. Maximum indoor air velocity is 0,12 m/s, maximum DR value is 10,61% (Category “A”: w_{max} < 0,18 m/s, DR < 15%).

- Average outdoor carbon-dioxide concentration is 440 ppm, thus to meet requirements for category “A” concentration should be lower than 900 ppm.

According measuring results indoor comfort level is “A” in the investigated office space.

Indoor comfort guidelines described in CR 1752 gives three different levels (A, B and C) to classify an enclosure. In Hungary it has been an issue the opportunity and way of providing indoor comfort at level “A”. Due to strict requirements, there is a need of design and maintenance at high level of accuracy.

New office buildings are typically designed with significant glassed area and decreasing heat capacity. Fluctuations of indoor / outdoor loads make automatic control system crucial. Further consideration has to be made in accordance of required cooling capacity of chilled ceiling systems. Chilled ceiling is one of the most complicated HVAC system. To avoid harmful condensation on cooling panel proper design is essential. Atypical water temperature ranges (e.g. 15/17 °C) give a further aspect of complicity of system design. Higher water temperature level (17/19^{o}C) have found sufficient to provide proper thermal comfort in summer. In this case condensation could not be an issue.

Based on our research, using chilled ceiling system it is possible to provide indoor comfort at level “A”. Investment cost is higher, although thermal comfort parameters are meet standard. Moreover there is a decreasing level of dissatisfaction. Measured PMV index has been slightly lower than neutral value (Figure 2.). Therefore thermal comfort can be said slightly cool. Given value of PMV can be controlled using automatic control system. In summer there is a common problem with relative high indoor air temperature in office buildings. However it has not to be proved in this case, thanks to proper design, implementation, hydraulic balancing and maintenance of HVAC system.

ACNOWLEDGEMENT

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Measurement of Flow Characteristics of a Ceiling Fan with Varying Rotational Speed

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SUMMARY

This paper conducts an experimental measuring of the airflow generated by a modern ceiling fan that has an electronic device to vary the instant rotation speed for simulating natural winds. The measuring results are then used to calculate the mean air velocity, standard deviations, turbulence intensity, equivalent frequency, power spectrum, and other comfort evaluation parameters. The results have shown that, by adopting the natural wind mode, even at a lower mean air velocity, a substantial comfort level can be achieved while consumes much less electric energy. In the consideration of achieving a sustainable built environment, this finding is of great importance. This study also completes a so-called *1/f* power spectrum analysis to identify the feature of natural wind generated. It seems that we can achieve environmental sustainability by further exploring the characteristics of natural environment.

INTRODUCTION

Ceiling fans are widespread used in hot or tropical climate zones for providing cooling and comfort. It consumes relatively low amount of energy in comparison to air-conditioning units. Although in these days, air-conditioning is preferable to operating a ceiling fan in modern office or residential buildings. Complementarily using ceiling fans as additional air circulating devices in air-conditioned rooms can often lead to the same comfort level at an elevated temperature setting, which means a potential saving of energy. Whereas, as ceiling fans are such common appliances nowadays, little effort has been paid on the new fan designs and its consequent influence on room airflow. This paper conducts an experimental measuring of the airflow produced by a modern ceiling fan which has an electronic device to vary the instant rotation speed to simulate natural winds.

The interest of this paper is to investigate the draught caused by a rotating ceiling fan because the space leans to have higher mean velocity. Therefore, thermal comfort considering only mean velocity and temperature as was done in a central air-conditioning system is not enough without considering draught. Draught is defined as an unwanted feeling due to locally convective cooling of the human body caused by air movement. In fact, the sensation of draught has direct relation to the airflow fluctuation, for examples, turbulence intensity and frequency of fluctuation. Several studies relevant to draught and airflow have been performed by Fanger and Christensen [1] and Fanger et al. [2]. In these studies, the percentage of people dissatisfied due to draught was defined by the following equation:

$$DR = (3.14 + 0.37\bar{v}Tu)(34 - t)(\bar{v} - 0.05)^{0.62}, \qquad (1)$$

where *DR* is the percentage of people dissatisfied, $\overline{v}$ mean velocity, *t* air temperature and *Tu* turbulence intensity. Recent study [3] shows that the frequency of air velocity has great impact on draught feeling. In that article, an equivalent frequency parameter was defined to describe the frequency characteristics of the airflow. It indicates that most people are sensitive to airflow at an equivalent frequency between 0.2 and 0.6 Hz. For spaces using a central air conditioning system, the air speed is relatively uniform. Draught can be easily inhibited in such a well-designed system. However, it is not a similar case for the space, using a unitary air conditioner, where persisting air speed might exist in occupied zone. Shih et al. [4] studied the draught induced by a split-type air-conditioner. Their results show that, for such a type of air conditioners, the equivalent frequency falls between 0.3Hz and 0.6Hz, which is in the uncomfortable range.

Recent research in personalized air conditioning systems with fluctuating air movement [5] and simulated natural wind [6] brought out a concept which is different from that of common systems with steady-state air distribution. The fluctuation of airflow is one main factor that affects indoor comfort. According to the research results [7, 8, 9], the fluctuating airflow brings more cooling effect to the human bodies, especially the airflow with a frequency similar to that of natural wind. The research result of [6] shows that the distribution of a constant airflow at the outlet of an air moving devise is quite different from that of a simulated natural wind. Only the airflow reaches its full stage of diffusion ($v < 0.25$m/s), does the structure of the airflow approach to that of natural wind. Nevertheless, the velocity of the air current is too low and has little effect to the thermal comfort.

Power spectrum analysis is an essential tool to be used to analyze the patterns of the airflow. It evaluates the relationship between the frequency of occurrence of the velocity and corresponding magnitude (or the energy) of the air current. The criterion to be used to make the judgment is the slope of the spectrum curve (normally in natural log of base 10). The mathematical relationship of power spectrum density to the slope of spectrum curve can be expressed as $E(f) = 1/f$, where f is the frequency of occurrence of the velocity. According to the finding of relevant literatures, the power spectrum distribution of natural wind is in $1/f$ fluctuation and its slope falls in the range of -1.10 to -1.67. As to the airflow produced by the mechanical devises, its slope of power spectrum curve is approach to the range of $1/f^0$ fluctuation (namely, the range of white noise).

A transient thermal environment can be established by changing the air temperature and/or the air velocity. Simulated natural air movement can be used to offset high air temperature and to improve perceived air quality slightly. A new conditioning strategy focuses on the effective use of simulated natural wind, which has special characteristics in turbulence intensity, velocity distribution and power spectrum. A prediction of energy use shows that substantial energy saving can be achieved by effective use of transient air movement.

In this paper, we focus our experiment upon the flow phenomena of a ceiling fan since such an air devise is believed to be able to achieve a certain degree of thermal comfort with energy saving. The results of the experiment are then studied and evaluated respectively for mean air velocity, velocity distribution, turbulence intensity, equivalent frequency and power spectrum etc., of the airflow produced. The corresponding indices reflect some well-known approaches for evaluating the comfort and thermal pleasance in a warmer environment and for preventing cold draught, in particular, in an air-conditioned space.

METHODS

The ceiling fan to be studied was installed in an open space indoor, as shown in Figure 1, providing airflow blowing down to the space below it. A hot wire anemometer made by TSI Co., Model IFA300, displayed in Figure 1(b) was used to measure the airflow characteristics. The instrument has the maximum scanning rate up to 1MHz. But, in this study, the scanning rate of the anemometer was set to be 10Hz. which can best demonstrate the *1/f* features and avoid unwanted noises. So the data of air speed was recorded once every 0.1 seconds. The measuring time lasted approximately for 6.8 minutes and 4096 data sets were recorded for every single measuring point. Longer data logging time was also tried, but the deviations of results analysis are less than 5%.

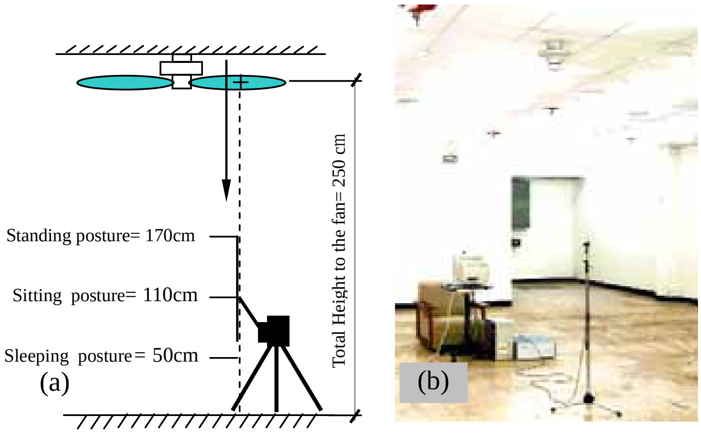


Figure 1. Layout of test site and some representative measurement heights

The speed of the air delivered is measured in a matrix consisted of several measuring points below the fan, as shown in Figure 2. A preliminary study for the fan speed is initiated where a series of measuring points are taken horizontally, 100 cm below the fan. Experimental setup shows that the ceiling fan is equipped with the fan blades in a spindle shape. The air distribution of the fan exhibits a unique pattern that the higher speed occurs somewhere below the center of the fan blade and the velocity declines as the measuring points are gradually away from the center, as Figure 2 shows. The measuring results are then used to calculate the mean air velocity, standard deviations, turbulence intensity, equivalent frequency, power spectrum, and other flow characteristic parameters.

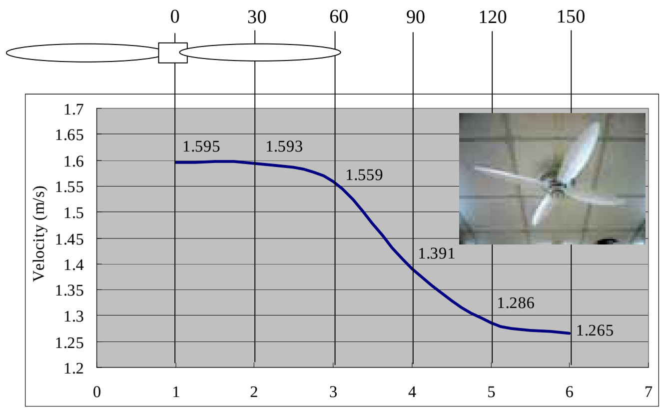


Figure 2. Profile of mean air velocity generated by the studied ceiling fan at height 150cm (fan was set at medium rotation speed)

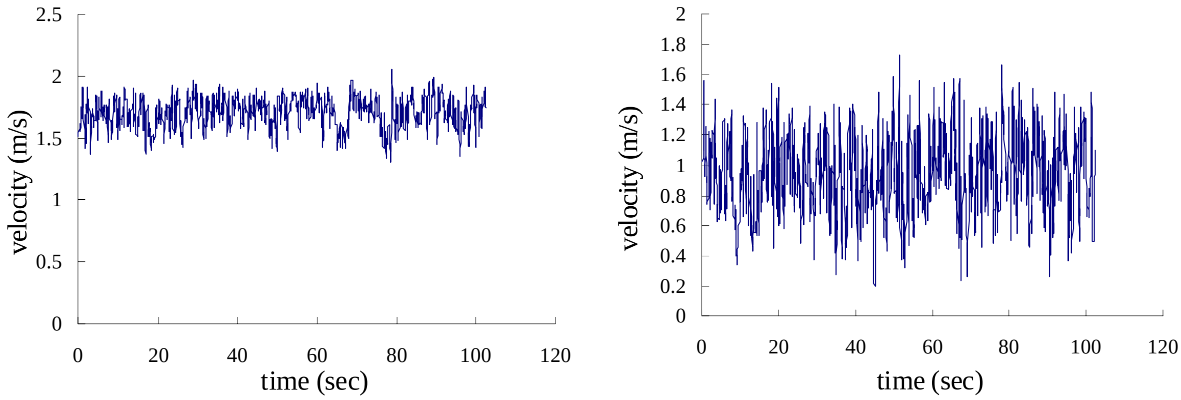
RESULTS

The ceiling fan studied in this paper, unlike the traditional one only operating at a constant rotation speed, can make fan blade subject to an instant speed change, with the aim to simulate nature breeze. It is claimed that the additional variable speed function can make occupants feel more comfortable in this indoor environment.

In order to find out the difference of air velocity variations for the ceiling fan operating at the constant speed mode and the variable speed mode, a measuring point located at the half of the blade length and 100 cm down from the fan blades was chosen to represent the flow characteristics in the occupied zone subject to direct air moving. The measuring results are shown in Figure 3-5 (a), correspondent to fan rotation in high, medium and low speeds. Basically, in the constant rotation speed mode, air flow measured at this location shows that the mean air velocities are around 1.4~1.7m/s, and the fluctuations are in the scopes of 0.2~0.3m/s.

In contrast, when the fan was operating in the variable speed mode, the air velocities measured, as shown in Figure 3-5(b) for different fan speed settings, all demonstrate the instantaneous velocity varying with time and show erratic change in fluctuation scale significantly larger than that of the constant speed cases. If we observe Figure 5(b) in more detail, the change of air velocity for the low-speed fan rotation condition was found with a specific fluctuation pattern. In some cases, the air speed measured even dropped to a very low speed to the extent about 0.2m/s, the condition lasting for about 20 seconds, and then the fan was quickly increased speed again. However, after observing the air velocity change for a longer time, it was found that the air speed pattern was not very repeated and the fan speed modulation could be explained as a random act.

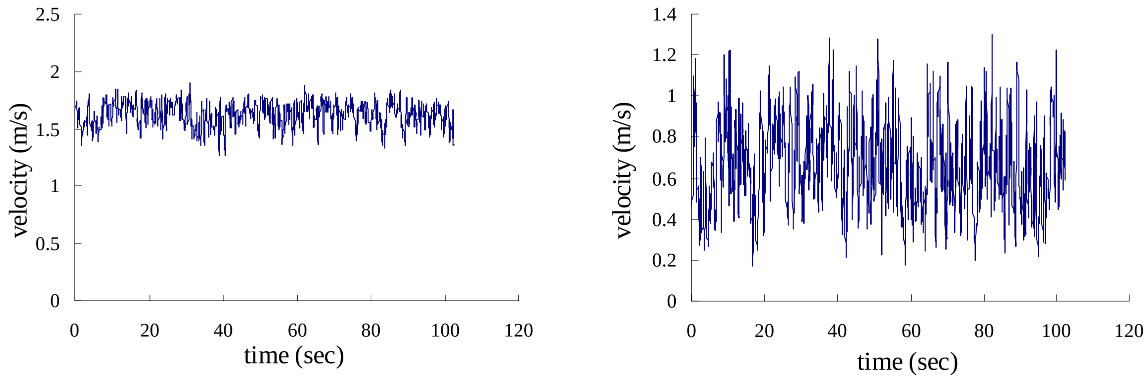
The air speed variation shown in Figure 3-5, by using some statistical analysis techniques, the data in time-domain can be converted to some flow characteristics in frequency-domain, and the results are shown in Table 1. Obviously, when the fan operating in natural wind mode, the mean air velocities are comparatively lower than those of constant speed mode, but the standard deviations of velocity and acceleration are roughly 2~3 times higher. The combined effect of a lower mean air velocity and a larger flow fluctuations results in a much stronger turbulence intensity for the natural wind cases. As indicated in Equation (1), turbulence intensity is the main cause, beside the temperature and speed differences, to introduce cold draught which has a direct impact on occupant’s feelings of comfort. In order to calculate the percentage of people dissatisfied with the draught. An average indoor temperature of 28^{o}C was chosen for the places using the tested ceiling fan in which the room temperature can be kept at a higher level than the normal setting in a traditional air-conditioned environment. The calculation results are shown in Table 1 in term of parameter *DR*. Surprisedly, though with higher turbulence intensity, the natural wind cases present lower *DR* values, mainly because of smaller mean air velocities. Hence, based on these results, fan operating in the natural wind mode can be inferred to achieve higher comfort levels and energy conservation.



(a) constant speed mode

(b) natural wind mode

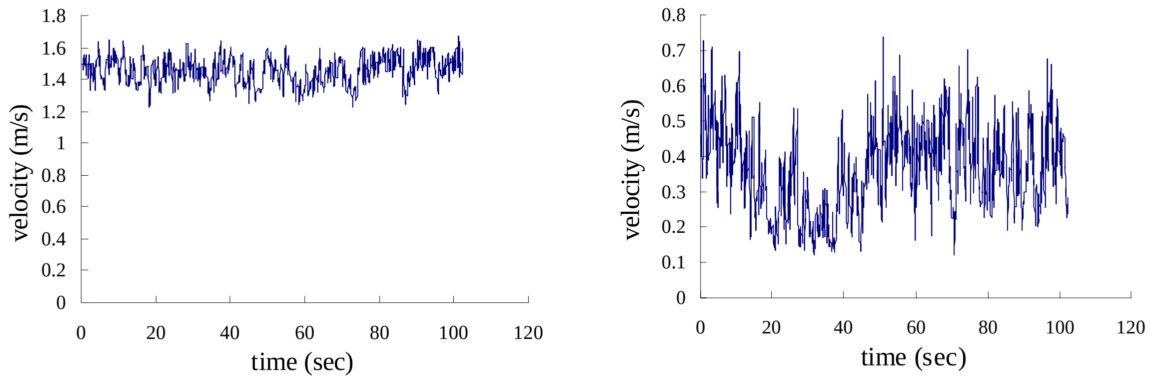
Figure 3. Velocity fluctuations for ceiling fan rotating at the high-speed setting



(a) constant speed mode

(b) natural wind mod

Figure 4. Velocity fluctuations for ceiling fan rotating at the medium-speed setting



(a) constant speed mode

(b) natural wind mode

Figure 5. Velocity fluctuations for ceiling fan rotating at the low-speed setting

To further explore the flow characteristics of the main air flow and the induced air motion, six data points were selected to calculate the parameters, as listed in Table 2, to stand for the features in two different flow regions. As the velocity profile shown in Figure 2, measuring points, PT-8,20,32 were positioned at half of the fan blades and 60,120,180 cm under the fan rotation plane. These positions can be interpreted as the heights of the fan discharge, standing and sitting postures of occupant, respectively, in which are under the influence of direct fan blowing. The other three measuring points, PT-10,22,34, were measured at the similar heights of the former group but are about 30 cm away from the tip of fan blade. In this region, the air motion is mainly induced by the main air stream such that the mean velocities are relatively small. Interesting phenomena can be found in Table 2 that, in the natural wind mode, the mean air velocity differences between these two regions are clearly larger than those in the constant speed mode. When the probe was being moved outside of the coverage of the rotating fan blades, the measured wind velocities rapidly decreased to about 0.2m/s, a speed often encountered in a central air conditioning system. In addition, the issue of discomfort caused by draught (*i.e. DR*) becomes almost negligible.

Table 1. Statistical analysis on the flow characteristics of a ceiling fan which operates in constant speed mode and natural wind mode respectively (at room temperature 28 ^oC)

(a) constant speed mode			
Statistics parameters	high-speed	medium-speed	low-speed
MEAN	1.7041	1.6242	1.4537
SD-v	0.1253	0.1058	0.0834
SD-a	1.1020	0.9108	0.6068
Tu	0.0735	0.0651	0.0573
DR	26.1185	25.2718	23.4763
fe	1.4003	1.3705	1.1586
1/f	1.00	0.98	1.20
(b) natural wind mode			
Statistics parameters	high-speed	medium-speed	low-speed
MEAN	0.9400	0.6658	0.3236
SD-v	0.2602	0.2167	0.1217
SD-a	2.7014	1.9572	0.7999
Tu	0.2767	0.3255	0.3761
DR	18.0645	14.3047	8.5555
fe	1.6527	1.4374	1.0462
1/f	0.63	0.83	1.10

The frequency of flow fluctuations will also affect occupant's perception of comfort. To identify the influence from the frequency aspects, two common statistical analysis methods are used: the first one is the so-called equivalent frequency. It is a function of the instant changes of wind velocity and the corresponding acceleration. According to the quoted literatures, equivalent frequency in the range of 0.2-0.6 Hz, the human body is most likely to suffer from the cold draught. The second one is the so-called *1/f* fluctuation index, which is the slope of linear equations in a logarithm plot of speed vs. frequency obtained from the Fourier transformation of air velocity. Based on the literatures, when this factor is less than 1.0, it represents that the air stream is still in the process of developing; when larger than 1.0, the air movement is caused by induction; when closer to 1.0, the air flow characteristics is more similar to a natural wind spectrum, and the human comfort level will be increased. Table 2 shows the calculation results for the ceiling fan rotating in a constant speed and in variable speed modes. The equivalent frequency (*fe*) are not in the aforementioned frequency range which is most likely to cause uncomfortable draught. In the aspect of *1/f* analysis, the results for most cases are all close to 1.0. The ceiling fan might demonstrate a more uniform air flow in comparison to a conventional jet-flow-typed air distribution device due to the slow rotation speed and large fan blade of a ceiling fan. No matters operating in the constant speed or the natural wind mode, the measured air velocity, after making Fourier transform, are found contribute insignificant differences on the slopes of *1/f* curves for different conditions. It is indeed a surprising finding.

Table 2. Statistical analysis on the flow characteristics at two lines of measuring points representing the circumstances of direct wind blast and induced air motion respectively (at room temperature 28 ^{o}C)

(a) constant speed mode						
Statistics	PT-8	PT-20	PT-32	PT-10	PT-22	PT-34
MEAN	1.7433	1.5999	1.5447	1.3326	1.4006	1.4034
SD-v	0.1244	0.1301	0.1392	0.0466	0.1018	0.1094
SD-a	1.1898	1.1072	0.9606	0.2212	0.6257	0.5668
Tu	0.0713	0.0813	0.0901	0.0350	0.0727	0.0780
DR	26.4982	25.1001	24.5681	22.1046	22.9712	23.0211
fe	1.5224	1.3543	1.0983	0.7548	0.9781	0.8244
1/f	0.8000	0.9100	1.0000	1.3000	1.2000	1.2000
(b) natural wind mode						
Statistics	PT-8	PT-20	PT-32	PT-10	PT-22	PT-34
MEAN	0.7649	0.6641	0.5053	0.2279	0.1932	0.2466
SD-v	0.3452	0.3123	0.2477	0.0828	0.0692	0.1193
SD-a	2.3599	2.2190	1.5464	0.5040	0.3096	0.5919
Tu	0.4514	0.4703	0.4903	0.3634	0.3583	0.4836
DR	15.9227	14.4370	11.9047	6.5218	5.6929	6.9689
fe	1.0879	1.1307	0.9934	0.9686	0.7117	0.7900
1/f	0.9200	0.8800	0.9500	1.2000	1.3000	1.2000

DISCUSSION

In this paper, the measurement of a new ceiling fan, which can vary its instant motor rotation speed to simulate a natural wind pattern, has shown that by adopting the natural wind mode operation, even at a lower mean air velocity, a substantial increase of airflow turbulence intensity can be easily achieved. Hence it, in terms, results in a better comfort level while consumes much less electric energy. In the consideration of achieving a sustainable built environment, this finding is of great importance. But according to the literatures, the so-called natural wind must present itself as a linear equation with the negative slope close to a value of 1.0 in a logarithm plot of velocity vs. frequency. It is surprised that the characteristic values have no significant difference no matter the ceiling fan is operating in a constant speed or natural wind mode. The results might indicate that the judgment of comfort should utilize multiple analysis methods, rather than a single target analysis method. A coherent argument about the description of comfort for the present study seems not fully established yet. However, it is believed that through continuous in-depth studies, we can achieve environmental sustainability by further understanding the characteristics of natural environment.

ACKNOWLEDGEMENT

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Numerical modelling of air supply and air flow pattern of a room that contains a gas appliance with open combustion chamber

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SUMMARY

In the last decade, quite a few carbon-monoxide intoxications occurred in Hungary due to the inadequate operation of gas appliances with open combustion chamber, connected to chimneys. These cases emphasized the importance of faultless air supply of the appliances and the safe removal of the incipient flue gases. This problem gave reason for the modelling of air supply, temperature and velocity distributions in the space that contains the gas appliance. For the modelling of changes caused by the variations in the inside or outside ambient conditions, numerical modelling can be used. With the help of CFD, the phenomena can be studied in what is virtually a computational environment.
CFD modelling gives results for the changes in the magnitude and direction of air velocity in the room and between the air inlet and the appliance, temperature distribution in the room and from the weather factors, the effect of wind on the operation of the air inlet.

INTRODUCTION

"B" type gas appliances have an open combustion chamber; combustion air comes from the room in which the equipment operates, while flue gases leave through a chimney. The two primary groups of "B" type gas appliances according to the European grouping are [8]:

– Appliances with atmospheric burner and draught hood, connected to a chimney with natural draught (e.g. B_{11}, Figure 1.),
– Appliances, which have burners installed with ventilators, connected directly to the chimney, without draught hood (e.g. B_{23}, B_{33}, B_{53}).

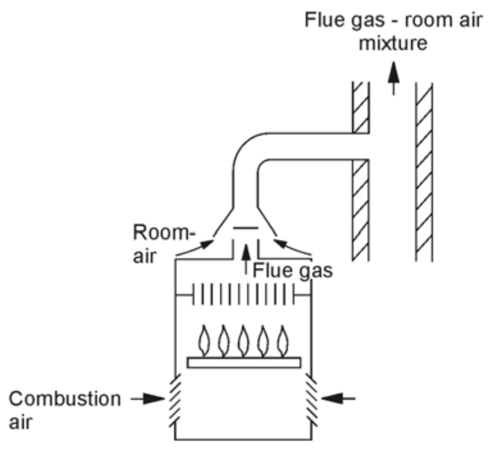

Figure 1. B_{11} type gas appliances, with opened combustion chamber, connected to a chimney

9-10 million gas appliances are estimated to operate in Hungary, most of which connected to a chimney and have open combustion chamber (B_{11} type gas appliances). In the case of these appliances, flue gas has immediate contact with the air of the room in which the machine is installed. Thus, if the air-flow conditions are unfavourable, the flue gas may re-enter the space.
Air flows into the room due to the draught in the chimney. To be able to determine the operating point of the system the mathematical model of the chimney, the gas appliance, the room and the air inlet has to be worked out. However, the simultaneous or variance-based examination of several factors cannot be carried out analytically because of the large number of equations and their complexity (differential and integral equations etc.). For the modelling of changes caused by the variations in the inside or outside ambient conditions, numerical modelling can be used. With the help of CFD, the phenomena can be studied in what is virtually a computational environment.
CFD modelling gives results for the changes in the magnitude and direction of air velocity in the room and between the air inlet and the appliance, temperature distribution in the room and from the weather factors, the effect of wind on the operation of the air inlet.

METHODS

Steps of the CFD modelling:
- creating the geometry of the model,
- stating the differential equations for the numeric model,
- developing the CFD model,
- modelling the air supply and flue gas removal of a gas appliance for different conditions and operation modes, compute the air velocity and temperature distribution in the room.

First step of investigations: creating the geometric model
For the modelling of the B_{11} type gas appliance a conventionally sized room is used, in which the appliance is the only equipment (Figure 2). A volume of the room is 15 m^3, and its size in detail is: 2 m (width), 2,5 m (length), 3 m (height). The windows and doors of the room are air-tight structures made of wood or plastic, sealed with several layers of rubber sealing. Outside air can barely or cannot enter at all in the room through natural (gravitational) means. The air necessary for combustion is provided via air inlets that in the model were inserted in the window frame in different ways, under and above the window.

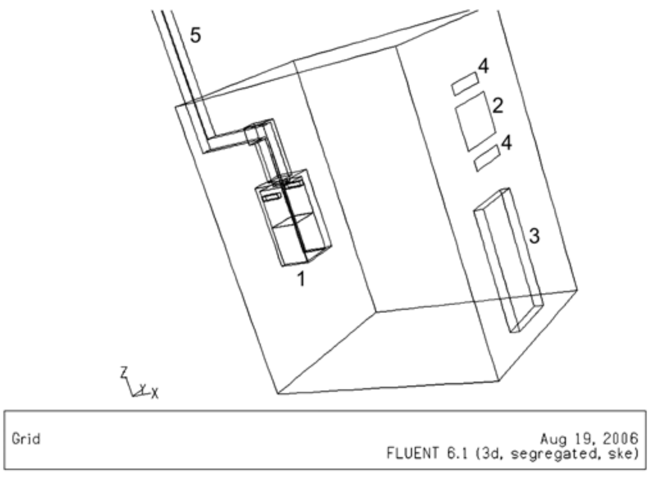

Figure 2. The geometric model used for the examination of B_{11} type gas appliances
1 – wall-mounted gas appliances, 2 – window, 3 – radiator, 4 – air inlet, 5 – chimney

The *U*-value of the external wall is 0.45 W/m²·K, while the window has a *U*-value of 1.4 W/m²·K.
Under the window a radiator is situated that is controlled by a thermostatic radiator valve which adjusts the heat loss so that the desired room temperature is achieved.
Nominal heat output of the gas appliances in the investigation are: 12 kW, 24 kW, 28 kW and 36 kW.
The connecting flue pipe consist of: 0,5 m long vertical section, bend, 1 m long horizontal section. The pipe is made of aluminium and has a maximum absolute roughness of 1 mm.
The chimney is situated partly in the heated space and partly outside.
The outdoor air temperature is -15 °C, which is the best condition regarding the chimney but is the worst from the room's comfort point of view.
Figure 3 shows the main sizes of the geometric model that was shown in Figure 2.

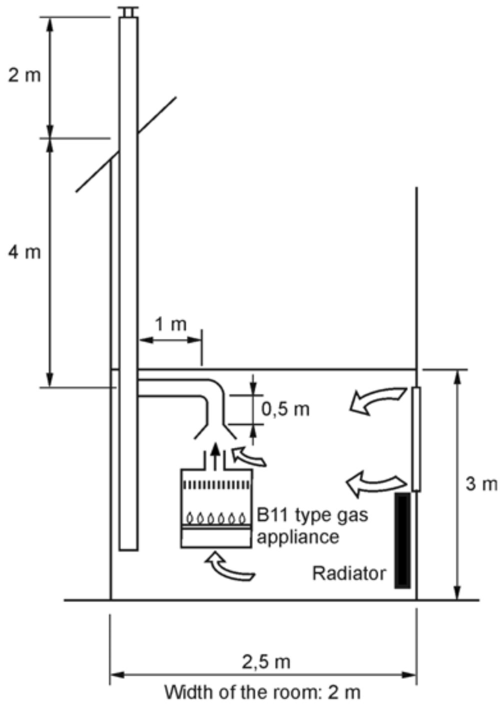

Figure 3. The main sizes of the geometric model

Stating the differential equations for the numeric model
The numeric model, based on the geometric model, was developed by adding principal initial and boundary conditions.
The air movements of closed areas are described by the differential equations of continuity and *Navier-Stokes*. The thermo balance of the areas is expressed by the equation of energy; its distribution of concentration is described by the differential equation of material balance. As we are talking about turbulent air conduction, also the proportion of the kinetic energy and the dissipation (k-ε) of the airflow has to be determined. Resulting from a system of equations, this is the mathematical model of closed spaces.
Assuming an incompressible agent the listed equations are formed as follows:

Continuity:

$$\mathrm{div}\,(\rho \cdot u_\mathrm{i}) = 0\,, \qquad (1)$$

where ρ is the air density and u_i are the air velocity components in *x, y, z* direction.

Equation of movement:

$$\frac{\partial}{\partial x_i}(\rho \cdot u_i \cdot u_j) = \frac{\partial}{\partial x_i}\left((\mu + \mu_t)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right) - \\ - \frac{\partial}{\partial x_j}\left(p + \frac{2}{3} \cdot \rho \cdot k \cdot \delta_{ij}\right) + g_i(\rho_x - \rho) , \qquad (2)$$

where μ is the viscosity, and μ_t is the turbulent viscosity, p is the pressure, k is the kinetic energy, and δ_{ij} is the *Kronecker* symbol.

Equation of energy:

$$\frac{\partial}{\partial x_i}(\rho \cdot u_i \cdot h) = \frac{\partial}{\partial x_i}\left\{\left(\frac{\mu}{\sigma} + \frac{\mu_t}{\sigma_t}\right)\frac{\partial h}{\partial x_i}\right\} + Q , \qquad (3)$$

where h is the enthalpy, Q is the quantity of heat per volume, σ_t is a factor, depends on *Prandtl*- and *Schmidt*-numbers.

The turbulent viscosity:

$$\mu_t = K \cdot \rho \cdot \frac{k^2}{\varepsilon} , \qquad (4)$$

where K is a constant, ε is the dissipation of kinetic energy.

Turbulent kinetic energy:

$$\frac{\partial}{\partial x_i}(\rho \cdot u_i \cdot k) = \frac{\partial}{\partial x_i}\left\{\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_i}\right\} - \\ - K_4 \cdot \rho \cdot \varepsilon + \mu_t \frac{\partial u_i}{\partial x_j}\left\{\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right\} + F , \qquad (5)$$

where σ_k is the kinetic energy factor.

Dissipation of turbulent kinetic energy:

$$\frac{\partial}{\partial x_i}(\rho \cdot u_i \cdot \varepsilon) = \frac{\partial}{\partial x_i}\left\{\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_i}\right\} - K_2 \cdot \rho \cdot \frac{\varepsilon^2}{k} + \\ + K_1 \cdot \mu_t \frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\frac{\varepsilon}{k} + K_3 \cdot F \cdot \frac{\varepsilon}{k} , \qquad (6)$$

where

$$F = g_i\left\{\beta \frac{\mu_t}{\sigma_t}\frac{\partial T}{\partial x_i} + \beta_c \frac{\mu_t}{\sigma_{ct}}\frac{\partial C}{\partial x_i}\right\} . \qquad (7)$$

Standard *k-ε* turbulence model

The *k-ε* transport equation is created from *Navier-Stokes*-equation on condition that the turbulence effect dominates over the whole flow field. The *k-ε* turbulence model ensures the option to operate turbulence effects as transport equations.
The continuity equation for incompressible and source-free medium:

$$\frac{\partial u_i}{\partial x_i} = 0 \ , \qquad (8)$$

where u_i are velocity components, x_i are coordinates, $i = 1, 2, 3$.

The conservation of momentum equations use *Newton's* movement laws. The resultant of external forces, affecting the elementary volume, equals to the resultant of total momentum's growth and total outgoing impulse from the elementary cell with reference to same elementary volume. These external forces on one hand are external stresses on the surface of the primary cell, on the other hand split force effects, like the force effect resulting from gravity:

$$\rho \cdot \frac{\partial u_i}{\partial t} + \rho \cdot \frac{\partial}{\partial x_j}(u_i u_j) + \frac{\partial p}{\partial x_i} - \frac{\partial \tau_{ij}}{\partial x_j} - \rho \cdot F_i = 0 \ , \qquad (9)$$

where τ is symmetrical liquid viscosity stress tensor, $\rho \cdot F_i$ is split force effect (e.g. gravity), for our purposes it is considered to be zero.

Liquid viscosity stress tensor in *Newton*'s medium:

$$\tau_{ij} = \mu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \ , \qquad (10)$$

where μ is the dynamic viscosity, Ns/m^2.

Equations (9), (10), and (11) describe *Newton*'s medium flow in laminar and turbulent case. If the computations were based on these equations, the model would have such a fine resolution for the investigation of smaller and greater fluctuations that in the end the necessary calculation power would be greater what an average computer could handle. Because of this, the *Navier-Stokes* equation's time average modification has to be used. However, it can only be used for the calculation of large-scale fluctuations. Small fluctuations have to be described with the help of imminent or empirical methods. In 1883, *Reynolds* proposed and introduced the *f(x,t)* value, which could manage the fluctuation's size with an average in time.

Reynold's filter can be stated in a more general form, where *f(x,t)*'s first component is the large-scale fluctuation's average in the time, $\bar{f}(x,t)$, while the other component is the small-scale fluctuation's average in time $f'(x,t)$:

$$f(x,t) = \bar{f}(x,t) + f'(x,t) \ . \qquad (11)$$

This average-creating method can be understood as filter permeable at the bottom, which, in function of time, filters small-scale fluctuations.

The modified *Navier-Stokes* equation system and the continuity equation is as follows:

$$\frac{\partial \overline{u}_i}{\partial x_i} = 0 \quad \text{and} \tag{12}$$

$$\frac{\partial \overline{u}_i}{\partial t} + \frac{\partial(\overline{u}_i \overline{u}_j)}{\partial x_j} + \frac{\partial}{\partial x_j}(R_{ij} - \frac{1}{\rho}\overline{\tau}_{ij}) + \frac{1}{\rho}\frac{\partial \overline{p}}{\partial x_i} = 0 \ , \tag{13}$$

where

$$R_{ij} = \left\langle u_i' \ u_j' \right\rangle \ , \ u_i' = u_i - \overline{u}_i \ , \ p' = p - \overline{p} \ , i, j = 1, 2, 3$$

By introducing the concept of turbulent viscosity, which connects *Reynolds* stress and the gradient of the spatial mean velocity, and, following the suggestion of *Boussinesq* from 1887, the following can be stated:

$$-R_{ij} = \nu_t \left(\frac{\partial \overline{u}_i}{\partial x_j} \frac{\partial \overline{u}_j}{\partial x_i} \right) - \frac{2}{3} k \cdot \delta_{ij} \ , \tag{14}$$

where ν_t is the turbulent viscosity in m^2/s.

Turbulent medium kinetic energy:

$$k = \frac{1}{2}\sum R_{ii} = \frac{1}{2}\langle u_1'u_1' + u_2'u_2' + u_3'u_3' \rangle . \tag{15}$$

Using these terms, the definition of the R_{ij} value is simplified to the calculation of the turbulent viscosity. However, turbulent viscosity depends on flow and not on the medium. The turbulent viscosity after the dimension analysis is:

$$\nu_t = \frac{\mu_t}{\rho} = C_\mu \frac{k^2}{\varepsilon} . \tag{16}$$

With the *k-ε turbulence model* it becomes possible to manage turbulent effects as a transport equation. It is an important advantage that numerical methods can handle transport equations and thus, besides the known transport (diffusion) processes, turbulence can be modelled as well.

Yet, the *k-ε turbulence model* does not provide satisfactory accuracy in the case of flows in the wall region. Therefore, the application of wall law cannot be avoided, adding more equations to the equation system.

RESULTS

Figure 4 and 5 show the temperature and velocity distributions and the temperature flow lines in the room.
It can be seen on Figure 4 that the air coming through the air inlet is heated up quickly and the temperature in the occupied zone is between 20 and 22 °C. Figure 5 shows that in the occupied zone velocities are far below 0.1 m/s.

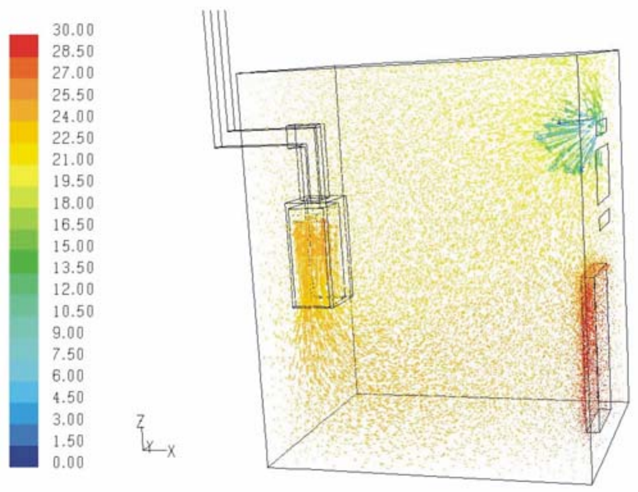

Figure 4. Temperature distribution in the room

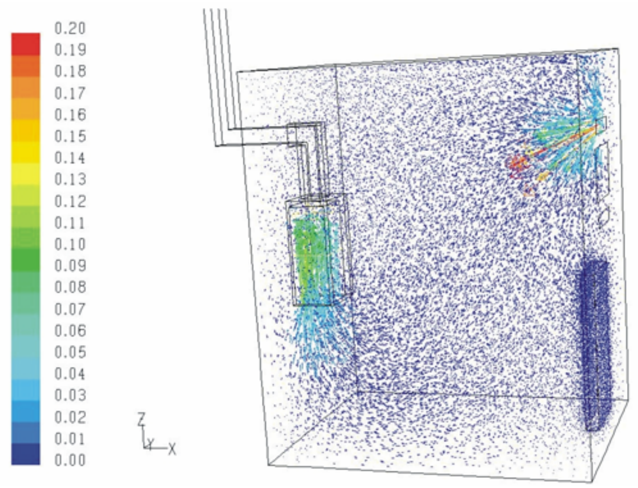

Figure 5. Velocity distribution in the room

The numerical data from the CFD simulation are collected in Table 1.

Table 1. Results of the modelling for winter case

Nominal heat output of the appliance, kW	12	24	28	36
Mass flow of the flue gas, kg/h	22.3	49.3	75.6	108
Temperature of the flue gas leaving the appliance, °C	101	121	122	118
Temperature of the flue gas in the beginning of the chimney's indoor section, °C	88	104	112	108
Temperature of the flue gas in the beginning of the chimney's outdoor section, °C	70	75	89	82
Temperature of the flue gas at the outlet point, °C	60	62	84	71
Air flow rate, m^3/h	23	45	53	73.3
Pressure difference by the air-inlet, Pa	3.9	4.4	4.5	4.5
Average air temperature of the room, °C	24.1	23.6	23.5	23.6
Average air velocity in the room, m/s	<0.1	<0.1	<0.1	<0.1

DISCUSSION

Based on the data introduced in the table and on the distributions indicated in the figures the following can be stated:

- the average air temperature in the occupied zone is adequate (24 °C) and uniform,
- the relative air velocity in the occupied zone is far below 0.1 m/s,
- the air velocity in the occupied zone has a uniform distribution,
- 30 cm from the air-inlet the entering fresh air has a temperature that hardly differs from the average air temperature of the room,
- it is reasonable to place a heat-transfer appliance (accurately sized radiator) under the air-inlet so that the cool zone of the air-inlet can be reduced.

The investigation about appliances' air supply and the publication of the design aiding results is of importance as in Hungary numerous B_{11} type appliances with atmospheric burner and draught hood, connected to a chimney with natural draught operate.

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Field Survey of Thermal Environment And Occupancy Condition of Passengers in Railway Station

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SUMMARY

Field surveys on thermal environment were carried out at Station T which is a large station in Tokyo, mainly in summer 2006. It appeared that the thermal environment in concourse is closely related to passenger's comfort. As the result of surveys, it was clarified that thermal environment in the concourse was not acceptable enough for passengers. At most place and time, thermal environment in concourse was easily over 32°C in SET*, upper limit of acceptable range for passengers. In addition, thermal environment in the concourse was widely distributed. But passengers stayed in relatively uncomfortable place for the reasons of meeting people. For improving thermal environment and comfort in the station, it should be considered that the detail of thermal environment and occupancy characteristic of passengers.

INTRODUCTION

Most of the people in Japan use railway stations everyday to commute to their office or school. Since the stations can be easily crowded especially in a rush-hour, it has been designed for smooth transit. On the other hand, railway stations lately started to draw attentions as commercial opportunities. Not only small shops but also big shopping malls are introduced, which includes café, restaurant, bookstore, boutique, travel agency and so on. Therefore the usage of stations is being changed for the passenger to stay longer.
Therefore the comfort in stations is getting more important. However its thermal environment is not comfortable enough especially in summer. As a result of a previous field survey in several stations in 2004[1], air temperature and radiant temperature of the concourse was generally higher than outdoors and air velocity in concourse was distributed on lower range. From these facts, thermal environment of summer is especially a problem in concourse. In addition, as a result of questionnaire to passengers about thermal sensation and acceptability, the upper limit of acceptable range for passengers was found to be 32°C in SET*. In order to solve this situation, introducing an air-conditioning system for the whole station-building would cause a huge energy loss due to the open structure of stations. Being both comfort and less energy consumption required, more studies for the current environment of the station and its solution are necessary.
In this research we investigated about Station T, which has a plan to introduce a shopping mall inside next year, whereas lots of compliment exists for the hot environment in summer. Field survey on thermal environment and questionnaire survey were conducted mainly in summer, and thermal environment and comfort was quantitatively evaluated.

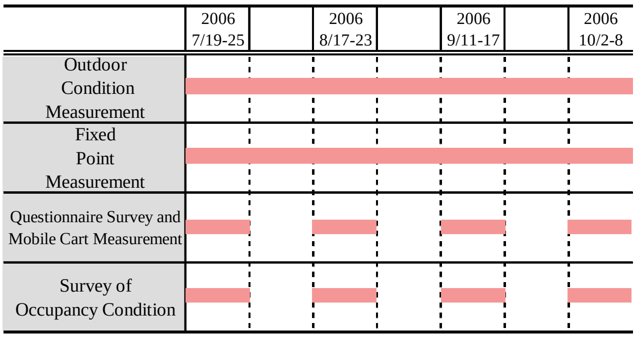
METHODS

Field surveys were carried out during July to October 2006 in Station T. Descriptions of the station are given in Table 1. Station T is located in Tokyo and has a large-scale with 4800m^2 floor area. There are many skylights made of polycarbonate in concourse. Field survey mainly consists of outdoor condition measurement, fixed point measurement, questionnaire survey with mobile cart measurement, and survey of occupancy condition. Mobile cart measurement was conducted for measuring thermal environment around passenger answering questionnaire. Survey periods of each item are given in Table 2 and measuring instruments are shown in Figure 1.

Table 1. Description of the station

Station Name	Tachikawa
Photo	
Daily Passengers	148,000
Concorse Type	Bridge
Number of Platforms	4
Floor Area	4800m^2

Table 2. Survey periods



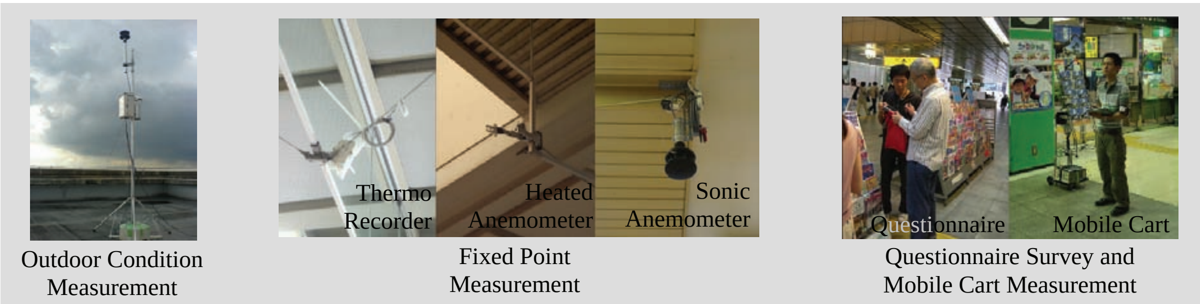


Figure 1. Measurement instruments

Thermal Environment Measurement: Measurement items of outdoor condition measurement, fixed point measurement and mobile cart measurement are given in Table 3. Since Station T have large floor area, it was thought that thermal environment in the concourse have a wide distribution. Therefore many measuring instruments were set up evenly for fixed point measurement. Floor plan of Station T and measurement points is shown in Figure 2.

Table 3. Measurement items

	Measurement Items	Instruments
Outdoor Condition Measurement	Air Temperature	Thermo Recorder
	Humidity	Thermo Recorder
	Air Velocity	Sonic Anemometer
	Air Direction	Sonic Anemometer
	Solar Radiation	Solar meter
Fixed Point Measurement (Height=3m)	Air Temperature	Thermo Recorder
	Humidity	Thermo Recorder
	Air Velocity	Heated Anemometer
	Air Direction	Sonic Anemometer
Mobile Cart Measurement (Height=1.1m)	Air Temperature	C-C Thermocouples
	Humidity	Thermo Recorder
	Air Velocity	Heated Anemometer
	Total Radiation	Directional Radiometer (0.3-40μm)
	Solar Radiation	Silicon Pyranometer (0.4-1.1μm)

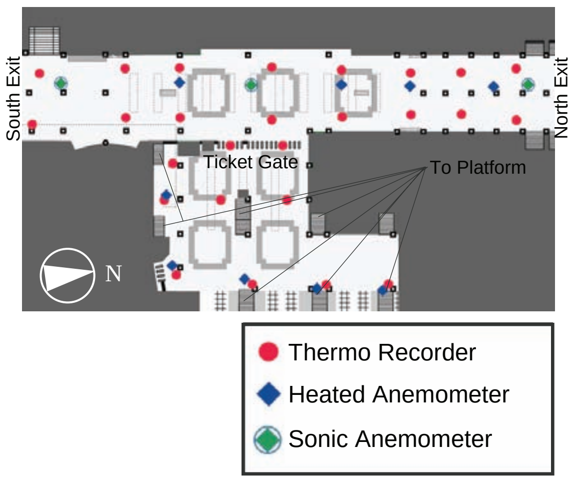


Figure 2. Measurement points

Survey of the Passengers: Questionnaire survey and survey of stay place were carried out in concentrated survey period. Items of questionnaire survey are given in Table 4. 1,960 answers were collected throughout the survey. Survey of stay place was conducted every hour in 10 to 18 o'clock. In this research, occupants plotted on the sheet of floor plan.

Table 4. Items of questionnaire survey

Background	Sex, Age, Occupation, Items of clothing , Stay place , Purpose of stay
Psychological Responses (Scales)	General Comfort (+3 Very comfortable / +2 Comfortable / +1 Slightly comfortable / 0 Neutral / -1 Slightly uncomfortable / -2 Uncomfortable / -3 Very uncomfortable)
	Thermal Sensation (+3 Hot / +2 Warm / +1 Slightly warm / 0 Neutral / -1 Slightly cool /-2 Cool / -3
	Thermal Preference (Cooler - As it is - Warmer)
	Acceptability (Acceptable - Not acceptable)

RESULTS

Weather Condition

Daily mean temperature in measurement period is plotted in Figure 3, together with the average year temperature variation. In 2006, although the end of the rainy season was delayed, outdoor temperature in summer tended to be higher than that of average year. Since outdoor temperature in concentrated survey period of august was especially high, result of field surveys of this period are mainly shown afterwards.

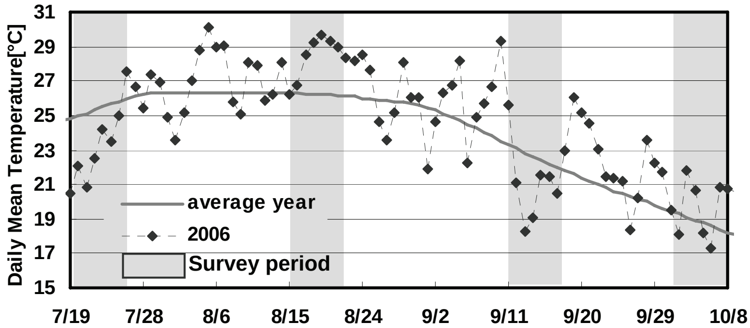


Figure 3. Daily outdoor temperature

Temporal Change of Thermal Environment in Concourse

Temporal change of air temperature and radiant temperature are shown here. It paid attention to the difference between indoor and outdoor environment.

a) Air Temperature: Daily temporal change of air temperature in the concourse is plotted in Figure 4, together with external air temperature and solar radiation. In sunny day of august, average air temperature of concourse was almost equal to that of outdoor at noon. But difference of indoor and outdoor temperature grew bigger in the night to early morning. In cloudy day, air temperature in the concourse was over 2 degrees higher than that of outdoor through almost a day.

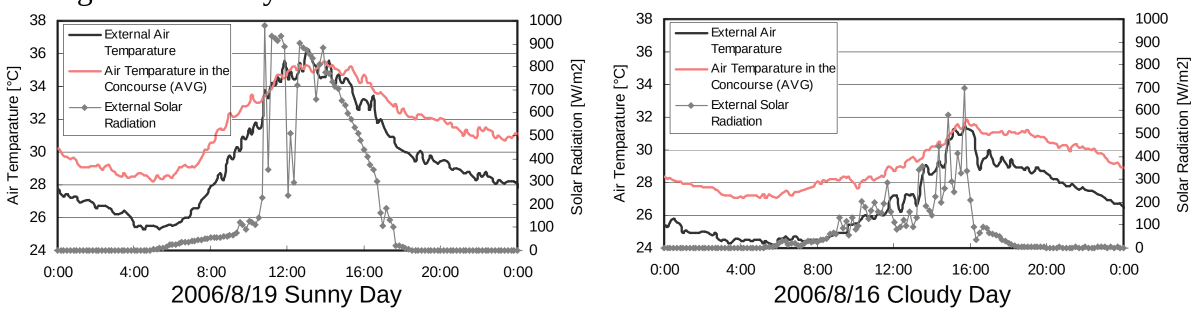


Figure 4. Temporal change of air temperature

b) Radiant Temperature: Mean radiant temperature (MRT) of the concourse and the platform is plotted against air temperature in Figure 5 and thermal images in the concourse are shown Figure 6. In this case, weighted-mean radiation of six directions was used as MRT. MRT was generally 2 degrees higher than air temperature in the concourse. According to thermal image of daytime, the surface temperature in the vicinity of skylights reached 42°C. In the evening, the surface temperature of skylights and ceiling was still higher than air temperature. It is thought that these high surface temperatures related to high MRT in the concourse.

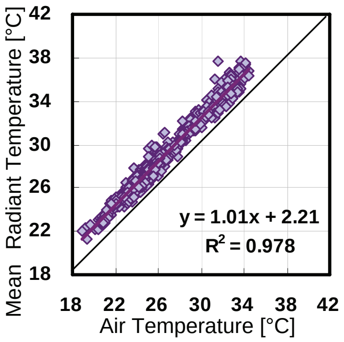


Figure 5. Air temperature and mean radiant temperature

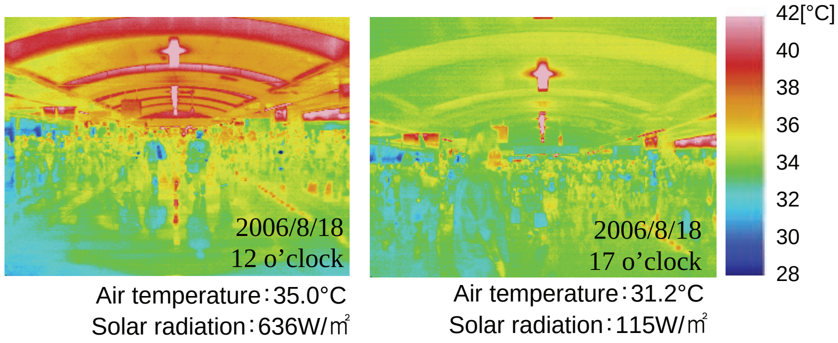


Figure 6. Thermal images

Distribution of Thermal Environment in Concourse

Spatial distribution of thermal environment became clear from the result of thermal environment measurement in each measurement points. Distribution of air temperature, air velocity and SET* shown in this thesis, because tendency of these distribution was especially clear.

a) Air Temperature: Distribution of air temperature in the concourse is shown in Figure 7. At 12 o'clock, air temperature exceeded 33°C in almost all places of the concourse. At 18 o'clock, air temperature in the concourse decreased overall. But it was still higher than that of outdoor and reached 35°C in the hottest place.

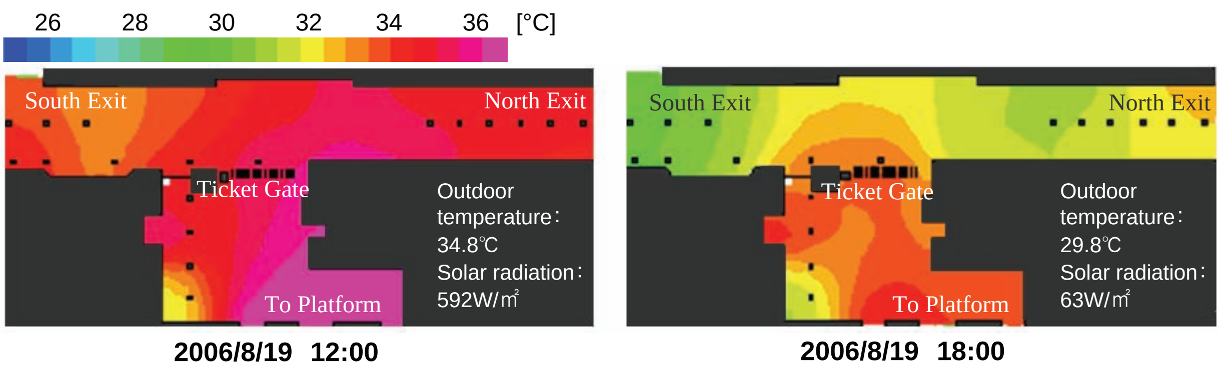


Figure 7. Distribution of air temperature

b) Air Velocity: Air velocity in the concourse was generally distributed in lower level, regardless of external air velocity. Distribution of air velocity in the concourse is shown in Figure 8. Since the external wind blew from the south at either time zone, air velocity in the vicinity of the south exit was comparatively large. But the wind from south didn't reach the center of the concourse. Therefore air velocity around the ticket gate was too small.

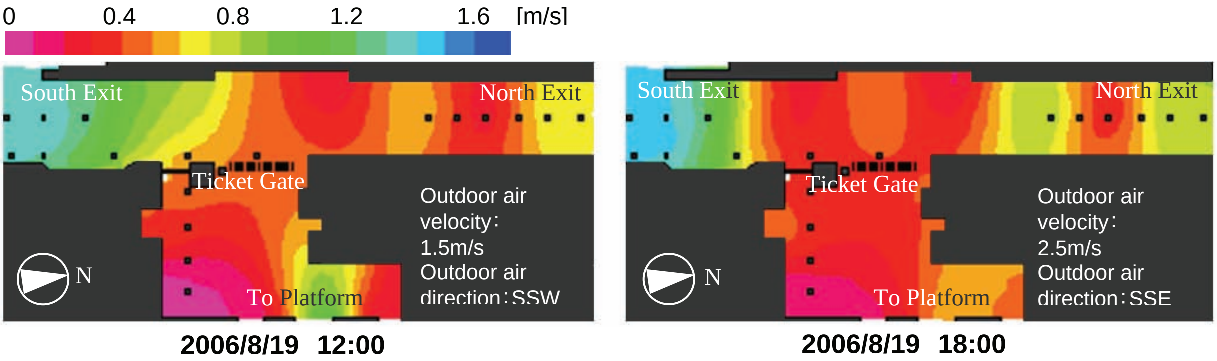


Figure 8. Distribution of air velocity

c) SET*: SET* in the concourse was calculated from measured items (air temperature, humidity, radiant temperature and air velocity) and assumptive clothing insulation (0.5clo) and metabolic rate (1.4met). Distribution of SET* in the concourse is shown in Figure 9. It showed the tendency to look like Distribution of air temperature. SET* in the vicinity of the south exit was low comparatively. It is thought that this is because the influence of the air velocity. According to the past investigation, the acceptability range for passengers was found to be 32°C or less in SET*. But in such a sunny day, it found that SET* in concourse was over 32°C in much place even in the evening.

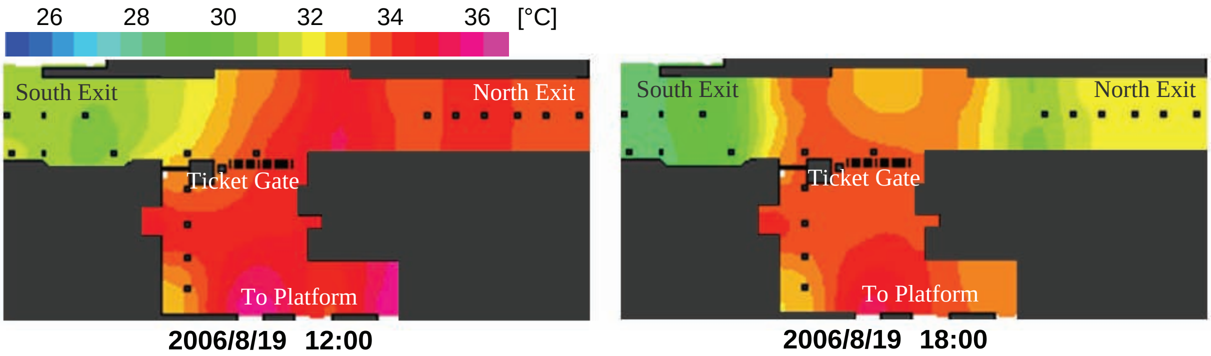


Figure 9. Distribution of SET*

Occupancy Condition

Characteristic of occupants and its occupancy in Station T are shown here.

a) Attribution of Occupants: Attributions of staying passenger are shown in Figure 10. These are from answer of questionnaire to passenger. Total number of answers was 1960, and answers were collected from various people impartially. The ratio of female was a little larger than that of male. Seeing according to the age, the ratio below 20's was large. But a certain amount of answer was collected from each age.

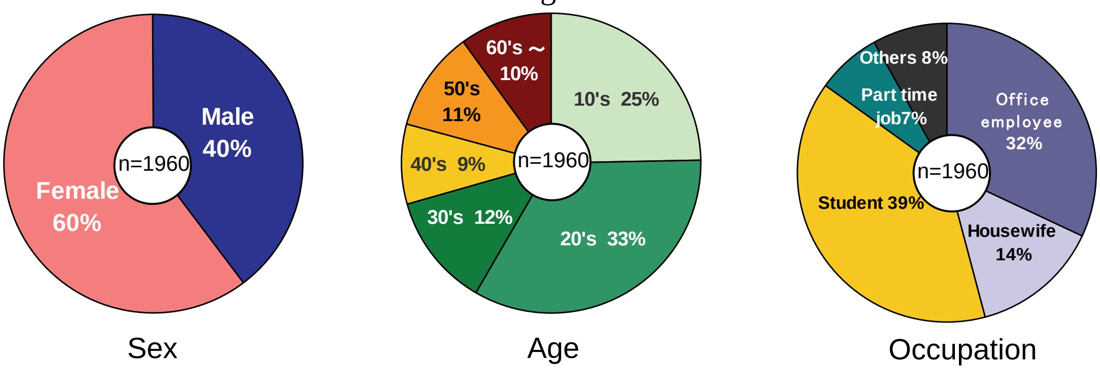


Figure 10. Attribution of occupants

b) Number of Occupants: Number of occupants in concourse is shown in Figure 11. These are from the result of survey of stay place. The transition pattern of number of occupants was almost constant regardless of the month. It appears that numbers of occupants grows most in the evenings, when number of passenger was most.

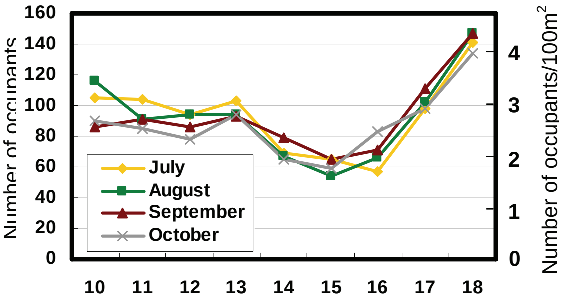


Figure 11. Number of occupants

c) Stay Place and Purpose of Passengers: Results of survey of stay place are shown in Figure 12. The tendency is clearly seen in it. The number of occupants was large at the center of the passage and near the wall. It is thought that passengers willingly stayed there for meeting people. Results of questionnaire about stay are shown in Figure 13. Much of passengers answered yes about question of “Do you wait time?” In addition, the question of “Why are you here?” was set up for the passengers who answered yes about the previous question.

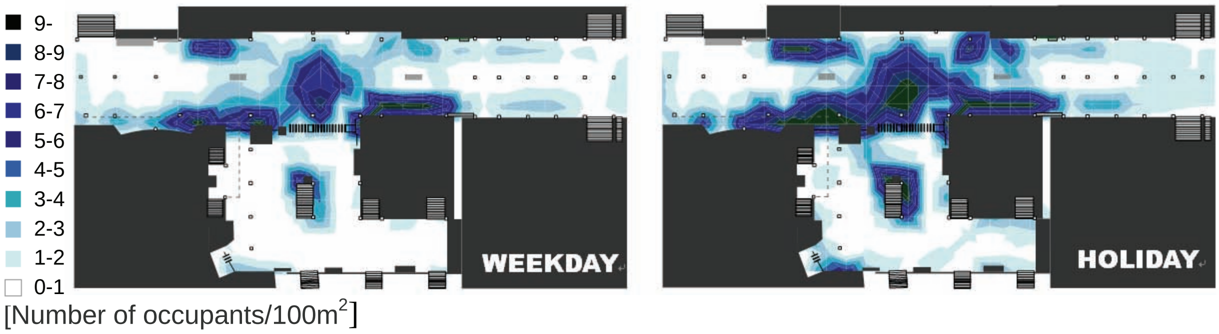


Figure 12. Stay place of occupants

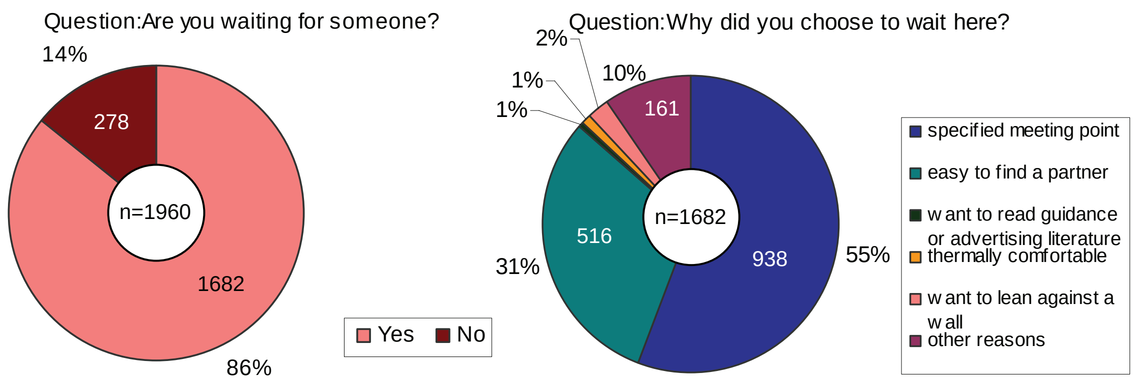


Figure 13. Purpose of occupancy

As a result of that, many of passengers answered “Specified” or “Easy to find partner”. It can be said that these people couldn’t change stay place easily regardless of comfort. On the other hand, a little number of passengers answered “cool, comfortable” or “want to lean over”. In a word, it is found to be that passengers had not decided the stay place for the comfort.

Thermal Comfort Condition and Adaptation

Thermal comfort condition and Adaptation of occupants in Station T are shown here, including comparison with the result of past investigation in 2004.

a) Air Temperature and Clothing Insulation: Clothing insulation of passengers who answered questionnaire is shown in Figure 14. The data set of air temperature in concourse by the interval of 1 was averaged, and corresponding clothing insulation was averaged. Circle shown in Figure 14 describes the number of corresponding subjects. Clothing insulation increased due to the effect of high temperature. When air temperature was over 31°C, average clothing insulation became constant about 0.5clo.

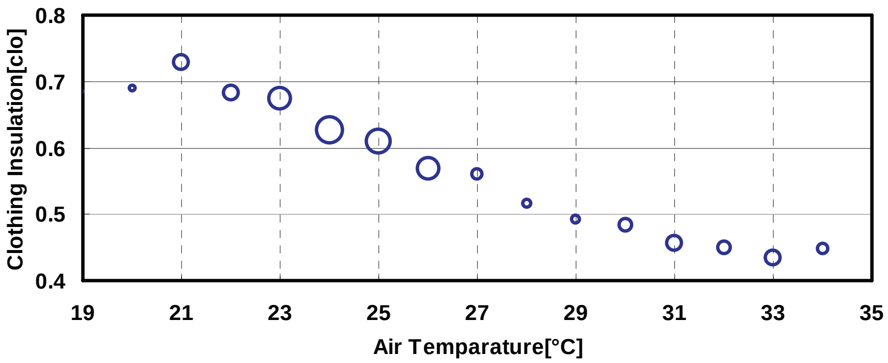


Figure 14. Clothing insulation of passengers

b) Thermal Sensation: The correlation about thermal sensation and percentage of uncomfortable or percentage of "Not acceptable" was shown in Figure 15. The data set of thermal sensation by the interval of 1 is from answer of questionnaire survey. Circle shown in Figure 15 describes the number of corresponding subjects. In figure of thermal sensation and percentage of uncomfortable, uncomfortable means answer of "Slightly uncomfortable", "Uncomfortable" and "Very uncomfortable". The clearest correlation was seen in this figure. Percentage of Uncomfortable increased with the rising of thermal sensation. The correlation appeared in the other figure. When thermal sensation is +3 (Hot), percentages of "Not accepted" was obviously large. In this way, it is thought that the thermal environment in concourse is closely related to passenger's comfort.

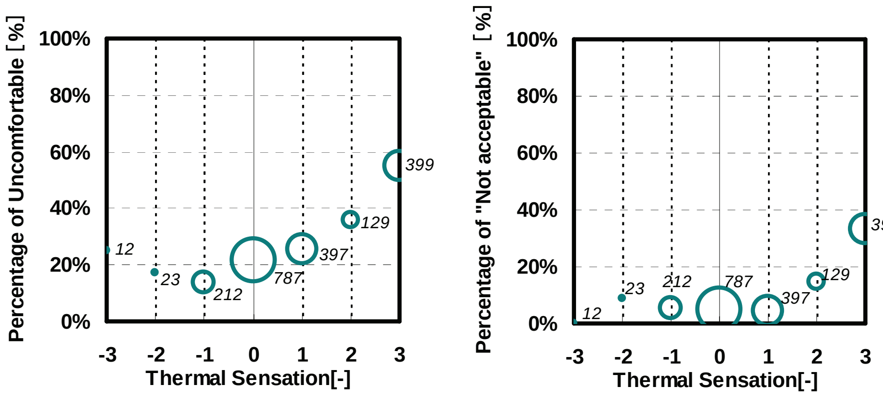


Figure 15. Thermal sensation

c) SET* and Thermal Comfort Condition: The correlation about SET* in concourse and percentage of vote about thermal sensation is shown in Figure 16. SET* was calculated assuming the metabolic rate of 1.4met for all passenger. In this figure, "Thermally Uncomfortable" means voting ±2, ±3 in thermal sensation and voting -1,-2,-3 in general comfort. "Thermally Unacceptable" means voting ±2, ±3 in thermal sensation and answering "Not acceptable" in acceptability. Two lines in Figure 16 drawn by result of past investigation in 2004 and its probit analysis. The data set of SET* in concourse by the interval of 1 was averaged, and corresponding percentage of vote averaged. Circle shown in Figure 5 describes the number of corresponding percentage of vote. The lines and circles showed clear correlation. From these results, there is a possibility of setting a constant standard about thermal environment and comfort in train station. When considering the result in the preceding clause, thermal environment in the concourse was not acceptable enough for passengers.

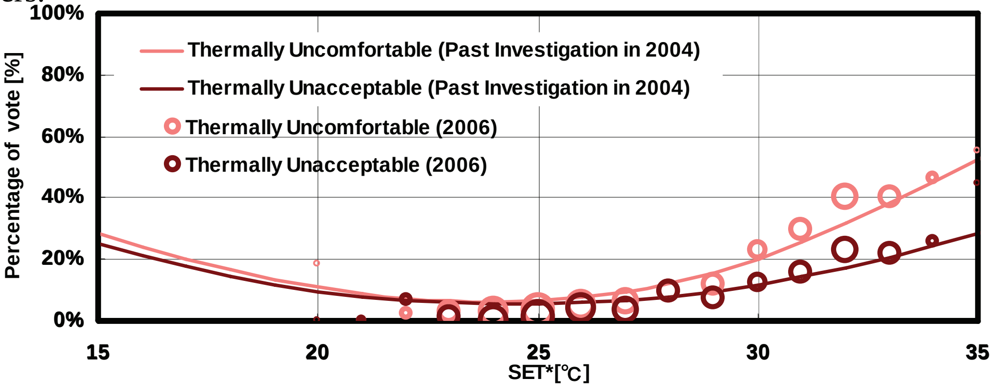


Figure 16. SET* and thermal condition

It is understood that there are the clear correlation with thermal environment and comfort from these results. In addition, contradiction was not seen between result in 2006 and past investigation. So these results can be used as index of thermal comfort in train stations. But temporal change of environment and occupants, together with spatial distribution of thermal environment must be considered when thinking of the improvement of thermal environment.

DISCUSSION

As a result of field survey, thermal environment in the concourse of Station T have many problems. The environment in the Station T was generally worse than outdoors due to the higher air temperature and radiant temperature, and lower air velocity. High sensible temperature over 32 °C in SET* was recorded often in summer, when lots of people voted that they couldn’t tolerate thermal environment. In sunny day of summer, it found that SET* in concourse was over 32°C in much place even in the evening. Moreover, comparing Figure 9 with Figure 12, it was clarified that passengers stayed in the place where the environment is comparatively bad. Considering results of questionnaire about stay, passengers couldn’t change stay place liberally. It is clear showed in Figure 4 that the difference of air temperature inside and outside of the station was depending on time. At time zone when this difference is large, thermal environment in concourse could be made improvement by means of bringing in outdoor air. When the external temperature is also high, it is possible to use spot cooling system in stay place of passengers. It is thought that the thermal environment in concourse is closely related to passenger’s comfort, so it should be considered that the detail of thermal environment and occupancy condition of passengers to improve thermal environment and comfort in railway station.

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Smoke removal in uni-storey smoke control system

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ABSTRACT

There is pronounced interest of engineers in new systems for storey smoke control system. The operational principle of this system is based on independent smoke removal from each storey. Independent ducts with fans and outlets are built for each floor of the building. For this approach it is important to safety remove smoke from the building. One of the proposed approaches is to use nozzle type outlets characterized with high flow and velocity to evacuate combustion products from the building. At the same time, due to system safety requirements it is crucial to dispose combustion products at a distance from building walls. The research interest in smoke removal systems at the Warsaw University of Technology was recently oriented for testing smoke outlets. The primary focus was on analysis of the performance of these elements taking into account flows, diameters and throws with respects to those typically applied in smoke removal systems. Therefore, in contrary to typical diffusers tests our studies included long-throw jets. The air streams generated by ventilation nozzles working as torch outlets were examined in the spacious (high cubic capacity) hall using specially prepared test site. Obtained results were used as a basis for evaluation of effectiveness of smoke removal for the smoke removal system operation.

INTRODUCTION

Uni-storey smoke control system can operate interdentally from other systems or in connection with building ventilation system. Modern offices spaces are built in the form of open space, that is arranged by user according to their needs. In buildings like this smoke exhaust are located in corridors. Smoke evacuation system is mainly assuring smoke removal towards staircase.

Compensating air, due to overpressure in the staircase, is flowing from the staircase, and then from the smoke removal grilles located beneath the corridor ceilings. The number of smoke removal grilles used is dependent on corridor length and shape; however, it should be sufficient to efficiently remove smoke in the event of fire in any of the rooms accessible from the corridor. Uni-storey smoke control system is especially attractive arrangement for buildings with high storey area.

The major advantages of uni-storey smoke control system are:

- individual smoke removal system for every storey guarantees better tightness of each level,
- there is no connection between ductworks of smoke control system of proximate storeys,
- the system can be limited to selected storeys only (important from the fire protection point of view)

- the system cost can be reduced, because smaller number of fire dampers and other elements are required,
- reduced inertia of system, due to shorter distance between the exhaust element and the smoke control fan,
- system is more flexible and easy to control,
- the exhaust ventilation ducts of air-conditioning system can be used as a part of smoke control system provided that the elements of installation are resistant to high temperature.

The new idea of smoke control system has a lot of advantages, but also disadvantages have to be considered. The major disadvantages are as follows:

- the higher number of smaller smoke control fans are needed,
- if different smoke control systems are used in the same building, the proper connection between them have to be assured,
- the precise assessment of removal conditions is required, followed by careful choice of type and localisation of exhaust outlets,
- the requirements for supply airflow: volume and localization need to be specified.

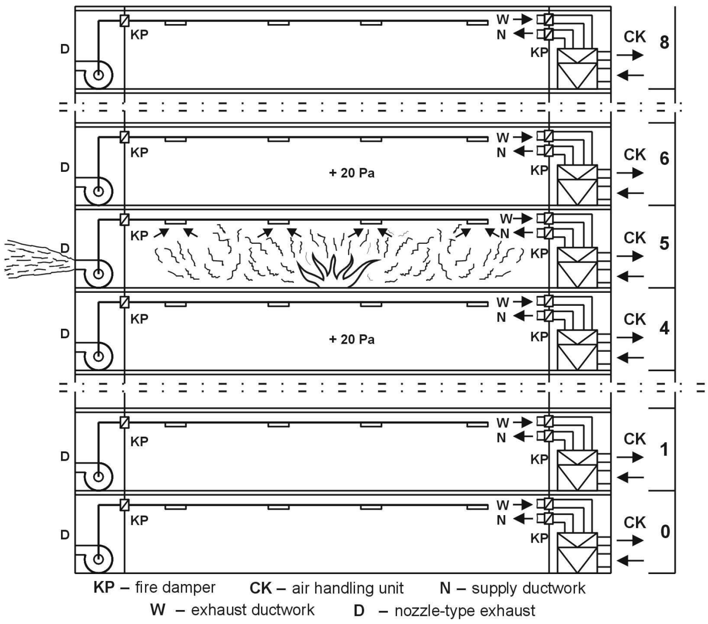

Fig. 1. Uni-storey smoke control system

SMOKE OUTLETS

Smoke control fans can be situated in relatively small room. However, it is required that the minimal distances from the walls are observed as to provide necessary access for assembly and maintenance, it also required that enough room for smoke removal ducts is reserved. Fans serving single storey are relatively small, therefore shock absorption and building protection from vibration transmission is it more easy.

One of the key problems to be solved is safe and effective smoke evacuation from the building. The choice of outlet type used is dependent on architectural building layout, position in surrounding area and general, typical, weather conditions, especially wind direction.

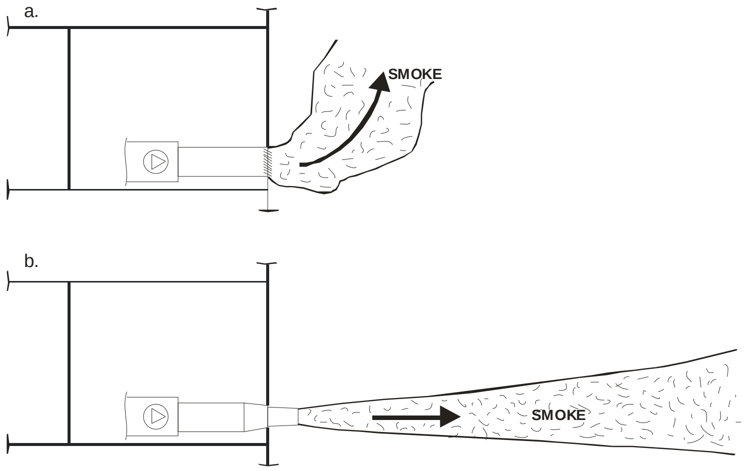

Fig. 2. Smoke outlets. a) shutter type outlet, b) long throw jet type outlet.

Using classical type wall outlet, smoke is discharged at a relatively small distance from the building, with low velocity. The localization of outlet has principal effect on its efficiency and operational safety. Optimal localization of the outlet is at leeward side of the building as well as at the outer wall, where is no window. In practice, fulfilling all above conditions is quite difficult. Therefore, a convenient solution is to locate smoke outlet either at the wall where is no windows or at the wall with windows that cannot be opened, away from other buildings. The influence of wind should be considered taking into account predominant wind direction in the considered area. Inappropriate localization of wall outlet brings a risk of inefficient smoke removal and in consequence can lead to smoke introduction to above storey.

In order to minimize the effect of wind direction and to make the system less sensitive to localization of the outlet, nozzle type outlets are used. The latter assures high rate of smoke removal. The air flow velocity (at outlet plane) achieved during tests conducted at the Institute of Heating and Ventilation, Warsaw University of Technology using outlets of diameter ϕ300 and ϕ400 mm were within the range from 8 to 16 m/s.

TESTS

As mentioned above the typical elements for storey smoke removal system is nozzle type wall outlets, used to evacuate combustion products from the building. The tests conducted were focused on properties these type nozzles in terms of volumes and diameters and on achieving appropriate smoke evacuation ranges for storey smoke removal system. Taking safety of the system into consideration it is important to discharge combustion products at the distance from the building walls. Therefore, in contrary to typical ventilation system elements, these tests were using long throw jets. The air streams extracted trough nozzles, working as torch outlets were examined. The measurements were conducted in a spacious, high cubic capacity, test-room using specially prepared test stand. The test room dimensions a=132m, b=32m, h=7m were appropriate to obtain long throw air jets, air tightness requirements were fulfilled, too. The natural air movement in the room was negligible. Two nozzles of efficiency sufficient for studied smoke extraction system were tested. The aim of the study was to characterize the reduction in axial velocity of air stream as well as to determine the throw of streams, for which the maximal airflow velocity was reduced to about 0.2m/s.

a) b)
Fig. 3. Test room: a) general view, b) measurements setup.

METHODS

The measurement set-up (Fig. 4.) included air supply system consisting of: fan equipped with speed control element, damper, measuring duct with measuring orifice, expanding box as well as multipoint anemometric system for air velocity measurement. Expanding box was raised to obtain air streams not disturbed by the building barriers. Ventilation nozzles used of diameter 300 and 400mm were fulfilling criteria for geometrically similar nozzles. The measuring system was composed of six thermoanemometric probes with transducers mounted on a vertical measuring stand, 5 m high, and data recording system. Probes used allowed us to measure the air streams within the range from 0.05 to 10 m/s and to compensate temperature with in the range from –10 to +50 ^{o}C. The air flow volume was determined using orifice connected to micomanometer. The characteristic of the reduction in axial velocity of air stream, for different air flow volumes of involved elements as well as characteristic of the jet throw was obtained. During tests the airflow velocities at cross-section of the stream were determined for two air flow volumes for each element (3000m^3/h, 4000m^3/h). The measuring stand with probes was located at the geometric axis of the air stream, at defined distance from the outlet. The axis of the air stream was determined by air flow visualization and trial measurements orientated to find the best position of the measuring stand, resulting in maximal air flow velocities at the define cross section.

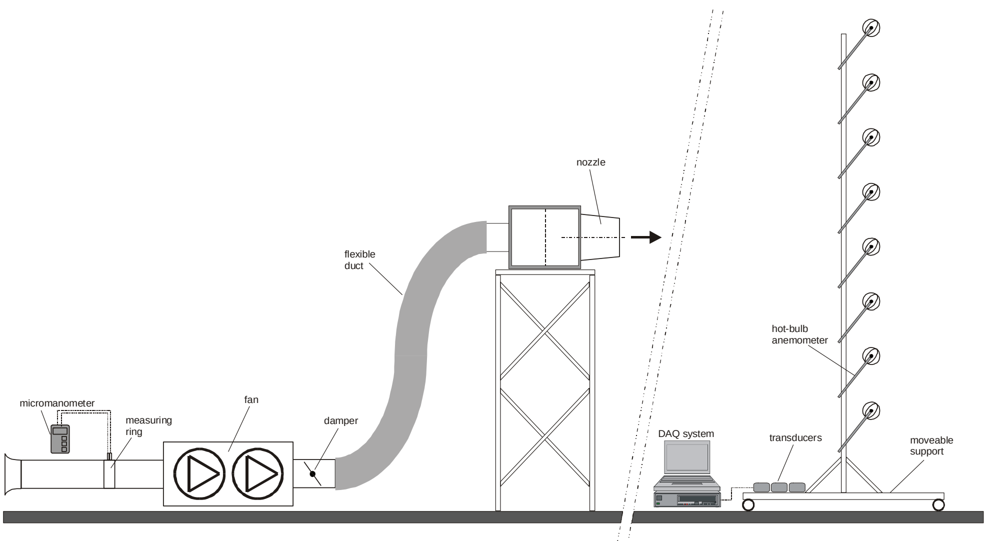

Fig. 4. Measurement set-up

RESULTS

The profiles of air flow velocities and ranges of air streams were determined. The results - profiles of air flow velocities at cross-sections located at certain distance x [m] from tested nozzle are presented. The profiles determined by airflow velocities were approximated, Fig 5 and 6.

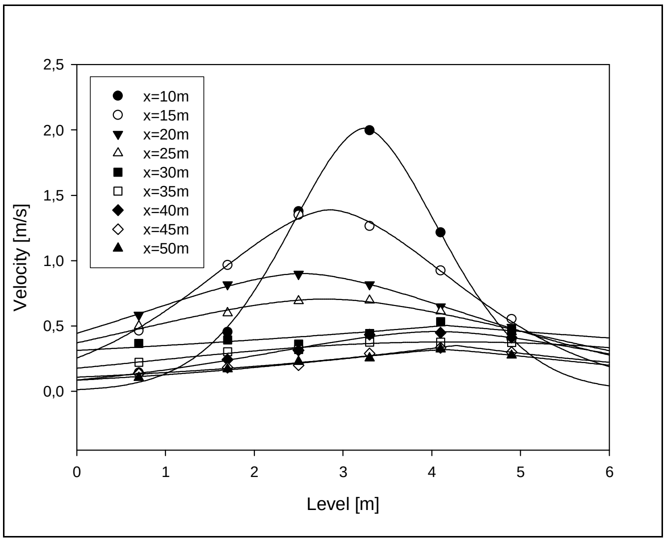

Fig. 5. Air flow velocity profiles, nozzle diameter 300mm, V=3000m^3/h

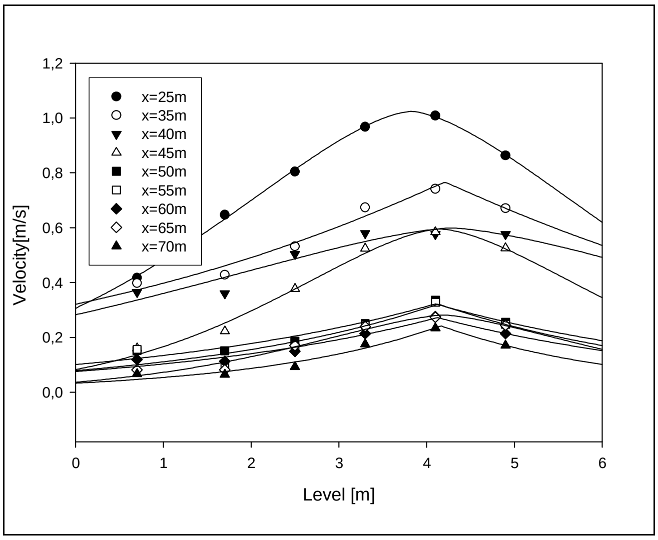

Fig. 6. Air flow velocity profiles, nozzle diameter 300mm, V=4000m^3/h

Fig 7 presents the airflow distribution at the stream axis obtained for the nozzle of diameter ϕ300mm. The initial airflow velocity was equal to 12.2 m/s for a 3000m^3/h stream. Results presented in Fig 7 show that velocity 0.4m/s is reached at the distance close to 40m. At the distance 25m the velocity is equal to 0.8m/s, this value is considered as enough resistant for disturbances caused by average wind speed. The distance from external wall is enough to achieve smoke dispersion. In the same time smoke stream is significantly thinned, due to high level of induction leading to suction of surrounding air and to increase of stream volume.

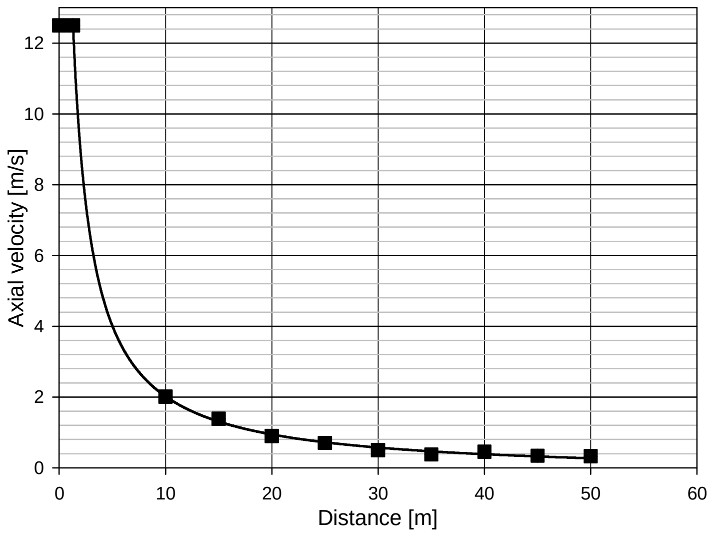

Fig.7. The distribution of axial velocity, nozzle Ø300mm, V=3000m^3/h.

For the stream 4000m^3/h evacuated through the same outlet the air velocity at the nozzle is equal to 16 m/s and the velocity 0.4m/s in stream is reached at the distance close to 50m, Fig 8. Thus the choice of the nozzle diameter need to adjusted according to stream volume of exhausted smoke to keep the appropriate air velocity at the nozzle outlet for the surrounding environment, i.e. construction and decoration type, anticipated influence for surrounding buildings and interferences form wind.

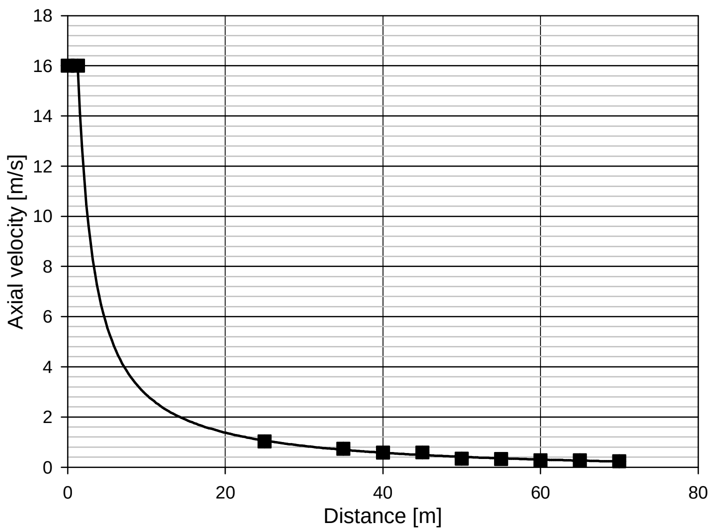

Fig. 8. The distribution of axial velocity, nozzle Ø300mm, V=4000m^3/h.

CONCLUSIONS

1. The analysis of experimental data allowed calculation of jets throws, i.e. the distances from the nozzle outlet at which the maximal air velocity is reduced to assumed limiting value of 0.4m/s. Thus defined throws for tested elements and volumes were close to 50m.

2. In order to prevent the occurrence of wind interferences, which can reduce the smoke discharge distance, the high velocities at the nozzle outlet need to be maintained.
3. Experimental data allowed us to determine the distribution of velocities at certain distance from outlet plane as profiles approximated by Gauss type function. At longer distances from the nozzle the profiles showed only minor fluctuations of air velocities at cross section.

SUMMARY

Uni-storey smoke removal system is an interesting alternative for classical approaches. In some cases this is the only possibility to evacuate smoke, especially when there is no enough room in the building for vertical ducts and central plant room. While setting the smoke removal system layout the special attention should be paid to safe removal of smoke from the building. Above presented results can help to pick up the right diameters and velocities in wall outlets of the system.

The research project concerning uni-storey smoke control systems, financed by State Committee of Science Research (project for years from 2003 to 2005) was realized in the Institute of Heating and Ventilation in Warsaw University of Technology.

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Numerical and Experimental Analysis of Elliptic Finned-Tube Heat Exchangers under Misted Conditions

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ABTRACT

The numerical and experimental analysis was carried out to study the thermal-hydraulic characteristics in elliptical finned-tube heat exchanger under the dry and air/water spray cooled system. Three kinds of major/minor axis ratios (2.5、2.8 and 3) were examined for the elliptic finned-tube heat exchangers and the results were also compared with the corresponding circular finned-tube having the same perimeter. The numerical results for the pressure and heat transfer coefficient at various inlet frontal velocities (1~5 m/s) are shown and compared with the available experimental data. The numerical results indicated that the pressure drop of circular finned-tube heat exchanger is 3 times that of the elliptic finned-tube heat exchanger, while the heat transfer coefficient of the circular finned-tube is 1.51 times that of the elliptic finned-tube. The heat transfer coefficient per unit of pressure drop for elliptic finned-tube is 1.6-2.5 times of circular finned-tube. When the air/water spray system is applied, the heat transfer could increase up to more than 30%

Keywords：elliptic finned-tube heat exchanger, mist flow , thermal-hydraulic characteristics .

INTRODUCTION

Numerous designs for industrial heat exchangers involving a bank of tubes in a fluid crossflow were applied. Among many types of finned-tube heat exchangers, those constructed of circular and elliptic fins have been used in many industries. The advantage to use elliptic finned-tube is that it can provide a lower pressure drop when the fluid flows around the tube banks. But the disadvantage is that elliptic tubes are generally restricted to low-pressure applications on the tube side. There have been a number of studies [1-5] on the characteristics of the flow field and heat transfer adjacent to a bare elliptic tube. The dry coil correlations for a staggered circular tube layout are the Briggs and Young [6] correlation for heat transfer, and the Robinson and Briggs [7] correlation for pressure drop. The convective heat and mass transfer coefficient and friction factor for circular finned-tube heat exchanger under dry and wet conditions were reported by Idem et al. [8, 9].

For the elliptic finned-tube heat exchanger, Brauer [10] was the first author to study the thermal-characteristics with an axis ratio of 1.77 under dry condition. Dunwoody [11] had discussed the variation of local Nusselt number with different axis ratio under laminar flow and constant wall temperature. In order to enhance the heat transfer, the dry/wet schemes were wildly used in many industrial applications. MaClaine-Cross and Bank [12, 13] analyzed the heat and mass transfer characteristics in a wet surface of the heat exchangers. Finlay and McMillan [14] obtained the data with the tube bank, and reported the significant effect of the water spray in heat transfer enhancement. The formula for the design of mist cooled heat exchangers on the assumption that the tubes are fully wet was proposed by Oshima et al. [15].

In this study, the 3-D thermal hydraulic simulation for the elliptic finned-tube heat exchangers was developed. The effect of heat transfer and pressure drop coefficients due to different axis ratio of elliptic fins was also discussed. A water spray system was established for the elliptic finned-tube heat exchanger and the spray effect on the thermal hydraulic characteristics were also discussed.

EXPERIMENTAL SETUP AND DATA REDUCTION

Three types of finned-tube configurations, as shown in Figure 1, were tested in the present study. The physical model and the configuration were shown in Figure2 and Figure 3. Their detail geometric parameters are tabulated in Table 1. The experiments were conducted in an induced open wind tunnel as shown in Figure 4. This setup was based on the ASHRAE 41.2 standard [16]. The air flow was driven by a 3.73-kW (5-hp) centrifugal fan with an inverter to provide various inlet velocities. An air-straightened equalizer and a mixer were provided to minimize the effect of flow maldistribution. The air temperatures at the inlet and the exit zones across the test section were measured by K-type thermocouples the accuracy of which was approximately 0.2 %. The pressure of the test coil was detected by a precision YOKOGAWA differential pressure transducer, reading to 0.1Pa.

The working fluid on the tube side was hot water and the temperature of the water was controlled by a thermostat reservoir. The hot water inlet temperature was controlled at 75℃; for the spray condition, the chilled water was controlled at 20℃. Both the inlet and outlet temperature of the working fluid (water) were measured by two precalibrated RTDs (pt100). Their accuracy was within ±0.05℃. The water volumetric flow rate was measured by a magnetic volume flow meter with 0.002 L/s resolution. A data acquisition system (hybrid recorder) was used to collect and convert the entire data signals and then transmit the converted signals through a GPIB interface to the host computer for further operation. The energy balance between the air and tube side was ±5% for dry condition and ±7% for spray condition.

For spray condition, water was sprayed into the stream of air. The so-called "steady–state method" was employed in determining the heat transfer performance of the core. The water spray water system which includes water pump, air - compressor and 3 spray nozzles was introduced into the air tunnel about 0.5 m upstream of the heat-exchanger. The spray nozzles used were IKEUCHI two-fluid fan-shaped (type BIMV 8002) for small flow rate. The permitted spray rate was 1.0 to 10.2 L/hr depending on the pressure of air and water at the back of the nozzles, as shown in table 1. To obtain the average heat transfer coefficients h from the measured experimental data, the ε -NTU (effectiveness-number of transfer unit) method [17] under the cross overall counterflow with both fluids unmixed was applied.

THREE-DIMENSIONAL MATHEMATICAL ANALYSIS

Figure 3 designates the computational domain, and the coordinate system is also illustrated in the figure. The dimensionless equations for continuity, momentum, and energy may be express in tensor form as

$$\frac{\partial \overline{u_i}}{\partial x_i} = 0 \quad (1)$$

$$\frac{\partial}{\partial x_j}\rho(\overline{u_i}\,\overline{u_j}) = -\frac{\partial \overline{p}}{\partial x_i} + \frac{\partial}{\partial x_j}[\mu_{eff}(\frac{\partial \overline{u_i}}{\partial x_j} + \frac{\partial \overline{u_j}}{\partial x_i}) - \rho\overline{u_i' u_j'}] \quad (2)$$

$$\frac{\partial}{\partial x_j}\rho c_p(\overline{u_j}\,\overline{T}) = \overline{u_j}\frac{\partial \overline{p}}{\partial x_j} + \overline{u_j'}\frac{\partial \overline{p'}}{\partial x_j} + \frac{\partial}{\partial x_j}(k\frac{\partial \overline{T}}{\partial x_j} - \rho c_p\,\overline{u_j' T'}) \quad (3)$$

In this study, the k-ε turbulent model was introduced to simulate the flow field more accurately. The k-ε turbulent is shown as below

$$\frac{\partial}{\partial x_j}\left(\rho\overline{u_j}k\right) = \frac{\partial}{\partial x_j}(\frac{\mu_{eff}}{\sigma_k}\frac{\partial k}{\partial x_j}) + \rho(P_r - \varepsilon) \quad (4)$$

$$\frac{\partial}{\partial x_j}\left(\rho\overline{u_j}\varepsilon\right) = \frac{\partial}{\partial x_j}(\frac{\mu_{eff}}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial x_j}) + \rho\frac{\varepsilon}{k}[(c_1 + c_3\frac{P_r}{\varepsilon})P_r - c_2\varepsilon] \quad (5)$$

$$P_r = \frac{\mu_t}{\rho}[2(\frac{\partial u_i}{\partial x_i})^2 + (\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i})^2 - \frac{2}{3}(\nabla u_i)^2] \quad (6)$$

Because the governing equations are elliptic in spatial coordinates, boundary conditions are required for all boundaries of the computation domain. At the up-stream boundary, located the distance as tube outside diameter from the first row tube, uniform flow velocity u_{in} and temperature T_{in} are assumed. At the down-stream end of the computational domain, the Neumann boundary condition was applied.

At the solid surface, no-slip conditions and constant temperature T_{tube} are specified. At the symmetry plane, normal velocity and the temperature variation along the normal direction is set to be zero. The local heat transfer coefficient could be expressed in dimensionless form by the Nusselt number Nu, define as

$$Nu = \frac{h \cdot H}{k} = \frac{q'' \cdot H}{k \cdot (T_w - T_b)} = (\frac{\partial(\frac{\overline{T}}{\overline{T_b}})}{\partial n})_{wall} \quad (9)$$

Where k is the thermal conductivity of the fluid. To represent the pressure drop, the friction factor f was introduced as below

$$f = \frac{p_{in} - p}{\frac{1}{2}\rho u_{in}^2} \times \frac{H}{4L} \quad (10)$$

where p_{in} is the pressure at the inlet plane.

In this study, a body-fitted coordinate system along with a multiblock system was used to generate a general curvilinear coordinates system numerically by solving Laplace equations with proper control of grid densities. A control-volume-based finite-difference formulation was used to solve the governing equations. The system of finite-difference equations were solved iteratively using the SIMPLER algorithm [18]. A gird system, as shown as Figure 5, for the elliptic finned-tube bank was adopted in the computation domain. To ensure the accuracy and validity of the numerical results, three different grid systems, 158×12×8, 190×15×10, and 228×18×12 for the elliptic finned-tube bank were tested. It was found that the relative error for the total amount of heat transfer was less than 1.2%. Iterative computations were performed on a computer with Pentium 4 1.8G CPU. Typical CPU time was 3-4 hours for each case.

RESULT AND DISCUSSION

Experimental and numerical simulations of thermal-hydraulic characteristics of three types of elliptic finned-tube heat exchangers with different fin axis ratio (3.1, 2.8, and 2.5) were presented. The pressure drop and average heat transfer coefficient of circular finned-tube heat exchanger were also being compared with the experimental and numerical results of elliptic finned-tube heat exchangers.

The pressure drops of the three different axis ratios of elliptic finned-tube heat exchangers for different values of inlet frontal velocity are shown in Figure 6. As shown in Figure 6, the value of the pressure drop increased as a parabolic curve with an increase of the frontal velocity. In general, the experimental results are 18-23% higher than the numerical results. The pressure drop of the circular finned-tube heat exchanger is 2.4 times the value of elliptic finned-tube heat exchanger with axis ratio 2.5; 2.9 times the value of elliptic finned-tube heat exchanger with axis ratio 2.8; and 3.2 times the value of elliptic finned-tube heat exchanger with axis ratio 3.1.

Figure 7 presents the variations of the average heat transfer coefficient $\bar{h}$ with the frontal velocity for the three types of finned-tube heat exchangers, respectively. Also shown in the figure for comparison is the circular finned-tube heat exchanger data. It is shown that the numerical results are 32-45% higher than the experimental results. This is probably due to the fact that the actual boundary conditions for the tube surfaces in the experiment do not occur under constant wall temperature. As shown in the figure, the average heat transfer coefficient $\bar{h}$ of the circular finned-tube is about 49% higher than the value of the elliptic finned-tube heat exchanger with axis ratio 2.5; 52% higher than the value of elliptic finned-tube heat exchanger with axis ratio 2.8; and 56% higher than the value of elliptic finned-tube heat exchanger with axis ratio 3.1.

The numerical and experimental results for the heat transfer coefficient per unit pressure drop $h/\Delta P$ versus the inlet frontal velocity are shown in Figure 8. Both the experimental and numerical predictions indicate that the $h/\Delta P$ ratio decreases with the increase of the frontal velocity, and this ratio for elliptic fined tube is about 55-150% higher than for the circular finned tube. Figure 9 shows the heat enhancement while applying the air/water spray system under different air/water ratios. As shown in the figure, when the air/water ratio are 0.15% and 0.3%, the amounts of heat transfer are 16% and 33%, respectively, higher than the dry condition for the elliptic finned tube with axis ratio 2.5; 18% and 34%, respectively, higher than the dry condition for the elliptic finned tube with axis ratio 2.8; 25% and 38%, respectively, higher than the dry condition for the elliptic finned tube with axis ratio 3.1.

CONCLUSION

Experimental and numerical predictions of the thermal-hydraulic characteristics of elliptic finned-tube heat exchangers are presented. For the dry conditions, the pressure drop of circular finned-tube heat exchanger are respectively 2.4, 2.9 and 3.2 times the values of elliptic finned-tube heat exchanger with axis ratio 2.5, 2.8 and 3.1. The average heat transfer coefficient $\bar{h}$ of circular finned-tube are respectively about 49%, 52% and 56% higher than the values of elliptic finned-tube heat exchangers with axis ratios 2.5, 2.8 and 3.1. The heat transfer coefficient per unit pressure drop $h/\Delta P$ for elliptic fined tube is about 55-150% higher than for the circular finned tube. The heat transfer could be enhanced while applying the air/water spray system. When the air/water ratio are

0.15% and 0.3%, the amounts of heat transfer are 16% and 33%, respectively, higher than the dry condition for the elliptic finned tube with axis ratio 2.5; 18% and 34%, respectively, higher than the dry condition for the elliptic finned tube with axis ratio 2.8; 25% and 38%, respectively, higher than the dry condition for the elliptic finned tube with axis ratio 3.1.

ACKNOWLEDGEMENTS

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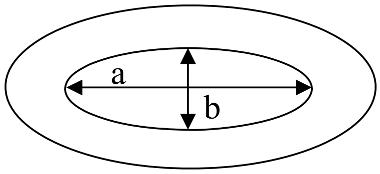
Table 1. Geometric Parameters of the Test Sections

	Circular Finned-Tube H.X.	Elliptic Finned-Tube H. X.		
Axis Ratio	1.0	2.5	2.8	3.1
Tube Outside Diameter (mm)	27	36.8×14.6	37.6×13.2	38×12
Width of Heat Exchanger (mm)	266	226	226	226
Fin Thickness (mm)	0.5	0.5	0.5	0.5
Fin Height (mm)	7	7	7	7
Fin Pitch (fins/in)	8	8	8	8
Transverse Tube Spacing (mm)	42	34	34	34
Longitudinal Tube Spacing (mm)	37	50	50	50
Tube numbers	24	32	32	32
Pass numbers	4	4	4	4
Hydraulic Diameter Re_{Dh}(mm)	6.72	7.63	8.33	8.83

Figure 1. Test Section



$$Ar = \frac{a}{b} = 2.5, 2.8, 3.1$$

Figure 2. Elliptic Finned-Tube Configuration

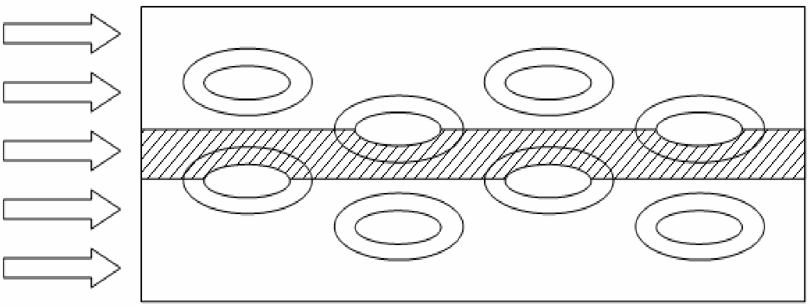
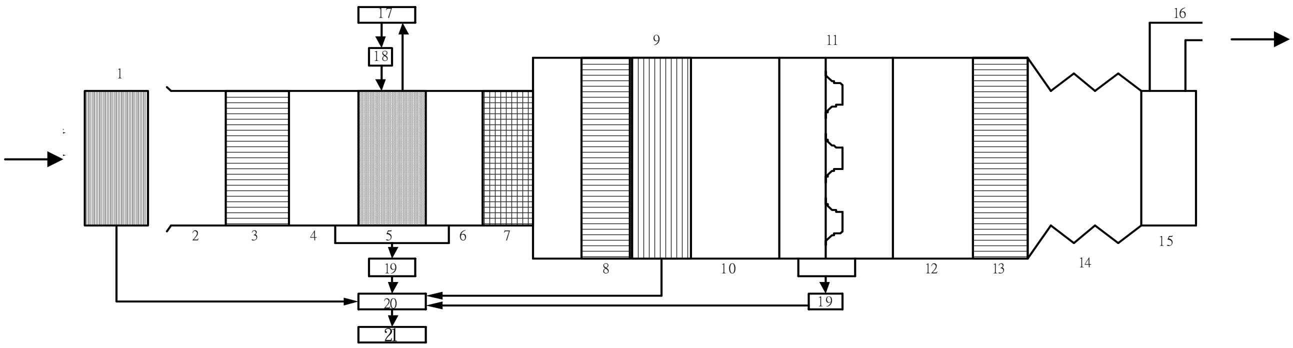


Figure 3. Computational Domain



11. air side inlet temperature measuring station
12. inlet area
13. air straightener
14. pressure tap (inlet)
15. test section
16. pressure tap (outlet)
17. air mixer
18. air straightener
19. air side outlet temperature measuring station
20. nozzle pressure tap (inlet)
21. multiple nozzles plate

1. nozzle pressure tap (outlet)
2. air straightener
3. flexible duct
4. variable exhaust fan system
5. discharge
6. thermostat reservoir
7. volumetric flow meter
8. differential pressure transducer
9. data acquisition system
10. working computer

Figure 4. Schematic Diagram of the Experimental Setup

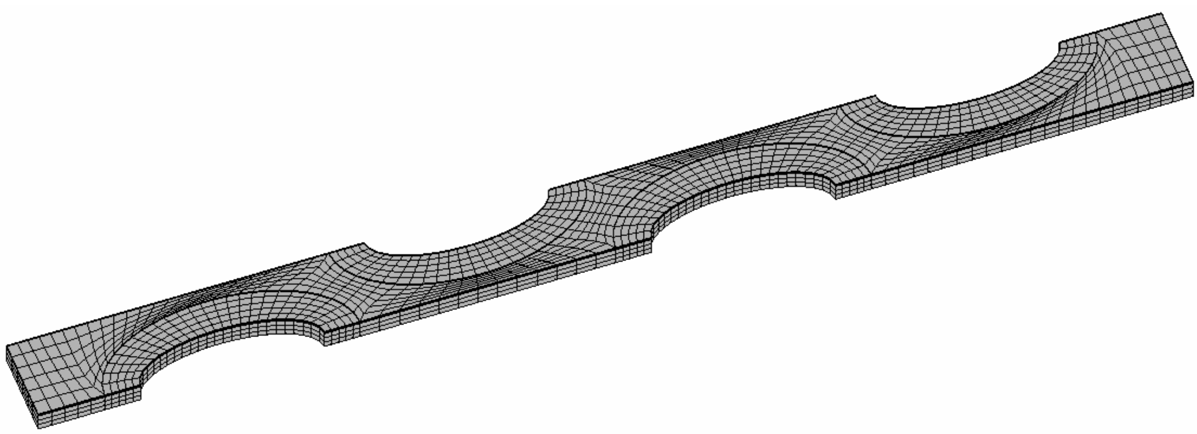


Figure 5. Computation Grid System for Elliptic Finned-Tube Banks

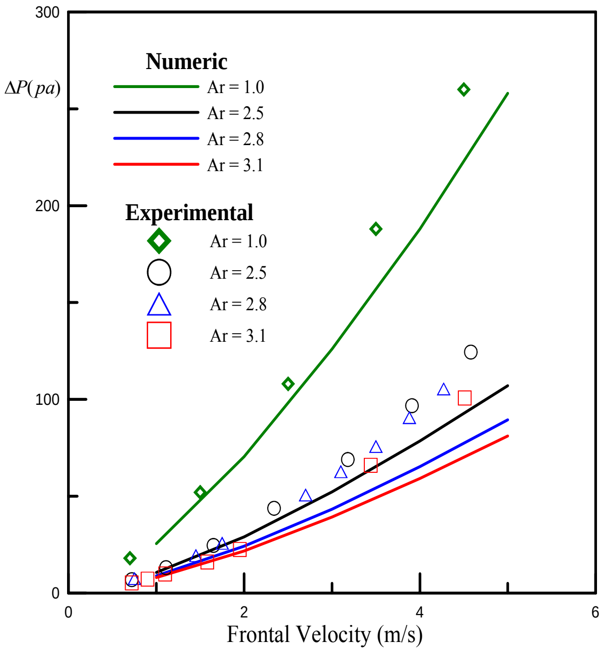


Figure 6. Pressure drops for different values of frontal velocity

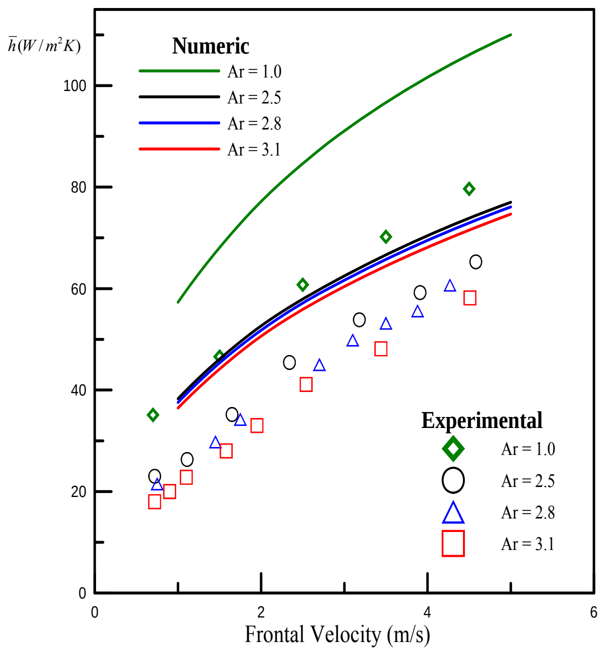


Figure 7. Heat transfer coefficient for different values of frontal velocity

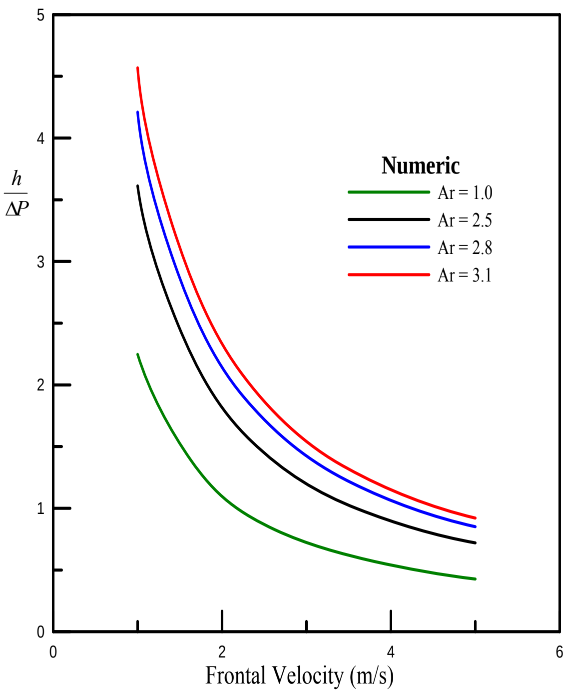


Figure 8. Heat transfer coefficient per unit pressure drop for different values of frontal velocity

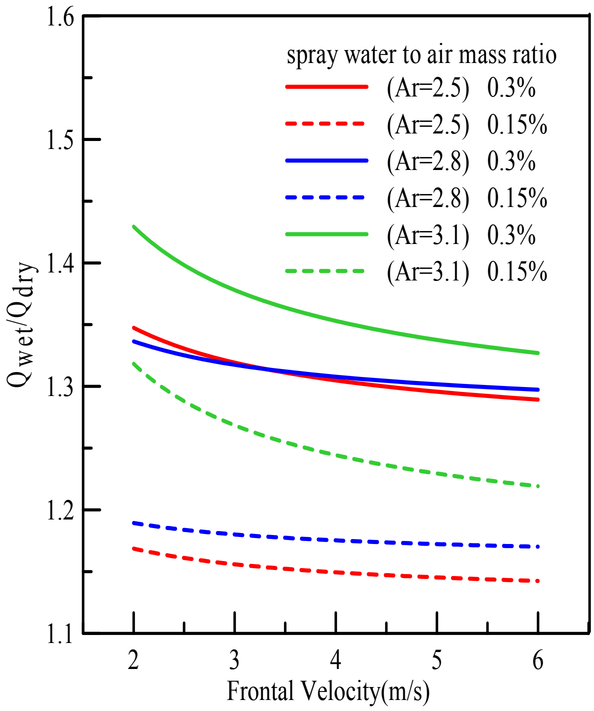


Figure 9. Heat enhancement by different axis ratios and spray ratios varies with frontal velocity

Temperature Distribution of Rotary Heat Recovery Units

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SUMMARY

This paper deals with measurements of temperature distribution of rotary heat recovery units. A special grid of thermocouples has been made to measure air temperature in equally distributed sampling points on each side of a heat recovery wheel. Air flow rates and the amount of recirculation air (leakage) were measured using a tracer gas. The results show that the temperature profile is generally dependent on both the rotational speed of the wheel and the flow rate. Measurements also show that the placement of temperature sensors in the duct can be significant, since wrong placements may lead to both faulty or non-optimized control of the units, and wrong readings of the temperature efficiency. Propitious sensor placements have been proposed.

INTRODUCTION

There are many types of air-to-air heat recovery units (HRU) which are in use in ventilation systems (see for instance Schild [1]). Rotary heat recovery exchangers are widely used, and the units are known for their high efficiency and trouble-free operation. Temperature efficiencies above 80% are not uncommon. For balanced ventilation systems, the recovered heat power can be at the same rate. Unfortunately these units are also infamous for transmission of polluted air from exhaust to supply. Due to the rotation of the device, there will always be a direct and indirect connection between exhaust and fresh supply air. Cross-contamination through rotary heat recovery units comes from two mechanisms; (1) Leakage, (2) Carryover. The former is dependent on the differential static pressure across the wheel, and the placement of supply and exhaust fans may affect its value significantly. The latter (carryover) occurs as a result of air being entrained within the wheel volume. Contaminated air is then carried into the other, clean air stream. The degree of carryover air is strongly dependent on whether the unit has a purge section or not. Rotor speed is normally relatively low and in a range of 3-15 rpm. The cross sectional temperature distribution on the supply or discharge side of a rotary HRU is in many cases non-uniform. This means that placement of temperature sensors for control or for calculation of energy transfer and efficiencies is crucial.

SITE DESCRIPTION AND MEASUREMENT SETUP

Flow and temperature measurements where performed in a laboratory test ventilation system (the heat recovery AHU part of the system is shown in figure 1). This is a ABB EC2000 unit with a non-hygroscopic rotary wheel. Flow rates where measured with a tracer gas. A tracer gas (SF_6) was released into the air, and concentrations upstream and downstream of the dosing points determined the flow rate and cross-contamination of the heat recovery unit (according to figure 2).

Figure 1. The AHU subsystem containing the rotary heat recovery unit

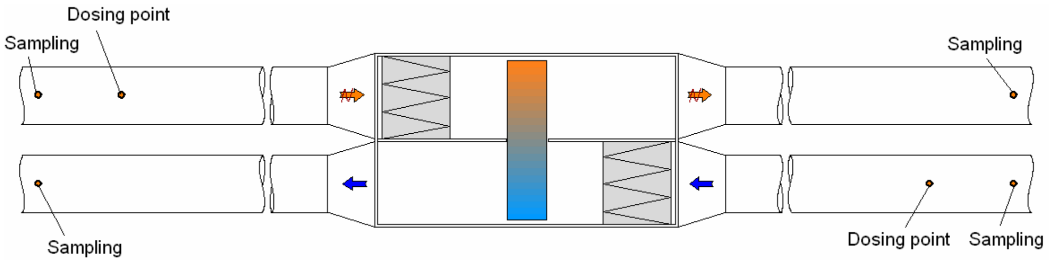

Figure 2. Flow measurement setup and tracer gas sampling points, Eian/Sørensen [2]

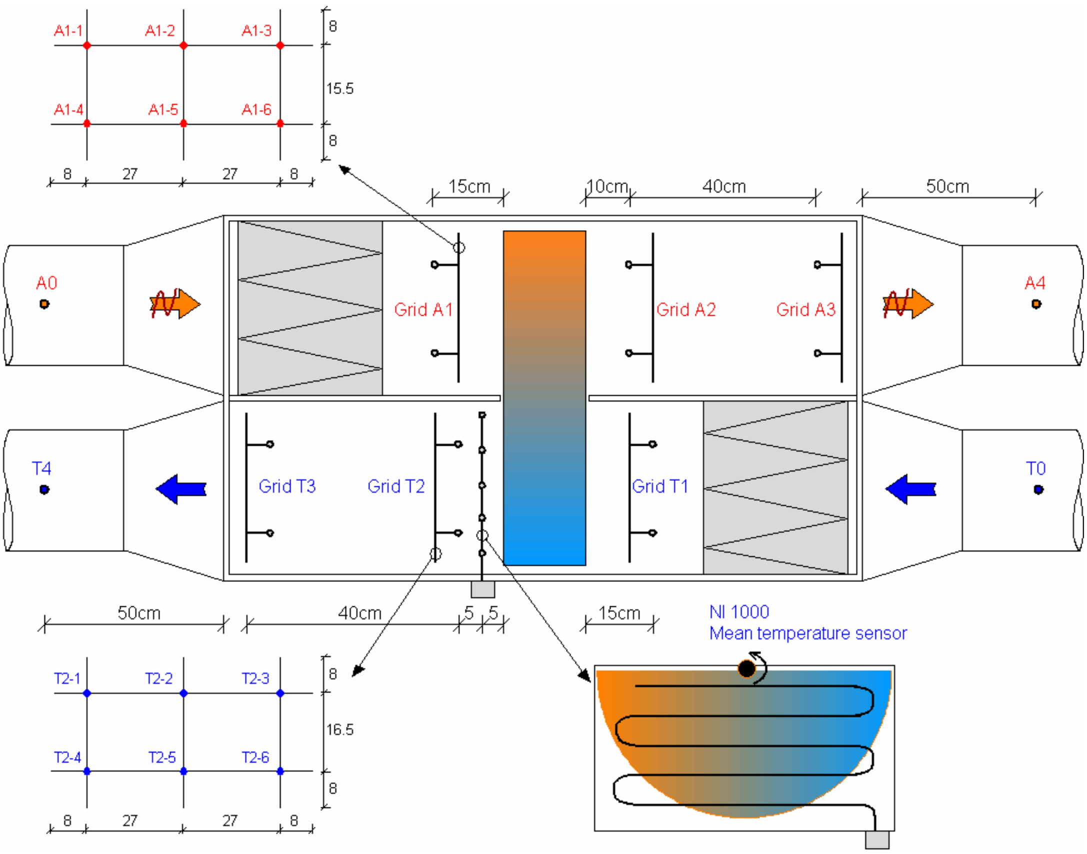

Figure 3. Temperature sampling points

The temperature measurement system (refer to figure 3) consisted of:

- Three grids of thermocouples on the intake and supply side (grid T1, T2 and T3), each containing 6 sample points (figure 3 and 4)
- Three grids of thermocouples on the exhaust and discharge side (grid A1, A2 and A3), each containing 6 sample points
- Temperature in the ducts T0, T4, A0 and A4.
- A NI1000 mean temperature sensor on the exhaust side (this sensor was not used in this study)

Figure 4. Picture of a thermocouple grid in front of the rotary heat recovery unit.

RESULTS AND DISCUSSION

Flow rates

Figure 5 shows the flow rates that were measured for different speeds of the fan and heat recovery unit. Figure 6 shows the percentage return air being recirculated to the supply side.

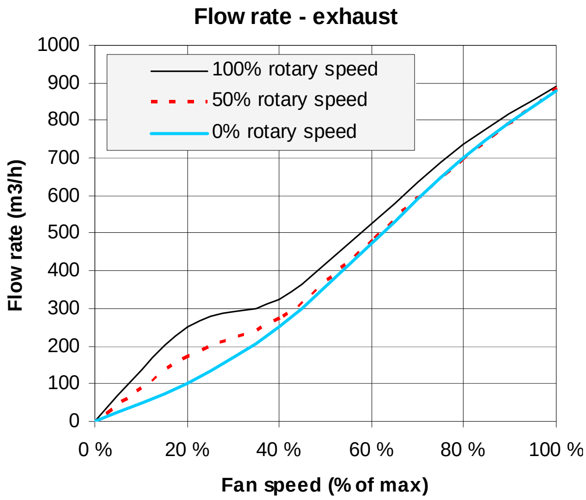

Figure 5. Flow rates measured in the exhaust

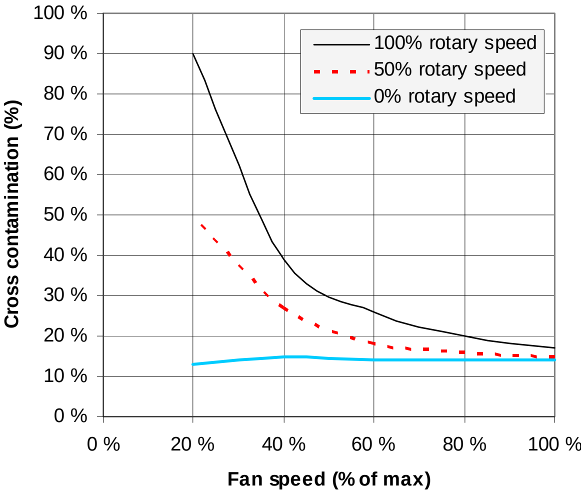

Figure 6. Cross contamination (%)

A large amount of recirculated return air was measured. The large amount of return air was mainly a result of the rotary HRU having no purge sector and an unfavorable placement of the fans in the system. This resulted in a low static pressure on the intake side of the HRU and a relative high pressure on the exhaust side, forcing air to cross to the supply side. The latter caused a 12-15% air recirculation. It was also noticed that the amount of recirculation increased significantly with increased speed of the HRU wheel. This indicates that carryover air stands for a large amount of total recirculation. Furthermore, at low fan speeds, the relative amount of recirculated air increased significantly. While the fans were running at only 20% of maximum speed, and the HRU was at full speed, there were almost no fresh outdoor air on the supply side (90% recirculation). Thus this confirms the importance of obtaining proper pressure conditions around the rotating wheel. The results also suggest that for VAV systems running on low fan speeds, the amount of return air can become very high and thus unacceptable.

Measurements with the HRU not rotating show an almost constant amount of recirculated air (independent of fan speed). Hence, the increase of recirculation was mainly caused by carryover air in the rotating wheel. Furthermore, the amount of carryover air from the supply to the exhaust was quite noticeable, as can be seen in figure 5 (since flow rates on the exhaust side varied with the speed of the rotating wheel).

Similar measurements were performed on the supply side of the HRU, showing that the ventilation system was very good balanced with respect to supply and exhaust flow rates.

Temperature measurements

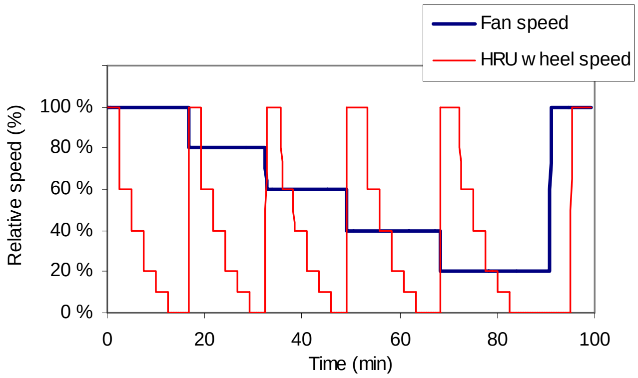

Figure 7: Fan and HRU wheel speeds (% of max) during temperature measurements

Two measurements series were performed, each resulting in responses from altered fan speeds (0-100% of max, 20% step) and HRU wheel speeds (0%, 10%, 20%, 40%, 60% and 100% of max), see figure 7. The outdoor temperature was fairly constant around 0°C. Figure 8 shows the temperature measurements as a function of time from grid T2 (supply side, closest to the wheel) for measurement series 2. At a first glance, it is obvious that the temperature varied quite noticeable across the section/grid for a given fan and HRU wheel speed. If looking closer it is possible to see the variations in temperature due to the alternations of the wheel rotary speed. The most conspicuous temperature drops happened when the rotary HRU was shut down to 0% speed. When this occurred, the temperature on the supply side naturally dropped towards the outdoor temperature.

In figure 9-12, the results have been arranged so that they correspond to the location of the actual sampling point. The two upper rows of graphs correlate to the grids on the supply side of the HRU wheel, grid T2 and T3, while the two lower rows of graphs correlate to the grids on the discharge side, grid A2 and A3. In addition, the duct temperatures (T4 and A4) are plotted on the supply and discharge graphs respectively. Since these sampling points are located away from the wheel (in the ducts), they are used as references to decide whether a temperature value is too low or too high in the grids. Each graph has been numbered in accordance with the setup given by figure 3. The direction of the HRU is also shown. Figure 9 and 10 show steady state temperature levels (according to the variations in fan speed and wheel speed defined by figure 7) for each sampling point at 20% respectively 100% fan speed, as a function of the relative HRU wheel speed. Likewise, figure 11 and 12 show temperature levels for each sampling point at 20% respectively 100% HRU speed, as a function of the relative fan speed.

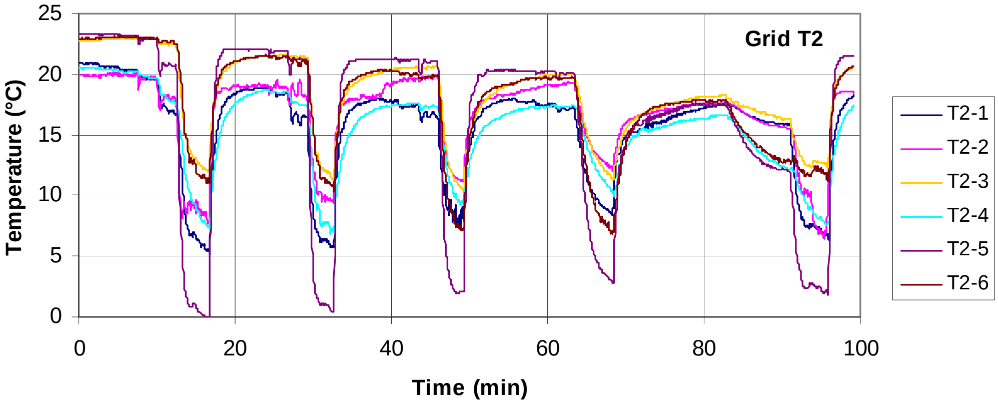

Figure 8. Temperature measurements, grid T2 (supply side, close to wheel).

From figure 9 and 10 it can be seen that the HRU speed is of minor importance as long as it is above 20% (in this case, 20% speed corresponds to an absolute speed of 2.25 rpm). Furthermore, while the fans run on low speed (20% of max) the cross sectional stratification bedding is small on the supply side, and the temperature distribution is fairly uniform. As the HRU speed is increased, temperatures on the supply side decrease slightly. Most likely this is caused by the large increase of recirculation air (refer to the flow measurements discussed earlier). However, in a system with an efficient purge sector this tendency will not show. On the discharge side, the distributions are more scattered, with highest values near the center line of the wheel.

When the fan speed is increased (figure 10), the cross sectional temperatures on both the supply and discharge sides of the wheel become increasingly scattered, and a cross sectional stratification is revealed. In this situation, the air is warmest near the center line of the wheel.

From the next two figures (figure 11 and 12) it can be seen that the fan speed of course affect the temperature levels significantly. While the fans run on low speed (20% of max) the cross sectional stratification bedding is small on the supply side, and the temperature distribution is fairly uniform. There is however a tendency of high temperatures near the center of the unit and low temperature around the edges. On the discharge side, the warmest air is near the edge of the wheel, close to the center line. Table 1 shows a simple summary of the temperature levels from figure 9-12, as a color map of grid T2 and A2. Levels are shown as +/- differences from the duct levels T4 and A4, which are believed to be more representative for the actual air temperature.

Table 1. Map of temperatures (°C) close to HRU wheel (supply and discharge side).

Fig. 9, fan 20% Varying HRU			Fig. 10, fan 100% Varying HRU			Fig. 11, HRU 20% Varying fan			Fig 12, HRU 100% Varying fan			
+1	0	-1	+1	-2	-1	+1	-2	-1	0	-1	-1	Supply
0	0	-1	+1	+2	-1	+1	+2	-1	0	+1	-3	
+2	0	+2	+6	+2	+8	+6	+2	+8	+5	+2	+8	Discharge
0	-2	0	+6	-3	0	+6	-3	0	+3	-2	+1	

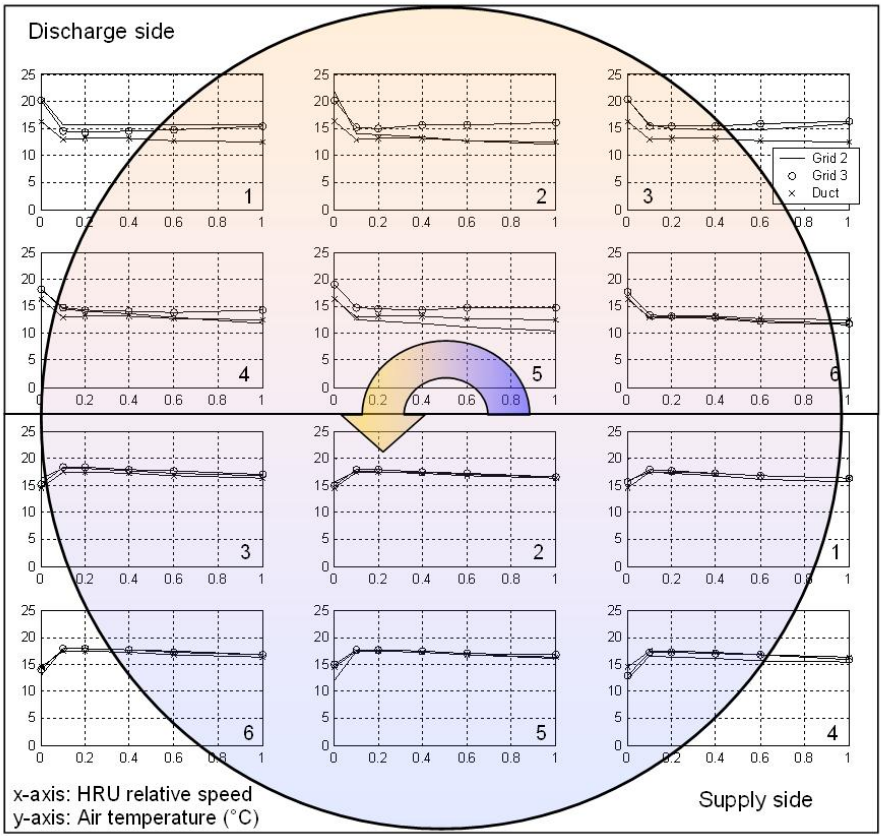

Figure 9. Air temperatures, T2-T4 and A2-A4 (supply and discharge sides) as a function of HRU rotational speed at 20% fan speed (HRU view from intake side).

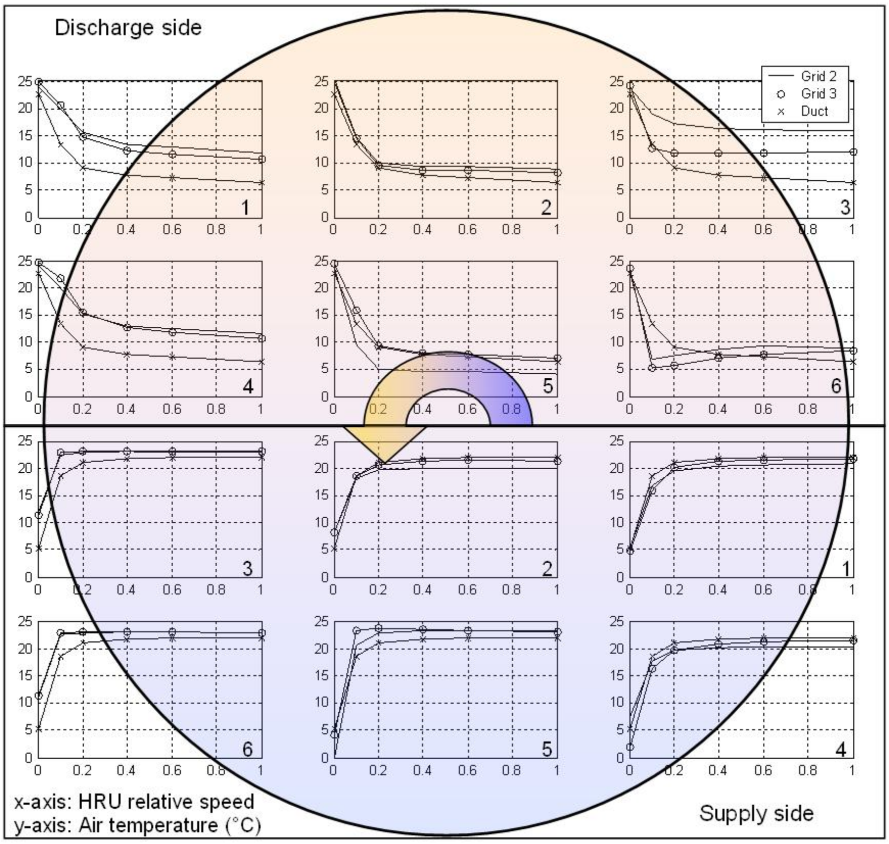

Figure 10. Air temperatures, T2-T4 and A2-A4 (supply and discharge sides) as a function of HRU rotational speed at 100% fan speed (HRU view from intake side).

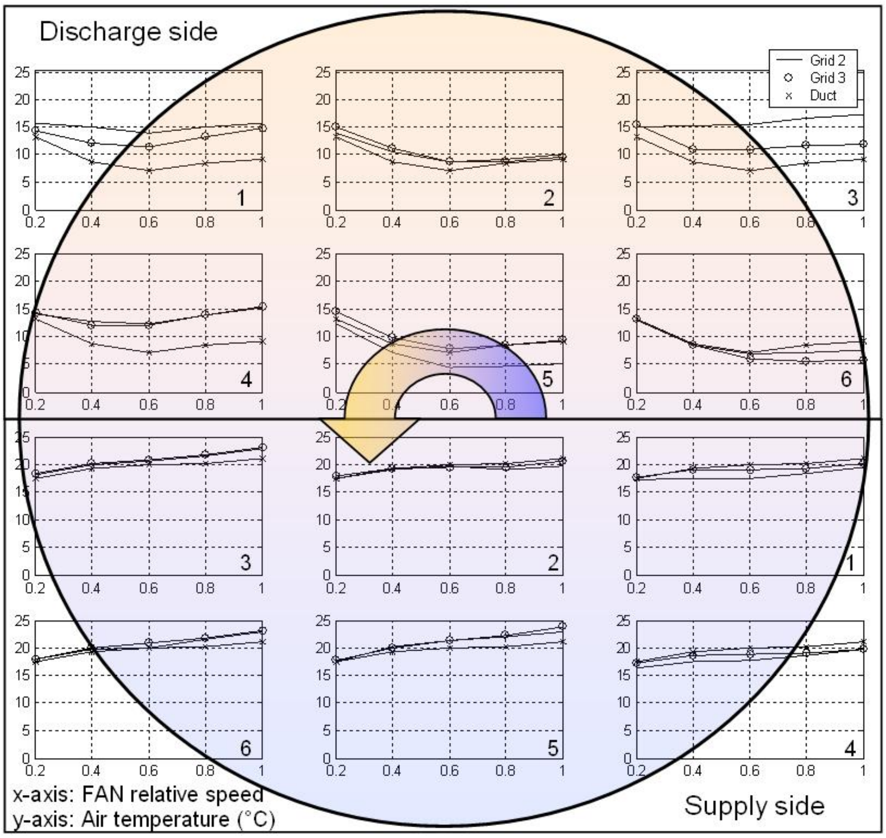

Figure 11. Air temperatures, T2-T4 and A2-A4 (supply and discharge sides) as a function of relative fan speed at 20% HRU rotational speed (HRU view from intake side).

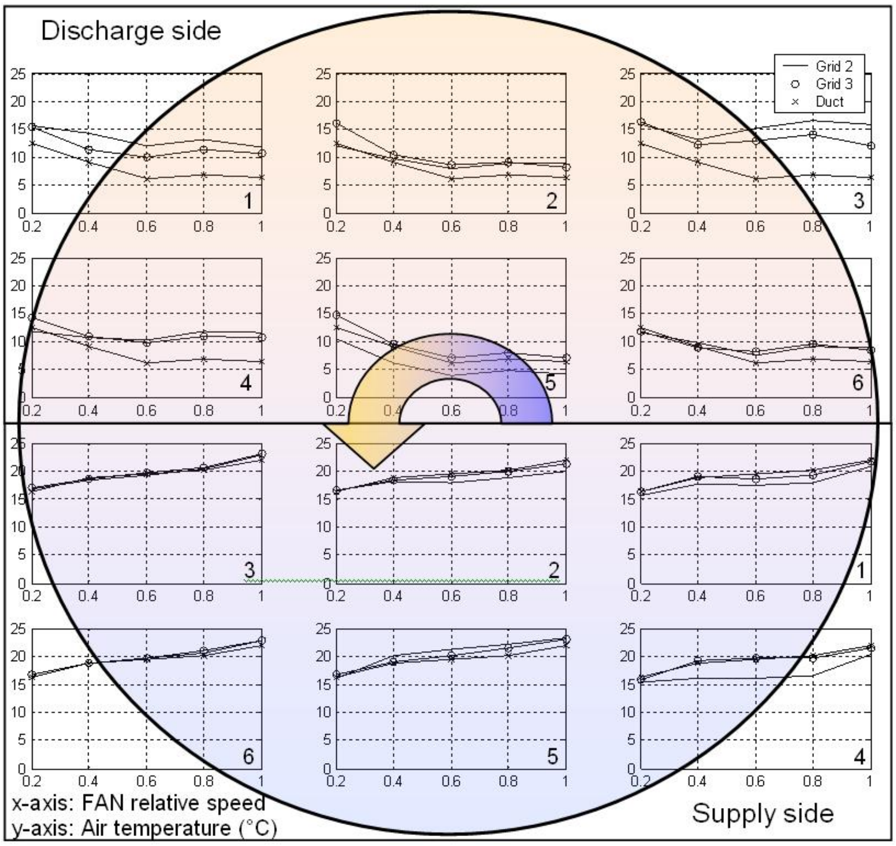

Figure 12. Air temperatures, T2-T4 and A2-A4 (supply and discharge sides) as a function of relative fan speed at 100% HRU rotational speed (HRU view from intake side).

Placement of temperature sensors in the duct

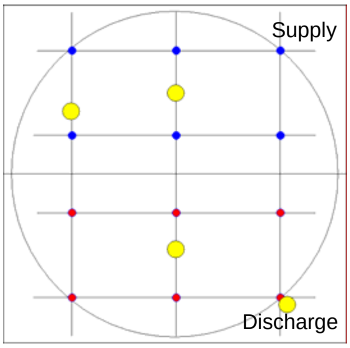

Figure 13. Placement of sensors

The ideal placement of a temperature sensor used for control or efficiency calculations is away from the wheel so it is not affected by stratified air nor by radiation from the wheel media. However, this is rarely convenient or possible, so the compromise must be to find a spot near the wheel (on all sides) that approximately resembles the actual air temperature as if it was measured away and shielded from the device. From table 1, an assessment of sensor placement is possible. For this actual HRU wheel it seems most appropriate to place the supply side and discharge side sensors as shown by the yellow spots in figure 13. It should also be mentioned that the temperatures on the exhaust and intake side of the wheel were fairly uniform, but small variations can be expected due to radiation from the wheel and in cases with leakage. A similar assessment may be required here.

Possible sources of error

Measurements should be carried out on several different heat recovery units. Also, the number of repeated measurements series are low, and it is thus not possible to establish a basis for categorical conclusions. Nevertheless, the results serve to pinpoint typical tendencies.

CONCLUSIONS

The flow rate measurements results suggest that for ventilation systems running on low fan speeds (VAV), the amount of recirculation of air from the exhaust to the supply can, due to *carryover*, become very high and thus unacceptable.

The rotational speed of the heat recovery unit has shown to be of minor importance as long as the wheel is rotating. For this particular case, above 20% of maximum speed.

The ideal placement of a temperature sensor used for control or efficiency calculations is away from the wheel so it is not affected by cross-sectional stratification bedding, nor by radiation from the wheel media. For the particular HRU in question, measurement results suggest that the sensors on the supply and discharge side should be placed approximately in the middle of the AHU duct, or displaced to the sides.

FURTHER WORK

This work represents the first report in a project on air-to-air heat recovery devices carried out at Narvik University College. Future work will address temperature and energy efficiency, transient conditions, control, modelling and simulation.

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Using field synergy theory to discuss the performance of mass transfer in dehumidifying air-conditioner

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ABSTRACT

This paper analyses the problem of improving the performance of the mass transfer withthe standpoint of field synergy theory. It mainly studies with the view that adjusting an angle between velocity field and density field could enhance the performance of dehumidifying air-conditioner. At the same time, it cites one kind of the structure of screw type beehive wad dehumidifying airconditioner and screw baffler dehumidifying air-conditioner which is fixed with an angle, validates its performance of intensifying mass transfer and discusses the extensive applications in engineering. The elementary result is that the mass transfer effect will be better when the screw inclination angle is between 25°and 45°.And the total mass transfer coefficient could enhance 20% 30% in most cases.

Key words: dehumidifying air-conditioner ,field synergy theory, convection and mass transfer

INTRODUCTION

Field synergy theory [1] pointed out that the size of the heat transfer intensity was decided not only by the gradient of temperature, the speed and the nature of the fluid but also by the synergic degree of the velocity field and the hot flow field . Namely, The fluid flows may both strengthen the heat transfer coefficient, and weaken the heat transfer coefficient, so changing and controlling the synergy of the speed of flow and the heat flow may control the intensity in the convection of heat transfer. The mass transfer process has also occurred in field synergy theory. Under the same condition, changing the included angle of the velocity field and the concentration gradient field may enhance the mass transfer intensity. This article uses the field synergy theory's viewpoint of mass transfer to re-know and study the performance of dehumidifying airconditioner, and through improving the synergic degree of the fluid velocity field and the concentration gradient field enhance the dehumidifying air-conditioner's performance. It is applied into the practice of projects.

THE BASE OF MASS TRANSFER'S FIELD SYNERGY THEORY

Regarding to the three dimensional and the stable state's convection mass transfer, its differential equation is:

$$u\frac{\partial \rho_A}{\partial x}+v\frac{\partial \rho_A}{\partial y}+w\frac{\partial \rho_A}{\partial z}=D\left[\frac{\partial^2 \rho_A}{\partial x^2}+v\frac{\partial^2 \rho_A}{\partial y^2}+w\frac{\partial^2 \rho_A}{\partial z^2}\right] \tag{1}$$

Among the above formula, u v w = the velocity of x, y, z -dimension's coordinates , m/s, D = the diffusion coefficient of part A m²/s and ρ_A = the diffusion coefficient of part A kg/m³.

According to the boundary layer theory, the density boundary layer's convection and mass transfer quantity is equal to close-wall layer's diffusion mass transfer quantity, formula (1) is integrated from 0 to the boundary layer thickness δ_C in rectangular coordinates y, according to the density boundary layer and write down in the vector description form as follows:

$$\int_0^{\delta_c} U\nabla\rho_A dy = \cos\alpha \int_0^{\delta_c} |U||\nabla\rho_A| dy = -D_{grad}\rho A\big|_0 \qquad (2)$$

In the above formula,

U = the velocity vector of the close-wall layer's mobile fluid m/s
$\nabla\rho_A$ = fluid's concentration gradient kg/m³
α = the included angle of the velocity vector and concentration gradient vector
δ_C = the thickness of boundary layer m
D_{grad} = the diffusion coefficient of concentration boundary layer m²/s

In the above equation ,the vector form of source equivalent interior production rate is:

$$U \bullet \nabla\rho_A = |U| \bullet |\nabla\rho_A| \cos\alpha \qquad (3)$$

From the above equation, we can concluded that the dot metrix size has relation to the module of the two vector and their included angle. Under the condition of invariable mold, the smaller the included angle is, the larger the scalar product is. Namely, with the same energy consumption of mass transfer, we adjusts the included angle α between the velocity field and the density gradient field to achieve the goal of strengthening mass transfer.

THE FIELD SYNERGY THEORY OF DEHUMIDIFYING AIR-CONDITIONER'S MASS TRANSFER

The mass transfer model of two dimensional laminar flow

In the same temperature, stable state, non-internal production rate, humid air will produce the laminar boundary layer when wet air has flowed the absorbent surface in the laminar way, as shown in Figure 1, which belongs to the two-dimensional convection and mass transfer situation. Among them, the air velocity changes only along x direction, the concentration increases along the y direction, integrates the both sides to coordinates y according to its corresponding differential equation:

$$\int^{c} U \bullet \nabla\rho_A dy = D\frac{\partial\rho_A}{\partial y} + C \qquad (4)$$

In the above formula, the character C is constant , the fluid velocity is along horizontal direction, the concentration gradient is vertical upwards, and the included angle between the two vector is 90 degree.

From $U \bullet \nabla\rho_A = |U| \bullet |\nabla\rho_A| \cos 90° = 0$, we can see that the mass transfer quantity that is caused by the equivalent source item is zero. When diffusion coefficient D is invariable, the concentration was taken on linear distribution along the y direction in the boundary layer.

This kind of situation equates the simple diffusing phenomena which proliferates mutually in the quality. Namely, through close-wall layer, the mass transfer quantity was only able to be the pure diffusing phenomena with the correspondence of mass transfer quantity, convection has not any function to strengthening the mass transfer, in other words, fluid flowing in this time was the same as the static situation. As a result, dehumidifying or desiccating equipment can be optimized from this period and it has further instructed significance for energy saving.

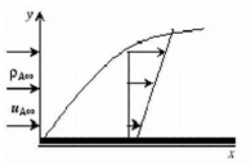


Figure1.

Extremely mass transfer model

When laminar humid air vertically flows from the porous big area plate, as shown in Figure 2, the incident flow passes through in turn from left side and right side. By now, on both sides of the board included: the left side of the boundary layer has hole pumping effect, its thickness attenuates inevitably Right side of the boundary layer has hole injector head effect, thickness will possibly increase a lot. The research indicates [3], on the left side of the plate, the boundary layer's velocity direction is parallel to the concentration gradient direction but their direction is opposite, and the velocity direction was the same as the mass transfer direction (mass transfer), it has prevented the boundary layer from accumulating, but it has the hole pumping effect and causes the boundary layer to become much thin, thus strengthens the mass transfer in the greatest degree, this is one kind of ideal situation on the right side of the plate, the boundary layer's velocity direction is parallel to the concentration gradient direction and their direction is the same , the velocity direction was opposite to the mass transfer direction (mass transfer), but has the hole pumping effect and causes the boundary layer to become much thick, thus weakens the mass transfer in the greatest degree, decreases the mass flow density, makes the mass transfer to be weak, its effect is so far inferior to the static mass transfer effects.

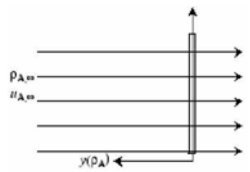


Figure 2.

MODEL DESIGN AND APPLICATION

Discussion

According to the mass transfer field synergy theory, we can conclude that the velocity field and the density field which fixed a suitable included angle will be beneficial to strengthen mass transfer effect. This article is going along with a model design in view of the dehumidifying airconditioner present research situation and proposing the application tentative plan.

4.1 Design the dehumidifying airconditioner's beehive wad as the screw type

The common dehumidifying airconditioner with beehive wad was beehive wad straight channel , speaking to the dehumidifying air-conditioner's beehive wad straight channel, fluid velocity gradient is vertical to the concentration gradient, their vector included angle α is 90 degree, according to the formula $U \bullet \nabla \rho_A = |U| \bullet |\nabla \rho_A| \cos 90° = 0$, the mass transfer quantity is only correspondence to the pure diffusing phenomena mass transfer quantity, convection has not any effect to strengthen mass transfer, just as one kind of static phenomenon Presently we designs it as the screw channel (as shown in Figure 3, the contrasts of the two kind channels as shown in Figure 4). Due to the certain included angle between the dehumidifying air-conditioner screw beehive wad 's velocity gradient field and concentration gradient field, namely, $\cos\alpha \neq 0$ thus we can greatly enhance the dehumidifying airconditioner's mass transfer performance. Literature [4] has given the moisture absorption capacity's research study when the air flow through the straight channel and the screw type channel. And their tentative data curve are shown in figure 5. See from Figure 5, under certain air flow velocity, the moisture absorption capacity of the screw channel is far higher than the straight channel's. Thus it validates that the screw channel's dehumidifying effect is distinctly enhanced.

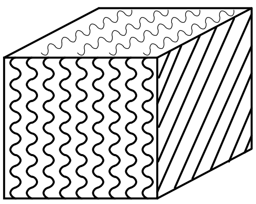
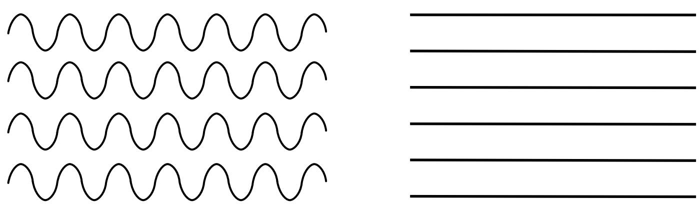


Figure3 the screw beehive wad.



(a) straight channel. (b)screw type channel.

Figure 4 the beehive wad sketch map.

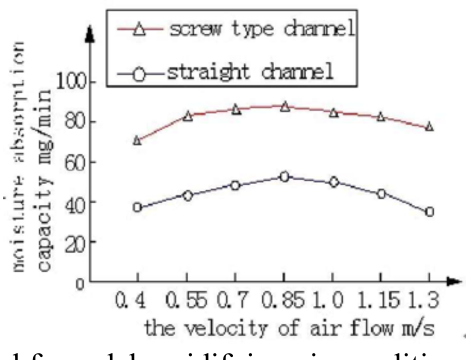


Figure 5 The different channel form dehumidifying air-conditioner's moisture absorption capacity along with the velocity of air flow change curve.

4.2 Design the dehumidifying air-conditioner as the screw type baffle plate fixed with inclination angle *β*

Regarding to the situation mentioned above, it is very difficult to realize in the actual project when the laminar humid air vertically flows through the porous plate of the extremely mass transfer model situation. Because the air vertically flows through the two-sided perforated plate situation, although having holes, it enormously increases the fluid flow's resistance. Considering to designing the form of an inclination angle between the board and the air flow direction in screw type baffle plate, as shown in figure 6, it may cause an included angle β between the velocity gradient and the concentration gradient in the screw type channel's columnar air flow ,thus influences the boundary layer's formation, strengthens dehumidifying air-conditioner's mass transfer performance, and the smaller the included angle is ,the better performance of mass transfer is.

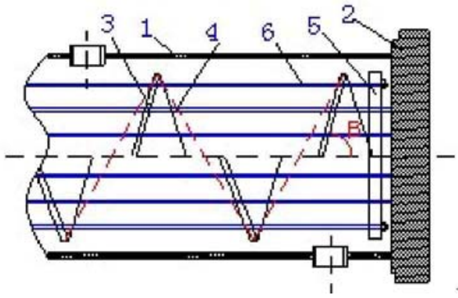


Figure 6 The screw baffle plate sketch map. 1-shell body 2- Tube plate 3-baffle plate 4-spoiler plate 5- Base plate 6-distance tube β-screw angle.

In relative to the traditional vertical arc plate, the screw type baffle plate has completely changed 180°time after time to turn back the transverse mobile way, make the fluid cause an included angle between velocity field and the density field , attenuate the boundary layer, reduce the thermal resistance, thus greatly enhance the mass transfer efficiency. It is suitable to deal with the solid particles, dusts, silts containing in the fluid. According to the related literature reported, the total mass transfer coefficient could enhance 20% -30% in most cases byusing the screw type baffle plate of dehumidifying air-conditioner, especially when the screw inclination angle is 25° - 45° the mass transfer effect is better.

CONCLUSION AND APPLICATION

It is significant to analyze the dehumidifying air-conditioner's mass transfer problem with field synergy theory. Because the convection and mass transfer process of the dehumidifying air-conditioner happens under the velocity field and the density field's together action, when fluid flow through the dehumidifying air-conditioner, we can make use of the included angle between the concentration gradient and the velocity gradient to control the mass transfer intensity. Under certain conditions, we design dehumidifying air-conditioner to adopt to have a certain angle screw beehive wad structure and screw type baffle plate fixed with an inclination angle of the dehumidifying air-conditioner form, the research indicated that it will be actually able to enhance the mass transfer performance. As its design is not very complex, it is suit for large scale production and promotion, it also has a broad prospect in the field of application.

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Efficiency Investigation of an Induction Motor drive system with Three Different Types of Frequency Converters with focus on HVAC applications

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SUMMARY

This paper discusses the efficiency of induction motor drives for HVAC applications, based on measurements made on three 4kW induction motor drives. Both converter efficiency as well as induction motor efficiency is studied and in particular the total system efficiency is determined. Measurements have been performed on a 4kW induction motor fed by three different types of frequency converters. Converters A and B use an open loop, constant flux control, where B has a L-C filter on its output. Converter C uses an open loop, field weakening algorithm but also a special pulse width modulation (PWM) technique in order to reduce the switching losses in the converter. Simulations and measurements have shown that an efficiency improvement of the system can be achieved using a field weakening algorithm which improves the efficiency at light load with 18% at 30Hz which in particular is of importance in HVAC applications. At full load the difference in efficiency is negligible between system A and C but 3% lower using system B due to the sinus filter. Efficiency improvements were also accomplished by using the special PWM technique. It was also found that it is important to make a complete optimization of the energy efficiency. Using system B, with L-C filter, gives higher machine efficiency but a low overall efficiency due to high converter losses. As a general conclusion, it has been shown that system C was the best solution from an efficiency point of view for a typical HVAC application.

INTRODUCTION

Electrical machines for HVAC applications contributes to approximately 40% (320TWh) of the energy consumption of electrical machines in EU(1996) [1]. The flow from pumps and fans are often mechanically controlled by valves, which leads to a reduced efficiency of the system. A better approach is to allow the pump/fan to determine the flow [2]. This is done by using a frequency converter, adjusting the voltage and the frequency of the motor to meet the demand of the load. This type of drive system has become more common throughout the years due to the energy savings that can be made. The energy efficiency of an induction machine (IM) is relatively high at optimal load and speed. However, in many situations, the load is not optimal. A common practice regarding dimensioning of drive systems are to use a 10% marginal and then choose the next available size above this limit [3]. As a result, the drive system can in some cases become highly over dimensioned and will not be operating at its optimum. Furthermore, drive system used in HVAC applications are often season/weather dependent and will be operating at light load during extended time periods [4]. It is therefore of great importance to optimize the energy efficiency over the whole operating range. Hence, it is of interest to investigate efficiency improvements, both in the frequency converter but also in the IM which can be done with design and control of the system. However, this paper will be focused on the issues regarding control. There are several control techniques in order to minimize the losses in the converter. Techniques related to the minimization of the switching losses have been proposed in [5, 6]. However, the described technique in this paper is only mentioned briefly.

Efficiency improvements of the IM can be made using optimal volt/frequency control which is described in detail in [4] and will just be mentioned briefly in this paper.

The purpose of this paper is to investigate the different loss components in a drive system consisting of an IM fed by a frequency converter. Moreover a goal is to use measurements and simulation to determine the difference using three types of converter technologies.

SYSTEM DESCRIPTION

Measurements where made on three different types of frequency converters feeding an IM. The different systems are referred to as A, B and C.

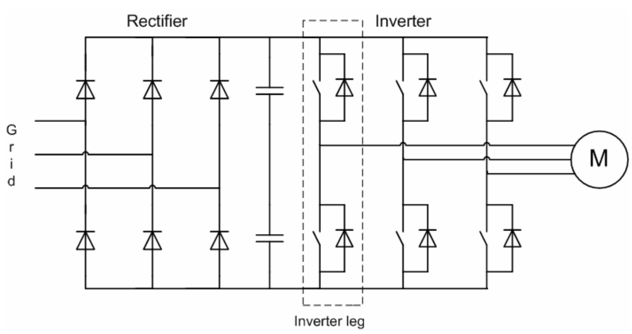
The frequency converter used in system A was a 4kW converter using constant voltage/frequency ratio.

Converter B contained a 4kW frequency converter with same control as system A. The converter was also equipped with an internal L-C filter on its output, which reduced the harmonic content significantly compared to system A and C.

Converter C used a field weakening algorithm but also a special pulse width modulation (PWM) technique in order to reduce the switching losses in the converter. At each half period the switching was stopped for 60° when the phase voltage was around its peak voltage.

LOSS COMPONENTS IN AN INDUCTION MOTOR DRIVE

The drive system of consideration is an IM fed with a frequency converter, shown in figure 1. The different losses that occur in the IM are resistive losses in the stator and rotor, core, mechanical harmonic and stray losses.



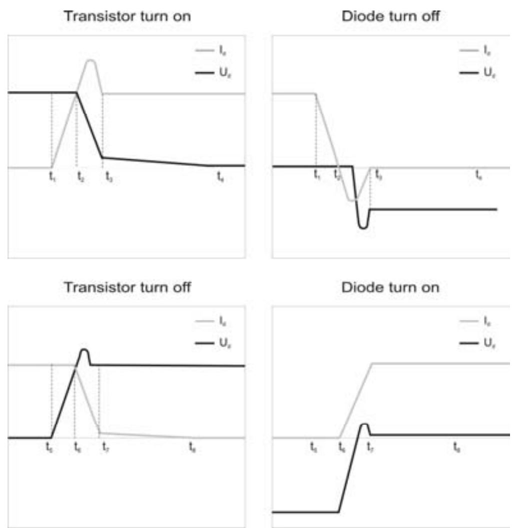


Figure 1a) Schematic figure of a frequency converter feeding an IM.

b) Turn on and turn off voltage and current waveforms

These losses has been measured separately according to the IEEE standard 112-1991 method B [7].

The different loss components in a frequency converter are much more complicated to measure/estimate due to the compact construction of the device. Hence, the measurement in this study did not consider these losses separately. Instead the total loss components of the frequency converters were measured. The loss components are on state losses in the diode rectifier and in the inverter stage and switching losses in the inverter stage.

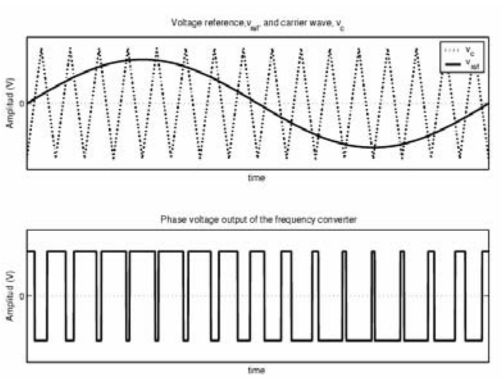
IMPROVEMENTS IN ENERGY EFFICIENCY

The improvement in energy efficiency can be divided into two broad groups, the design of a drive system and the control of the system. This section will describe the factors that improve the energy efficiency related to system C which both falls under the category of control.

PWM switching strategy

The most commonly used control technique in converters is Pulse Width Modulation (PWM). The PWM voltage is a pulse train of fixed magnitude and frequency with variable pulse width. The pattern is created by comparing a modulating carrier wave with a reference wave. If the frequency of the carrier wave is increased the switching frequency of the converter also increases. The choice of switching frequency are balance between switching losses in the converter and harmonic losses as well as torque pulsation in the IM, (of course physical limitations of the device must be taken into account). Furthermore, the switching losses also depend on the load type, which in this case is inductive. Typical voltage and current waveforms for turn on and turn off of an inverter leg feeding an inductive load can be seen in figure 1b.

This paper will investigate a discontinuous PWM technique which stops the switching at different time durations during the peak voltage, figures 2a) and b) shows examples of the continuous and discontinuous PWM scheme respectively. However, during continuous PWM sampling, at the so called over modulation, the switching stops for a certain time interval, ie when the voltage reference amplitude is higher than the carrier wave. This occurs at frequencies of approximately 40Hz for the investigated case when constant voltage/frequency ratio is used.



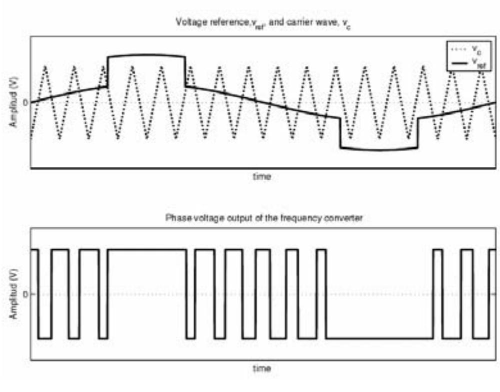


Figure 2a) Continuous PWM b) Discontinuous PWM

Optimal Voltage/frequency control

Improvements in energy efficiency of the IM can be made with different control strategies. A simple control strategy is the constant air-gap flux control which keeps the ratio between voltage and frequency (V/Hz) constant at all loads, (system A and B). However, every loading situation can be achieved with various combinations of voltage amplitude and frequency and it can be shown that there is an optimal frequency and an optimal voltage at each loading point [8,9]. The goal is to control the voltage and the frequency in order to optimize the balance between the copper and iron losses [4]. Different control strategies can be divided into three groups, *Power factor control* [10,11], *model based control* [12-14] and *search method* [15-18]. Evaluation and description of the different methods can be found in [4]. A previous work [4] has shown that the improvements in efficiency has a larger effect at light loads and are therefore suitable for HVAC systems where the power demand varies with the square or the cube of the speed. Furthermore, many HVAC applications, as mentioned in the introduction, can be assumed to operate on reduced load for a long period of time. Furthermore, over dimensioning of HVAC system also contribute to lower efficiency in constant air-gap flux control since the motor now constantly is lightly loaded. Figure 3 shows a simulation of the efficiency using optimal control and constant voltage/frequency control. The applied load is representing a pump/fan.

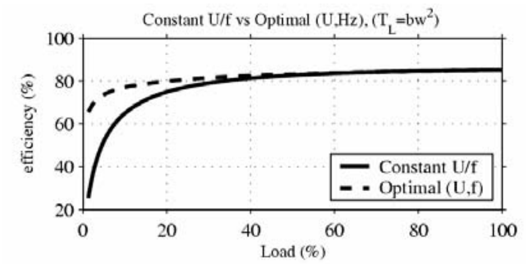


Figure 3 Optimal control and constant voltage/frequency control of an IM. The load is represents a pump/fan

Simulation of inverter switching losses

Calculation of the switching losses were made using Matlab Simulink ®. A state space model of the three phase IM used during the measurement where constructed, fed by a frequency converter using a switching frequency of 18kHz. At each switching instant, current magnitude and the conducting device where detected (i.e. if the current was flowing in a transistor or the freewheeling diode). The losses were then calculated using the voltage and current waveforms shown in Figure 1b using the datasheet of an IGBT type IRGPS40B120UDP.

TESTEQUIPMENT

The measurements were performed on one standard 4kW, 4 pole IM, and three different types of frequency converters A, B and C. The drive system was loaded using a DC machine and the test procedure was equal to all system setups. The various loadings were represented by sweeping the frequency between 10Hz and 60Hz with 5Hz increments. The drive system was operated at the same loads at each frequency, with the exception for the over rated loading points at the lower frequencies.

Table 1 Measurement equipment

Type	**Model**
Torque transducer	Tn 30
Power analyze	Norma 61D2
Power analyze	Yokogawa WT
Digital Oscilloscope	Lecroy 9304 CM
Ohmmeter	CM 1703
Stroboscope	1531 AB
Data aquision card	PC - MIO - 16E -1

MEASUREMENT RESULT AND ANALYSIS

Measurements were performed according to the description found in the previous section. Figure 4 a) and b) shows the different loss components for system C at 50Hz and varying loads. It can be noted that the mechanical and core losses are increasing with increasing load. This is due to the fact that the voltage in converter C is decreased at decreasing load, which differs from system A and B which has a constant core loss component due to the constant V/Hz control (the mechanical losses can be assumed constant in all three systems at a fix frequency). The resistive loss component are naturally increased with increasing load so is the converter loss component due to the increased power transfer.

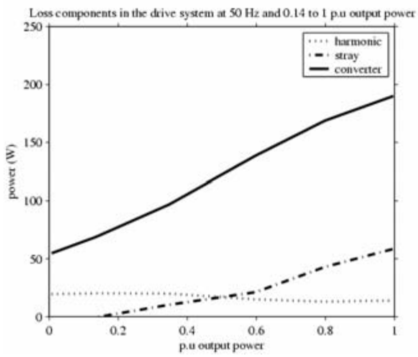
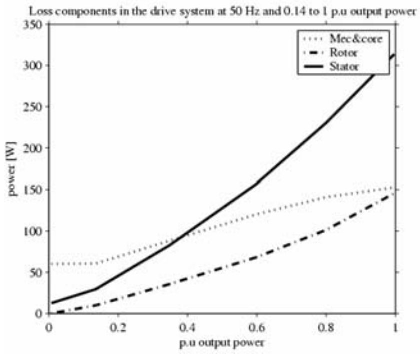


Figure 4a)Measured harmonic, stray and converter losses at 50Hz



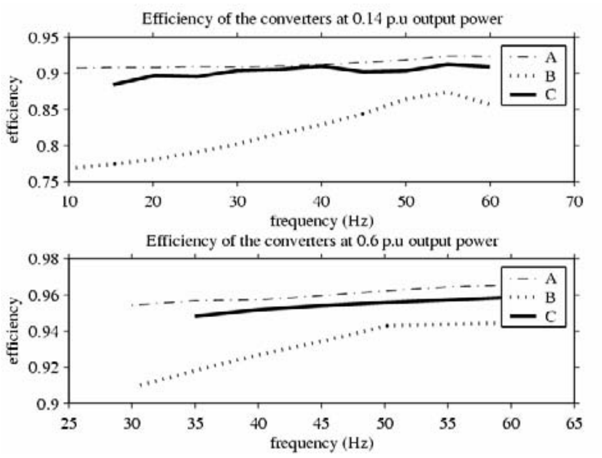
b) Measured mechanical+core, rotor and stator losses

Efficiency of the frequency converters

Figure 5a) shows the efficiency of the three converters at two load situations, 0.14 pu and 0.6 pu output power respectively. It can be seen that the difference in the efficiency decreases as the load increases and that converter B has the lowest efficiency while converter A has the highest.

Efficiency of the IM

Figure 5b) shows the efficiency of the IM in the three cases. It can be noted that system A and B has the lowest and the highest efficiency respectively. It can also be noted that the difference is higher at light load. This is due to the field weakening algorithm used by converter C as was explained in the previous section. The converter lowers the output voltage at lower load. As a result, the iron losses are reduced significantly compared to the other systems.



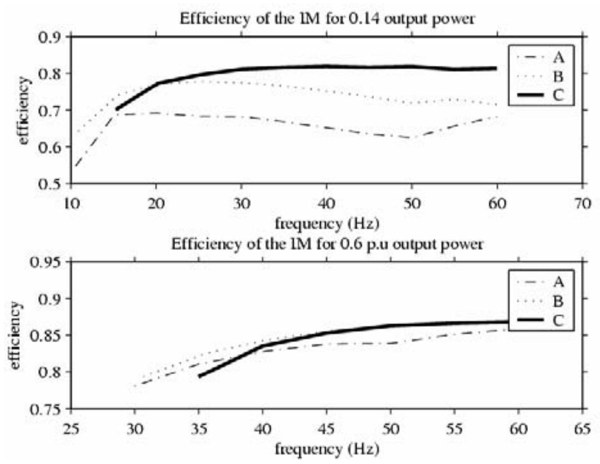
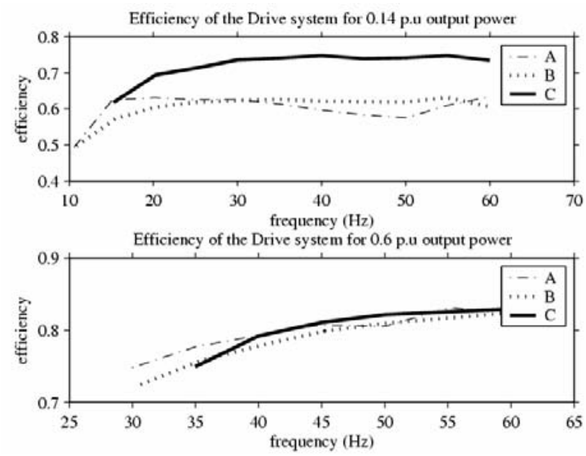


Figure 5a) Calculated efficiency of the converters using measurement data b) Calculated efficiency of the IM



c) Calculated efficiency of the drive system using measurement

Efficiency of the whole drive system

Figure 5c) shows the efficiency of the whole drive system. System C has the highest efficiency due to its field weakening algorithm but also due to its PWM switching scheme.

COMPARIOSON BETWEEN SYSTEM A, B AND C FOR HVAC APPLICATIONS

Figure 6 shows a typical annual load cycle for a variable air volume (VAV) system according to [19]. The operating time is assumed to 8760h (one year). It is further assumed that the system is 25% over dimensioned. The estimated energy consumption using the measurement data is presented in table 2.

Table 2

System	Yearly consumption (kWh)	Difference (kWh)
A	19270	152
B	19368	250
C	19118	0

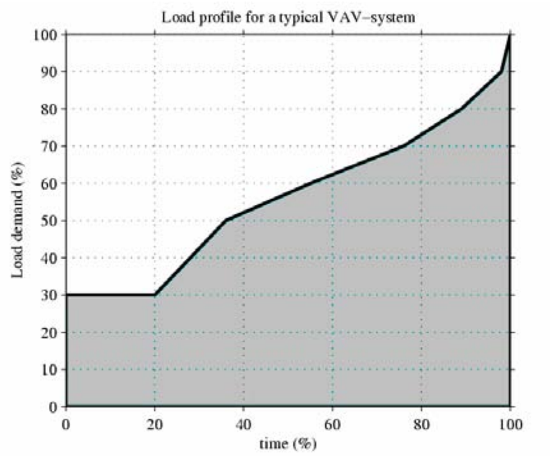


Figure 6 Annual load profile of a VAV-system

Simulation result

The above VAV-system was adopted as the load to the simulation model described in previous section. The switch stop at each half period was set to 0-80 degrees with 20 degrees increments. Figure 7 shows a decrease in the switching losses as the switch stop period increases, as expected. It can also be noted that the losses starts to decrease at certain load points which is due to the natural switch stop at over modulation mentioned earlier.

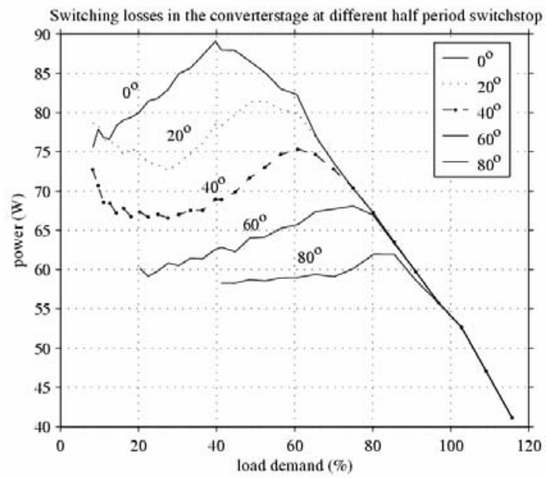


Figure 7 Switching losses in the converter at different half period switch stop. The IM is feeding the fan.

CONCLUTIONS

It was found that it is important to make a complete optimization of the energy efficiency. Using system B, with its special filter, gives higher machine efficiency but a low overall efficiency due to high converter losses. As a general conclusion, it has been shown that system C was the best solution from an efficiency point of view for a typical HVAC application. This is mainly due to the field weakening algorithm. Regarding the switching techniques improvements can be made. However, the relative difference was found small using this type of IGBT and switching frequency, especially at high load due to the over modulation of the converter.

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An Experimental Study on the Effect of Outdoor Temperature and Humidity Conditions on the Performance of a Heat Recovery Ventilator

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SUMMARY

The purpose of the present paper is to investigate the effect of outdoor weather conditions on the performance of a plate-type heat recovery ventilator. The performance should not be affected in a theoretical point of view. However, the performance varies in real applications, because of air leakage, motor heat generation, and etc. Experiments have been conducted to measure the sensible, latent, and enthalpy efficiencies by varying outdoor temperature and humidity conditions with the indoor conditions fixed at the standard heating or cooling conditions. The coefficient of energy has been introduced to quantify recovered energy in comparison with the electric power consumption.
Results indicate that the temperature exchange efficiency under winter conditions shows larger values than in summer conditions due to the heat generation by an internal fan. With the heat gain eliminated, the modified temperature efficiency remains almost constant regardless of outdoor temperature conditions. The enthalpy efficiency can exhibit very large values or negative values in case outdoor conditions are in the vicinity of the indoor enthalpy line. The direction of heat flow, in such a case, can be opposite to that of moisture flow between two air streams. Discussions are included about various interesting features of the psychrometric processes occurring in a heat recovery ventilator for various outdoor temperature and humidity conditions.

INTRODUCTION

Heat recovery ventilators are basically air-to-air heat exchangers to retrieve energy from building exhaust air. They are used to provide fresh ventilation air while saving heating and cooling energy. Heat recovery ventilators are expected to install commonly in apartment houses as well as in office buildings in Korea. The Act of Indoor Air Quality for Public Buildings [1] has been legislated and the Standards for Building Facilities [2] have been amended recently so as to include proper ventilation systems in public buildings and residential buildings.

The attestations for HRV's have been conducted by the Korea Energy Management Corporation [3] and the Korea Association of Air-conditioning Refrigerating and Sanitary Engineers [4]. However, the test conditions were different from each other and there were resolution problems in reporting test results depending on the temperature and humidity conditions and their uncertainties. A research has been conducted to provide theoretical backgrounds for the revision of the test protocol.

Theoretically speaking, heat recovery efficiency should be unchanged regardless of the indoor and outdoor temperature conditions. There are previous investigations that showed the dependence on the test conditions. Allan et al.[5] performed uncertainty analysis for an air-to-air heat exchanger installed in a commercial building and reported the efficiency can vary depending on outdoor weather conditions. Yoo et al.[6] compared the efficiencies of various paper plate-type heat exchangers for given outdoor conditions. Yee et al.[7] compared KS and JIS attestation standards and conducted sensitivity analysis. They raised a sensitivity issue related to the standard test conditions.

It is the objective of the present paper to investigate the effect of outdoor temperature and humidity conditions and their uncertainties on the performance results and their uncertainties so as to provide background information for revising the test protocol and to understand psychrometric processes occurring inside the heat recovery ventilator further.

METHODS

Experimental Apparatus

The heat recovery ventilator used in the present study is a ceiling-mount permeable fixed-plate heat exchanger. The HRV is one of typical kinds sold for apartment houses in the market. It has an average performance in terms of efficiency, capacity, and noise level. The measured values of the airflow rate and the electric consumption are 307CMH and 205W, respectively.

Experiments are conducted in Korea Test Laboratory. The test facility consists of two constant temperature chambers. The capacity of the calorimeter is 80MW, and the temperature range is -50^{o}C~100 ^{o}C. The HRV is installed between two chambers as shown in Figure 1. Temperature measuring points are located three diameters away from the duct inlet. The accuracy in measuring temperature is within ±0.1^{o}C. Flow rates are measured beforehand according to the test method by KS A 0612 [8]. Return airflow rate is adjusted to match the supply airflow rate within ±5%. The internal leakage rate is measured by a tracer gas method. A non-disperse infra-red detector is used to measure carbon dioxide concentration in each duct. The measurement range of the detector is 0~1000PPM, and the uncertainty is considered to be within ±2%.

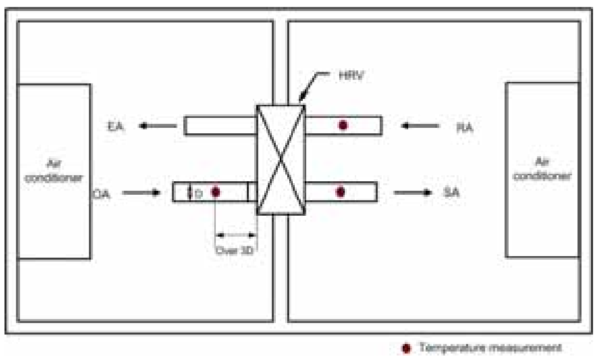

Figure 1. Schematic drawing of the test facility for heat recovery ventilators.

Experimental Conditions and Procedure

Table 1 shows the standard indoor and outdoor conditions by KARSE. The indoor condition is fixed for either heating or cooling condition, and various outdoor conditions are applied. The experimental conditions for indoor and outdoor are shown in Figure 2. The experimental number 'C' represents a cooling condition, and 'H' represents a heating condition. C0 and H0 are the standard test conditions by KARSE for cooling and heating conditions, respectively. C1 and C4 have the same humidity ratios with C0. C2 has the same temperature with C0, and C3 has the same enthalpy with C0. There is no difference between indoor and outdoor humidity ratios at C2, and therefore there is no moisture transfer. For heating conditions, H1 is the case when the relative humidity is the same with H0, and H2 and H3 are the cases when the enthalpy is the same with H0. Finally, H4 is a severe weather condition when condensation is expected to occur inside the heat exchanger module.

Table 1. Test conditions for HRV.

	Indoor condition		Outdoor condition	
	T_{dry}(°C)	T_{wet}(°C)	T_{dry}(°C)	T_{wet}(°C)
Cooling	24±0.3	17±0.2	35±0.3	24±0.2
Heating	22±0.3	13.9±0.2	2±0.3	0.44±0.2

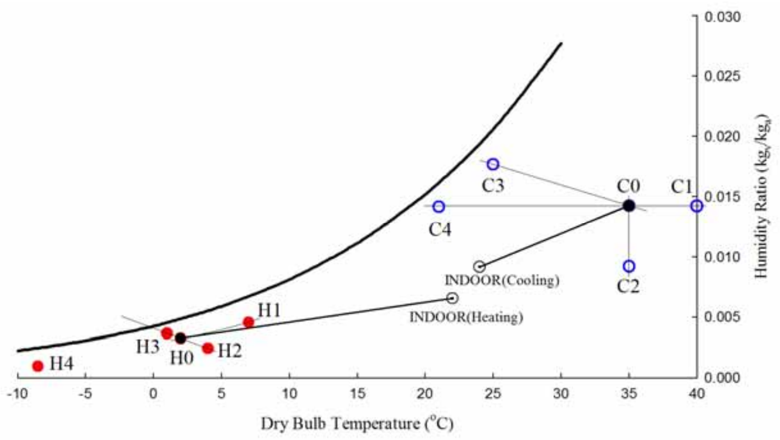

Figure 2. Test conditions shown on a psychrometric chart.

Each experiment is conducted after more than two hours when a steady state is reached for a given test condition. Dry-bulb and wet-bulb temperatures are measured for 30 minutes and averaged. The humidity ratio is obtained from the average dry-bulb and wet-bulb temperatures and the saturated vapor pressure at the temperature [9].

The temperature efficiency is defined as the ratio of the sensible heat recovered to the sensible energy difference between indoor and outdoor air.

$$\eta_T = \frac{T_{OA} - T_{SA}}{T_{OA} - T_{RA}} \times 100 \qquad (1)$$

The humidity efficiency is defined in the same way for the latent heat. The enthalpy efficiency is the efficiency including both sensible and latent heat recovered by the heat exchanger.

$$\eta_h = \frac{h_{OA} - h_{SA}}{h_{OA} - h_{RA}} \times 100 \qquad (2)$$

The effective enthalpy efficiency indicates the performance of energy transfer between supply and exhaust air excluding that by leakage. When the leakage ratio is zero, the effective efficiency is identical with the original enthalpy efficiency. As the leakage ratio increases, the supply air temperature approaches to the indoor air condition. When the leakage ratio is 100%, then the enthalpy efficiency becomes 100% as well. The effective enthalpy efficiency is always lower than the enthalpy efficiency. It is a function of leakage ratio as well as enthalpy efficiency, as is shown in Equation (3).

$$\eta_e = \frac{\eta_h - \eta_q}{100 - \eta_q} \times 100\,, \qquad (3)$$

where η_q is the leakage ratio. Leakage ratio is the volumetric leakage rate divided by the supply airflow rate. It is measured beforehand by the tracer gas method according to the following equation. Carbon dioxide is injected into the return air duct and concentrations are measured in each duct.

$$\eta_q = \frac{C_{SA} - C_{OA}}{C_{RA} - C_{OA}} \times 100\,, \qquad (4)$$

The coefficient of energy (COE) is defined as the recovered energy to the electric energy consumption. It indicates the cost benefits and usability of a heat recovery ventilator. The COE is not a unique property of a HRV but depends on test condition. As the indoor-outdoor enthalpy difference increases, the amount of recovered energy increases, while the denominator remains nearly constant. Therefore, the COE value changes from one test protocol to another.

$$COE = \frac{\rho \eta_e Q_e \left| h_{OA} - h_{RA} \right|}{W} \qquad (5)$$

It should be noted that the airflow rate in the equation is the effective airflow rate, which is the supply airflow rate subtracted by the leakage airflow rate.

Uncertainty Analysis

Uncertainty analysis shows the effect of individual measured quantity on the efficiency results. The measured quantities are indoor and outdoor dry-bulb and wet-bulb temperatures. The enthalpies are calculated from the measured quantities. The uncertainties in temperature efficiency, humidity efficiency, and enthalpy efficiency are analyzed according to the root-mean-square method as followings.

$$\frac{\Delta Y}{Y} = \sqrt{\sum_{i-1}^{n} \left(\frac{\partial Y}{\partial x_i} \frac{x_i}{Y} \frac{\Delta x_i}{x_i} \right)^2} \qquad (6)$$

RESULTS AND DISCUSSION

Table 2 shows the experimental results [10]. Various efficiencies and COE are shown with uncertainties associated. The humidity efficiency for C2 is not shown in the table, since there is no difference between indoor and outdoor humidity levels. Figure 3 shows the temperature and humidity efficiency results with uncertainty ranges. The horizontal axis shows the outdoor-indoor temperature difference, which is negative for heating conditions and positive for cooling conditions. As can be expected the error bar is large when the indoor-outdoor temperature difference is small. For most cases, temperature efficiency is in the range of 50~70%, whereas humidity efficiency is between 20~40%. It can be noticed the uncertainty in humidity efficiency is greater than that in temperature efficiency. This is due to the fact that the uncertainties in measuring humidity ratios are larger than those in measuring temperatures.

Table 2. Experimental results on various efficiencies and COE .

Experimental No	Temperature efficiency (%±%P)	Humidity efficiency (%±%P)	Enthalpy efficiency (%±%P)	Effective enthalpy efficiency (%±%P)	Coefficient of Energy (COE)
C0	58.75±1.09	20.49±3.95	38.72±1.86	32.66±2.05	3.47±0.23%
C1	61.35±0.78	19.17±3.98	42.53±1.57	36.85±1.73	4.70±0.24%
C2	57.68±1.13	-	61.68±3.56	57.89±3.91	2.81±0.23%
C3	-51.71±21.47	27.84±2.24	24.72±2.10	17.28±2.30	1.75±0.24%
C4	97.61±4.59	25.02±3.65	0.53±5.17	-9.31±-5.68	-0.38±0.23%
H0	69.46±0.63	29.27±3.49	57.43±0.98	53.22±1.08	7.02±0.19%
H1	71.69±0.84	30.34±6.65	61.86±1.50	58.09±1.65	5.33±0.19%
H2	69.93±0.70	26.19±2.86	54.03±0.97	49.48±1.06	6.47±0.18%
H3	69.27±0.60	31.32±4.08	59.48±1.00	55.47±1.10	7.27±0.19%
H4	67.32±0.41	34.47±1.94	56.97±0.58	52.72±0.64	10.94±0.21%

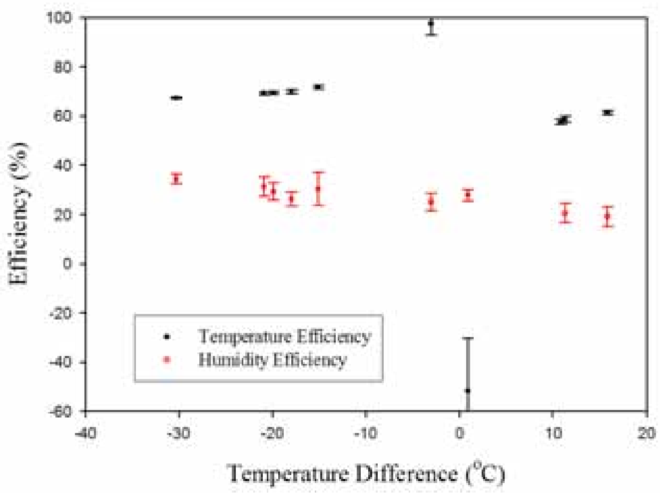

Figure 3. Temperature efficiency and humidity efficiency with respect to the indoor-outdoor temperature difference.

It is interesting to notice the temperature efficiency is negative for C3. This means the supply air has higher temperature than the incoming outdoor air for the heating condition. This is due to the temperature increase by a fan motor when the indoor-outdoor temperature difference is small.

As can be observed in Table 2, the effective enthalpy efficiency is lower than the enthalpy efficiency. Both of them are in the range of 30~60%. The COE is in the range of 2~10. As the power consumption of a fan motor is nearly constant, the COE can be considered to be proportional to the indoor-outdoor enthalpy difference.

Figure 4 shows processes on a psychrometric chart when outdoor air is passing through a HRV. The outdoor air at O reaches S' as it exchanges heat with return air. When temperature efficiency is greater than humidity efficiency, the slope of the line OS' is less steep than the line OR which represents the sensible heat ratio (SHR) of the space heating/cooling load.

The point moves from S' to S, as the heat generated by a motor is supplied to supply air. The temperature efficiency can be represented by the ratio of the length Rc to bc, consequently. In case the temperature rise is large enough so that the point b is beyond the point c, the temperature efficiency becomes negative, as for C3. Similarly, the humidity efficiency is the ratio of Od and Oc.

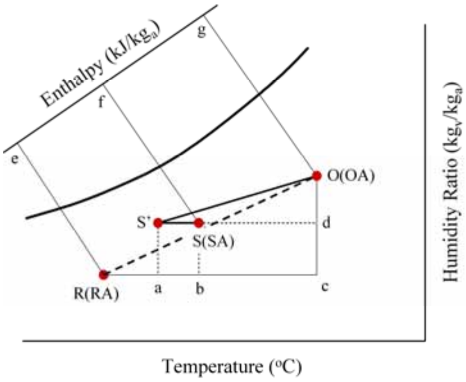

Figure 4. Process lines on a psychrometric chart inside a heat recovery ventilator.

The modified temperature efficiency is defined as the temperature efficiency with the internal heat gain removed, as shown in Equation (7). It is assumed that the heat gain is 50% of the power consumption of a fan motor. Yoo et al. [6] has estimated 40~50% in calculating internal heat gain.

$$\eta = \frac{T_{OA}\left(T_{SA} - \Delta T_m\right)}{T_{OA} - T_{RA}} = \frac{\overline{ac}}{\overline{Rc}} \qquad (7)$$

As can be observed in Figure 5, the modified temperature efficiency is nearly constant, i.e. 64~68%, regardless of the outdoor conditions. It is interesting to notice the effective enthalpy efficiency of the case of C4. The outdoor temperature is lower but the humidity ratio is larger than the indoor condition. The direction of heat transfer is opposite to that of moisture transfer through heat exchange medium. As the enthalpy of C4 is quite close to that of indoor condition, the enthalpy efficiency can have any value with large uncertainty. In case, the enthalpy efficiency is lower than leakage ratio, the effective efficiency becomes negative.

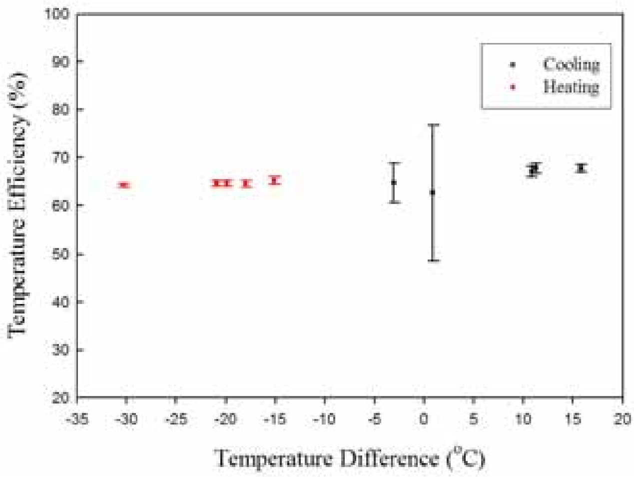

Figure 5. Temperature efficiency modified with the heat gain eliminated.

A psychrometric chart can be subdivided into several regions as shown in Figure 6. D is the region where most weather data fall on. A is the region where the effective enthalpy efficiency exceeds 100%, and B and C are the regions of negative efficiency. The enthalpy efficiency is positive in C but less than leakage ratio, so that the effective value is negative. The lines between these regions have specific slopes. The solid line is an enthalpy line.

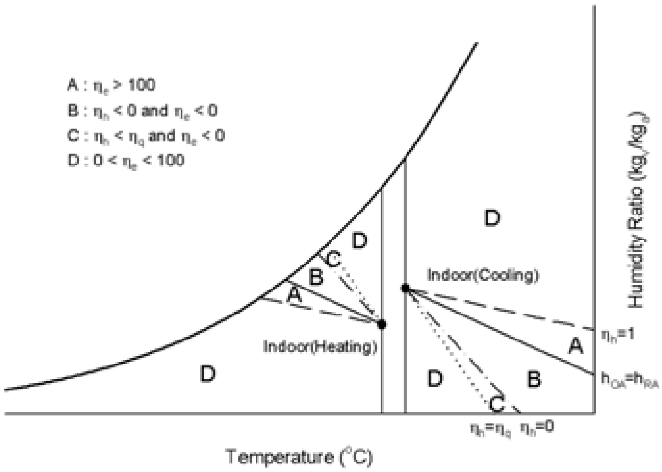

Figure 6. Psychrometric regions categorizing enthalpy efficiency.

CONCLUSIONS

Various outdoor temperature and humidity conditions are applied to a plate-type heat recovery ventilator to investigate the effect of weather conditions on various efficiencies and their uncertainties.

Firstly, the humidity efficiency is lower than the temperature efficiency for all cases regardless of heating or cooling weather conditions. The uncertainty of humidity efficiency is greater than that of temperature efficiency, since the uncertainty in measuring humidity is greater than that in measuring temperature.

One of the reasons that the efficiencies show different values depending on outdoor temperature and humidity is due to the heat gain by the internal heat generation of a fan motor. That is the reason why the temperature efficiency is greater for a winter condition than a summer condition.

When the indoor-outdoor temperature difference is not large, it is possible that the temperature efficiency can show negative values. Results show that the modified temperature efficiency with the internal heat gain eliminated remains nearly constant and within the uncertainty range.

The enthalpy efficiency can be negative or over 100% when the indoor-outdoor enthalpy difference is small. The effective enthalpy efficiency can be negative in case the enthalpy efficiency is less than the leakage ratio.

NOMENCLATURE

C : Tracer gas concentration [ppm]
C_p : Specific heat at constant P [kJ/kg·K]
h : Enthalpy [kJ/kg(DA)]
h_f : Saturated liquid enthalpy [kJ/kg]
h_{fg} : Latent heat [kJ/kg]
h_g : Saturated vapor enthalpy [kJ/kg]
P : Pressure [Pa]
Q : Airflow rate [m^3/h]
T : Temperature [^oC]
W : Electric consumption [W]
w : Absolute humidity [kg/kg(DA)]
x_i : Measured value
Y : Result

Greek

ρ : Supply air density [kg/m^3]
η : Efficiency

Subscripts

d : Dry air
e : Effective quantity
h : Total enthalpy
m : Motor
q : Air leakage
w : Moist air
OA : Outdoor air
RA : Return air
SA : Supply air

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13 June 2007 at 10:00 - 11:30

A09
Ventilation for special environments

Air Change Rate Control of Ventilated Ceiling concerning Heat Load in Commercial Electrical Kitchen

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SUMMARY

Ventilated ceiling system is an energy-saving replacement ventilation system for maintaining comfortable working environment in kitchens. The technology was introduced from Germany, but there have been no clearly determined relationships between their designs in an electric kitchen and the ventilation design standards and no established design methods. This study aims to establish design methods of ventilated ceiling systems appropriate for electric kitchens by assessing the thermal environment created and energy consumed by the systems while changing the amount of ventilation and the temperature of the air supplied. The ventilated ceiling system formed thermal stratification, which improved displacement ventilation, and maintained a good thermal environment in the working area. To meet the standards of the Ministry of Health, Labour and Welfare, the system should be operated at a ventilation rate of 25 times per hour with supply air at 16^{o}C or a ventilation rate of 20 to 40 times per hour with supply air at or below 20^{o}C.

INTRODUCTION

Ventilated ceiling system is an energy-saving replacement ventilation system for maintaining comfortable working environment in kitchens. The technology was introduced from Germany, but there have been no clearly determined relationships between their designs in an electric kitchen and the ventilation design standards and no established design methods. This study aims to establish design methods of ventilated ceiling systems appropriate for electric kitchens by assessing the thermal environment created and energy consumed by the systems while changing the amount of ventilation and the temperature of the air supplied.

METHODS

The study was conducted in the main kitchen on the 19th floor of K Building (Osaka City, Osaka Prefecture) and involved:
1) Monitoring the temperature and humidity of the kitchen and the amount of cooling water supplied to the air conditioner while ordinarily using the kitchen and changing the amount and temperature of air supply,
2) Analyzing the monitored data in terms of both the indoor environment and the amount of cooling water consumed, and

3) Constructing a model for analyzing air flows that almost reproduced the monitored results, and using the model to comparatively analyze and estimate the system performances at different heat emissions, which represented an assumed gas kitchen.
A plan of the kitchen and the positions of kitchen equipment are shown in Figure 1. The kitchens were zoned into the heating zone, dishing zone, and rice-cooking zone from the left to right in Figure 1. The ceiling was the ventilated ceiling system. A list of the kitchen equipment is shown in Table 1. The total electric power capacity of the equipment was 155.1 kW (448A). The amount of ventilation supplied is shown in Table 2 for each zone. The design ventilation was determined by the manufacturer by following the standards for air conditioner and ventilation equipment (VDI) in Germany. An energy-saving method is used to control the amount of ventilation based on the air current used by the kitchen equipment listed in Table 1. Controlling the amount of ventilation so as to be proportional to the electric current consumed in the kitchen may result in rises in the room temperature after the usage of the equipment since the equipment does not quickly cool down. To prevent the problem, ventilation control was tested which involved controlling the ventilation based on electric current consumption at the time point certain period prior to the control as shown in Figure 2. When ventilation was to be increased, the delay control was not used but the ventilation was increased spontaneously to ensure safety. The monitored changes in the automatic control output signals to the supply fans when the electric current consumption sharply dropped from 160A to 6A are shown in Figure 3. Approximately 60 seconds (10 minutes) after the drop in current consumption, the ventilation was reduced to the set value.
A large amount of vapor produced from brazing pans and other equipment was confirmed to have been attracted by the cold air supplied form nearby outlet slits, causing rises in the cold air temperature and dew on the surface of the slit metal plates. To minimize the mixing of the vapor and the cold air, a hanging wall was installed to separate the heat-producing grills from the slit metal down to a height of 150 mm.
The points where temperature and humidity were monitored are shown in Figure 4. The monitoring conditions are shown in Table 3.

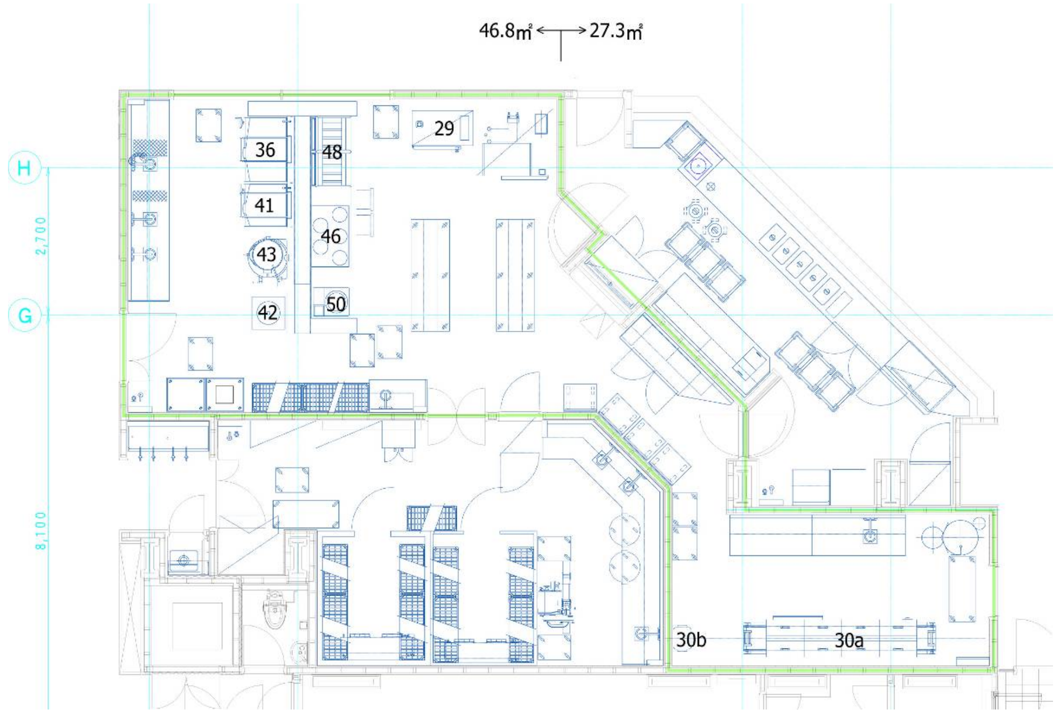

Figure 1. Plan of the kitchen

Table 1. List of principal kitchen equipment

	Article	Electric power capacity
29	Steam convection oven	36.8kW
30a	Multiplex IH rice cooker	31.5kW
30b	IH rice cooker	
36	Blazing pan	18.0kW
41	Blazing pan	12.0kW
42	Cylinder stove	10.0kW
43	Steam kettle	9.8kW
46	Electromagnetic cooker	10.2 + 11.2kW
48	Electromagnetic fryer	10.0kW
50	Electromagnetic Chinese oven	5.6kW
Total		155.1kW (448A) (A) = (kW) ÷ (√3×200)

Table 2. Amount of ventilation for each zone (Mean ceiling height: 2.55 m)

Zone	Area	Capacity	Ventilation volume	Ventilation rate
Heating zone plus dishing zone	46.8 m^2	119.4m^3	9,000 m^3/h	75.4 times/hour
Rice cooking zone	27.3 m^2	69.6m^3	3,600 m^3/h	51.7 times/hour
Subtotal	74.1m^2	189.0m^3	12,600 m^3/h	66.7 times/hour
Preparation zone	-		1,580 m^3/h	
Total			14,180 m^3/h	

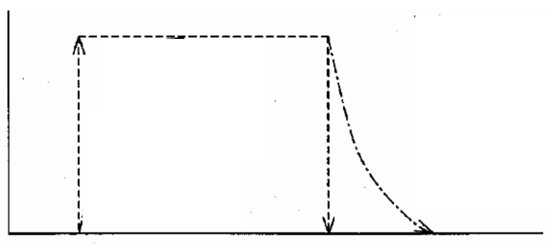

Figure 2. Schematic illustration of delay ventilation control

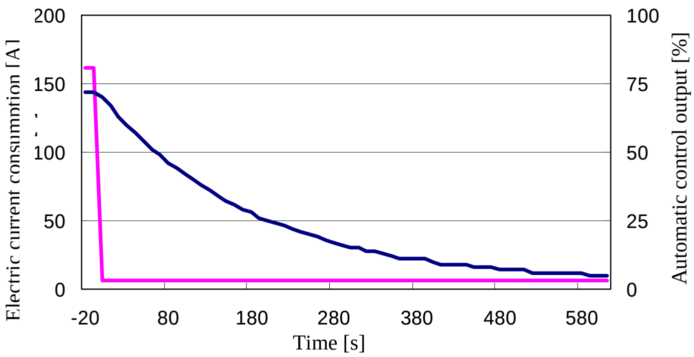

Figure 3. Monitored ventilation during delay ventilation control

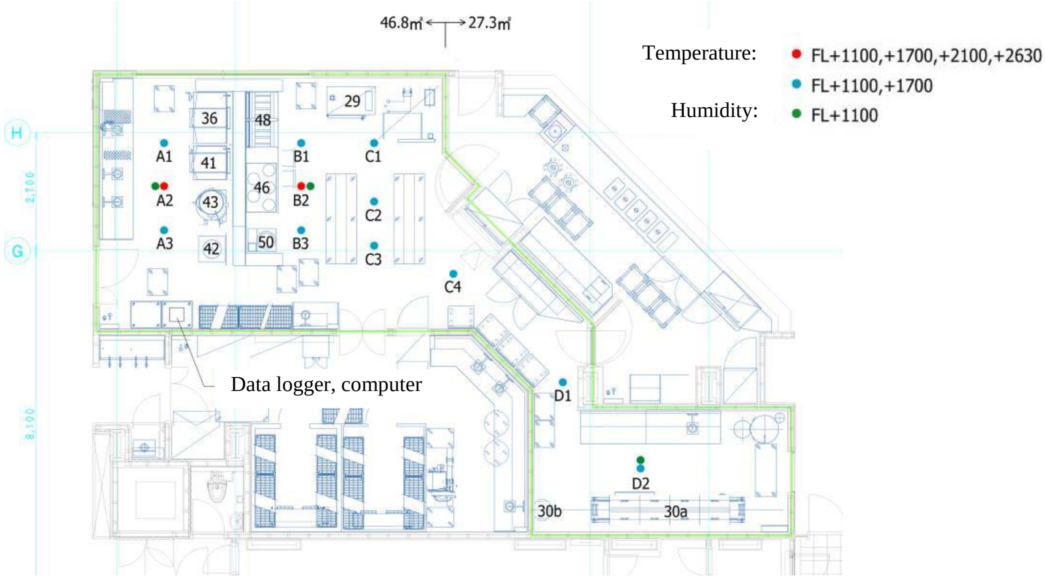

Figure 4. Points of monitoring (floor plan)

Table 3. Monitoring conditions

	Ventilation rate	Supply air temperature	Delay control	Hanging wall	Date of monitoring	Representative day
Case 1	18 to 45 times/h	22℃	w/o delay control	w/o hanging wall	7/10～7/13	7/11
Case 2	25 times/h	18℃	-	w/o hanging wall	7/18～7/21,24	7/19
Case 3	25 times/h	16℃	-	w/o hanging wall	7/25～7/26	7/26
Case 4	25 times/h	14℃	-	w/o hanging wall	7/27～7/28	7/28
Case 5	20 to 40 times/h	18℃	w/ delay control	w/o hanging wall	8/1,3	8/3
Case 6	20 to 40 times/h	16℃	w/ delay control	w/o hanging wall	8/2,4	8/2
Case 7	25 times/h	18℃	-	w/ hanging wall	8/8	8/8
Case 8	25 times/h	16℃	-	w/ hanging wall	8/9	8/9
Case 9	20 to 40 times/h	20℃	w/ delay control	w/ hanging wall	8/10	8/10
Case 10	20 to 40 times/h	18℃	w/ delay control	w/ hanging wall	8/11	8/11

RESULTS

Analysis of measurements

Figure 5 shows the electric current consumption in the kitchen and inverter output. When no delay control was performed, the data aligned on a proportional line. On the other hand, when delay control was used, the data scattered above the proportional line because the ventilation was immediately increased when the electric current consumption increased. Figure 6 shows the ventilation rate for variable air flow and effects of delay control. The mean ventilation rate was 32.1 times per hour when the air flow was allowed to vary within the range of 18 to 45 times per hour and was 33.6 times per hour when the air flow was allowed to vary within the range of 20 to 40 times per hour. The changes in air flow were smooth for the case in which air flow was 20 to 40 times per hour and delay control was used. The electric current consumption in the kitchen and temperature at A2 are shown in Figures 7 and 8. The temperature fluctuated less with variable air flow (Figure 8) than with fixed air flow (Figure 7). This was likely attributable to the effects of replacement ventilation and to changes in ventilation rate according to the heat produced from the kitchen equipment. The electric current consumption in the kitchen and humidity at A2 are shown in Figures 9 and 10. The humidity fluctuated less with variable air flow (Figure 10) than with fixed air flow (Figure 9), but was more stable at temperatures shown in Figure 8.

Comprehensive comparison

The mean heat quantity of the cooling water, electric current consumption in the kitchen, room temperature at A2, etc. are comparatively shown in Figure 11 for Cases 1 to 10 (excluding Case 7 in which some data were missing), for a hood surface speed of 0.3 m/s, and with air for gas combustion. The values for Cases 1 to 6 are the means for six hours between 7:00 to 13:00, when the loads were high, and those for cases 8 to 10 are the means for five hours between 8:00 to 13:00. The heat quantity of the cooling water was set to be the heat necessary for cooling the outdoor air of 30^{o}C, 70% humidity and 78.16 kJ/kg to the temperature of the supply air determined for each case to eliminate effects from the differences in outdoor air temperature. The power for moving the fan was calculated using:

$$kW = Q(T.P.) / 6{,}120\eta_T$$

where, *kW*: the power for moving the fan [kW], *Q*: air flow [m^3/min], *T.P.*: total pressure of the fan (= 120 mmAq, total of the air supplying and exhausting fans), and η_T: the fan total efficiency (= 0.5).

The results showed that: (1) The primary energy consumption was the smallest in Case 9 (ventilation rate of 20 to 40 times, supply air temperature of 20^{o}C). Since the electric current consumption in Case 9 was large, the small energy consumption was not attributable to small internal loads. (2) Even with a ventilation rate of about 25 times per hour, which was determined by multiplying the total electric capacity of the kitchen equipment and 30 m^3/kW, the ventilated ceiling system could satisfy the standards of the Ministry of Health, Labour and Welfare by setting the temperature of the supply air at 16^{o}C. (3) The temperature of the supply air, as well as ventilation rate, largely affected the energy consumption by cooling water. For example, the energy consumption by cooling water differed by 30% and 40% between Case 6, in which the ventilation rate was 20 to 40 times and the temperature of the supply air was 16^{o}C, and the case in which the ventilation rate was the same but the supply air was at 20^{o}C. (4) The power of the fans accounted for a small percentage of only 5 to 10% of the energy consumed by cooling water. (5) Ordinary hoods required much more energy for cooling water since the ventilation rate was larger than the ventilated ceiling system. (6) With Case 9, which consumed the least amount of energy, a gas kitchen, in which a theoretical amount of air was assumed to be burnt, required a double amount of energy for cooling water.

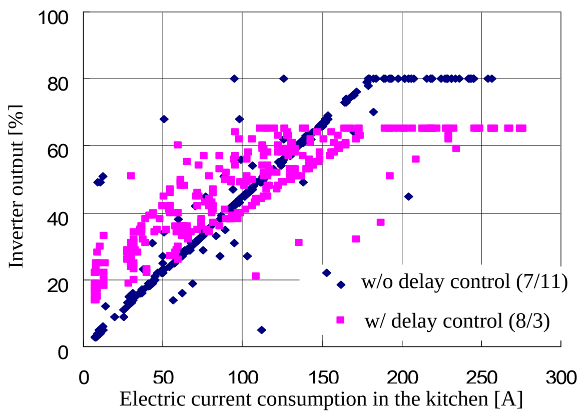

Figure 5. Relationship between electric current consumption in the kitchen and inverter output

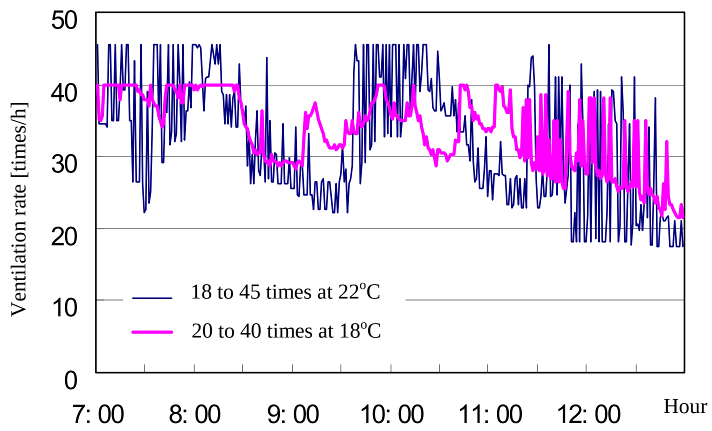

Figure 6. Time historical changes in ventilation rate for variable air flow operation

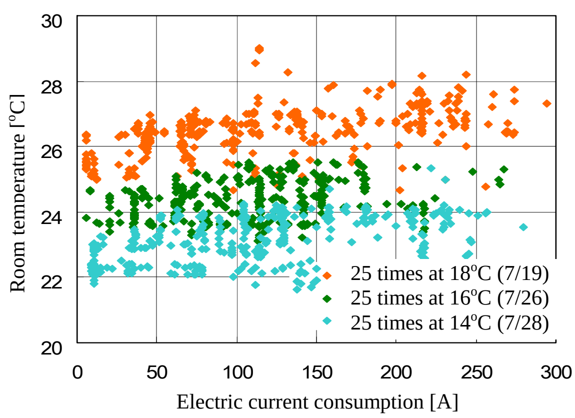

Figure 7. Relationship between electric current consumed in the kitchen and room temperature (for fixed air flow)

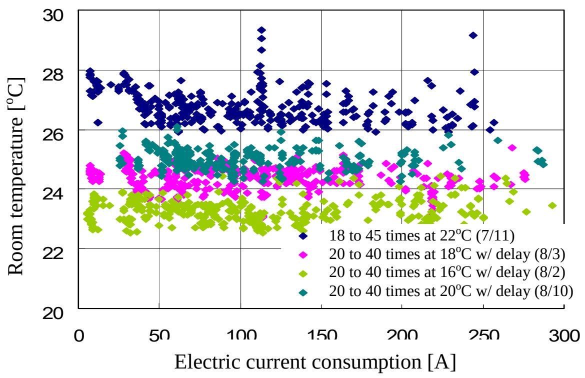

Figure 8. Relationship between electric current consumed in the kitchen and room temperature (for variable air flow)

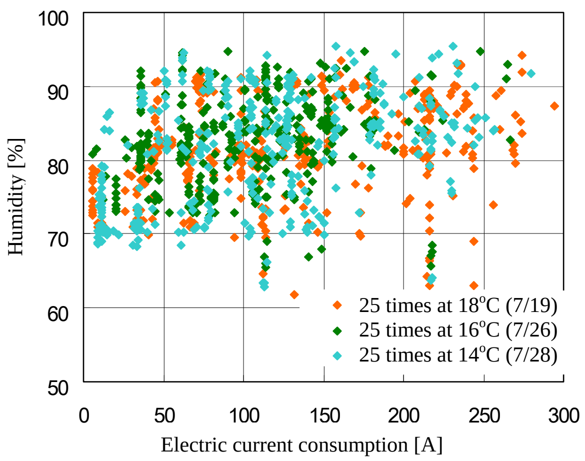

Figure 9. Relationship between electric current consumed in the kitchen and humidity (for fixed air flow)

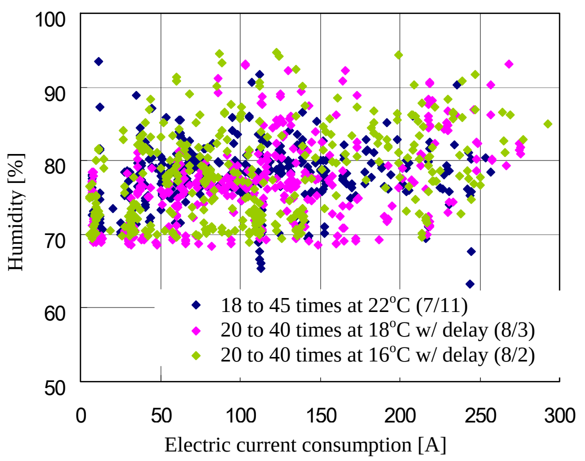

Figure 10. Relationship between electric current consumed in the kitchen and humidity (for variable air flow)

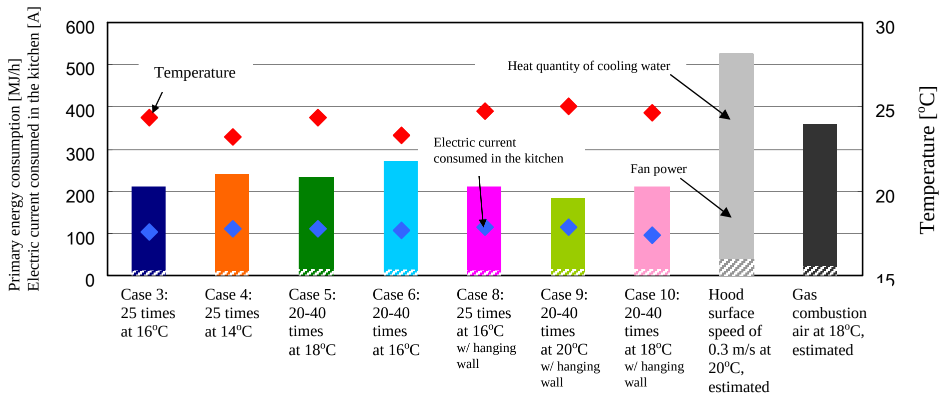

Figure 11. Comparison between Cases 1 to 19, ordinary hood, and gas kitchen

DISCUSSION

The ventilated ceiling system formed thermal stratification, which improved displacement ventilation, and maintained a good thermal environment in the working area. To meet the standards of the Ministry of Health, Labour and Welfare, the system should be operated at a ventilation rate of 25 times per hour with supply air at 16°C or a ventilation rate of 20 to 40 times per hour with supply air at or below 20°C. Of the cases tested, a ventilation rate of 20 to 40 times per hour with supply air at 20°C resulted in the smallest heat quantity in the cooling water. The mean ventilation rate in the variable air flow operation of 10 to 40 times per hour was 33.4 times per hour. Ordinary hoods and gas kitchens required large ventilation and thus resulted in large heat quantities in the cooling water. Variable air flow operation with delay control resulted in small room temperature fluctuations. Since the ventilation rate was kept to be at least 20 times per hour, the room temperature became too low after the termination of kitchen works. Thus the system should be desirably controlled by raising the minimum ventilation rate and/or raising the temperature of the supply air. Since the amount of cooling water used depended largely on the temperature of the supply air, the set temperature should be decided by considering the running costs and comfort of kitchen workers as well as the standards of the Ministry of Health, Labour and Welfare. The kitchen investigated in this study mainly performed secondary processing of precooked foods and cooked no stir-fried foods, which need large fire power. The system met the standards of the Ministry of Health, Labour and Welfare even at a constant ventilation rate of 25 times per hour in the kitchen, but there may be kitchens in which the standards cannot be met depending on the works to be performed. On the other hand, variable air flow operation is widely feasibly since the ventilation can be changed depending on the usage of kitchens via the software program for determining the electric current and ventilation rate.

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Ventilation and Industrial Hygienic Measures to Reduce Chemical Exposure in Car Repair Painting Shops

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SUMMARY

The main target of this research project was to find technical and industrial hygienic solutions for small car body repair shops to reduce worker's exposure to chemicals during the painting process. Worker's exposure to isocyanates and solvents was measured during painting with portable instruments in five car body shops of different ventilation and occupational hygiene levels. Ventilation performance was studied as well. Concentrations of isocyanates were high in the painter's breathing zone when solvent-based paints were sprayed. Also, the varnish applied on water-based paints contains isocyanates, and therefore their concentrations in the air were high during spraying. The highest solvent concentrations in the breathing zone were measured during the cleanup of the spray gun. Therefore, it is highly recommended to use a paint spray respirator and proper gloves when cleaning up the spray gun. In addition, use of a fume cupboard in paint mixing and spray gun cleanup is preferable . Primers were commonly applied to car bodies in a poorly ventilated preparation room without the use of a proper breathing respirator. Therefore, personal exposure to solvents was often higher in the preparation room than in the spray booth. It is recommended that all painting should be done in the spray booth.

INTRODUCTION

Workers are still exposed to hazardous isocyanates and solvents in car body repair shops despite the implementation of water based paints. The main target of this research project was to develop technical and industrial hygienic solutions for small car body repair shops to reduce worker's exposure to these chemicals during the painting process. Because the main target was in small companies, it was also important to find economical solutions. It was assumed that the painters' personal exposure to chemicals can be substantially reduced by quite simple adjustments to the working process. For example, previous studies by the research group have shown that it is quite common in Finland to spray primers to car bodies outside the painting booth. It was also assumed that ventilation performance could be improved by some minor development of the system. This presentation is based on the research project called "Preventive ventilation measures to reduce chemical exposure in car painting shops".

METHODS

A preliminary study was done in 22 car painting shops in southwestern part of Finland in order to find out the level of both the ventilation techniques and the occupational hygiene. The pre-study consisted of visual observations, interviews of the painters and observations of their work, examination of the ventilation method, smoke tests (pressure conditions, air flow directions) and observation of the used chemicals (paints, varnish and solvents). A lot of attention was paid on personal working manners, used chemicals and personal protective equipment.

After the preliminary study, five typical car painting shops were selected for more detailed investigations. They all had different types of ventilation systems. Worker's exposure to isocyanates and solvents was measured during the painting process with portable instruments. The same car hood was painted in all five shops. Ventilation performance was studied by measuring the air flow rate and the air velocity. Air flow pattern was visualized with smoke tests.

Table 1. Description of the car paint shops.

Car Painting Shop	1	2	3	4	5
Air change rate (1/h)	0.7	14	123	260	240
Characte-rization	natural ventilation	locally made booth	old factory made booth	good shop	high quality shop
Supply air distribution	leakage air flow rate	ceiling (2x2 m²)	whole ceiling	whole ceiling	whole ceiling
Exhaust location	near ceiling	6 pcs h=1,0 m	on the floor level	on the floor level	on the floor level
Water based paints				x	x
Paint mixing	grinding room, no ventilation	no mixing, use of factory mixed paints	paint mixing room, poor ventilation	paint mixing room, poor ventilation	paint mixing room, good ventilation
Spray gun wash	grinding room	grinding room	grinding room	paint mixing room, poor fume cupboard	paint mixing room, good fume cupboard

Total concentration of solvents was measured with VOC monitor (PID) at painter's breathing zone and the measurement signal was combined to video. This was done in order to find out the most exposive working tasks. Also the effect of different working methods was examined.

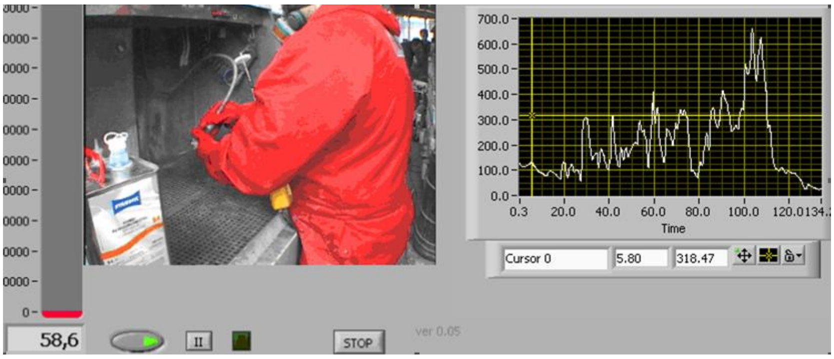


Figure 1. Combined measurement signal and video

Furthermore, computational fluid dynamics (CFD) was used in order to compare the effect of different air distribution methods and exhaust placements to concentration distribution and air flow pattern. Calculations were done in a steady state situation just after the car hood had been painted. Calculations were made with ANSYS CFX 10.0. In the basic situation (case 1), supply air was blown from the whole ceiling area and the exhausts were on the floor level. In case 2, supply air was blown to the room from a smaller area (2x2 m^2) of the roof and exhausts were on the wall at the height of 0.5 m. Booth air concentrations were compared to exhaust air concentration in both cases.

RESULTS

Main results of the pre-study

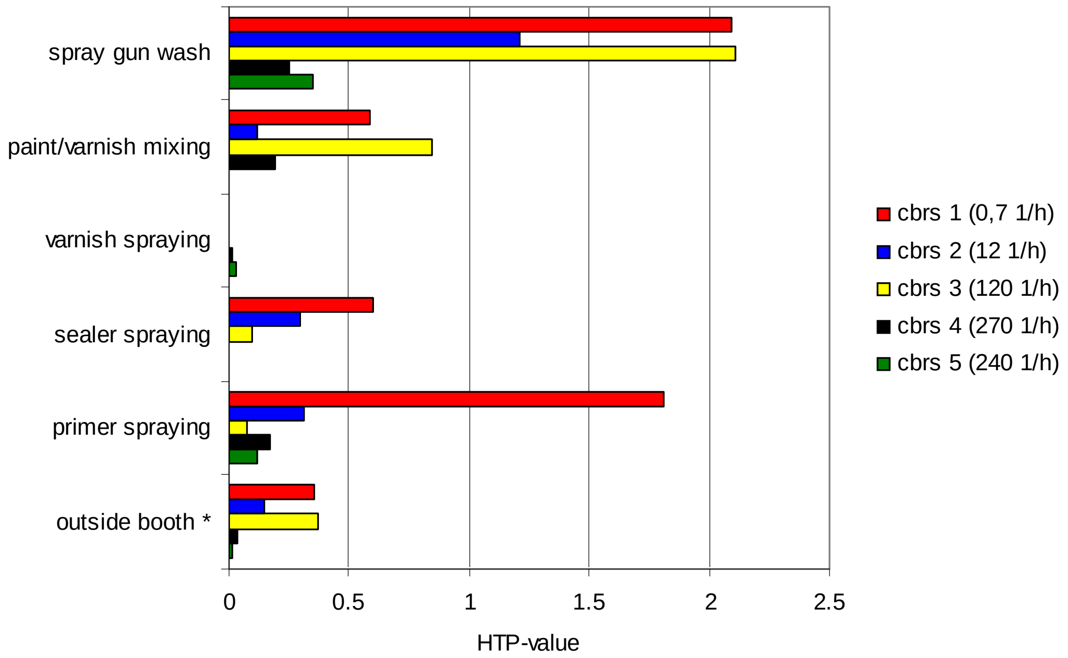
Water based paints were used only in the four biggest of the studied car painting shops. Other shops were still using traditional solvent based paints. Even in cases where water based paints were used, the varnish applied on paints still contained isocyanates. The hardeners of traditional paints also contain isocyanates (HDI). Primers were sprayed commonly outside the spray booth in premises where ventilation was often quite poor.

The use of respirators varied in different car body repair shops. Painters used half mask during painting in most of the cases, but only in some cases in spray gun washing and paint mixing.

Main results of the study

The results of the measured mean solvent concentrations with samplers in different work tasks are presented in Figure 2 and the measured isocyanate concentrations are presented in Table 2.

A typical example of the time variation of measured solvent concentrations is presented in Figure 3.



* Compared to Finnish 8 h OEL, others 15 min OEL

Figure 2. Measured solvent concentrations in different car body repair shops

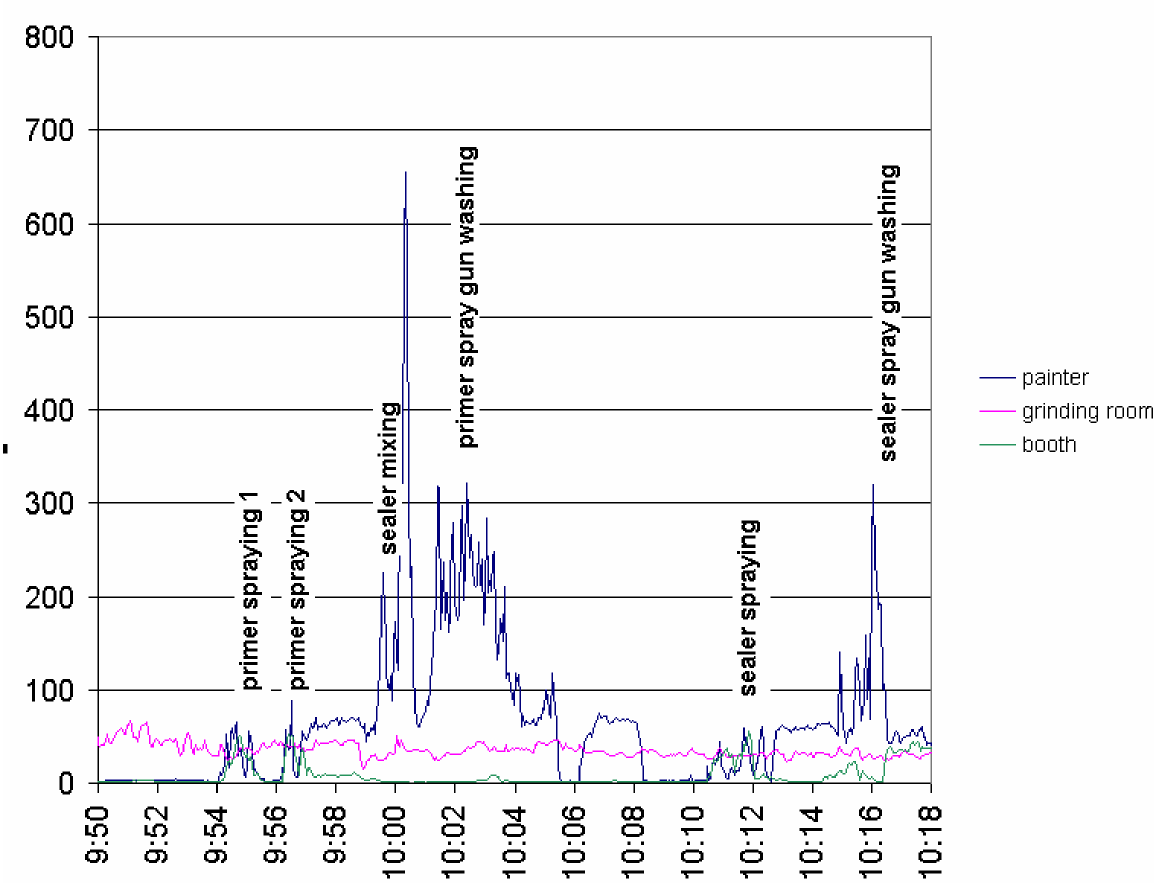


Figure 3. Measured time variation of relative solvent concentrations during different work tasks.

The highest solvent concentrations were measured during spray gun washing. High concentrations were measured also in mixing and liquid pouring. Special attention was paid to spray gun washing. It was examined with combined video and measurement signal. Personal working customs affected significantly painter's exposure to solvents. Worker's exposure to solvents was also remarkable in the grinding room when compared to the 8 hours exposure limit. This finding is important, because painters don't use personal protective equipment. Solvent concentration in the spray booth was below 20 % of TLV (Threshold Limit Value) when air exchange rate exceeded 100 1/h.

Concentrations of isocyanates were high in the painter's breathing zone when solvent-based paints were sprayed. Measured levels exceeded 5 – 170 times the TLV. The varnish applied on water-based paints contains isocyanates and therefore their concentrations were high during spraying (2 times the TLV). Concentrations of isocyanates were very low in sealer/varnish mixing and spray gun washing.

Table 2. Isocyanate concentration in painter's breathing zone
(Finnish $TLV_{15\ min}$ = 0.035 mg NCO / m³)

work phase	n	Isocyanate concentration level (mg NCO / m³)
primer spraying	3	0.003 – 0.42
sealer spraying in the booth	3	0.18 – 6.0
varnish spraying in the booth	2	0.044 – 0.070
sealer/varnish mixing	4	< 0.001
spray gun washing	5	< 0.001

The results of cfd-calculations are shown in Figure 4 and Figure 5.

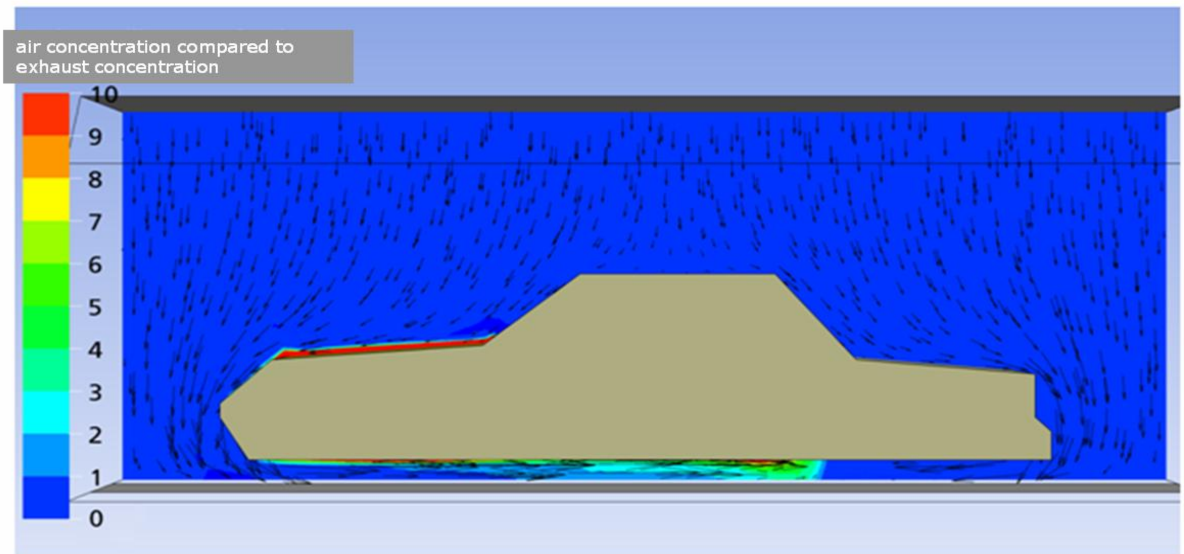


Figure 4. Calculated solvent concentrations in case 1, where supply air was blown from the whole ceiling area and the exhaust was placed on the floor level.

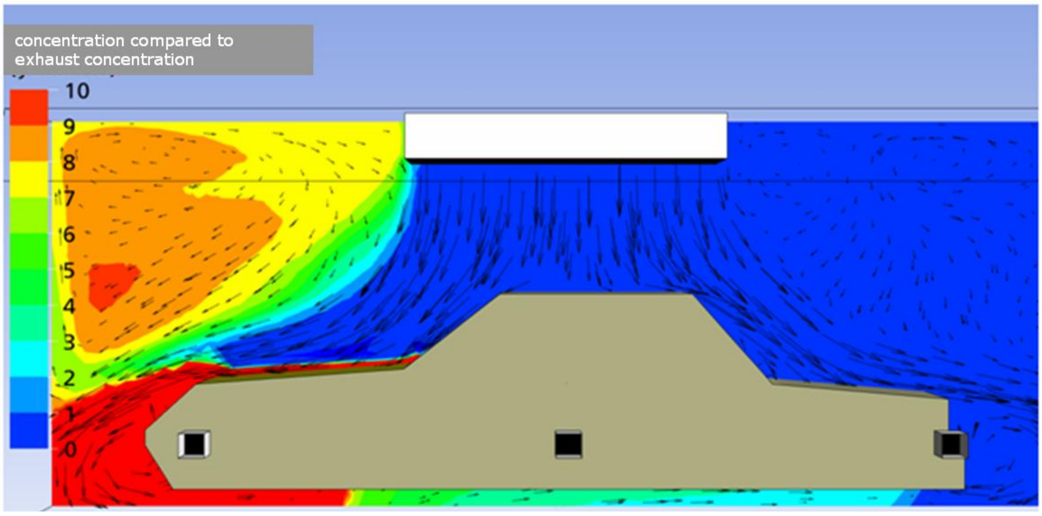


Figure 5. Calculated solvent concentrations in case 2, where supply air was blown from a small ceiling area and the exhaust was placed on the walls at the height of 0.5 m (black squares).

In case 1 the air flow in the booth is directed towards the exhausts, which are placed on the floor level. High solvent concentrations are located just above the painted hood and on the floor level. In case 2, where the supply air was blown from a smaller ceiling area, a large eddy distributes solvents widely to the booth. Smoke tests in real booths gave flow directions similar to the modeling.

DISCUSSION

Highest solvent concentrations were measured during the cleanup of the spray gun. Therefore, use of a proper paint spray respirator and gloves is essential when cleaning the spray gun. Paint mixing and spray gun washing should be done in a fume cupboard or in a separate under pressured room. The effectiveness of fume cupboard is not always adequate. Especially, cleaner solvent spraying from the spray gun often spreads outside the fume cupboard. Therefore, it should be possible to spray the solvent directly to the exhaust. Moreover, open solvent cans should be placed in a fume cupboard or in an under pressured space.

In the spray booth the solvent concentrations were below 20 % of the Finnish TLV when air exchange rate exceeded 100 1/h. Primers were commonly sprayed in a poorly ventilated preparation room without a breathing respirator. Therefore, personal exposure compared to the 8 h exposure limit to solvents was often higher in the preparation room than in the spray booth. All spraying should be done in the booth to minimize unnecessary exposure to solvents.

Concentrations of isocyanates were very high in the painter's breathing zone when solvent-based paints were sprayed. Measured values exceeded 5 – 170 times the Finnish TLV.
The varnish applied on water-based paints contains isocyanates and therefore their concentrations were high during spraying. Measured levels were twice the Finnish TLV for isocyanates.

The most effective way to ventilate the spray booth is the use of downdraft. Supply air should be blown from the entire ceiling and the exhaust should be placed on the floor level. The booth should be slightly under pressured to prevent contaminant leakage from the booth.

The most serious inadequacies of ventilation were found outside the spraying booth. More attention should be paid on the ventilation of paint mixing, spray gun mixing, paint storage and grinding premises.

It is essential to improve the workers' awareness of the health hazards of the chemicals and the importance of using personal protective equipment.

ACKNOWLEDGEMENT

The financial support of The Finnish Work Environment Fund is highly appreciated.

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Ventilation Concepts in Operating Rooms An Innovative Research Project

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SUMMARY

The current state of building services engineering in the healthcare sector is characterized by ambiguity and prejudice. There is a conflict, with inadequate knowledge of the necessity and effectiveness of ventilation-based protection concepts on the one hand – with the associated investment and operating costs if an integral view is adopted – and the economics of antibiotic prophylaxis and infection treatment costs on the other.

Keywords: HVAC systems in hospital, engineering in health facilities, operating room
Category: Innovative technologies and solutions / Innovative research project

INTRODUCTION

The "GiG" building services engineering in healthcare project is an interdisciplinary research and development project. The cost transparency and lump compensations per treatment unit required by healthcare policies are giving rise to new strategies and new cost/benefit considerations. Having to treat customers (i.e. patients) for as short a time as possible is becoming an increasingly attractive proposition. In order to achieve this, building services equipment as a whole must comply with requirements on hygiene.

The continuing spread of multi-resistant pathogens and the corresponding rise in the use of increasingly expensive antibiotics in hospitals and other healthcare facilities must be sustainably reduced.

This project will make contributions to building services engineering research and development with the aim of reducing both healthcare and building services costs in Switzerland by CHF 24 MN per annum. It is a project of international consequence in terms of private enterprise, research and health policy.

The major foreign research partners are represented by Prof. Dr. Ing. Rüdiger Külpmann, TFH Berlin and Dr. med. habil. Peter Lüderitz, AYSID Hygiene-Institut, Berlin. The main domestic partner is the Innovation Promotion Agency KTI/CTI of the Swiss Confederation, which is providing half of the project financing.

Surgical procedures make the highest demands on the sterility of the air in operating rooms, but not in the anterooms. Today we concentrate on creating the best possible conditions (asepsis) for the operating team in the vicinity of the operative field and instrument tables.

Heating, ventilation and air-conditioning systems are necessary for reasons of
- physiology / physical comfort (wellbeing)
- hospital-specific indoor air hygiene (infection prophylaxis),
- the technical process (functionality and safety requirements).

The primary purpose of air-conditioning is to ensure optimal conditions in terms of physiology and occupational health. The latter will require increasing attention in future. Bearing in mind all the routes to infection, the current expense in terms of ventilation technology to reduce the germ number of the indoor air outside the operating room appears excessively high, especially with regard to investment costs and ongoing maintenance costs.

New air-conditioning technology concepts can also be put to meaningful use in healthcare facilities. The basis is provided by requirements for hygienic air quality in hospitals as reformulated by hygienists, and the associated simplification for ventilation and air-conditioning technology from the point of view of hospital hygiene. High demands on the sterility and protective action of the air pertain only to the immediate vicinity of invasive procedures, to the care of immunosuppressed patients, to handling of hazardous airborne materials, and to isolation care. Therefore, interest in highly sterile air or directed airflows focuses on precisely defined areas in modern hospital hygiene. Requirements are function-specific, and they differ from room to room.
In the characterization of the necessary air treatment equipment, therefore, it seems useful to make a distinction according to characteristics of physical comfort and infection prophylaxis.

Assuming that, downstream of the upright filters, the supply air entering the conditioned space is sterile, the operating team, the nurses and the patient must be considered to be the primary emitters of microbes. Medical equipment and furnishings mainly represent thermal loads. Air treatment systems must also meet "normal" requirements in addition to hospital-specific requirements.

Evaluations, student dissertations and laboratory tests have revealed that existing concepts have clearly lacked an overall approach. Results and measurements are unsatisfactory in all respects.

Status of our own research

Development activities, and the resultant new Swiss SWKI 99-3 directive, represent the philosophy that air hygiene should concentrate only on areas where there is a risk to patients, instead of operating aseptically in all areas.

This philosophy has two important goals: to provide the best possible patient protection with the least possible energy expenditure and, at the same time, to make an intensive effort to get under control the costs of treating the infections and the problems of antibiotic resistance occurring in hospitals.

The basic idea of OR ventilation promoted by the new Swiss SWKI 99-3 directive stems from the experience that a very wide variety of procedures are performed in many hospitals – from urological procedures, which are of lesser significance in terms of air hygiene, to the highly demanding implantation of prosthetic joints. On the basis of numerous publications, the conviction has become prevalent that only a large LAF (Laminar Air Flow) ceiling panel with very good shielding effect is suitable for demanding procedures. On that basis, there are no resultant special requirements for the other areas of the OR and even in the environment of the LAF field itself in terms of ventilation technology. Consequently, activities with increased hygienic requirements must only be performed in the OR, specifically in the LAF zone only.

Until now, independent technical and/or hygienic investigations have been performed in different ways, with different task definitions and according to different procedures (mainly in specific, already completed and no longer modifiable clinical operating rooms, under time pressure and only if the clinic was kind enough to grant access to the room during off-peak hours). Therefore, the individual investigations are barely comparable with each other. Clinical issues always took precedence, and structural or technical changes were barely possible. The results of investigations performed so far in the development departments of LAF outlet manufacturers, for example, are not generally available for scientific evaluation and use, with the exception that the influence of structural factors or of OR lighting was taken into account in those investigations.

Within the hospitals section of the Swiss administration's Energie 2000 program, memoranda generated during the operational optimization phases of the hospitals clearly reflect the unsatisfactory state of hygiene-relevant spaces.

This means that, although some development has taken place, there has as yet been no systematic and comprehensive, application-oriented research using exact and reproducible scientific methods.

State of the art (national, international)

The national situation is reflected directly at the international level. German, Austrian and Swiss hygienists and engineers have been working together for some time to discuss requirements and technical possibilities, at least in the German-speaking region.

Additionally, the Swiss standardization body for the construction sector, SIA, has expressed willingness to intensively support the standardization efforts of CEN (based on Switzerland and Germany). The German Association of Engineers, VDI, took over the current version of SWKI 99-3 and published it in Germany in 2004 as the applicable directive for hospitals and building services engineering (draft VDI 2167).

Since 2003/04, therefore, there has been a single directive that is able to reflect the present and future state of the art. The blanket deployment of this instrument/knowledge is to be accomplished via the network/center of competence of HTA Lucerne and the Berlin University of Applied Science TFH as well as the project partners from the industry.

An ad hoc CEN meeting took place in March 2004 to discuss the question of research and standardization of building services engineering in hospitals. The European CEN partners that attended were unanimous: today's knowledge is somewhere between imprecise and incorrect, and must be urgently revised. Working Group WG 13 of Technical Committee TC 156 officially began its work in November 2004. The convenor of the WG is the author of this document.

RESEARCH GOALS

The research goal is to analyze the contamination paths of airborne germs (particles) and to eliminate them via new and innovative ventilation concepts. Since ventilation investigations for operating rooms cannot be done in scale model environments, and there are no simulation tools today that deliver sufficiently precise results, it is clear that today's OR ventilation technology must be investigated in the laboratory.

Investigations (among others in 2003/2004 term papers and dissertations of the HVAC/S department of HTA Lucerne) have shown that new systems with laminar flow ceilings do not always function as desired. OR lighting, for example, can cause undesirable mixing. Therefore, research and development work is needed.

Ventilation and measurement methods should be developed further, and not only theoretical but also experimental work is necessary.

There is no laboratory test facility for OR ventilation in Europe today. Professional associations, the industry, hospitals and German institutes are interested in our OR ventilation laboratory test facility.

The hospital hygiene project is of far-reaching international significance, since most hospitals (over 500 in Switzerland alone) are entering the renovation phase of their life cycles, and new, focused standards of hygiene need to be reviewed and developed further. Complementary work is required in the research sector. Airborne contamination paths and the ingress of germs attached to airborne particles must be investigated. Initial investigations have shown that insufficient attention has been paid to particle ingress from the sides and to contamination of the operating table from below, and that such ingress is being found to be considerable.

For this purpose, a test facility for OR ventilation is being set up to facilitate work in the field of OR qualification, hygiene, airflow investigations and metrology.

We have divided the overall project into five sub-projects:

Sub-project 1: heating and ventilation engineering concepts
Sub-project 2: OR lighting (flow optimization, new development)
Sub-project 3: measuring methods (quality controlling, quality assurance)
Sub-project 4: CFD (flow simulations as a supporting resource)
Sub-project 5: knowledge transfer and implementation

In addition to these research and development activities, an accreditation center for measurement of operating rooms or for companies providing such services is to be established at HTA Lucerne in cooperation with TFH Berlin.

NEW PERSPECTIVES

The current state and understanding of building services engineering in the healthcare sector is characterized by ambiguity and prejudice. There is a conflict, with inadequate knowledge of the necessity and effectiveness of ventilation-based protection concepts on the one hand – with the associated investment and operating costs if an integral view is adopted – and the economics of antibiotic prophylaxis and infection treatment costs on the other.
The research project is intended to contribute to the objectification and clarification of the situation.

What is new is above all the possibility to perform systematic investigations in a realistic operating room without the constraints on time and scope that real operating rooms impose. This alone makes it possible for the first time to perform comprehensive scientific investigations. Also new is the aspect that the investigations include the entirety of the OR equipment instead of individual aspects with findings that are not transferable to actual practice.

Therefore, the desire for an overall view of building services engineering and hygiene, and with it the desire for a functioning OR unit, also becomes the assignment. How can a concept be developed that meets both the needs of hygiene and the usual requirements on building services equipment? What will the future protection concept for hygiene and building services be like in order to achieve the defined characteristic values?

The goals have been divided into the five sub-projects described above for better understanding and simpler coordination. They will be accomplished individually and integrated at a higher, interdisciplinary level.

ALL PARTICIPANTS ARE WINNERS

Contribution to healthcare

Cost savings for health insurance providers and premium payers, as well as a reduction in patient suffering due to:

Infections: prevention of hospital acquired infections.
Hospitalization times: patients will spend less time in hospital.
Treatment: patients will not always require additional treatment with expensive antibiotics, and staff can concentrate on treating the basic condition.

Hospital and healthcare cost reduction:

Hygiene: Concentration on (in future, verifiably effective) hygienic protection concepts where there is a real risk. Cost reduction through reduced use of antibiotics, which is also appropriate in the light of the antibiotic resistance problem. (MRSA – multiresistant staphylococcus aureus, which occurs in hospitals in particular, is a serious problem, since these bacteria are resistant to most antibiotics)

Energy: avoiding needless asepsis in all areas means reduced air volume and less expense for hygiene in no-risk areas in the vicinity of invasive procedures (i.e. asepsis in the OR only, and not in the entire green zone).

Investment: The enclosed space (building investment) for central building services and ventilation shafts will be reduced. The outdoor air treatment equipment will be reduced to the dimensions necessary to meet the actual requirements for hygienic ventilation. Relatively little space is required between or above the conditioned spaces for the air circulation equipment and the necessary silencers (but it must be considered by the architects!). The ventilation equipment can be optimized selectively. Less investment expense will be necessary on average. Simplifications in the concepts will be possible.
Operating costs: Energy savings with local air recirculation instead of vast volumes of outdoor air that require constant heating or cooling. Reduced expense of preventive and corrective maintenance.

Medical product liability: Room air treatment systems make an important contribution in the field of medical product liability.
The sterility chain must be assured via organizational measures and improvements in ventilation. Sterile items must be protected against contamination, from unpacking to readying on instruments tables to use.
Lump compensations will necessitate a philosophical change in healthcare that will make investments in correct building services inevitable. If doctors are no longer paid according to hours worked but according to lump sums for each service provided (e.g. hip replacement) it will become economically relevant if a patent acquires an infection or can be discharged a few days earlier.

Goal = benefit for industry partners:

The goal of the industry partners is to optimize or redesign their products according to the redefined building services engineering requirements, i.e. taking the aspects of hygiene-relevance into account.

This will help them to achieve or maintain competitiveness, and create sustainable jobs in the healthcare supply sector while also reducing costs.

Goals from the hospitals' perspective:

Cost reductions are to be achieved in the OR area due to a highly sterile environment (reduced use of antibiotics) as well as energy savings on OR ventilation.

In the coming years, more than 600 operating rooms will be due for renovation in Switzerland alone; over 8'000 in the whole of Europe. The ongoing costs, which have until now been caused by poor hygiene, partially due to building services equipment, could be reduced in this context.

Direct cost reduction:

We have estimated the cost reduction in Switzerland and Germany based on reports from the statistical authorities in both countries as well as on statistics from the Energie 2000 program and dissertations from HTA Lucerne and TFH Berlin. With optimal use of building services equipment, i.e. through optimization and deployment of new products, costs could be reduced by an amount in excess of CHF 12 MN (a factor of 10 more than that in Germany, i.e. CHF 240 MN) per annum.

Benefit from economic recovery:

As a result of the new SWKI 99-3 directive for hospitals, which has been applicable in Switzerland since 11/2003 (VDI draft 2167), products are being withdrawn from the market and replaced with new developments (e.g. OR lamps, LAF outlets, media supply units, etc.).

The inevitable result is better and cheaper medical care for our population. This benefit is naturally also transferable to surgical day hospitals and outpatient surgeries.

Air Distribution Strategy Impact on Operating Room Infection Control

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SUMMARY

This paper discusses the impact of different air distribution strategies on infection control in operating rooms. The quality of air in an operating room is primarily assessed with regard to how effective the air distribution strategy is in minimizing the possibility of airborne particles causing infection to the patient. The ASHRAE Research Transaction paper titled, "Comparison of Operating Room Ventilation Systems in the Protection of Surgical Sites" [1], is reviewed as a starting point. A number of air distribution strategies are then reviewed and a qualitative assessment of each strategy will be presented. The design concepts to be reviewed were:

- Type of supply air diffusers
- Exhaust/ Return air locations
- Lighting and medical equipment location effects

Constant air exchange rates and supply air temperatures were assumed in the case studies, which include: 1) The new operating rooms at University of Massachusetts (UMass) Memorial in Worcester, MA, USA in which an air curtain/ laminar flow supply air diffusion concept was implemented: and 2) The new Hospital for Special Surgery in New York City in which laminar flow panels with manually variable airflow direction have been installed to study impacts on patient outcomes and infection control. Computational Fluid Dynamics (CFD) models of each of the case studies are presented with documented findings which support the qualitative assessment.

INTRODUCTION

Infection Control Overview

The assessment of indoor air quality depends on the type of space being assessed. In a healthcare environment, indoor air quality is primarily assessed by minimizing the effects of the rate at which the ventilation distribution system either protects the patient from infection, or conversely, contributes to infection risks. While infection control is clearly the primary focus in air distribution strategies, thermal comfort issues should also play a role in assessing the quality and effectiveness of the air distribution system, even in a healthcare environment. This is especially true in what many consider to be the most critical of spaces in a hospital- the operating room- where the thermal comfort of the patient and medical personnel in the operating room are a very different issue.

Recognizing that patient outcomes may be affected by the thermal comfort of the operating room occupants, infection control is still the main concern. One recent study [2] has shown that 2.54% of patients that have invasive procedures experience complications related to post operative infections. While it is commonly held, and for good reason, that the major concerns are the sterile procedures used by the healthcare practitioners, the air distribution and exchange rate strategy can have a significant effect on the patient outcome as it relates to the likelihood of complications due to infection.

Intuitively, indoor air quality would not appear to be a primary concern in a critical environment like an operating room. Instead, issues such as the quality of the lighting, room layout and how potentially life saving equipment is placed, sterile procedures, and overall cleanliness of the room would seem to be paramount. However, as evidenced by the detailed requirements outlined in the American Institute of Architects (AIA) Guidelines for Healthcare [3], the concepts for how air is distributed, filtered or exhausted, and how often the air is changed in the room are very important.

In an operating room, the AIA Guidelines specify a minimum exchange rate of 15 air changes per hour. It is important to understand that the guidelines only address minimum standards for the air distribution and thermal control systems. Most hospitals have standards that require higher air exchange rates than the minimum AIA standards. The most recent version is the 2006 AIA release of "Guidelines for Design and Construction of Hospital and Health Care Facilities".[3] There are other guidelines, such as Association for Heating Refrigeration and Air Conditioning Engineers (ASHRAE) and the Center for Disease Control (CDC) in the U.S. These guidelines include ASHRAE's new design manual "HVAC Design Manual for Hospitals and Clinics" [4] and CDC's "Guidelines for Environmental Infection Control in Health-Care Facilities". These new standards are being updated due to the Infection Control Risk Assessment (ICRA) [5] which is "conducted by a panel with expertise in infection control, risk management, facility design, construction, ventilation, safety, and epidemiology" as required by AIA Guidelines [3].

In addition to these guidelines, there are studies, both theoretical and evidenced based, that show that increasing the air exchange rate reduces the amount of contaminated air in a space and, therefore, reduces infection risks for the patient. In a study done by The Center for Heath Design, it was found that infections are transmitted through air, surface contact and water [3].

In addition to the evidenced based support for the importance of higher air exchange rates, there have been recent ASHRAE-supported studies which used some of the latest software techniques to attempt to model air distribution concepts in operating rooms to determine theoretical optimization of the air distribution concept, as they relate to infection control. Specifically, one by Memarzadeh and Manning [1] attempted to address this topic from a computer modeling framework utilizing computational fluid dynamics software.

The intent of this paper is to extend concepts presented in this theoretical study to two real-life systems designed and constructed within the past year; the new operating rooms at the University of Massachusetts Memorial Lakeside Addition in Worcester Massachusetts, and the new operating rooms at the Hospital for Special Surgery in New York City.

Summary of the paper "Comparison of Operating Room Ventilation Systems in the Protection of the Surgical Site [1]"

The main focus of this paper was to examine the different outcomes of a common operating room with two variables, first being the diffuser locations and second being the air change rate per hour (ACH). The reader is referred to the study for detailed criteria regarding assumptions for modeling the space and equipment in the room.

This study evaluated eleven systems, each of which consisted of a different diffuser set up, a different ACH, or a combination of the two. During the simulations performed for this study, particle tracking was done from three different sources. These particle origination sources were the surgical staff leaning over the patient, the circulating nurse and the tracked particles known as "main" which represented the particles passed from general surgical activity around the table.

The three different particle start locations were tracked with each of the eleven possible ventilation systems. The outcome of the research was analyzed in two different categories based on whether the particle was removed or if it hit the surgical site or back table. For the first situation of the particle being vented from the room after one hour it was found that the laminar flow diffuser with 150 ACH system was the best for all three particle cases. The most inefficient concept was the U-shaped diffusers.

The second category for evaluating the different system concepts was based on if the particle hit the surgical site or the back table proved that the laminar flow diffuser with mixed (high and low) exhaust / return elevations was the most efficient at keeping the particles from hitting the surgical site, however only had a mediocre performance at keeping particles from hitting the back table. The most inefficient at keeping the particles from the surgical site was the conventional diffusers.

From this research it can be concluded that the best systems were the laminar flow systems. While the outcome varied based on location of the diffusers, it was not significant enough to firmly conclude that the high and low exhausts systems should not be used. However, the updated AIA Guidelines 2006 recommend that the high and low return/exhaust strategy is preferred [3].

RESEARCH

The Memarzadeh and Manning study advanced the understanding of air flow concepts by identifying the thermal plume from the patient, doctor and nurses as a critical factor to be addressed. Assuming these findings to be accurate, it is important to note a couple of other important factors in assessing laminar flow concepts in an operating room that are often assumed to be negligible. These are the effects of the lighting and the obstructions (booms and movement of surgeons) on laminar flow systems. This study attempted to model the effect of these variables in the context of the two case studies. The assessment is done utilizing similar CFD analyses. The IES Macroflow [6] software was used. The particulate count was not addressed, rather, air flow patterns from clean to "less clean" spaces were modeled. Key model parameters and assumptions were:

- 8m x 7.7m x 4m (24 ft x 23 ft x 12 ft)
- One surgical staff members
- One patient
- One surgical stand
- Three surgical lights
- Two monitors (and stands)
- One operating table

The following sections summarize the results of the CFD studies as applied to the two field site case studies.

UMass Memorial Case Study

The UMass Memorial air distribution concept utilizes the standard laminar downdraft flow concept delivering air at a low velocity directly above the patient area, as shown in Figure 1.

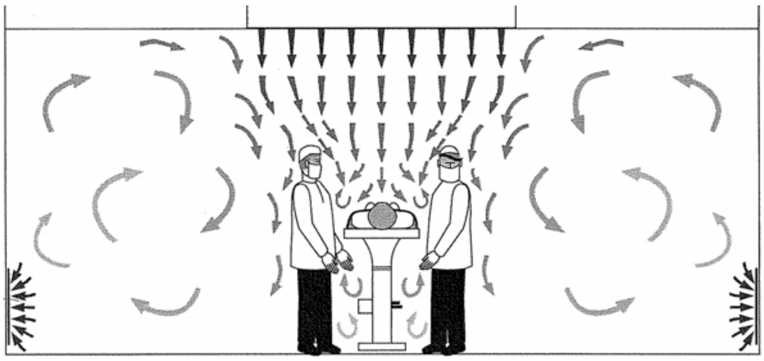

Figure 1. Standard laminar flow system concept without curtain. [2]

In addition, a linear supply air flow can be provided to apply an air curtain at the boundary of the operating area, protecting the clean area of the space from the less clean area of the space. (See Figure 2.)

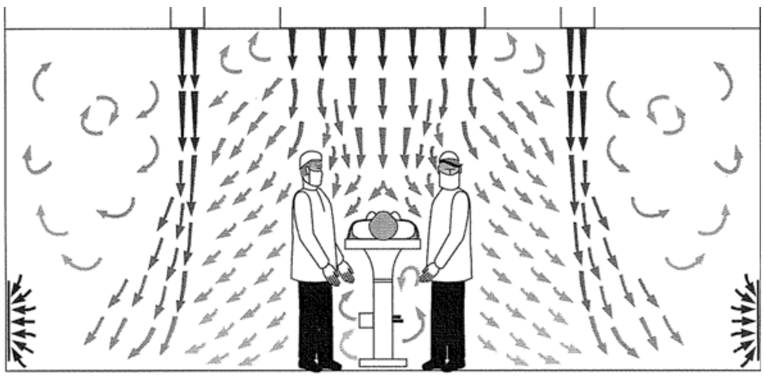

Figure 2. Laminar flow system with air curtain concept. [2]

Return grilles are located low in opposite corners of the space, and the room is under a positive pressure, consistent with the clean to less clean concept. The central air handling system has high efficiency particulate absorbing (HEPA) filters. The air is returned from the space. Of the flow needed to achieve 25 ACH; approximately 33% is brought in from the outside. This exceeds the minimum requirements of 15 ACH and 3 ACH of outside air required by the AIA Guidelines.

There are three medical booms in the laminar flow surgical area. The thermal plume from the patient is assumed to be 0.04 kW (136 Btu/hr), and the heat from the lights (which are located at the ceiling between the laminar flow diffusers and the linear, air curtain diffusers) is assumed to be radiating at 0.32 kW/m^2 (100 Btu/hr-ft^2). The heat from the lighting on each boom is estimated at 0.41 kW/m^2 (130 Btu/hr-ft^2).

The CFD model results (Figure 3) indicates that the particle path does flow from the clean to less clean area. The positive effect of the linear diffuser concept is clearly indicated. The air curtain has effectively isolated the clean zone, and has also provided an increase in air change rate in the patient zone by inducing air flow away from the clean zone. This results in an increase in the effective air exchange rate within the clean zone from 25 ACH to well above 40 to 50 ACH [2]. It is important to have as little space as possible between the linear diffusers as to reduce the induction of less clean air into the airstream and to avoid stagnant or low velocity areas which will trap contaminants [2]. As noted in the reference to the ASHRAE research studies on this topic [1], this results in a reduced likelihood of patient infection.

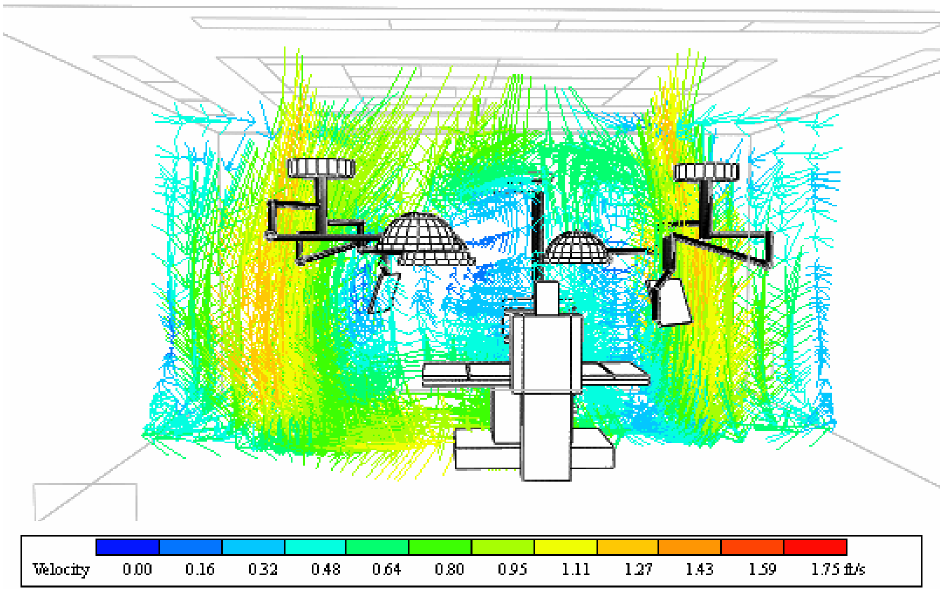

Figure 3. Velocity profile from the UMass study showing the laminar flow air curtain surrounding surgical area.

The next two figures (Figure 4 and 5) represent the particle path from the location directly above the patient to the return/exhaust location. The path shows that the particle disperses from a point directly above the patient and is trapped by the air current, flowing in a relatively linear path to the less clean zone, and then out of the room. The CFD models provide theoretical results that support the air distribution manufacturer’s technical literature [2] with respect to the air curtain concept. The results further validate the ASHRAE study [1] that advocates for more than 20 ACH.

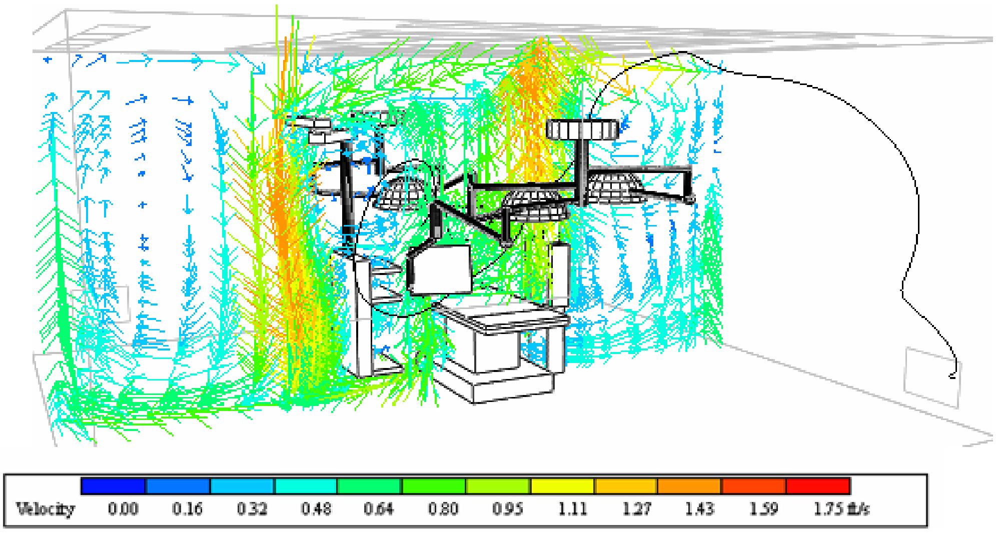

Figure 4. Velocity profile from UMass study showing particle being removed from surgical area.

Hospital for Special Surgery Case Study

The New York City's Hospital for Special Surgery (HSS) air distribution concept is based on the laminar flow concept, but does not utilize the air curtain concept previously described. The variations include diffusers that can be adjusted to direct air flow, and an enclosed plexi-glass curtain that isolates the clean zone. The ceiling layout is altered regarding the diffuser, light and grille locations. The motorized directional vanes on the laminar flow panels are manually adjusted by the surgical team to allow for the air flow to change from the vertical direction to a 45-degree deflection. This feature is utilized when the surgical team is required to work directly above the patient and vertical laminar flow would cause for the air particles to be carried directly into the patient area. The 45 degree adjustment results in the particles to be angled away from the surgical area. The arrangement of the room, as far as equipment and people, was modeled similarly to the UMass Memorial model. However, changes were made to the supply and return diffusers as well as the lighting locations to reflect the actual room layout. The layout of the HSS (Figure 6) consists of eight 0.6m x 1.3m supply diffusers and two 0.6m x 1m supply diffusers located directly above the operation table. Surrounding the diffusers are ten 0.6m x 1.3m fluorescent lights. In addition there are three 0.3m x 0.3m fluorescent lights located in the room. The return grilles are located at opposite corners of the space.

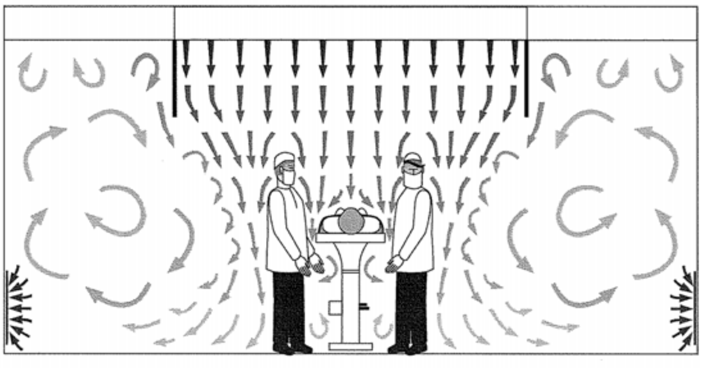

Figure 5. Laminar flow system with physical curtain [4].

The physical curtain provides a barrier between the clean zone and the less clean zone. The curtain seen in Figure 5 is a partial curtain which is used to help direct the laminar flow air towards the surgical area. As an improvement on this method, the design of HSS includes a physical curtain which surrounds the surgical zone on all four sides. Instead of hanging from the ceiling to only a few feet below the ceiling, the design includes the curtain hanging from the ceiling to 8" above finished floor.

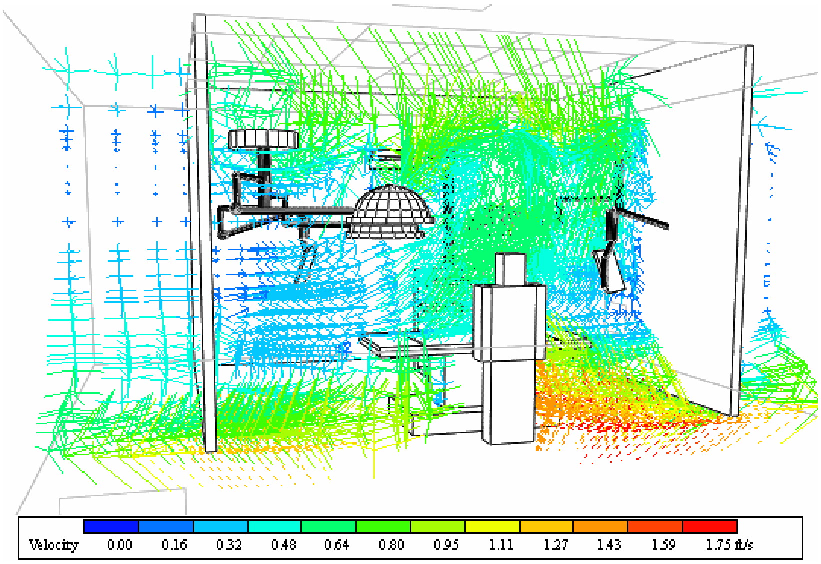

Figure 6. Velocity profile for HSS study showing general air movement coming from ceiling diffusers and under curtain and out of the surgical area.

The CFD models clearly indicate that the directional flow from the laminar diffusers creates a customized air flow dynamic in the room. There is a significant increase in velocity at the floor where the plexi-glass curtain acts in a similar fashion to a slotted return/exhaust opening. It is also evident that the equipment in the room causes turbulence that inhibits the direct removal of particles. The directional diffuser concept could be optimized by the use of similar studies.

SUMMARY

The case studies and analyses developed based on actual operating room installations show that the theoretical air distribution will optimize the concept of maximizing air exchange rates and directing particles in the air from the critical clean area above the patient to the less clean areas at the outer parts of the room. The systems also isolate any particles already located in the less clean areas in such a way that there is a very low probability that they will be re-entrained into the critical clean zone.

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Case Study: A Controlled Ventilation Solution for Laboratories at The Department of Natural Science of Tallinn University of Technology

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SUMMARY

This case study deals with the renovation project of ventilation system in the Natural Science Building of Tallinn University of Technology, covering the established aims and principles of technical solutions, in general, and results of monitoring. The reconstruction project involved the task of creating a new ventilation system for required indoor climate of laboratories, avoiding dissemination of harmful materials to rooms. In addition, it was required to keep the operations costs approximately at an acceptable level.

There are two separate central supply-exhaust air ventilation systems for general air exchange and for local ventilation systems. To assess the operating systems and to achieve the established aims, special testing of joint work between fume cupboards and main ventilation was conducted, using airflow smoke to emphasize the effect.

Results of monitoring demonstrate that investments for these ventilation systems were justified and the installed ventilation systems corresponded to our technical and sanitary-hygienic expectations.

INTRODUCTION

The main target of the ventilation systems for a laboratory is to guarantee a healthy working environment, avoiding dissemination of harmful materials from fume cupboards to rooms and between the rooms.

At the same time, an optimum energy consumption of the ventilation systems is required.

This case study deals with the renovation project of ventilation in the Natural Science Building of Tallinn University of Technology, covering the established aims and principles of technical solutions, in general, and results of monitoring. In this building there are mainly educational and scientific laboratories. Additionally there are some office and maintenance rooms. The present study only deals with subjects pertaining to the laboratories.

The reconstructed building has 4 floors with 7900 m^2 of total area. The laboratories hold 97 fume cupboards, 18 local exhaust - ventilation systems for specific purposes and 45 cupboards for chemical reagents, all situated dispersed in 50 different rooms.

TECHICAL SOLUTION

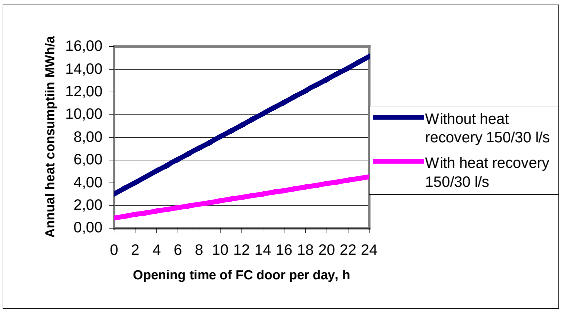
Target Requirements

The following target requirements were adhered to while working out the technical solution:

- The constant air speed 0,5 m/s in fume cupboard holes will be restored after a couple of seconds;
- The air pressure balance in the rooms;
- The maximum inside temperature 25 °C;The maximum CO_2 900 ppm;
- The optimum energy consumption

Energy Consumption and Fume Cupboard's Door Opening Time

Several earlier studies have shown that it´s necessary to guarantee constant air speed 0,5 m/s in fume cupboard holes for the fume cupboards to work effectively. The dimensioning of the fume cupboards in the present study is based on assumption that the design air flow of one equipment is 150 l/s while the door is opened and 30 l/s in stand-by. The longer fume cupboard door is opened, the larger is the consumption of heat and electric energy. The figures below illustrate estimated consumption of heat and electric energy per fume cupboard in a year, depending on the length of time the door is open during astronomical day in Estonian climate conditions.



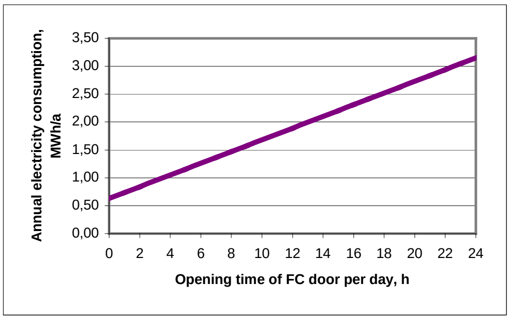


Figure 1 Annual heat (left) and electricity (right) energy consumption per fume cupboards (FC) dependig of FC's door opening time.

The usage of heat recovery allows for significant savings in heat energy. The supply and exhaust air may not mix in the heat recovery system.

Design Solution

Two separate supply-exhaust air ventilation systems are designed for the laboratories: for general air exchange and for local ventilation systems. Both systems have recuperative indirect heat recovery units with a circulating fluid medium. To extract dangerous and abrasive materials, additional separate local exhaust ventilation systems are provided.

Air handling units (AHU) are situated in the maintenance room of the last floor. The AHU-s are equipped with heating and cooling coil and heat recovery. The main ducts of the ventilation are situated in the technical room of the last floor where the vertical risers are going down from. The risers are branching off horisontally on every floor, in the start of branches there are constant pressure dumpers. Generally one horisontal branch services 1 to 4 rooms (laboratories). Figure 2 illustrates the global ventilation solution.

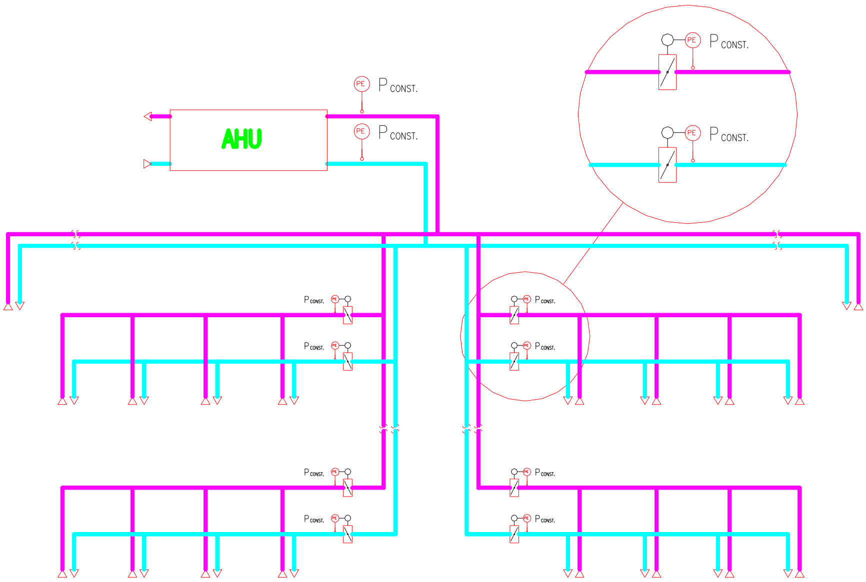


Figure 2. General scheme of ventilation ducts

There is a local automation system in every laboratory room. Automation of ventilation systems is characterized by a short-time response – 1…3 seconds. Indoor conditions of the rooms during some changes (e.g. after regulation of the size of fume hoods opening), the required air pressure balance in the rooms and constant air speed in fume cupboard holes will be restored after some seconds. The system of automation has several additional functions, such as to ensure the minimum main air exchange rate in the rooms (so-called standby conditions), signalisation about system disturbances and non-economical operation in part of the rooms to guarantee the required level of CO_2 concentration or indoor air temperature. Figure 3 illustrates the ventilation control system of a typical laboratory room.

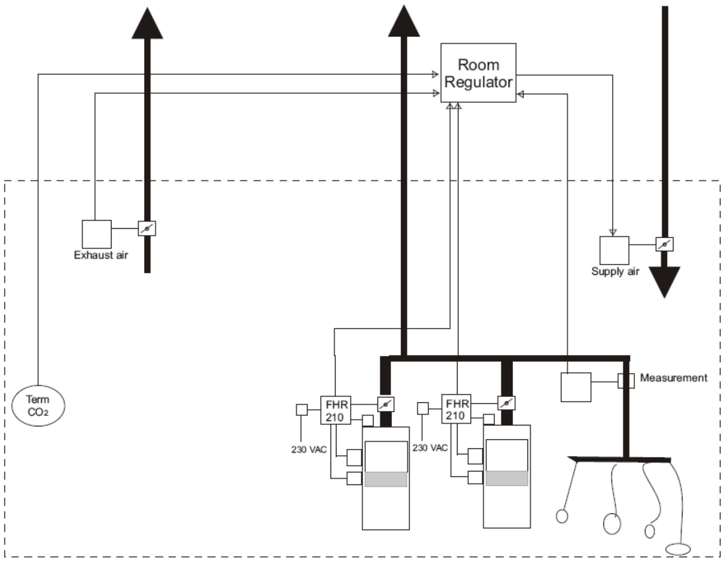


Figure 3. The ventilation control principle of a typical laboratory room

RESULTS

To assess the operating systems and to achieve the established aims, there was conducted a special test of collaboration between fume cupboards and main ventilation, using air flow smoke to emphasize the effect. During this monitoring, CO_2 concentration and air temperature in the rooms and the air speed values in the fume cupboard openings were scrutinised.

In the educational laboratories the concentration of CO_2 stayed below the target value of 900 ppm during mass attendance of students (Fig. 4).

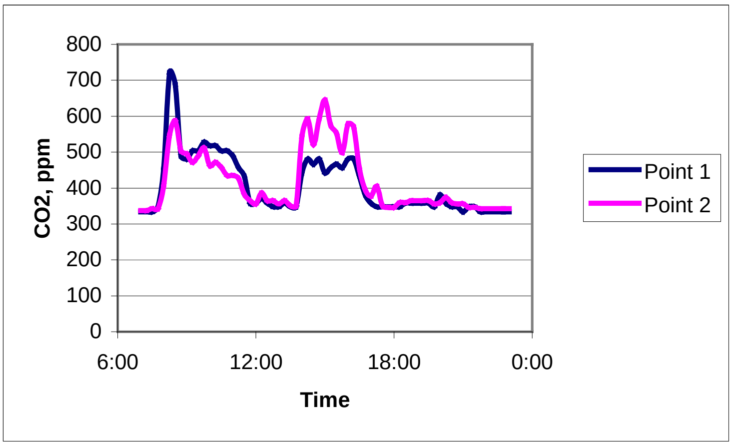


Figure 4. The concentration of CO_2 in the room air of educational laboratories during a typical school day.

The temperatures of indoor air stayed generally in between 21 ... 25 ^oC (Fig. 5).

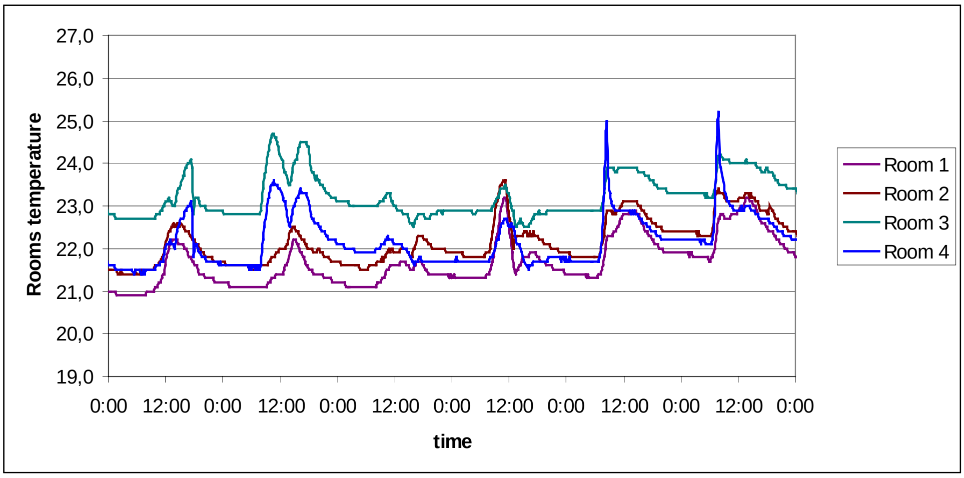


Figure 5. The temperatures of indoor air in different rooms.

Smoke tests proved that necessary over/underpressure in between different rooms was predominantly assured. During significant changes of air amount in the ventilation system (the opening of numerous fume cupboard doors at the same time), necessary pressure balance stabilised after up to 10 seconds.

In the course of monitoring air movement speed in fume cupboard holes was measured during different usage regimens. Necessary speed (0,5 m/s) stabilised in the work holes after 1…5 seconds. Changing the positions of the fume cupboard doors in neighbouring rooms connected to the ventilation system practically did not influence air speed in other fume cupboard holes connected to the same system.

The total heat energy consumption in 2006 of the building was 1 752 MWh and specific consumption was 222 kWh/m^2 (55 kWh/m^3). There is no official statistics in Estonia on energy consumption of laboratory buildings. The specific consumption is in the same order of magnitude as the indices of Estonian dwelling fond. Considering the air exchange in laboratories is significantly larger than in dwellings, the achieved energy expenditure is satisfactory.

Results of monitoring demonstrate that investments into these ventilation systems were justified and the installed ventilation systems corresponds to our technical and sanitary-hygienic expectations.

CONCLUSIONS

From the viewpoint of both investments and operation, it´s expedient to prefer central ventilation systems. A great number of fume cupboards and other local exhaust devices can be connected into a global ventilation system – in case of the case study there were 60-70. Separate local exhaust ventilation systems have to be provided to extract dangerous and abrasive materials. With the help of special short-time response regulation valves and respective room-specific automatics system, necessary indoor air quality requirements can be guaranteed in the laboratories with reasonable investment- and operation costs. While designing such ventilation systems, long ventilation stub networks should be avoided.

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Energy Saving Potentials In Laboratory Facilities In The Context Of Safe Environment

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SUMMARY

Laboratory facilities are known as significant energy consumers. High ventilation rates are required for providing safe and comfortable environment as well as optimum operation of the equipment. Air from laboratories involving chemical activity is exhausted to the outdoors and should be replaced by outside air. Recirculation is not considered a good practice with regard to safety. Protection of the outside surroundings is an additional environmental challenge. The purpose of the study is to reveal various ways of saving energy through minimizing the ventilation rates in laboratory facilities in the contest with providing safety conditions in laboratory space. Two key contributors are analyzed: the air flow requirements from laboratory exhaust equipment and the minimum air change recommended by laboratory design guides or engineering and safety practice.

INTRODUCTION

Providing safe environment, the health and comfort of the occupants is the primary goal of the design of laboratory air conditioning system [1]. Among air exhaust equipment in laboratories the fume hoods (fume cupboards) represent the most important energy consumer. Used to prevent the exposure to hazardous materials, the fume hoods could exhaust to the exterior a considerable amount of conditioned air. Reducing the air flow, while maintaining safe conditions, has been an enduring objective of HVAC industry. Most of the latest innovations in fume hood design aim to reduce the face velocity or to restrict the hood opening area. Their implementation in laboratory practice has been received with mixed reactions revealing psychological and habitual resistance from some the users. The face velocity accepted in practice as satisfying safety conditions may vary from 0.3 to 0.6 m/s. The most common design value is 0.5 m/s. Adopting low velocities of this range is not always received with confidence by the safety managers and the users comfortable with traditional values employed over the years. However, 20% to 40% reduction is generally acknowledged in practice at least for periods when the operators are not present in front of the hood. On the other hand, the measure of minimizing the hood opening area challenges the users on accepting restricted access to the work area. Some tend not to be appreciative, viewing it as work restraint. Limitations of the vertical sash opening or various combinations of vertical and horizontal sashes have been developed. Reducing the energy cost of operating the fume hoods is a multidisciplinary mission involving the designers, the managers and the users.

On the safety side, it is common in the design practice to impose a minimum air change rate (air changes per hour -ACH) as an operating limit of the ventilation system . The recommended values for various types of laboratories fit into a range of 4 to 15 ACH, sometimes higher. No method or procedure, other than practical considerations and observation, is known to be the source of these values. Adjusting this range on more justified basis has been objective of energy saving in laboratory ventilation. The concentration of hazardous substances in a laboratory environment is the major criterion in discussing the minimum ACH. The release of chemicals inside the laboratory environment could only happen by accident. A spill of a chemical could result from defective material, equipment malfunction or negligence. Laboratory protocols do not allow the use of hazardous materials in open laboratory space. Chemicals must always be handled in containment equipment such as fume hoods, biological safety cabinets or glove boxes. Regardless of the limits of the design of laboratory ventilation system, the air dilution or replacement could not protect personnel from exposure to concentrated bursts [2]. If this point of view is acknowledged, the current minimum air change rates in laboratories could be reconsidered based on calculations and measurements of the concentration levels.

METHODS AND RESULTS

Methods of reducing the laboratory fume hood exhaust flow rates are analyzed considering a variety of design solutions and operational measures. The possibility of minimizing the air flow is discussed from the safety view point, by evaluating the concentration of hazardous substances and comparison with the permissible safety limits.

Reducing the air flow of at fume hoods (cupboards)

A comparison of various fume hood design solutions for reducing the access area is presented in Table 1. A 1,800 mm wide fume hood is used for reference. A laboratory of $90\,m^3$ and $3\,m$ ceiling height is considered for evaluating the corresponding number of air changes per hour (ACH). In the concept of modular laboratory design the space is equivalent to a laboratory module.

Table 1. Comparison of design solutions

Design solution	Face velocity	Face opening	Air flow	Air flow reduction	ACH
	m/s	m^2	l/s	%	
Full open sash - 70 cm open	0.5	1.20	600	-	25
Vertical sash - 45 cm open	0.5	0.76	380	36	15
4 horizontal sashes on 2 tracks	0.5	0.57	285	52	12
Vertical sash at 45 cm open low velocity	0.4	0.57	285	52	12
Vertical sash at 45 cm open minimum velocity	0.3	0.57	215	64	9
Combination of vertical and reduced height horizontal sashes on 2 tracks	0.5	0.33	165	72	7

It results that this measure has a potential of reducing the flow rates more than three times of full open sash reference case. It should be noted that the rates of air change remain high. The room yet changes 7 volumes of air even in the most extreme saving solution. Further reductions are possible with the user’s input in employing the features offered by design along with a relentless sash position management.

The variable air flow control solution based on sash position (VAV) is a traditional measure of reducing the energy consumption of exhaust equipment. Employing the benefits of this solution involves a dependable attitude of the user of maintaining low or closed sash position. The technology of automatic sash position control has been developed; the sash moves safely as sensing the presence at the hood. The ideal scenario is that the sash should be actually open only during the loading, unloading or interacting with the process inside.

The current concept of low-velocity hoods is based on constant flow. Optimizing the pattern of the flow inside the hood enclosure is used to enhance the containment at lower face velocities. Such hoods have passed the required safety tests at velocities as low as 0.3 m/s. In the case of constant air flow (CV) hoods, the only means of reducing the energy consumption is using two position control settings for occupied/non-occupied periods. During the non occupied time, the hood face velocity is usually reduced to 60%.

A comparison of various methods under different operating conditions using as reference a conventional hood operating at constant flow (CV), with full open sash is illustrated in Fig. 1 A good management of keeping the sash closed 20 hours a day can potentially reduce the flow to 20%. For the 90 m^3 reference laboratory this represents a drop from 25 to 5 ACH

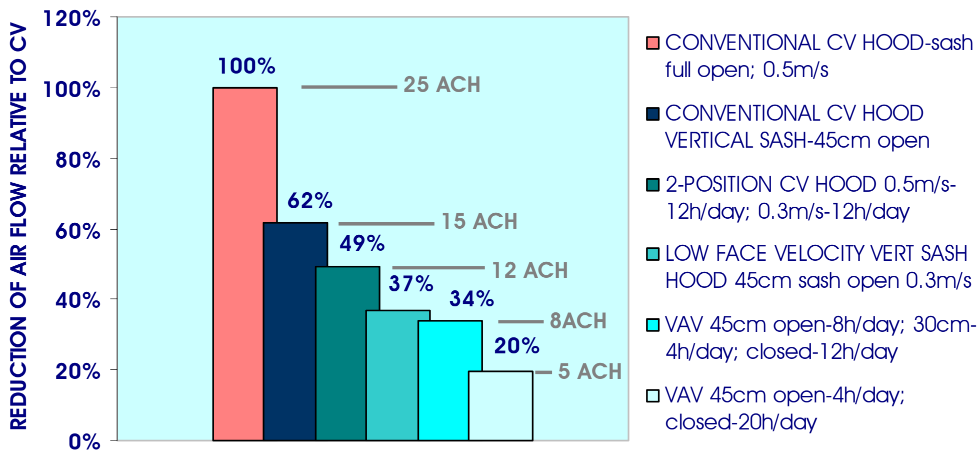

Figure 1. 24 hour potential savings using various fume hood design and operating conditions

Figure 2 shows the average air flow and air changes per hour in a 3-module laboratory (270 m^3) with the same hood in the similar operating configurations.

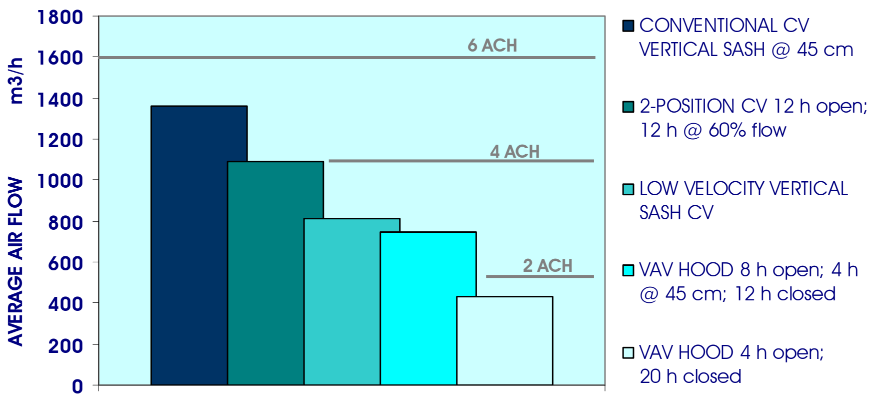

Fig. 2. Average air change rates from one 1,800 mm fume hood in a 3-module laboratory

With the chemical activity safely contained inside the fume hood the volume of air in the laboratory environment could be changed to as low as 2 to 4 times every hour. Reducing the exhaust flow from fume hoods should not be a concern with regard to safety of the outside environment. Along with the substantial dilution in the fume hood, combining the effluents from fume hoods into a single exhaust system provides further dilution and reduces the energy consumption [3] [4].

It could be concluded that various design solutions, along with proper fume hood usage and management, offer great opportunities of reducing the room air changes and consequently the energy consumption in laboratory environment. The remaining question is whether the low level of air change possible to be employed through these measures assures a safe environment in the laboratory, outside the fume hood. The next section intends to offer a response by focusing on the concentration in laboratory environment.

Minimum air change in laboratories

The discussion of minimum air changes per hour (ACH) is primarily related to the level of concentration of hazardous materials in the laboratory environment. An extreme condition of an accidental spill is analyzed.

The concentration of chemicals is neither steady, nor homogeneous because of continuous change of air patterns affected by the air diffusing system, people's movement or doors opening. The development of a generalized model that quantifies the influence of all interfering factors inside the space is not possible. In case of a spill, the maximum concentration at the source is diminished but the distribution in space is not known. Permissible exposure limits imposed by the safety codes must be satisfied regardless the location.

In safety practice, Permissible Exposure Limits (PEL) criterion was introduced, along with other similar criteria, for defining the concentration of chemicals in air under which it is believed that workers could be exposed without adverse effects.

This section evaluates the concentration produced by evaporation of a chemical liquid spill. The results are compared with permissible exposure limits (PEL) and correlated with air changes (ACH) in the laboratory space. The procedure of the analysis is described in Sandru and Xing [5]. The rate of evaporation of $1\,m^2$ of liquid area is calculated and used for calculating the average volumetric concentration in the $90\,m^3$ volume of the reference laboratory. 20 chemicals frequently used in laboratory practice, in the range from low to extreme health hazard are examined. Heat-mass transfer analogy was applied considering steady state conditions of a parallel flow over smooth liquid surface, at uniform liquid and air temperature. The mass flow rate from convective evaporation was determined by equation

$$\dot{m}_v = h_m A_v \left(\rho_{v,s} - \rho_{v,\infty}\right), \qquad (1)$$

where h_m is the average mass transfer coefficient. A_v is the area of mass transfer, $\rho_{v,s}$ is the vapor density at liquid-air interface and $\rho_{v,\infty}$ the vapor density in the free air stream.

The mass transfer coefficient was evaluated from heat and mass transfer analogy [6]

$$h_m = Sh \frac{D_v}{L} \qquad (2)$$

Sherwood number (Sh) was calculated considering parallel flow over a flat plate and D_v, the diffusion coefficient of vapor into air selected from CRC Handbook [7]

Volumetric concentration C_V (ppm) was calculated as

$$C_V = \frac{\dot{m}_v / \rho_{v,s}}{\dot{Q}} \times 10^6, \qquad (3)$$

where $\dot{Q}$ is the air flow rate in the space (m^3 / h).

The results are illustrated in Figure 3. Remarkably low air changes are shown to be sufficient for diluting the moderate and low hazardous substances. However the highly hazardous chemicals could not be safely diluted witin the air change rates currently accepted in laboratory design. It is obvious that increasing the ventilation air flow could not eliminate the hazard generated by an accidental spill.

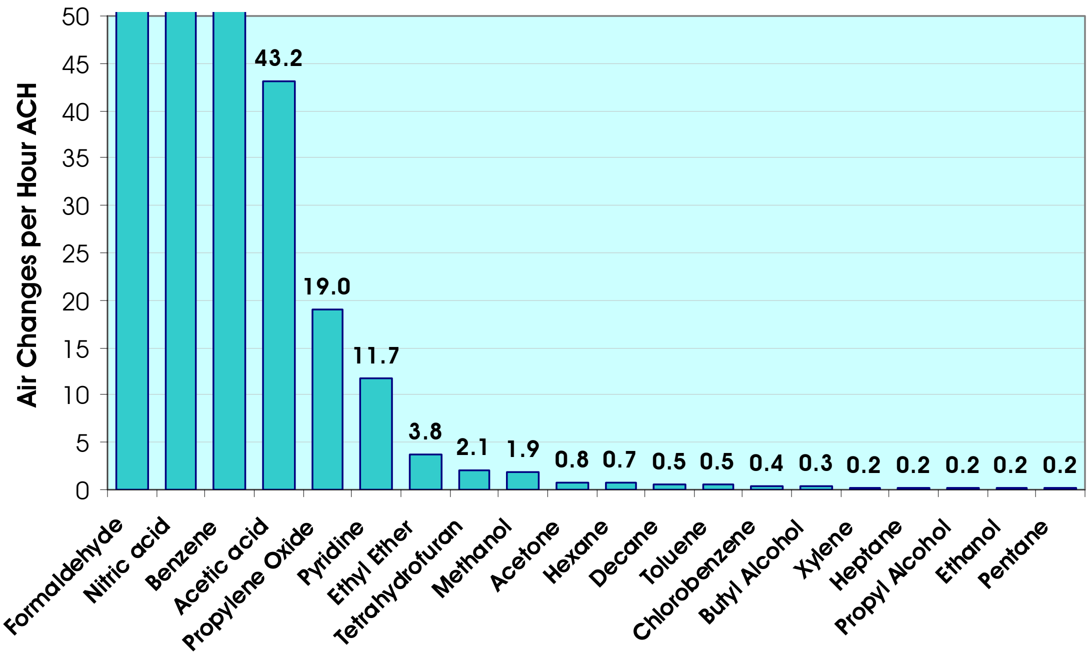

Figure 3. Air changes per hour required to attain permissible exposure limits. Evaporation of 1 m^2 of spill in 90 m^3 laboratory space.

Evaporation from an open tray of 0.04 m^2 in the 90 m^3 reference laboratory space is also evaluated for comparison. As in the case of the large spill the air velocity was selected at 0.2 m/s, simulating possible air drafts from laboratory ventilation. The results displayed in Figure 4 lead to the similar conclusions. With one exception, the low evaporation rate brings the concentration below the PEL at air changes within the current range of 4-15 ACH.

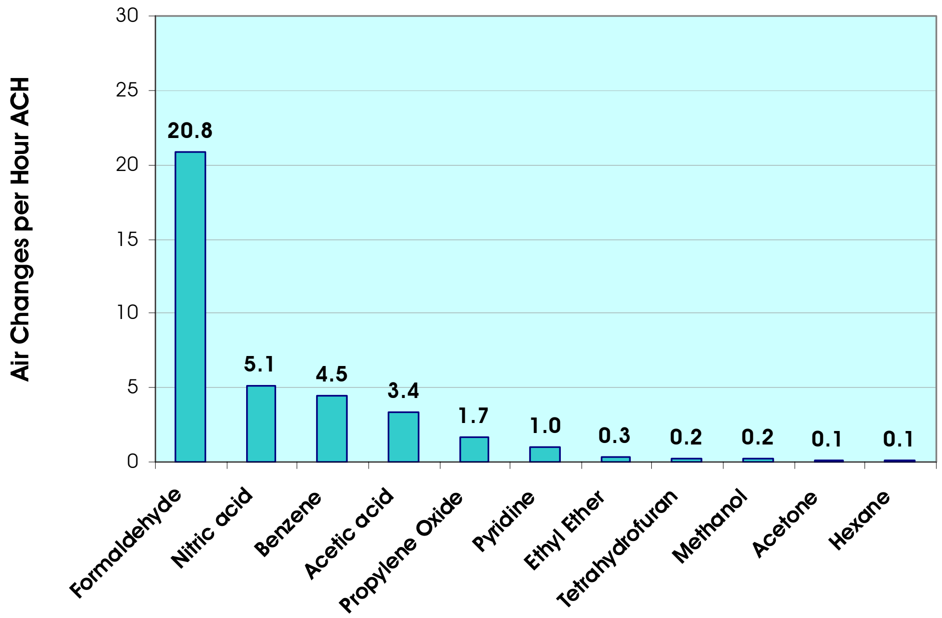

Figure 4. Air changes per hour required to attain permissible exposure limits. Evaporation from 0.04 m^2 open tray in $90\,m^3$ laboratory space.

DISCUSSION

Two ways of reducing the energy consumption related to laboratory ventilation have been analyzed. Various methods of reducing the average air flow exhausted from laboratory fume hoods have been compared. Current design solutions available in industry along with user's discipline in operation the fume hoods offer great potentials for energy saving. The analysis reveals the fact that a fume hood could operate at average flow rates as low as 20% of maximum flow in ideal conditions of operation.

Safety aspects of saving energy by adopting low flow rates in laboratories handling hazardous substances are discussed. The air changes per hour necessary to maintain the laboratory environment below permissible exposure limits are calculated in the case of an accidental chemical spill. The evaluation is theoretical considering the homogeneous distribution of concentration in space. The odor criterion has not been considered. The analysis proves that, regardless the level of the air flow, most hazardous chemicals could not be diluted to safe limits. Maintaining high flow rates could not avoid the hazard. Purging the space with high emergency flow rates could only provide a faster dilution before the space could be reoccupied. The results also show that extremely low air changes are sufficient to dilute the moderate and low hazardous substances. In this context could be observed that:
High flow rates are not justified to ventilate the space where chemicals are not present. Lower than traditional minimum flow rates could be considered in designing the laboratory ventilation system. The observations are based on the fact that the laboratory written code of behaviors does not allow the use of hazardous materials in open laboratory space and always requires immediate evacuation of the space in case of an accidental spill.

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Numerical Investigation on Ventilation Strategy for Laboratories: A Novel Approach to Control Thermal Comfort Using Cooling Panels

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SUMMARY

A novel ventilation system which introduces bench exhausts and radiant cooling panels is proposed. The benefits of using bench exhausts and radiant cooling panel system in removing heat from equipment intensive laboratory are quantitatively evaluated using CFD simulations. This study resulted in a recommendation for a ventilation strategy using a combination of ceiling exhausts, bench exhausts and ceiling radiant cooling panels that appears to give the best thermal condition and energy saving in the laboratory. The system offers the potential for application in new and existing research laboratories where a large portion of the space cooling load in a laboratory is a result of the heat produced by the research equipment on the bench.

INTRODUCTION

Laboratories are usually equipment intensive. The ventilation flow rate required to cool these laboratories is higher than in a less equipment intensive zone of the building. A large portion of the space cooling load in a laboratory is a result of the heat produced by the research equipment on the bench. If the heat can be captured at the source, its impact on space cooling load and the resulting HVAC requirements and cooling cost will be reduced.

Traditionally, HVAC systems are designed as All-Air Systems. These systems achieve the tasks of ventilating and cooling a building by convection only, which means that air is used to ventilate the buildings in order to maintain a high level of indoor air quality as well as provide thermal comfort in the buildings. An alternative is to separate the tasks of ventilation and thermal space conditioning through a combination of radiation and convection inside the building by using the forced air to fulfill the ventilation requirements and radiant cooling panels to provide most of the cooling. This is because radiant cooling can be more energy-efficient than air-based systems [1-2]. It requires less parasitic energy (pump and fan energy) to remove heat from a space. Since the walls are radiantly cooled, the air temperature can be warmer to achieve the same level of thermal comfort. The warmer air temperature results in lower energy loss to the outdoors. In addition, the radiant cooling system also reduces noise and drafts of air-based HVAC systems. The preferred installation of the radiant cooling panel is ceiling mounted, as this reduces air stratification and facilitates collection of condensation. To investigate the possibility of further reduction in cooling cost, the combination of benchtop exhaust system and ceiling mounted radiant cooling panels needs to be considered. The purpose of this study is

- to assess the performances of bench exhaust system and ceiling mounted radiant cooling panels in achieving required thermal comfort with reduced ventilation flow rate.
- to evaluate the saving of annual cooling cost by using bench exhausts and cooling panel at reduced ventilation flow rate.

METHODOLOGY

Computational Fluid Dynamics (CFD) uses numerical procedure to solve the conservation equations, Equation (1), that govern the airflow and heat transfer in a space. It is a very powerful and efficient methodology to investigate temperature and flow field in a room where there are many parameters involved [3-4]. Therefore, CFD is employed in this study [5].

$$\frac{\partial}{\partial t}(\rho\varphi) + div\,(\rho\,\vec{V}\,\varphi - \Gamma_{\varphi}\,\mathrm{grad}\,\varphi) = S_{\varphi} \qquad (1)$$

Where :

ρ = density
$\vec{V}$ = velocity vector
φ = dependent variable
Γ_{φ} = exchange coefficient (laminar + turbulent)
S_{φ} = source or sink term

The airflow in a ventilated laboratory is turbulent. In this study, the turbulence is simulated with the k-ε model [6-7] that represents the most appropriate choice because of its extensive use in other research applications, such as predicting mixing rate of a jet flow and modeling airflow in urban open space [8-9].

The Predicted Percentage Dissatisfied (PPD) introduced by Fanger and given in the ASHRAE guide [10] is a widely used index in assessing the thermal comfort [11]. This index can be estimated by equations that are based on an empirical investigation of how people react to environments. As each individual has a different perception of the climate produced in a building, any given climate is unlikely to be considered satisfactory by all. Therefore, 80% occupant satisfaction is considered good, or a PPD of less than 20% is good. The equations implemented in the study shown below are taken from Fanger's equations for Predicted Mean Vote (PMV) and PPD as given in BS EN ISO 7730: 1995.

<u>Definitions</u>

$$PMV = (0.303e^{-0.036M} + 0.028)\{(M - W) - 3.05\times10^{-3}[5733 - 6.99(M - W) - p_a] - 0.42[(M - W) - 58.15]1.7\times10^{-5}M(5867 - p_a) - 0.0014M(34 - t_a) - 3.96\times10^{-8}f_{cl}[(t_{cl} + 273)^4 - (t_r + 273)^4] + f_{cl}h_c(t_{cl} - + t_a)\} \qquad (2)$$

where

$t_{cl} = 35.7 - 0.028(M - W) - I_{cl}\{(3.96\times10^{-8}f_{cl}[(t_{cl} + 273)^4 - (t_r + 273)^4] + f_{cl}h_c(t_{cl} - t_a)\}$

$h_c = 2.38(t_{cl} - t_a)^{0.25}$ or $h_c = 12.1v^{0.5}$ whichever is the greater

$f_{cl} = 1.00 + 1.29I_{cl}$ for $I_{cl} \leq 0.078\ m^2KW^{-1}$ or $f_{cl} = 1.05 + 0.645I_{cl}$ for $I_{cl} > 0.078\ m^2KW^{-1}$

$$PPD = 100 - 95e^{-n} \quad (3)$$
where $n = 0.03353PMV^4 + 0.2179PMV^2$

List of Symbols

PMV = Predicted Mean Vote
PPD = Predicted Percentage Dissatisfied
M = Metabolic rate (W/m^2 of the body area)
W = External work (W/m^2 of the body area (= 0 in most cases))
Icl = thermal resistance of clothing (m^2KW-1)
fcl = Ratio of clothed surface area to nude surface area
ta = Air temperature (°C)
tr = Mean radiant temperature (°C)
v = Air velocity relative to the body (ms-1)
pa = Partial water vapor pressure in Pa
hc = Convective heat transfer coefficient (Wm^2K)
tcl = Clothing surface Temperature (°C)

CASE DESCRIPTION

A generic laboratory with a conventional air distribution system, shown in Figure 1, was developed as the baseline laboratory model. The same laboratory space was then modeled with the bench exhaust ventilation scheme at different exhaust flow rates and cooling panels of different arrangements for more than 30 cases among which 16 cases were presented in this paper as listed in Table 1. The bench exhausts used in this study were continuous slots along the length of the benches, mounted beneath shelves of the bench. The cooling panels were flushed on the ceiling above the bench top and aisle in as shown in Figure 1. One set that was mounted above the bench top, called bench panels (light blue colored in Figure 1), included 3 panels. The central panel was 0.6m wide, and the two against the side walls were 0.3m wide. The other set, called aisle panels, was mounted above the two aisles. These cooling panels were maintained at 13.9°C temperature. The total heat generation from the bench devices was 5808W. The lighting heat sources were 2275W. The sensible heat from each occupant was assumed to be 80W. Solar loading from south-facing windows on the external wall was divided as 1160W transmitted into the room and 1273W absorbed by the glass and the external wall section. The supply temperature was 11.1°C for all cases.

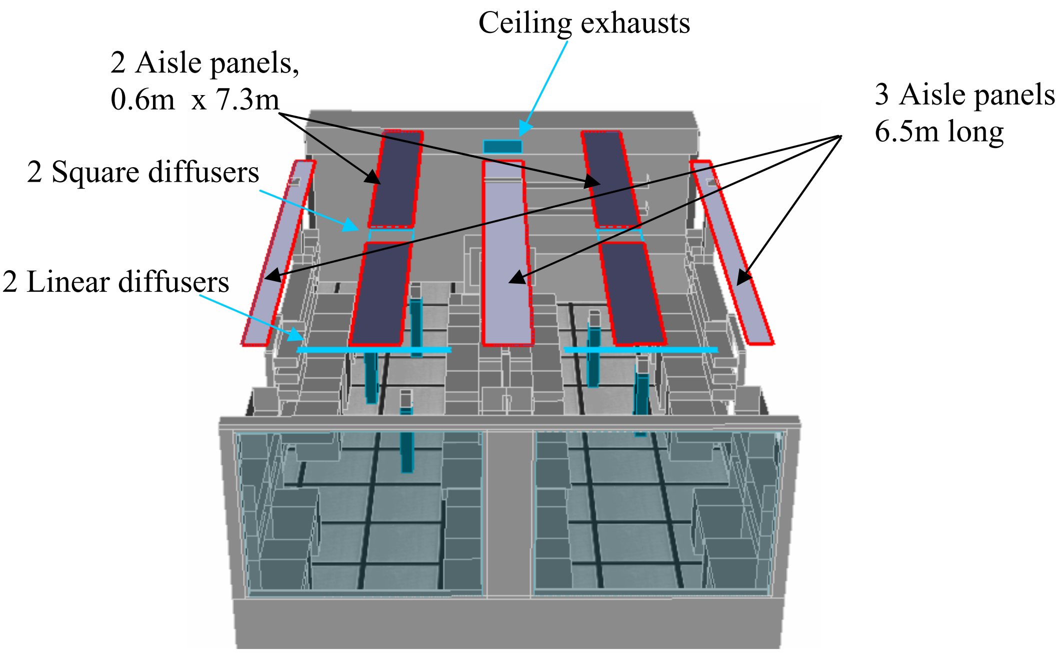


Figure 1. Laboratory layout

Table 1. Case description and results

	Cooling panels	ACH	Bench exh. CFM	Ceiling exh. CFM	Average Air T (°C)		Average PPD (%)	
					Walking zone	Bench zone	Walking zone	Bench zone
Baseline	-	13	0	-1750	21.1	22.2	12.8	12.8
Case1	-	8	-800	-370	24.0	25.4	7.6	13.0
Case2	Bench Panels	8	-800	-370	23.5	24.9	7.3	8.8
Case3	Aisle panels	8	-800	-370	23.5	24.9	7.3	8.9
Case4	Bench & Aisle	8	-800	-370	23.1	24.6	7.9	7.9
Case5	-	6	-800	-130	26.4	27.6	12.8	25.1
Case6	Bench Panels	6	-800	-130	25.5	26.8	9.1	16.0
Case7	Aisle panels	6	-800	-130	25.4	26.6	8.6	14.9
Case8	Bench & Aisle	6	-800	-130	24.9	26.2	7.4	12.3
Case9	-	5	-600	-208	27.4	28.7	16.7	30.0
Case10	Bench Panels	5	-600	-208	26.7	28.2	13.4	24.4
Case11	Aisle panels	5	-600	-208	26.5	28.0	12.3	23.1
Case12	Bench & Aisle	5	-600	-208	26.1	27.5	10.0	18.6
Case13	-	4	-600	-86	28.8	30.3	28.1	47.4
Case14	Bench Panels	4	-600	-86	28.0	29.3	19.5	32.1
Case15	Aisle panels	4	-600	-86	27.9	29.2	18.5	31.2
Case16	Bench & Aisle	4	-600	-86	27.5	28.8	15.0	26.7

RESULTS

In order to evaluate the performance of bench exhausts and cooling panels, the occupied zone is divided into two, the walking zone and the bench zone, as highlighted in a) and b) of Figure 2, respectively. The walking zone covers the regions of aisles and the doorways from the floor to

1.8m above, and the bench zone includes the volumes from the top of the bench to 1.8m from the floor.

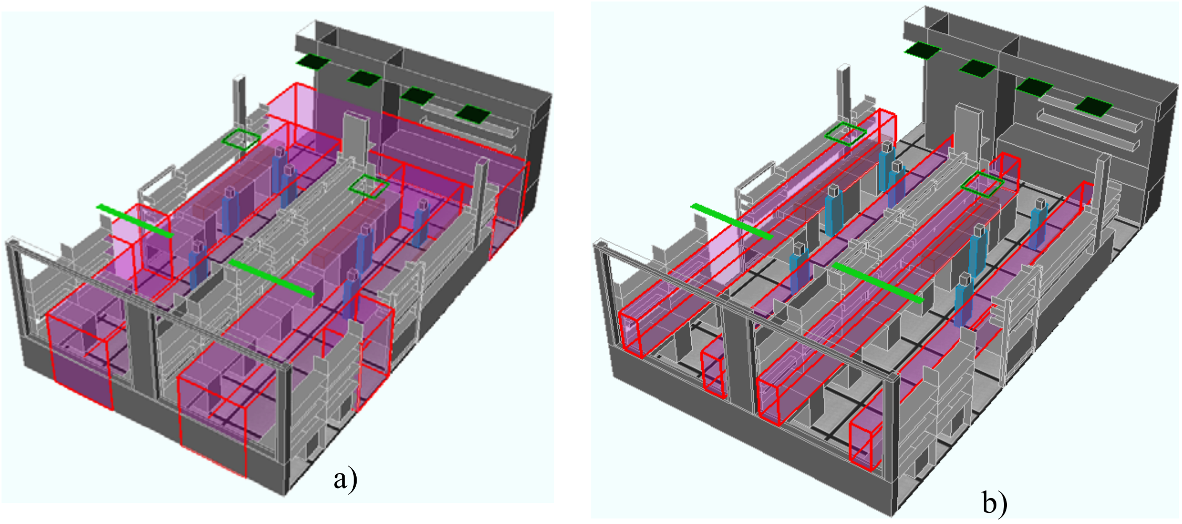
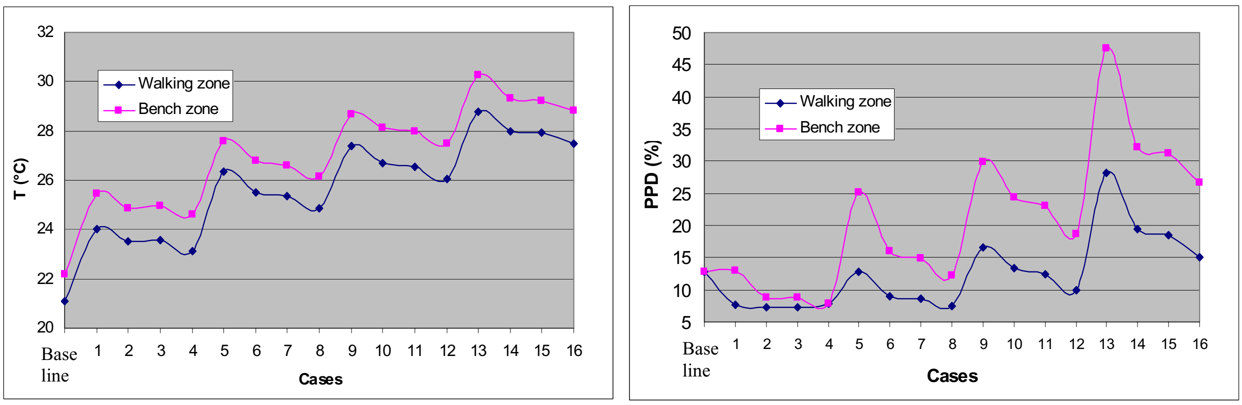


Figure 2. Two sets of occupied zones. a) Walking zone includes the 5 highlighted regions; b) Bench zone is the volume above the bench top in the 4 highlighted areas

Let's first discuss the effects of bench exhaust on thermal comfort. The average PPD and temperature in the two occupied zones are summarized in Table 1. They are also graphically presented in Figures 3 to demonstrate the trends. Table 1 shows that the PPD in the occupied zone is 12.8% in the baseline case with 13 ACH. When using bench exhausts, the PPD in the walking zone drops to 7.6% in Case 1 even at reduced supply flow rate of 8 ACH. When further reducing the supply flow rate to 6 ACH (see Case 5), the average air temperature in occupied zones increases by 2.3°C, and PPD, especially in bench zone becomes significantly higher. This indicates that, when using bench exhausts, the supply flow rate can be reduced from 13 ACH to 8 ACH to achieve similar level of thermal comfort in the lab.



a) b)

Figure 3. Average a) air temperature and b) PPD in occupied zones

Secondly, we would like to evaluate the performance of cooling panels. Comparing the cases of same flow rate with and without cooling panel, Cases 1 through 5 for example, shows that the cooling panels reduce the average air temperature in the occupied zones by about 0.8°C to 1.4°C. When cooling panels are used, sensible heat is removed from the room by both ventilation and radiation. Through radiative heat transfer, the heat emitted by occupants is absorbed by the cooling panel. Therefore, even the average air temperature in the occupied zone is reduced by only 1°C, the thermal comfort can be improved significantly as seen in Table 1 or Figure 3. It is especially so when the supply flow rate is lower, for example Case 13 and Case 16. Table 1 shows that at 8 ACH, the PPD in the walking zone increases slightly if the two sets of cooling panel are used together. This is because the air temperature is already slightly lower than the desired temperature of 23.5°C (74.3°C), cooler temperature of surrounding surfaces can have negative impact on thermal comfort. With aisle panels, the air temperature is, in general, lower in both walking and bench zones than with bench panels since the total surface area of aisle panels is larger. The average PPD in the two occupied zones at 6 ACH all drop below 20% with any cooling panel arrangements. At 5 ACH, however, it requires both bench and aisle panels working together to bring PPD below 20% in occupied zones. At 4 ACH, even combination of the two sets of cooling panels cannot bring the PPD to lower than 20% unless more cooling panels are installed.

OPERATING COST REDUCTION

The ventilation flow rate required for equipment-intensive laboratories to be thermally comfortable can be as high as 13 ACH, referring to the baseline case. With the proposed bench exhausts, the ventilation rate can be reduced to 8 ACH (Case 1) to achieve similar level of thermal comfort. The combination of bench exhausts and ceiling mounted radiant cooling panels can further bring the ventilation flow rate down to 5 ACH as seen in Case 12. Figure 4 illustrates the annual cooling costs of these three cases estimated for a typical lab located in Washington DC area.

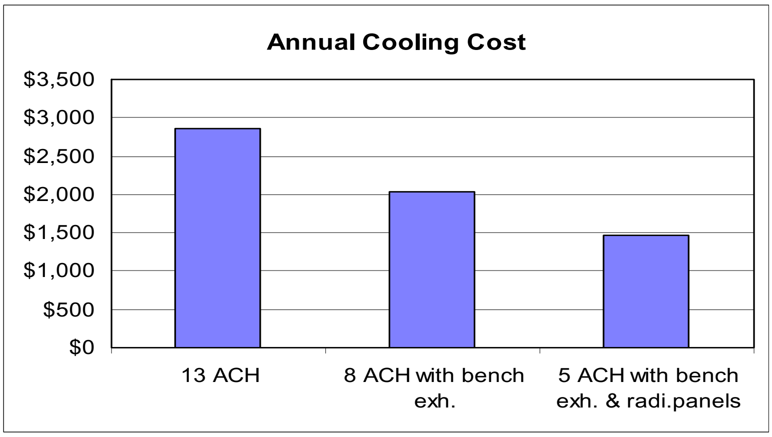


Figure 4 Saving in annual cooling cost of a lab

Comparing 13 ACH and 8 ACH, both without cooling panels, the total saving in annual cooling operating cost by utilizing bench exhausts alone is around 29%. Comparing the cooling costs of 8 ACH without cooling panel and 5 ACH with cooling panels, the saving by using cooling panels alone is about 28%. The total saving for annual cooling season when using bench exhausts together with radiant cooling panels is around 49%. The following conditions/assumptions are used in this cost calculation.

- The outdoor condition is taken from weather data in Washington DC.
- The cooling season is considered to be 4489 hours annually.
- The percentage of outdoor air is 70% for 13 ACH case and 100% for other two cases.
- Supply air temperature is 11.1°C (52°F), and the desired room temperature is 23°C (73.5°F) which is used as return air temperature in the calculation.
- Cooling load per CFM is considered to be the difference in air enthalpy when entering and leaving the HVAC system. Perfect duct insulation is assumed.
- The ventilation flow rate (CFM) in Table 1 is the flow rate required during the peak load of the day. The average cooling load of a day is assumed to be 64.3% of the day's peak load. So is the ventilation flow rate used in the cost calculation.
- The cost of electricity is 0.1$ / KWH, fuel is 8.0$ /100Btu; chilled water generation efficiency is 1.0KW / TON; fan efficiency is 68%.

DISCUSSION

The following conclusions are drawn from this study:

- For laboratories with high bench heat sources, using bench exhausts in conjunction with ceiling exhausts improves the thermal condition in the occupied zone at even relatively low total ventilation flow rates.
- Without cooling panels, the case of 8 ACH with 30% to ceiling exhausts and 70% to bench exhausts (Case 1) appears to give the best thermal condition in the laboratory.
- With bench and aisle radiant cooling panels, the ventilation flow rate can be reduced to 5 ACH while maintaining the PPD in occupied zone below 20%. The cost reduction of annual cooling associated with using bench exhausts and radiant cooling panels is about 49% for a typical lab located in Washington DC, USA.

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Performance Based Design for Ventilation Systems of Operating Rooms Supported by Numerical Simulation - Discussing the Methodology

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SUMMARY

This paper discusses a methodology to support the performance assessment of ventilation systems of operating rooms (ORs), with a further focus on the application of numerical simulation tools in this design assessment. The performance based design should provide good information quality that advises the needs and expectation of the stake holders. The methods consist of literature review, the development of an information matrix to assess ventilation systems of ORs, and two case studies that will elucidate the approach and verify its applicability. The methodology chosen combines the performance base approach with the Building Evaluation Domain Model. The results will permit to identify which type of numerical simulation tools is best applicable for the assessment of a specific performance indicator and how the assessment should be performed. The conclusions will indicate directions for future work and which areas need to be focused on.

INTRODUCTION

The ventilation system in an OR is responsible for keeping a good indoor air quality in ORs. Previous research has shown that some types of ventilation systems used in ORs can contribute to the reduction of surgical site infection [1, 2, 3, 4 and 5]. They furthermore establish working conditions for the operating staff.

Several different ventilation systems/strategies are available for application in ORs. However, no general assessment methodology is available to compare different ventilation systems for a given design problem towards identified performance requirements (e.g. indoor air quality). The focus in the research therefore is on the design decision support information that will provide a methodology to support the design assessment for the ventilation system in ORs.

The objective is to discuss a methodology to support designers and decision makers in the objective performance assessment of here applied as an example, the ventilation design for ORs. The performance based should allow for a good information quality, and balances the needs and expectations of the stake holders. It will permit application for different types of ventilation system and surgeries.

This paper discusses the primary results of a PhD study [6]. After a brief introduction, the method will be described which has been used in order to develop the “Performance Based Design for Ventilation Systems of ORs”. The result will be presented in a matrix, and will be used to discuss some points and two case studies. Finally the conclusions and directions for future work.

METHODS

The methods consist of literature review, the development and the discussion of an information matrix of the assessment of design for a ventilation system of ORs supported by numerical simulation, and finally, two case studies will be presented to elucidate the approach and to verify the applicability of the evaluation proposed.

The development of the performance assessment methodology has a basis in the Building Evaluation Domain Model (BEDM) that was developed by Mallory-Hill, 2004 (see Figure 1). This model can be visualized as a three dimensional matrix. The three axes in this case refer to ‘Human Systems’, ‘Building Systems’ and ‘Architectural System’ levels. A subdivision is provided for each axis. The Human System level (HSL) focuses on the stakeholders ranging from the individual occupants to the global community and the performance specifications that are set by them. Naturally, the individual occupant is most interested in the basic performance value (e.g. working conditions). For the global community, e.g., the functional value is more important. The Building System level (BSL) is subdivided in several levels that relate to the number of changes that may occur during the building life time. Finally, the Architectural System level (ASL) has levels that subdivide the level of detail for decision making.

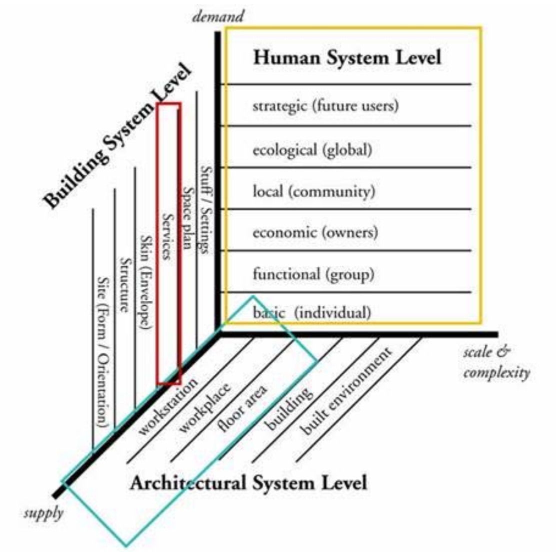

Figure 1. Building Evaluation Domain Model [7]

For example, for the ASL, the operating room is divided in workstations, workplace and floor area. The workstations now are defined as the zones of the patient, surgeon and auxiliary nurse, and supporting staff including the anaesthesiologist. The workplace encompasses the total environment of the operating room, while the floor area includes also the hallways, antechamber, storage and other adjacent areas that are directly connected to the operating room. For the HSL in principle all levels may be included with respect to the operating room. Focus, naturally, is on the individual, group, future users and owner level. The individual refers to the individual persons in the operating room and the group to the surgical team as a whole. The future users, in this case, need to accommodate the needs of many different occupants and, for example, deals with the adaptability for developments in the medical field.

The BEDM encompasses the whole building. For the assessment of the ventilation system in an operating room, only parts of the model are of interest. Figure 1 indicates the parts of the model that at first instance will be taken into account. One can see that the attention is focused to one slice of the matrix.

Next step is to identify for each point in the model the performance requirements that can be set. Figure 2 presents a part of the Performance Assessment Methodology. One can notice the axes of the BEDM. Now for each point Functional requirements have to be identified and performance indicators have to be related to these requirements. For this development literature study, in combination with expert interviews and information from attending surgical operations have been used. The list of requirements and indicators is quite large, nevertheless, several indicators overlap. Furthermore, only those indicators that have an evaluation possibility in the design (simulation) and use (measurement) phase are regarded, in order to adhere to the performance based approach.

The final result will permit to identify which type of numerical simulation tool is best applicable for the assessment of a specific performance indicator and how the assessment should be performed.

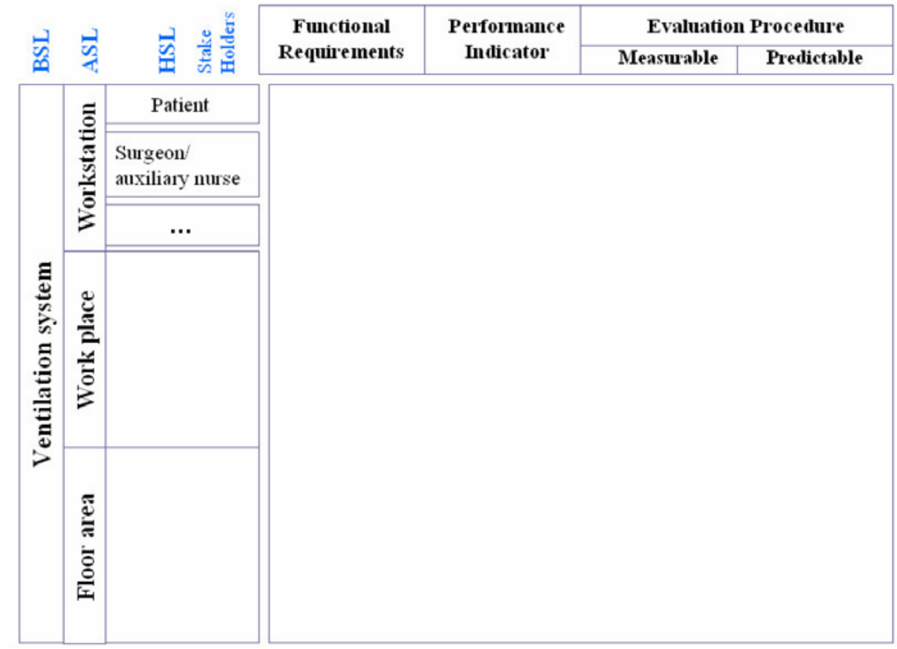

Figure 2. Performance Assessment Methodology [6]

RESULTS AND DISCUSSION

Figure 3 presents a part of the matrix that has been developed. Notice that it specifically addresses the performance assessment of ventilation systems for an OR. Compared to Figure 2, three columns have been included: OR zones, Performance requirements and How. The ‘OR zones’ column was included to define the workstation for each stake holder. The assumption is that in ORs several subzones can be identified which can be given different Functional requirements. This column furthermore is important to identify stake holders that have more than one workstation, e.g. the surgeon. Next, the Performance requirement column indicates the physical parameters that should be assessed in order to determine the performance indicator. The columns ‘How?’ describe the steps that are required to evaluate the performance indicator using experiments and numerical simulation tools. This may be read as recipes for performing the assessment. Although several performance requirements have been evaluated, this paper will focus on the surgeon thermal comfort in the zone 2. The correspondent row is identified in grey in Figure 3. The example will be used to describe the procedure.

		Stake Holders	OR zones (involved)	Functional Requirement	Performance Requirement	Performance Indicators	Evaluation Procedure: Measurable	How?	Predictable	How?
VENTILATION SYSTEMS	WORKSTATION	Patient	Zone 1	Safety (SSI rate)	Airborne Infection	CFU/m^3		M1		P1
					Hypothermia			M2		P2
		Surgeon	Zone 1 and Zone 2	Ergonomic (health, productivity and satisfaction)	Indoor Air Quality			M3		P3
								M4		P4
								M5		P5
					Thermal comfort	PMV index	Yes, but indirectly	M6	Yes	P6
						Local Thermal Comfort	Yes	M7	Yes	P7
					Visual comfort	LUX / CRI		M8		P8
					Acoustic comfort	dBA / PNC / PSIL		M9		P9
		Auxiliary nurse	Zone 1 and Zone 2	Ergonomic (health, productivity and satisfaction)	Indoor Air Quality					
					Thermal comfort	PMV index				
						Local Thermal Comfort				
					Visual comfort	LUX / CRI				
					Acoustic comfort	dBA / PNC / PSIL				
		...								

Figure 3. Performance Based Design for Ventilation Systems of ORs – draft version [6]

In order to better understand the evaluation and discussion of the literature review, two points will be explored: how previous researches assessed surgeon thermal comfort and what parameters were considered in the analysis. After that, the case studies will be discussed.

The thermal comfort in relation to environmental variables depends on metabolic rate, clothing, air temperature, mean radiant temperature, air velocity and air humidity [8]. In ORs these variables can change in accordance with the type of surgery. Managing the thermal discomfort of the surgeon is complex, because it is usually not possible to reduce the amount of clothing, to alter the activity being performed, or to reduce the various heat sources used during the surgery. This problem has been discussed and evaluated by several researchers [4, 9, 10, 11, 12, 13, 14, 15, 16, 17, 18 and 19]. These researchers evaluated the surgeon thermal comfort using Fanger's PMV model [20], with the exception of one study that used a thermal physiological model. Many of these studies were experimental and only few used numerical simulation to predict the thermal comfort. The surgeon thermal comfort reported in the literature review was between slightly warm and very hot - this difference resulted of the type of surgery performed, some times of the ventilation system used, or of the environment variables that were not necessarily the same.

Next, two case studies will be presented to verify if the approach proposed in Figure 3 is applicable, and with them, more details will be discussed. The focus is on the last column "How" for a specific performance indicator - the thermal comfort. This step is divided in three parts: first describing how the performance indicator can be assessed using numerical simulation; second describing the case studies; finally, the application will be discussed.
In accordance with Figure 3, the surgeon thermal comfort can be predicted, e.g., using PMV index and Local Thermal Comfort. The Local Discomfort can result of draught, vertical air

temperature difference, warm and cool floors, and radiant asymmetry. The PMV index can be calculated using Building Energy Simulation models (BES), while the Local Thermal Comfort generally would require the use of the Computational Fluid Dynamic (CFD) technique. These approaches have different resolution level.

When the OR is in the use phase, the variables at the surgeon workstation can be assessed through experiments. In case the OR is in the design phase, it will be possible to predict the variables using numerical tools. If one applies CFD it is possible to zoom in at the surgeon workstation which will permit to look at the problem in detail, identifying the variables and parameters that influence the thermal comfort. From the literature and observations in ORs identified the parameters which need to be taken into account include: characteristics of the OR, medical equipment (type and number), position of the operating table and of the patient; requirements; layout performed in the surgery, people (number and function), types of clothes and ventilation system design. Another important parameter to consider is the surgical lights effects.

In the case studies, two types of surgery are evaluated: a Hip and a Foot, both orthopaedic surgeries. The characteristics of the OR: 36 m^2 and h = 3.50 m. A 'Large downflow Plenum' ventilation system is used and its area is represented thought the dash-line, corresponding to 9 m^2. The layouts performed in each surgery are reported in Figure 4, which were designed through observations in ORs. The numbers correspond to: surgeon (1); others people of the staff (2, 3, 4, 5 and 11); instrument tables (7); back table (8); and equipment – defibrillators and accessories (9), anaesthesia, respiratory ventilator and vital signs monitor (10), and X-ray (12). The surgical lights are also considered above the surgeon focussed at the wound.

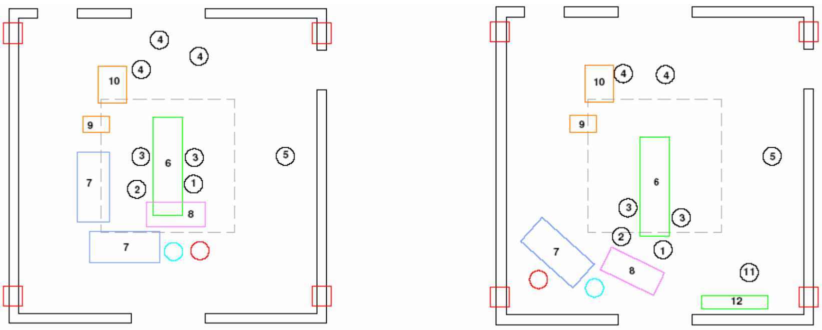

Figure 4. Layout Hip and Foot surgeries, respectively

In both surgeries the surgeon thermal comfort can be calculated using the PMV index. However, a more accurate evaluation, and in order to identify the local thermal discomfort caused by the supply air and the surgical lights will only be possible using CFD. In accordance with the layout performed in the Hip surgery, the draught could have a big influence on the local thermal discomfort, because the surgeon is positioned in the downflow area. Therefore, zooming in at this zone, it will be possible to see if the thermal discomfort is significant. In the Foot surgery the draught probably is not significant as the surgeon is not positioned in the downflow area. Therefore, calculating the PMV index would be enough. However, if we want to assess the local thermal discomfort caused by the temperature gradient, it is important to use the CFD. In the Foot surgery, the surgeon is in a mix zone, the same situation that occurs when there is mixing ventilation system in the OR. Probably, in the mix zone the surgeon will present more thermal discomfort than the surgeon below the supply air, because the air temperature will be higher and the velocity of the air is lower.

CONCLUSION

The purpose of this paper was to discuss the methodology and the approach to support the design assessment for ventilation systems in ORs. The BEDM was taken as the point of departure for the development of the Performance Based Design. The numerical simulation is a useful and important tool in design phase, permitting to observe if the ventilation system adhere to the needs for specifics surgeries. In the evaluation procedure, it was identified the types of numerical tools that could be applicable for the assessment of the surgeon thermal comfort. The case study permitted to identify benefits from the method and to verify the applicability of the evaluation proposed.

In the other hand, for the example discussed, although previous studies evaluated the thermal comfort in ORs, there is yet a strong need for future research to identify performance indicators that permit to assess efficiently the thermal comfort in ORs. Next step will be to complete the matrix presented in Figure 3, to evaluate other stake holders, performance requirements and indicators, and the other architectural system levels.

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Validation of a Neutral Pressure Isolation Room

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SUMMARY

The principles behind a new design of isolation room are set out. The process of validating the design is described using a range of physical modelling techniques. A CFD model was also constructed and compared to the basic operational mode. Characterisation of the internal airflow patterns was used to assess the level of mixing under steady state conditions. The techniques were extended to cover the response of the design to various challenges such as door opening. There are two reasons for using an isolation room – the patient is a source of infection and the patient is at risk from infection. The concept of a protection factor is introduced as a measure of the effectiveness in protecting staff or a patient against a source of infection. The PPVL isolation room provides a protection factor of 10^5 between the corridor outside and the isolation room (all doors closed) and a factor of 10^3 within the isolation room.

INTRODUCTION

The UK Department of Health (UKDH) has issued guidance for the design of a neutral pressure isolation room. The design incorporates a positively pressurised ventilated lobby (PPVL). The PPVL provides a barrier to airborne infection originating within the isolation room (i.e. equivalent to the negative type isolation rooms) and a barrier to airborne infection originating in the corridor (i.e. equivalent to the positive type). A static pressure differential between the isolation room and the adjacent corridor of close to zero is intended in the design - hence the term neutral pressure isolation room. [N.B. It is probable that the UKDH will adopt the terminology of PPVL rather neutral to make its purpose clear and unambiguous].

Air is supplied mechanically into the lobby through a central ceiling mounted supply diffuser. The supply flow rate was 220 $l.s^{-1}$. The majority of the air supplied to the lobby passes into the isolation room (160 $l.s^{-1}$) via a pressure stabiliser fitted above the door between the lobby and the isolation room. A proportion of the air leaves the lobby and enters the corridor via gaps around the door. Air also enters the isolation room via gaps around the door below the pressure stabiliser. Air is extracted from a ceiling mounted grille in the ensuite bathroom and so air is drawn from the isolation room via a low-level transfer grille in the ensuite door. The extract flow rate was 160 $l.s^{-1}$. The design intent is for around 10 air changes per hour ($ac.h.^{-1}$) to be delivered to the isolation room with good mixing, which is equivalent to 60 $ac.h^{-1}$ in the lobby.

The mechanical stabiliser is set to maintain a nominal pressure difference of +10 Pa between the lobby and the isolation room when all doors are closed. As the static pressure differential between the isolation room and the corridor is intended to be close to zero, the static pressure in the lobby is also designed to be around +10 Pa above the corridor when all the doors are closed.

In a PPVL isolation room, effectiveness is achieved by means of diluting airborne pathogens in the vicinity of the patient by a well-mixed pattern of room air movement. Therefore, the contaminant is removed from the isolation room by means of the mechanical extract system. Tracer gas (nitrous oxide) was used to simulate airborne infections. It was released into the isolation room and its concentrations were measured around and outside the room using tubes connected to analysers, to investigate the dilution, transport and removal of a contaminant. Further tests were done with the source in the lobby and with doors being opened and closed to simulate personnel movement.

Dilution ventilation is only one of the protective features afforded by the design. Challenges such as high-speed ejection of infectious material (e.g. a cough, sneeze or excretion of bodily fluids) or direct contact routes of infection are handled by use of appropriate PPE and/or barrier nursing procedures. Other factors come into play when considering the situation where the patient is at risk from their surroundings. However, this paper is not concerned with operational and maintenance issues [1] or the rationale for selecting a particular ventilation strategy.

The following schematic is taken from the recommended guidelines [1], upon which the principles of design and operation were based, where the minimum requirements for such a facility are listed.

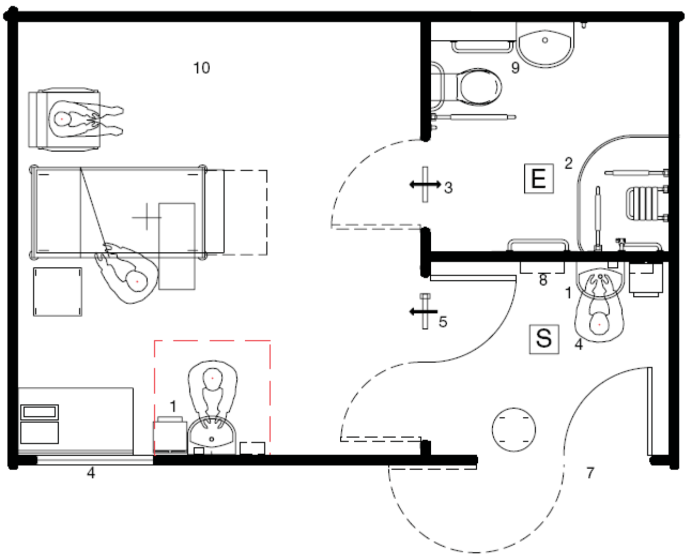


Figure 1. Single Room for Isolation [1]

The independent validation exercise was carried out to build and test a PPVL isolation room in a laboratory. It was decided in the outset of the project to validate the performance of the PPVL design with the following key issues:

1. Can the PPVL concept be built (within the test facility environment)?
2. Is it a robust design (from the point of view of construction, commissioning, maintenance, integration and testing – as far as the laboratory permits?)
3. Are the preferred design flow patterns achieved within the space, and are they stable?
4. Is the performance of the facility compromised with the lobby door left open to the main isolation room?
5. How do the internal conditions withstand challenges from door openings and personnel movement?
6. Is the thermal comfort of the patient and staff acceptable under normal operating conditions?
7. Is the performance acceptable with regards to infection control?
8. How does layout of furniture affect the room performance?

METHOD

Test facility

The design was based upon a specific configuration of PPVL isolation room, adapted for the laboratory environment. The isolation facility was constructed within the boundaries of an existing purpose built chamber, in order to maintain stable conditions and hence repeatable results. As far as practically possible, preferred suppliers and fitters of the Health Service were used to fit out the chamber. This helped to aid the authenticity of the project and to highlight any key issues with building the design. The construction included door fitting and sealing, pressure stabiliser balancing, and the use of hospital furniture and fittings within the isolation room, ensuite and lobby. The adjacent laboratory space still within the outer environmental chamber was used to model the corridor of the hospital.

Figure 2. View of completed PPVL test facility

Measures to provide a stable and quantitative environment for the test facility were employed from the onset to gain repeatable results. Cables and sample tubes were passed into the chamber through sealed glands, data acquisition and power metering of all heat loads (IT, occupancy and lighting) within the facility was used, and volumetric flow rates from the mechanical supply and extract were monitored using Air Movement and Control Association (AMCA) standard nozzle boxes. The dedicated data logging system with custom designed software took regular spot measurements of air temperature, air speed, power consumption, internal and external wall surface temperatures and inter-room static pressure differentials. Further details on the techniques used within the project are presented elsewhere, see [2] and [3].

Test Procedure

The principal aim of the test facility was to establish whether the ventilation patterns inherent in the isolation room were operating as a single, well-mixed zone. Sufficient mixing of fresh air from the lobby with the room air, effective extraction from the ensuite, and maintaining the pressure barrier with the stabilisers, were the key aspects in making the design work.

The design also required a wholly stable environment from which tests could be repeated with confidence in producing repeatable test results. All instrumentation used that is key to understanding the physics of the facility was calibrated at regular intervals to provide confidence in the results.

Multiple routes to validate the design were explored throughout the project, including:

- Air leakage performance through pressurisation to commissioning standard in [1].
- Room Air Movement (RAM) survey to measure temperature and air speed over a regular 11 x 13 grid at 11 heights within the facility. This provides information to calculate the thermal comfort parameters, such as draught risk.
- Flow visualisation using smoke pencils for local effects, and smoke generators for facility air mixing visualisation.
- Gas Tracer Tests (GTT) to study the release of nitrous oxide (N_2O) into the isolation room as the source of contaminant from the patient, analysed at multiple points within the facility through analysers. Protection levels and air change rates were deduced.
- Computational Fluid Dynamics (CFD) to accurately model the facility as built and to compare predictive computed results against the physical results found with the RAM surveys.

RESULTS

Fundamentally, it was found that the chosen configuration could be built, and that the isolation room does operate as a single zone and as a well-mixed space under standard conditions.

The design provides a nominal 10 ac.h^{-1} from the mechanical ventilation supply to the lobby, working with the pressure stabilisers. The test facility exhibited approximately 8 ac.h^{-1} within the isolation room, demonstrating the predominantly well-mixed behaviour that was predicted.

The main findings are reported below as an overview. Figure 3 shows air temperature and air speed results from two RAM surveys within the facility, one under standard operating conditions (Test 5), and the other when the lobby door to the isolation room has been left open (Test 6). It would not be expected that the door would be left open for a prolonged period, however tracer gas tests demonstrated that the isolation room still behaved as a single zone, but with air interchange between the lobby and the isolation room. The mixing in the isolation room (door closed) was characterised by a factor of 0.89; the corresponding factor for door open was 0.53.

The plots in Figure 3 demonstrate the supply air jet from the balancing diffuser entering the isolation room (door closed), and highlight the uniform distribution of air temperature within the entire space. Satisfactory levels of thermal comfort are experienced across the floor space. Opening the lobby door removes the supply jet as expected, and introduces a level of stratification to the facility. Analysis of the results was performed for four tests, with results being averaged for each anemometer height within the zone that were conducted under thermally steady state conditions:

Test 3: All doors closed, no furniture in room, original pressure stabiliser used
Test 4: All doors closed, no furniture in room, alternative pressure stabiliser used
Test 5: All doors closed, furniture introduced, alternative pressure stabiliser used
Test 6: Lobby to isolation room door open, with furniture in room

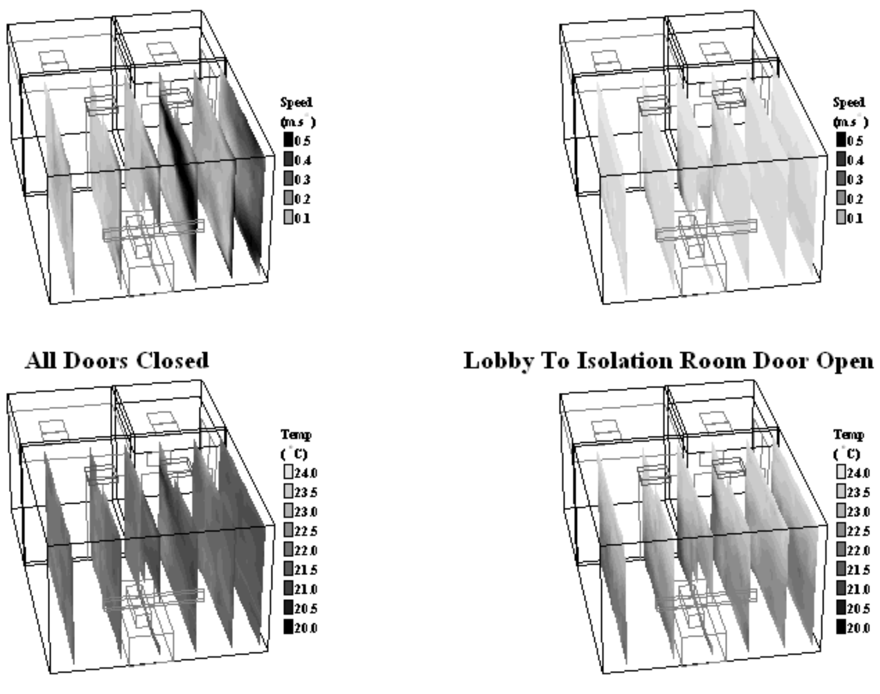
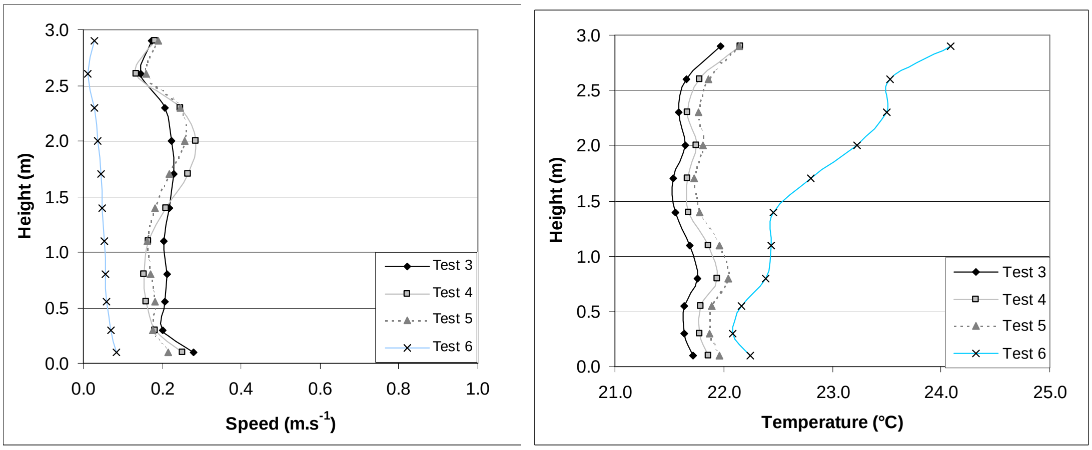


Figure 3. RAM Survey visualisation plots, Tests 5 and 6.

Figure 4 demonstrates the room averaged air temperature and air speed profiles at the anemometer heights within the facility. Good levels of air mixing within the space can clearly be seen in Tests 3, 4 and 5, with the significant temperature stratification present in Test 6 when the lobby to isolation room door is opened.



a) b)

Figure 4. RAM Survey thermal analysis. a) Speed ($m.s^{-1}$), b) Temperature (°C).

The distribution of contaminant (tracer gas) with the patient as a source has been studied and reported. [4]

Single failure modes were explored within this project and included supply and extract fan failures. The simulated supply fan failure indicated no significant leak of tracer gas into the corridor, with the isolation room remaining effectively well mixed and under negative pressure. Extract fan failure lead to increased levels of detection within the room as expected, but the lobby still provided a layer of protection to the corridor. A patient vulnerable to infection would not be at a significantly higher risk as long as the doors remained closed in this particular failure mode. The worst-case failure mode explored within this project is for the lobby door to be left propped open. Tests show that under these conditions the tracer gas can pass into the lobby, with barely detectable levels reaching the corridor.

Determination of the level of protection presented by the isolation facility (from the PPVL) was a major objective in the project. Minimum detection levels of 10ppm (parts per million) of N_2O against 100% (10^6 ppm), implies a dilution ratio of 10^5. This is expressed as the protection factor.

The principal conclusion is that the PPVL design is considered to be validated. The main qualifications to this principal conclusion are set out below:

- Full size room with ensuite successfully constructed following published guidance. Construction follows ethos of guidance with standard lighting and HVAC components. Specific recommendations were followed with regard to key features such as pressure stabilisers.
- Isolation room functioned as expected once commissioned.
- Preferred flow patterns were characterised by smoke, anemometry techniques, CFD prediction (base case) and tracer gas tests. Stability and repeatability were clearly demonstrated.

- Transient issues were addressed mainly through remote door operation to simulate personnel movement. Only scenarios compatible with proper use were considered (e.g. scenario such as both doors open was not relevant).
- Standard thermal comfort indices provide standard basis for assessment and comparison but individuals vary widely in their perception. Medical conditions may have significant impact. Potential mismatch between patient and nurse clothing level and metabolic rate is a consideration applying to all types of isolation room.
- Airborne infection covered for scenarios where tracer gas is a good model for the infection mechanism. A 10^5 protection factor was introduced and established.

DISCUSSION

The test facility demonstrated exhibited well-mixed (turbulent) air distribution, operating as a single zone of movement under standard operating conditions. The introduction of transient situations (opening and closing of doors) proved successful in indicating that the isolation room can return to its initial well-mixed state when disturbed.

The protection factor was introduced as a measured degree of effectiveness in protecting staff and patients against a source of contamination.

There are two reasons for using an isolation room – the patient is a source of infection and the patient is at risk from infection. The PPVL isolation room provides a protection factor of 10^5 between the corridor outside and the isolation room (all doors closed). Conformance with air tightness standards in construction coupled with the operation of the isolation room itself at near neutral pressure differential relative to the corridor is a further safety factor.

There is a further role other than isolation. The patient has to be nursed and medical examinations and minor procedures have to be carried out. Where the patient is a source of infection, potential hazards must be reduced to provide a safe and ideally comfortable environment. The dilution ventilation strategy is based on dilution of airborne pathogens by the incoming fresh air. The validation using tracer gas has shown that normalised concentrations within the isolation room (doors closed) are within a range of 0.5 and 2.0. This expresses the typical variation in concentrations that would be expected in the working zone of nursing staff away from the immediate source of aerosol ejection. The alternative view is to regard the ventilation within the isolation rooms providing a conservative protection factor of 10^3. For the patient at risk from infection, a similar protection factor can be assigned with regard to the dilution ventilation as viewed by the patient.

It has been established that a configuration of the design presented in HBN4 [1] can be built and operated as intended, through the use of multiple established validation techniques. The PPVL concept provides a quantifiable barrier of protection for the patient from the corridor, and for the inhabitants of the corridor from the patient. The design as tested will operate correctly as an isolation room if built as an isolated system. Sanitation, mechanical ventilation and power supplies need to be independently sourced for it to function effectively for extended periods of time.

The PPVL isolation room design offers real potential to control infection in hospitals contributing to a healthy and productive environment.

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Knowledge from running of air conditioning in clean spaces

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ABSTRACT

This text contains attainments from indoor climate and air conditioning equipment monitoring in clean rooms in the public health service. The editor is designer of HVAC systems in the modern hospital near Brno in the Czech Republic. The text includes the experiences from the design phase as well as from the function the hospital. Monitoring of the HVAC systems enables during 2 years operation efficiency evaluation of heat recovery system from waste air. We are able to evaluate its economy effect. And benefit of heat recovery system depends of course of outdoor climatic conditions.

INTRODUCTION

The general requirement for energy consumption lowering in buildings includes also the requirement to bring down the energy consumption for heating. Czech technical standard for thermal protection of buildings is permanently innovated and the requirements for thermal-technical properties of buildings are strict. It is necessary to deal with lowering the heat losses by ventilation, especially in buildings where the air change rate is higher. By current standard for the envelope of the building the heat loss by ventilation is higher than the heat loss caused by transmission already for air change equal to one. That is the reason to intensify dealing with systems for energetic demand for ventilation lowering – systems for heat recovery. Other possibilities that follow are related to the operation of the equipment – the choice of the optimal temperature of the inlet air with respect to the required indoor temperature, taking the advantage of the operating hours of the equipment by using damping mode with lowered air supply. If it is not possible to set the equipment away in non-operating hours due to hygienic reasons, it is advisable to lower the air temperature (similarly as when heating) at damping mode.

The eldest and simplest method of heat recovery of air is mixing when a part of the airflow circulates and another part is mixed with the outside air. This is not a fortunate solution, when ventilation is concerned, especially for large systems operating a greater number of rooms. The air circulation spreads the noxious agents from a single room into the other areas of the building via air conditioning. This is evident e.g. while air-borne spread of cold disease. Therefore the mixing of air of ventilation and air conditioning systems should be nowadays replaced by hygienically more favorable methods.

EXPERIMENT AND METHODOLOGY

Most frequently used exchangers for recycling of heat from outlet air are plate heat exchangers in counter-flow or cross-flow connection. These particular heat exchangers in air conditioning units that operate clean spaces were subjected to detailed observation while running in reality. A special attention was devoted to these effects:

- Efficiency of heat recovery from outlet air. Based on measurements of necessary variables by Measurement and Regulation system the efficiency of heat recovery was calculated and heat savings were evaluated.
- Heat loss by transportation of air by piping and in air conditioning unit.

Individual fields were elaborated experimentally based on measured variables provided by Measurement and Regulation system of the object, some variables were derived and calculated. The system measures and saves the values in the interval of 6 minutes.

HEATING OF THE AIR IN THE RECUPERATOR AND HEAT SAVINGS

Air conditioning equipment from two operating rooms was chosen for monitoring. The airflow of the inlet air is 6200 m^3/h; the airflow of the outlet air is 5750 m^3/h.
The building control system permanently measures and saves the temperature of the outer (suction) air, inlet and outlet air and waste air behind recuperator. The humidity of outlet air is measured as well. The variables are indicated on fig. 1. The air temperature behind the heat exchanger on the inlet airside was calculated from equality of heat flows of inlet and outlet air:

$$V_p\left(t_e^{/}-t_e\right)=V_o\left(t_i-t_i^{/}\right) \qquad (1)$$

Airflows are determined from measurement of speed in piping made by testcrosses. Based on this measurement the speed of the fan is adjusted in order to reach constant airflow by means of conversion transducer.
Because of this the airflow in piping is constant without the influence of pressure relations in piping network. The measurement of speed is made on the treated inlet airside and the outlet air leaving the room before the AC unit side. The temperature of inlet and outlet air is similar; the difference is not bigger than 5 K. Hence the influence of different thermal capacities of airflows and their densities can be neglected.

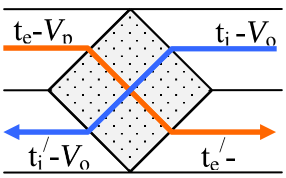

Fig. 1 Observed variables of the plate

An individual problem is the condensation of water vapor on the outlet airside while observing the plate exchangers efficiency. No moistening of inlet air was applied in monitored period; therefore the humidity of outlet air was low (under 30%, commonly about 25%, consequently dew point lower than 5 °C). Because of this low humidity and assumed efficiency of heat recycling up to 60% it was possible to exclude the condensation of water vapor on the exchanger.

The graph representing the temperature relationships while cooling of the waste air on the plate exchanger is shown on fig. 3. We can also see the outlet air temperature variation corresponding to room temperature variation. In operation period the room temperature is 24,5 °C, otherwise 19,5 °C. In regard to high air change rate (23x per hour) the temperature rise and drop is very fast, temperature difference of 6K in operation field is balanced in 15 minutes. Lowering of temperature in non-operating hours is another advisable saving measure. However it is necessary to synchronize the temperature modes of ventilation (air conditioning) system and heating system. From measurement it is obvious that if the heating elements are not adjusted accordingly, the heating system automatically increases its power to compensate the loss by ventilation. Thus the temperature of outlet air remains the same in operating and non-operating hours.
It is possible to calculate the heat savings based on measured and calculated variables in the formula 2:

$$Q = \int_{\tau} V_p \rho c \left(t_e^{/} - t_e\right) d\tau = \int_{\tau} V_o \rho c \left(t_i - t_i^{/}\right) d\tau \qquad (2)$$

At the average outer temperature is 3,0 °C the thermal energy savings of heat recovery system by 50%.
It is necessary to mention another factor. When using the heat recovery system, the Building control system should contain control of the plate exchanger bypass and provision of the antifreeze safety. But if the heat recovery system substitutes air mixing, the mentioned modifications of MaR are equivalent to the control of mixing valve.
Thermal efficiency of the exchanger is defined by the formula [3]:

$$\eta = \frac{V_o \left(h_i - h_i^{/}\right)}{V_o . h_i - V_e . h_e} = \frac{V_e \left(h_e^{/} - h_e^{/}\right)}{V_o . h_i - V_e . h_e} \qquad (3)$$

Above the assumption makes simplified formula:

$$\eta = \frac{t_i - t_i^{/}}{t_i - t_e} \qquad (4)$$

In monitored period the outer temperature varied from -1 °C to +5 °C. The average efficiency of the exchanger was 50%. This low efficiency is given by low humidity of outlet air (to 30%), so that there is no condensation of water vapor by cooling of outlet air. For such conditions the manufacturer states the efficiency to be 52%. It is necessary to respect the real efficiency while thinking about the importance heat recovery equipment saves 1,18 GJ of thermal energy per day. If the thermal energy is acquired from a central heat supply system the savings per day can be enumerated to 13 Euro. We assume this day as a representative day for whole heating period. For a total number of 195 days would the year savings of one heat recovery unit be 2 555 Czk. The length of the heating period is derived from average monthly temperatures according in Czech Republic.
The pressure loss of the chamber of the plate exchanger in optimal operating conditions does not exceed 200 Pa, in monitored case it is 120 Pa. Total pressure of a supply fan is 1 200 Pa, which means at given airflow and efficiency of the fan the power of 2,79 kW. At the decrease of supply pressure by 120 Pa the required power drops to 2,48 kW. The real consumption of energy for the fan driving is determined according to the engine power assigned to the fan. The lowering of total pressure of the fan would mean in this case the lowering of required

engine power from 4,0 to 3,5 kW. The equipment runs 24 hours per day, 10 hours of it to 100% and 14 hours to 50% air power. In this operating mode is the difference of electrical energy consumption 7,8 kWh/day for supply fan and 2,3 kWh/day for outlet fan. The difference of electrical energy consumption makes 2 890 kWh per year. The operation of the fans in the AC unit without the plate exchanger would be cheaper by 286 Euro, considering market price of electrical energy. The increase of supply pressure of fans lowers of heat recovery. In despite of relatively low efficiency the thermal energy savings mentioned above are significant.

HEAT LOSS BY PIPING AND IN AIR CONDITIONING UNIT

If we look at possibilities for savings of thermal energy for air conditioning, it is necessary to be concerned with heat losses in air conditioning distribution system. Average values of heat transmission coefficient shows table 1.

Tab. 1 Heat transmission coefficient of air conditioning units and their insulated piping

Insulation	Heat transmission coefficient (W/m^2K)
AC unit panel cover PUR panel 50 mm	0,57
AC unit panel cover PUR panel 25 mm	0,95
Steel piping with insulation from mineral wool 60 mm	0,67
Steel piping with insulation from mineral wool 40 mm	1,0

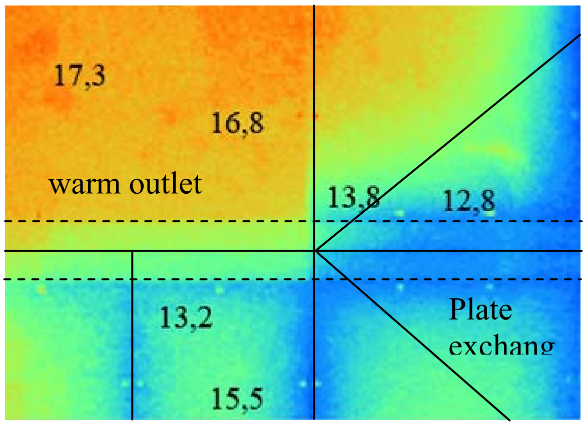

Fig. 2 Thermal field of the cover panel of AC unit with indicated chambers (temperature of sur-roundings 17 °C,

The heat loss through AC unit panel cover is neglected if the AC unit is placed in interior heated environment. The thermally treated air passes only through short part of the unit (exchangers are usually placed at the end of the formation) and the heat loss does not exceed 50 W. Fig. 2 shows thermal image of an AC unit. Even though is the total heat loss not significant, it is obvious that the temperature field of the outer cover of the panel corresponds to the supply air temperature. The AC unit is modular, made of chambers. From the thermal insulation point of view are the critical places connections of the chambers, since there are the stiffening elements. The temperature difference in the middle and in corners is 5 K by temperature difference of 19 K between the warm (waste) air and fresh (outer) air. Condensation of water vapor can occur under unfavorable thermal moisture conditions.
The piping in the machine room has thermal insulation of the thickness of 60 mm. besides heat losses and risk of condensation it is necessary to take into account the noise propagation. At the temperature difference between the air in the machine room and in the piping 10 K is the heat loss for 50 m^2 of the heat transfer surface of the piping 330 W. For average conditions (above mentioned example) this heat loss does not mean a change greater than 0,3 K in the temperature of supply air. Thermal imaging proved a good quality of the insulations, which should be nowadays standard in HVAC field.

Current design standard of thermal resistance of constructions of building envelopes ensures that even in unheated spaces of buildings the temperature approaches the temperature in surrounding rooms. In ordinary circumstances the piping does not pass though different spaces concerning temperature. If the temperature of the supply air does not differ from the temperature of the surroundings by more than 4 °C it is not necessary to insulate the piping. In running of public health services the ventilation systems usually take over the role of air conditioning and therefore the piping is insulated because of the summer temperature conditions. We can summarize that air transportation by piping is no more a source of heat losses.

RESULT

From all of the heat losses in air-conditioning equipment that was analyzed above is the most significant one a heat loss by ventilation, which can be effectively lowered by means of heat recovery system. The values presented in this article were obtained by monitoring of real air conditioning equipment. Air conditioning in health service is specific with regard to operation time and higher air change rates. These are favorable circumstances for usage of plate exchangers that ensure hygienically clean running by separation of the inlet and outlet airflows. In this case study were found annual savings of thermal energy under real climatic conditions in the CR 2 270 Euro. Considering the price of a chamber with a plate exchanger to be 3 100 Euro, the payback of this measure is very short.
Another suitable measure is consequent respecting of working hours in attended spaces. Only then it is possible to use lowered air supply and lowered temperature in damping time. Mutual correspondence between set values of air conditioning and heating is essential.

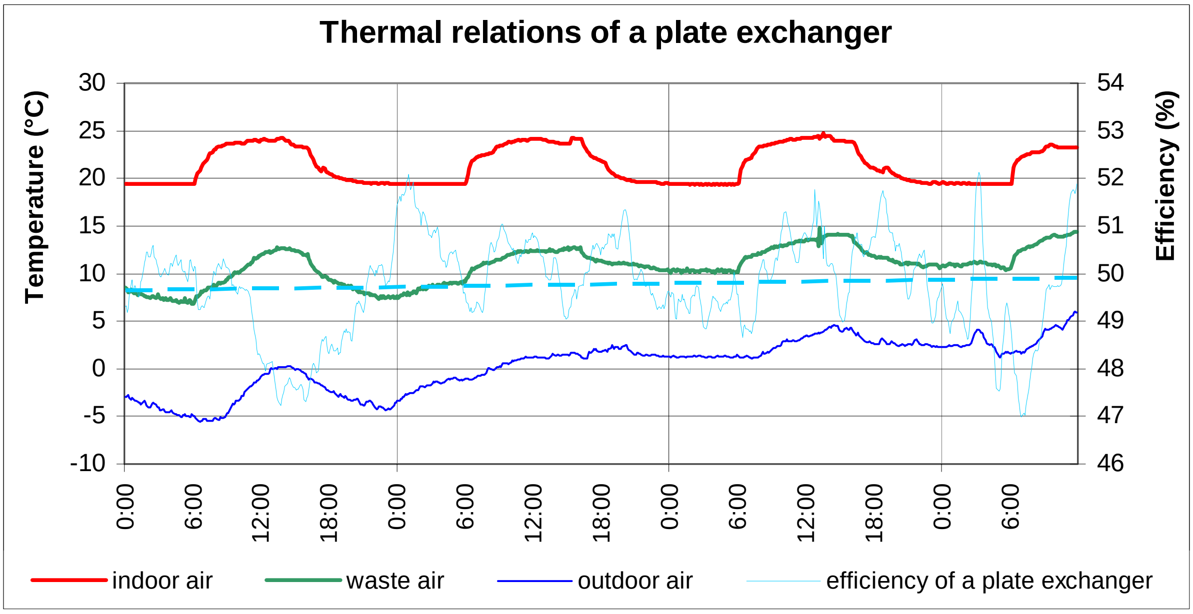

Fig.3 Thermal relations of a heat recovery exchanger in Januar 2006

DISCUSSION

Efficiency off-heat reverse obtaining can markedly be lowered by low outdoor temperature with effect antifreeze exchanger safety. By temperature fall of waste air below 5°C is outdoor air input partly or completely bypass. Surface temperature of exchanger could be in this time lower then 0°C (figure 4). Heat exchanger is for this time off the function. Temperature 5°C waste air can be reach with outdoor temperature about -7°C (figure 3). The real efficiency with low outdoor temperatures is accordance with lower regulating antifreeze safety exchanger.

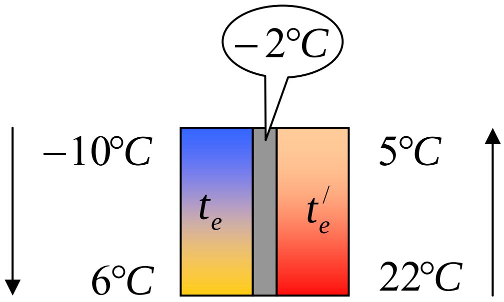

Fig.4 Air and surface temperatures of exchanger

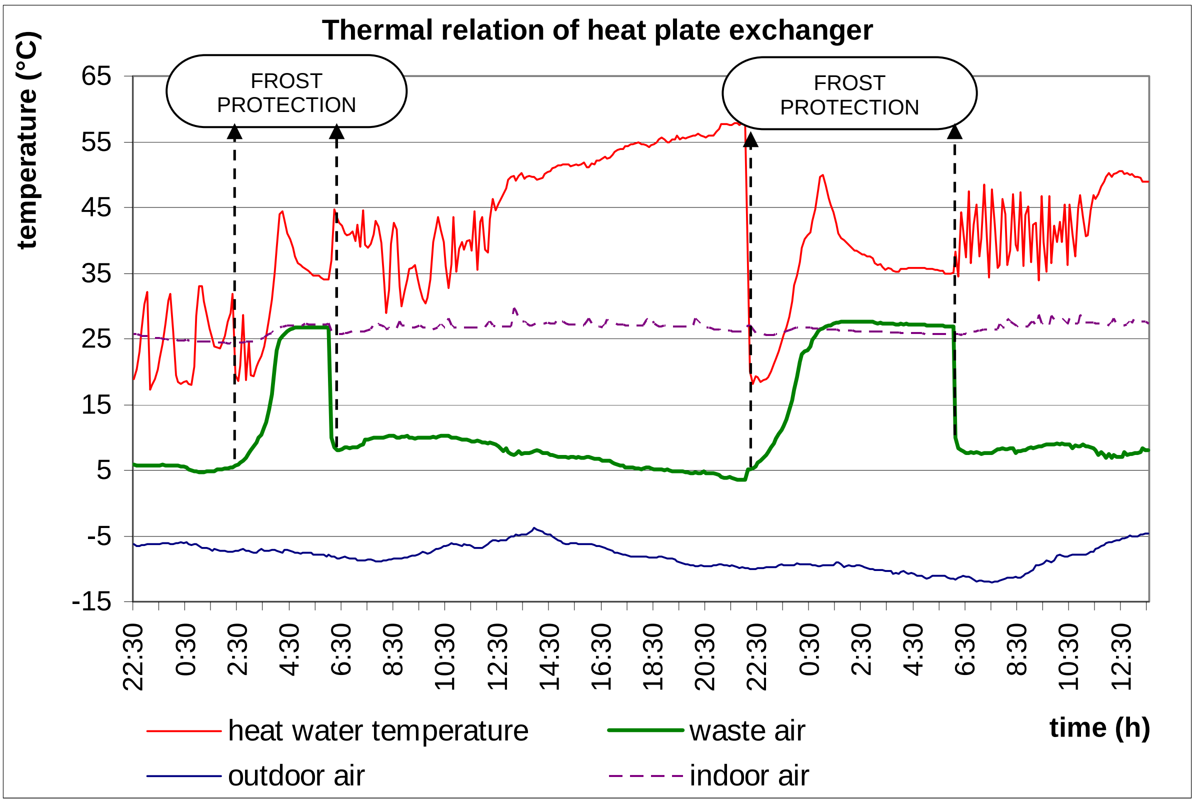

Fig. 5 Thermal relation of heat recovery exchanger by antifreeze safety

NOMENCLATURE

Variables:
V ... airflow (m^3/h)
Q ... heat power (W)
t ... temperature (°C)
h … enthalpy (kJ/kg)
τ ... time (s)

Indices:
i ... inner
e ... outer
Ψ ... efficiency (%)
o ... outlet (air)
p ... inlet (h)

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Ventilation Considerations for Indoor Environmental Quality for a Control Center

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SUMMARY

This paper presents design considerations of ventilation and indoor environmental quality (IEQ) for a control center. Raised access floors are extensively used to provide flexibilities for control cables and wires required by a control center. Underfloor air distribution system (UFAD) is more practice and economical than an overhead mixing system. It improves indoor ventilation effectiveness due to the air stratification of thermal plume, and it provides a better indoor environment for the equipment and occupied people. From the analyses and calculations for a control center, when the load density is less than 40 btuh/s.f., and the throw height of the floor panel diffuser is less than 4', the ventilation effectiveness is up to 1.2, similar to the displacement ventilation. When the load density is between 40 and 60 btuh /s.f, and the throw height of the floor panel diffuser is between 4' and 6', the stratification of thermal plume is still generated, the ventilation effectiveness is approximately 1.1. When the load density is over 60 btuh/s.f., the throw height of the floor panel diffuser is over 6'. Without the air stratified layer, the ventilation effectiveness is 1.0, equal to perfect mixing air distribution. In the high supply airflow rate situation, ventilation airflow for providing an acceptable IAQ around people should be maintained at least 10% of supply air, in order to provide at least 5 cfm fresh air for people who can obtain nearby.

INTRODUCTION

A control center differs from a data processing center because it is continuously occupied by people performing various activities whereas a data processing center is infrequently occupied by one or two persons for brief periods. Its heat load is similar to the data processing center, there are high density heat loads emitted from control devices. Then, it is more complicated than the data processing center because an acceptable indoor air quality is required for the occupants' health and thermal comfort.

High density heat loads from computer workstations and control devices result in high air flow rates in a control center. Outdoor air flow for ventilation is mixed in supply air flow, ventilation air percent of supply airflow is decreased with the increase of supply air when the required outdoor air flow rate 15 cfm per person is used. Ventilation air that people can catch in the breathing is greatly decreased. Hence, the ratio of ventilation to supply air should be hold at some proportion.

Raised access floors are built to accommodate the large number of electrical cables and wires required by the control center. Under floor space is available, it is reasonable and economical to use for an underfloor air distribution system (UFAD) for air conditioning of the control center. UFAD uses raised access floors as a conditioned supply air plenum, the conditioned air is delivered to the occupied space through diffusers installed in the floor, that can be relocated easily.

Compared with displacement ventilation, UFAD uses air buoyancy that helps to drive air up to return openings, and air stratification of thermal plumes to improve indoor air quality. Furthermore, when supply air is mixed with room air in this system, much higher cooling loads can be handled. Hence, an underfloor air distribution system is specially suited for a control center.

Figure 1 shows a sketch for an underfloor air distribution system for a control center. Floor panel diffusers throw conditioned air up to the occupied space. When the throw height is proper, the stratification of thermal plume is formed. The ventilation effectiveness and indoor air quality are improved and outdoor air flow rate may be decreased.

For this project, the area of the control center is approximately 1520 s.f., there are 8 workstations, 5 wall videos and 14 people occupancy. The load densities will be analyzed from 40 btuh/s.f. to 110 btuh/s.f..

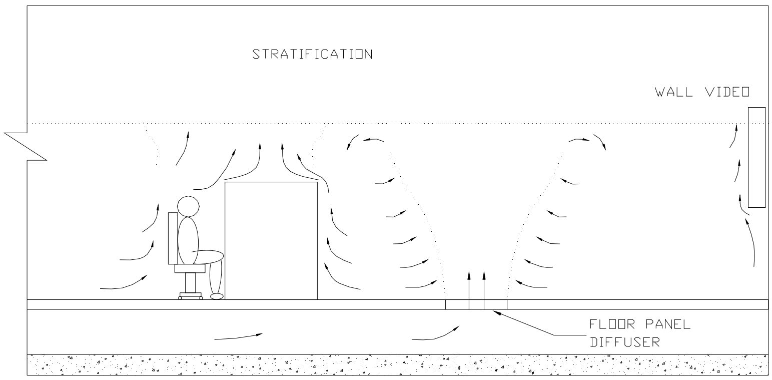

Figure 1 Sketch for Underfloor Air Distribution for a Control Center

VENTILATION CONSIDERATIONS

Outdoor air is brought to an occupied space by an air handling unit, complying with ASHRAE Standard 62. Conditioned air mixed with the acceptable outdoor air in a pressurized underfloor air plenum through floor panel diffusers is delivered to an occupied space, that includes the breathing zone around people under 6’ height and the above area. If the conditioned air with the fresh outdoor air is concentrated to send to the breathing zone, the ventilation improves the indoor air quality around people. The air velocity and temperature distributions directly affect the ventilation for improving an indoor air quality. Furthermore, ratio of ventilation air to total supply air should be kept to provide the minimum fresh air nearby people who can catch in the breathing.

AIR VELOCITY DISTRIBUTIONS

An area of a floor panel diffuser for this control center is 2’x2’ with a 25 % free area. For a load density 55 btuh/s.f., required airflow rate is 300 CFM for a floor panel diffuser at the air temperature difference 15 °F between the supply and return air. There are total 17 floor panel diffusers in this control center. The air velocity through the perforated floor panel is 300 fpm with an effective area 1 square feet. Table 1 presents the air flow rate for a floor panel diffuser and the required total pressure in the pressurized underfloor plenum at different load conditions.

Table 1 Air flow Rate at Perforated Discharge of a Floor Panel Diffuser with a 25 % Free Area at Different Load Densities and Total Pressure Required in the Pressurized Plenum for the Control Center

Load Density btuh/s.f.	40	55	75	90	110
Airflow Rate per Floor Panel Diffuser, cfm	230	300	400	500	600
Velocity at Perforated Discharge, fpm	230	300	400	500	600
Required Total Pressure at the Pressurized Underfloor Plenum, W.G.	0.04	0.07	0.13	0.20	0.29

By calculating the pressure losses through the perforated panels plus velocity pressure, the total pressure 0.072" w.g. in the pressurized underfloor plenum is required to deliver the air flow rate 300 CFM to the occupied space. When the airflow rate increases, the total pressure of the supply air plenum is required to increase according to the power of the flow rate. If the air flow rate is over 600 cfm, the floor panel diffuser with a 25 % free area requires the total pressure over 0.29 W.G. to carry the conditioned air to the space and results in high energy consumption of fan power and noise issues. At the situation, a floor panel diffuser with a higher free area is required to decrease the pressure losses through the diffuser.

When air is discharged from perforated panels, the constant velocity core formed by coalescence of the individual jets extends a distance from the panel face. For the panel diffuser throw the air to the space, the air stream velocity above the floor panels is expressed at the equation (1).

$$V_x = 1.2V_0\sqrt{C_d R_{fa}} \tag{1}$$

where:
V_x is an average initial velocity at the perforated discharge.
C_d is a discharge coefficient.
R_{fa} is a ratio of free area to gross area.

The centerline velocity of a floor panel diffuser in jet expansions zone 2 and zone 3 are expressed in equations (2) and (3) below.

$$Vx=V_0\,(1.13KH_0\,/X)^{1/2} \tag{2}$$

$$Vx=V_0\,(1.13KA_0^{1/2}/X) \tag{3}$$

Where:
A_0 is an effective area of stream, ft^2.
H_0 is an effective width of jet at outlet, ft.
K is a centerline velocity constant.
V_0 is an average initial velocity at the outlet, fpm.
V_x is a centerline velocity at distance X from outlet, fpm.
Figure 2 shows the air velocity distributions at the centerline of jet expansion at the room height when the floor panel diffuser delivers the airflow rates at the range of 230 CFM to 600

CFM, the load densities have a range of the 40 btuh/s.f. to 110 btuh/s.f.. After the air velocity at the centerline reaches 50 fpm, the velocity and temperature are decreased rapidly. The throw height is obtained at the air velocity 50fpm. If the air throw height is under a breathing zone height, the stratification of thermal plume is generated over the breathing zone. Ventilation air originated from the outdoor is concentrated and distributed in the breathing zone of the occupied area, it is not distributed in the un-breathing zone in the stratified area. The ventilation effectiveness in breathing zone is improved.

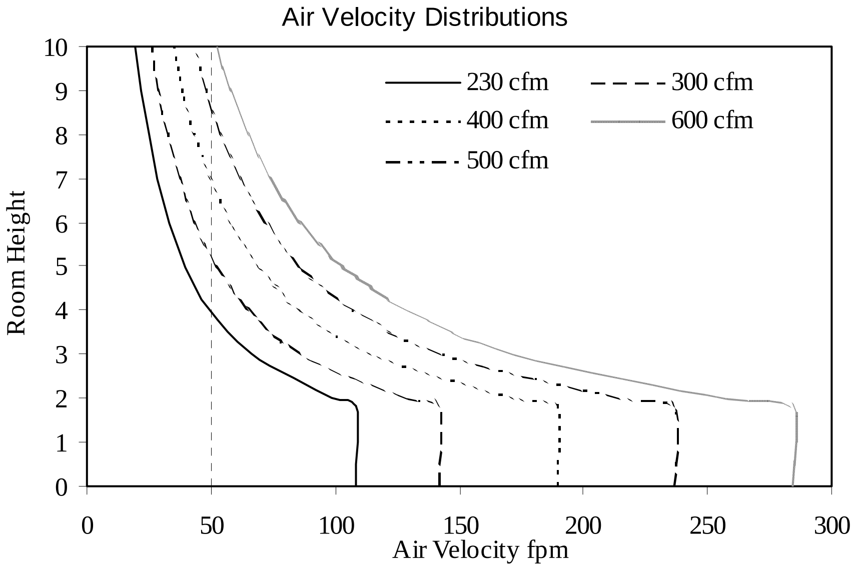

Figure 2 Air Velocity Distributions at the Room Height for a Floor Panel Diffuser.

For the air flow rate 300 CFM and load density 55 btuh/s.f., the throw height is approximately 5 feet, while the throw height is close to 9 feet at the air flow rate 500 CFM and the load density approximately 90 btuh/s.f. When the throw height is 5 feet, the stratification of the thermal plume may be formed above 6 feet. Furthermore, if the load density is under 40 btuh/sf and the air flow rate is under 230 fpm, the throw height is under 4 feet, that is height of main equipment and control devices. When the stratification of thermal plume is generated, the ventilation for this situation is similar to the displacement ventilation. When the load density is over 90 btuh/s.f., the throw height is over 8’, the stratified layer of thermal plume disappears because the air velocity is turbulent in the area. Although the stratification disappears, the room air is fully mixed with the conditioned air from the underfloor plenum. This case is equivalent to an overhead air mixing system, the ventilation air is uniformly distributed to the whole control center space.

In the stratified case, the ventilation air is distributed at the breathing zone under the stratification. It increases the ratio of ventilation air to the room air at the breathing zone, if the stratification of thermal plume is formed at 6 feet height (it is above the breathing zone). When higher the stratification is formed, less the ratio of ventilation air to the room air at the breathing zone. The ventilation air is decreased until reaching the ventilation effectiveness of uniformly mixing ventilation.

AIR TEMPERATURE DISTRIBUTIONS

An air temperature distribution is similar to an air velocity distribution for the underfloor air delivery. The temperature difference at the distance X is presented in equation (4).

$$\Delta Tx=\Delta T_0\,(0.73V_x/V_0) \qquad (4)$$

ΔT_0 presents a temperature difference between an initial temperature and ambient temperature, V_x and V_0 present velocity at an initial and velocity at a distance X, respectively. An air temperature changes with an air throw distance. The air temperature difference with the ambient air at the throw height for the control center is stated in Table 2. When the load density is 55 btuh/s.f. or under, the temperature difference is 3.7 °F or over at the throw height. At that height, the air density is higher than the ambient air density. The air starts to move down, and is drawn by heat source, and makes the space air movement. If the ventilation is brought around people, the ventilation effectiveness is over 1.0. When the air temperature difference is approximately 1.5 F at the throw height 8.5' for the load density 90 btuh/s.f., the air is fully mixed with the room temperature, including the breathing zone and zone over that in the room. The ventilation effectiveness equals to 1.0.

Table 2 Air Temperature Differences at the Throw Heights for the Floor Panel Diffuser.

	Velocity at Perforated Panel Discharge, fpm				
	230	300	400	500	600
Throw Height, ft	4	5	7	8.5	10
Temperature Difference,°F	3.3	2.7	1.9	1.5	1.3

VENTILATION EFFECTIVENESS

Ventilation effectiveness describes the air distribution system's ability to remove internally generated pollutants from an occupied space, it is defined by the fraction of the outdoor air delivered to the space that reaches the occupied zone. When the ventilation air is perfectly mixed with the air in a space, the ventilation effectiveness reaches 1.0. Furthermore, the ventilation effectiveness can be greater than that which would be realized with perfect mixing. For displacement ventilation, the ventilation effectiveness is 1.2. For an UFAD system, the ventilation effectiveness should be over perfectly mixing ventilation and under displacement ventilation.

Table 3 states the ventilation effectiveness for this control center at different load conditions. When the loads density is under 40 btuh/s.f., and the thrown height of design panel diffuser is lower than 4', the ventilation effectiveness is approximately 1.2, similarly displacement ventilation. When the load density is between 40 btuh/s.f. and 60 btuh/s.f., the ventilation effectiveness would be over 1.0 and under 1.2. The stratification of thermal plume may be formed. When the load density is over 60 btuh/s.f. in this case, the ventilation effectiveness is 1.0, that equals to the perfect mixing system.

Table 3 Ventilation Effectiveness at Different Load Conditions for the Control Center.

	Ventilation Effectiveness
Ceiling Supply, Perfect Mixing	1.0
Displacement Ventilation	1.2
Floor Supply, Throw Height < 4.0'. Load Density < 40 btuh/s.f.	1.2
Floor Supply, Throw Height > 4' and < 6.0'. Load Density > 40 btuh/s/f. and <60 btuh/s.f.	1.1
Floor Supply, Throw Height > 6'. Load Density > 60 btuh/s.f.	1.0

THE MINIMUM VENTILATION AIRFLOW RATE

High density loads in a control center result in high density supply airflow rates, and low ventilation air percent of total supply air. If the minimum outdoor airflow per person 5 cfm is required in the breathing area, when heat release of people seating work is sensible 280 btuh and latent 270 btuh, then the approximately air flow rate 50 cfm is required to carry the heat out for a temperature rise 5 F. Its outdoor air percent of supply airflow is approximately 10 %. When airflow rate is increased in HVAC system, the ratio of outdoor air to supply air is decreased if outdoor airflow rate 15 cfm per person is used. For a 55 btuh/s.f. heat load, the percent of supply air is approximately 4% around people. According to air flow rate 50 cfm passing through person for heat release, fresh air that person can catch is 2 cfm. Outdoor air around people breathing area is low to result in IAQ problems. For this case, the minimum outdoor air percent of supply air is maintained at 10%, similar to the above normal situation that 5cfm ventilation airflow rate can be obtained per person, then the air flow for ventilation is greatly increased.

Table 3 shows the outdoor airflow rates in various load conditions. Outdoor airflow rate in a control center should be higher than required 15 cfm per person even the ventilation effectiveness is considered.

Table 3 Outdoor Air Flow Rate for the Control Center when Ventilation Airflow is 10% of Supply Airflow.

Load Density	Required Ventilation Airflow by ASHRAE 62	Supply Airflow Rate	Ventilation Airflow Rate
btuh/s.f.	cfm	cfm	cfm
40	175	3900	325
55	190	5200	475
90	210	8500	850

CONCLUSION

The stratification of thermal plume improves indoor air quality and ventilation effectiveness. The ventilation effectiveness in this project is obtained by analysis and calculations of air velocity and temperature distributions. In conclusion, when the load density is less than 40 btuh/s.f., and the throw height of the floor panel diffuser is less than 4’, that is control equipment and devices’ heights. The ventilation effectiveness is up to 1.2, it is similar to the displacement ventilation. When the load density is between 40 and 60 btuh /s.f, and the throw height of the floor panel diffuser is between 4’ and 6’, the stratification of thermal plume is still generated, the ventilation effectiveness is approximately 1.1. When the load density is over 60 btuh/s.f., the throw height of the floor panel diffuser is over 6’ (breathing zone), the ventilation effectiveness is 1.0, it equals to perfect mixing air distribution. For high supply airflow rates of a control center, the ventilation air should be maintained at least 10% of supply air for providing an acceptable indoor air quality around people who can obtain the minimum 5 cfm fresh air in the breathing.

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2. ASHRAE Standard 62.1, 2001.
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Improving exhaust rates in machine tools based on flow simulations

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ABSTRACT

Three dimensional simulation of flow and material dispersion processes inside the enclosures of machine tools opens up perspectives of assessment not by far achievable experimentally. Visualisations of these processes in terms of velocity and directions, temperature and pressure allow for an in-depth understanding of the processes and their consequences in view of a reduction of emitted metal-working fluid aerosols. This is demonstrated by one example from the machine tool laboratory (Werkzeugmaschinenlabor, WZL) at the University of Aachen (Rheinisch-Westfälische Technische Hochschule, RWTH).

PROBLEM DESCRIPTION

The cooling lubricants (CL) on machine tools are flowing into the production hall and the workplace in the following ways:

- via residual concentration in the so-called clean air,
- via the heated or hot swarf and work pieces wetted by cooling lubricants (CL),
- via holes and leaks at the spin-off area of rotating pieces by exerting local overpressure,
- when opening the door at the end of a processing cycle.

To avoid the release of the cooling lubricants via holes and leaks it must be guaranteed that a negative pressure exists at all these holes or leaks. This is one of the main function of an exhaust air system which is used for such machine tools (typical air changes are 300 h^{-1} and more).

Additional functions of the exhaust air system consist as well in limiting the cooling lubricants concentration for reason of explosion protection as in removing thermal loads. The latter case is only of importance to the minimal quality of lubricating system and to the dry processing because of the fact that the essential thermal loads are to be absorbed by flooded cooling/lubricating.

One of the difficulty is to respect the dependency of the separators efficiencies on the air flow. With an increased air flow the efficiencies are principally decreasing and that’s why the cooling lubricants components in the clean air are as well increasing.

In the majority of cases, it’s easy to avoid the direct spin of swarf and/or cooling lubricants particles. If the exhaust air openings are not clearly arranged outside of the spin area, they must be equipped with a closing device whose diameter is greater than that of the exhaust air openings.

To obviate the access of cooling lubricants particles to the separator, it's necessary that so-called splash water separators or pre-separators are directly installed at the exhaust air openings at the enclosure.

The releasing of the cooling lubricants particles through the swarf can be evaded by integrating the swarf removal into the exhaust air system. This will be demonstrated later.

ANALYSIS OF FLOW PATTERN

Figure 1 shows one of the engine lathes in the machine tool laboratory at the University of Aachen (RWTH), which were tested exemplarily within the scope of a research project promoted by the association of commercial and industrial workers' compensation insurance carriers (HVBG). Figure 2 pictures a machine for flow simulation with the essential fixtures, holes and leaks.

Fig. 1: Engine lathe at the machine tool laboratory(WLZ) at the University of Aachen (RWTH)

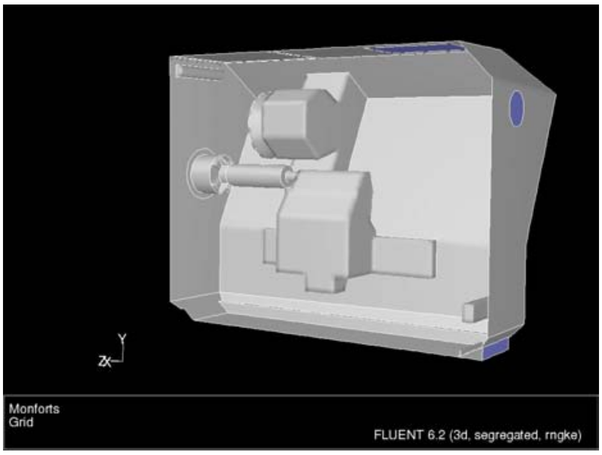


Fig. 2: Engine lathe model for flow simulation.

In the following figures 3 – 8 the conducting path lines and the appropriate pressure ratios inside the enclosure concerning as well the exhaust air flow of 650 m³/h as the maximum spindle speed of the machine of 3,000 rpm are presented.

The ratios for the following cases are shown:

- the exhaust air system is working only (no rotating pieces)
- the spindle etc. is rotating only (without exhaust air flow) and
- the combination of both processes corresponding with the real operation

Fig. 3 shows the conducting path lines which are starting from the opening for the swarf removal. In case that only the exhaust air system is working, half of the after flowing air is going through this hole into the enclosure. The air flow locates itself on the ground and streams to the left by conducting then from the side wall upwards and from the ceiling to the right side of the exhaust air opening.

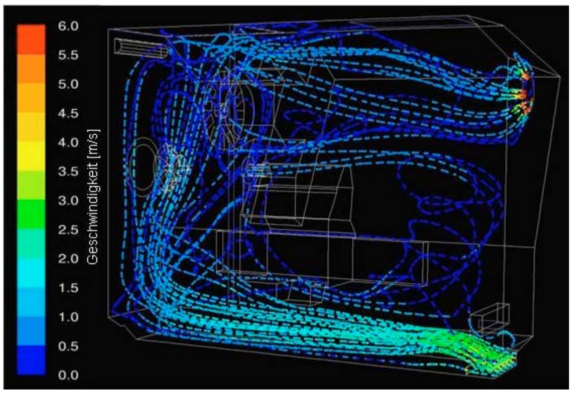


Fig. 3: Conducting path lines intra-machine during on-line operation of the exhaust air system.

The appropriate pressure distribution inside of the exemplified tested engine lathe is shown in fig. 4. On average, a negative pressure of 7 Pa is exerting on the ambience. It can be recognised that the negative pressure at the redirection places in the bottom left corner and top corner is weaker because of the dynamic pressure.

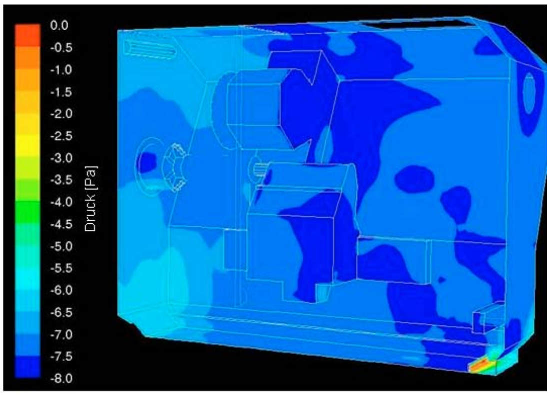


Fig. 4: Pressure distribution intra-machine during on-line operation of the exhaust air system.

During off-line operation of the exhaust air system the gripping jaws are working like a propeller which is radially displacing the air. The after flow happens axially over the fixed work piece. Starting from the jaw chuck two opposed air rolls are developing.

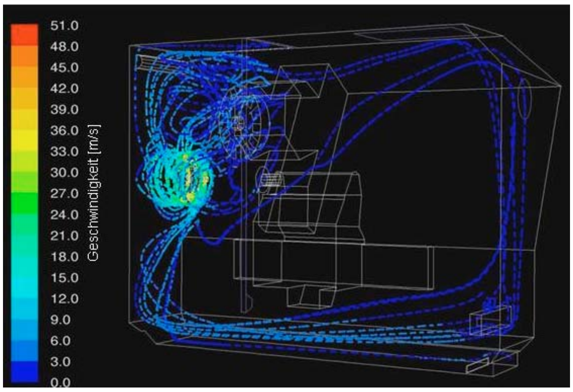


Fig. 5: Conducting path lines intra-machine during off-line operation of the exhaust air system and a spindle speed of 3,000 rpm.

- In the lock joints zone of the slide door, which is lockable from right to left, an over-pressure is occurring (see fig. 6). The air loaded by cooling lubricants (CL) can escape at these places.

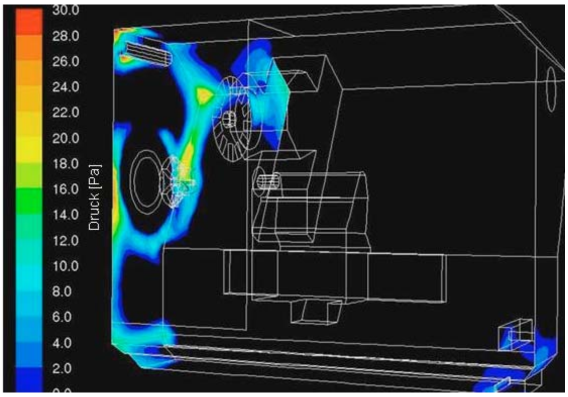


Fig. 6: pressure distribution intra-machine during off-line operation of the exhaust air system and a spindle speed of 3,000 rpm.

By combining both processes which are appropriate to simulate the real operation, clear air rolls can’t be more recognised (see fig. 7). The flow covers in that case the whole enclosure.

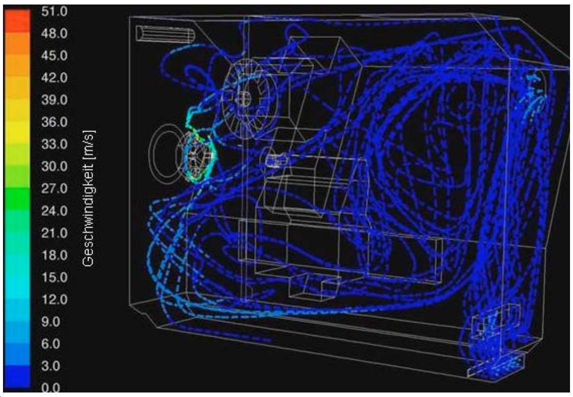


Fig. 7: conducting path lines intra-machine during on-line operation of the exhaust air system and a spindle speed of 3,000 rpm.

The negative pressure generated by the exhaust air ventilator is not enough to avoid the flow of the air loaded by cooling lubricants (CL) at the door joints (see fig. 8).

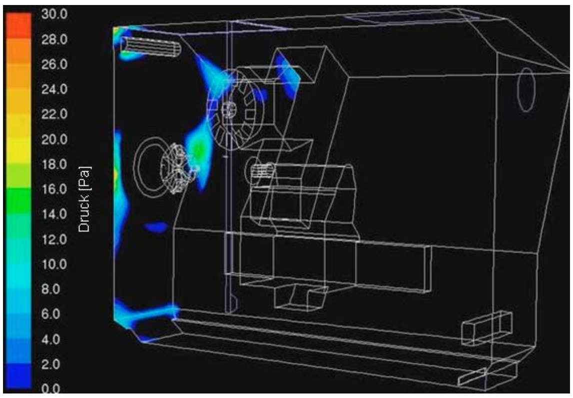


Fig. 8: pressure distribution intra-machine during on-line operation of the exhaust air system and a spindle speed of 3,000 rpm.

METHOD OF CALCULATING THE MINIMUM AIR FLOWS

By comparing the pressure ratios of spindle and work piece rotations without exhaust air applied with the appropriate pressure ratios during the operation of the exhaust air system the correlation between the static pressure at the potential places of leaks inside the enclosure and the air flow can be deduced. The diameter of the exhaust air opening can be practically ignored. Fig. 9 illustrates this correlation.

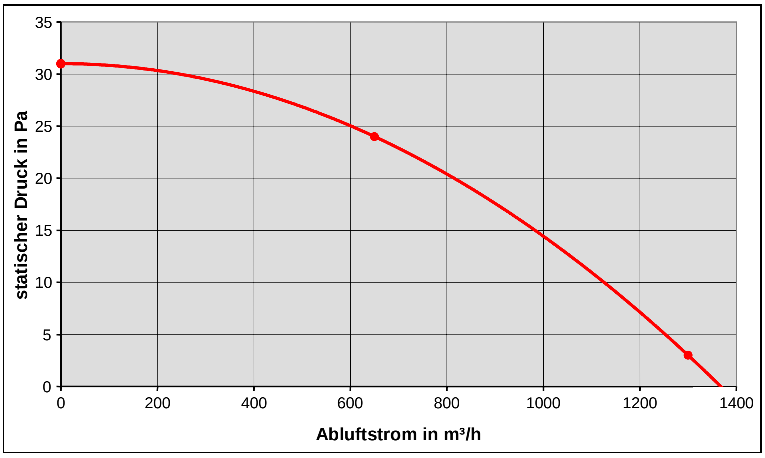


Fig. 9: Graphical determination of the minimum air flow for an engine lathe with a spindle speed of 3,000 rpm.

For determining the behaviour, two points are enough because the dependence between the 2 quantities is quadratic. During the operation with a maximum rotational speed of 3,000 rpm the machine reaches a static pressure of 0 Pascal at the critical points when the air flow comes to 1,350 m³/h.

Fig. 10 illustrates the correlation between minimum air flow and spindle speed. If there exists no or minor thermal energy inside the enclosure and if there are no pieces rotating, an air flow is not necessary, so that the zero point is given. Because of the linear correlation of the both variables another point is sufficient for the determination of a diagram.

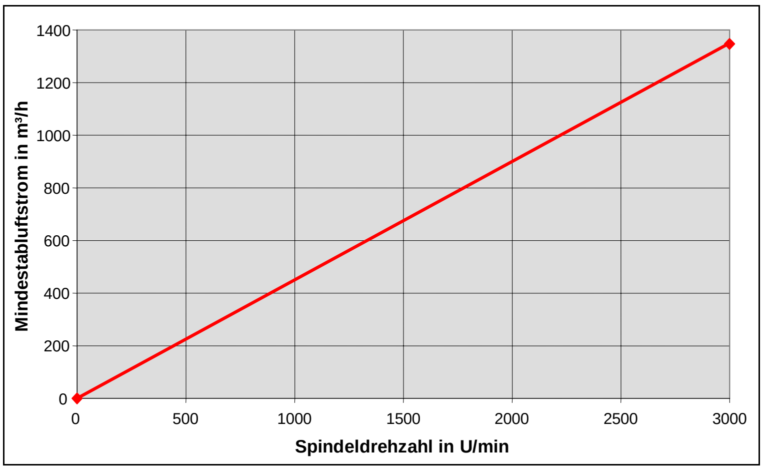


Fig. 10: Dependence of the minimum air flow on the spindle speed.

During the tests on the machine in the laboratory WZL carried out by the Institute for occupational safety and health of the employer's liability insurance association (BGIA) the maximum speed during the smoothing came up to 950 rpm. The consequence arising out of the fig. 10 is the fact that under these conditions a minimum air flow should be necessary.

If the air flow mustn't be hold in a higher position for other reasons (like limiting the cooling lubricants concentration for reason of explosion protection or for reason of the necessary removing of thermal loads), it is possible to connect the air flow to the spindle speed and to influence favourably the separator degree at the separator.

AIR FLOW INFLUENCE ON THE DISTRIBUTION OF THE CL-CONCENTRATION INSIDE THE ENCLOSURE

The following figures 11 - 14 demonstrate the distribution of the cooling lubricants concentration in a longitudinal level of the enclosure for the reference case and 3 versions. The scaling of the cooling lubricants concentrations is standardised on the average concentration in the exhaust air. Therefore, the read zones show which local concentrations are three times higher than that in the exhaust air.

In the reference case (see fig. 11) the smoothing procedure takes places with a speed of 950 rpm. This corresponds to a cutting speed of 400 m/s which is based on the actual workpiece diameter. The exhaust air (650 m³/h) dissipates via the provided exhaust air opening (right on the top). The distribution is uneven and the average concentration inside the enclosure is high. The concentration is at a ratio of 1,4 : 1 of that in the exhaust air.

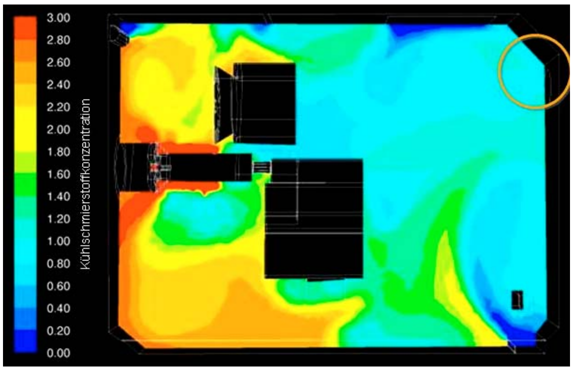


Fig. 11:Distribution of the cooling lubricants concentration in the reference case.

At variance with the given reference case, the exhaust air opening is closed in version I (see fig. 12). The exhaust air system is closed at the opening for the automatic swarf removal. The zones with high concentrations are smaller than in the reference case so that the average concentration inside the enclosure is considerable lower. The concentration is at a ratio of 0,9 : 1 of that in the exhaust air.

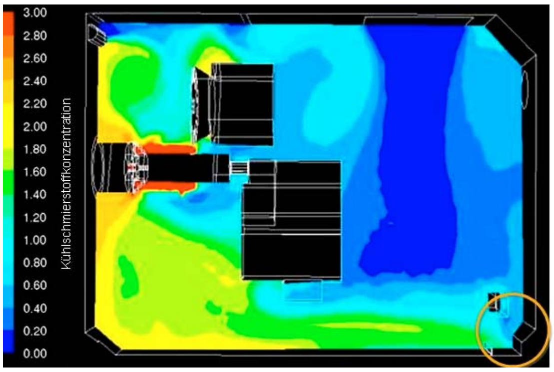


Fig. 12:Distribution of the cooling lubricants concentration for the version I.

In version II the door joints on the top are provided as exhaust air openings (see fig. 13). Compared with the version 1 the ratios are once more ameliorated. The average concentration inside the enclosure is at a ratio of 0,4 : 1 of that in the exhaust air.

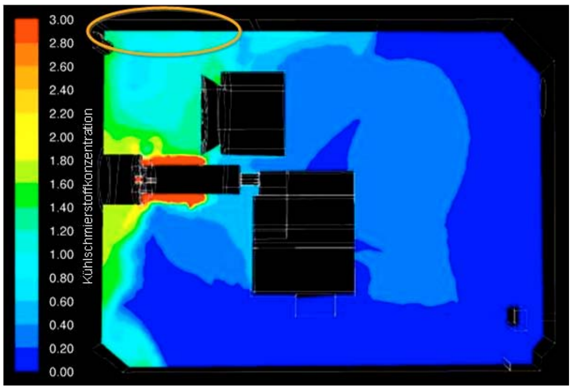


Fig. 13: Distribution of the cooling lubricants concentration for the version II.

In version III the air in the opening zone for the swarf removal is supplied in such a way that a really hydraulic short-circuit occurs up to the exhaust air opening (see fig. 14). According to this the average concentration inside the enclosure is at a ratio of 1,5 : 1 of that in the exhaust air.

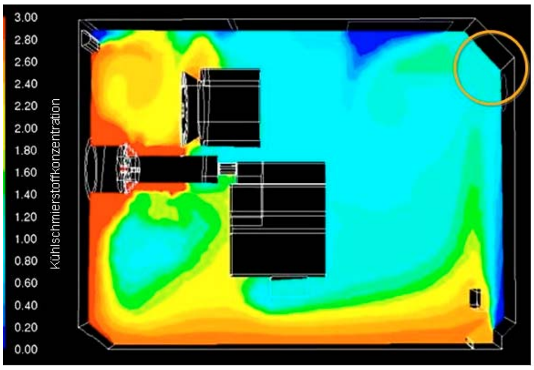


Fig. 14: Distribution of the cooling lubricants concentration for the version III.

The cooling lubricants concentration constitutes no suitable criterion for the arrangement of the exhaust air openings inside of the enclosure. In conjunction with the circulation of the supply air the arrangement has a considerable influence on the distribution and on the average concentration inside of the enclosure. This fact is important with regard to reasons of explosion protection and to the starting conditions at the moment of opening the door.

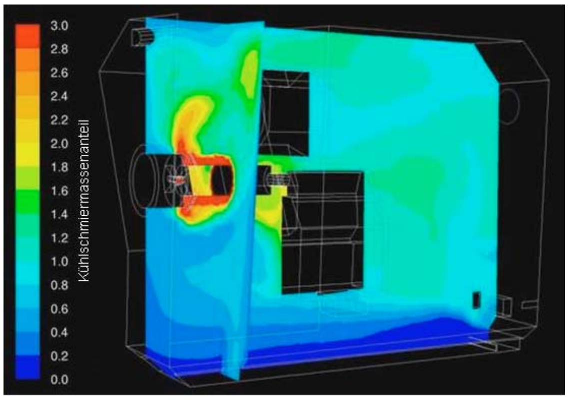
TRANSIENT PROCEDURES DURING THE OPENING OF THE ENCLOSURE'S DOOR AT THE END OF THE MACHINING

The procedures during the opening of the door are individually very differing and depend on the machine's geometry as well as on the following criterions:

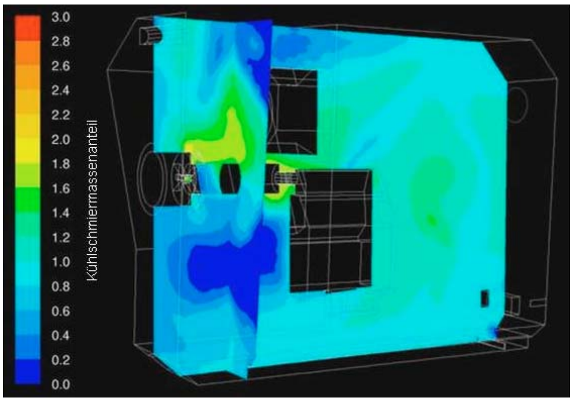
- velocity and direction of air streams just before opening the door; this momentum drop down fast but have an impact on the procedure anyway;
- thermal air streams along more or less hot tools and work pieces;
- air flow and air routing when the door is open; both can be different from the steady state phase of machining.

Fig. 26 demonstrates the time-dependant CL-concentration in the enclosure for two orthogonal section planes for the phase of opened door at the end of the machining. The four figures show the distribution of CL-concentration immediately after the end of the machining and one, four and ten seconds after that moment. The figures give an impression of how the CL loaded air gets exhausted from the enclosure.

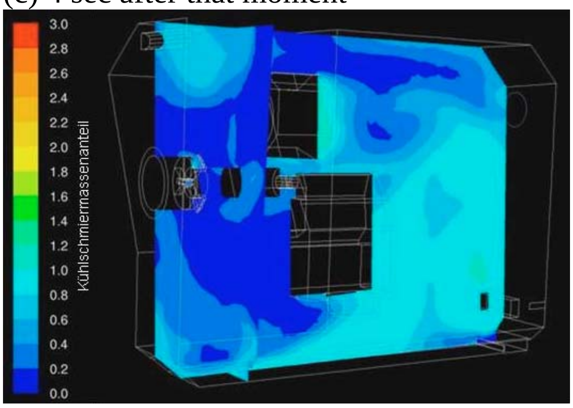
(a) immediately after opening of the door



(b) 1 sec after that moment



(c) 4 sec after that moment



(d) ~10 sec after that moment

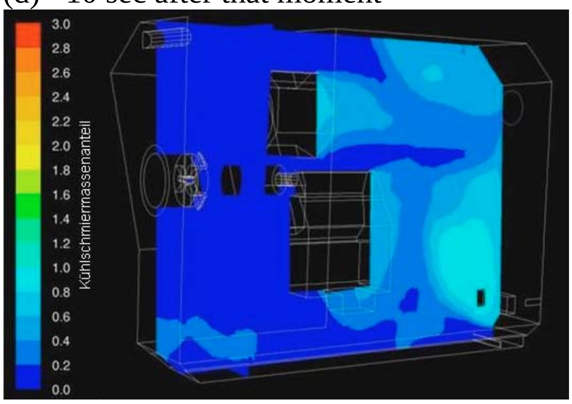
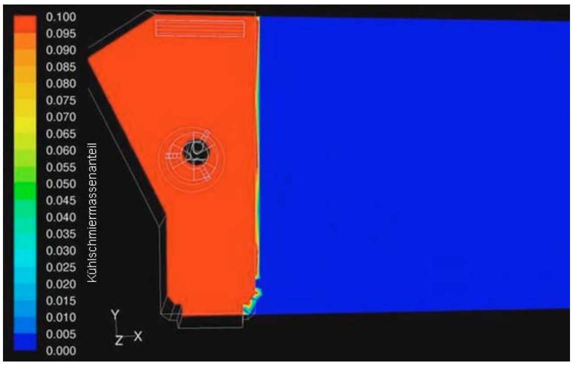


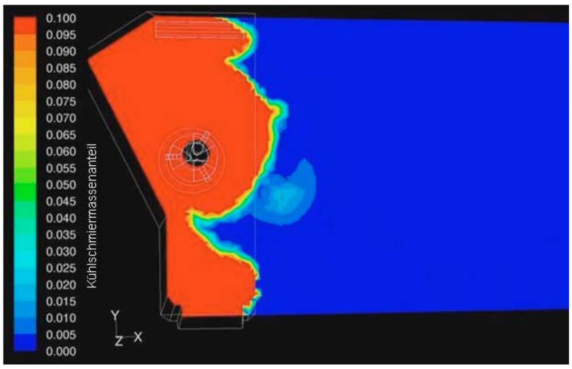
Fig. 26: time-dependant distribution of CL-concentration in the enclosure after opening of the door at the end of the machining.

Fig. 27 for the same moments as fig. 26 shows the distribution of CL-concentration in the enclosure and in the worker's area. The section plane is placed in the middle of the doorway. The scaling is different from the one in fig. 26; the maximum value shown is 0.1 of the CL-concentration in the exhaust air. The figure demonstrates clearly how CL loaded air streams into the worker's area.

(a) immediately after opening of the door



(b) 1 sec after that moment



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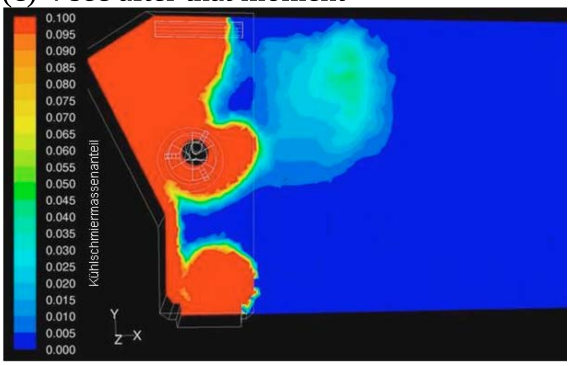

(d) ~ 10 sec after that moment

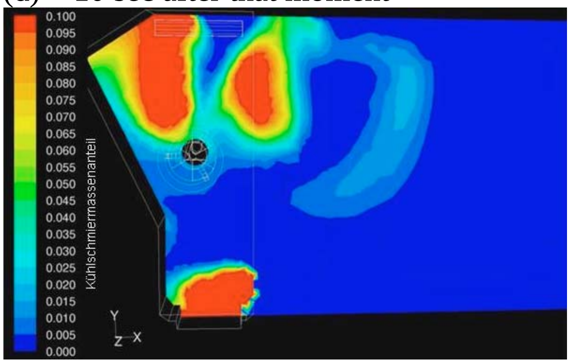

Fig. 27: time-dependant distribution of CL-concentration in the enclosure and in the worker's area after opening of the door at the end of the machining.

SUMMARY

- Rotating parts have an important impact on the pressure distribution. The exhaust air flow can be varied depending on their rotation speed.
- The concentration of CL in the exhaust air is no valid criteria for the layout of the air discharge opening.
- The air routing has a strong impact on level and distribution of CL-concentration inside of the enclosure. Then this concentration is important regarding explosion prevention and concerning the phase of opening the enclosure's door.
- Due to short response time the CL-concentration gets stationary very fast which is important for the weighing of the emissions during machining on the one hand and during opening of the door on the other hand.

Computational Study of Contaminant Control by Multi-slotted Hoods in an Industrial Exhaust System

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SUMMARY

The subject of this paper is the numerical simulation of the capture of contaminants with a local exhaust system. The objective is to evaluate the influence of multi-slotted hoods on the capture efficiency and its impact on indoor air quality. The ventilation in the proximity of slots and the contaminant dispersion throughout the room were predicted using the commercial tool "Fluent". Characteristics of the polluted air were examined for different geometrical opening configurations for a two-dimensional model. Two parameters were examined: the number of slots and the slot width. Two-dimensional simulations revealed that, at a constant volumetric exhaust flow rate, a gradual reduction in contaminant concentration is experienced throughout the room as the number of slots increases. It was found that there is an optimal number of slots that reduces this concentration to 12% compared with a non-slotted hood system. The slot width analysis also revealed the existence of an optimum value for which the contaminant reduction is maximal. Following this basic assessment, computations were performed on a more realistic, three-dimensional model. These preliminary calculations indicate that flow computations need to be three-dimensional to address real-life situations more accurately.

INTRODUCTION

Industrial exhaust ventilation systems are used to remove airborne contaminants consisting of particulates, vapor and gases, all of which can create an unsafe, unhealthy or undesirable atmosphere for workers. There are two types of exhaust systems: *general exhaust ventilation* (GEV), in which an entire work space is exhausted without considering any specific operation; and *local exhaust ventilation* (LEV), in which the contaminant is controlled at its source [13]. In the industry, LEV is the preferred method because the flow rates, and therefore the costs, are lower than in GEV. Moreover, with GEV only, it may be difficult to achieve the high level of contaminant control that is needed to reduce the worker's exposure near a contaminant source. The main concept used in an LEV to select the adequate volumetric flow to with draw air through a hood is the capture velocity. This is the velocity of the air at the point of contaminant generation. The contaminant enters the moving air stream at the point of generation and is conducted along with the air into the hood. Traditionally, the design of LEV hoods is based on the empirical velocity formulas [4,13] that give the capture velocity profile at the front of the hood. While useful, this method does not quantitatively take into account the effects on the efficiency of the LEV of the momentum of the contaminant source, disruptive air currents and obstacles in the flow field. For this reason, a better option is to study indoor air quality using computational fluid dynamics (CFD), because CFD is capable of providing information on factors such as the distribution of flow and concentration at all grid points, independently of the complexity of the geometry. The single rectangular slot is highly applied since it is very efficient for capturing large amounts of contaminants. CFD has been used on this type of slot to assess the performance of LEV systems and to define new design guidelines [1,8,10]. Kulmala [2] investigated the accuracy of the numerical simulation of an air flow field generated by a rectangular exhaust opening. The calculations were made using the standard k-ε model, and the results were verified with laser Doppler measurement. Multi-slotted hoods are often used on exhaust systems, and it is widely excepted that the slots only help to distribute air over the hood face and do not influence capture efficiency. In this paper, we use CFD to analyze the influence of slot number and slot width on capture efficiency. We first examine the effectiveness of the multi-slotted hood in a 2-D model, and then we apply similar concepts to the 3-D case of a worker positioned near a contaminant source.

BASIC GEOMETRY

The 2-D room geometry, including a hood configuration with 6 slots, is depicted in Fig. 1 (not to scale). The contaminant source is considered to spread material over a surface 2 m in length. The area to be studied is near the hood, and the dimensions of the room have been increased to impose boundary conditions (BC) that will not affect the region of interest.

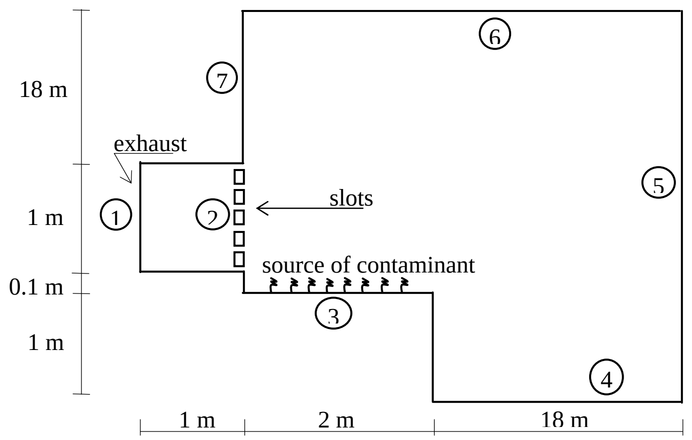


Figure 1: Geometry with slots.

NUMERICAL METHOD

Governing Equations

The aim of the numerical prediction is to solve the governing set of partial differential equations representing an isothermal incompressible flow without buoyancy effects. The equations that describe the flow of a fluid, heat and concentration within an enclosure are based on the conservation of mass, momentum, energy and species concentration. The general form of transport equation for property Φ is:

$$\underbrace{\frac{\partial(\rho\Phi)}{\partial t}}_{Transient} + \underbrace{\frac{\partial}{\partial x_i}(\rho\Phi U_i)}_{Convection} = \underbrace{\frac{\partial}{\partial x_i}\left(\Gamma_\Phi \frac{\partial\Phi}{\partial x_i}\right)}_{Diffusion} + \underbrace{S_\Phi}_{Source} \text{ ; i=1, 2, 3}$$

In this expression, ρ represents the density of the fluid, Γ_Φ the diffusion coefficient, S_Φ the source term and U_i the velocity components along the x_i coordinates. From this general form, the Navier-Stokes, energy and species concentration equations can be obtained. Namely:

Conservation of mass:

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho U_i}{\partial x_i} = 0$$

Conservation of momentum:

$$\frac{\partial \rho U}{\partial t} + \frac{\partial \left(\rho U_i U_j\right)}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\mu \frac{\partial U_i}{\partial x_j}\right) + g_i\left(\rho - \rho_0\right)$$

Conservation of energy:

$$\frac{\partial \rho H}{\partial t} + \frac{\partial}{\partial x_i}\left(\rho U_i H\right) = \frac{\partial}{\partial x_i}\left(\lambda \frac{\partial T}{\partial x_i}\right) + \frac{\partial P}{\partial t}$$

Conservation of species:

$$\frac{\partial}{\partial x_i}\left(\rho \overline{U}_i \overline{C}\right) = \frac{\partial}{\partial x_i}\left(D \frac{\partial \overline{C}}{\partial x_i} - \rho \overline{\mu_i c'}\right)$$

where P, H, T, μ, g, λ, C and D denote pressure, enthalpy, temperature, dynamic viscosity, gravity acceleration, thermal diffusivity, concentration and molecular diffusivity respectively. The contaminant used in this study is a gas.

To handle turbulence, the renormalization group (RNG) $k-\varepsilon$ was selected. The SIMPLE algorithm was used for the pressure-velocity coupling. Boundary conditions and meshing aspects are discussed in the next section.

Grid and boundary conditions

In order to obtain a numerical solution of the above equations, a descretization of the room is required, followed by the application of a solver. Because solutions will be sought using the commercial package Fluent, the companion software Gambit was applied to generate the grid. Based on the topology room and of the slotted hood, hybrid grids, both structured and unstructured, were applied. An adaptive mesh refinement algorithm around the slots was employed, which permits a more accurate representation of the boundaries.

In order to compute a solution, boundary conditions need to be applied. In this study, the most relevant parameter is at the exhaust boundary. To enforce an adequate exhaust mass flow rate, an adequate capture velocity must be established. Based on successful experiments, ASHRAE [13] proposes ranges of capture velocities for several industrial operations under specific conditions of contaminant dispersion. In this case, we consider the contaminant source as an evaporation tank, which means that the contaminant is released into still air with essentially no velocity. The proposed capture velocity range is 0.25 – 0.5 m/s [13]. To ensure the corresponding exhaust mass flow rate, the transport equations were solved several times until a velocity of 0.35 m/s was obtained at a reference point. This was located 1 m from the hood face and 0.5 m above the surface of the source of contaminant.

At the outlet (side 1 in Fig 1), a mass flow rate of 0.1 kg/s was imposed. This same rate was also imposed at the inlet (side 5). Solid walls were considered on all the other surfaces. A constant standard temperature of 300^o K was considered at both the inlet and the outlet. All solid walls where considered to be adiabatic. Table 1 summarizes the boundary conditions.

Table 1: Boundary Conditions

1	Exhaust	Mass flow - outlet = 0.1 kg/s
2	Hood slots	Wall - adiabatic
3	Contaminant source	Wall - adiabatic with diffusion flux. Mass fraction = 1
4,6,7	Wall	Wall - adiabatic
5	Inlet	Mass flow - inlet = 0.1 kg/s

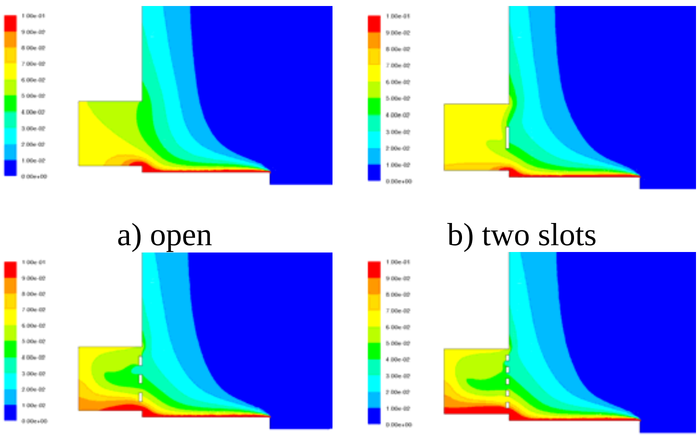
Below, we present the solution of the governing equations for 2-D and 3-D geometries using the commercial package Fluent [11].

2-D CALCULATIONS

These calculations were carried out to assess the impact of the number of slots and the slot width.

Influence of the number of slots

This test is quite simple and involves conducting computations of hood performance by changing the number of openings at the hood inlet. Fig. 2 shows iso-values of species concentration for various numbers of slots for a constant outlet mass flow rate of 0.1 kg/s. Although this qualitative view indicates better suction with the increase in the number of openings, a quantitative analysis is required. This result is illustrated in Fig. 3, where the total mass of the contaminant in the entire room has been plotted against the number of slots. Based on this simple representation, it appears that the optimum outcome corresponds to 6 slots, and that a further increase in their number does not help to reduce the species concentration. On the contrary, it amplifies it.



a) open b) two slots

c) four slots d) six slots

Figure 2: Iso-values of species concentration. The outlet mass flow rate is constant, at 0.1 kg/s.

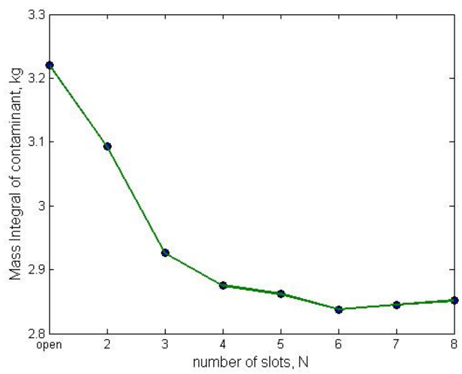
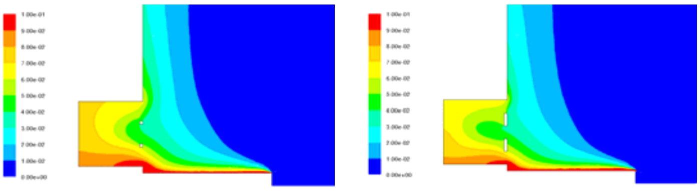


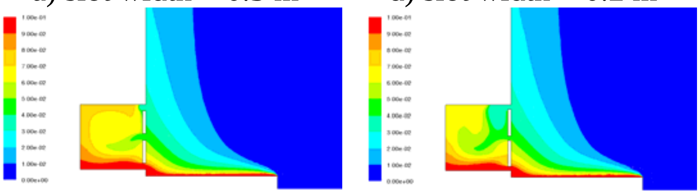
Figure 3: Mass integral of contaminants for various numbers of slots.

Influence of slot width

A second test was conducted to analyze the impact of slot width on species concentration. Only the case of a hood with two slots was studied.



a) slot width = 0.3 m d) slot width = 0.2 m



g) slot width = 0.1 m h) slot width = 0.07 m

Figure 4. Iso-values of contaminant concentration for slot widths of 0.3, 0.2, 0.1 and 0.07 m.

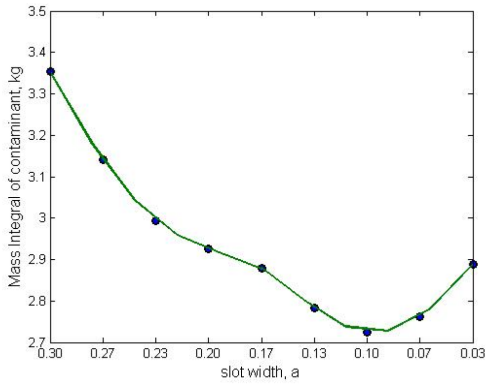


Figure 5: Mass integral of contaminants for the variation in slot width.

Again, this is a naïve test, in which the slot width varied from 0.3 m to 0.03 m. As in the previous test, Fig. 4 shows the simulation results by means of iso-values of concentration. As expected, the differences are noticeable behind the hood inlet. However, what is needed is quantitative information on the room. This is presented in Fig. 5 as the total contaminant in terms of slot width. The curve indicates that the slot width plays an obvious role, and that the optimum width is 0.1 m.

Using these results as a basic platform, we proceeded with 3-D modeling.

3-D MODELING

Because a 2-D world corresponds to “a slice” of a 3-D geometry, a first calculation in three dimensions was carried out by considering a wall-to-wall hood with various numbers of slots. Thus, a vertical plane passing through the middle of the hood corresponds to the 2-D world previously analyzed. An illustration of this idea is shown in Fig. 6, where iso-values of the species concentration obtained when using four slots are shown. As expected, these results coincide with the 2-D calculations.

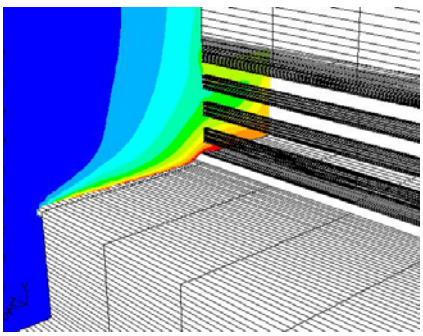


Figure 6: Contours of the mass fraction of the contaminant at mid-plane.

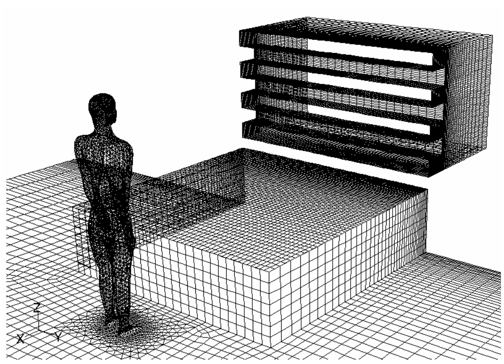


Figure 7: Meshing the worker and the hood.

After completing this basic verification, a more realistic geometry was considered (Fig. 7), which includes a mannequin positioned near contaminants released from the horizontal surface and controlled by a lateral hood. In this type of 3-D modeling, mannequins have been introduced ([5] [6]) to investigate numerically the influence of different body shapes on the exhaust efficiency. In these studies, it was found that the use of such a simplified human body is satisfactory when focusing on the global airflow pattern in a ventilated room. In our study, the model is intended to closely represent an actual worker standing in a particular spot. With this more realistic shape, the airflow pattern around the human body may also be addressed. Because there are many possible locations for the person, a position facing the grid exhaust was chosen for a preliminary test.

Fig. 8 shows qualitative results by means of path lines near the hood colored according to the mass fraction concentration. Details around the body are shown in Fig. 9, where the swirling nature of the flow in front of the face can be appreciated.

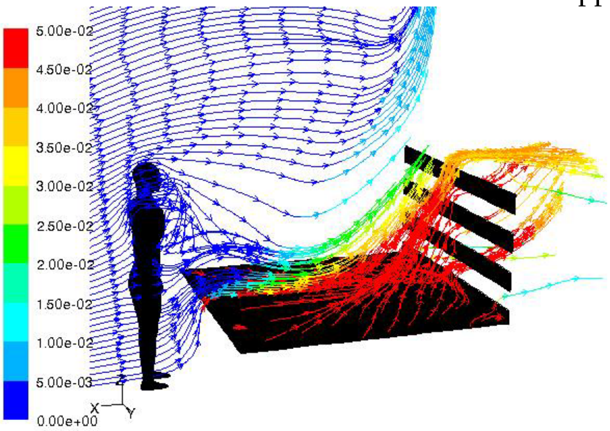


Figure 8: Path lines colored by the mass fraction of the contaminant.

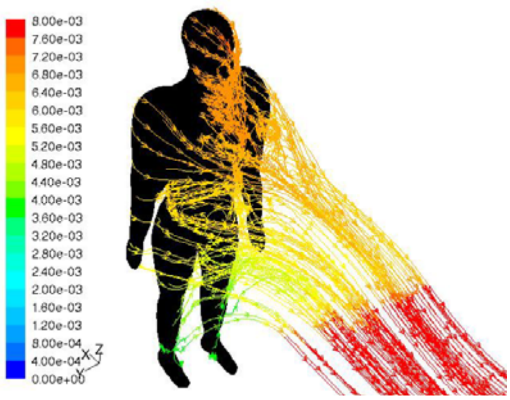


Figure 9: Path lines near the mannequin colored by the mass fraction of the contaminant.

Quantitative results are displayed in Fig. 10, where the total mass integral of the contaminant is shown against the number of slots.

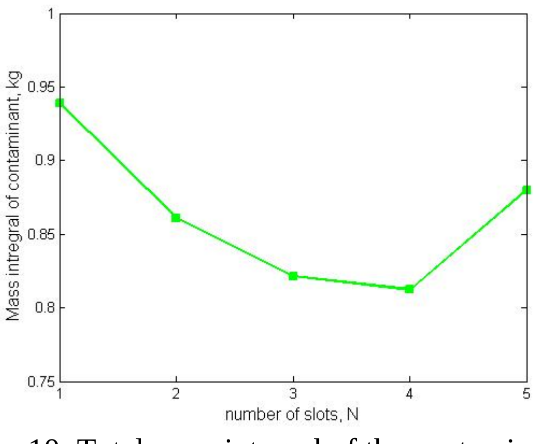


Figure 10: Total mass integral of the contaminants for different numbers of slots.

This particular configuration for a mannequin facing the hood shows a similar behavior to that found for the 2D case; that is, a predicted ideal number of slots to achieve a better configuration of the contaminant’s control system.

Note that the number of slots calculated (4) is not the same as that found in the 2-D analysis (6). This has been attributed to two factors: first, the width of the hood, for which the lateral effects do not exist in 2-D; second, to the presence and position of the worker, which were not considered in the 2-D analysis either. From this particular result, it becomes apparent that more elaborate 3-D investigations need to be carried out.

CONCLUSION

Two- and three-dimensional studies where conducted using CFD to investigate the benefit (12% concentration reduction) of slotted hoods in controlling contaminant levels in working environments. The commercial software Fluent was used to perform the simulations. An optimum number of slots to reduce the contaminant concentration was found in both cases. Differences in the predicted number of slots indicate that hood size and the presence and position of a worker require a variety of 3-D calculations to assess the impact of the number of slots in more realistic situations.

ACKNOWLEDGMENT

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Smoke separation with Air curtains analysed using CFD-simulations

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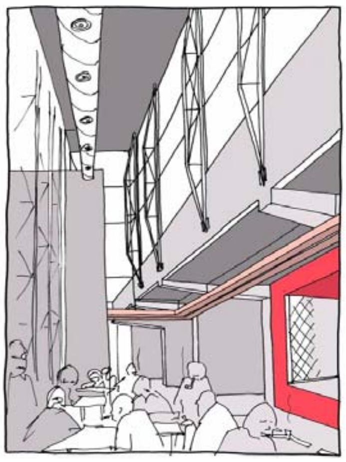
SUMMARY

Cfd-calculations were performed to test whether it is possible to separate a non-smoking zone from a smoking zone using an air curtain. The cfd-calculations resulted in the following conclusions: 1. A larger exhaust flow is best. 2. An air curtain with a low air curtain velocity and an air intake from the smoke zone (as opposed to an air intake from the non-smoking zone) has the lowest smoke concentration in the non-smoking zone. 3. For a maximum of 40 smokers and a ventilation flow compatible with the Dutch building law (1500 m^3/h for this smoke area) an air curtain velocity of 0.75 m/s is the best solution for the smoke zone in the hall of the faculty of Architecture in Delft. 4. With higher air curtain velocities, the air curtain will become a smoke curtain if the exhaust flow is not large enough.

INTRODUCTION

Based on the Tobacco law of 2002, smoking in many buildings is only allowed in separate smoking rooms. However, by physically separating the smokers and the non-smokers, the visual, social and function unity is lost. This problem can be solved by making smoke areas in common rooms where the smoke-free zone is separated by air curtains in stead of walls. In this article computational fluid dynamics (CFD) simulations are presented which show the effectiveness of an air curtain for various air-curtain configurations and various ventilation flows.



All simulations were performed for the hall of the faculty of Architecture of the Delft University of Technology (The Netherlands), see figure 1. The smoke area is situated on the west side of the common hall and has a floor area of 80 m2 with about 40 seats for the smokers. Three sides of the smoke area are outfitted with air curtains, the fourth side of the smoke area consists of a glass facade. A mechanical exhaust is situated on the façade side. A lowered ceiling is applied at a height of 3 meters, compatible with the average room height in the Netherlands.

Figure 1. The smoke room in the hall of the Architecture faculty of the Delft University of Technology

THEORY

The air curtain is not only a separation between the smoking and the non-smoking zone. The air, which is blown downward by the air curtain, mixes with air from both the non-smoking zone and air from the smoking zone, see figure 10. This is not a negligible amount but can amount to 90 % of the air volume flow from a single air curtain [3]. This has the unwanted side-effect that the higher the air curtain velocity, the larger the amount of air that mixes with the air curtain air.

From the schematic drawing in figure 10 it is possible to understand why an air curtain which circulates smoking air (top drawing) has a better performance than an air curtain which circulates non-smoking air (middle drawing). The total amount of air which flows from the bottom of the air curtain to the non-smoking zone due to the air curtain is larger with a recirculating air curtain which draws its air from the non-smoking zone. This airflow from the bottom of the air curtain to the non-smoking zone does have a fair amount of unwanted smoke due to the entrainment (mixing) of air on either side of the air curtain.

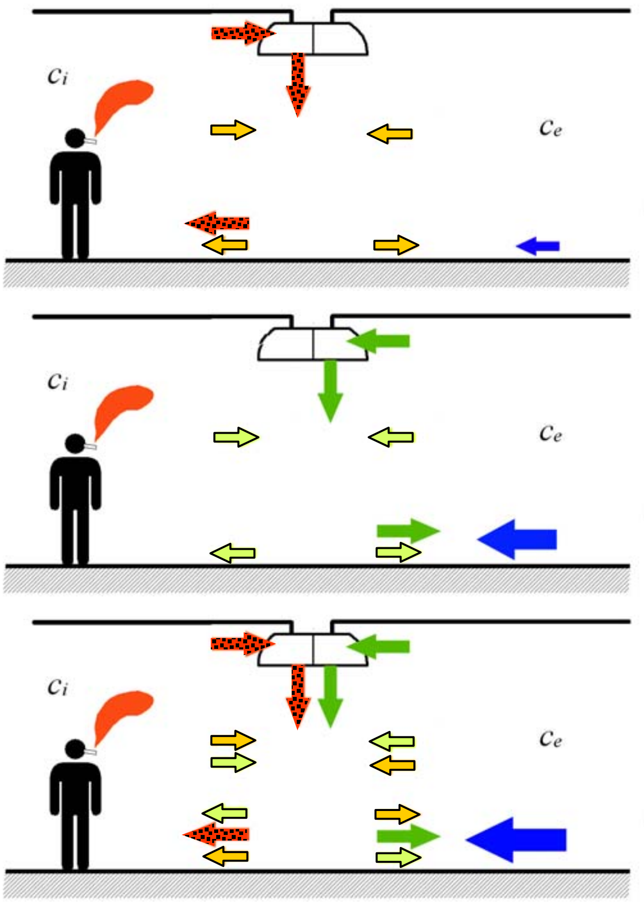


Figure 2. Schematic drawing of the working of the different air curtain configurations. From top to bottom: the smoking side air curtain, the non-smoking side air curtain and the double air curtain. The framed arrows represent the induced air. The dark arrows represent the necessary amount of exhaust air which is necessary to countermand the induction effect.

Having a larger or equal amount of ventilation air flowing in the opposite direction should be present to prevent the smoke-mixed air from reaching the non-smoking zone. A lower exhaust ventilation amount can therefore be enough to prevent the smoke-mixed air from a smoking-side air curtain to reach the non-smoking zone but not enough to prevent the smoke-mixed air from a non-smoking side air curtain to reach the non-smoking zone. Similarly, a higher air curtain velocity requires a larger exhaust ventilation flow to counterbalance the effect of a higher air curtain velocity.

As a criterion for obtaining a low smoke concentration in the non-smoking zone, the parameter G can be defined. This parameter depends on the kind of air curtain, the air curtain velocity and the amount of air curtain ventilation: $G = \frac{FQ_0}{Q_v}$. Q_0 is the air curtain flow. F Q_0 is the amount of air flow from the bottom of the air curtain to the non-smoking zone, Q_v is the amount of exhaust ventilation. The criterion is then G < 1 which indicates that the amount of air flow from the bottom of the air curtain to the non smoking zone is less than the amount of exhaust ventilation. F is the entrainment factor of the air curtain. This factor can be approximated by 0.9 for a smoking side air curtain, by 1.8 for the non-smoking side air curtain and by 2.7/2 for the double air curtain.

METHODS

Simulating cigarette smoke and a smoker

The smoker is simulated by block of 0.2 by 0.2 by 1.7 m^3 with a heat production of 80 W. The smoker is situated at a distance of 1.5 m from the west façade, see figure 3. The smoker exhales 1 m^3/h at a height of 1 m over a surface of 4.0 cm2 in horizontal direction facing the air curtain. The exhaled air has a temperature of 35 °C and a velocity of 0.7 m/s. The exhaled air has a concentration of 1.152 mg nicotine per kg air [2]. Nicotine is assumed to have the same density as the air (1.19 kg/m^3).

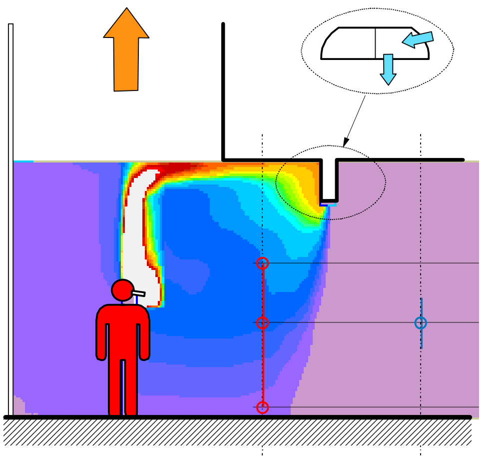


Figure 3. Schematic drawing of the smoker as a source in the cfd-simulations

Modelling the hall

Only a vertical cross section of the hall (from glass plane to glass plane, see figure 4) is modelled. In the middle of the model a smoker is situated. Perpendicular to the cross section two adiabatic boundaries are defined. The outside air is assumed to have a temperature of 20 °C. The simulated model has a length of 13.2 m a width of 10 m and a height of 3.5 m, thus a volume of 462 m^3.

The exhaust is modelled as an exhaust area of 0.3 m. over the entire width of the glass, see the upward arrow in figure 4. The inflow is modelled as an opening of 4.2 by 10 m^2 (again over the entire width of the glass), see the downward arrow in figure 4. For a maximum number of people of 40, the exhaust is simulated with a capacity of 1500, 3000 en 4500 m^3/h. This correspond to approximately to one to three times the Dutch Building Regulations. However, the amount of 4500 m^3/h is height and should preferably not be applied due to corresponding large energy costs and a risk at high air velocities (draught).

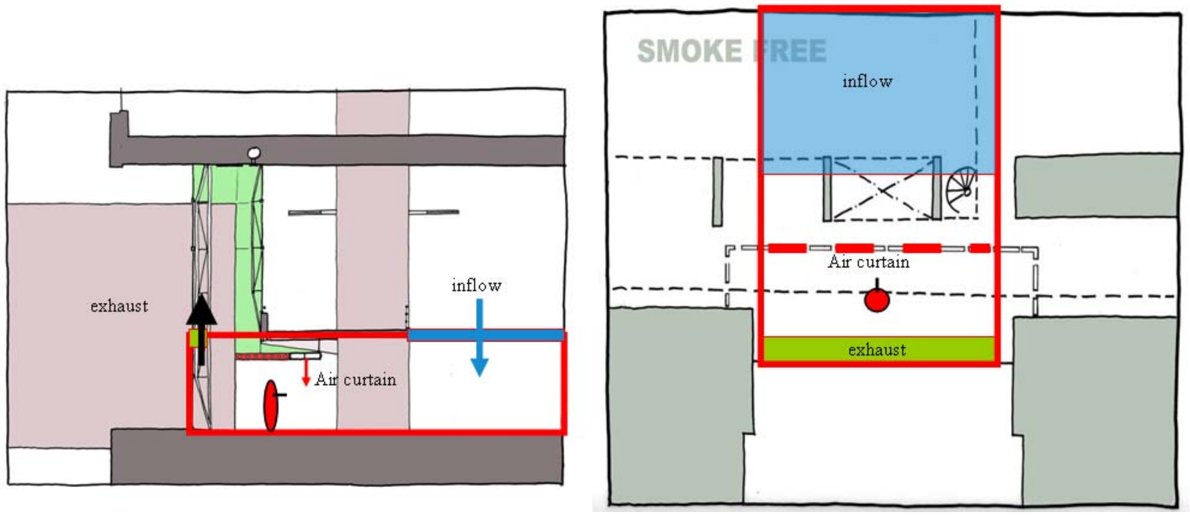


Figure 4. Cross section and floor plan of the hall of the Architecture faculty of the Delft University of Technology.

Modelling the air curtain

Cfd-simulations have been performed for 21 combinations of air curtain configurations (7) and exhaust volume flows (3) in order to calculate the lowest smoke concentration in both the non-smoking zone, and the smoking zone. The simulated air curtain configurations are given in table 1.

Table 1. The simulated variants and their names, totalling 7 air curtain combinations.

Name	**Air curtain**	**Air curtain velocity (m/s)**	**Air curtain volume (m^3/h)**
Er_0.75	Non-smoking side	0.75	2700
D_0.75	Double	0.75	2700
Er_0.5	Non-smoking side	0.5	1800
D_0.5	Double	0.5	1800
El_0.75	Smoking side	0.75	2700
El_0.5	Smoking side	0.5	1800
no	No air curtain	0.0	0

The air curtains are simulated as simple air curtains without any heating or cooling capacity. The simulated downward velocities from the air curtain(s) are: 0.0, 0.5 en 0.75 m/s with corresponding air curtain volume flows of 0 m^3/h, 1800 m^3/h and 2700 m^3/h. The total installed air curtain configuration consists of two air curtains. One inside ring with an inflow from the smoking area, and one outside ring with an inflow from the non-smoking area, see figure 2. The air curtain size is 0.2 by 0.6 m^2 over the entire width of the glass plane at a distance of 4.2 meters from the glass plane. Both the outside and the inside ring are simulated as recirculation elements in the cfd-simulation.

Points

To quantify the results, the smoke concentrations have been calculated at three positions in the smoking area and 1 position in the non-smoking area, see figure 5. The points are taken in the smoking zone at a height of 0.1 m., 1.1 m. (sitting position) and 1.8 meter (standing position). The point in the non-smoking room [ce] is taken at a height of 1.1. m.

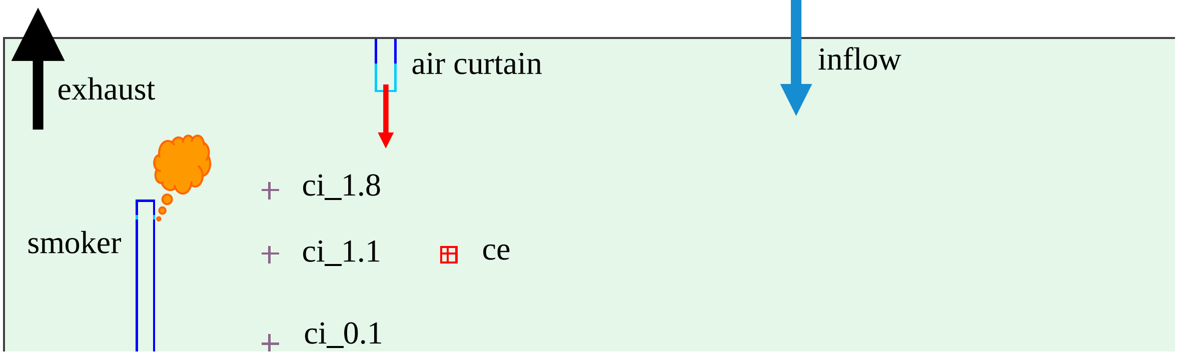


Figure 5. Position of all measured points. At 3 meters from the left glass wall in the smoking zone the point ci_0.1, ci_1.1, ci_1.8 at a height of 0.1, 1.1 en 1.8 meters; at 5 meters form the left glass wall in the non-smoking zone, the point ce at a height of 1.1 meters.

CFD-simulations

The simulations were performed with the commercial computerprogramm Flovent [2], version 5.1. Flovent uses a rectangular grid. A single simulation is performed with around 140.000 grid cells, the k-ε model, and a calculation time of around 2 hours until convergence on a Pentium computer.

RESULTS

In figure 6 the cfd-simulations are shown. In figure 6 both the smoke concentration and the route of certain selected air particles from the inflow of clean air (see the square in figure 7) to the exhaust point is shown for various configurations. The legend of figure 6 ranges from light (purple, 0.0 kg smoke/kg air) to dark (red, 8.4 10^{-6} kg smoke/kg air). A smoke concentration of 8.4 10^{-6} kg/kg air corresponds to a percentage of 8.4 10^{-6} kg / 1.512 mg = 0.5 % of the smoke concentration which is exhaled by the smoker. Areas with a smoke concentration higher than 0.5 % are white.

The lighter (purple) area is larger for a larger exhaust flow; a larger exhaust flow therefore leads to a lower smoke concentration over a larger area. When the flow pattern of the clean air is compared with the smoke concentrations in the non-smoking room, it can be seen that the lowest concentrations are obtained when the clean air passes the air curtain with the smallest distance from the floor. This is exceptionally clear for a smoking side air curtain with a velocity of 0.5 m/s and an exhaust flow of 4500 m^3/h and for the no air curtain configuration with an exhaust flow of 4500 m^3/h.

In figure 7 the smoke concentration in a number of points is shown as a function of the air curtain configuration. As expected from figure 6, the smoke concentrations decrease with increasing exhaust volume flow. Also concluded from figure 6 is the fact that the smoking side air curtain (El) is to be preferred over a non-smoking side air curtain or a double air curtain. A non-smoking side air curtain is simulated to have a higher smoke concentration than a double air curtain for most velocities and exhaust flows.

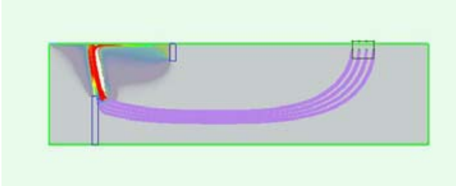
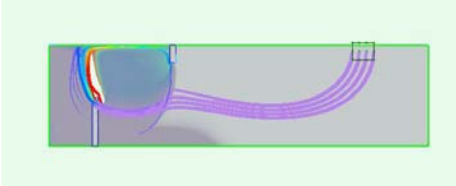
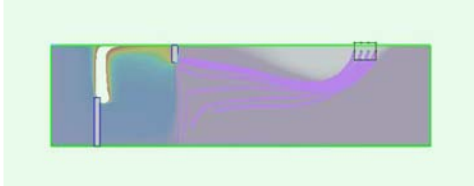
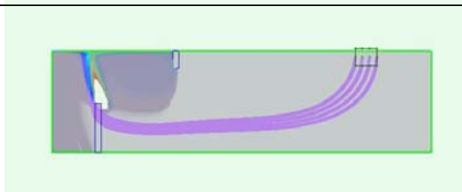
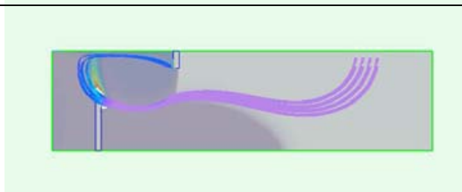
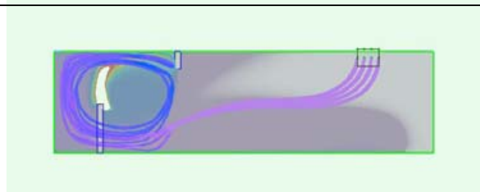
	Exhaust flow: 4500 m^3/h	Exhaust flow: 1500 m^3/h
no		
Er_0.5		
El_0.5		
El_0.75		

Figure 6. Simulated smoke concentrations and simulated airflow patterns for certain inflow air particles for a number of air curtain configurations and two exhaust flows(1500 m^3/h and 4500 m^3/h).

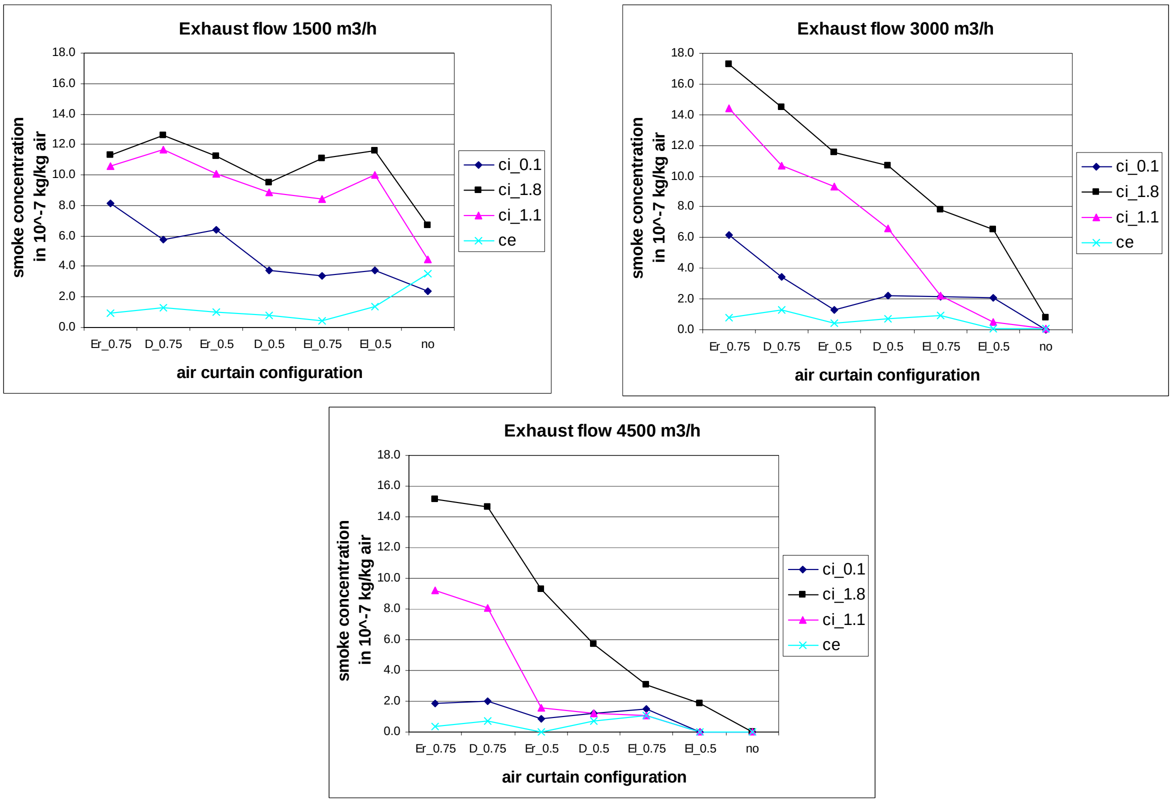


Figure 7. Simulated smoke concentrations ce, ci_0.1, ci_1.1 en ci_1.8 as a function of the air curtain configuration for three exhaust flows (4500, 3000 and 1500 m^3/h).

Contrary to expectations by a general public, the lowest possible air curtain velocity should be applied in order to obtain the lowest smoke concentration in the non-smoking and the smoking room. For high exhaust flows, this velocity is close to 0.0 m/s, for lower exhaust flows this value increases due to the adverse effects of diffusion of smoke to the non-smoking zone.

DISCUSSION

The most important value which determines the smoke nuisance is the smoke concentration in the non-smoking zone: ce. In figure 8, this value is presented as a function of the above defined G (= FQ_0/Q_v). Figure 8 shows that the value of G indeed influences the smoke concentration in the non-smoking zone (ce). However, for one point the cfd-simulated ce concentration is far too high although G is smaller than 1. This is the situation where there is no air curtain (FQ_0 = 0 and thus G = 0) and where the exhaust flow is not high enough to counterbalance the effect of diffusion (1500 m^3/h). For G-values between 0.8 and 1.0, the cfd-simulations show an increase in smoke concentration in the non-smoking zone. In practice a criterion of G < 0.8 seems more appropriate from these cfd calculations.

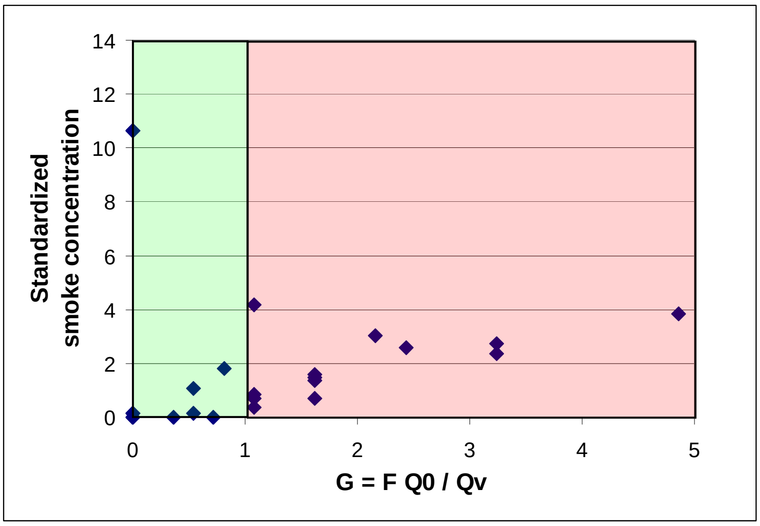


Figure 8. Standardized smoke concentrations in the non-smoking zone as a function of G. The smoke concentration in the non-smoking zone (ce)is standardized with respect to the smoke concentration in the exhaust air at an exhaust flow of 4500 m3/h.

That the G value is an appropriate criterion can also be seen when two simulations with the same G-value are compared. Compare for example the configuration of a double air curtain with an air curtain velocity of 0.5 m/s and an exhaust flow of 3000 m3/h and the double air curtain configuration with an air curtain velocity of 0.75 m/s and an exhaust flow of 4500 m3/h, see figure 8. Both simulations look remarkably alike.

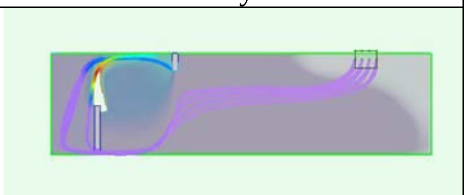
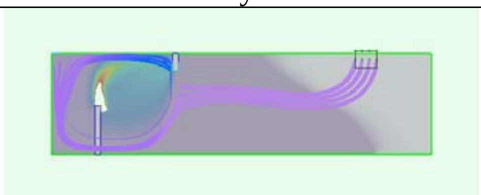
	Exhaust flow: 3000 m^3/h Air curtain velocity: 0.5 m/s	Exhaust flow: 4500 m^3/h Air curtain velocity: 0.75 m/s
Double air curtain		

Figure 9. Comparing two simulations with the same G-value.

CONCLUSIONS

From the cfd-calculations the following conclusions can be deduced:

- A larger ventilation flow is best.
- An air curtain with a low outlet velocity and an air intake from the smoke zone (as opposed to an air intake from the non-smoking zone) has the lowest smoke concentration in the non-smoking zone.
- For the maximum of 40 smokers and a ventilation flow compatible with the Dutch building law (1500 m^3/h for this smoke area) an air curtain velocity of 0.75 m/s is the best solution for the smoke zone in the hall of the faculty of Architecture in Delft, according to the CFD-simulations.
- With higher air curtain velocities, the air curtain will become a smoke curtain if the exhaust flow is not large enough.

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Experimental study of particle concentrations in an underground station

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SUMMARY

Three experiments were carried out in an underground station of Paris. The aim was to evaluate the influence of three parameters on particle number concentrations: ventilation, passengers and trains. These experiments took place by night when traffic stopped to freely study all the three parameters individually. Different measurements have been performed: particle number concentrations, air velocity, CO_2 concentration and particle mass concentration (PM_{10}).
First results do not show any influence of ventilation when turned off and on during one night. Passengers' activity and especially trains pass-by have an influence on particle concentration. Due to the fact that fine particle concentration levels are not reproducible, results should be interpreted very carefully. But as influences can be observed, this study could be completed by another one that could permit to distinguish emission and transport of particles, especially particles with a diameter less than 1µm that represent the major part of particle number concentrations.

INTRODUCTION

Indoor air quality in offices, schools and houses has been studied for years. It is not the case of the indoor air quality in underground train stations, which is still not well understood. Different studies have been carried out in several underground stations (Stockholm [1], Helsinki [2], London, New York, Rome,..) to evaluate mass concentration level. Measurements performed in Helsinki also mentioned particle number concentrations (10 nm<Diameter<500 nm) in the station. All these studies have been carried out to better investigate personal exposure. But it is also interesting to understand what kind of phenomenon influences particle concentrations in order to better appreciate station particle's sources.
To this end, three experiments have been carried out, in order to find the influence of three different parameters, namely ventilation, passengers and trains on the pollutant concentrations. These experiments took place by night when public access is forbidden in order that these three parameters could be individually investigated.

METHODS

The three measurements were performed in Magenta station (RER line E) at the same place at the end of one platform. The ventilation system of the station is described on Figure 1, arrows schematise the air circulation. This air circulation is the least favourable mode in terms of particles concentration.

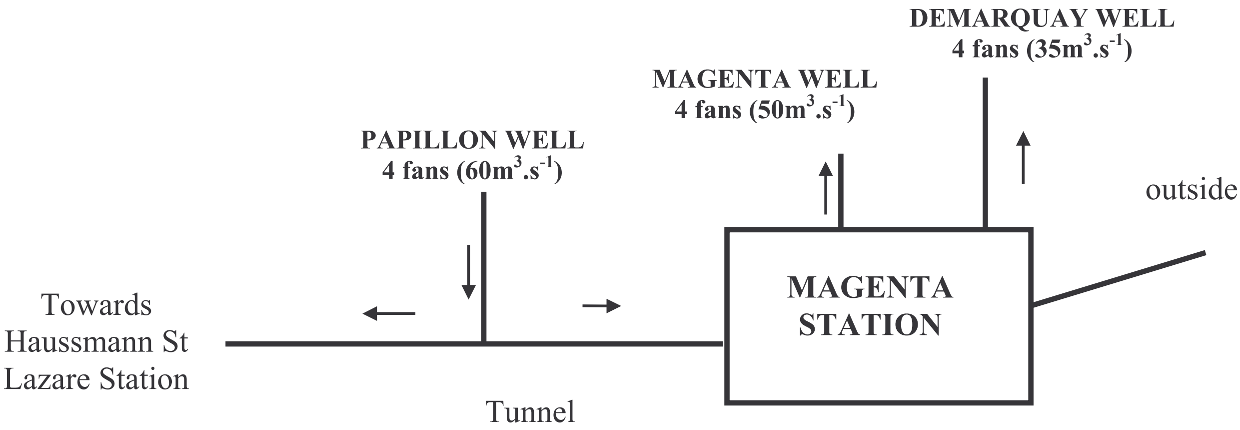

Figure 1: description of the ventilation system.

Three particles counter were arranged at three different heights (0.50, 1.10 and 1.70m) and another one was arranged at 1.70m height 1m further. An anemometer was also placed 1m from the three counters. Measurements of particle mass concentration (PM_{10}) and CO_2 concentration were also performed. Measurement devices were placed as shown on Figure 2.

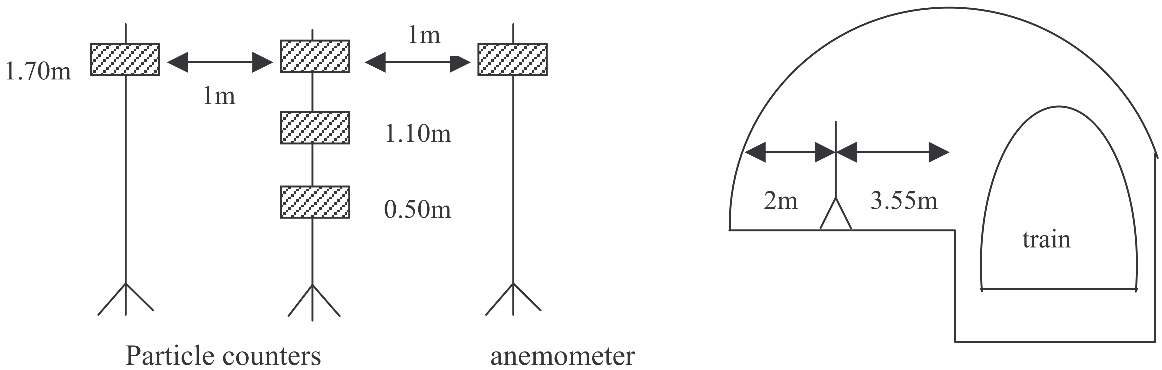

Figure 2: arrangement of the material.

During the first experiment, Demarquay, Papillon and Magenta well fans were turned off between 11:05pm and 11:35pm. Fans were turned on between 2:55am and 4:50am.
During the second experiment, to study the influence of passengers' activity on particle concentrations, forty subjects followed three scenarios. After the traffic stopped and a waiting time of approximately 45' (time for particle concentrations decreasing to urban level), the first one consisted for the forty subjects in staying in front of captors during 15'without moving to assess a possible natural emission. During the second one, named "off-peak period", four groups of ten subjects walked one after the others during 4' each in front of sensors. The last scenario, named "rush hour", consisted in walking all the forty subjects in front of captors during 15'.
During the last experiment, to study the influence of train pass-by on particle concentrations, two scenarios were repeated: pass-by without stopping and pass-by with braking and stopping.
The timetable of the different pass-by is shown in Table 1.

Table 1: Timetable of the different pass-by.

	braking 1	stop 1	braking 2	stop 2
entry	2h18min57s	3h02min40s	3h47min34s	4h32min31s
stop	2h19min24s	/	3h48min03s	/
Setting in motion	2h20min30s	/	3h49min04s	/
departure	2h20min56s	3h03min07s	3h49min33s	4h32min57s

RESULTS

Ventilation

During the measurement period the relative humidity ranged from 47% to 57% and the temperature from 11.4 to 13.4°C.
Firstly, measurements of particle concentrations show higher concentrations for particles ranged between 0,3 and 0,4µm than those for bigger height classes. During the night, fine particle (0,3 to 0,4µm) concentrations part increase whereas the others decrease (Table 5 and Table 3). Globally, air becomes impoverished in particles and PM_{10} concentration in the station reduces to the urban level.

Table 2: Granulometric distribution in Magenta at 00h

size (µm)	0.30-0.40	0.40-0.50	0.50-0.65	0.65-0.80	0.80-1.0	1.0-1.6	1.6-2.0	2.0-3.0
Concentration (part.L-1)	68299	27018	19704	7570	4100	1525	800	734
Repartition (%)	52,5%	20,8%	15,2%	5,8%	3,2%	1,2%	0,6%	0,6%

size (µm)	3.0-4.0	4.0-5.0	5.0-7.5	7.5-10.0	10.0-15.0	15.0-20.0	>20.0	total
Concentration (part.L-1)	180	65	30	4	1	1	0	130031
Repartition (%)	0,1%	<0.1%	<0.1%	<0.1%	<0.1%	<0.1%	0,0%	100,0%

Table 3: Granulometric distribution in Magenta at 2h30

size (µm)	0.30-0.40	0.40-0.50	0.50-0.65	0.65-0.80	0.80-1.0	1.0-1.6	1.6-2.0	2.0-3.0
Concentration (part.L-1)	33550	7441	4272	2360	1805	785	609	543
Repartition (%)	65,2%	14,5%	8,3%	4,6%	3,5%	1,5%	1,2%	1,1%

size (µm)	3.0-4.0	4.0-5.0	5.0-7.5	7.5-10.0	10.0-15.0	15.0-20.0	>20.0	total
Concentration (part.L-1)	61	26	11	0	0	0	0	51463
Repartition (%)	0,1%	0,1%	<0.1%	0,0%	0,0%	0,0%	0.0%	100,0%

Disturbances due to trains pass-by have an important effect on particle concentrations (less than 1µm). As one can see on Figure 3 and Figure 4, between 00:00am and 01:00am and after traffic started, the particle concentrations variations fit with trains pass-by. When considering air velocity (that also fits with trains pass-by), ranged from 1.0 to 2.2m.s^{-1} on nearest track of measurement's one before traffic stopped and up to 4m.s^{-1} after traffic start on measurement's track, it is easier to understand that particles are under high trains' disturbances. When there is no traffic in the station, air velocity ranged from 0.02 to 0.6m.s^{-1}. Then, particle concentrations decrease to lower level on 45', as shown on Figure 3.

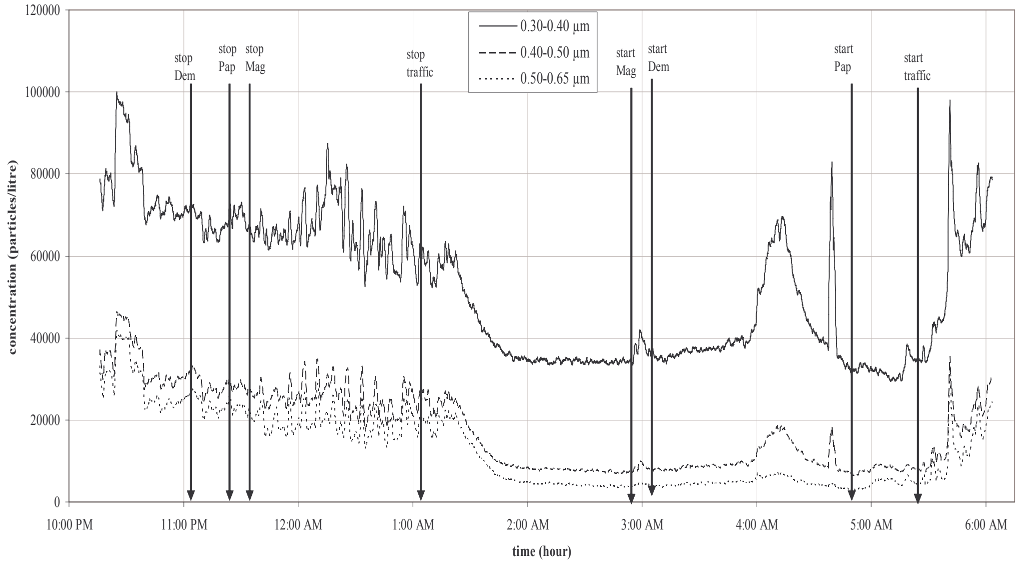

Figure 3: particle concentrations variations at 1.70m.

When looking on a smallest period during the night (on Figure 4), after 11:30 pm, 1' after each train entry on nearest track of measurement's one (when train sets in motion) corresponds to a particle concentrations decrease. This can be explained by the fact that particles move outside the station with train's wake.
As fine particle concentrations are relatively close for each measurement's height, only fine particle concentrations at 1.70m is represented on Figure 4.

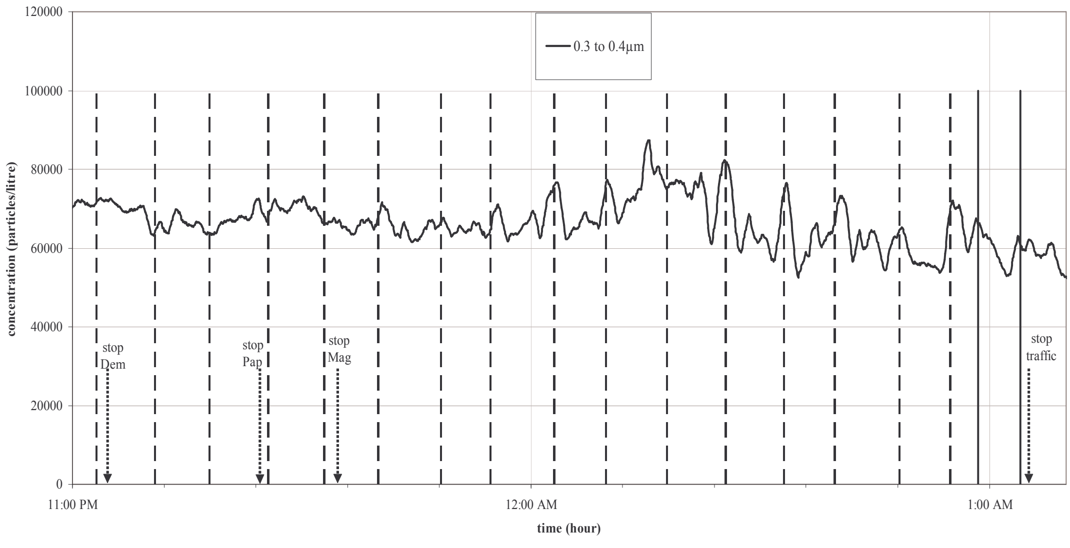

Figure 4: trains entry on nearest track of measurement's one and particle concentration at 1.70m between 11:00pm and 1:15am

: trains entry on the nearest track of measurement's one
: unstopped trains on nearest track of measurement's one
stop Dem/Pap/Mag: stop of Demarquay, Papillon and Magenta well fans
stop traffic: end of traffic
Before traffic stopped, measurements also show that relative humidity increase (approximately 3%) when trains enter in the station. This can be due to passengers when

coming-out from trains or to trains if outside relative humidity is higher than inside one. Because of low variations of CO_2 concentrations, we cannot conclude on the passengers' influence on relative humidity.

Passengers

During the measurement period the relative humidity ranged from 32% to 41% and the temperature from 10 to 12°C.

Concentrations of particles ranged between 0.30 and 0.80µm were very high and others very low during this experiment. As only particles ranged from 1.6 to 4.0µm can be analysed, other particle concentrations variations have not been represented here. Fine particle concentrations were three times higher during this experiment than during the first one. This may be due to "fire tests" that took place the day before. It consisted in smoking out the station and turning on fire ventilation.

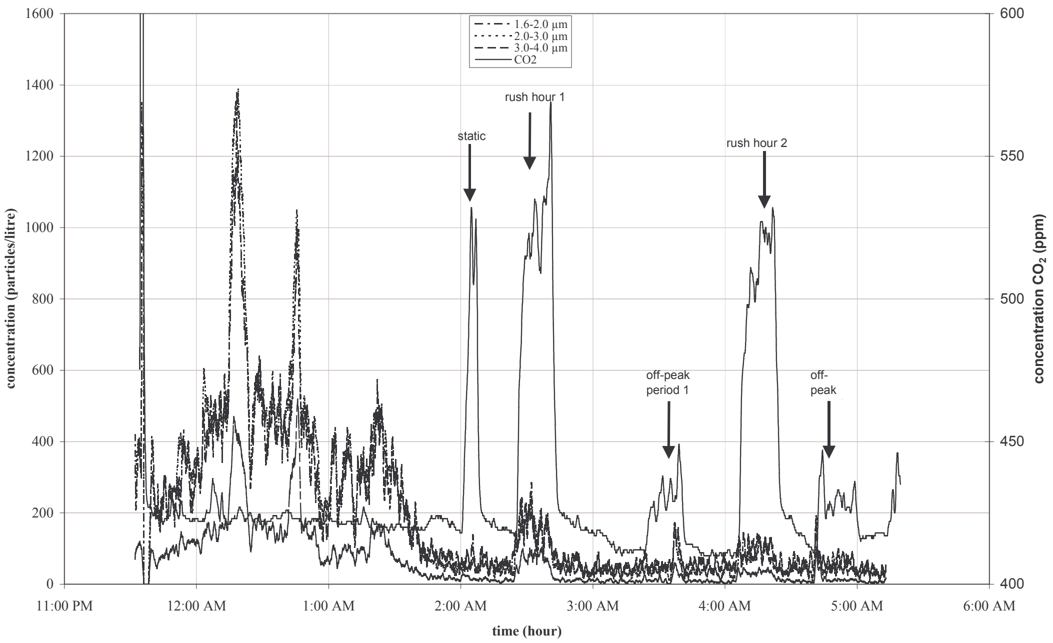

Figure 5 : particle (1,6 to 4,0µm) concentrations variations and CO_2 concentrations variations at 1,70m

Figure 5 shows concentrations of particles ranged from 1.6 to 4.0µm and CO_2 concentration (human presence's indicator).

Five CO_2 concentration peaks can be observed. It corresponds to the different scenarios: static, rush hour and off-peak period.

First rush hour peak (2:30am) is more marked than the second one (4:15am) and off-peak period are spotted with a thin peak at 3:40am and 4:40am.

For the first rush hour scenario, the concentrations variations around the peak are given in Table 4.

According to particle height, concentrations vary from a factor 2 to 10 around the peak.

Although the variations around the second peak of rush hour scenario are less marked, one can observe an influence of subjects' activity on particle concentrations.

Table 4: mean concentrations (part.L^{-1}) during the first rush hour between 2:00am and 3:00am.

time	1.6-2.0 µm	2.0-3.0 µm	3.0-4.0 µm	4.0-5.0 µm	5.0-7.5 µm	7.5-10.0 µm
2:00-2:24am	62	45	14	6	4	1
2:25-2:45am	139	131	57	38	29	11
2:46-3:00am	55	38	11	6	3	1

For the first off-peak period, variations around the peak are given in Table 5.
According to particles' height, concentrations vary from a factor 1.4 to 3 around the peak. Nevertheless, for particle ranged from 5 to 10 µm, concentrations values are in the measurement uncertainty. Concentration variation must be carefully considered.

Table 5: mean concentrations (part.L^{-1}) during the first off-peak period between 3:00am and 4:00am.

heures	1.6-2.0 µm	2.0-3.0 µm	3.0-4.0 µm	4.0-5.0 µm	5.0-7.5 µm	7.5-10.0 µm
3:00-3:23am	50	33	9	5	2	1
3:24-3:39am	68	52	17	10	7	3
3:40-4:00am	55	33	9	4	2	1

For these particles' heights, the influence of subjects' activity is less marked for off-peak period than for rush hour. During the two different scenarios, air velocities are relatively close, ranged from 0.3 to 0.5m.s^{-1} when people walked and close to 0.2m.s^{-1} when there was no activity in the station.
No interpretation can be done on others particles' height because of the unexpected concentrations during this experiment.
As in the first experiment, the same relative humidity variations (approximately 3%) have also been observed during this second campaign.

Trains
During the measurement period the relative humidity ranged from 56% to 64% and the temperature from 16 to 18°C.
Four pass-by were performed during this test: two when breaking and stopping and two other without stopping. Measurements give four different signatures for the four pass-by.
Fine particles are one more time present in majority, even during pass-by trains.

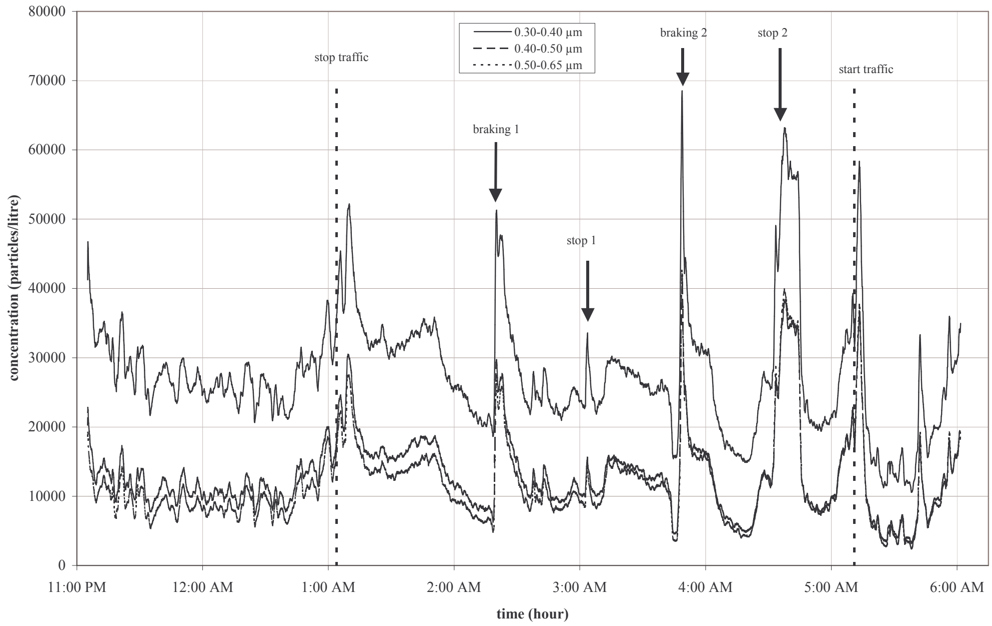

Figure 6 : particle concentrations variations at 0.50m

During this last experiment, the concentrations of particles ranged between 0.3 and 0.65µm are approximately two times lower than during the first one.
Table 6 gives granulometric distribution during this third experimentation. It shows, as during the first one, fines particles represent the major part of particles numbers.

Table 6: Granulometric distribution during the experimentation at 00h

size (µm)	0.30-0.40	0.40-0.50	0.50-0.65	0.65-0.80	0.80-1.0	1.0-1.6	1.6-2.0	2.0-3.0
Concentration (part.L-1)	24984	9632	7940	3350	2120	790	337	362
Repartition (%)	50.33%	19.41%	16.00%	6.75%	4.27%	1.59%	0.68%	0.73%

size(µm)	3.0-4.0	4.0-5.0	5.0-7.5	7.5-10.0	10.0-15.0	15.0-20.0	>20.0	total
Concentration (part.L-1)	78	31	8	4	0	0	0	49636
Repartition (%)	0.16%	0.06%	0.02%	0.01%	0.00%	0.00%	0.00%	100%

As we can see on Figure 6, trains' signatures due to the four pass-by are not reproducible two by two in spite of the same conditions (same operator, same speed…). One pass-by, the first without stopping, does not cause an important signature.
The other three pass-by cause fast increase and decrease particle concentrations. These variations can be observed for the whole particles' height and are probably due to the transport caused by trains' air mixing, as air velocities are up to 4.5m.s^{-1} on measurement's track, as during the first experiment.

The difference which can be observed between the two last pass-by in term of maximum concentrations represents approximately 5400part.L^{-1} for particles ranged between 0.3 and 0.4µm. As trains pass-by are not reproducible, it is difficult to maintain this difference represents the breaking emission's part.

As during the previous experiments, the same relative humidity variations(approximately 3% to 4%) have also been observed during the period of trains pass-by. As there were no passengers in the station during this test, we can conclude that this variation is due to trains pass-by.

DISCUSSION

The first experiment did not show an effect of ventilation on particle concentration. Ventilation may not have an influence on resuspended particles but it may be have an impact on the accumulation of particle concentration. So it could be interesting to turn off the station's ventilation during several days or weeks if it's possible.
The second experiment permitted to evaluate the passengers' influence. As this campaign was polluted by "fire tests", we can only observe trends on a few particle heights. This experiment could be completed by another one that should take place in a more enclosed space with controlled conditions before starting it again in an underground station.
The last experiment gave an idea of trains pass-by influence on particle concentration. This source is probably the most important in the station. But it is also the most complex because it is difficult to estimate the part of emission, transport and resuspension.

Many studies have been carried out to determine PM_{10} and $PM_{2.5}$ levels in underground stations. Particle number concentrations are rarely performed. Such measurements have been done by P.Aarnio [1] to compare it to urban level. In our study, such measurements have been carried out to characterize different particle sources.
To have more information on particle transport, it could be interesting to measure the concentration level in tunnel to compare respective pollution of tunnels and station. But punctual measurements could not be sufficient to understand such difficult phenomenon as particle transport and resuspension.
Nevertheless, these three experiments show particles less than 1µm represent the major part of particle number concentration in the underground station. In order to reduce their concentrations, it is very important to better understand their origins and behaviour.

ACKNOWLEDGEMENT

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Application of Natural Ventilation in Cattle Barns

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SUMMARY

The design of buildings for animal keeping and the dimensioning of ventilation equipment for such buildings require knowledge about heat, moisture and CO_2 production of the animals. In addition to these parameters the emission streams of odour and gases are interesting on the part of action on the environment. Therefore basic research was carried out in different cattle barns, different designed buildings and different number of animals under one roof. Special investigations were done in two naturally ventilated livestock buildings for 288 dairy cows and for 251 beef cattle. The paper reported about the measuring methods and the results regarding the microclimate parameters and the ammonia emission streams.

INTRODUCTION

The design of buildings for animal keeping and the dimensioning of ventilation equipment for such buildings require knowledge about heat, moisture and CO_2 production of the animals. In addition to these parameters the emission streams of odour and gases are interesting on the part of action on the environment. Therefore basic research was carried out in different cattle barns, different designed buildings and different number of animals under one roof. Special investigations were done in two naturally ventilated livestock buildings for 288 dairy cows and for 251 beef cattle.
The climatic parameters are measured by data logger. The air flow pattern is made visible by smoke and the air velocity in the animal zone is measured by hot wire anemometer. Inside the building the gas concentrations (NH_3, CO_2, CH_4 and N_2O) will be recorded by a Multi-Gas-Monitor. To determine the air volume stream the air exchange rate is measured by tracer gas technique. The ATB has developed a special method and technology using the radioactive gas Krypton 85. Up to 40 measuring sensors can be used at the same time – with the measuring frequency of 1 second. High air exchange rates can be measured.
The paper will show the correlation between outside wind speed and air exchange rate. The large openings lead to high air volume streams and therefore the temperature difference between inside and outside is small. Also the ammonia concentration inside the building is very low because of the high air exchange rate. But from the high air volume stream results a high ammonia emission stream. In our paper we will show the air flow pattern and the measured air velocity in the animal zone.

BASICS

The necessary ventilation of livestock buildings can be realised by using forced ventilation, natural ventilation or the combination of both kinds of ventilation. In cattle barns the most commonly used ventilation system is the natural ventilation. In some cases an additional forced ventilation system is used – especially during the summer period if the outside temperature is high and the outside air velocity is low. Under such weather conditions the additional ventilation system is used to achieve more air movement in the animal zone to improve the heat dissipation of the animals. During wintertime such system can be used for a better mixing between fresh air and room air for a better using of the heat production of the animals inside the building.

The driving forces of the natural ventilation are the thermal lift and the outside wind. Both forces produce pressure differences at the building and effect the air stream trough the openings in the structure of the building. The thermal lift results from the temperature difference between inside and outside air (see Figure 1).

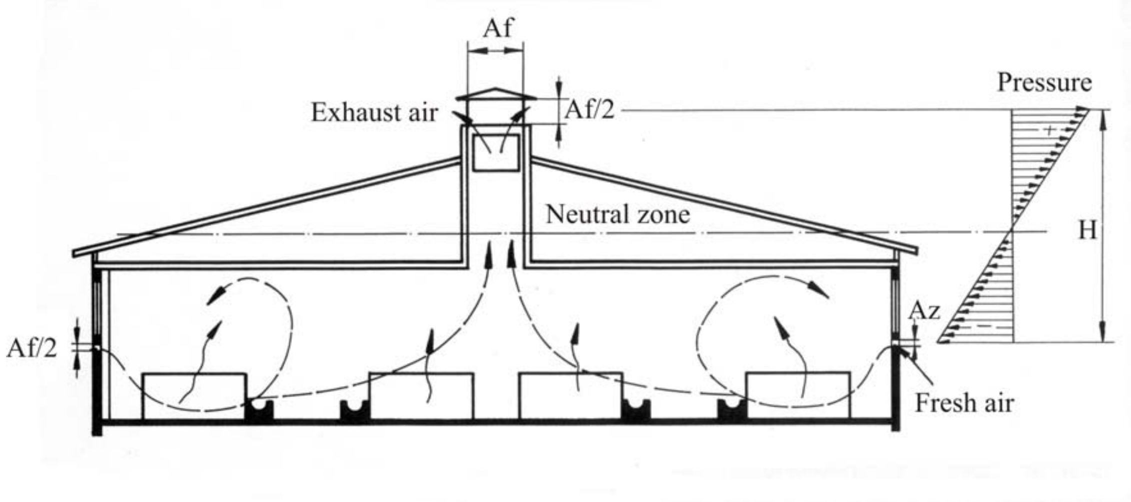


Figure 1. Schematic presentation of thermal lift in an animal house with shaft ventilation.

The pressure difference is proportional to the density difference (a result of the temperature difference between inside and outside air) and the height difference between air outlet and air inlet:

$$\Delta p_{th} = g\ H\ \Delta \rho \qquad (1)$$

Where Δp_{th} is the pressure difference, g is the gravity, H is the height difference between air outlet and air inlet and $\Delta \rho$ is the density difference between inside and outside air.

The air flow around the livestock building because of the outside wind effects pressure differences at the building (Figure 2).

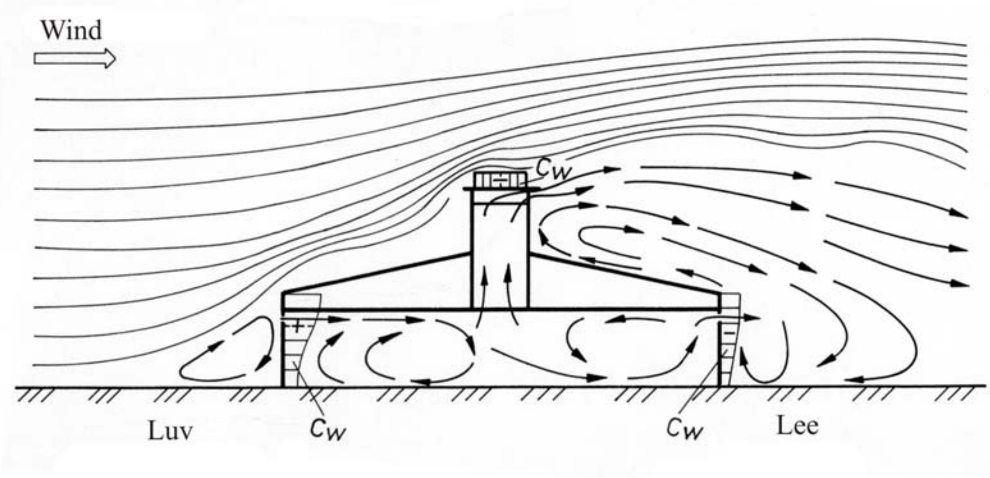


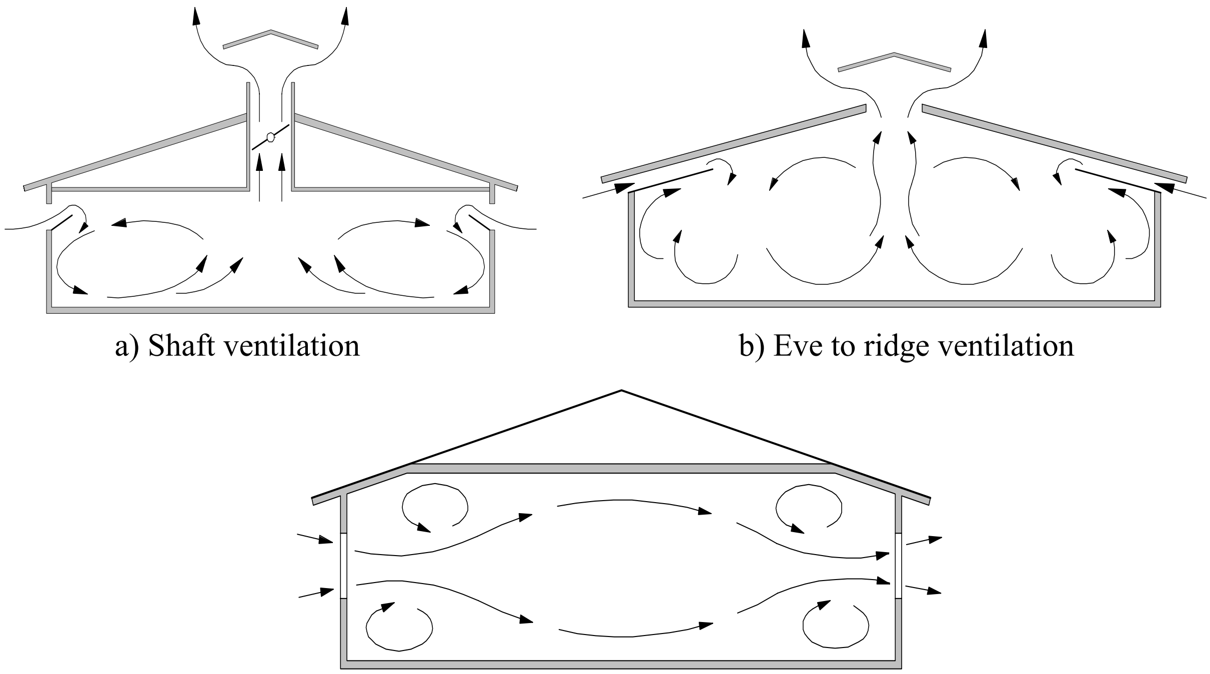
Figure 2. Schematic presentation of the wind effect on air flow trough the building.

The wind pressure results from the dynamic pressure of the wind can be written in the simplest way as the following:

$$p_W = c_W \frac{\rho}{2} w^2 \qquad (2)$$

Where p_W is the wind pressure, c_W is the coefficient of the wind pressure and w is the wind velocity. It is to consider that the coefficient of the wind pressure c_W depends on different influencing factors such as the wind profile, the location of the opening at the building and the shape of the opening. This coefficient must be determined by empirical methods.

Different types of natural ventilation are applied in farm buildings (Figure 3).



a) Shaft ventilation

b) Eve to ridge ventilation

c) Wind induced cross ventilation

Figure 3. Different types of natural ventilation used in animal houses

INVESTIGATED CATTLE BARNS

Investigated building 1

First measuring period (16 – 24 March, 2004):
Size: 74.4 m long and 34.2 m wide; room volume 19 731 m³
Number of animals: 288 dairy cows
Ventilation: natural ventilation; ridge opening; openings in side walls controlled by curtains; doors with adjustable windbreakers
Second measuring period (24 April – 04 May, 2006):
Size: (extended) 96.15 m long and 34.2 m wide; room volume 25 499 m³

Number of animals: 359 dairy cows (animal mass: 655.5 kg per animal; milk performance: 33.6 l/d per cow)

Ventilation: natural ventilation; ridge opening; openings in side walls controlled by curtains; doors with adjustable windbreakers; 3 additional ceiling fans

Investigated building 2

Size: 86.4 m long and 11.91 m wide; room volume 3 972 m³
Number of animals: 251 beef cattle
Ventilation: natural ventilation; shaft ventilation; openings in side walls (windows, which can be opened by hand); doors, which can be closed or opened by hand

Methods

Climate parameters

Run measurements for air temperature and humidity sensors connected with data logger. Wind outside is measured by ultrasonic anemometer and inside by thermistor sensor.

Air volume stream

- CO_2 – balance method [1]
- Especially during summer, when air change rates are high and cattle barns are ventilated naturally the small number of measuring locations and the small temporal resolution leads to higher measuring uncertainties. Therefore a special measuring system for Krypton 85 as tracer gas has been installed at ATB. With this system it is possible to measure the concentration of Krypton 85 in the barn at 40 locations in parallel. The shortest measuring interval per location is 1 second. The tracer gas is distributed in the building in thinned solution and as fast as possible. This is done manually, that means the dosing device is moved evenly across the entire surface of the livestock building. Then the tracer gas is mixed with the air of the livestock building by waving big cardboard fans. It is recommended during this time to close all openings. After that, the openings of the building must be opened quickly and the decay of the concentration of Krypton 85 is measured. The exponent of the decay function is the air change rate. In connection with the volume of the livestock building, the volume flow can be determined. This method is described in [2]. 18 measuring points (MP) are evenly distributed in 3.5 m height (each 3 MP in line across to the longitudinal axis of the building).

Concentrations

Run measurements for CO_2, NH_3, CH_4 and N_2O are realized by means of a Multigas-Monitor. Using a multiplexer it is possible to connect up to 12 measuring points. The time resolution drops with the increase of the number of measurement points and amount to approx. 12 minutes in case of 12 measurement points.

Emission mass flow

The emission stream from the livestock building is the product of both, the difference in concentration between emitted air and fresh air and the air flow rate.
One needs to take into consideration that both the concentration and the air volume stream vary permanently. That means the emission mass flow is temporal variable. To indicate the temporal run of the emission mass flow rate both parameters have to be measured the air volume stream and the concentration. The product will be calculated from both factors for each time step.

RESULTS

Investigated building 1

Climate parameters

The location of the investigated cow shed is in the north of Germany near the Baltic See. In this region there is more wind than in other parts of Germany. The air volume stream is mostly much higher than the minimum ventilation rate. Figure 4 shows the run of inside and outside temperature.

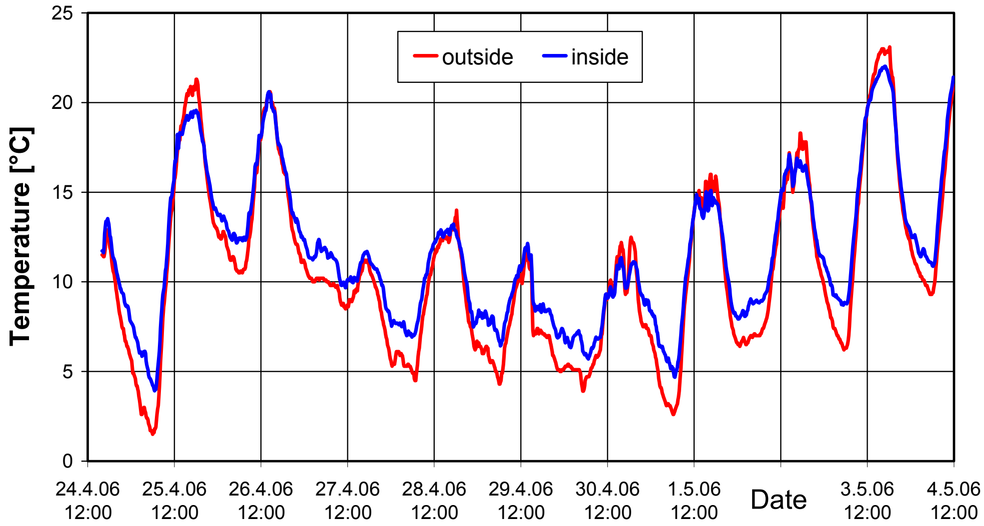


Figure 4: Run of inside and outside temperature during the second measuring period (24 April – 04 May 2006). Inside values are averaged over 4 MP.

The difference between inside and outside temperature is small and the inside temperature follows the variation of the outside temperature. The same pattern can also be seen for outside temperatures lower than 0°C. The air flow pattern and also the air velocity in the animal zone are strongly influenced by the outside wind situation (Figure 5).

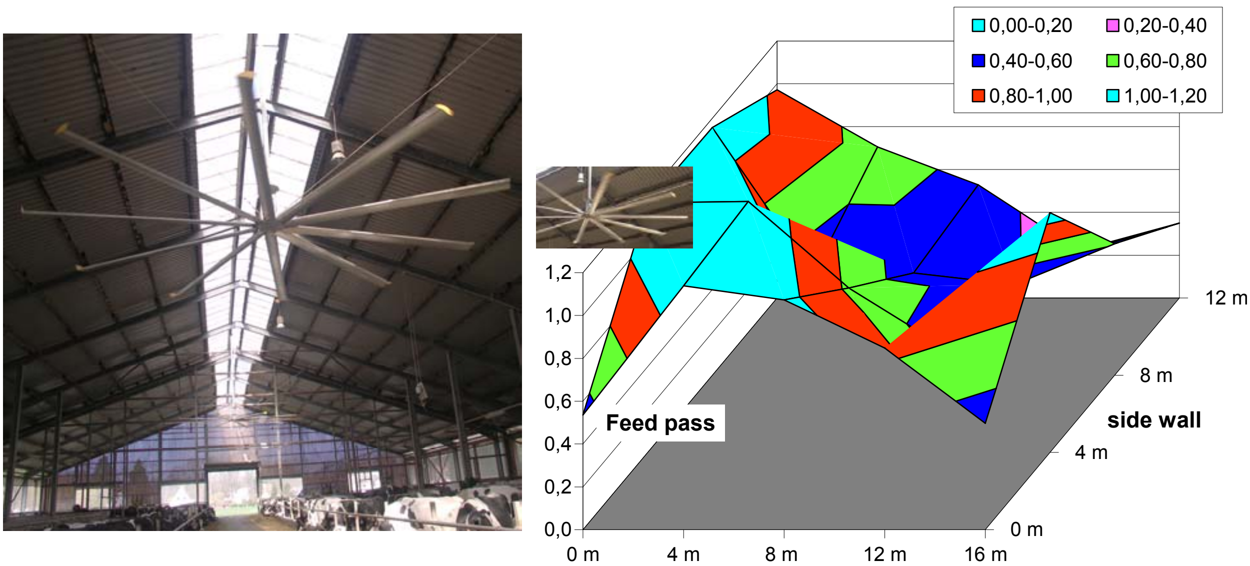


Figure 5: Ceiling fan (left) and air velocity field 1.2 m above the floor (04 May 2006). The velocity values are average values about 2 minutes

On 04 May 2006 the air velocity in the animal zone was measured with three different speed steps of the ceiling fan (speed step 1; 4.5; 9). Figure 5 shows the fan (left) and an example for speed step 4.5 (right).
Directly beneath the fan the air comes up and the velocity is low (0.5 m/s). In a distance to the fan between 3 m and 6 m the air velocity increases up to 1 m/s. In a bigger distance the velocity decreases again and near the side wall there are the highest values caused by the outside wind near the big openings in the side wall. In the German standard DIN 18910-1 an air velocity up to 0.6 m/s is permissible – but this standard regards only to closed livestock buildings.

Air volume stream

The volume stream is measured and calculated by CO_2-balance and the tracer gas method using Krypton 85 as tracer gas. The openings in the side walls and gable walls lead to a very high air exchange rate. In both measuring periods the air exchange rate (measured by Krypton 85) differed between 17.3 h^{-1} (341.346 m³/h => 1 185 m³/h per animal) and 89.2 h^{-1} (2 274 511 m³/h => 6 336 m³/h per animal). These air flow rates per animal are much higher than required in the standard DIN 18910-1 (minimum air flow rate = 84,3 m³/h per cow and maximum air flow rate = 333 m³/h per cow). Figure 6 shows results of air stream measurement by CO_2-balance and by tracer gas method using Krypton 85 as tracer gas. The CO_2 analyser cannot follow the fast fluctuation of air change, but in the most cases exist a good correspondence between both methods. The very high measured value by the Krypton method (11:35 – 11:40) results from a bad mixing between tracer gas and room air. The Krypton gas was dosed with open walls and doors, so that tracer gas already escaped from the room during dosing time.

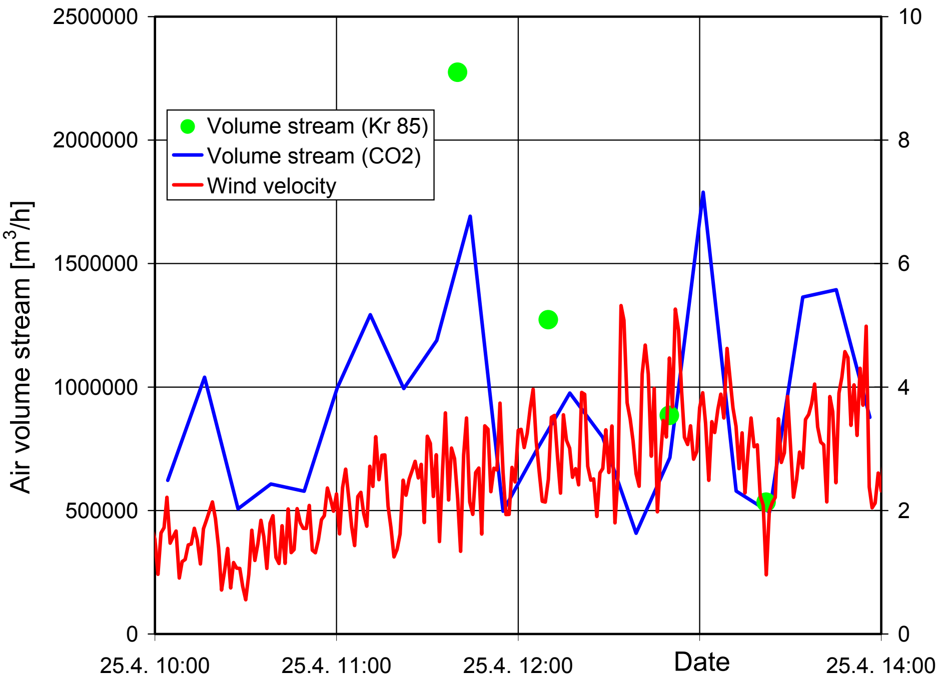


Figure 6: Air volume stream measured by CO_2-balance and air exchange rates determined by the decay method (Krypton 85) in a short time period.

The comparison of both measuring methods shows that the application of CO_2-balance can be used for assessment of air volume stream trough naturally ventilated large animal houses also for long periods of time. The correlation between air volume stream and wind velocity over both measuring periods shows the strong influence of wind velocity on the air change rate.

Concentrations

The high air volume stream through the cow shed leads to low gas concentrations. Sometimes the inside concentration is close to the outside concentration. In both measuring periods the CO_2 concentration is most of the time lower than 1000 mg/m^3 and the average value for NH_3 is 1.10 mg/m^3 in the first period and 1.16 mg/m^3 in the second period. For both gases peaks have been observed. These peaks appear especially during feeding time. Reasons for that are the higher animal activity and also the emission from the fresh feed. Investigations in the "emission laboratory" of the ATB have shown that the fresh feed emits CO_2 and NH_3.

Emission mass flow

The emission mass flow is calculated with the measured NH_3 concentration and by the determined air volume stream (CO_2 balance) during the whole periods. In spite of low NH_3 concentration a relatively high emission stream results – for the first period 2.6 g/(h LU) and for the second period 1.4 g/(h LU).

Investigated building 2

Climate parameters

The measured climate parameters are comparable to building 1. The maximum values inside and outside are nearly the same but during night time the temperature inside the house decreases not so deep than the outside temperature.

Air volume stream

The measured air volume stream is shown in Figure 7.

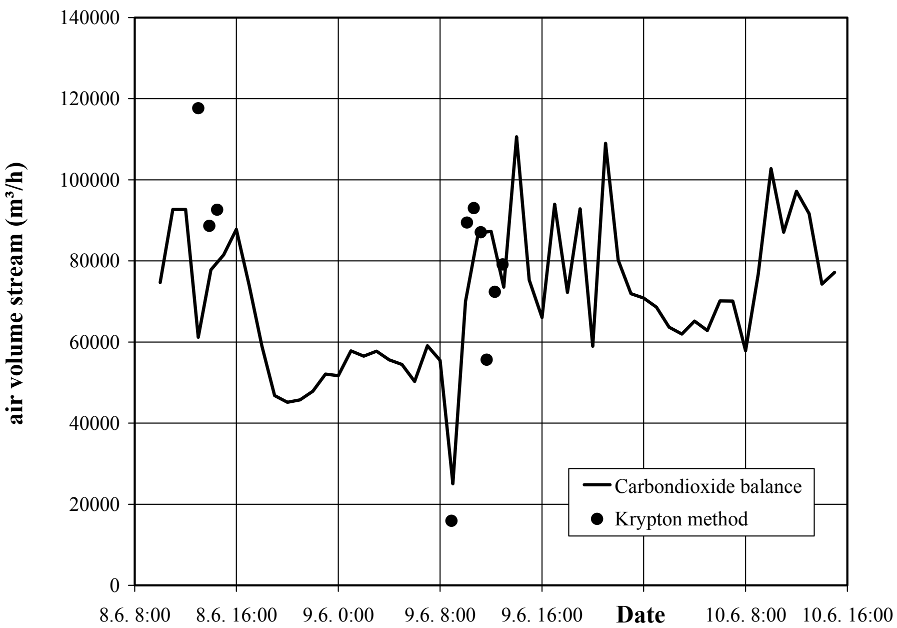


Figure 7: Air volume stream in a beef cattle house – determined by carbon dioxide balance and by Krypton method (June 2004)

Concentrations

During feeding time in the morning the ammonia concentration increases dramatically up to 25 mg/m^3. During the day the concentration decreases to 2 mg/m^3 in the night time.

Emission mass flow

The emission mass flow is calculated with the measured NH_3 concentration and by the determined air volume stream (CO_2 balance) during the measured period. The high NH_3 concentration in the morning lead to a relatively high emission stream of 1.4 g/(h LU).

CONCLUSION

- In naturally ventilated livestock buildings the air volume stream through the building is strongly influenced by the size of the openings in the building and by the outside wind velocity.
- The inside temperature follows the outside temperature, but the oscillation of the inside temperature is a little bit damped.
- By means of ceiling fans the air movement in the animal zone can be increased especially during summer to reduce the heat stress.
- The Krypton method is applicable to determine air exchange rates in naturally ventilated livestock buildings - for short term measurements – high air exchange rates in large buildings are measurable.
- The CO_2-balance method is also applicable for natural ventilation to assess the air volume stream for longer periods of time.
- In naturally ventilated livestock buildings the gas concentrations are low but in spite of high air volume streams the emission streams are relatively high.

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13 June 2007 at 15:00 - 16:30

A10
Maintenance and improvement of ventilation systems

Modern Concepts for Refurbishment of Ventilation and Air- Conditioning Systems

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SUMMARY

A research project has been carried out to develop new methods and technical solutions to refurbish ventilation and air-conditioning systems of office and commercial buildings built during 1970`and 1980`. More than twenty refurbishment concepts have been developed, which deal with all parts of ventilation and air-conditioning system (air-intake, filtering, cleaning, tightening and balancing of ductwork, noise control, air distribution, demand based ventilation, night cooling, adding mechanical cooling etc) and which can be carried out one by one or all together depending of the actual needs. Special attention has been paid for the installation work on site to minimize the disturbance and nuisance to users of the building. Therefore dustless and noiseless installation methods have been developed. For each concepts design criterias and installation rules are given. Finally the commissioning procedure for each concept has been discribed. This project has been carried out in Finland, where building owners, manfacturers, contractors and designers have been working together with researches to achieve means to refurbish old buildings to meet todays reguirements of modern indoor air climate.

INTRODUCTION

There exist a great number of office, commercial and public buildings, built during 1970` and 1980`, which do not any more meet the users and building owners requirement concerning indoor air quality and climate. Those buildings are not anymore competitive in the market, where modern buildings with latest ventilation and air-conditioning technology are available.

These ventilation and air-conditioning systems shall be refurbished without disturbing the users. This project deals with public and private offices, commercial buildings and schools built during 1970` and 1980` in Finland. In this project refurbishment concepts to improve performance of ventilation systems has been developed. An integral part of the work has been to develop a procedure in which way and means refurbishment of ventilation and air-conditioning systems can be carried out in actual refurbishment projects.

METHODS

Typical ventilation systems and their problems

Analyses of ventilation systems, built during those two decades have shown that they have many common, recognized technical features, which are not any more acceptable by the building users and owners. Additionally one can recognize the same complains, which users raise continuously, when indoor air quality is concern. The main factors in this field are:

- poor indoor air quality

- high noise level
- draft
- too high room temperature
- lack of individual temperature control
- unbalanced ventilation system
- unclean ventilation system
- high energy consumption

Addition to problems with ventilation and air-conditioning these buildings have other features, which were not considered here. They are such as moisture damages, used hazardous materials etc. which problems showed out to be often bigger than inadequate ventilation and temperature control.

All these factors, combined with technical problems in service and maintenance has forced the building owners to reconsider the value of the building and take necessary steps to update the ventilation and air conditioning system to meet their user's needs.

<u>Development of refurbishment concepts and procedure to carry refurbishment project</u>

Refurbishment concepts to improve indoor air conditions were selected based on the analysis of existing buildings. Concept development varied a lot, depending on concept and its features. Anyhow, all concepts were finally tested in actual construction conditions. Procedure to carry out refurbishment project was developed simultaneously with the concepts but concentrating more design, installation and management features.

Research approach to determine the concepts have been:

- design
- studies and models
- laboratory tests
- field studies
- modeling and simulation
- prototypes
- pilot projects

RESULTS

<u>Refurbishment concepts of ventilation and air-conditioning systems</u>

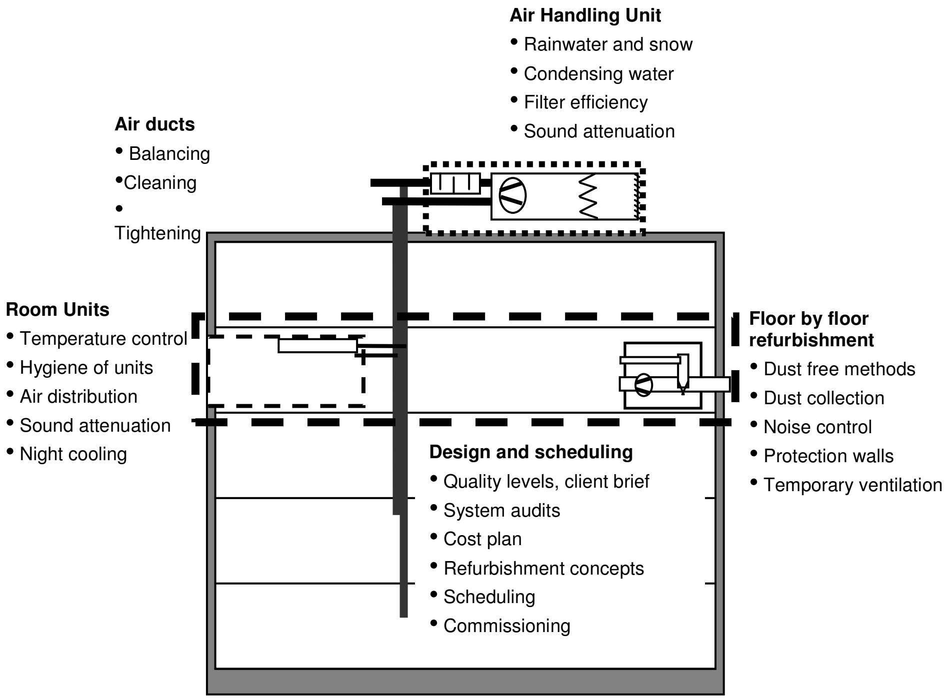

Figure 1. Refurbishment concepts of ventilation and air-conditioning systems.

The developed concepts are:

1. Concepts related to ventilation and air conditioning systems such as:
    - Balancing of air flows
    - Decreasing the noise levels
    - Increasing the air flow rates
    - Demand based ventilation
    - Adding the mechanical cooling
    - Night ventilation and cooling
2. Concepts based on spatial needs in offices and schools such as:
    - Air distribution in an office room
    - Air distribution in open office
    - Air distribution in classroom concepts
    - The hygiene of local cooling units
3. Concepts based on improvements in mechanical rooms such as:
    - Prevention of snow to enter the air handling units to keep the filters dry
    - Improvement of filtering efficiency
    - Cooling coil draining solutions
4. Concepts related to ducting such as:
    - Tightening of sheet medal air ducts
    - Cleaning of sheet metal ducts
    - Cleaning, tightening and layering of hollow core concrete slabs used as air ducts
    - Layering of noise attenuators
5. Concepts related to work on site such as:
    - Minimizing of noise and silent working methods

- Minimizing of dust generation and dustless working methods
- Temporary ventilation systems

Procedure to carry out the refurbishment concepts

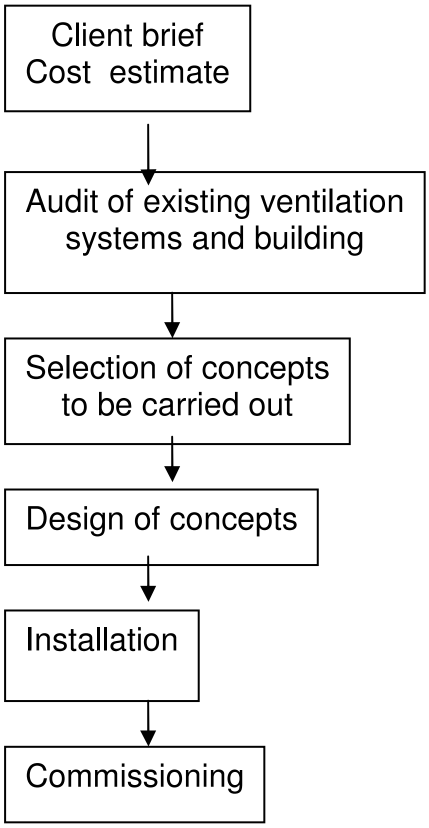

Figure 2. Procedure to carry out the refurbishment concepts of ventilation and air-conditioning systems.

The developed procedure has the following steps:

- client brief and cost estimate
- audit of existing ventilation system
- selection of concepts to be carried out
- design of concepts
- contract to carry out of concepts
- commissioning
- use of the building and systems

Client brief is the phase of the project, where client oriented design analysis is made concerning not only indoor air conditions and other performance of building services systems, but also such features as flexibility, durability, ecology, energy and security of the building. Cost estimate is essential part of client brief.

Audit of existing ventilation system is necessary to determine before starting the refurbishment work:

- to find out the existing condition of ventilation system
- to make sure, that the design targets can be achieved
- to make sure, that the actual work on site can be done

Selection of concept to bee carried out in that specific case is based on feasibility study, client brief and audit of ventilation system. Concepts can be divided into three groups based on the purpose of the refurbishment.

- To remove the nuisance of the user, such as noise, draft, poor air quality, too high temperature etc.
- Increase of indoor air quality, such as from satisfactory level to good level etc.
- Technical improvement, such as changing the old component, improving the energy efficiency etc.

Concepts most often deal with multipurpose targets, so that the concentration is to select a total improvement instead of individual improvement.

Design principles of each concept are given in the design manual. Design manual includes results of the researchers´ work, which have been done in the project. Additionally good design and construction practices related to the concepts have been collected into the manual.

The design manual includes the following items:

- Target values for indoor air climate
- Other targets, such as flexibility levels, environmental requirements, cost targets etc.
- Design principles for each concept (dimensioning, calculation methods, materials, working methods etc.)
- Instructions, how to handle that part of ventilation system, which is not covered by the concept itself
- Construction documents for the bidding (specification, drawings etc.)
- Quality control documents
- Maintenance documents

In construction phase special attention is paid for installation methods, so that minimum disturbance to the users of the building will take place. Particularly dustless and noiseless working methods are used.

Related to each refurbishment concept instructions are given in construction documents, which are the technical requirements and working methods to carry out the concept.

Commissioning and handling over procedure consists of the following actions, which will be carried out by the supervisor with close co-operation with the contractor:

- Approval of materials, products and components provided by the contractor
- Checking of installations
- Functional tests
- Test runs
- Indoor air quality and other technical measurement (temperature, air velocities, noise, air-flows, tightness, cleanliness etc.)

A checklist has been prepared for each refurbishment concept to clarify, which are the essential technical features for the concept. At the end of the refurbishment project both design and commissioning documents will be attached to buildings service and maintenance documentation.

DISCUSSION

As the result of this research and development project, twenty concepts for refurbishment of ventilation systems have been developed. These concepts cover several critical areas of ventilation systems.

Partial refurbishment requires effective use of existing ductwork. The ductwork must meet modern requirements for cleanliness, air tightness and adjustability.

Noise from fans and terminal devices is one of the most common complaints. Therefore several concepts to eliminate this problem were developed.

Improving thermal comfort is one essential step which is needed to take when heading towards updated indoor air quality. This project focused on means to add mechanical cooling to existing building, for nighttime free cooling and air distribution solutions for draughtless indoor conditions.

Refurbishment of ventilation systems while building is in use means that new methods, tools and systems has to be developed to minimize disturbance from noise and dust. Additionally partitioning systems and temporary ventilation is needed guarantee safe and healthy conditions for the users also during the refurbishment work.

Cleanliness of supply air is an essential performance criterion for good ventilation system. In order to get rid of old systems hygiene problems, concepts concerning filtering efficiency, snow and water penetration to air-handling units and local filtering systems were developed.

The results of the project have been collected to design manual, which has three parts:

1. Instructions to carry out the refurbishment project for building owners and end users
2. Design, installation and commissioning rules to carry out of refurbishment project
3. Researchers reports of refurbishment concepts

ACKNOWLEDGEMENT

This development project “Modern Concepts for Refurbishment of Ventilation systems” was financed by Finnish building services industry, contractors and consultants, building owners and Technology Development Centre in Finland (Tekes). Without their financial support the results of this project would not have been realized.

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Protecting the HVAC System During Construction: An Industry Standard of Care for Contractors

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SUMMARY

The relationship between construction activities and an increased risk of microbial contamination and degradation of air quality in new building construction and renovation projects was studied. Poorly designed or installed ventilation systems and inferior workmanship during construction were determined to have an adverse impact on the performance of the building ventilation system. This presentation will provide guidelines to protect HVAC system components during building construction and renovation projects and for building commissioning procedures for building ventilation systems. It also underscores the importance of educating the building owner on proper operation and maintenance of building mechanical systems as part of the building commissioning and transfer process.

INTRODUCTION

Heating, ventilation and air conditioning (HVAC) system components often collect significant amounts of debris and particulates during construction activities within a building. A faulty or contaminated HVAC system can lead to poor humidity and temperature control and cause degradation of indoor air in the building. Newly installed HVAC systems or those in buildings undergoing renovation must be protected before the system is permitted to operate and should be verified to be clean [1]. Protecting HVAC system components from water, dust and debris during construction activities is the major control method to prevent degradation of indoor air quality and microbial contamination. Building commissioning procedures should include quality assurance testing of the ventilation system, prior to building transfer to the owner, to ensure that it is properly tested, and balanced and functioning according to building design parameters. A systematic approach is required to address reported ventilation problems during the warranty period. This presentation describes an industrial hygienist's role in building construction and renovation, from project planning through testing, and the contribution an industrial hygienist can make to protect HVAC systems from airborne contaminants during construction.

METHODS

The focus of this presentation is on protecting building HVAC systems. Major considerations to prevent adverse impacts on the performance of the building ventilation system are presented by the author based on microbial investigations and case studies involving construction of commercial and residential buildings located throughout North America and the Pacific Rim.

The HVAC system includes the interior surfaces of a building's air distribution system that services conditioned indoor spaces and/or occupied areas. This includes the entire building ventilation air conveyance system. The return air grilles; return air ducts to the air handler unit (AHU); variable air volume boxes; interior surfaces of the induction, convection, fan-coil units and the AHU; and mixing boxes, heat exchangers, condensate drain pans, humidifiers and dehumidifiers, supply air ducts, fans, fan housings, fan blades, air wash systems, spray drift eliminators, turning vanes, filters, filter housings, reheat coils, ductwork, and supply diffusers are all considered part of the HVAC system. Protecting the integrity of the HVAC system in construction starts with proper building and ventilation system design. Proper building design includes choice of proper building materials; specifications for building construction with detailed attention to critical tasks such as waterproofing, vapor barriers and HVAC systems; procedures for reviewing plans and specifications with explicit consideration of water infiltration and potential mold-related exposure from construction materials, methods, and HVAC system requirements; and procedures for reporting building design weaknesses that can cause water intrusion during construction.

Moisture control in building ventilation design requires diligent engineering. The temperature and relative humidity in climate-controlled areas of the building should be designed within a range considered both comfortable and healthy according to consensus guidelines[2]. Design of the HVAC system, natural airflow, and temperature fluctuations can also help to control moisture condensation and humidity on jobsites. Proper design starts with appropriate material selection. Some materials are more amendable to cleaning during and after construction. For example, unlined galvanized sheet metal and borosilicate glass ducts are corrosion-resistant and cleanable, compared to other duct material, such as fiberglass ductboard and flex duct. Flex duct is extremely difficult to clean because it contains a tremendous amount of surface area inherent in its expandable accordion design. Anti-microbial silver ion-based paints are available to coat sheet metal to minimize mold growth in ductwork[3].

Contractors should provide detailed shop drawings and specifications for the building envelope, including exterior cladding, roofs, windows, walls, doors, and interfaces between the building features and the HVAC system components. For example, Figure 1 shows an exterior wall cavity prior to closing on a multi-story residential construction site. The wall cavity is clean, with no organic debris and minimal dust. This is especially important in places where you have condensation buildup; for example, around the package terminal air conditioning (PTAC) unit housing. PTAC units should be snugly fitted and sealed in the wall with weatherproof silicone sealant, and have flashing and drainage to the exterior for proper condensate pan overflow.

Figure 1. PTAC unit housing in multi-story residential building.

Strategies to protect HVAC systems during construction are broadly categorized into proper ventilation system design, quality construction methods and work practices, and adequate building commissioning procedures. Design should include easy access to HVAC components, which is essential for proper maintenance and inspection of HVAC systems. Access doors or panels must be provided for all components including ductwork, plenum walls, drain pans, upstream and downstream areas of filters and fan coil units, and terminal boxes [4]. Construction work practices must include proper sequencing of work to protect work-in-progress and to ensure that porous materials, such as fiberglass-lined duct, are installed after the roof structures are dried-in and the building envelope is intact. The time between material delivery and installation should be minimal. Contractors should place emphasis on protection of building materials delivered to and stored at the jobsite from water, dust and debris, and mold growth. Figure 2 demonstrates poor work practices as evidenced by water damage to stockpiled fiberglass duct wrap stored outdoors and exposed to ambient conditions and inclement weather.

Figure 2. Water-damaged fiberglass duct wrap stored outdoors.

Pre-construction planning should include implementation of weather protection plans in project schedules. For example, tenting of the building envelopes using polyethylene sheeting in building construction is a control strategy routinely used to protect exterior wall assemblies during damp, rainy seasons in the U.S. Pacific Northwest.

Because water availability is one of the most critical factors controlling microbial amplification in indoor environments, prompt correction and elimination of water intrusion/ moisture sources are imperative. Protecting HVAC system materials and components, and work-in-progress from water, moisture, construction dust and debris is critical to prevent microbial growth and degradation of indoor air. Figure 3 shows proper storage of an AHU in a large commercial building prior to installation, i.e., elevated above the concrete slab floor to protect from water intrusion and covered to shield from construction dust and debris. Figure 4 shows a 30-centimeter diameter branch entry opening to a duct near a supply drop, covered with black plastic, to protect the ductwork interior during construction. Equipment and material movement and a lack of adequate space on construction jobsites can also result in physical damage to HVAC system components. Stored materials and work-in-progress on HVAC systems should be inspected on a daily basis and safeguarded from physical damage.

Figure 3. Proper AHU storage – pre-installation.

Figure 4. Protected branch entry duct opening.

Proper location of outdoor air intakes (OAIs) and cooling towers are of special concern. The location of the OAI can significantly affect the nature of outdoor bioaerosols that enter a building. Rooftop OAIs are especially vulnerable to bioaerosol sources such as cooling towers, sanitary vents, building exhausts, and standing water. Cooling towers located close to or directly upwind of OAIs are potential bioaerosol sources. To minimize entrainment of cooling tower mist, it is recommended that OAIs and other building openings be located at least 7.5 meters and preferably 15 meters upwind of, and horizontally separated from, cooling towers [5]. Even OAI placement at this distance does not ensure that aerosols in the exhaust air / drift will not be entrained. Airflow patterns and prevailing wind direction are important factors affecting drift entrainment. Install drift eliminators on cooling towers to intercept water droplets where air is discharged to limit dissemination of bacteria, such as *Legionella*, from water-cooled heat transfer systems. The cooling tower shown in Figure 5 is constructed of materials that can be readily disinfected and is located in an area that is remote from building air intakes, and easy to inspect and maintain.

Figure 5. Cooling tower installation.

Building renovations performed without proper planning and design can also result in degradation of indoor air caused by moisture problems and subsequent mold growth. An indoor air quality study in the United States in 2001 found that some indoor air quality problems in office buildings are the result of contractor errors and omissions including inappropriate balancing and reassessment of HVAC systems during building renovation or modification [6].

Contractors should reduce exposure to liability by contract review, proactive risk transfer methods, and independent third-party construction oversight. Contracts should be carefully reviewed to verify that they: adequately protect the contractor from construction defects or negligence of subcontractors; specifically address the responsibility for repairs; and contain enforceable language providing for indemnification of damages. Contractors should document activities at certain critical stages of construction to establish proof of proper construction practices. Superintendents should take photographs of work-in-progress to document conditions during construction, for example, prior to closing ceiling plenums and wall cavities and covering work with interior finishes, furnishings and contents. Damaged materials should be properly marked and replaced during construction, as shown in Figure 6. Comprehensive field notes and inspection checklists should be retained on file for the project. Contractors should require subcontractors to provide documentation that materials and HVAC system components are installed per manufacturer's instructions and engineering design drawings. All of this documentation will be valuable in mitigating damages and associated costs. In addition, third-party consultants should be retained to perform independent peer review of the HVAC system design and to make inspections during construction. They should document HVAC system commissioning procedures and inspect the completed project prior to transfer to the owner.

Figure 6. Water-damaged duct riser – marked for removal.

DISCUSSION

Post-construction activities should include final cleaning of all interior finishes and functional testing of building mechanical systems. Protect the HVAC system from dust, dirt and water during construction and make certain it is clean afterwards. Cleaning is the final defense in managing indoor environmental quality and preventing microbial contamination. After the areas of source management and building operations have been addressed, cleaning is still necessary. The importance of this was demonstrated in a year-long study of cleaning effectiveness in a multi-use building without evident infrastructure problems. The routine use of high efficiency vacuum cleaners, damp dusting, and improved cleaning products – particularly in high traffic areas – along with attention to leaks and spills, resulted in meaningful decreases in particulate and microbial contamination [7].

Building commissioning procedures of the completed project should include quality assurance testing of the HVAC system to ensure that it is properly tested and balanced and functioning according to building design parameters. Procedures and methodologies for documenting and verifying the performance of HVAC systems according to design are published by the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE) [8]. Replace pre-filters and filters on all air moving equipment in the HVAC system after construction is completed. Filters should be of the highest grade compatible with the HVAC system and the air handler fan – with a 50% to 70% dust spot efficiency rating. Ensure that complex computer-controlled systems for building ventilation have been properly programmed. Operate the HVAC system for 7 to 14 days to stabilize its operation prior to building occupancy.

In addition to these guidelines, testing the HVAC system for acceptance by the owner may include a microbial risk assessment performed on various HVAC system components most susceptible to contamination during construction – including supply air ductwork and plenums, heat exchangers, air washers and humidification devices, AHUs, filters, drain pans, and outdoor air intakes. The assessment should always be performed by a qualified environmental health professional, such as a Certified Industrial Hygienist (CIH), with experience in potential health implications of indoor contaminants, including microbial growth and a familiarity with commercial and residential HVAC systems. If needed, based upon the results of the microbial assessment, remedial action should be taken prior to occupancy of the building. The risk assessment should include a visual inspection of HVAC system components to ensure that they are clean and dry – with no signs of water intrusion, physical damage, or mold growth – and microbial sampling of a representative portion of the building and HVAC units. Procedures to verify cleanliness of the HVAC system proposed by the author should include:

- A visual inspection should be conducted to make certain HVAC system components are clean after construction with no residual dust or visible contamination. Test and balance the system per industry standards. Observations of airflow and pressure differentials should be made after system startup.

- Fungal spore samples should be collected pre-HVAC system operation. The system should then be turned on for 60 to 120 minutes to stabilize its operation, followed by the collection of post-HVAC system fungal spore samples from several random areas of the building. Pre-HVAC and post-HVAC samples should also be collected in areas of the building where the highest building occupant load or population density is expected. Air sampling is performed using a calibrated air sampling pump operating at 28.3 LPM. Fungal spore samples should be collected from the ambient indoor air and the outdoor air intake. One sample should be collected from the ambient indoor room air; another at a supply air diffuser; and another near the outdoor air intake. An additional outdoor sample should be taken at ground level outside the building for reference control purposes. Each sample should be collected for two minutes. The microbial samples should be analyzed by an Environmental Microbiology Accredited Laboratory using cultured methods or Quantitative Polymerase Chain Reaction (Q-PCR) analysis. Although more costly, Q-PCR analysis appears to be the most likely technology to be used in the future because it provides very rapid turnaround and greater accuracy, sensitivity, and precision in identifying microbial species present; but conventional techniques are also acceptable. Microbial air samples are interpreted by comparing the types, distribution and levels of fungal spores found in pre-HVAC system operation samples to post-HVAC system operation samples and to outdoor control areas. General guidelines for interpretation of fungal spore data specify that indoor levels of mold spores should be less than outdoors. The biodiversity of fungal spores indoors should also be similar to outdoors. The levels of mold spores and fungal genre in the post-HVAC samples should be less than and similar to those in the pre-HVAC samples.

- Sampling of cooling towers and domestic water supplies for the presence of *L. pneumophila* and other waterborne bacterial pathogens is also recommended. The location of the OAIs and air pathways within the building should be identified and evaluated to prevent aerosolization of *Legionella*-containing water droplets in the building. Water samples should be collected using sterile sampling containers with Sodium Thiosulfate preservative provided by an AIHA-accredited laboratory. Samples should be analyzed for the presence of *Legionella* using a combination of culture analysis and Deoxyribonucleic Acid (DNA) sequencing.

Third-party inspections should be performed by qualified professionals to review shop drawings and to inspect the integrity of the building envelope during and after construction – to identify any defects that could cause potential problems with water intrusion, moisture accumulation, mold growth or structural integrity. A systematic approach for addressing reported problems and customer complaints during the warranty period should be developed to help prevent small problems from becoming big. Educate the building owner on proper operation and maintenance of all building systems. Train the building owner's property management and building engineers/maintenance staff on preventive maintenance of the HVAC system; humidity control; ventilation of moisture producing processes; inspections and maintenance of the building envelope; and operation of the plumbing, heating, fire sprinkler and mechanical systems, including outside irrigation and landscaping. Train building maintenance staff in mold prevention methods for the new building after completion. Prompt attention to water intrusion should be emphasized with the owner and maintenance staff to prevent and minimize costly structural damage and mold growth, and to increase the life span of the building.

Air quality can be enhanced by properly designed, operated and maintained building ventilation systems. Fanger suggested that air quality be defined by the extent to which human requirements are met [9]. Sources of contaminants and the effectiveness of control measures, including HVAC systems, should continue to be evaluated to assess indoor air quality after building occupancy.

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Cleaning initiation criteria for heating, ventilation and air conditioning (HVAC) systems in non-industrial buildings

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SUMMARY

The objectives of this study are to reproduce in the laboratory different levels of cleanliness in non-porous ducts of HVAC systems, to compare a new method for sampling surface dust in ducts with those methods cited in the literature, to compare the numerical evaluation method to the visual method, and to propose objective cleaning initiation criteria. For each of the simulated cleanliness conditions, a committee of specialists did a visual assessment based on a scale of 3, where level 1 meant normal, 2 meant above normal, and 3 meant serious. According to these assessments, the established initiation criteria correspond to 0.2 g/m² for the NADCA method, 0.3 g/m² for the ASPEC method, and 0,6 g/m² for the new method. These criteria are significantly different ($p \leq 0.05$). Any of the surface sampling methods can be used, insofar as the corresponding initiation criterion is used.

INTRODUCTION

Ventilation systems can be potential sources of pollutants due to the accumulation of dust in their air systems. Building managers have to deal with a wide range of proposals from specialized cleaning companies and have difficulty arriving at a decision because there is no recognized or standardized method for assessing a system's dust contamination. Ventilation systems must therefore be maintained under optimal cleanliness conditions. For the optimal maintenance of facilities, it is therefore important to be able to measure the amount of dust that has been deposited in ventilation networks. In all cases, an objective diagnosis avoids unnecessary network cleaning, or if decontamination is required, allows the cleaning methods to be chosen (1).

In the United States and Canada, the initiation of air system cleaning is currently based on visual inspection (2,3). However, these criteria are subjective and rather impractical for major cleaning work. In 2005, the American National Air Duct Cleaner Association (NADCA) published criteria for cleanliness acceptance after cleaning. However, these criteria are inadequate if you want to know when to start cleaning HVAC system networks (4). As well, these criteria can only be applied on rigid and non-porous surfaces, meaning smooth surfaces (e.g., metallic surfaces).

The Association pour la prévention et l'étude de la contamination (ASPEC, association for the prevention and study of contamination) in France, has published a guide on methods for keeping non-porous air systems for clean rooms and related controlled environments clean (1). In this guide, the initiation criteria for tertiary environments (office buildings) and the methods used are reported for different countries. Table 1 presents these criteria.

Table 1. Criteria for initiating cleaning of non-porous ducts (1)

Country	Cleaning initiation criterion based on surface density (g/m^2)	Cleaning initiation criterion based on thickness (µm)	Post-cleaning acceptance criterion (g/m^2)	Sampling method
United States (NADCA 2005)	-	-	0.075	Surface sampling on membrane at 15 L/min (open cassette)
Great Britain (1998)	Blowing : 1 Exhaust: 6	Blowing : 60 Extraction: 180	0.1	Surface sampling on membrane for at 15 L/min
Finland (1995)	Blowing 2 Exhaust: 5	-	-	Surface sampling on membrane at 15 L/min (sampling tube)
France (2004)	Blowing : 0.4 Exhaust: 6	-	0.1	Surface sampling on membrane at 15 L/min (sampling tube)

This table shows that the criteria are accompanied by different dust sampling methods, consequently making comparisons difficult. According to ASPEC, these methods can be applied only to rigid and non-porous ducts of sufficient dimensions, i.e., larger than 30 cm in diameter for round components; in addition, the ducts must be horizontal; and finally, the walls must be dry (1). Sampling must be done on a layer of dust distributed on the bottom surface, and not on an accumulation of dust (1). Furthermore, the sampling methods have some deficiencies, mainly the absorption of moisture from the air by the cellulose ester membranes, and the adhesion of dust on the walls of the cassettes and sampling tubes.

One method that would eliminate these two problems would involve weighing a complete sampling cassette such as the IOM cassette (SKC Inc. Eighty Four, PA, USA) equipped with a cellulose ester membrane. However, this surface sampling method has not yet been evaluated. We will compare this cassette with the sampling systems mentioned in the literature in order to choose the most accurate method.

The objective of this activity is therefore to propose a methodology for measuring the dust contamination of ducts in order to compare the numerical value to the value obtained from the visual evaluation corresponding to the limit value that determines the need for cleaning.

METHODOLOGY

The steps in this project are: to develop a dust contamination chamber and technique, to choose the most appropriate surface sampling method for dust, and to determine a limit concentration for initiating cleaning.

Development of the dust contamination chamber

A preliminary study undertaken during the summer of 2004 at the IRSST led to the development of a chamber for the laboratory simulation of dust contamination of ducts (5). Figure 1 presents a diagram of this chamber.

This chamber was designed with smooth and non-porous surfaces. It is also equipped with a PALAS RBG 1000 dust generator. The standard dust used is the dust recommended by ASHRAE (5). It consists of (6):

- 72% fine test dust (Arizona road dust).
- 23% carbon powder (Molocco black).
- 5% No. 7 cotton linters.

The work undertaken by the IRSST in 2004 has shown that with this chamber, known and uniform concentrations of dusts can be obtained (5).

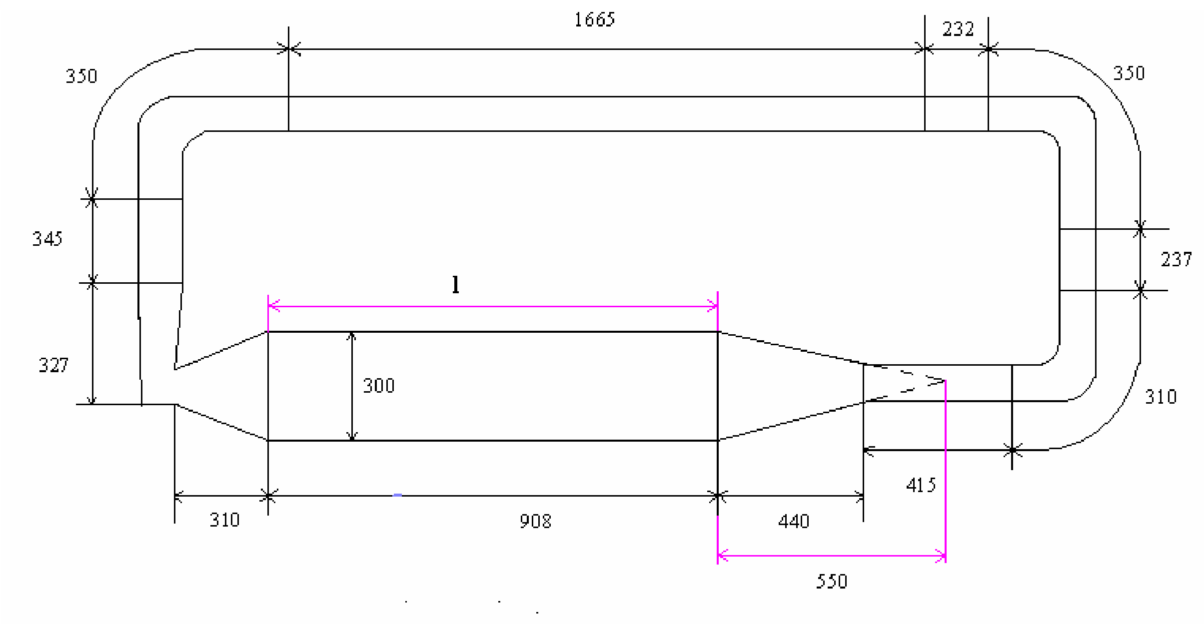

Figure 1. Diagram of the actual dust chamber (in mm)

However, to optimize this system, different improvements will have to be made in order to improve its performance. The following modifications will be made to the system (7,8):

- maximum extension of the sections
- elimination of asperities and interior joints
- replacement of the fan in order to eliminate leaks

Choice of sampling methods

Twelve control substrates (aluminum filters 47 mm in diameter, MSP Corp., Shoreview, MN, USA) uniformly distributed over the entire surface of the base were used for each dust contamination test. Figure 2 presents their location on the plate.

Three sampling methods were chosen for the dust contamination chamber. The first principle was the NADCA principle (4). It involves vaccuming dust over a predefined 100 cm² duct surface from a template 0.381 mm thick and of collecting it on a preweighed 0.8 µm cellulose ester membrane in an open cassette 37 mm in diameter (SKC Inc. Eighty Four, PA, USA), in order to determine the difference in surface density (Figure 3).

Figure 2. Distribution of control substrates

Figure 3. NADCA method

The second sampling method was the ASPEC method (1). It consists of a 0.8-µm pore size cellulose ester membrane in a 37-mm closed cassette (SKC Inc. Eighty Four, PA, USA) connected to a beveled tube. The aspirated duct surface is 50 cm^2. Figure 4 presents this method.

The third method is the method recommended by the IRSST. It uses an IOM cassette (SKC Inc. Eighty Four, PA, USA). In the latter, the entire 25-mm-diameter cassette including a 0.8-µm pore size polyvinyl chloride membrane is weighed (SKC Inc. Eighty Four, PA, USA). This avoids underestimations of weight. To be able to compare this method to the NADCA method, the cassette is slipped onto a template with a distance of 1.5 mm between the sampling cassette and the aspirated surface. Figure 5 presents this latter method. All the sampling flows were 15 L/min.

Figure 4. ASPEC method

Figure 5. IRSST method

Experimental plan

9 samples (3 per method) were collected each time in the dust chamber in order to compare the methods (paired Student's t-tests over a bilateral distribution). In total, 22 different tests were compiled. The tests were carried out on the natural logarithms due to the log-normal distributions of the data. Mulhausen and Damiano (1998) estimated that 6 to 10 measurements are required to suitably estimate a mean and a standard deviation (9). These tests were conducted by varying the fan operating time and the amount of dust in the generator. The average velocity of the air in the dust contamination chamber, measured with a hot-wire anemometer (TSI Inc., model 8384, Shoreview, MN, USA), was 0.59 (±0.15) m/sec (calculated using 50 measurement points). The precision of this instrument is ±3%. Weighing was done using the standard IRSST method (10) whose minimum reported value is 25 µg.

Visual inspection

In parallel with this dust sampling, a committee of experts from different disciplines related to air quality did a subjective evaluation based on visual assessment of the dust contamination. In addition to the four people in charge of this activity, this committee consisted of four other specialists (chemist, certified hygienist, building engineer and microbiologist), all from the IRSST. This assessment was based on direct visualization of the deposits with concentrations unknown to the committee. It involved a 3-level scale, where level 1 or normal, is characterized by clean ducts or ducts with a thin uniform layer of dust; level 2 or above normal, is characterized by a uniform layer and localized accumulations; and level 3 or serious, is characterized by significant accumulations (11,12). Level 2 corresponds to the limit concentration (or concentration range) for initiating cleaning.

RESULTS AND DISCUSSION

Initiation criterion

Table 3 presents the results of the corrected weighings of the control substrates. The minimum value used for concentration calculations is the method's reported minimum value, 25 µg divided by the square root of 2 (13,14).

Table 2. Corrected weighings[1] (mg) for the aluminum control substrates for the 22 tests

1st third of the base	2nd third of the base	3rd third of the base
0.004	0.020	0.111
0.004	0.020	0.111
36.375	34.661	30.895
5.509	5.329	5.883
12.545	11.976	10.053
3.625	3.501	3.218
20.579	19.366	17.902
8.510	8.410	7.342
4.825	4.939	4.857
3.228	3.273	3.289
2.367	2.749	2.775
5.576	5.995	5.703
6.464	6.551	6.307
12.979	12.913	11.773
24.551	23.127	21.096
5.535	5.765	5.024
2.546	2.882	2.780
7.368	7.183	7.235
9.360	9.158	8.072
5.399	5.412	5.150
28.045	25.659	23.138
31.430	27.336	25.362

[1]: Each of these values represents an average of six weighings

The paired Student's "t" comparison tests on the normal logarithms between the first third and second third ($p \leq 0.185$), between the first third and third third ($p \leq 0.279$), and between the second third and third third ($p \leq 0.393$) are all non-significant. The deposits are therefore considered as uniform along the entire base.

Table 3 presents the comparisons of the average votes of the committee of experts in relation to the average weighings for each of the methods evaluated, including the control substrates. Therefore, for an average vote of 1.6/3, or approaching level 2 which is characterized by a uniform layer and localized accumulations, the corresponding values are 0.2 g/m² for the NADCA method, 0.3 g/m² for the ASPEC method, and 0.6 g/m² for the IRSST method. All these methods are significantly different ($p \leq 0.05$) from one another. Despite the fact that there is generally a significant difference between the weighings for the control substrates and the IRSST method, the same value is obtained for these two methods, or 0.6 g/m² for a vote of 1.6.

Table 3. Results of the evaluation of the 3 methods (g/m²) in relation to the vote of the committee of experts

Substrate	IOM (IRSST)	ASPEC	NADCA	Committee's vote (n=6)
0.264	0.402	0.075	0.069	1.0
0.274	0.235	0.086	0.066	1.0
0.326	0.205	0.131	0.031	1.3
0.346	0.110	0.114	0.073	1.1
0.488	0.151	0.244	0.126	1.2
0.538	0.411	0.226	0.146	1.7
0.545	0.517	0.289	0.238	1.4
0.558	0.423	0.187	0.186	1.0
0.575	0.624	0.286	0.200	1.6
0.647	0.414	0.244	0.162	1.8
0.727	0.612	0.343	0.171	2.1
0.811	0.628	0.364	0.234	1.7
0.885	0.569	0.430	0.325	1.9
1.152	1.077	0.786	0.494	1.3
1.257	0.970	0.797	0.490	2.6
1.928	1.479	1.222	1.087	2.3
2.294	1.530	1.292	1.338	2.8
2.557	1.382	1.576	1.550	2.9
2.805	2.407	1.174	1.607	2.6
3.399	3.136	2.470	1.923	2.0

CONCLUSION

We can therefore state that the numerical criteria for initiating cleaning of smooth non-porous ducts were determined from a laboratory study. Clearly, the method recommended by the IRSST requires some adjustments before being used commercially. In addition, the method's conditions of application will have to be determined for existing smooth non-porous ducts from a ventilation system and with the actual dust from occupied buildings.

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Cleanliness of ventilation systems - a REHVA guidebook

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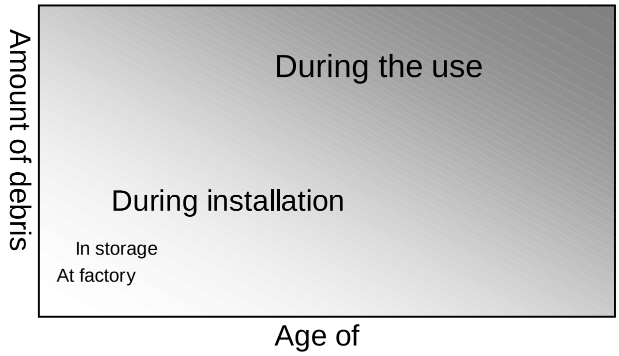
SUMMARY

The published REHVA guidebook provides comprehensive up-to-date information about design features, criteria for cleanliness, inspection and cleaning instructions of ventilation systems. The guidebook is aimed at practitioners, designers and those who are setting criteria for cleanliness of ventilation systems. The design practice includes guides to construct a clean ventilation system and which cleanliness can be maintained during whole lifetime of the building. This provides to take account the cleanliness in building construction and installation processes as well as demands for proper maintenance actions including the functional checks and cleaning. A pathway for verification of the cleanliness and measuring methods are presented. Training practices for inspection and maintenance personnel in different countries are also introduced. The content of the guidebook is a consensus of the state of art information reviewed and discussed with the European specialists representing practice and science.

INTRODUCTION

Dust and other contaminants accumulate on the surfaces of the ventilation system during construction and operation and it is revealed that the ventilation systems itself may be potential sources of pollutants in buildings. Dusty surfaces may increase energy consumption, decrease air flow rate and cause malfunction problems to ventilation system. Additionally, contamination in the supply air duct may cause negative health impacts to occupants. Thus, the ventilation system has to be inspected regularly and cleaned whenever the inspection has revealed that the amount of accumulated dust has exceeded the acceptable limit. Ventilation systems, which convey very dusty and/or fire hazardous contaminants must be cleaned frequently.

During the last few years there has been increasing awareness of the importance of cleanliness in air ducts. Dust accumulation in newly installed air ducts was found to be high when attention had not been paid to the cleanliness of the ducts during construction. Dust and other impurities in new ducts originate particularly from installation work and when the particle concentration is high at the building site. After the construction process, pollutants accumulate on duct surfaces during the operation of the ventilation system. During operation, the main causes of dust accumulation in the supply air duct are polluted out-door air and inadequate maintenance of the filters (Figure 1). Highly efficient filtration protects supply air ducts during operation of the system and thus can increase the length of the intervals between cleaning.



(a)

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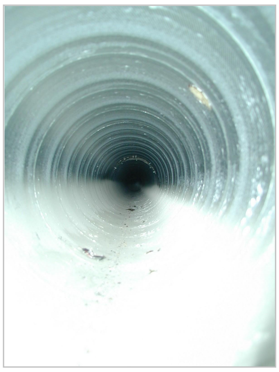


Figure 1. (a) Principle of dust accumulation on ventilation system after the components are manufactured. (b) Typical dust accumulation on round air duct.

The guidebook gathers the scientific and practical information in a way that it is easy to utilize during the many phases of the work towards cleaner ventilation systems. These guidelines will give guidance to practitioners on how to take the cleanliness aspects properly into account in design, construction, commissioning and operation.

CRITERIA FOR CLEANLINESS

In some countries official guidelines are given for cleanliness criteria for ventilation systems. However, most of these guidelines focus on fire safety and seek to minimise the amount of flammable material in ducts, especially in exhaust-air ducts. In only a few countries cleanliness criteria are specified for hygienic reasons. Different are given for new and existing systems. E.g. European standard gives hygiene requirements and recommendations for air-handling units (AHU), different components and sections of AHUs [1].

The guideline for **new systems** focuses on design and installations instructions and the demands for cleanliness of components. It aims that the new ventilation system is clean after installation and building construction as well as the system can be kept clean during the whole lifetime of the building. Limit values for dust accumulation in newly installed air ducts and systems are presented in Table 1. The production and installation of components for new ventilation systems is very important. Manufacture of sheet metal components includes critical phases because lubrication is needed between a tool and the metal sheet. Oil residues may serve as nutrients for micro-organisms which produce odour. During operation a surface with oil residues tends to accumulate dust faster, and the resulting dust layer is sticky and difficult to remove. In a system it is important to use only equipment and components which do not emit harmful substances, fibres or odours.

The following three major contaminants from all components which may reduce IAQ should be kept to minimum:

- residues of lubricants
- dust accumulated during manufacture and installation, or debris from construction
- micro organisms, particularly when toxic species are present and conditions during installation and storage are favourable for their survival and growth.

Table 1. Limit values for dust accumulation in newly installed air ducts and air-handling systems.

Country	Application	Category	Prior to clean ducts	Evaluating method	Reference
Finland	Supply	P1 [a] P2 [a]	1 g/m² 2.5 g/m²	Visual [d] or Vacuum test [e]	[2]
Germany	General	High [b] Middle [b] Basic [b]	Visually unclean Visually unclean Heavy dirt	Visual [f] Visual [f] Visual [f]	[3]
Norway	Supply	Class A [c] Class B [c]	3% 5%	Optical [g] Optical [g]	[4]
Sweden	Supply	–	1 g/m²	Not mentioned [h]	[5]
USA	General	–	Visually unclean	Visual [i]	[6]

Categories:
[a] cleanliness categories, [b] cleanliness levels, [c] cleanliness class levels
Evaluating methods:
[d] Visual inspection with a reference scale as the primary method [7]
[e] Vacuum test (FiSIAQ -test 2) [8]
[f] Requirements for specific categories (A and B) and training to inspectors before they are authorised to inspection work [9]
[g] Optical method with gelatine tapes [10]
[h] Requirements for specific qualification (classes K and N) and experience to inspectors before they are authorised to inspection work
[i] Requirements for qualification and experience of inspectors before they are authorised to inspection work

More detailed requirements for single components and ducts are given in a Finnish guideline [2]. Limit value for oil residues in ducts and the components manufactured by cutting, bending or jointing the limit value is 0.05 g/m^2, for parts that needs deep drawn the limit is 0.3 g/m^2. The amount of surface dust shall not exceed 0.5 g/m^2.

The emissions of new filters are controlled according to odour emissions and it is stated that low polluting new filters shall be used. Naturally, the efficiency of filter is most important feature and it shall be tested according to [11].

Control of cleanliness is a part of maintenance of **existing ventilation systems**. The majority of national maintenance guidelines do not include cleanliness inspections, rather they presents intervals for checking and maintenance. The hygiene of moist surfaces, such as humidifiers and cooling coils, needs to control to avoid or detect microbial contamination of the surfaces or water reservoirs. Criteria for the quality of humidifier water are specified, e. g., in the German guideline [12].

Cleanliness target values for existing air ducts are presented in Table 2. In the guidebook gives more detailed cleanliness control instructions for filters, heating and cooling coils, humidifiers and cooling towers.

RECOMMENDED DESIGN PRACTICES

Ventilation system design should be a "design for lifetime" rather than a "design for construction". This means that the cleanliness criteria should be defined and documented so that the maintenance of the system allows maintaining of cleanliness during the whole lifetime of the building.

Table 2. Current target values on the hygiene of air ducts presented as amount of accumulated dust.

Country	Application	Prior to clean ducts	After cleaning	Evaluation method	Reference
USA	(1)		0.1 g/m²	Filter with vacuum	[13]
Great Britain	Supply Air	1 g/m² 60 µm	0.1 g/m²	Filter with vacuum	[14]
	Recirculation Air	1 g/m² 60 µm		Filter with vacuum	
	Exhaust Air	6 g/m² 180 µm		Filter with vacuum	
Sweden	Supply Air	1 g/m²		not mentioned	[5]
Japan	Supply Air		1 g/m²	Wiping with cloth	[15]
Finland	Supply Air	2g/m² 5 g/m²		Scrape/Filter with vacuum	[2]
Germany	general	20 g/m²	10 g/m²	Scrape/Filter with vacuum	[9]

For new buildings the required standards for the indoor air quality (IAQ) should be high. This means that all designers have to consider the IAQ in their design solutions. In aiming at high IAQ the ventilation system, the designer should plan sufficient air-flow rates, adequate cleaning of the supply air, i.e. filtration, and ensure the system is constructed using clean components and a clean installation technique.

Design should also include sufficient guidance and instructions for

- assessment of cleanliness and need for cleaning
- default cleaning intervals
- cleaning method
- checking after cleaning, both for cleanliness and system functioning.

Guidebook presents design principles for different parts in ventilation systems. For example space for maintenance needs to be taken account in design of lay out in AHU room, Figure 2.

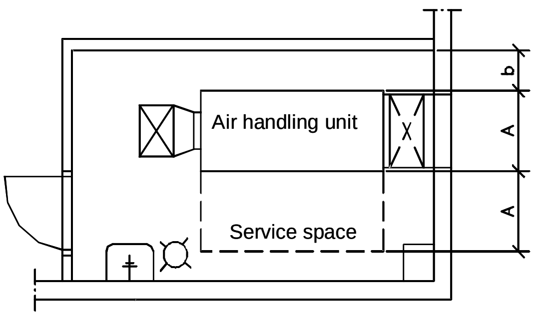


Figure 2. Sufficient space is required for maintenance access

INSTALLING CLEAN VENTILATION SYSTEMS

Ventilation systems are exposed to dust and debris during some phases of the system's history. The building process and installation are the most critical phases when dust from the building construction is likely to contaminate the system. Prevention of dust entering the partially built ventilation system is an effective technique for ensuring a clean system in the new building.

The best results are obtained by implementing the following actions:

- install the system during construction phases when no dust producing work is being carried out
- store uninstalled ducts and components under a cover or in a shed to prevent dust and debris from entering the components (Figure 3a)
- remove the packing materials or coverings just before installation
- check that the environment remains clean during the installation work
- cap or recap the installed openings, duct ends or airways during breaks in the installation
- only use cutting methods that do not produce metal shavings or powder
- where contamination has occurred, clean before closing the system
- check the cleanliness before commissioning of the system.

According to field studies the protection methods to keep the system clean during the installation process are very advantageous. In figure 3(b) the amount of accumulated dust is presented after each construction phase; P1 refer to technique in which attention is paid on the cleanliness during construction process and P2 refers to the method in which the installation is carried out without special care. At its best the amount of accumulated dust found in the duct work has been around the detection limits of the measuring methods.

(a) 

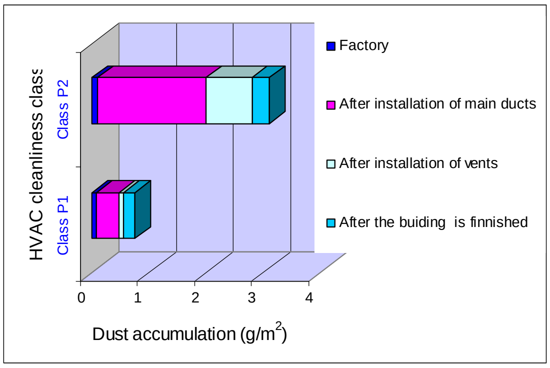
(b) 

Figure 3. (a) An example of protection of air ducts during storage at building site (left).
(b) Effect of different installation processes on dust accumulation on ventilation ducts. Class P1 and P2 are cleanliness classes expressed in [2].

VERIFICATION OF CLEANLINESS

In newly installed ventilation systems, the verification of cleanliness is required during the commissioning process to confirm that the building contractor has built a system that is as clean as the building owner has demanded. In existing buildings, the periodic inspection of cleanliness is most often done to evaluate the need for cleaning. The verification process is shown in the schematic of Figure 4(a).

The evaluation of cleanliness begins with an inspection plan. The content of the plan depends on the size and design features of the system. Firstly the evaluation is done visually, aided by a source of light, possibly also using a mirror and some tools. The level of cleanliness is compared against a cleanliness scale (Figure 4(b)) which contains six illustrations showing different amounts of dust accumulation.

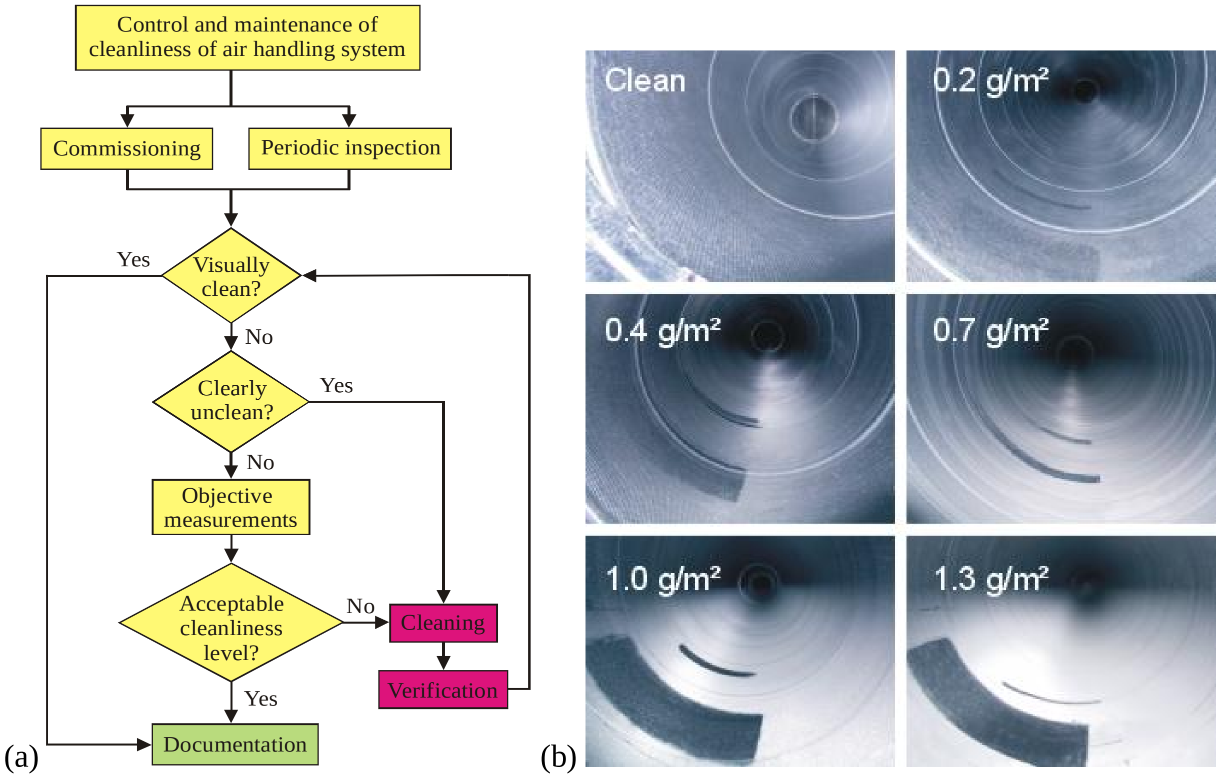


Figure 4 (a) A pathway for verification of cleanliness. (b) A reference scale for visual inspection of new installations [7].

If the contractor, duct cleaner or building owner is not satisfied with the results of the visual inspection, a more objective method of assessment should be used; this is carried out by collecting and weighing the dust deposited on a known area of the duct surface. For round ducts the dust is collected from the lower half or side quarter (as seen in figure 4(b)), and in rectangular air ducts from the bottom of the duct.

During the sampling notes from visual observations shall be done and also photographs e.g. with digital camera will aid the reporting and decision making if the system shall be cleaned or not. Finally, the results of the evaluation should be documented, with photographic evidence.

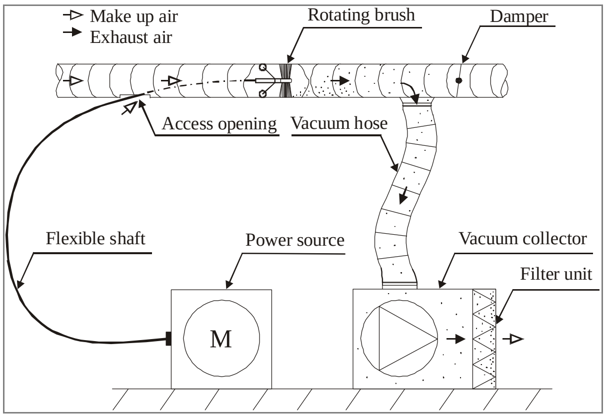
Different parameters or contaminants need different methods. Several methods for dust deposits are developed; vacuum sampling with loosening the dust by scraping is recommended in the guidebook. Microbial contamination on surfaces is useful to sample by swiping technique followed by cultivation of the sample. The microbial concentration in water samples are analyzed with the cultivation techniques as used in water hygiene control.

CLEANING OF VENTILATION SYSTEMS

The ventilation system should be cleaned according to a cleaning plan. The plan consists of a selection of methods suitable for the different components and surface materials. The methods should be selected so as to avoid damage to surfaces and components being cleaned.

A basic distinction is made between dry and wet cleaning. Compressed air cleaning, mechanical brushing, hand vacuuming are examples of dry cleaning methods. Compressed air is used e.g. for cleaning narrow gaps, e.g. between the fins of heat exchangers, or for irregular surfaces of other parts of ventilation systems. Mechanical brushing with a vacuum technique to remove the loosened dust is the most common cleaning method for spiral seamed ducts. With proper equipment the mechanical dry cleaning method is an effective technique to remove loose dry dust from the surface of air ducts (Figure 6).

Hand vacuuming is suitable for small areas like filter banks or the surfaces of silencers.

(a) 

(b) 

Figure 6. (a) A schematic picture of mechanical brushing with a vacuum collector. (b) Duct cleaning with mechanical brushing.

Wet cleaning methods are used, for example, for cleaning terminal devices after they have been removed from the duct work. Wet cleaning methods can also be used for the cleaning of heat recovery units if they are installed in a location where water can be drained from the air handling unit.

Cleaning robots are also useful in both in round and rectangular air ducts. The robots are usually equipped with a camera which provides an opportunity to see the progress of the cleaning work.

EDUCATION TO MAINTAIN CLEANLINESS

Guidebook gives a brief introduction to the education programs for hygiene inspections and maintenance of ventilation systems. However, as training procedures and legislation vary from country to country it is difficult to harmonize training practices. As training is imperative it is important to focus on local codes and guidelines, although an international training programme would be beneficial.

Another REHVA Guidebook “Hygiene Requirements for air-conditioning systems and units” presents an example of a training system established in Germany, Austria and Switzerland. EVHA guidebook states the training demands from point of view of cleaning professionals.

RELATIONSHIPS TO OTHER GUIDEBOOKS AND STANDARDS

The REHVA guidebook (nr8) is not an official guideline or regulation; but recommends the best proven practices for maintaining hygienic and clean ventilation systems. However, this guidebook does present some recommended minimum or maximum values for different parameters, such as default values. National regulations, which may require more stringent values should always be followed. The guidebook is useful for practitioners who like to follow the recent international practices. It offers a wide approach to the factors affecting indoor air quality and is aiming to conform to the European standards of HVAC maintenance. Standard EN 12097 gives requirements for ductwork design and construction in order to ensure the cleanability of the system, focusing on the size and location of access openings.

The other REHVA guidebook "Hygiene requirements for ventilation and air conditioning systems and units" (nr 9) goes in more detail to hygiene and health issues of the systems and components. The European Ventilation Hygiene Association guidebooks "EVHA Guide to cleaning and hygiene management of ventilation systems" and "EVHA Good practice document for grease extract cleaning" are targeted to cleaners.

ACKNOWLEDGEMENT

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Man Made Mineral Fiber Emission from HVAC-components

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SUMMARY

Man made mineral fibers (MMMFs) that can cause irritation in upper respiratory tract, eyes and skin, can be emitted to indoor air from the HVAC system. A Finnish project was set up to measure fiber emissions and to develop design and materials of the HVAC systems to improve indoor air quality. Within the project laboratory tests and field measurements were carried out.

As a result, new methods of testing the emission rates of the MMMFs from HVAC-components have been developed. With these methods HVAC-components can be classified within the Finnish Indoor Air Classification. For instance, in the vibration test the limit value will be 10 fibers/m^3.

Also, there were differences between fiber emissions in the studied buildings. The fiber concentrations (length > 20 µm) in the supply air ranged between 0.01 – 85 fibers/m^3. The building with the highest emissions had also many workers suffering from respiratory and skin symptoms.

INTRODUCTION

Man made mineral fibers (MMMFs) that can cause irritation in upper respiratory tract, eyes and skin, can be emitted to indoor air from the HVAC system. A Finnish project "Particle discharges of air-conditioning systems: health risks, measurement and product development" was set up to measure fiber emissions and to develop design and materials of the HVAC components to improve indoor air quality. This was implemented by measuring fiber emissions and exposures in buildings and by emission tests in laboratory. Also the sampling and analytical methods were developed and the limit values of fiber emissions were specified.

Preliminary results of the projects have been introduced in Clima2005-conference in Lausanne [1], [2]. This paper deals with the final results.

METHODS

Laboratory measurements

In the laboratory tests fiber emissions from HVAC-components (round silencers) were examined by several methods. The components were sent to VTT by the HVAC-companies that participated in the project. In every silencer the fiberglass cloth and perforated sheet metal prevented the emission of fibers (glass or rock wool) from the attenuation material.

Fiber emissions were measured in accordance with the Finnish Classification of Indoor Climate 2000 [3].

The product to be examined was installed as a part of the test-duct (ϕ160 mm). The other parts were the adjustable fan, HEPA-filter, flow measuring device and polypropylene filter cloth, Figure 1. The cloth sampled all the fibers that emitted from the HVAC-component.

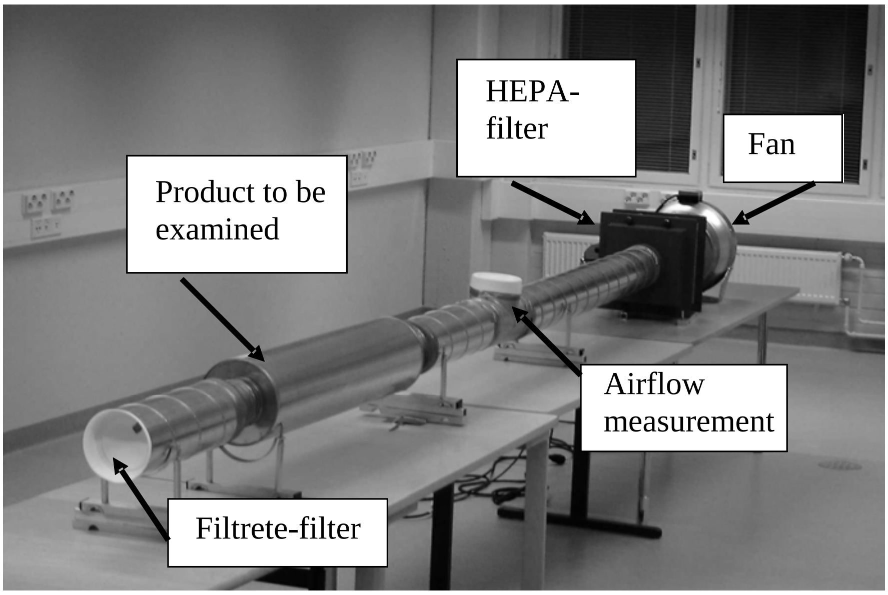

Figure 1. The measuring device for mineral fiber emissions (airflow flushing test).

As the filter cloth had been placed, the first measuring period was the initial flushing of 0.5 h. This was carried out to ensure that eventual fibers from the manufacture, transportation and storage do not have an effect on the actual tests. The airflow rate during the initial flushing was 50 dm^3/s. After this initial flushing the cloth was changed for the actual tests of several methods.

The airflow flushing tests lasted 1 h and 24 h. After each test the filter cloth was changed. The airflow rate during the actual tests was 80 dm^3/s. As the diameter of the test duct was 160 mm, the air velocity in the duct was about 4 m/s.

Besides, the emissions of fibers were measured by pressure shock of about 100 Pa. This was carried out by turning the damper in the test-duct from open to closed position in turn during 5 minutes. Every turn lasted about 5 s. The airflow rate during the tests was 80 dm^3/s.

Other tests were a vibration test of 1 hour and a cleanliness or brushing test of 5 minutes. The airflow rate during both tests was 80 dm^3/s.

In the vibration test the round silencer to be tested was installed on a carrier that caused a frequency of 25 – 35 Hz (amplitude about 5 cm) on the silencer, Figure 2.

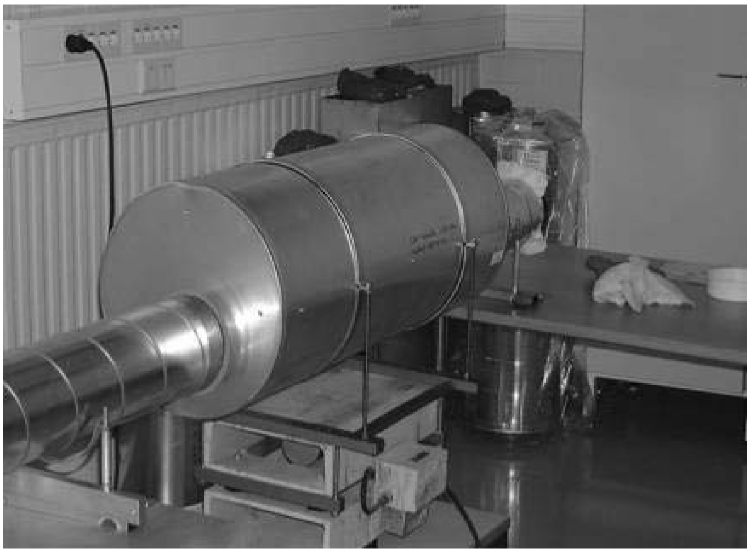

Figure 2. Round silencer on the carrier for the vibration test.

In the cleaning or brushing test the cleaning brush was pushed back and forth 10 times in the test duct and the silencer, Figure 3. A drilling machine was used to rotate the brush. The brushing lasted 5 minutes. After that an airflow flushing test was performed during 24 hours. The airflow rate during these tests was again 80 dm^3/s.

Figure 3. The measuring device for the cleaning or brushing test.

The samples were prepared for fiber (length > 20 μm) counting with polarization light microscope equipped with phase contrast optics (magnification 100X). The analyses were made in the Finnish Institute of Occupational Health. A new analytical method has been developed, where organic polypropylene fibers and inorganic glass or rock wool fibers can be separated. The results were given as fibers/cm^2 on the polypropylene filter.

The concentration of the fibers in the airflow can be calculated by Equation 1.

$$C_V = \frac{C_A \times A}{q_V \times T}, \qquad (1)$$

where C_V is the concentration of fibers in the airflow [fibers/m^3], C_A is the accumulation of fibers on the polypropylene filter [fibers/m^2], A is the cross-sectional area of the polypropylene filter (or the duct) [m^2], q_V is the airflow rate [m^3/s] and T is the measuring time [s].

Field measurements

MMMF emission from the ventilation equipments was surveyed in 10 buildings: one school, one day-care centre, one library, one community centre and six office buildings. Preliminary samples had shown that settled dust in every building contained MMMFs. In some of the buildings people were known to have prolonged sick-building symptoms, the cause of which was suspected to be MMMFs. Some buildings were quite new without excessive irritation symptoms among occupants. In each building, two to four rooms were selected for the fiber measurements in supply air and in settled dust. During preliminary visit selected surfaces were cleaned before the sampling of settled dust. The studied buildings had mechanical supply and exhaust ventilation systems. Ventilation rates were in accordance with The National Building Code of Finland.

Fiber emissions from HVAC-components were surveyed by measuring the MMMF concentration in supply air. As in the laboratory tests, samples were collected on a polypropylene filter cloth (3M Filtrete™). Now the cloth was installed on the supply air terminal device with tape, Figure 4. Air flow rate was measured after the installation. The sampling time was 2 - 4 days.

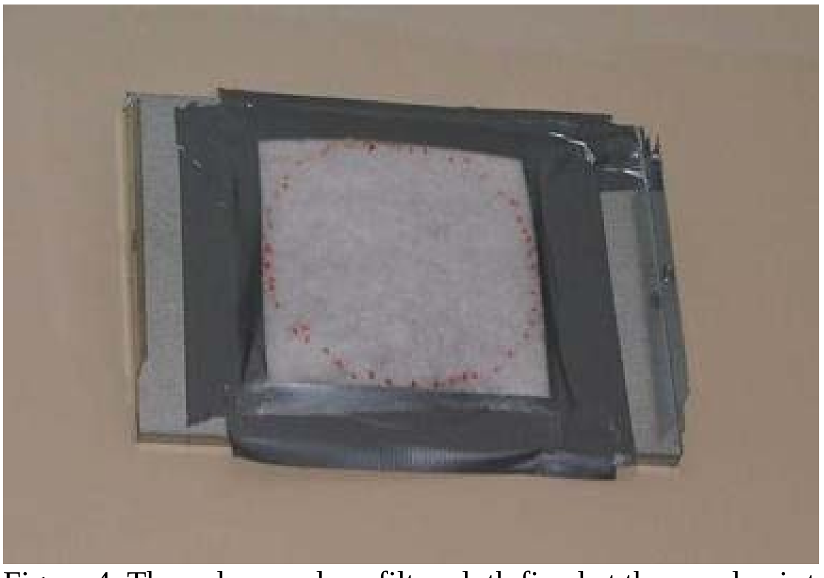

Figure 4. The polypropylene filter cloth fixed at the supply air terminal device.

Settled dust samples were collected with special tape (BM-Dustlifters®, tape area 14 cm^2), which was pressed against the surface (desks, shelves, supply air terminal devices). MMMFs (length >20 μm) were counted with a stereomicroscope (magnification 80-100X). The results were given as MMMF/cm^2.

In addition, the composition and origin of the dust collected from the ducts and indoor air were determined by electron microscope. Also the structure, functioning, materials and cleanliness of the ventilation equipment were examined in each building.

RESULTS

Laboratory measurements

Results of the airflow flushing test of 24 hours in Figure 5 reveals the difference in fiber emissions between various silencer types.

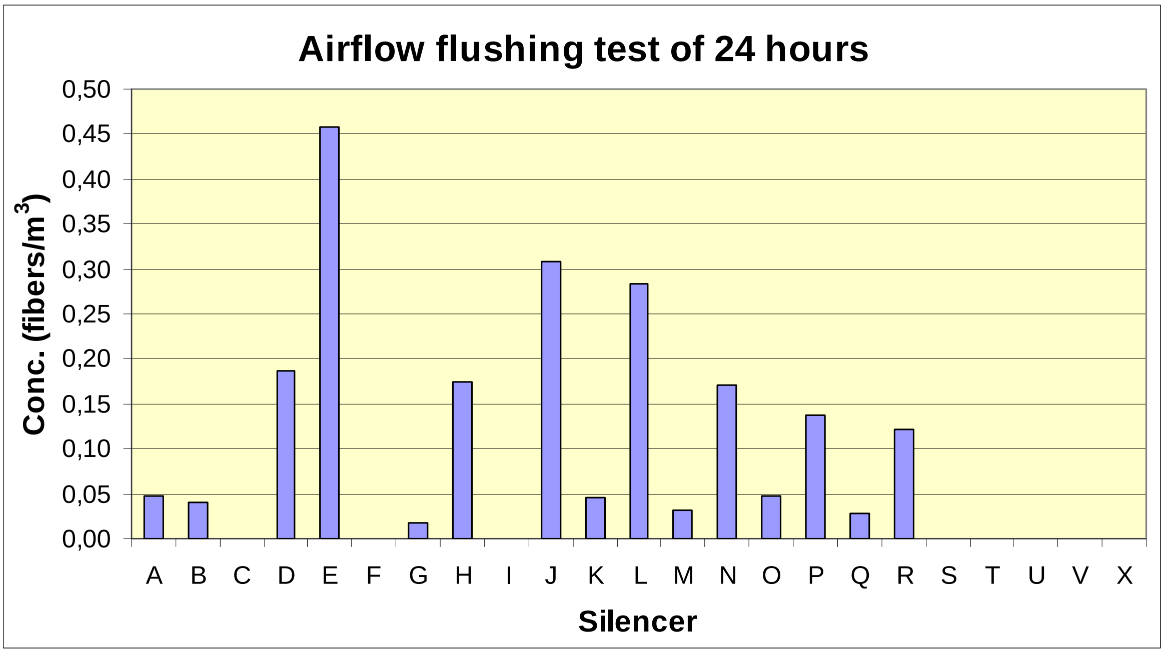

Figure 5. An example of the results that reveals the difference between various silencers.

In Figure 6 all results are collected and average fiber concentration in each tests are highlighted.

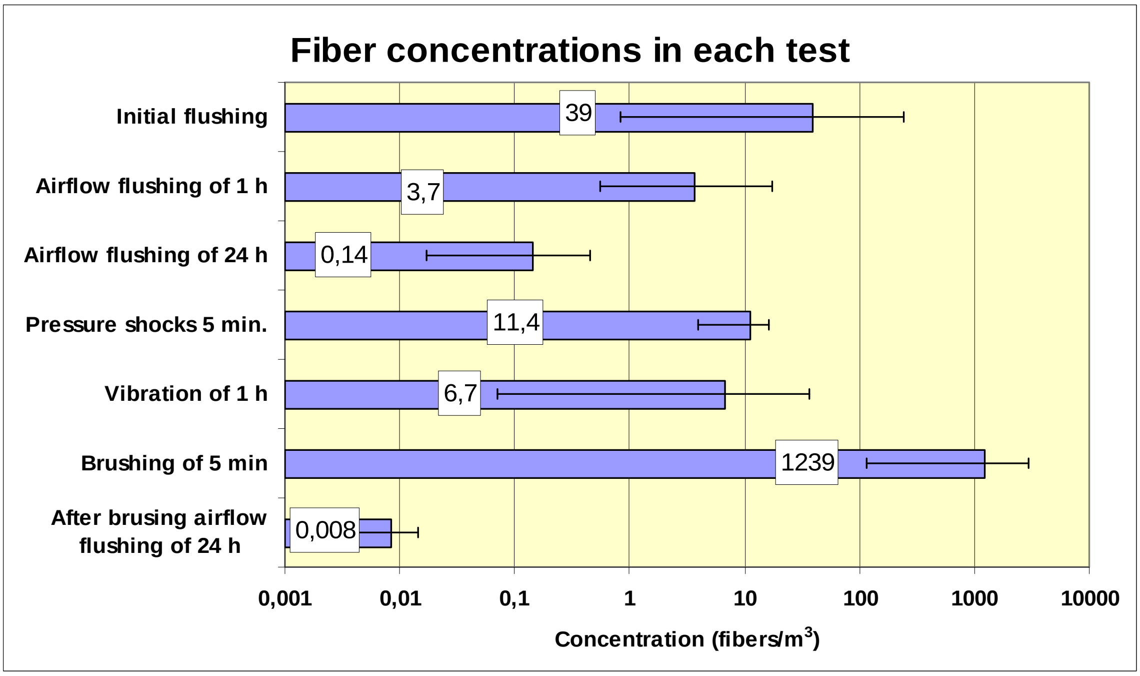

Figure 6. Fiber concentrations of different test methods. The averages and variations of the results are seen in the columns.

According to Figure 6 the highest fiber concentrations during the initial flushing were about 1000 times as much as the lowest value during the airflow flushing of 1 h test. In the airflow flushing of 24 h the range of fiber concentration was between 0.02 – 0.46 fibers/m^3. Highest fiber concentrations were during the brushing tests, over 1000 fibers/m^3. However, during the airflow flushing tests of 24 after the brushing tests, concentrations were again less than 0.01 fibers/m^3.

Field measurements

The concentrations of MMMF in supply air and in settled dust of ten studied buildings are presented in Table 1.

Table 1. MMMF (length > 20 µm) concentrations in the supply air and surface fiber density (during 14 days).

Building	Number of samples (conc. / density)	MMMF concentration (fibers/m^3)	MMMF density (fibers/cm^2)
School	9 / 7	0.3 – 1.8	
Office A	3 / 5	<0.07	<0.1 – 0.2
Office B	5 / 7	<0.06	<0.1 – 0.1
Office C	16 / 3	0.2 – 85	0.1 – 2.6
Office D	2 / 4	0.1 – 0.3	<0.1
Library	2 / 6	0.7 – 1.2	<0.1 – 0.3
Office E	3 / 9	0.03 – 0.12	0.1 – 1.1
Day-care centre	3 / 5	0.1 – 0.4	0.2 – 0.4
Community centre	3 / 3	0.01 – 0.10	0.1 – 0.5
Office F	3 / 4	6.6 – 10.1	0.2 – 0.7

The fiber concentrations (length > 20 µm) ranged between 0.01 – 85 fibers/m^3. Concentrations higher than 1 fiber/m^3 were measured in four buildings.

The surface fiber densities (length > 20 µm) during 14 days ranged between <0.1 – 2.6 fibers/cm^2. In several surface samples, the density exceeded the value of 0.2 fibers/cm^2. The detection limit is 0.1 fibers/cm^2.

The emitted dust composed mainly of various mineral particles, soot and organic material.

DISCUSSION

In laboratory the fiber concentrations in airflow flushing tests of 24 h ranged between 0.02 - 0.46 fibers/m^3. However, one must remember that only fibers that are longer than 20 µm have been taken into account. Therefore these concentration values are not comparable with the present limit value of 0.01 fibers/cm^3 or 10000 fibers/m^3 presented in the Finnish Classification of Indoor Climate 2000 [3]. This limit value originated from the clearance value of asbestos fiber measurement after asbestos removal and it contains all respirable fibers.

In conclusion, a new method of testing and analyzing the concentration of MMMFs from HVAC-components has been developed. With this method HVAC-components can be classified within the Finnish Classification of Indoor Climate. Within this project the limit values of fibers will be updated as

- 10 fibers/m^3 in vibration test
- 0.1 fibers/m^3 in airflow flushing of 24 h test after the brushing.

With these values about 90 % of sound dampers that were examined in the project can be considered as acceptable.

In the studied buildings fiber concentrations of supply air were mainly under 1 fiber/m^3. Surface fiber densities during 14 days were up to 2.6 fibers/cm^2. The sound dampers of ventilation equipment were one reason to elevated fiber emissions in the studied buildings. Workers suffered from airway and skin symptoms in the building where the highest fiber concentrations of supply air were detected. In many rooms the surface fiber density exceeded the recommended value 0.2 fibers/cm^2 [4]. The sound damping sheets on the roofs in some buildings may have increased the surface fiber density.

The new method of sampling and analyzing the concentration of MMMFs that was used in laboratory can be applied in the field measurements. With this method, the limit value of 1 fiber/m^3 in supply air is recommended. The recommended value of surface fiber density during 14 days is 0.2 fibers/cm^2. In buildings, where these limit values are exceeded, finding fiber sources and solving the possibility to reduce them, are strongly recommended.

ACKNOWLEDGEMENT

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Fire insulations of ventilation equipment and shafts as a source of man-made mineral fibres

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SUMMARY

The sources of man-made mineral fibres (MMMFs) settled on furniture surfaces and in supply air were determined in an office building. The impact of renovation on indoor air and the perception of air quality by the occupants were studied. Renovation improved the air-tightness of the inner parts of structures. Settled MMMFs were collected with gelatine tape by pressure and deposition techniques. MMMFs in supply air were collected using the filter installed to the supply diffuser. Sources of MMMFs were determined by comparing the found fibres to the fibres of insulation materials in the building using SEM/EDS. Occupants' perceptions of air quality and how it affected their wellbeing were collected using a questionnaire. The results showed that MMMFs originated mainly from fire insulations outside the main air ducts in shaft structures. The renovation improved indoor air quality and reduced the number complaints and symptoms of occupants. This study indicates that it is important to maintain the air-tightness of building preventing air leakage from shaft structures to indoor air.

INTRODUCTION

Man-made mineral fibres (MMMFs), which are also referred to as man-made vitreous fibres (MMVFs), are amorphous silicates manufactured from glass, rock, or other mineral [1, 2]. The concentrations of man-made mineral fibres in indoor air and the sources of mineral fibres have been studied in Europe and in the United States since the 1980´s. Based on current knowledge, the sources of man-made mineral fibres can be thermal, sound, acoustical and fire protection insulations in building structures and ventilation equipments. Although MMMFs are not considered to have a carcinogenic effect [4], high levels have been related to symptoms experienced by occupants in particular irritation of the respiratory tracts, eyes and the skin. Exposure to fibres occurs mainly through surfaces [3].

The purpose of the study was 1) to identify the sources of MMMFs settled on furniture surfaces in the office building and 2) to assess the impact of renovation on the indoor air quality and on the experiences of occupants. This study is the part of comprehensive study that focuses on the condition of indoor air quality and climate of the building.

METHODS

A 8(9)-storey office building in which occupants perceived building-related symptoms was studied. The stone-structured building comprised of three sections connected to each other by an intermediate part. The study building had a mechanical air handling system. The main air ducts of ventilation units were located in vertical shafts located in the centre of the each building sections.

Man-made mineral fibres settled on room surfaces were sampled with gelatine tape (tape size 2 cm x 7 cm) by pressing the tape against surface or by allowing fibres deposit on the tape placed on the surface. The deposition time was one week. MMMFs were calculated with a light microscope. Two replicate samples were taken from each measurement point. The results were expressed as fibres/cm^2.

Man-made mineral fibres in supply air were collected using the polypropylene filter cloth that was attached to the supply diffuser. Mineral fibres were calculated using light microscopy. The samples were taken in different parts of the building complex, in the service areas of different ventilation units. Results were expressed as fibres/m^3.

The sources of MMMFs were determined using scanning electron microscope with X-ray microanalysis (SEM/EDS). The sampled MMMFs were compared to the insulation materials of structures and ventilation equipments.

Complaints and symptoms experienced by occupants were collected using the Indoor Air Questionnaire (modified MM-40 questionnaire). Complaints and symptoms attributed to the building every week were reported from the past three months. The inquiry was conducted twice, before and after renovation in 2004 and in 2006, and in one part of building five times between years 2001 and 2006. The percentage of answered questionnaires ranged from 53 % to 60 % yielding 250 and 310 answers.

After determining the initial quality of the indoor air, the building was renovated. The feedthroughs of the vertical shafts and the manholes were sealed and an under pressure was created in the shafts. Old transfer air vents leading from rooms to the corridors were also sealed. The air-tightness of structures was checked with tracer gas, sulphur hexafluoride gas, and gas analyser after renovation.

In the statistical analysis the results were tested with the testing method of proportional sample (questionnaires) and with the *t*-test (mineral fibre concentrations).

RESULTS

Man-made mineral fibres on surfaces

A significant decrease in settled man-made fibres ($p<0.001$) was observed after renovation of an office building (Table 1). The decrease in concentration levels had taken place in 2006 both as regards the number of samples exceeding the target maximum value of 0.20 fibres/cm^2 and as regards the average concentration level. In 2004 the variation of the MMMF concentrations was high ranging from <0.07 fibres/cm^2 to 100 fibres/cm^2. The high single concentrations may be due to the basic repairs of ventilation equipments in process simultaneously in part of the building and insufficient protection during work. This becomes evident in the check-up measurements carried out after the repairs in 2004 whereby equally high single concentrations were not observed.

Table 1. Concentrations of man-made mineral fibres before (2004) and after (2006) renovation works.

	Pressure samples		**Deposition samples**	
	2004	**2006**	**2004/2 (check-up measurements)**	**2006**
Number of samples	137	80	19	80
Average fibres/cm^2	4.27	0.07	0.18	<0.07
Median, fibres/cm^2	0.29	<0.07	0.07	<0.07
Minimum, fibres/cm^2	<0.07	<0.07	<0.07	<0.07
Maximum, fibres/cm^2	100	0.29	1.43	0.29
Samples exceeding target value of 0.20 fibres/cm^2 , %	58	4	11	1

Man-made mineral fibres in supply air

In general, the mineral fibre concentrations in supply air were low. The MMMF concentrations varied from <1 fibre/m^3 to 21.4 fibres/m^3. It was found that the MMMF concentrations were below the target value of 1.0 fibre/m^3 suggested by Finnish Institute of Occupational Health [5] in 80 % of samples.

Sources of man-made mineral fibres

X-ray microanalysis of settled man-made mineral fibres showed that MMMFs originated mainly from the fire insulations outside the main air ducts of the vertical shafts. Some fibres originated from the insulation materials of partition walls and from old transfer air vents in 2004, but not after renovation in 2006. MMMFs in supply air and on furniture surfaces originated from the same source i.e. the fire insulations outside main air ducts of vertical shafts.

Indoor Air Questionnaire

In general, the amount of building related symptoms and environmental complaints decreased after renovation (Figures 1 and 2). No exceptional amount of building-related symptoms (above 20 %) and environmental complaints (above 40 %) emerged in autumn 2005 or in the spring months of winter 2006. The amount of reported eye irritation symptoms and dry air perception diminished significantly ($p<0.05$) in the winter spring months (in 2006 compared to 2004). Similarly, the dry skin symptoms and dry air perception diminished significantly ($p<0.001$) in autumn (in 2005 compared to 2001). In late winter in 2006 more symptoms were reported than in autumn 2005. In winter time indoor air is dry in Finnish buildings and may increase the perception of dry air.

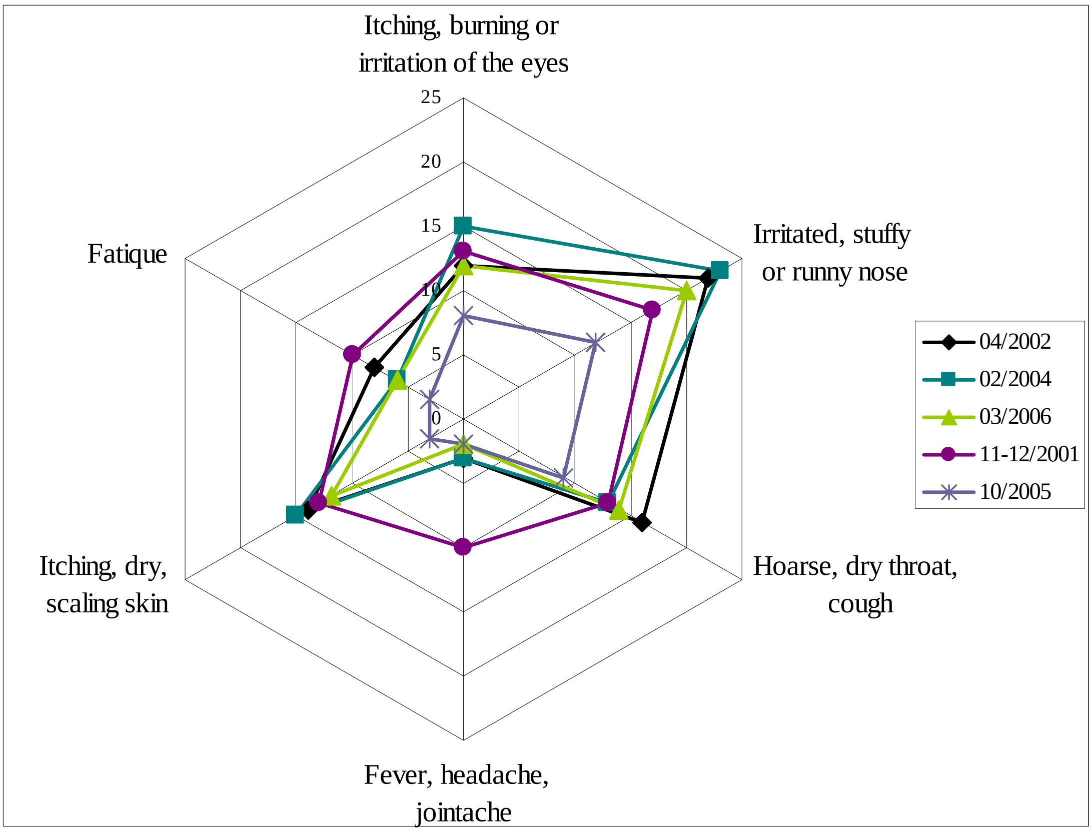

Figure 1. Building related symptoms in the study area of the building. After the inquiry of 02/2004 the comprehensive measures were made in the building.

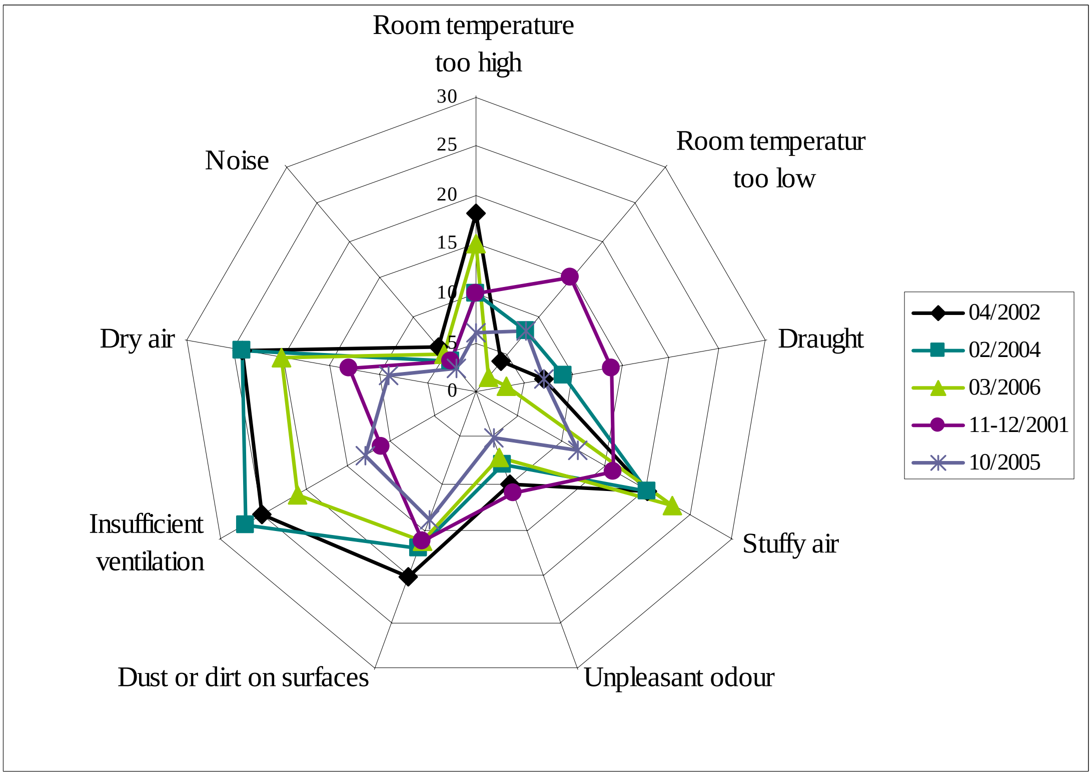

Figure 2. Environmental complaints in study area of the building. After the inquiry of 02/2004 the comprehensive measures were made in the building.

DISCUSSION

These results showed that man-made mineral fibres found in dust settled on furniture surfaces and in supply air originated mainly from the fire insulation materials outside the main air ducts of the vertical shafts and also to a lesser extent from the insulations of the partition walls and from transfer air vents. This study indicated importance of renovation for indoor air quality. The indoor air quality improved, and, complaints and symptoms experienced by occupants decreased as the emissions of MMMFs decreased.

Thus the renovation work of sealing the inner parts of the structures towards the shaft structures, can be considered useful. The present study does not provide adequate evidence to conclude the reasons for symptoms and complaints. The symptoms and complaints may have been caused by mineral fibres, or they may also be related to the migration of the air into indoor air through non-airtight structural components. The air flowing through the structures contains various impurities originating from building materials or for example from microbes attached to the inner walls of shaft structures. The joints between the indoor spaces of the building and the shaft structures were non-tight and air flowed towards indoor spaces from the shaft structures. Buildings with mechanical ventilation system have mainly negative pressure. Based on this research results it is important in buildings with mechanical ventilation system to maintain the air-tightness of the inner parts of structures e.g. towards shaft and air conduit structures.

ACKNOWLEDGEMENTS

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Air quality by VAV HVAC system before and after cleaning. Case study.

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SUMMARY

Since the study by P.O Fanger (1988) [1] we know that the Heating Ventilating and Air Conditioning (HVAC) system could be responsible for a large amount of indoor air pollution and Sick Build Syndrome (SBS). The pollution could become from filters, cooling coils and dust accumulated on duct surfaces in systems with poor maintenance.
While the importance of maintenance of air handling units and replacement of air filters is well recognized in Portugal, the cleanliness of ducts is sometimes forgotten. Research is needed on standard methods to measure the surface pollution and criteria to appreciate the cleanliness of duct surface, as well as the requirement for duct disinfection after an adequate mechanical cleaning.
In this paper the results of a study undertaken in a 9 year old office building in Lisbon area with a VAV system are presented, including the methods used to measure the air quality and the surface pollution. The results show that mechanical cleaning contributes to a large reduction in dust concentration in surfaces and in the air supplied to spaces. The concentration of bacteria and moulds in surfaces and in the air was quite low and therefore in this building chemical disinfection is not required.
A continuous audit of the system is recommended (with visual inspection for instance and some measurements) to choose between cleaning some components or the complete system and some suggestions about the criteria to be adopted before cleaning ducts are also offered.

INTRODUCTION

To promote good indoor air quality in buildings we need to supply an adequate air flow rate to dilute and remove pollutants and the supplied air should also be fresh and clean.
In the study of Fanger [1] in 1988 it was shown that HVAC systems could be responsible for 42% of the pollution sensed by occupants. The pollution of HVAC systems has many sources, namely air filters, heating and cooling coils, humidifiers, water condensation pans and ventilations ducts.
The contamination of inert HVAC components (ex. components made of steel, copper, aluminum, etc) could be associated with the transport of contaminants germinated in air handling units, high humidity levels on surfaces, oil residues on surfaces, accumulated dust on ducts which in the presence of water allow the germination of microorganisms.
To reduce the risk of HVAC contamination is proposed [2] an annual audit of HVAC system, periodic replacement of air filters, cleaning of humidifiers, heat exchangers, water condensation pan, etc. Regarding duct it is recommended the cleaning when necessary, without defining the criteria to decide when it will be necessary. Duct cleaning was first

introduced to reduce the fire hazard or the blockage of duct by the dust which reduces the air flow. Nowadays, the impact on the air quality supplied to the building, is also a concern (figure 1).
Regarding the impact of duct pollution in Indoor Air Quality (IAQ) currently there isn't information available that could allow the definition of safe limits for dust or microorganism accumulation in duct surfaces indicating good/bad air quality supplied to spaces. Some previous work suggests that duct cleaning has a minor effect in IAQ [3, 4].
In this study we were interested in showing, using quantitative measurements, the impact of HVAC and especially duct cleaning in the air quality supplied to the building and in the pollution of the duct surfaces.

Dirty duct

Clean duct

Figure 1. Ducts

CRITERIA TO EVALUATE DUCT CLEANLINESS

The simplest method to appreciate duct cleaning is the visual inspection, which allows the identification of a large number of anomalies. For instance, dust concentration of 5 to 10 g/m^2 are thick and easily seen [3]. When dust concentration is lower than 0.1 g/m^2, a cleaned duct is observed.
Quantitative methods could be used to appreciate more objectively the dust level inside ducts, namely: the vacuum test method, the gravimetric tape or the optical method [3,4]. To measure microorganism levels contact plates are used [5,6].
To measure dust accumulation, Holopainen [5] showed that the most precise method is the Vacuum test. The tape test method could be used for lower concentrations (lower than 3 g/m^2).
As already referred, a correlation between surface dust concentration and it's impact on indoor air quality, it has not yet been defined. For that reason, quantitative methods have been adopted essentially to determine adequate duct cleaning.
The NADCA association [7] refers that ducts are clean when the measured surface dust concentration (obtained with the vacuum test method with air flow rate of 15 l/min and with the cassette (filter holder) sliding over a structure not touching the surface of the duct) is lower than 0.075 g/m^2.
The APIRAC association [8] considers that acceptable cleanliness of ducts is achieved when the obtained concentration of surface dust is smaller than 1 g/m^2. APIRAC's method uses a collection airflow of 25 l/min and the cassette contact directly with the surface of the duct.
FISIAQ considers two cleanliness classes [9] in new air conditioning ducts: for the P1 class the limit for the concentration of surface dust is 1.0 g/m^2 and for the P2 class the limit is 2.5 g/m^2, if we consider existing ducts limits are 2.0 g/m^2 and 5.0 g/m^2 for the classes P1 and P2 respectively. For P1 category oil residues must be lower than 50 mg/m^2.

In the United Kingdom [9] the maximum limit allowed is de 1.0 g/m^2 in supply ducts and 6 g/m^2 in exhaust ducts, measured according NADCA method.
A technical note from Chow et al [10] recommends cleaning when surface dust measured with the NADCA method detects concentrations higher than 6 g/m^2 in exhaust ducts and higher than 1 g/m^2 for supply air duct or exhaust ducts on a system with air recirculation
There are no reference limits for microbiological contamination. Luoma et al reports [4] presents some results with mean concentrations of fungi ranging from 0.05 to 11 cfu/cm^2 and bacterial concentrations ranging from 0.03 to 13 cfu/cm^2.
A Norwegian expert group [3] presents three risk classes for surface microbiological contamination. For fungi, concentrations lower than 1000 cfu/g of dust presents a low risk, for concentrations smaller than 3000 cfu/g of dust present medium risk and, finally, the risk will be high for concentrations higher than 3000 cfu/g. For bacteria, low risk is defined for a concentration lower than 6000 cfu/g, medium risk for concentrations lower than 10000 cfu/g and high risk will be for concentrations higher than 10000 cfu/g .
Regarding recommendations for surface cleanness, we have rules for surfaces in contact with cold stored foodstuffs [11], which states that the microbiologic contamination should be lower than 1 cfu/cm^2 to be in the excellent class, a microbiological contamination lower than 10 cfu/cm^2 to be in the good class. Clean Rooms for pharmaceutical industry have four classes: A - 0.04 cfu/cm^2, B - 0.2 cfu/cm^2, C - 1.0 cfu/cm^2, D - 2.0 cfu/cm^2 [11].

DESCRIPTION OF THE HVAC SYSTEM

This study was conducted in a 9 years old office building in Lisbon area, with a VAV ventilation system. The study was carried out on the east side of the building with the air handling unit is placed at the roof, figure 2. This system is 100% outdoor air (without recirculation). The air handling unit has a nominal capacity of 45 690 m^3/h and is connected to a duct 1.5 m × 0.9 m. In this air handling unit a two step filtration scheme is applied, with filters of class G3 and F9 [12].
The system is made with galvanized steel ducts. At the exit of air handling unit and in the vertical shaft the duct has rectangular cross section. For air distribution in every floor one main duct spiral oval is used, (figure 3). The connection of VAV box to the oval duct is made with oval or round ducts. The connection of air supply diffusers to the VAV box is made with flexible ducts. Inside the VAV box a sound insulation material is used.

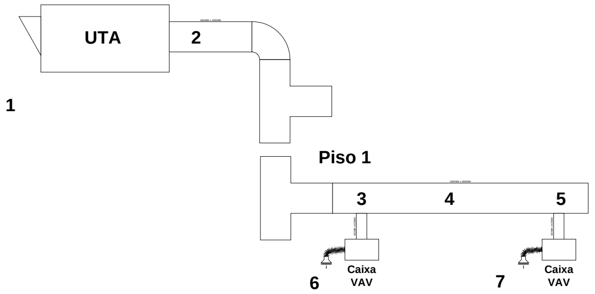

Figure 2. Schema of supply air duct and measurement location

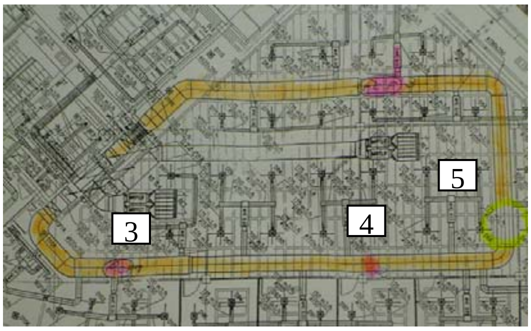

Figure 3. Supply air duct and measurement location

METHODS

To appreciate the effect of 9 years of HVAC system use on the quality of supplied air to the rooms, and to test the effect of HVAC cleaning without disinfection, concentrations of bacteria, fungi and particulate matter (dust) were determined in air and inner surfaces of air ducts, before and after cleaning.

To obtain the worst possible scenario, we studied the first floor of the building, because it's bigger the distance to the air treatment unit and dirtiest ducts are expected.
Figure 2 and 3 represent the measuring points for air determinations (1 to 7) and inner duct surface determinations (2,3,4 and 5). The selection was made in order to obtain information about the effect of HVAC components in the quality of supplied air and also to study correlations between surface and air pollution.
To obtain an estimation of airflow, we measured air velocity profile at the same points where the air contamination was assessed.

Surface Contamination

To collect surface samples for microbiological or dust analysis, the HVAC system was turned off and holes were opened in ducts at selected locations to give assess to inside of ducts. The measurements were done on inner surfaces near the holes.
Sampling of viable microbiological samples on the surfaces was performed using bioMerieux contact plates filled with the appropriate culture media, Count-Tact (ref. 43 501) to collect bacteria and Count-Tact Sobouraud Glucose Chloranfenicol with neutralizer (CTSCD, ref. 43 580) to collect fungi. At least two samples at each measuring point were taken.
The collection of surface particulate matter (accumulated dust) was done according with the APIRAC procedure [8], using the vacuum test method. It was used the ROBImech equipment, with a flow rate of 25 l/min. Filters were weighted before and after collection, at the laboratory.
The amount of dust inside ducts is not homogeneous, presenting higher concentrations near the joints of the steel duct. The methods for measuring the accumulated dust do not specify the sampling places. Therefore, to correct this possible source of error, we performed collections in both "clean" and "dirty" points.

Air Contamination

The collection of total particulate matter in the air was performed using pumps calibrated for a 2 l/min flow rate and membrane filters previously weighted. The collection time was approximately 12 hours.
Sampling of viable microorganisms inside the ducts was performed using impact samplers. Andersen-N6 impactor, calibrated to 25.32 l/min, was used inside the ducts and a MAS 100 impactor (Merck) calibrated to 100 l/min used to collect microorganisms at the terminal devices in unoccupied spaces because human beings are important sources of bacterial contamination.
To culture fungi Malt Extract Agar with chloranfenicol was used and Trypticase Soy Agar was used for bacterial counts. Plates were incubated at 27ºC and 37 ºC for fungi and bacteria respectively.
Inside the ducts the air velocity was measured using an Airflow thermo-anemometer model TA 5.

RESULTS AND DISCUSSION

The measurements before system cleaning were carried out on the 18th and 19th of March 2005 (figure 4 and 5 – Before Clean). The measurements after cleaning were performed on the 5th and 6th of May, 2005 (figure 4 and 5 – After Clean).
In figure 4 the results of measurements in air are presented, namely: bacteria, fungi and total particulate matter. In figure 5 the results of measurements in duct surfaces before and after

cleaning are shown. The results of bacteria and fungi are the average of two measurements, while the particles are the average of 12 h measurement.

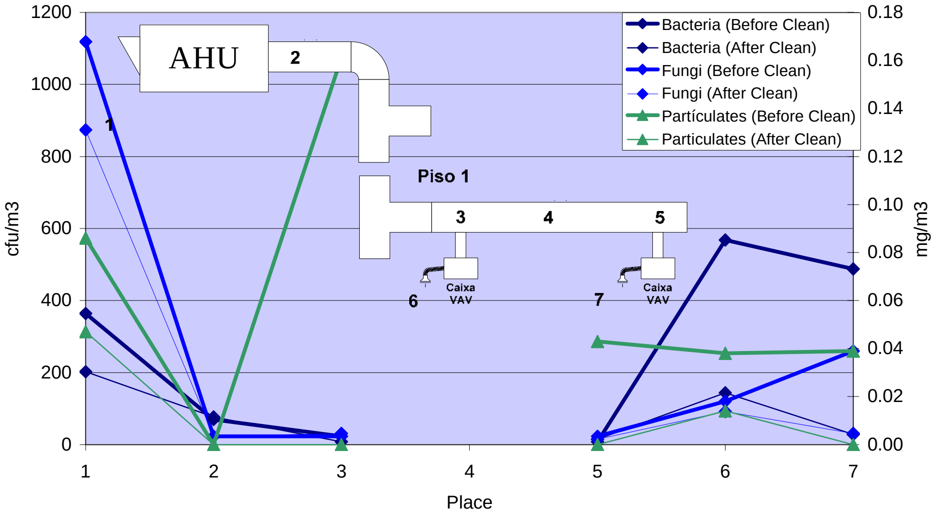

Figure 4. Measurement in air

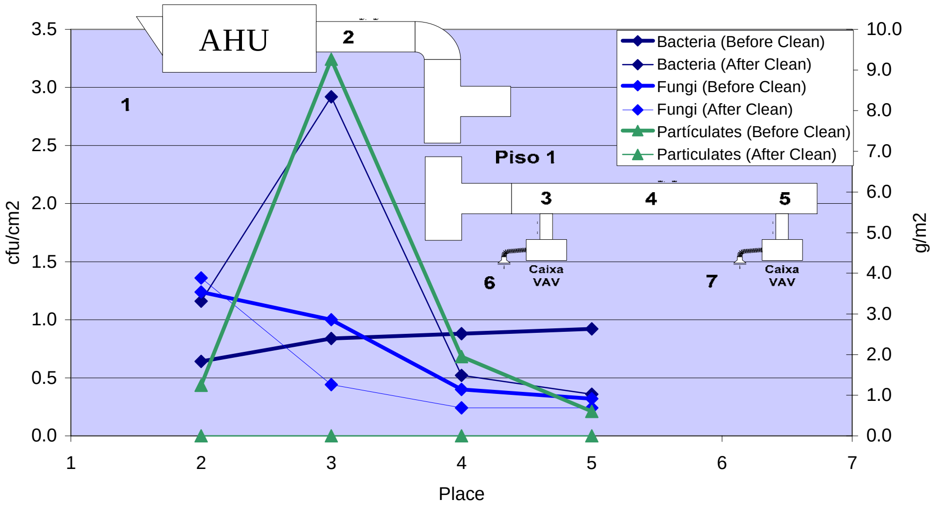

Figure 5. Measurements in duct surface

General appreciation

The duct network of the HVAC system under study was relatively clean, with a surface dust concentration under $2g/m^2$ and a microbial concentration inside the range classified as excellent for surfaces in contact with food, class C for "clean rooms" and close to the lowest levels referred by Luoma et al [4].
In the following paragraphs we present the main conclusions regarding the effect of mechanical duct cleaning in supplied air.

Air handling unit

Comparing the results obtained outdoors (A1 point) with the results obtained beyond the air handling unit (A2 point), it's possible to conclude that filters were effective in the removal of contaminants, as we can see for the reduction of pollutants (microorganisms and particulate matter).

The results obtained before and after cleaning (points A1 and A2) were not significantly different, despite the cleaning.

Ducts

To appreciate the effect of duct network in the contamination of air, we compare the results obtained beyond the air handling unit (point 2) and 1st floor entrance (point 3) and between point 3 and point 5 the remote point of duct system.

There is no difference between the results obtained for microbiological contamination in point 2 (after passing the air handling unit) and 3 (1st floor entrance), although there is an increase on particulate matter concentrations (from not detected to 0.16 mg/m^3). This increase on particulate matter in air may be associated to the displacement of particles settled at the bottom of the vertical shaft

Analyzing the results obtained in surfaces we verify that bacterial and fungal concentrations are almost constant between samples collected after passing the air handling unit and 1st floor entrance point. However there is a larger dust deposition at the entrance of 1st floor (10 g/m^2) than after passing the air handling unit (1.3 g/m^2). The observed difference may be due to the air flow bend near the collection point, which promotes the settlement of particles, and also to the large particle concentration in air in this zone.

Between the entrance in the 1st floor and the remote point (points 3 and 5) a reduction in particle matter concentration in air is registered (0.16 mg/m^3 to 0.04 mg/m^3), with almost the same concentration of micro-organisms. The reduction of particle concentration in air is associated with the reduction of average air velocity along the duct, which promotes the settlement of heavier particles. Regarding surface concentration, we detect the same changes as in air, *ie*, the micro-organism concentrations are almost constant and there is higher concentration of dust at the beginning of duct (10 g/m^2) than in the most remote point (2.0 g/m^2).

Analysing these results we conclude that there isn't a direct correlation between particle and micro-organism concentration in duct surface, because the micro-organisms remain almost constant in several collection points, while the particle concentration presented a large variation. Regarding particle concentration in air and in surface, there is a direct correlation.

With those results we can see that mechanical cleaning of duct surface is effective to remove particles, since we obtain a substantial reduction in surface particle concentration after mechanical cleaning and we also obtain a large reduction of particle concentration in air. After mechanical cleaning, the surface particle concentration complies with APIRAC criteria (≤1 g/m^2) and also with NADCA criteria (0.075 g/m^2), in spite of the large differences in measurement method of APIRAC and NADCA.

The micro-organism concentrations present slight variation (before and after mechanical cleaning). Because of the low micro-organism concentration (surface concentration of 0.2 and 1.4 cfu/cm^2 and concentration in air of 8 and 70 cfu/m^3) it seems that the impact in air quality is low and therefore chemical disinfection can be avoided.

In this study, we find that the collection point in duct surface could have an impact on the results obtained. Points very close to duct irregularities (the bending zone in spiral duct, joints, etc) could have large differences in dust concentration. The changes in dust deposit, however, didn't correspond to a change in micro-organisms concentration. We also notice that the higher micro-organism concentrations were found near the places of lower particle concentration.

VAV box and diffusers

To assess the impact of VAV boxes, flexible duct and diffusers we compared the results of measurement points 3/6 and 5/7.

From the results obtained at these points, we can see a large increase of micro-organisms in air between the main duct of the room and the exit at air diffusers. This indicates that VAV boxes, flexible duct and diffusers have good conditions to the germination of micro-organisms (namely bacteria) that are drawn by air flow. Regarding particle concentration between collection points 3/6 we can see that there is a settlement of particle in these components, which could create media for the development of micro-organisms if the humidity conditions are adequate, because the availability of water is the driving force in the growth of micro-organisms.

In this case study, with mechanical cleaning we detected a large decrease of particles and micro-organisms in the air supplied indoors. If we look at contamination classes of air [13], the air supplied before cleaning would belong to the intermediate contamination class and after cleaning would be classified as very low pollution.

The main focus of contamination of air supplied to the building was the branch VAV box and the flexible ducts, which were very dirty.

CONCLUSION AND RECOMMENDATIONS

In this case study the contamination of one VAV system and its impact on the air quality supplied to the building before and after mechanical cleaning, was analysed.

We found that after 9 years of use the system could be considered clean, even though this was not clean after construction. The concentration of micro-organisms and particles was generally low, and some points presented higher particles concentration but that still didn't compromise the air quality.

The main focus of pollution are the VAV box and/or flexible duct. Before the cleaning an air concentration of bacteria of 500 cfu/m^3 was measured and after cleaning this concentration decreased to 28 and 140 cfu/m^3. In spite of this reduction, after cleaning, these components remained the main source of air pollution.

For the dirt in duct surface we didn't found a correlation between particle concentration and micro-organism, because micro-organisms concentration remains almost constant from dirty collection points to clean collection points. However, we obtained a correlation between particle matter in air and duct surface.

Regarding the criteria for clean ducts, we found that the 1 g/m^2 limit [11] to start cleaning duct system is very stringent, because in this system (which presented higher values in some places) the duct was not source of pollution of air. In absence of other information, we think that it could be reasonable to adopt the class P2 criteria (5 g/m^2) [9] as the limit beyond that which cleaning the ducts is needed. As criteria to asses the cleanliness of the cleaning operation we could use the criteria of 1 g/m^2, or a lower value, for example 0.1 g/m^2, measured with the APIRAC method, because in this building both criteria were satisfied. Regarding the micro-organisms concentration in duct surface, we think that the contamination risk is low for ducts that comply with class D of pharmaceutical industries (2 cfu/cm^2) [11], taking into account the results obtained and the Norwegian recommendations [4]. For concentrations above this limit (2 cfu/cm^2), we think that a study of the impact of surface contamination in the contamination of air supplied to the building should be performed, before cleaning or disinfecting the ducts.

We found that the vacuum test method could be adopted as measurement method for the assessment of duct cleanliness regarding particles and that the mechanical cleaning is effective in the removal of particles and micro-organisms.

With these results we also show that before a complete system cleaning, some measurements should be done to find potential sources of pollution and clean only the contaminated components, cutting maintenance costs.
In the course of this work we also noticed the importance of the dirt left from construction. It is quite important to follow rules to protect HVAC during construction and probably do some cleaning after construction and before turning on the HVAC installation.

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Duct Cleaning: The Good, The Bad...and The Need for An International Consensus Standard

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SUMMARY

Cleaning ventilation system ductwork has been a topic of controversy for several years. Guidance documents are available for duct cleaning, but uniformity in application of remediation and cleaning methods is inconsistent in many cases. Means and methods for cleaning range from high-volume negative pressure vacuum cleaning of residential ductwork ...to use of sanitizing agents on the interior of galvanized ducts...to ultraviolet germicidal irradiation for treatment of re-circulated air and control of infectious viral aerosols in operating room suites. A comprehensive visual inspection of the components of the building ventilation system and a risk assessment to determine the nature and magnitude of contamination form the basis of effective remedial cleaning. This presentation describes a proposed industry standard of care for duct cleaning and a quantitative method to determine an acceptable level of cleanliness. The information presented underscores the need for an international consensus standard for duct cleaning.

INTRODUCTION

Over the past decade heating, ventilation and air conditioning (HVAC) system cleaning and decontamination to remove dust and biological growth has expanded into a large industry serving both residential and commercial building markets [1]. Guidelines for duct cleaning are available in Japan, Sweden, the United Kingdom and the United States, but are routinely violated by many contractors. Decisions to remediate ductwork are frequently driven by financial motives and are not always based on sound judgment. Improper remediation of HVAC systems can cause degradation of indoor air. Duct cleaning is often performed without a comprehensive visual inspection due to limited accessibility to HVAC system components, resulting in a failure to properly identify sources and implement corrective actions prior to cleaning.

The focus of this presentation is on mold-impacted and/or particle laden ductwork. HVAC systems are especially conducive to mold growth because they draw supply air, which usually contains fungi and moisture, into a building containing numerous organic materials that provide food sources for mold growth. The remediation of HVAC systems is an extremely sensitive undertaking due to the potential of disturbing and distributing spores or dust throughout the ventilation system.

HVAC system ductwork is an air-conveyance system that requires diligent cleaning methods and a quality assurance process to protect the health of building occupants. The purpose of this presentation is to provide guidance in the proper assessment and remediation of contaminated ductwork and to encourage development of international consensus standards.

METHODS
Building Ventilation Ductwork Components and Characteristics

In a typical building HVAC system, the fan pulls air from the occupied space through return air grilles into ductwork...then through the filter, heating and/or cooling coils and supply ductwork...then back into the occupied space. Various types of ducts and ductwork materials are available. Basic types include: straight metal sections; spiral-wound; flexible and expandable; and plastic made of polyvinyl chloride, reinforced fiberglass, or acrylonitrile butadiene styrene. Ductwork materials include: fiberglass, metal with fiberglass interior, metal externally wrapped with fiberglass, and insulated flex duct. The duct type and material will depend on the type and age of building and ventilation system, the nature of occupancy, the cost of materials and ease of installation, corrosion, temperature and other factors. Galvanized steel is probably the most common duct material; circular, rectangular and oval ducts the most common shapes. Some types and shapes of duct are capable of retaining more particulates regardless of their design velocity. For example fiberglass duct will generally accumulate more particles because of its rough, porous interior surface, compared to galvanized steel which has a relatively smooth surface except at section joints and seams. Duct manufacturers publish friction loss data, based on the surface material, which is useful in determining the potential for particulate loading on interior duct surfaces and the corresponding frequency recommended for maintenance and cleaning. The design of the ductwork will also determine the potential for interior particulate loading. Elbows, junctions, branch entries, and expansions and contractions are areas where there is greater potential for excessive surface deposits compared to straight sections.

Measurement of static pressure losses in a duct provides useful data to evaluate if particulates are being trapped in the duct. A pitot tube connected to an inclined manometer is the primary method of measuring duct static pressure. To reduce static pressure losses in ductwork design: (1) Branch entry angles should be designed at 30º – with a 45º maximum. (2) Expansions and contractions should be gradual – for example, one centimeter of duct diameter change for every five centimeters in length. (3) Rectangular duct should be as square as possible. (4) Seams on spiral-wound ductwork should be sealed with high or low pressure seals and pressure (leak) tested.

To evaluate the potential for particulate loading within HVAC ductwork, it is important to understand airflow in ventilation duct and duct velocity profiles. Airflow in ventilation duct is turbulent, not laminar. Turbulent airflow, in comparison to laminar airflow, is a function of the Reynolds Number – a dimensionless parameter represented by:

$$Re = \frac{VD\rho}{\mu}$$

Nomenclature:
V = air velocity (cm/sec)
D = duct diameter (cm)
ρ = air density (g/cm³)
μ = air viscosity (poise)

If $Re < 1200$: Airflow in duct is laminar, free of eddies and swirls, direction is predictable

If Re 1200 – 3000: Transitional airflow depends on acceleration or deceleration

If $Re > 3000$: Airflow is turbulent

Re in ducts is usually 100,000–1,000,000; therefore, it is always turbulent [2].

Dust in ventilation ducts usually has a very minor effect on air flow rates. However, heavy dust accumulations on critical components of the HVAC will limit air flow, for example dust on air supply diffusers, dampers and fan blades. Velocity profiles can help define the potential location of particulate accumulations in ductwork, and are best determined by taking duct velocity Pitot tube traverses. Velocity distributions inside ductwork are not uniform across the area of the duct due to factors such as obstacles in the airflow path and changes in airflow direction. Velocity is always zero at the wall surface of the duct. Velocity in duct must be corrected for the moisture content in the air, temperature and altitude. For air with no industrial particles, duct velocities of 9 to10 mps are considered optimum in main ducts, considering initial and operating costs, noise control, and naturally occurring dust transport. Air velocities in branches near room registers should be lower than in main ducts.

Sources of HVAC System Contamination

The cause(s) of mold growth must be identified and corrective action taken before remediation of either the building or the HVAC system is undertaken. Any identified HVAC deficiencies must be corrected. Sources of water intrusion, moisture accumulation, and/or surface condensation must be resolved prior to remediation. If the HVAC system and building problems are not resolved, recurrence of microbial amplification should be expected. Source identification and control form the basis for an effective and successful remediation.

Over 52% of all indoor air quality problems are caused by inadequate or improper ventilation [3] including poorly designed, installed, operated and maintained building ventilation systems. In some cases there can be widespread mold growth in a building, without a recognizable source of water intrusion, caused by improper moisture control in ventilation design. Proper building pressure management, ventilation system design engineering, and humidity control are critical elements to prevent mold and degradation of indoor air. Whenever the author performs an indoor air quality investigation, he frequently finds the source of the problem in one of “4 Ps” – pollutants, people, pathways or pressure. The latter two areas involve the ventilation system.
Common problems include:

- Poor air filter maintenance and efficiency – causing particulates to build up in ducts and at supply diffusers. Poorly maintained or improperly sized filters allow particulates (including ambient aerosols and those from local sources) to be distributed throughout a building, resulting in an increased risk of microbial contamination. When filters become clogged, the fans use more energy to operate and move less air, causing a detrimental effect on cooling, humidity control and proper air distribution. Filters in HVAC systems should be of the highest grade compatible with the system and the air handler fan – with a filter glass F6 to F7.
- HVAC units that are oversized for an area or a building – resulting in overcooling, system short-circuiting, improper dehumidification, condensation build-up, excess humidity and mold growth.
- Pressure differentials created by ventilation systems – notably if the system operates under negative pressure…as opposed to ideally operating under slightly positive pressure.
- Surface condensation in ductwork immediately downstream of cooling coils caused by moisture wicking off the coils. Condensation also occurs when the temperature of the surface of the ductwork falls below the dew point of the surrounding air. Particles not removed by the filters can collect on surfaces in the HVAC system resulting in microbial growth.
- Surface condensation on the exterior of sheet metal ductwork – caused by water vapor in the air; improper dehumidification; high relative humidity; lack of air circulation; openings in the building envelope; and poor indoor climate control.

- Duct leakage – due to loose-fitting joints and connections; improperly fabricated seams; physical damage to crawl spaces or attics; and leakage of return ducts below floor slabs allowing soil gases and moisture entry into ductwork.
- Use of fiberboard duct in warm, humid climates.
- Ductwork installed in vented attic spaces, especially in warm humid climates.
- Improper location of outdoor air intakes (OAIs) – for example, near exhaust air from kitchens, laundries, bathrooms, parking areas, loading docks, or dumpsters; or unprotected from birds and weather. Rooftop OAIs are especially vulnerable to bioaerosol sources from cooling towers, building exhausts, and standing water.
- Cooling towers providing heat transfer for the building's ventilation system that contain microbial growth on wet surfaces. Regardless of the recommended separation distance between the cooling tower and the OAIs (7.62 m), fastidious maintenance and treatment programs should be implemented to control growth of *Legionella* and other microorganisms in cooling towers [4].

Visual Inspection of HVAC Systems

Inspection of HVAC systems should be an integral part of building preventive maintenance. Inspection procedures and intervals should be based on building use/occupancy, length of heating and cooling season, building age and type of ventilation system, modifications to original building design, preventive maintenance and condition of the building and mechanical systems, and geographic location. In general, the older the ventilation system or building, the more frequent inspections should be conducted. Inspection intervals should not exceed one year. Geographic areas with climates having high humidity require more frequent HVAC system inspections, because of an increased risk of microbial contamination. Building occupancy also determines minimum inspection requirements. For example, a hospital is held to a higher standard of care than a retail outlet. HVAC inspections should include the AHUs, humidifiers and representative areas of the HVAC system components and ductwork [5]. Duct inspections should include a representative portion of supply system and return system ductwork components. Supply diffusers, mixing/control boxes, reheat coils, return air grilles, dampers, return plenums, and make-up air plenums should be inspected. Inspections should be conducted in a manner to prevent disturbance of settled dust and mold amplification. If mold contamination is suspected, then a qualified environmental health professional, such as a Certified Industrial Hygienist (CIH) with experience in performing microbial investigations, should be contacted to conduct an investigation. The cause should be identified and corrective action taken before remediation/cleaning. Safety precautions should be taken during the investigation to prevent aerosolization of fungal growth and microorganisms and cross contamination of other areas. In all cases, a conservative approach should be taken to protect the health and safety of building occupants. A moisture meter and an infrared thermal imaging camera to detect moisture in building materials, and a borescope (an inspection mirror with an illuminating light source to view spaces interstitial spaces, such as wall cavities and ductwork) are useful diagnostic tools. Procedures to identify problems with ductwork include checking supply diffusers for air movement using smoke tubes. If no air movement is detected, then check the fan motors. Improperly wired fan motors or reversed fan blades can cause fans to move air in the wrong direction. Inspect for closed dampers, clogged filters, open service openings, or ductwork leaks. Other specific causes for lack of airflow in ductwork may include air pressure losses from an inadequate fan motor, improper branch entry angles for ductwork, addition of exhaust ventilation hoods, installation of a duct turn or elbow too close to the fan inlet, and duct losses from friction and turbulence. If an inspection indicates that the HVAC system has excessive surface deposits or microbial contamination, then the system should be remediated.

DISCUSSION

Standard of Care for Duct Cleaning

A standard of care for duct cleaning starts with a comprehensive preventive maintenance program to mitigate conditions that created the need for cleaning. Consider installation of monitoring systems for proper climate control and periodic monitoring of air quality parameters – including temperature, relative humidity, air movement, pressure drop across a filter bank, and carbon dioxide – which are all surrogates for problems with degradation of indoor air. The installation of a manometer can provide an immediate indication of filter condition without performing a visual inspection.

The presence of dust in ductwork does not necessarily indicate microbial contamination. A small amount of dust on duct surfaces is normal. Problems with dust and other contamination in ductwork are a function of filtration efficiency, regular HVAC system maintenance, the rate of airflow, and housekeeping practices in the occupied space [6]. Condensation on the interior/exterior of ductwork, however, creates favorable conditions for microbial growth. If particulates accumulate in or on ductwork and the relative humidity reaches the dew point, then condensation occurs and the nutrients in the dust/dirt will cause microbial growth. Mold growth in HVAC systems and buildings requires timely remediation. Because water availability is a critical factor controlling microbial amplification in indoor environments, prompt correction and elimination of water intrusion/moisture sources are imperative.

Water-damaged or mold-impacted ductwork composed of porous or semi-porous materials – including fiberglass duct board, metal duct with fiberglass interior, metal duct externally wrapped with fiberglass insulation and insulated flex duct – should be removed and replaced under controlled conditions. Prior to removal engineering controls should be implemented. Engineering controls refer to equipment and procedures utilized to isolate a work area and to protect the occupants and the remediation workers – for example, temporary containment barriers, negative pressure enclosures, decontamination practices, specialized exhaust fans equipped with high efficiency particulate air (HEPA) filters, dehumidification equipment, and use of respiratory protection. Shut down all mechanical ventilation and HVAC system air handling equipment in the building prior to removal and remediation. All mechanical ventilation in the contaminated area, except that required to maintain the negative pressure, must be sealed off. All HVAC supply and return air vents, floor drains, doors, pipe chases, risers, and other penetrations within the rooms must be sealed with two layers of 6-mil fire retardant polyethylene sheeting to prevent migration of contaminants to other parts of the building. In general, controls are least stringent for building restoration and greatest for protection of sensitive populations [7]. For example, a greater standard of care is required to protect building occupants during remediation in schools and healthcare facilities.

Unlined sheet metal ductwork that is not physically damaged or corroded should be cleaned using mechanical agitation methods to dislodge attached particulate and debris and convey it to a collection device in a controlled manner. Agitation devices include: cable-driven brush, compressed air, and power water wash systems; pneumatic and electric driven brushes; and hand brushing tools. The collection device should be capable of creating a negative pressure differential between the ductwork and the surrounding area. It must maintain a minimum capture velocity to keep loosened particulate entrained during vacuuming, to prevent particle settling on the interior of HVAC system surfaces while being conveyed to the collection device. Minimum capture velocities depend upon the type of particulate and its aerodynamic diameter, particle size and density. Capture velocities required for dust transport range from

0.25 mps for small particulates in quiet air to 10 mps for large particulates released at high initial velocity into an area at very rapid air motion. The American Conference of Governmental Industrial Hygienists (ACGIH) *Industrial Ventilation: A Manual of Recommended Practice*, 26th ed., provides velocity requirements for contaminant removal.

Dry duct cleaning methods are preferred in almost all cases. Cleaning using water wash systems should be performed with extreme caution to prevent microbial growth. This method should be used only in places where porous insulation materials are not present on internal or external duct lining or in close proximity to the ductwork. Clean contaminated surfaces of non-porous ductwork using HEPA vacuums. Surfaces of the ductwork should then be damp-wiped with clean cloths using a water and detergent solution and HEPA vacuumed a second time. Ductwork components and the AHU should also be cleaned of mold-contamination and particulates. Return air grilles, supply diffusers, fan blades, and blower wheels should be cleaned and restored to their original position. Restorative drying equipment, such as portable power fans and dehumidifiers should be used to completely dry the ductwork after cleaning. The ductwork and HVAC system components should be clean and dry prior to re-starting the system. Biocides should not be used for cleaning ductwork. Biocides are not recommended for remediation because they: do not remove allergens and other metabolites from mold, do not kill all mold spores, and may pose health risk to some individuals.

Particle-laden ductwork composed of porous or semi-porous materials has a large surface area which can trap dust/dirt and absorb water compared to sheet metal ductwork. Procedures for cleaning sheet metal ducts should not be used for cleaning particulates from fiberglass-lined ductwork. Cleaning of lined ducts should be performed according to proven industry procedures described in the National Air Duct Cleaners Association (NADCA) *ACR 2006*: *Assessment, Cleaning and Restoration of HVAC Systems.* Lined ducts require a special type of cleaning to maintain their integrity and prevent damage to the lining. Fiberglass ductwork that shows evidence of damage, deterioration, delaminating, friable material, microbial growth, biological material, water damage or moisture accumulation should be replaced. The components being cleaned must be under a consistent negative pressure differential to the surrounding work area. It is extremely important that the cleaning procedure does not create abrasions, breaks, tears or other damage to fiberglass liner or duct board surfaces. Fiberglass materials that become wet during cleaning should be removed and the ductwork replaced.

Duct cleaning should be scheduled during periods when the building is unoccupied. Negative air pressure must be maintained at all times in the duct cleaning area to prevent migration of particulates and contaminants into occupied areas. Large high-volume vacuum equipment should be used with extreme care because negative pressure, together with limited airflow, can collapse ducts. Use existing duct system openings for cleaning unless the ductwork can be safely dismantled and removed under controlled conditions. Duct cleaning should be performed according to project-specific protocols described in a written mold remediation work plan. The plan should be developed by a qualified environmental health professional, such as a CIH, and detail specifications for scope of work, engineering controls, isolation and containment, proper personal protective equipment, and other protective measures necessary during the course of remediation. Specific protocols should be based on consensus guidelines developed by the Institute of Inspection, Cleaning and Restoration Certification (IICRC) S520 *Standard and Reference Guide for Professional Mold Remediation*; the U.S. Environmental Protection Agency (USEPA); NADCA ACR 2006; the New York City Department of Health; and other organizations. Remediation of mold impacted building materials and contents should be included in the work plan [8]. If the building containing the HVAC system has moderate, large or extensive areas of mold contamination, then HVAC remediation should be conducted after remediation of the building and contents is completed to prevent recontamination of the building's ventilation system. HVAC system remediation should be monitored by a qualified environmental health professional during all project phases, to ensure implementation of project-specific engineering controls and cleaning procedures. Maintaining the integrity of containments must be a high priority throughout the project.

Verification of Ductwork Cleanliness

Verification of ductwork cleanliness by means of a visual inspection should be performed by a qualified environmental health professional after cleaning, prior to HVAC system startup, to protect the health of building occupants. Measuring the effectiveness of cleaning may include a quantitative assessment as a secondary method if the results of the visual inspection are inconclusive. Visual inspection of the ductwork should be performed – to ensure that it is visibly clean, dry and free of contamination – while the containment is still intact. Service openings in the ductwork should be accessed to perform a visual inspection and closed afterward. The author typically performs a crude "white glove test" to help make a simple qualitative determination of cleanliness of non-porous surfaces (such as the interior of sheet metal duct immediately upstream of supply diffusers and downstream of the filter bank). The area within the containment should also be inspected to ensure that contaminated materials, visible dust and debris have been removed. There should be no malodors detected in the remediated area. If qualitative judgment in verifying ductwork cleanliness is not sufficient, then a quantitative assessment should be considered. Measuring the effectiveness of cleaning using secondary quantitative methods is best accomplished by a combination of methods: air sampling, surface sampling, laser particle counting, and moisture measurements. Samples and measurements for each method should be obtained before cleaning is performed to establish a baseline for comparison and after cleaning as part of post-remediation verification.

Air sampling is performed using a calibrated air sampling pump operating at a flow rate of 15.0 or 28.3 LPM. Fungal spore samples should be collected from the ambient indoor air and the outdoor air intake. One sample should be collected from the ambient indoor room air; another at a supply air diffuser; and another near the outdoor air intake. An additional outdoor sample should be taken at ground level outside the building for reference control purposes. Each sample should be collected for 2 to 5 minutes. The microbial samples should be analyzed by an Environmental Microbiology Accredited Laboratory using cultured methods, direct microscopy or Quantitative Polymerase Chain Reaction (Q-PCR) analysis. Although more costly, Q-PCR analysis appears to be the most likely technology to be used in the future because it provides very rapid turnaround and greater accuracy, sensitivity, and precision in identifying microbial species present. Microbial air samples are interpreted by comparing the types, distribution and levels of fungal spores found in suspect areas to non-suspect areas or outdoor control areas. General guidelines for interpretation of fungal spore data specify that indoor levels of mold spores should be less than outdoors. The biodiversity of fungal spores indoors should also be similar to outdoors. Surface sampling for fungal growth should be performed using sterile sampling swabs or tape lifts. A one square-centimeter surface should be sampled in each collection area. Samples should be collected on the interior of ductwork: one from the return and one from the supply air ducts. Samples should also be collected in a non-suspect/uncontaminated part of the building. The surface samples should be analyzed by an Environmental Microbiology Accredited Laboratory using cultured methods or direct microscopy. A general guideline for contamination using surface sampling is a comparison of samples collected from suspect and non-suspect areas. The sampling results should also be compared to those collected prior to duct cleaning. A decrease in spore counts should be observed after cleaning.

Particle counting is performed using a calibrated laser particle counter operating at a flow rate of 0.1 CFM. Particle measurements should be taken at approximately 50% of supply air diffusers and at a return air grille. Three samples should be taken at a representative return air grille. A sampling probe should be used for the return air measurements and an isokinetic probe for supply air measurements. Each sample should be collected for one minute. A printout should be generated from each sample enumerating individual particle sizes in the respirable size ranges of 0.3, 0.5, 0.7, 1.0, 2.0, and 5.0 µm. The data collected should be used to calculate the percentage increase or decrease in particulates measured in the air using the following formula:

$$\text{\% Increase or Decrease} = 100\left(\frac{\text{Supply Air}}{\text{Return Air}} - 1\right) \quad [9]$$

The data should be plotted on a modified ASHRAE filter efficiency performance chart. The data should also be compared to pre-cleaning measurements to evaluate the effectiveness of cleaning efforts. The data is also useful to evaluate the performance of the HVAC system filters and could demonstrate the need for upgrades in filter efficiency. Choice of the appropriate filter and proper maintenance are critical to keep ductwork clean.
The remediated ductwork should be both clean and dry after cleaning. Moisture measurements should be taken on the ductwork and components using infrared thermal imaging cameras or non-penetrating moisture meters. Ambient indoor conditions should be monitored during and after cleaning. Relative humidity in occupied spaces should be maintained below 60% and ideally 30 to 50% to increase comfort and minimize the growth of allergenic or pathogenic organisms. After post-remediation verification is completed and deemed to be satisfactory, new HVAC filters should be installed and the system operated to allow at least eight air changes prior to occupancy.

In conclusion, risk assessment, source identification, mechanical and building corrections, HVAC system cleaning, and post-remediation verification is a course of action that should be incorporated into best management practices for building HVAC systems throughout the world. Properly designed, operated and maintained HVAC system ductwork is paramount to achieve healthy indoor air in order to protect human health.

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Effectiveness of UV Radiation for Reducing *Aspergillus Niger* and *Actynomices* Contamination in Air-conditioning Systems.

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ABSTRACT

The effectiveness of UV radiation lamps in order to reduce fungal contamination of HEPA filters, to extend filter efficacy and to reduce maintenance costs, is experimentally studied by means of a dedicated air conditioning unit. An experimental HVAC system, with HEPA filters and UV-C lamps (λ=254 nm), was built. Two experiments were performed. After disinfection and control of airtightness, the internal surface of the HVAC system was contaminated (1) with *A. niger* spores and (2) with *Actynomices*. Temperature level was 300 K and Relative Humidity (RH) ranged from 30-90%. The results show that the addition of UV-C lamps to HVAC system reduces *A. niger* and *Actynomices* air concentrations; the effectiveness increases with the decrease of RH level in the HVAC system.

INTRODUCTION

The need of reducing the particulate contamination in hospital wards, especially the one biologically active, is of paramount interest for the health of both patients and hospital personnel, in the light of the recent UNI-EN 14644 ([1], [2], [3]) regulations on clean rooms and the associate controlled spaces, and also of regulations UNI EN 14698 [4] and [5] for the control of biocontamination. For the latter purpose several researchers have considered the possibility to adopt UV-C rays (wavelength λ=100÷280 nm) with the germicide lamps located inside the air ducts of the conditioning systems or directly on the roofs, with the purpose to inactivate the microorganisms present in the air flow ([6], [7], [8], [9], [10]).

The germicide capacity of UV-C rays is well known, and is explained by the absorption of such rays into the nucleic acids structure, especially for those with λ=254 nm. Many researchers have studied in detail their mode of action, and the susceptibility of several microorganisms to UV-C ([11], [12] ,[13], [14]).

During the disinfection procedures generally four types of survival curves have been observed for the microorganisms, described by linear-logarithmic graphs, having the log of the surviving microorganisms number plotted as a function of the employed treatment parameter (time of exposure or the doses of UV-C rays).

Such types of graphs are represented through linear curves A, curves with one shoulder B, curves with a tail (two stage survival curves C) or sigmoid curves D ([15], [16], [17], [18], [19]), such as qualitatively reported in Figure 1.

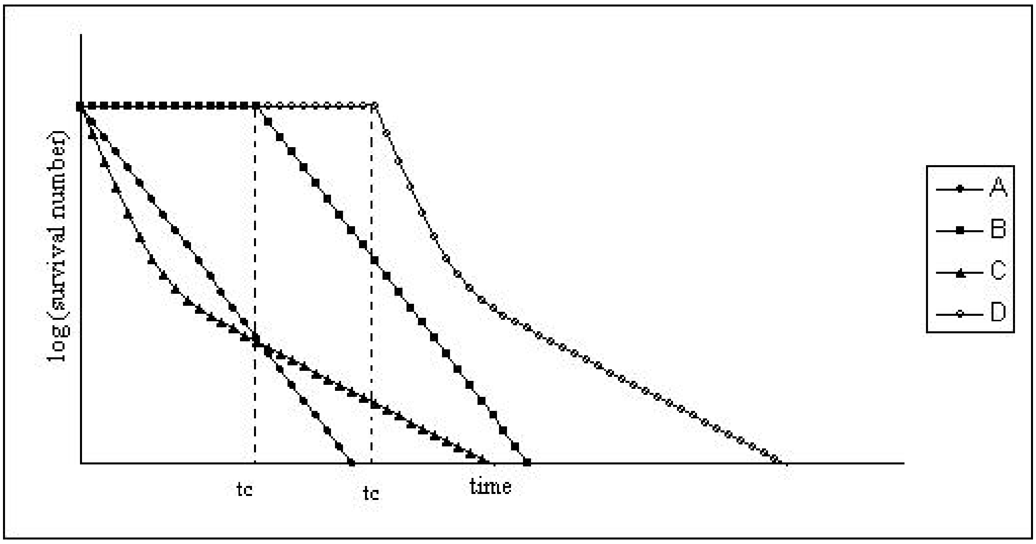


Figure 1- Survival curves (generic for bacteria or fungi):
linear (A), shoulder curve (B), two stage curve (C), sigmoid (D)

The mathematical models which represent such curves A, B, C, D can be expressed respectively as

$$N = N_0 e^{-\alpha t} \tag{1}$$

$$N = N_0 e^{-\alpha(t-t_c)} \quad \text{for } t \geq t_c \tag{2}$$

$$N = N_0\left[(1-F_0)e^{-\alpha_1 t} + F_0 e^{-\alpha_2 t}\right] \tag{3}$$

$$N = N_0\left[(1-F_0)e^{-\alpha_1(t-t_c)} + F_0 e^{-\alpha_2(t-t_c)}\right] \quad \text{for } t \geq t_c \tag{4}$$

where

- N and N_0 represent the microorganisms surviving at time t and those initially present at time t=0 respectively;
- α is a parameter proportional to the applied UV-C intensity and depends on the sensitivity of the microorganism to the UV-C rays exposure;
- F_0 represents the most resistant fraction, characterized by a lower sensitivity to the UV-C rays exposure, in a population of microorganisms, compared to the fraction (1 - F_0) less resistant to such exposure;
- t_c is the time during which microorganisms are substantially not inactivated

Among the different factors influencing the phenomenon of UV-C disinfection there are the air temperature and the air relative humidity ([20], [21], [22]).

The paper showes the results of an investigation on microorganisms *Aspergillus niger* and *Actinomices spp* exposed to UV-C both in the air stream and on the surface of HEPA filter. Inactivation curves at different values of relative humidity (RH) for the air (30%, 60% and 90%) are presented.

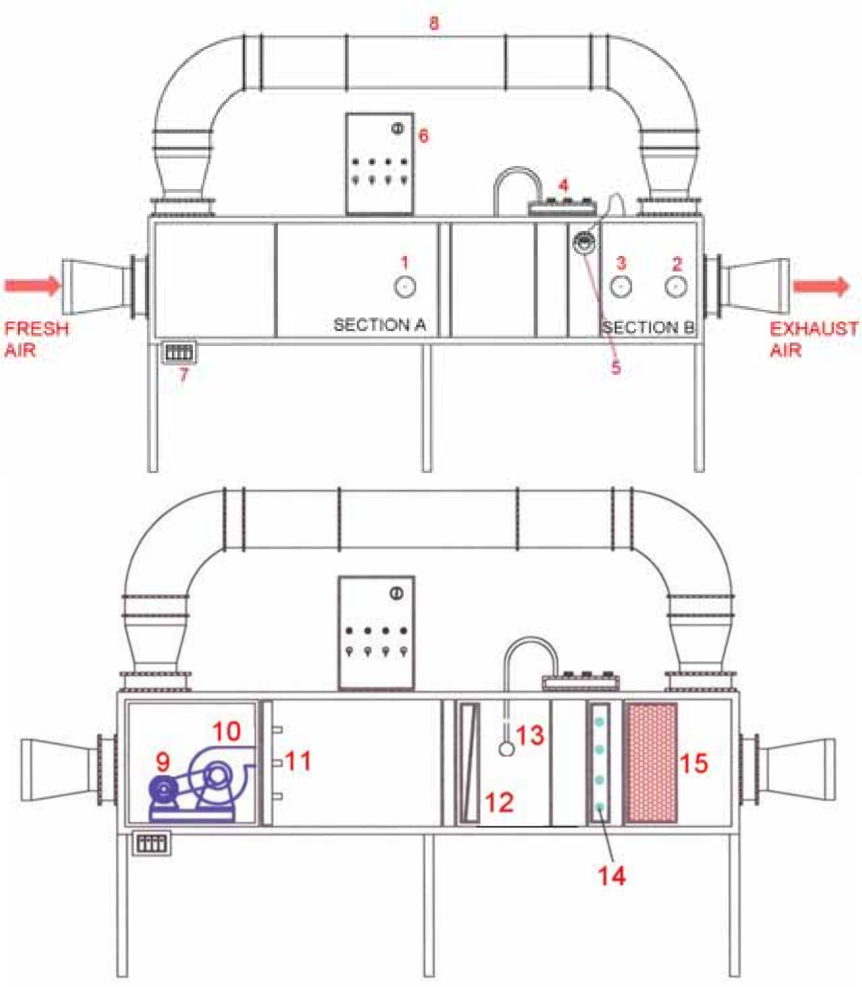
METHODS

Experimental Facility

The experimental apparatus is an air conditioner unit for air treatment (Figure 2), composed by

- A section including the fan together with his electric motor;
- A zone for microbiological charge (Colony Forming Units CFU);
- A section containing the air heating resistor and the steam humidifier with drops separator;
- A section containing the germicide lamps;
- A section containing the filtering media.

For each section there is the possibility of air sampling through an opening equipped with special butterfly valves, in order to avoid losses during the sampling. The box containing the Air Conditioning Unit (ACU) is made by galvanized steel sheets with finished surface; this is to avoid the growth of bio-film on the surfaces and to obtain a good reflection coefficient to increase the germicide action of the UV-C lamps installed.



Figures 2 - Experimental facility schemes. At top lateral view, at bottom lateral section.
1 air sampling intake A, 2 air sampling intake B, 3 inspection intake, 4 humidifier,
5 thermometer, 6 general electric panel, 7 UV-C lamps control, 8 air recirculation duct,
9 electric motor, 10 fan, 11 CFU plates, 12 heaters, 13 humidifier distributor,
14 germicide lamps UV-C, 15 HEPA filter

In the first section of the air conditioner unit ACU, along the air circuit, there is a fan with a static prevalence of 100 Pa and a capacity of 1,500 m^3/h, driven by a monophase electric motor fed at 220V.

In the second section are included

- Three armoured electric resistors, fed at 220 V, as heating device for the temperature control. They are regulated through a temperature probe and can absorb power up to 5000 W.
- The steam distribution for the relative humidity (RH) control of the circulating air into the ACU. It is an inox steel linear distributor which introduces steam obtained by a submersed electrodes humidifier that can reduce the water pollution by 99.8%. The humidifier is controlled by a humidity probe with modulating signal and absorbes a power of 5.8 kW supplied by an electrical transformer providing 8 kg/h of sterile steam. The humidifier with submersed electrodes is fed by the water line of the laboratory. This type of humidifiers operates through the Joule effect using as electrical resistance that of the water which is heated and evaporates. An anti foaming system (AFS) can detect and discard the foam produced in the cylinder. Due to the fact that the steam produced inside the ACU can condensate, the pipe of the distributor is positioned sub-horizontally, with a slope of 2-3% in order to avoid biological growth.

The third section houses the germicidal lamps mounted, with their standard supports for fluorescent lamps, on supports built ad hoc, standing on a squared frame. Four lamps (PHILIPS TUV UV - C G15T8 LONG LIFE, 15 W each) are used. They are tubular quartz lamps, 450 mm long, 26 mm diameter, fed at 220 V, with a specific UV-C emission of 40 μW/cm^2 and they are located on the air flow.

The last ACU section houses the aluminium sledges containing the HEPA filter employed during the experiments, with efficiency DOP of 99.995%**,** mounted at a distance of 150 mm away from the germicidal lamps.

MEASUREMENT PROTOCOLS

The objectives pursued during the experiments with UV-C as air disinfectant are the following:

- To verify the efficacy of the association of mechanical filters and UV-C apparatuses.
- To verify the microbial growth on the mechanical filter surface is avoided.
- To evaluate the opportunity of irradiating a filtering medium at high efficiency.
- To prolong, if possible, the operative life of mechanical filters.

To perform the research phases and obtain correct conclusions, it has been necessary to plan correctly the different steps. Before of each measurement campaign detection of possible air leakages from the plant, clean-up and disinfection of inner surfaces, disinfection of inaccessible parts using “Fumispore” was performed. After the introduction of fungal dose by means of the Colony Forming Units CFU and the start up of plant, three air samples was drawn in order to determine the initial situation. The air sampling was performed by means of a SAS device.

- PRELIMINARY PHASE: empty plant. Determination of the right dose of fungi to introduce into the ACU. Quantification of the natural decay of the microbial concentration due to three simultaneous events: (a) microdispersion outside the ACU, (b) deposit of fungi onto the internal surfaces, (c) lysis of the fungi due to the mechanical collision against the hard surfaces of the plant.
- FIRST EXPERIMENTAL CONDITIONS: plant operated with UV-C only.
- SECOND EXPERIMENTAL CONDITIONS: using Hepa filter inside the plant with UV-C not operating;
- THIRD EXPERIMENTAL CONDITIONS: using Hepa filter and operating UV simultaneously.

During the first, second and third phases, air sampling and sampling from the inner surfaces of ACU have been performed. For air we performed three samples for each sampling interval; the sampling interval was time 0, after 15 min, after 30 min, after 45 min, after 1, 2 4 and 6 hours. For the surfaces, several samples was drawn at the end of the single experiment.

RESULTS AND DISCUSSION

In this paper we report results regarding survival curves into the air samples, whereas those regarding the surface samples will be reported in future work. Also the opportunity of irradiating the filter at medium efficiency will be investigated in future work.

Figures 3, 4 and 5 report the inactivation curves toward time. The curves show the typical two stage shape with tail: it confirms what has been observed by many researchers, in particular Cerf 1977 [23], Smerage and Teixeira 1993 [24], Fujikawa and Itoh 1996 [25] and others ([26]). In general, in every microbial population there exists a small fraction resisting the UV-C radiations or other bactericidal agents. In that case, the mathematical model of inactivation can be represented by the relationship (3).

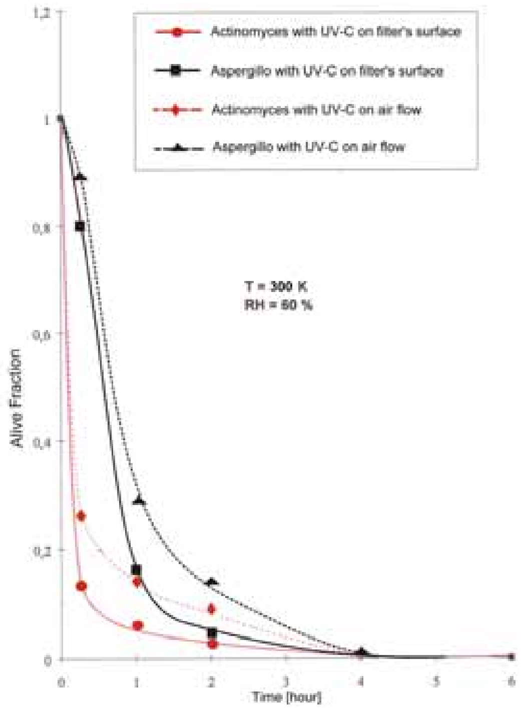


Figure 3 – Comparison of two radiation techniques for *Actinomices* and *Aspergillus niger*

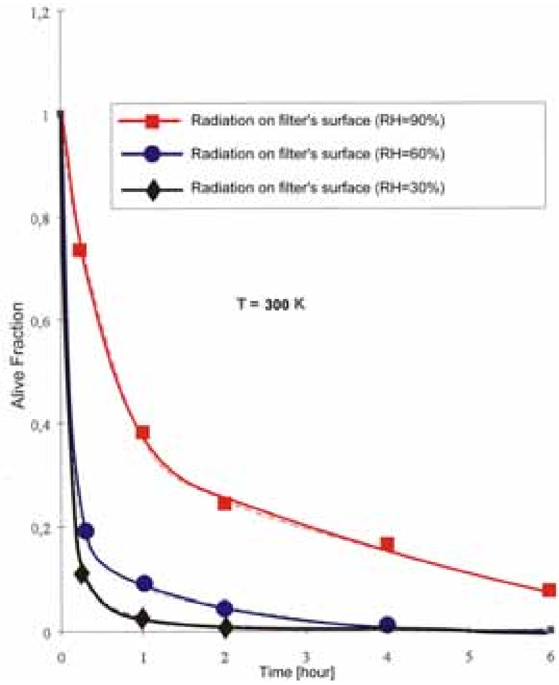


Figure 4 – Inactivation curves of *Actinomices*

Figure 3 shows the inactivation curves at 300 K and 60% RH for *Aspergillus niger* and *Actinomices*. It is clear that *Actinomyces* has a higher sensitivity compared to *Aspergillus niger*. Curves refer to measurements performed when irradiating the air stream and the HEPA filter surfaces respectively. The fraction of *Actinomyces* surviving 15 min later after irradiation of the air stream and of the filter results less than 25% and 15% respectively, whereas the fraction of *Aspergillus niger* it is still 90% and 80%. Irradiation of the filter surface gives better results than irradiating the air stream, as it appears evident from the much faster decay of the surviving population.

Decay curves of *Actinomyces* at 300 K temperature toward variation of RH are reported in Figure 4, while Figure 5 compares the decay curves of both microorganisms toward the variation of the test conditions, RH and modes of irradiation. When increasing irradiation, the fraction of surviving population increases neatly, and becomes much higher when RH is 90%.

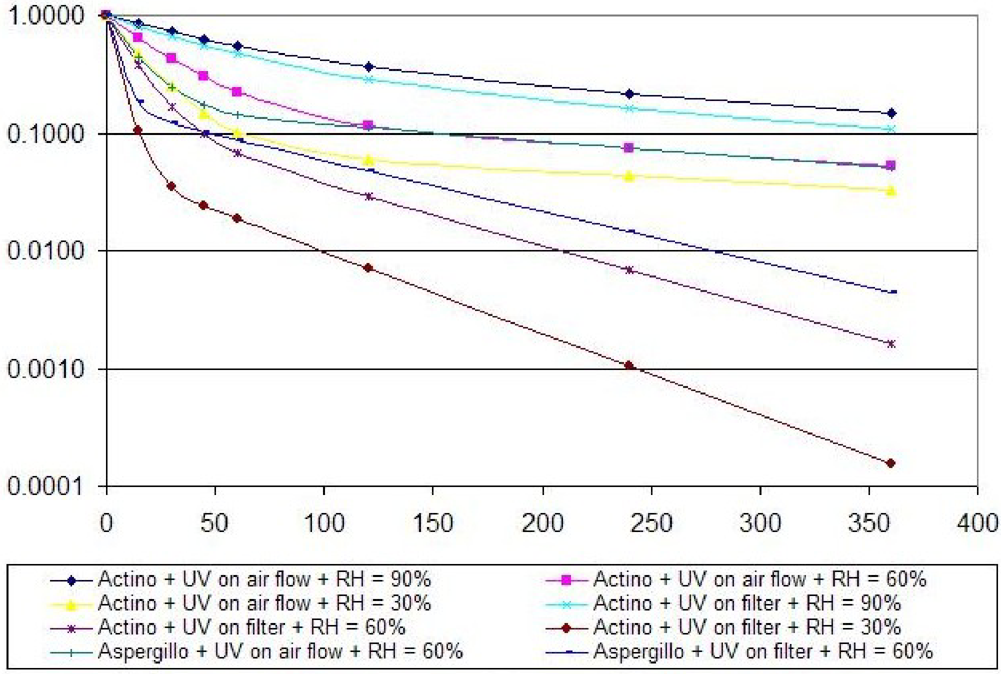


Figure 5 Decay curves of surviving Aspergillus niger and Actinomices fraction for different values of relative humidity and for two irradiation targets (air flow or filter surface)

At this value the differences observed in the decay curves with different irradiation procedures (irradiation of the air stream and irradiation on the filter surface) are quite small. Absorption of UV-C radiation by the steam becomes very high and, therefore, the effect of UV rays is reduced, especially for the more resistant population fraction. The tails do not differ significantly one from the other.

On the contrary, with low RH values (30%), the direct irradiation of the filter surface has amplifying effects on the decay of microrganisms, compared to the other irradiation procedure. Also the resistant population fraction decays faster, as it is demonstrated by the slope of the tail.

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The Prevention of the Snow Entrance to the HVAC-systems

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SUMMARY

Entrance of snow into the HVAC-systems is common problem in the arctic- and sub-arctic climates and may cause moistening or wetting of filters and outdoor air chambers which promotes microbe growth. Blocking the filters also increases pressure drop or even damages the filters. The design features and velocity of air in the outdoor air intake play key role when the problems occurs. To eliminate the snow entrance into the HVAC-systems the face velocity of the outdoor air shall be below 1 m/s at the intake louvre. The pressure caused by wind is one of the external conditions that influence the snow penetration through the intake louvre and some solutions for elimination of this problem are presented. If the snow, however, is entranced to the HVAC-system good design and hygiene control of the intake section of the HVAC-system is important which is discussed more detailed in this paper.

INTRODUCTION

Supply air filters are designed for collection particles such as pollen and other bioaerosols from the outdoor air. The better the filtration is more efficiency the concentration of particles in the supply air decreased. Besides decreased particle concentration in the air, the dust accumulation on the duct surfaces is minimized with proper filtration. Microbes of the outdoor air accumulate to the filters and especially during the warm seasons the concentrations of microbes increased. In normal operation conditions when the HVAC-system is running the filter is dry and air velocity through the filter is high enough that microbes will loose their viability. The life times of the microbes on the filter vary from few hours to few days. Non-viable spores of fungi and bacteria stayed on filter are comparable to the other particles which are not considered to cause adverse health effects. However, spores of the microbes are able to activate when conditions on the filter turns favourable for the microbial growth. These conditions could be possible it the air humidity is high or the filters get wet by the rain water or the melted snow and the supply air fan is shut down. If massive microbial growth occurs on the filters some of the spores will be released to down stream of the filter. The microbial growth is also possible on other parts of the HVAC-system if the moisture is present.

Control of humidity or water entrance to HVAC-system is important because excess moisture will be a risk for microbial contamination of the system [1]. When outdoor air intakes of the HVAC-systems were dry the microbial contamination of the HVAC-systems were not observed. The transportation of microbes via supply air ducts from the outdoor air intake to the supply and indoor air has been reported [2]. Hansen [3] has concluded that dry and clean outdoor air intakes are less likely to be microbiologically contaminated, and thus a good design and maintenance of the air intakes is a key factor for minimizing the risks from potential problems.

When the snow accumulates to the outdoor air chamber of the HVAC-system the surface of the snow can be at the same level than the outdoor air damper, ductwork and filter. In worst case the snow even totally blocks the outdoor air dampers and filters (picture 1). When the filter is blocked the supply air flow will decrease remarkably and the pressure difference in the building may be changed dramatically. This may cause serious indoor air quality problems in buildings like laboratories and hospitals. In some cases the pressure drop over the filter blocked with snow has increased so high that the filter has broken. The broken filter may release dust accumulated to the filter and this dust may spread to the supply and indoor air.

Picture 1. The filter blocked up with snow.

The humidity penetration to the HVAC-system is difficult predict because it is accidental and depends on weather conditions. In Norway 31 HVAC-systems were studied and in 15 of them supply air fans had humidity problems [4]. In our study we revealed a questionaire for the service people of about 270 HVAC-systems in Finland. Serious snow entrance problems existed annually on the average of 10 % (range 7-42%) of the studied HVAC-systems.

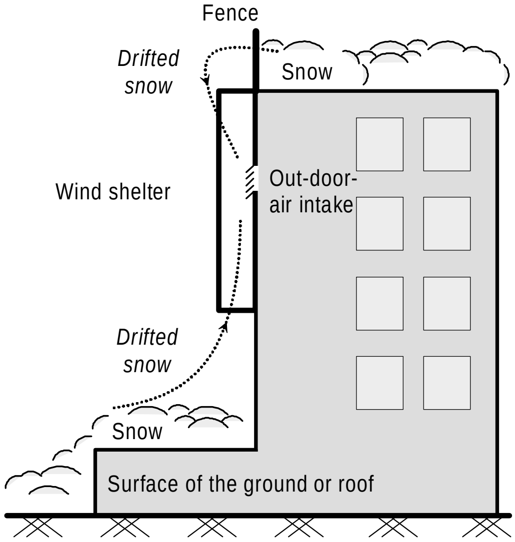
The penetration of the snow or rain water into the HVAC-system's outdoor air chamber is most probable in those where the face velocity of the air is too high and air intake is located incorrectly. The most serious problems in HVAC-systems are focused on the systems where the moisture penetrate through the air intake louvre and due to too high air velocity, too small outdoor air chamber or incorrectly located outdoor air duct the humidity drift to the filter.
The most serious snow entrance problems are detected in the HVAC-systems equipped with an alarm system which alarms when the pressure drop over the filter increases if outdoor air damper or the filter is fully or partial blocked up with the snow. In these cases, the alarms for pressure-drop indicate air flow limitation and malfunction of ventilation system. In the HVAC-systems which had not been equipped with the alarm system the snow entrance was difficult detect. These systems need checks of the outdoor air intake section after every snowfall.

DESIGN OF AIR INTAKE AGAINST THE SNOW ENTRANCE

Outdoor air intakes shall be located so that the outdoor air is as clean as possible. Instructions for location of the intakes have been given for example in Finish building code [5]. Also the maximum face velocity of the outdoor air on the intake shall not exceed 2 m/s.

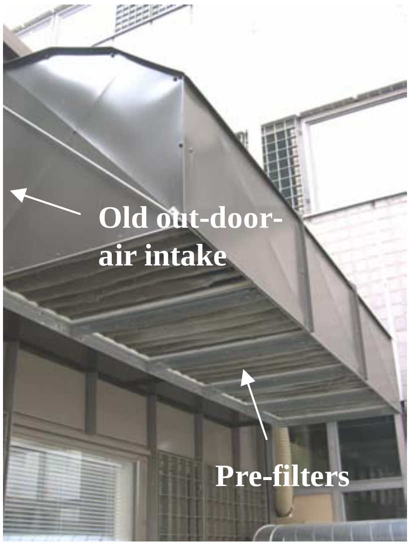
On the snowfall the falling speed of the snow flake is slow 0.2-1.0 m/s [6] and that's why the snow flakes drifts easily to the HVAC-system if the face velocity is too high in the air intake. Because of the slow falling speed the prevention of the snow penetration is more difficult than the rain water penetration and it is recommended that the face velocity of the outdoor air shall be below 1 m/s. If intake is located just above roof or ground, the snow will be driven to the system more easily (Picture 2). A proper location higher from the roof or ground levels prevents snow entrance. Also the wind shelters and fences could be used against the drifted snow (pictures 2 and 3).

The effect of the wind pressure on the penetration of the snow into the HVAC-system has not been studied, but effect might be remarkable because typical annually mean wind speed for example in Finland is 2.0-6.7 m/s and during stormy raining wind speed will be much faster. The pressure caused by wind may at times even exceed the pressure caused the supply air fan and that's why the air intake shall be located so that the wind pressure is low as possible. Good placements for air intakes are in inner yards of the building because the wind speed is typical lower than on outer facades. Also other buildings and plants especially conifers decrease the wind speed. The effect of the wind pressure could be reduced also with the wind shelters (Picture 2 and 3). The wind shelters shall be dimensioned wide enough or they shall be open only from bottom so that wind could not directly blow to the air intake. When the wind shelter is used it's important that the opening of the outdoor air intake is wide enough so that the air velocity and pressure at the opening is not too high.



Pictures 2 and 3. The wind shelters and the fence against the drifted snow and wind pressure.

There are many commercial prevention solutions for the snow and rain water penetration which are based on the mechanical separation or/and melting of the snow. The function of **mechanical louvres** based on the separation of the water and snow particles to the air flow with the labyrinth construction of the louvre. In some mechanical louvres **electric heating devices** are also used which melt the snow and increases louvre's prevention efficiency for snow. The function of the **needle heat exchanger** is also based on the melting of the snow and separation of the snow and melted water to the air flow. On the needle heat exchangers the needed energy is transferred from the exhaust air. **Pre-filters** are also applied for of snow penetration and for example the method in picture 4 is used for prevention of the snow entrance. With this solution one of very difficult snow penetration problem has solved. During the snow fall snow is collected to the pre-filter and after the snow fall snow will drip from the filter.



Picture 4. The pre-filter used for prevention of the snow entrance.

THE MINIMIZING OF THE PROBLEMS CAUSED BY THE MOISTURE AND SNOW PENETRATION

The outdoor air chamber shall be large enough to prevent the snow entrance to the supply air filter and into the HVAC-system which allows to snow settle down to the bottom of the chamber. For dimensions of the outdoor air chamber are not given in any instructions but size of the chamber shall be in relation to the air flow rate. The outdoor airduct and damper shall be located to the upper part of the chamber which increases the efficiency of snow capture in the chamber. The melt water shall be sewed from the chamber and the bottom of the chamber shall slope to the drain. However the enlargement of the outdoor air chamber is often difficult and expensive renovation solution because the chamber shall re-located. The hygiene of the outdoor air chamber shall taken account and all unnecessary material and debris shall be removed from the chamber. The coating material of the chamber's ceiling, walls and especially floor shall be hygiene, easily cleanable and moisture-resistant because on the dirty and moist materials are increases the risk of the microbial growth. For the same reason porous sound attenuation material in the outdoor air chamber is not recommended if there is the risk of the moisture penetration.

The snow entrance to the filters located too near of the outdoor air intake and chamber is more probable because the snow has no time to settle down before the filter. However if there is long outdoor air duct between the outdoor air chamber and filter care shall be taken for drainage of the melted water from duct. To ensure proper drying of the filter it shall install so there is no contact with the bottom of the filter chamber even when supply air fan is turned off. The operation time of the HVAC-system has effect to the moisture and the microbial contamination of the filters and it is concluded that the continuously operated HVAC-system creates less favourable conditions for microbial growth in filters than does the periodically operated system [7].

THE INSTRUCTIONS IF SNOW OR MOISTURE IS PENETRATED TO THE HVAC-SYSTEM

If the snow or moisture entrance to the HVAC-system and filters are filled up with the snow:

1. Supply air fan shall turn off
2. Wet or snowy filters shall remove as soon as possible
3. Filter shall be changed
4. Outdoor air chamber, -damper and other components of the HVAC-system shall check and snow and moisture shall remove
5. Supply air fan shall turn on

ACKNOWLEDGEMENT

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D02
Air-Conditioning systems

Global improvements of the energy efficiency of the European air conditioning stock: results from AuditAC project.

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SUMMARY

Following the continuous growth of the air conditioning market and the European Directive on the energy performance of buildings (EPBD), many changes are expected on the European air-conditioning market. The article 9 of the EPBD takes care about the Inspection of air-conditioning systems promoting possible improvement or replacement of the air-conditioning system. The AuditAC project reached its conclusion with a set of deliverables showing: the advantages from the new inspection system, how to benefit from the combination of air-conditioning (AC) systems audit and inspection, to promote the best practices and to help harmonising the implementation all over the European countries. The results are delivered in form of tools and guides freely accessible from the AuditAC web site. Ten technical guides and tools for owners, energy managers and auditors have been delivered with the aim to rise the awareness of the users about the energy savings subsequent to a “wise” management of the air-conditioning plant, to discover the energy conservation opportunities of an air conditioned building from a full list of possibilities, guiding auditors in audit procedures and supplying advices to identify the actual system performances and forecast energy savings.

INTRODUCTION

In the coming years the stock of Air Conditioning (AC) equipment in use in Europe will partly become obsolete. Most systems will be renovated for the first time (after 10-15 years of operation) and an opportunity exists to introduce higher efficiency systems. Out of the 2.200 Mm² of air-conditioned building area in use in 2010 in Europe, 800 Mm² will date by more than 15 years and will need urgent renewal (Figure 1). A SAVE Study [1] had the result that there is an energy saving of about 50 %. That means: AC – Systems are able to operate with about 50 % less energy.

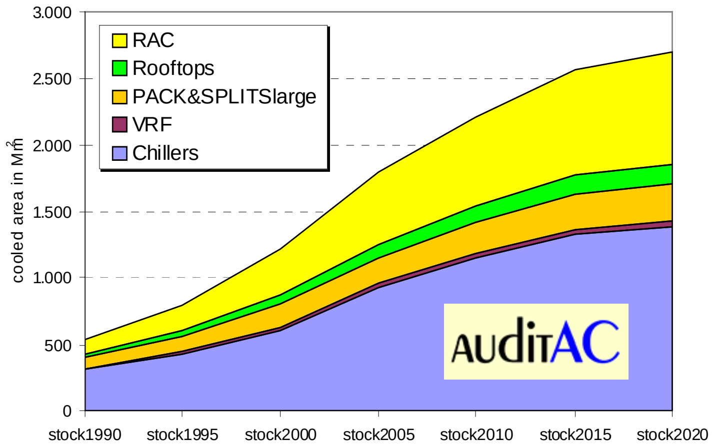


Figure 1 Estimated air-cooled area (Mm²) growth in Europe

This market will be influenced by the Directive 2002/91/EC of the European Parliament and of the Council of 16 December 2002 on the energy performance of buildings (EPBD [2]). According this new directive all over Europe, the building legislation will be changed to create higher energy efficiency within the European buildings.
The Directive takes care about the Inspection of air-conditioning systems:

Article 9- Inspection of air-conditioning systems

With regard to reducing energy consumption and limiting carbon dioxide emissions, Member States shall lay down the necessary measures to establish a regular inspection of air conditioning systems of an effective rated output of more than 12 kW.

This inspection shall include an assessment of the air-conditioning efficiency and the sizing compared to the cooling requirements of the building. Appropriate advice shall be provided to the users on possible improvement or replacement of the air-conditioning system and on alternative solutions.

AuditAC [3] is the short name for the two years project about "Field benchmarking and Market development for Audit methods in Air Conditioning", completed in December 2006. AuditAC was not a project about the implementation of Article 9 of the EPBD. It was much more a project about the questions that comes after the legislation on the national level. Within the focus of AuditAC was to support the European AC market to reach higher efficiency, by taking benefit of good inspections and subsequent audit.
Inspectors or auditors are external or internal engineers (or qualified technicians) in charge of establishing the status and proposing improvements on a plant which is not in failure mode. This is true for Air Conditioning as for any other piece of equipment. In case of failure, a maintainer should be called, not an auditor…Inspection or audit should be the initial stage of any of the following actions: starting or restarting operation of a plant, energy efficiency improvement, study of possible renovation, etc. The owner that decides to go through various stages to reach Energy Efficiency looks for answers to which only the audit procedure can answer: is there an opportunity? How large could be the benefit and the cost of my action? How make the best decision between several possibilities? How check that the measure is effective? AuditAC wants to supply the tools and help to best answer to these questions and make the owner confident facing the investment for energy efficiency.

AIR CONDITIONING ACTORS: HOW TO PROMOTE ENERGY EFFICIENCY

The initial review of existing pre audit and audit methods for air conditioning systems proved they were very scarce. A number of measures in the EPBD do not benefit from experience

feedback, among which the AC plants inspection. Inspection can be considered as a sort of short compulsory audit.
Audit behind inspection has been saw as a sporadically measure, although it should be repeated all along the plant lifetime in a continuous improvement strategy. Nowadays, there is always a factor giving birth to an audit. In the normal life of the plant it can only be a significant failure. For most plants an audit will only take place if the building suffers a deep renovation or ownership change. We see immediately that EU inspection from EPBD aims at introducing more audit opportunities.
If we look at the actors involved in the audit (Figure 2), only the owner or the manager is responsible for the decision but he rarely has the elements for the decision. These elements came from the auditor/inspector after exchanged information with all the other actors. Normally operators and maintainer do not have interest in the energy efficiency (except for specific O&M contracts). Finally best audit is a synergy from the different parts.

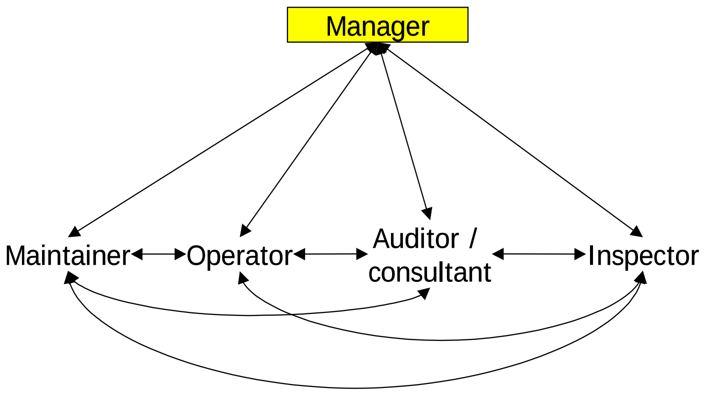


Figure 2 Actors and relationships in a situation of AC audit

So firstly, the AuditAC objectives were to disseminate information about the new measures and to rise the awareness of the AC actors about the benefits of the audit. A set of deliverables in form of technical guides explains which are the key points of energy efficiency in an AC plant (TG 1: Are you sure you are not paying for inefficient cooling?), makes light on the new inspection systems (TG 2: Energy Auditing of Air Conditioning Systems and the Energy Performance in Buildings Directive: what does the new regulation say?) and makes the owners more familiar with the AC aspects: equipments and comfort requirements (TG 3: System recognition guideline for field visit).

A training package (TP) including the information of the project results has been released in order to promote the AC audit. This TP contains 150 slides made in simple and non-technical language, an open version is proposed to users who want to take inspiration and ideas for their presentation and training. A default version is proposed to the large public (Figure 3): it starts from the basic of the air conditioning and the systems summary, then go through the inspection measures and finally through the air conditioning audit and the energy conservation opportunities. The training package has been tested initially internally to the group and published at the end if the project after validation.

Figure 3 Training package: title and index

Furthermore, AUDIBAC interface (Figure 4) allows to retrieve in the AuditAC case studies database the cases that fit more to real problem starting from simple parameters choice (system type, sector building etc.): it is a real shop window for the air conditioning audit! In simple layout the TG 10 documents includes all the AuditAC successful case studies of audited of AC buildings in a simple brochure.

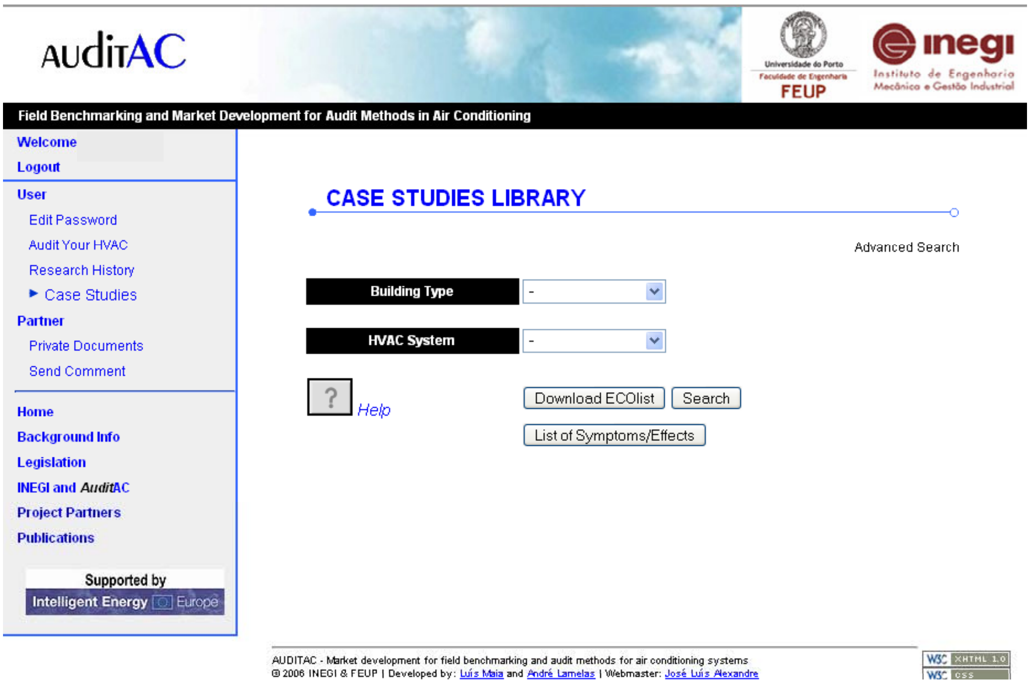


Figure 4 AUDIBAC database screen capture

All these documents are aimed to remove the prejudices of the owners about the air conditioning and rise the awareness of the opportunities to better operation of the systems maintaining or improving the comfort aspects. To improve the knowledge of the air conditioning is a first way to eliminate the barriers to the audit.

THE AUDIT PROCEDURES

In a second phase of the project we focused on the formal presentation of the characteristics of the procedures and the development of the procedures themselves. We made distinction between two levels of audit.

PRE-AUDIT

Pre-audit is a preliminary procedure mainly consisting in: inventory of thermal equipment in place and documentation, treatment of electricity consumption bills, disaggregation of AC data from others. We have found no existing real tool for pre-audit of air conditioning, except some check lists, outdated in some cases or mainly adapted to US practice. That led us to the definition within Auditac of a pre-audit procedure for air-conditioned buildings to support the pre-audit process (TG 4).
The procedure should preliminary investigate the possible of energy conservation opportunities (ECOs): we propose a full list of ECOs including actions at different levels: improvement through actions on Envelope and Loads, improvement through O&M, improvement through Building Energy Management System (BEMS), performance enhancement through adequate improvement works. Energy managers and owners can help the procedure with a continuous follow up of their consumptions and bills. TG 7 propose some simplified methodologies for non professional auditors to observe and track the AC operation and efficiency through the analysis of bills and consumptions: where auditor already find the owner doing a track of its consumption he will not need to do it again.
A simple spreadsheet calculator tool, called AC-cost, allow to have a first savings estimation due to four common actions in term of money savings based on real or estimated running costs. The sheet is easy to use; it helps the user to estimate the requested information when unknown and its results are easy to understand as they are showed in form of cost savings in euros, as gain of annual running cost and payback times in years.
The Customer advising tool (CAT) (Figure 5) allows to rapidly observe the effect of actions on the existing envelope, taking into account the building structure and location, a link allow simply to explore existing audit case in the database AUDIBAC.

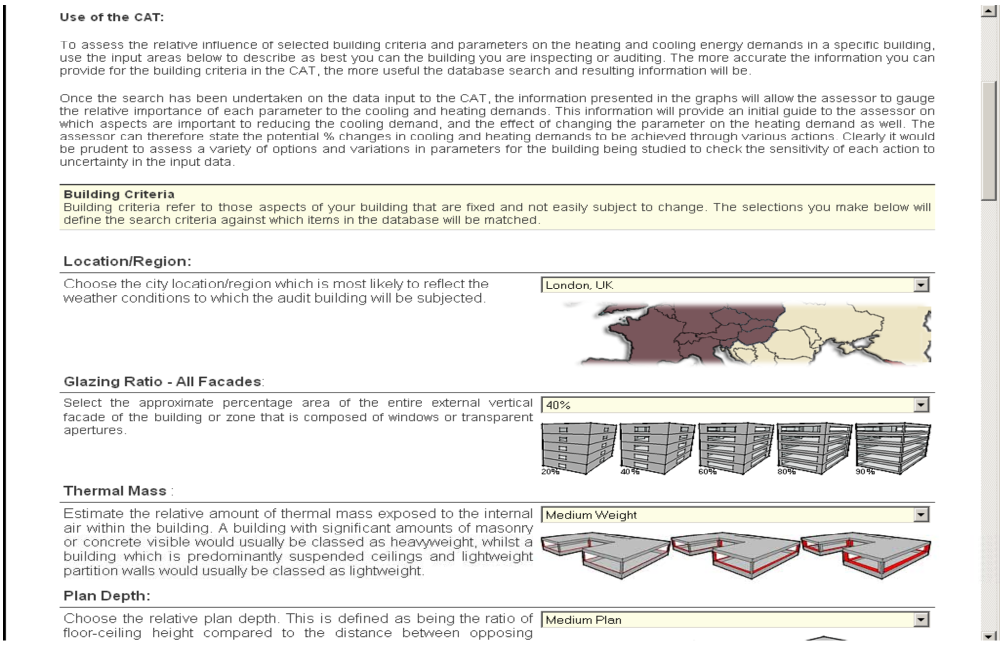


Figure 5 CAT tool screen capture

AUDIT

Thanks to the work on pre*audit, we may enter the audit phase going deeper in the selected ECOs in the previous procedure. There are tools to support the detailed audit of heating plants. We have found no specific tool for auditing air conditioning. Audit duration is a cost issues: more and simpler are the auditor tools, shorter and more efficient would be the audit. In this sense, we proposed several tools to help the auditor in the unfortunate audit work!

The AuditAC-EES model (Simplified load calculation tool defining the consumption of ideal system with the same comfort demands) will lead the auditor to estimate consumptions and the impact of the retained ECOs through the introduction of few parameters from the audited building (Figure 6). The AuditAC-EES model is accompanied by its explaining user's guide.

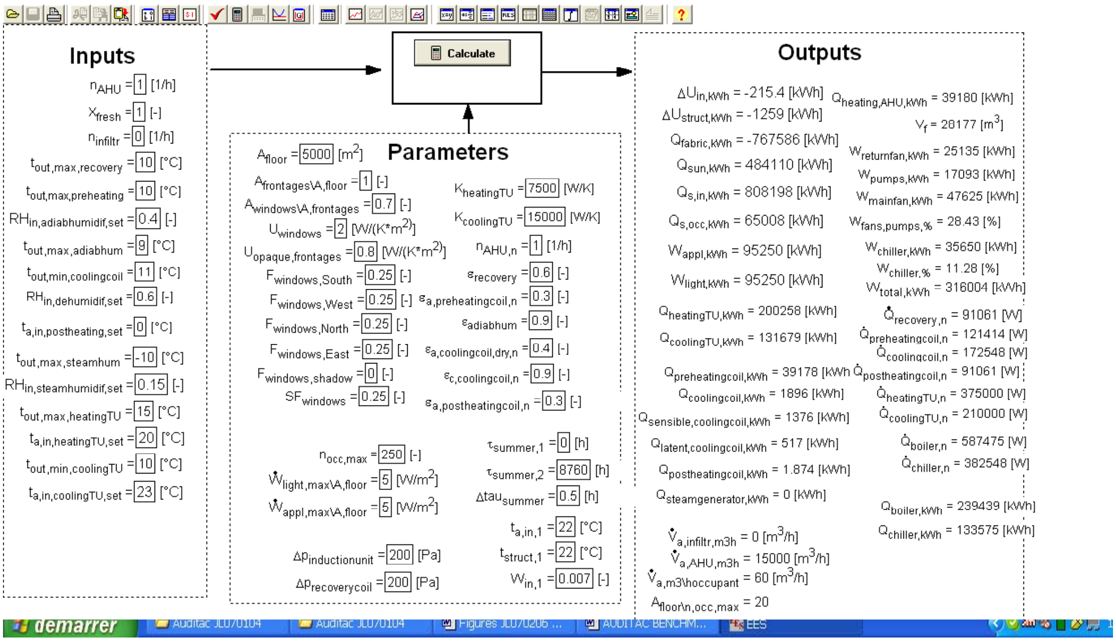
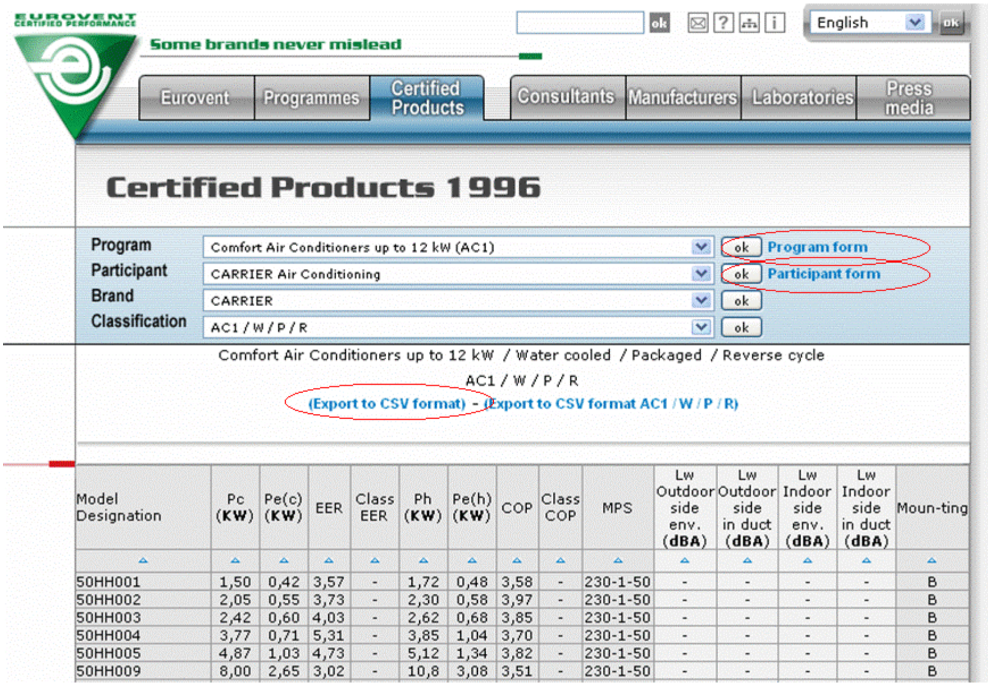


Figure 6 Inputs, parameters and outputs of the EES model

More help can come from the Eurovent-Certification database created in the AuditAC project. This database includes the Eurovent certified directories from 1995 until 2006 (an example in Figure 7). This directories includes certified values of performance for AC conditioners, auditor can retrieve data on the installed equipment when this is unknown, or retrieve data on the equipment of the same period and compare with performances of recent model and technologies looking for a better solution of replacement. How to benefit from this database is illustrated in the TG 6 "How to benefit from the Eurovent-Certification database and to retrieve past equipment data in the audit process".


EUROVENT CERTIFIED PERFORMANCE

Some brands never mislead

English

Eurovent | Programmes | Certified Products | Consultants | Manufacturers | Laboratories | Press media

Certified Products 1996

Program: Comfort Air Conditioners up to 12 kW (AC1) — ok — Program form

Participant: CARRIER Air Conditioning — ok — Participant form

Brand: CARRIER — ok

Classification: AC1 / W / P / R — ok

Comfort Air Conditioners up to 12 kW / Water cooled / Packaged / Reverse cycle

AC1 / W / P / R

(Export to CSV format) - (Export to CSV format AC1 / W / P / R)

Model Designation	Pc (KW)	Pe(c) (KW)	EER	Class EER	Ph (KW)	Pe(h) (KW)	COP	Class COP	MPS	Lw Outdoor side env. (dBA)	Lw Outdoor side in duct (dBA)	Lw Indoor side env. (dBA)	Lw Indoor side in duct (dBA)	Moun-ting
50HH001	1,50	0,42	3,57	-	1,72	0,48	3,58	-	230-1-50	-	-	-	-	B
50HH002	2,05	0,55	3,73	-	2,30	0,58	3,97	-	230-1-50	-	-	-	-	B
50HH003	2,42	0,60	4,03	-	2,62	0,68	3,85	-	230-1-50	-	-	-	-	B
50HH004	3,77	0,71	5,31	-	3,85	1,04	3,70	-	230-1-50	-	-	-	-	B
50HH005	4,87	1,03	4,73	-	5,12	1,34	3,82	-	230-1-50	-	-	-	-	B
50HH009	8,00	2,65	3,02	-	10,8	3,08	3,51	-	230-1-50	-	-	-	-	B

Figure 7 Eurovent-Certification database: example of data for certified air conditioners of capacity less than 12 kW in 1996

The auditor should find less barriers as possible during the procedure: the TG 8 “How manufacturers could help the unfortunate energy auditor”, intended to manufacturers, explains how a component can be used as measuring device is an inexpensive way just adding measurement points in catalogues and how they can help audit improving the documentation of the equipment and their nomenclature. TG 10 and the AUDIBAC tool can one more time support the decision of the auditor to investigate some opportunities on the base of the previous successful case study presented in the database.

ECONOMICS OF RENOVATION AND INSPECTION

When does decision take place? Saving running costs, reliability, what are the decision factors? How to give an energy efficiency orientation to the renovation of AC? The new factor is the EU inspection: the ways to implement it have been investigated. How to integrate it in the life cycle of a plant? TG 9 “How to integrate Energy Efficiency and AC inspection with full benefit in the structures in place” explores the relations between the AC actors and defines the different interests and how to combine it in the effort for the energy efficiency. The aspects of certification and training of inspectors are faced, showing the possibilities in some country. Discussion about inspection has be animated also grace to the estimation of the energy and economic saving of the inspection implementation in main EU AC market countries: figures for an engineer and a craftsman inspection has been presented in many international meeting arousing a new discussions and interest.

DISSEMINATION ACTIVITIES

All along the project the group developed dissemination activities at international and national level. On one hand we combine internal workshops with open national dissemination meeting with professionals of the host country: this allowed us to meet national problems and situations peculiar for each country and to add on the view point of the professional (installers, manufacturers, operators etc). This allowed also to have a more clear idea of the

level of dissemination of the new inspection and certification measures in countries were the measures are not yet applied (no country implemented the AC inspection measures in the 2006, all asked for three years of delay).
On the other hand the group met an international public participating actively to international events and conferences, promoting results and tools all along their development (AICARR congress 2006, ICEEB 06, Klimaforum'06 etc.).
All the presented papers and posters are available on the AuditAC project web site and can be freely downloaded. The group was in charge to deliver also periodically a newsletter (6 in total all along the project) about the development of the project and about the Directive implementation, author outside the project has been invited to cooperate and intervene about EPBD or other EU project EPBD related. Each country was responsible to create a contact list and deliver the newsletter to the national list.

CONCLUSIONS

AuditAC was part of the IEE projects to support the EPBD implementation and success. The objectives more than support the inspection were to develop the audit procedures for the AC, due to the lack of methodologies and tools. On one hand several tools and actions has been prepared to reduce the barriers to the audit promoting successful Case studies of AC audit and disseminating the contents of the audit procedures in order to motivate the owners and manager to face the investment for the AC renovation for energy efficiency. Moreover, a series of tool wants to help the auditor into the procedure to make shorter and efficient the procedure. Deliverables in form of document (Technical guides for owner and auditors), programmes (Tools), disseminating tools (Training Package). The material is freely downloadable on the project site that will be kept operative for two more years after the end of the project (December 2006).

ACKNOWLEDGMENT

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Taking flexibility into account in designing beam systems

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SUMMARY

In a modern office environment, balance is sought between working as individuals and the interaction between employees. It must be possible to appropriately combine various work methods, which means that partition walls and workstations should be flexibly adaptable, in order to best match the business model of each customer. The adaptability of office space is one of the central premises in designing a beam system. The systems must adapt to changed loads and partition wall locations. For a room system, adaptability means taking changes in the supply air flow, cooling effect and throw pattern of the supply air device into consideration. It must be possible to adjust the air flow supplied by the room device, which enables managing the maximum and average velocity in the occupied zone and thus reducing the risk of a draught.

INTRODUCTION

Recent studies have clearly proven the correlation between indoor air quality and the work performance of employees [1, 2]. Similarly, it has been demonstrated that the thermal conditions of the room have a significant impact on the productivity of work [3, 4]. Employee salaries and the potential change in productivity amount to many times the cost of a building technology system. The studies indicate that an investment in a better indoor environment is a profitable one, even with very minor productivity changes [5].

In addition to the indoor environment, the functionality of the workspace significantly affects the productivity of employees. Often a compromise must be made among the needs of the employee, team and organisation when arranging workspaces. Addressing the interaction and privacy needs of employees, both of which are important considerations in organisations, is particularly challenging. In general, it can be stated that, from the perspective of dispersing silent information (views, experiences, intuitions), fully autonomous workspaces do not support the business models of most companies. On the other hand, reducing the autonomy afforded by individual workspaces reduces acoustic privacy, which disturbs concentration.

Organisational changes in most companies are continuous and require flexible changes in work methods and workspaces. The traditional one-person office areas, or cells, and open offices, or hives, seen in traditional offices are today changing into spaces that are more suited to team work, referred to as dens or clubs (Table 1) [6]. In addition to this, information technology contributes to independence of time and location, transforming offices more into meeting places for sharing information. The office space must be utilised efficiently, and therefore a dedicated workstation is no longer deemed necessary for a worker who spends only a few hours a day at the office. Working at several workstations and at customer sites is becoming more common.

Table 1. Adapting space types and business processes in office buildings.

Space	Interaction	Autonomy	Operation	Example
Hive	low	low	customer service	call centre
Cell	low	high	support tasks	financial administration
Den	high	low	team work	media
Club	high	high	expert work	consultancy

The supply air flow required in office buildings is relatively low (1.5–2 l/s per m^2), with the exception of meeting rooms and similar spaces. On the other hand, cooling (50–70 W/m^2) is required for attaining a good indoor environment. A beam system offers the possibility of achieving the above objectives within economical life-cycle cost limits. Therefore, the beam system has become the most common system in office buildings. It should be borne in mind that the various systems should be applied according to the requirements in each space. For example, variable air systems should be selected for meeting rooms and displacement ventilation for auditoriums and common spaces. Thus can an optimal overall solution be obtained with respect to expenses, energy use and environmental impact [7].

This paper discusses the flexibility requirements imposed by changing office processes for a beam system and covers certain solution models that help to improve the adaptability and flexibility of the system as well as the indoor conditions in the room.

FLEXIBILITY IN A BEAM SYSTEM

In Scandinavian design, ventilation beams are typically installed without a suspended ceiling in the room space, in the longitudinal direction. The basis for the system is the column spacing (e.g., 8.1 m), which can accommodate three one-person office rooms or two larger rooms (Fig. 1). This is possible because the beam is located at the side of the room instead of in the middle of the room module.

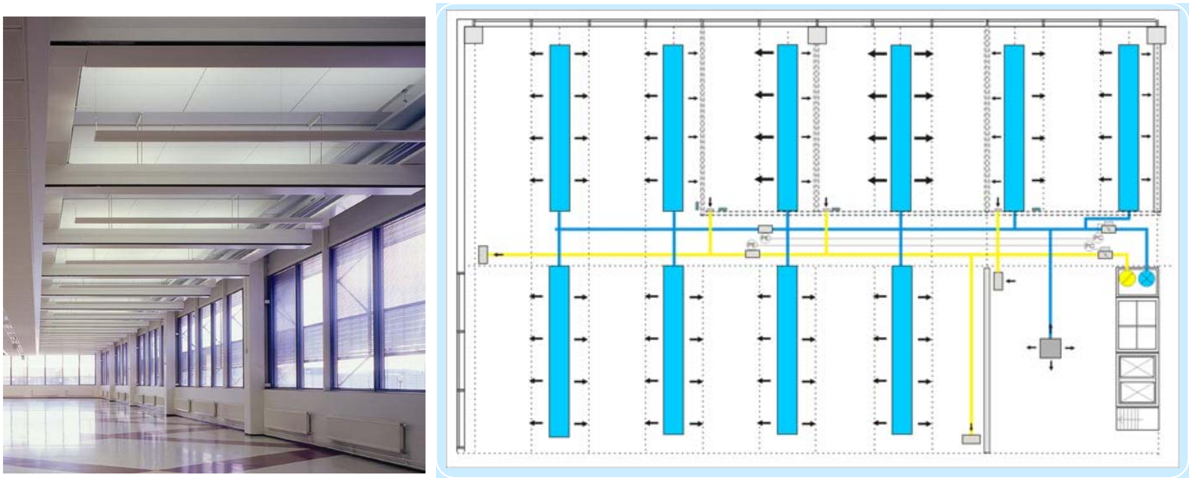
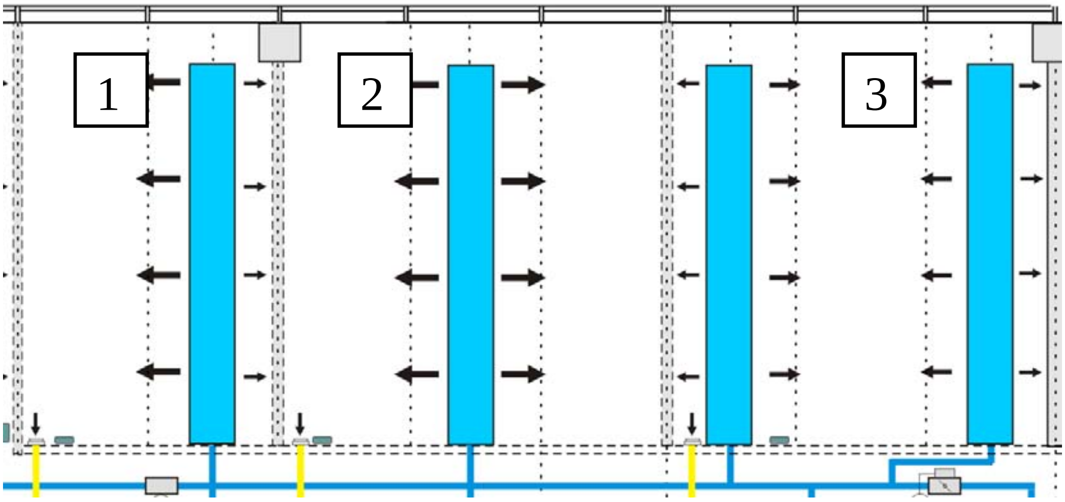


Figure 1. Typical Scandinavian office building module division.

The ventilation ductwork should be designed for constant pressure, which enables demand-based air flow control in meeting rooms. Pressure adjustment is implemented per zone or by individual ventilation device. It should be taken into consideration in the management of air flows in a meeting room that the pressure level in a beam system is 60–

100 Pa and the pressure level of a typical variable air system (MIV) is considerably higher than this. The variable air unit installed in the system must operate at the pressure level of the beams.

When the location of partition walls changes, the size of the room space and the supply air flow of the room device change as well. The typical office module division can result in three different cases: 1) one room device produces supply air for one room module, 2) it does so for 1.5 room modules, or 3) two room devices provide the required supply air flow for 1.5 room modules (Fig. 2). For maintaining a constant supply air flow (2 l/s per m^2), an adjustment range of 15–30 l/s is required of the room device. This range increases to 15–40 l/s if it must be possible to convert a one-person office room into a meeting room.



Room / Beam	Space Type	Primary airflow rate			
		Nozzles	HAQ	Total	
1 /	Office	15 l/s	5 l/s	20 l/s	2 l/s,m2
2 /	Office	15 l/s	15 l/s	30 l/s	2 l/s,m2
3 / Unit A	Office	15 l/s	0 l/s	15 l/s	2 l/s,m2
3/ Unit B	Office	15 l/s	0 l/s	15 l/s	2 l/s,m2
3 / Units A &B	Meeting room	15 l/s	0...45 l/s	15...60 l/s	6 l/s,m2

Figure 2. Impact of room module division on the office and meeting room air flow.

Ready adaptability of air flow and space arrangements also increases the need to manage air distribution such that it reflects the various space solutions. It must be possible to reduce the total air flow from a beam in situations where, for example, the room device is close to a partition wall and the distance of the workstation from the wall is short (Figure 2: room module 1). It should also be noted that, owing to individual differences between people, some people perceive even low air velocities as a draught. This means an increased need to manage the individual room conditions.

It is important to remember that the majority (70–80%) of the air flow distributed to the room is recycled indoor air induced by the ventilation beam through the thermal exchanger to obtain the required cooling effect. The supply air flow provided by the fans is only 20–30% of the total air flow. This means that if the supply air flow is 2 l/s per m^2, the volume of air continuously recycled in the room is 8 l/s per m^2. Therefore, an efficient way of managing room space air velocities is to reduce the induction ratio of the room device to an appropriate level. Figure 3 presents the principle of operation for the induction ratio adjustment in a ventilation beam.

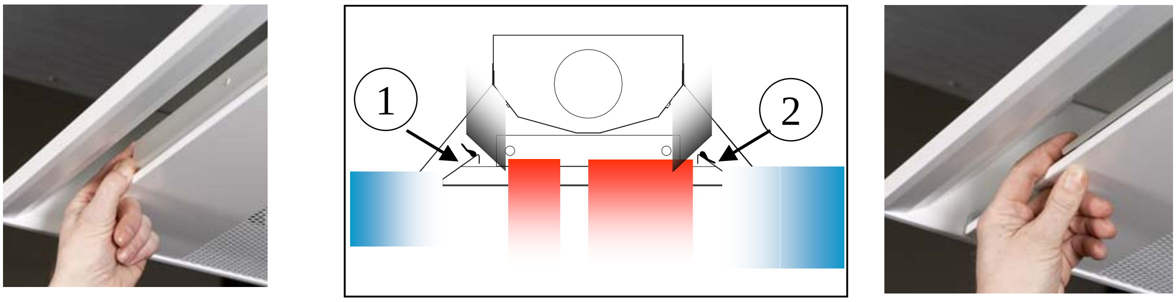


Figure 3. Principle behind induction ratio adjustment: 1) induction ratio adjustment on and 2) induction ratio adjustment off.

Figure 4 presents the impact of the induction ratio adjustment on the flow range of a CFD-simulated example room. The simulations indicate that induction ratio adjustment allows for reducing air velocities in the proximity of a window wall and at floor level. In Figure 4, induction adjustment is used on both sides of the beam. In practice, only one-sided induction ratio adjustment can be used, which lowers the effect of induction ratio adjustment on the room temperature.

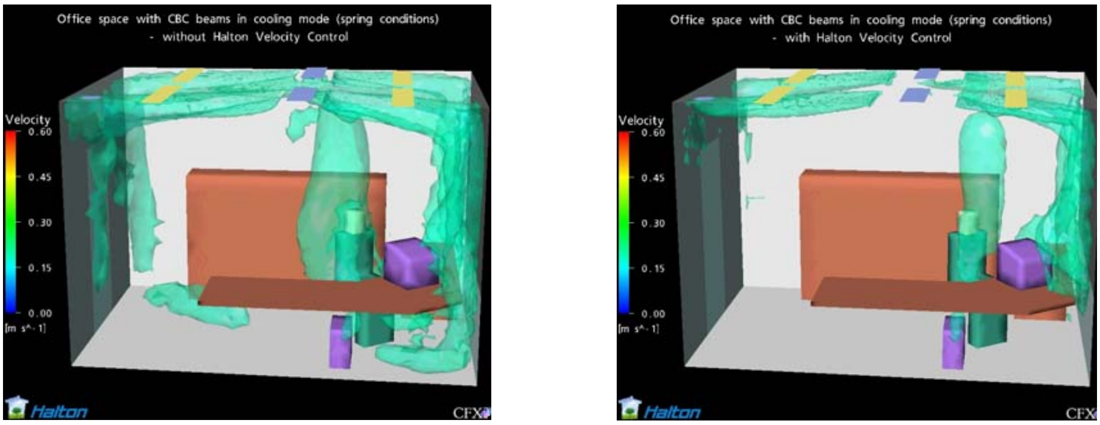


Figure 4. Impact of two-sided beam induction ratio adjustment on the air velocity for a sample room (threshold velocity: 0.25 m/s) and its temperature. Left: Induction ratio adjustment off. Right: Induction ratio adjustment on.

Case-studies carried out in laboratories examined the significance of induction adjustment in a situation where beams installed in a suspended ceiling were perpendicular to the window. According to the measurements made, induction adjustment could significantly reduce air velocities in the proximity of the employees (Fig. 5) [8].

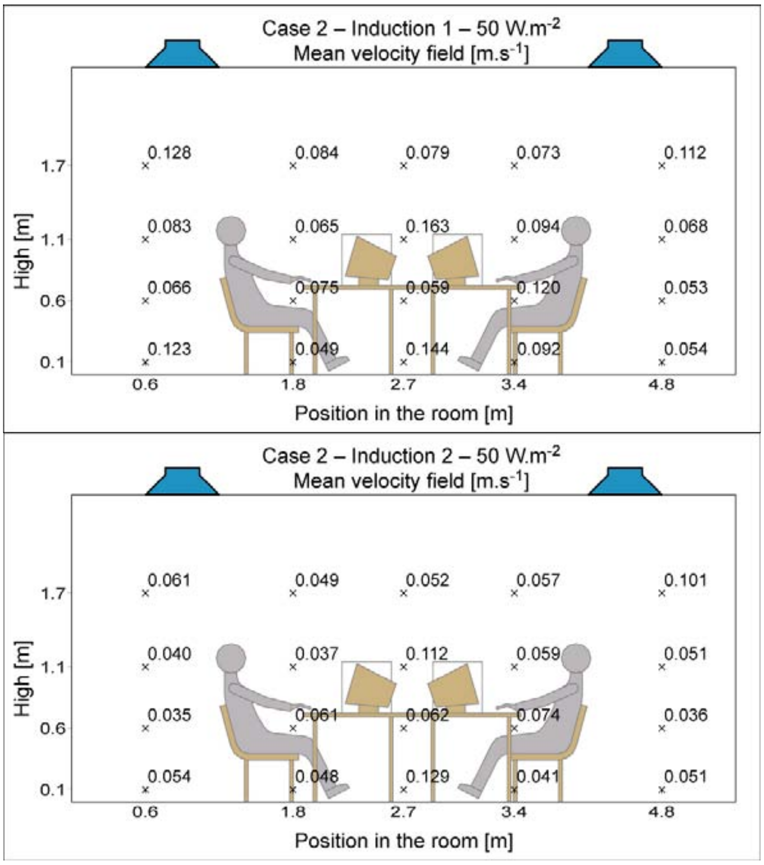


Figure 5. Impact of induction ratio adjustment on air velocity in laboratory tests.
Top image: Induction ratio adjustment off. Bottom image: Induction ratio adjustment on.

Induction ratio adjustment simultaneously affects air velocity and the cooling effect of the room device (by 5–15%). Depending on the design solution, the cooling effect of the room device can be decreased or increased on site depending on the induction adjustment position in the design situation. In comparing induction adjustment to adjusting the water flow in the beam, it can be determined that, when the goal is to reach the same air velocity, water flow adjustment results in a greater change in the cooling effect than induction adjustment does [9].

In addition to the maximum air velocity, induction ratio adjustment can reduce the average room air velocity. Figure 6 presents the impact of induction adjustment in a sample case on the average and maximum velocity [10].

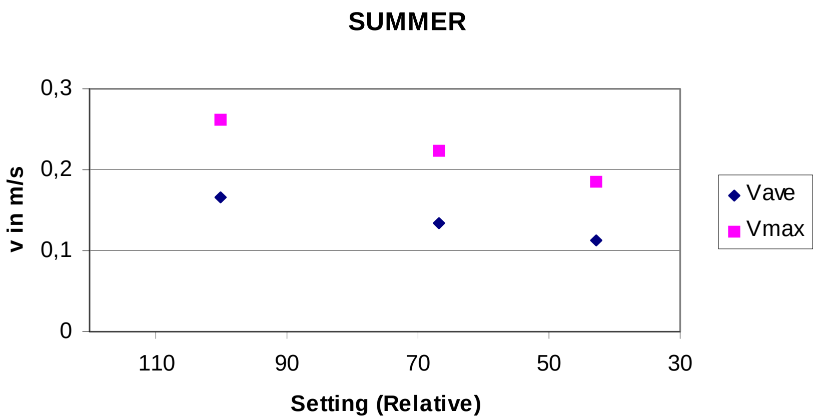


Figure 6. The impact of the total air flow supplied in the room on the maximum and average air velocity in the occupied zone as a function of the relative change of the induction ratio adjustment.

DISCUSSION

In a modern office environment, balance is sought between work performed by individuals and in interaction between employees. It must be possible to appropriately combine various work methods, so partition walls and workstations should be flexibly adaptable if they are to best meet the business needs of individual customers.

Adaptability of office space is one of the central requirements in the design of a beam system. The systems must be adjustable to address changed loads and partition wall positions. In design of a room system, this flexibility means taking account of supply air flow, cooling effect, and supply air device throw pattern changes. In addition, adjustability entails requirements concerning room automation, actuators and sensors. It must be possible to adjust the air flow supplied by the room device, which enables managing the maximum and average velocity in the occupied zone and thus reducing the draught risk.

ACKNOWLEDGEMENT

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IAQ and Energy Performance of the newly developed Single Coil Twin Fan air-conditioning and air distribution system – Results of a Field Trial

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SUMMARY

The Single Coil Twin Fan (SCTF) system is a newly developed air-conditioning and air distribution system that aims to deliver enhanced IAQ as well as save energy. The fundamental principle of this system is based on "demand ventilation" and "demand cooling", which is achieved in the individual occupied zones through localized temperature and carbon dioxide sensors positioned in the respective return grilles. The conditioning of the two air streams is achieved by a single compartmented cooling coil, which provides ease of control of the chilled water flow through the coil. Following a successful pilot study on a small-scale laboratory project in 2003 with an estimated energy saving potential of around 12%, a field trial of the SCTF system was embarked upon that involved an office floor area of about 2500 m^2 in a newly constructed office building within the campus of National University of Singapore. This paper presents the IAQ and energy data in this office area served by the SCTF system.

INTRODUCTION

The Single Coil Twin Fan (SCTF) concept involves two variable air volume (VAV) systems employing one compartmented cooling and dehumidifying coil [1, 2]. A schematic diagram of the SCTF air-conditioning and air distribution system is shown in Figure 1. The fresh air is conditioned in the "fresh air" compartment of the air handling unit (AHU) and distributed to the various VAV boxes that form part of the air distribution network. Each of these F/A VAV boxes is controlled by its own localized carbon dioxide (CO_2) sensors, which will ensure an adequate ventilation (F/A) provision at all times. As the main purpose of the F/A VAV box is to ensure adequate fresh air quantity based on occupant density, it helps in achieving energy conservation in the event of reduced occupant loads.

The return air from the various zones of the same distribution network is conditioned in the "recirculated air" compartment of the same AHU and distributed to a separate set of the various VAV boxes. Each of these R/A VAV boxes is controlled by its own localized zone thermostats, which addresses diversity in cooling loads, and consequently helps in achieving significant energy savings at part load operating conditions resulting from non-occupancy related factors. The conditioned Fresh Air and the conditioned Recirculated Air travel in parallel ducts and do not mix until just before the supply air diffusers in the mixing chamber of the modified VAV box. The compartmented coil achieves the required psychrometric performance of the two separate air streams throughout the operating range based on the concept of two VAV systems but one cooling coil.

DESCRIPTION OF BUILDING USED FOR FIELD-TRIAL

The building that is adopted for the field-trial of the SCTF system is a newly constructed low-rise office building located within the university campus. One half of the 3rd floor of this building, approximately 2,500 m^2, is used as the test-bed for the new air-conditioning and air distribution system. The SCTF system was installed in early 2005 and it has been in operation since June 2005. A schematic layout of the relevant portion of the 3rd floor of the building is presented in Figure 2. The office layout in Figure 2 comprises a combination of individual office rooms, low-level office cubicles, conference room, meeting rooms and open plan spaces. The return air is ducted from the various zones to the AHU room. The design cooling capacity of the compartmented coil was 86 kW for the F/A compartment and 233 kW for the R/A compartment, thus giving a total cooling capacity of 319 kW. The design F/A and R/A flow rates were 5,400 cmh and about 60,000 cmh respectively.

CONCEPTUAL DESIGN OF SCTF SYSTEM FOR THE BUILDING

The basis of design of the SCTF air distribution system is as follows:

- All the temperature thermostat and CO_2 sensors are duct mounted near the return air grille of the concerned occupied zones
- Every individual office room has its own temperature thermostat and CO_2 sensor
- All meeting rooms and conference rooms have their own temperature thermostat and CO_2 sensors
- The open plan areas are divided into clusters of imaginary thermal zones as shown in Figure 2. For each open-plan zone, a set of temperature thermostat and CO_2 sensor is to be provided that will control the requirements of R/A and F/A respectively

MEASUREMENT METHODS

As this is the first installation of the SCTF system on a large scale in a real building, it was important to measure the IAQ and energy performance of the system. Several experiments were conducted, during which the following measurements/data were obtained:

- Continuous measurements of CO_2, CO and TVOC at the sampling locations 1 through 5. Locations 1 through 4 are in the open plan office zones and location 5 is in an individual office room, as indicated in Figure 2. A multi-gas monitor, based on the photo-acoustic infra-red principle, is used for these measurements.
- Continuous monitoring of indoor air temperatures and relative humidity at the sampling locations 1 through 5. These are obtained using HOBO data loggers.
- Actual count of the occupancy rate in the open plan office areas at half hourly intervals during the experiment involving the dynamic response study
- Readings (frequency, hz) of the Variable Speed Drive (VSD) of both the “Fresh Air” and the “Recirculated Air” fans during the dynamic response study
- Compartmented cooling coil energy consumption obtained from Building Automation System (BAS) data, involving continuous measurements of chilled water supply and return temperatures to the coil and the chilled water flow rate.

RESULTS AND DISCUSSION

The IAQ and energy results are discussed and are based on the measurements obtained during the different interventions.

Continuous IAQ monitoring - Zone CO_2 set points at 700 ppm, (13th – 17th Nov, 2006)

The carbon dioxide, carbon monoxide and TVOC levels are plotted in Figures 3 through 5 respectively. The indoor temperature and RH levels are plotted in Figures 6 and 7 respectively. The concentration levels of indoor pollutants are generally within recommended Threshold Limit Values (TLV), which are also indicated in Figures 3-7 [3]. As the CO_2 set point in the zones is 700 ppm, the indoor CO_2 levels are of the order of 700-800 ppm during the normal operating hours of the day (Figure 3). The CO plot in Figure 4 reveals that the CO is well within the TLV of 9 ppm and that there are no sources of CO inside the building. It is observed from Figure 5 that the TVOC levels drop from an overnight build-up of 3.0-3.35 ppm to around 2.0-2.4 ppm when the AHU is started in the early morning. This clearly shows the increased dilution effect of a higher ventilation provision. During normal operating hours when the zones are occupied, the ventilation provided is able to maintain the indoor TVOC levels in the range of 1.8-2.4 ppm. The indoor temperature and humidity levels during normal operating hours are in the range of 20-23 °C (Figure 6) and about 50-58% RH respectively (Figure 7).

The total bacteria count and total yeast and mould count were obtained at the selected sampling locations and were generally within recommended Threshold Limit Values of 500 CFU/m^3. The Total Bacteria Count ranged between 70 and 370 CFU/m^3 and the Total Yeast and Mould count, which do not have any indoor source, were quite low in the range of 18-26 CFU/m^3.

The dynamic response of the SCTF system to varying set point requirements of CO_2 in the various occupied zones is seen from Figures 8. These experiments were conducted at two different set points of CO_2, - 500 ppm and 1000 ppm respectively. It is seen that the occupancy rate in the open plan office areas is about the same in both cases and that the indoor CO_2 level ranges between 500 and 750 ppm through the day when the CO_2 set point is 500 ppm. When the CO_2 set point is 1000 ppm, the indoor CO_2 level ranges between 520 and 800 ppm. Whilst the indoor CO_2 levels reached in both cases is quite similar and that the case involving CO_2 set point of 500 ppm is only marginally lower than that of 1000 ppm, it is seen that the fresh air fan operates at a higher frequency during 500 ppm set point of CO_2, implying the dynamic response capability of the system to provide more fresh air. It is to be noted that the CO_2 set point of 500 ppm is very close to ambient CO_2 levels, which typically ranges around the 400 ppm level in Singapore. A high recirculation rate of almost 90% further exacerbates the problem in the context that the design fresh air is only about 10% of total air flow.

Energy Measurements

The compartmented cooling coil energy consumption, expressed in terms of coil capacity (kW) at different time intervals, is presented in Figure 9. It is seen that the measured coil capacity is about 160 kW and this compares with the design capacity of 319 kW. It was also observed that the ΔT across the chilled water supply and return temperatures is rather high (about 9 °C) and the chilled water modulating valve was generally about 30% open. It is to be noted that fan energy consumption was not included in these measurements.

CONCLUSIONS

This paper presents the field data for IAQ and energy of a newly developed air-conditioning and air distribution system, called the Single Coil Twin Fan (SCTF) system. The results presented indicate that the SCTF system is able to provide adequate ventilation in a typical large office premises, based on "demand ventilation" and "demand cooling" in the individual occupied zones. The study has produced a clear demonstration of the dynamic response of the SCTF system. The thermal comfort parameters also indicate superior performance of the SCTF system, particularly in terms of the indoor RH levels, which is attributable to the enhanced dehumidifying performance of the compartmented cooling coil.

ACKNOWLEDGEMENTS

The financial support of the Prime Minister's Office, Singapore, under The Enterprise Challenge (TEC) scheme is gratefully acknowledged. The involvement of National University of Singapore as the Piloting Agency for the field trial is also acknowledged.

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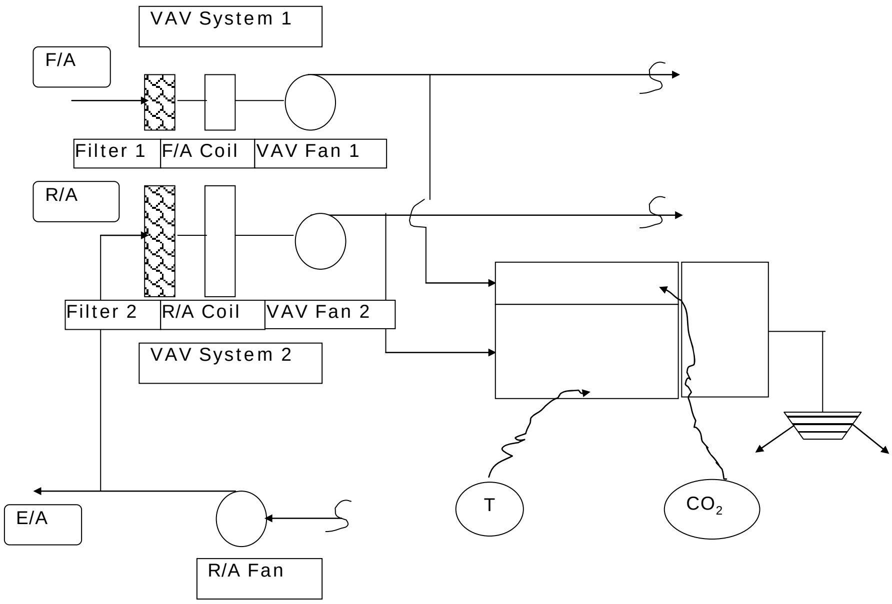


Figure 1 : SCTF air-conditioning and air distribution system

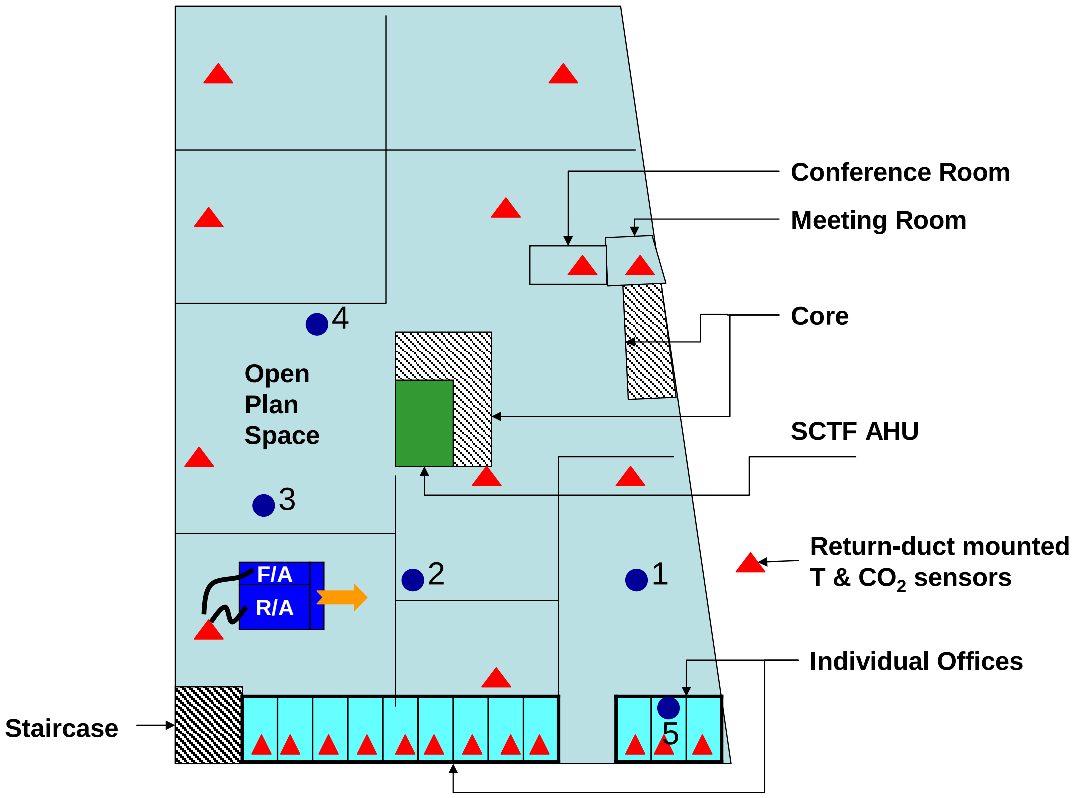


Figure 2 : Layout of the 3rd floor of the building employing the SCTF system

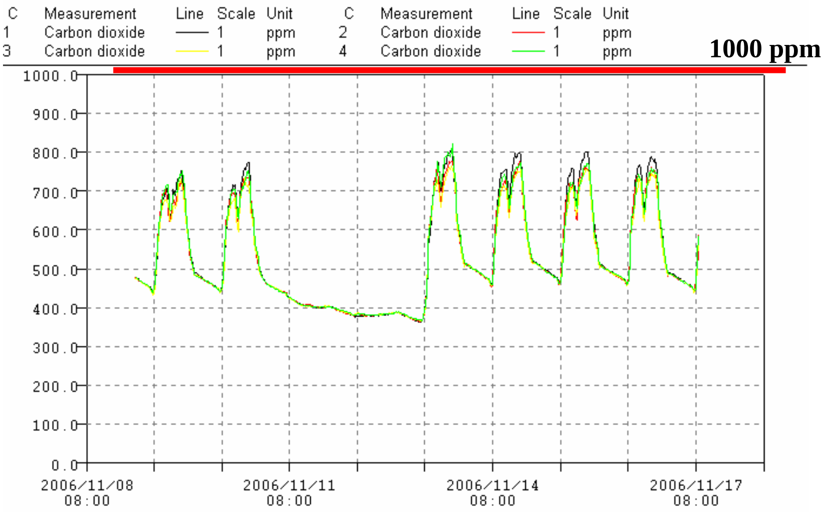


Figure 3 : Level 3 CO_2 levels

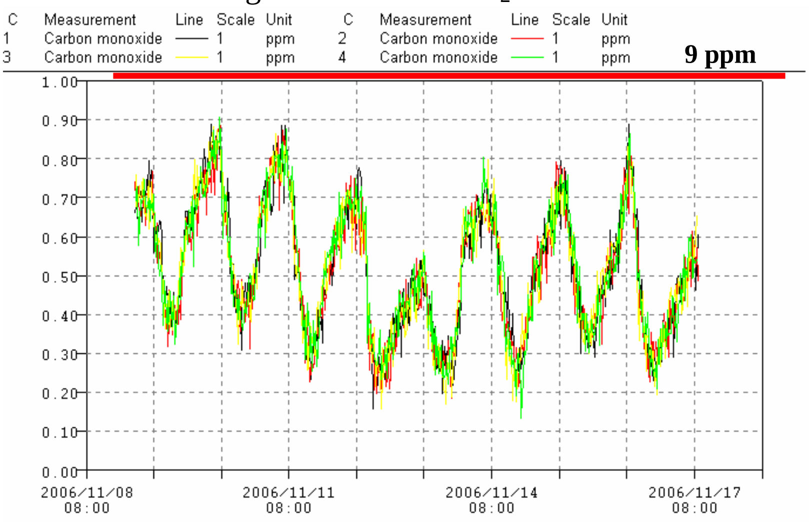


Figure 4 : Level 3 CO levels

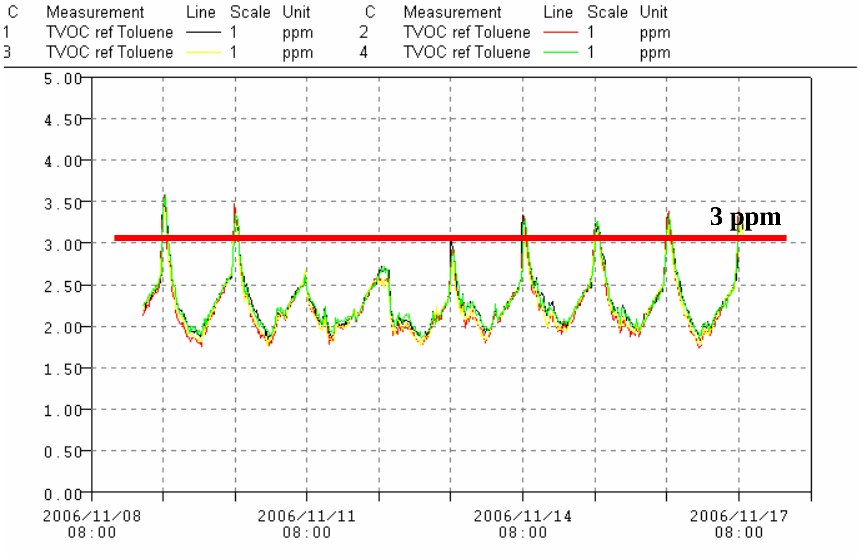


Figure 5 : Level 3 TVOC levels

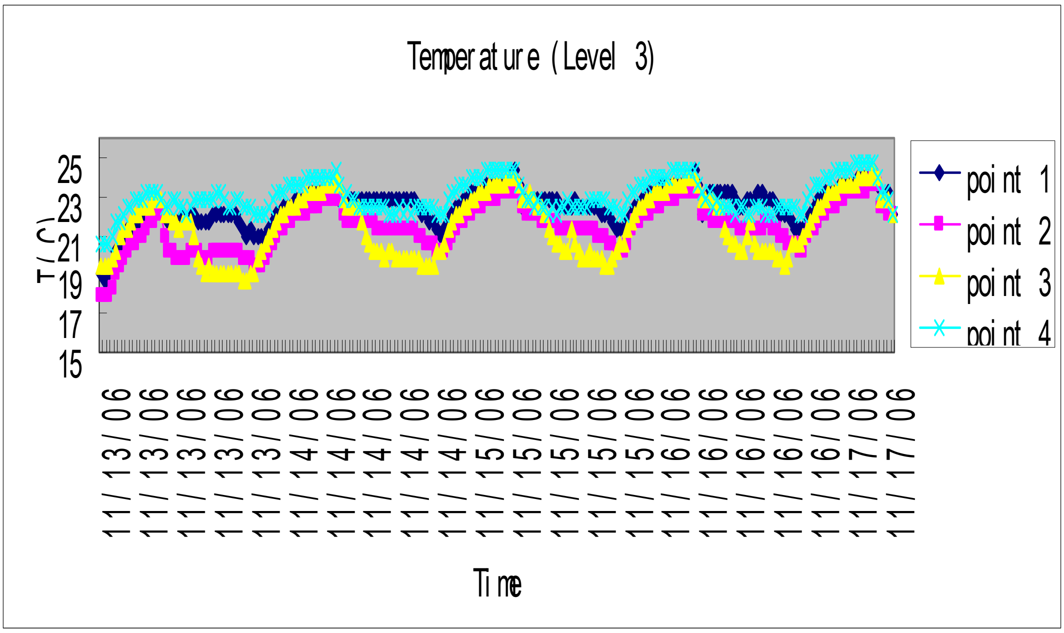


Figure 6 : Level 3 Temperature levels

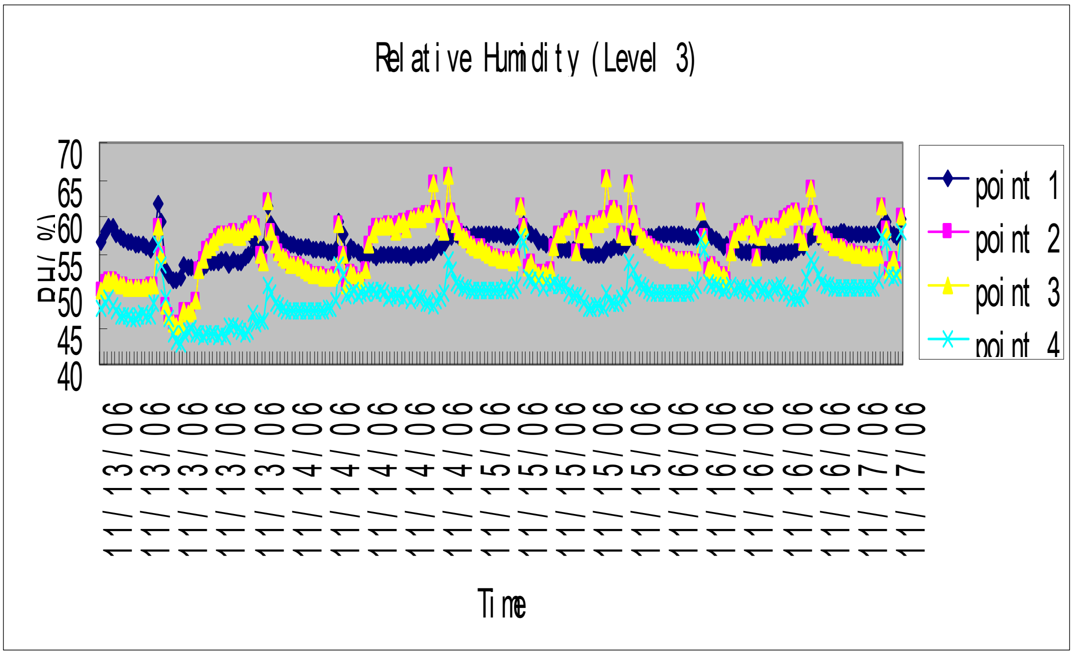
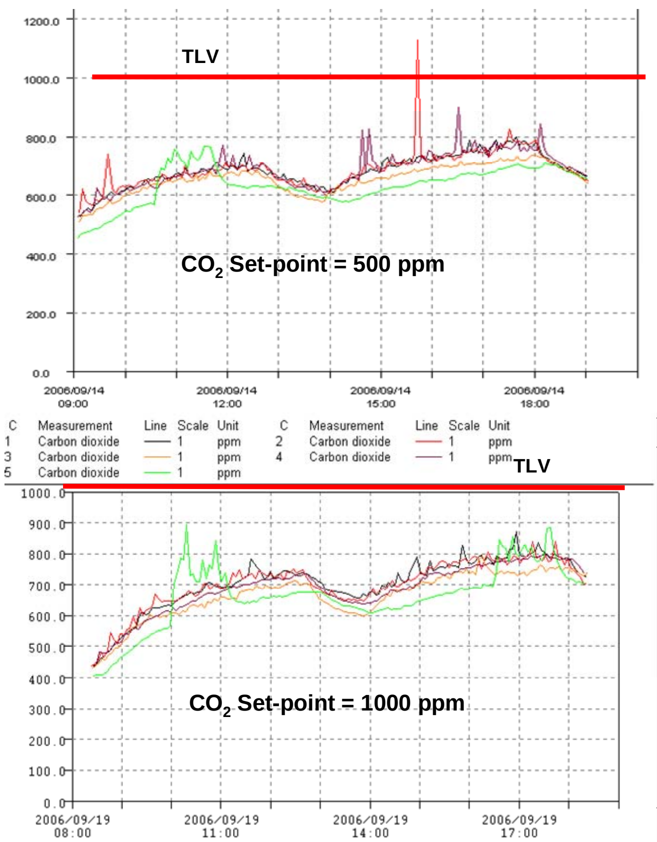


Figure 7 : Level 3 Relative humidity levels



Time	Occ Density	FA Fan(Hz)	RA Fan(Hz)
9:00	72	40.5	35.3
9:30	76	41	35.6
10:00	74	41.1	35.6
10:30	73	41.2	35.6
11:00	76	41.2	35.5
11:30	71	41.2	35.6
12:00	78	41.5	35.5
12:30	70	41.3	35.1
13:00	20	41.1	34.9
13:30	23	40.2	34.6
14:00	42	40.5	34.6
14:30	72	41.2	35.1
15:00	74	41.6	35.3
15:30	74	41.6	35.6
16:00	72	41.6	35.6
16:30	68	41.5	35.6
17:00	69	41.2	35.7
17:30	71	41.2	35.6

Time	Occ Density	FA Fan(Hz)	RA Fan(Hz)
9:00	75	27.3	32.1
9:30	75	27.4	31.9
10:00	82	27.3	31.9
10:30	75	27.3	31.9
11:00	77	27.4	31.9
11:30	78	27.3	31.9
12:00	70	27.1	31.9
12:30	79	27.1	31.7
13:00	15	27.1	31.5
13:30	30	27	31.2
14:00	48	27	31
14:30	74	27.4	31.9
15:00	74	27.3	31.9
15:30	74	27.4	32.1
16:00	74	27.3	31.9
16:30	78	27.3	32.2
17:00	75	27.2	31.9
17:30	76	27.2	31.5

Figure 8 : Dynamic response measurements

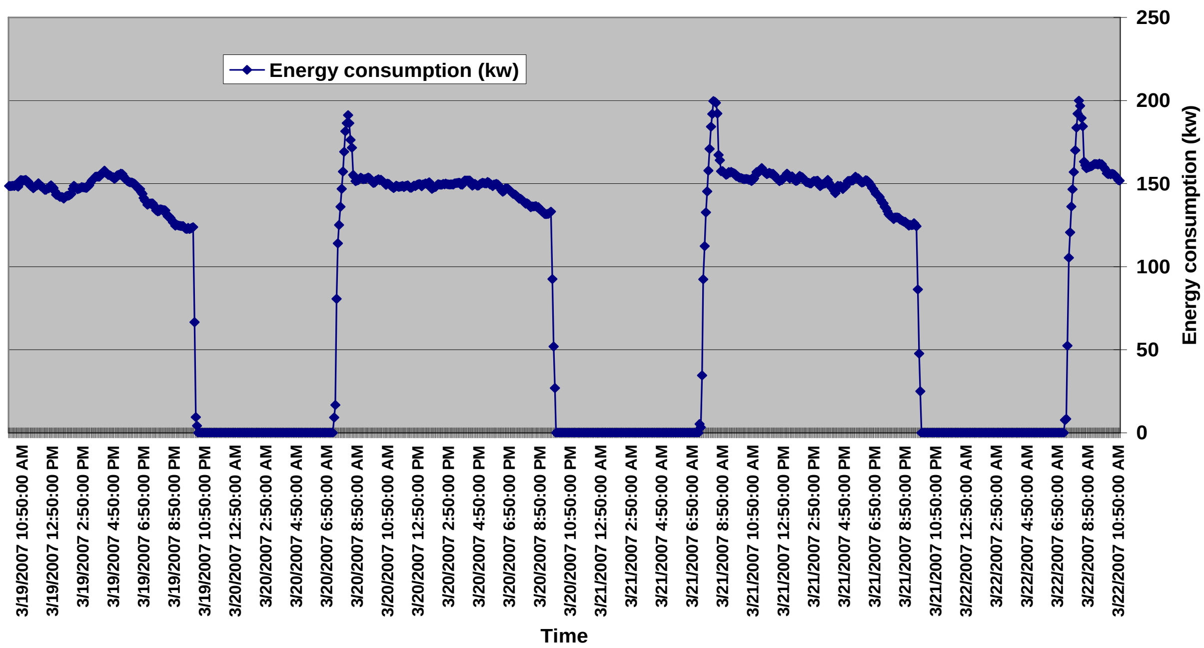


Figure 9 : Compartmented cooling coil energy consumption profile

Energy Evaluations between Air-Cooled Vs Water-Cooled Cooling Systems in Non-Residential Buildings

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SUMMARY

Present study is devoted to specifying the optimization of use of air-cooled vs. water-cooled cooling systems based on the climatic region classification. It proposes affecting design criteria's, which could be implemented in HVAC Uniform code for Middle East, and especially for the Egyptian HVAC and Energy codes. It yields selection criteria's for typical size ranges, where a water-cooled cooling systems are more economical than air-cooled systems and vise versa. This concept has impacts on energy efficiency and conservation. Study is concerned to the field survey, and energy analysis of non-residential buildings in Egypt, covering seven main climatic regions, on which, each region shall be defined for specific cooling system configurations. One of the conclusions is that for newly designed projects, located almost in all Egyptian climatic regions, with any cooling capacities ,could be designed based on water-cooled system with minimum initial and running cost taking into consideration the availability of different water sources in Egypt.

INTRODUCTION

Energy demands in buildings require clear identification of the energy measuring factors with potential verification of each factor in terms of energy saving and the economic revenue [1]. First, factors those required for comfort control and are easy to implement by codes. Second, factors that based on promoting changes in needs and habits of occupants, equipment, operating schedules and control of temperature settings. Third, those factors that are related to the Electro-Mechanical systems, which are the most effective terms on total energy, use and demands [2]. Field surveys, theoretical analyses and parametric runs were performed on different building types in Egypt [3]. Most of these analyses were related to the first group of factors. While, economic evaluations of cost impacts versus anticipated benefits were carried out by the local authorities [4]. The second group of factors has been preliminary treated, since these factors require adaptations in the habits of occupants. Finally, third group of factors was considered since these factors have strong potential on global energy efficiency without affecting citizen's habits. Egypt country was elected as a base case in present study.

Climatic Classifications

Egyptian climate as a sample case considered as Hot and Dry climate while the north region adjacent to Mediterranean Sea is hot and humid climate due the effects of sea [5]. Bioclimatic classifications based on temperatures, humidity and solar heat gains, for Egypt show main seven regional climates [6] on which human biological and physiological impacts were studied by many investigators to evaluate, and justify the acceptable human comfort limits [7]. Climatic Regions are indicated in Figure 1 .

Region [1],
Mediterranean Sea climates.
22-to-28^{O}C dbt & 50-to-80%RH
Region [2],
Upper and Lower west desert.
30-to-38^{O}C dbt & 40-to-60%RH
Region [3],
Upper Egypt valley at Sudan borders.
30-to-45^{O}C dbt & 15-to-40%RH
Region [4],
Southern-Upper Egypt valley.
31-to-42^{O}C dbt & 20-to-55%RH
Region [5],
Northern-Upper Egypt valley.
30-to-40^{O}C dbt & 30-to-55%RH
Region [6],
Delta Region.
22-to-37^{O}C dbt & 45-to-65%RH
Region [7],
Sinai, Red Sea Zone.
23-to-41^{O}C dbt & 17-to-50%RH

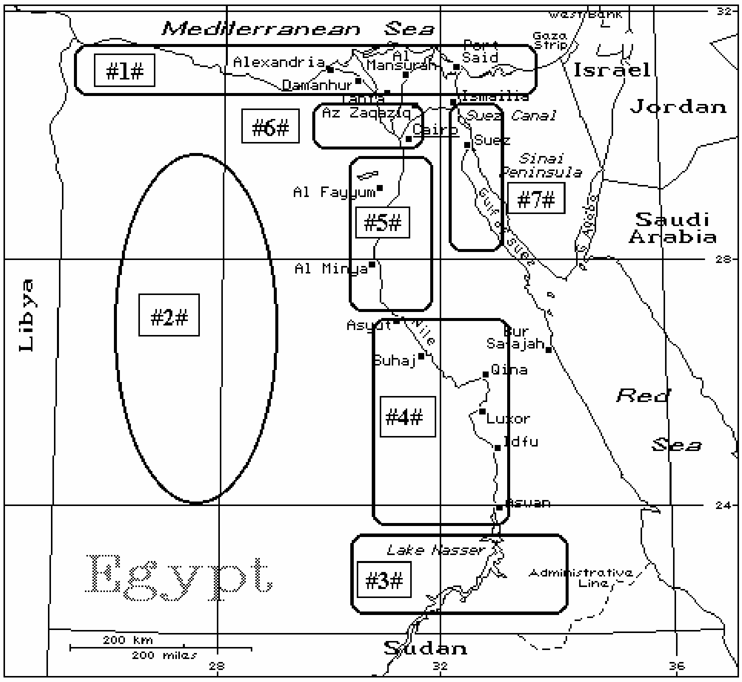

Figure 1. Climatic Regions Classifications in Egypt at Average Summer Conditions

Heat Release Concept

In dry coolers ambient air is used to remove heat by convection process, while in wet coolers heat removal achieved by ambient air in combination with water in evaporation processes. Where, heat release is a function of the wet bulb temperature, WBT, of ambient air and of air capacity for vapor transport. In a Hot & Dry climate, WBT is usually much lower than dry bulb temperature, DBT. Accordingly, lower effective heat sink temperature can be achieved with wet coolers than with dry coolers [8]. On the other hand, wet coolers require water feed & drainage systems, water treatment, filtration units, piping between cooling units and towers, and extra controls leading to additional complexity, together with installation and maintenance costs, that make it less acceptable and less economical than dry coolers. In Egypt cooling towers are not considered for installations smaller than 1500 kW [2].

Thermal load characteristics survey and common practices in Egypt [3] demonstrated ranges of required air-cooled cooling equipment capacities, especially when these equipment are located at ambient DBT over 45^{O}C with the presence of sandy storms. Figure 1 shows real energy efficiency ratios, EER's of in-service air cooled cooling units starting from ARI-Conditions and up to 48^{O}C outdoor operating conditions representing different locations in Egypt. Fitted curves listed in Figures 1 & 2 represent the following different cooling systems that have same cooling capacities, operating schedules and indoor conditions:

[A], Water cooled reciprocating or screw chillers ranged from 70kW-to-525kW nominal cooling capacities, new installations, sand filters and chemical treatment were utilized in condenser side, and operating at 80%-to-100% full loads.
[B], Water cooled reciprocating direct expansion units ranged from 17kW-to-100kW nominal cooling capacities, new installations, sand filters and chemical treatment were utilized in condenser side, and operating at full loads.

[C], Water cooled reciprocating or screw chillers ranged from 70kW-to-525kW nominal cooling capacities, old installations, sand filters and chemical treatment were not utilized in condenser side, and operating at 80%-to-90% full loads.
[D], Water cooled reciprocating direct expansion units ranged from 17kW-to-100kW nominal cooling capacities, old installations, sand filters and chemical treatment were not utilized in condenser side, and operating at 90% full loads.
[E], Air cooled reciprocating direct expansion units that are new &/or have five years old installations, operating at 80%-to-100% full load covering nominal cooling capacities from 17 kW-to-67 kW..
[F], Air cooled reciprocating water chillers that have five years old installation, operating at full load covering nominal cooling capacities from 17 kW-to-525 kW.

Figure 2 show that air-cooled cooling systems that ranged from 17 kW to 525 kW, When operated at tropical outdoor temperatures and with presence of sandy storms will have a new cooling ratings and performances lower than that listed in the original catalogues [2,8]. This means losing sensible cooling capacities and increase power consumptions significantly, On the other hand, the figures show that water cooled units have a stable cooling capacities as its operation principles are based on evaporative cooling that is affected only by wet bulb temperature. Wet bulb temperatures are almost stable and ranged from (22 ^{O}C-to-26^{O}C). Figure 3 shows actual behavior in term of the coefficient of performances, COP's of the same cooling systems compared with ARI-Conditions and up to 48 ^{O}C. Both Figure 2 and 3 show that water-cooled systems have good sustainability in hot and dry climate as they consume low energy up to 0.88 kW/T.R (1.0 T.R=3.5 kW cooling) ,when utilizing evaporative systems rated at 90% or more saturation efficiency [1].

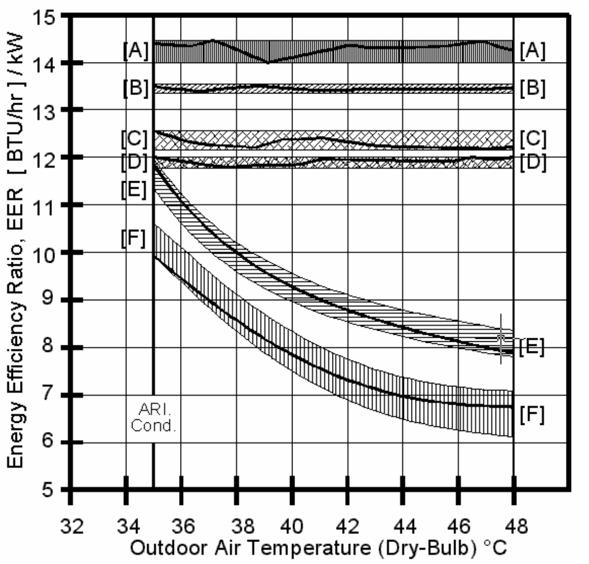

Figure 2, Changes in EER with increase of outdoor DBT compared with ARI-Conditions

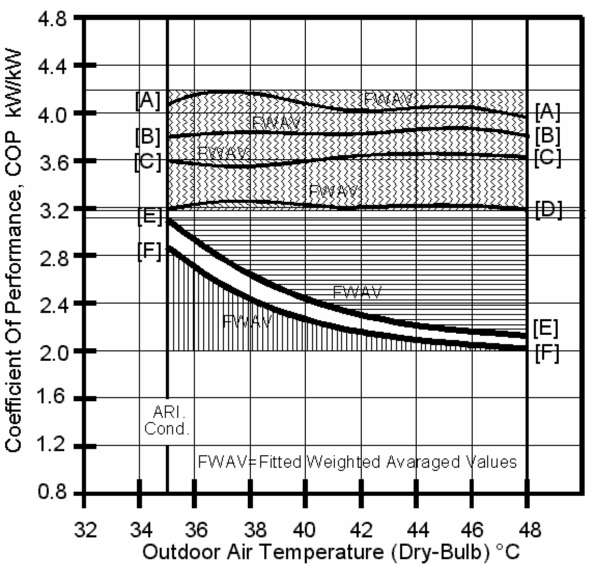

Figure 3, Changes in COP with increase of outdoor DBT compared with ARI-Conditions

Comparative Study Standardization

As this study primarily presents an economic evaluation of air-cooled and water-cooled systems, a number of assumptions are used to make the investigation more manageable and appropriate; the more important assumptions can be summarized as:

- In air-cooled units; the study covers direct expansion and chilled water systems for 100% loading units in operation up to 900 kW, 255 TR cooling capacity.

- In water-cooled units; the study covers direct expansion and chilled water systems for 100% loading units in operation range 500-to-3500 kW, 140-to-1000 TR cooling capacity.
- Ambient Conditions, dry bulb temperatures ranged from $39^{O}C$ to $42^{O}C$, & Wet Bulb temperatures ranged from $23^{O}C$ to $26^{O}C$.
- Indoor Conditions, dry bulb temperatures ranged from $24^{O}C$ to $26^{O}C$, & Relative humidity ranged from 45% to 55%.
- Systems Conditions, All units utilize refrigerant R-22 using Reciprocating or Screw compressors types while, water cooled types over 2000 kW, 570 TR cooling Capacity utilize centrifugal compressors.
- Supply air temperatures from direct expansion units are not more than $10^{O}C$.
- Supply/Return water temperatures in hilled water systems $6\ ^{O}C$ /$12\ ^{O}C$.
- Analyses for the life cycle cost, and maintenance do not consider any interest or inflation rates for simplification purposes and render the study consistent with other approaches and also due to the lack consistent projections for interest and national inflation rates.
- Annual operating costs consist of the cost of the electricity, water use and operation, maintenance charge per year of operation.
- Annual consumption of electricity and water are estimated on the basis of an equivalent full load operation of plant over a specified number of hours of operation per year.

Comparative Study Methodology

This study involves the evaluation of life cycle cost of air-cooled and water-cooled direct expansion and chiller plants of various sizes up to 3500 kW, 1000 TR. The cost analysis is performed in terms of customer [CI] and national [NI] cost indexes. Customer cost index reflects all expenses born by individual property owner the capital equipment of the chiller plant, installation, operation and maintenance over specific life time, and includes the new charges for installation and connecting the electrical capacity required by the plant. While national cost index reflects all expenses for equipment and labor at national level to install, operate and maintain cooling plant over same life time. It is rather similar to customer cost index except that the annual operating cost is estimated on the basis of the national cost for producing electricity and water as per the following governing equations:

$$CI = \{[CC+I]/TR\}+\{([LT]*[AC]+[NC]*[ER])/TR\} \quad \textbf{Eq.1}$$

$$NI = \{[CC+I]/TR\}+\{([LT]*[AC])/TR\}+\{[ER]*[CAE]\} \quad \textbf{Eq.2}$$

$$[AC]_{\text{air cooled}} = [ER*TR*HRE*EC]+OP+MC \quad \textbf{Eq.3,}\ \text{Customer}$$

$$[AC]_{\text{air cooled}} = [ER*TR*HRE*EN]+OP+MC \quad \textbf{Eq.4,}\ \text{National}$$

$$[AC]_{\text{water cooled}} = [ER*TR*HRE*EC]+[W*TR*HRE*WC]+OP+MC \quad \textbf{Eq.5,}\ \text{Customer}$$

$$[AC]_{\text{water cooled}} = [ER*TR*HRE*EN]+[W*TR*HRE*WN]+OP+MC \quad \textbf{Eq.6,}\ \text{National}$$

Where:

I	Installation cost **(L.E)**
CI	Customer cost Index **(L.E)**
CC	Capital equipment Cost **(L.E)**
TR	installed cooling capacity **(TR)**
LT	Life Time of equipment **(Years)**
AC	Annual operating Cost **(L.E)**
NC	New connection fee per kW Capacity **(L.E/kW)**
ER	Electrical Requirement per unit (T.R) capacity **(kW/TR)**
HRE	Equivalent hours of operation per year at full load **(h/yr)**
EC	Electricity Cost to customer **(L.E/kWh)**
W	Water consumption rate per unit (T.R) capacity **(l/h/TR)**
WC	Water Cost to customer **(L.E/m^3)**
OP	Cost of operator per year **(L.E)**

MC Maintenance Cost per year (labor + spare parts), **(L.E)**
CAE Cost of installing additional electrical capacity **(L.E/kW)**
EN National cost for producing electrical energy **(L.E/kWh)**
WN National cost for producing domestic water (**L.E/M^3)**

It is evident that accurate evaluations of customer and national cost indices requires detailed and accurate information of all the component costs that contribute on life cycle cost. This information was provided by HVAC contractors, equipment agents and consultants.

ENERGY ANALYSIS

a. Air-Cooled Units are available in one size of up to 800 kW, 255 TR in single packaged units. The majority of the units on the market are driven by reciprocating or screw compressors, one or more compressors per unit, for most installations in the range of 350-to-1750 kW, 100-to-500 TR . An analysis of the packaged units available on the market reveals a kW/TR requirement, excluding the chilled water pump and the air side, as shown in Figure 4. The figure indicates that the electrical requirements fall between 1.40-to-1.70 kW/TR.

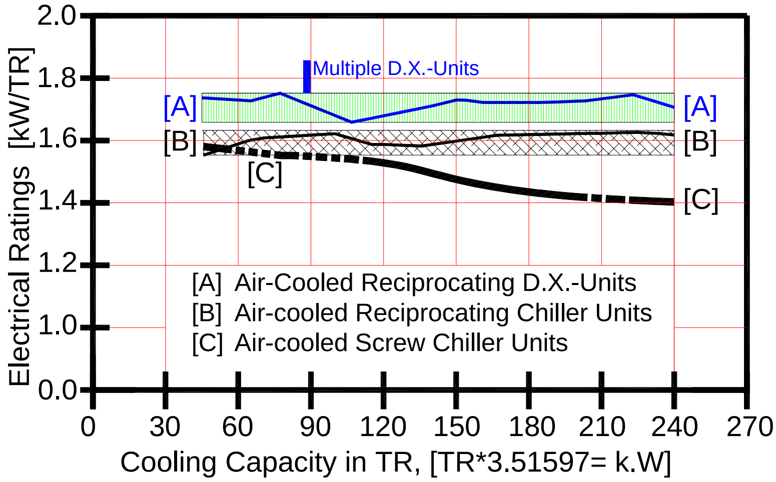

Figure 4, Electrical Average Ratings of Air-Cooled Units

b. Water-Cooled Units are available in sizes of up to 2000 kW, 570 TR in packaged units.

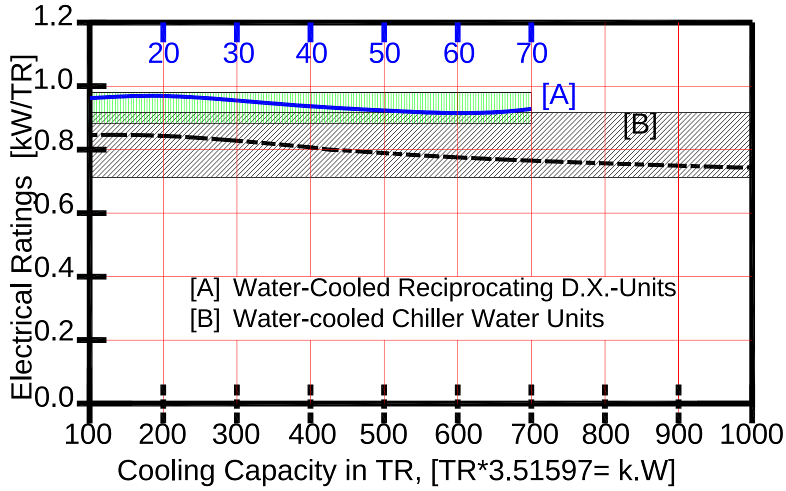

Figure 5, Electrical Average Ratings of Water Cooled Units

Reciprocating or screw compressors are used in the small size units while centrifugal compressors are common in the large size units. Due to the diversity in the type and size of compressors and other components, the electrical requirement of the cooling plant is expected to vary accordingly.

Cooling plant components that require electricity are primary the compressor, the condenser water pump and cooling tower fan. These components in water-cooled units are rather detached physically and operationally and their sizing is somewhat site dependent. Collected data reveal a spread in the energy requirements of the D.X or chiller only, as shown in Figure 5. The kW/TR requirement falls generally between 0.80-to-0.90, kW/TR. There is a tendency for the kW/TR requirement to drop with the increase in the size of the chiller though this drop is rather insignificant.

c. Cooling Towers have similar analysis of electrical requirements of the cooling tower fans for cooling towers of 350-to-7000 kW, 100-to-2000 TR in capacity gives the spread in results as shown in Figure 6. It is observed that the kW/TR requirement is around 0.1 kW/TR for 750 kW, 215 TR capacities, and falling off to 0.094 kW/TR for a 3000 kW, 850 TR capacity unit. These values are common for the usually utilized induced draft cooling tower design.

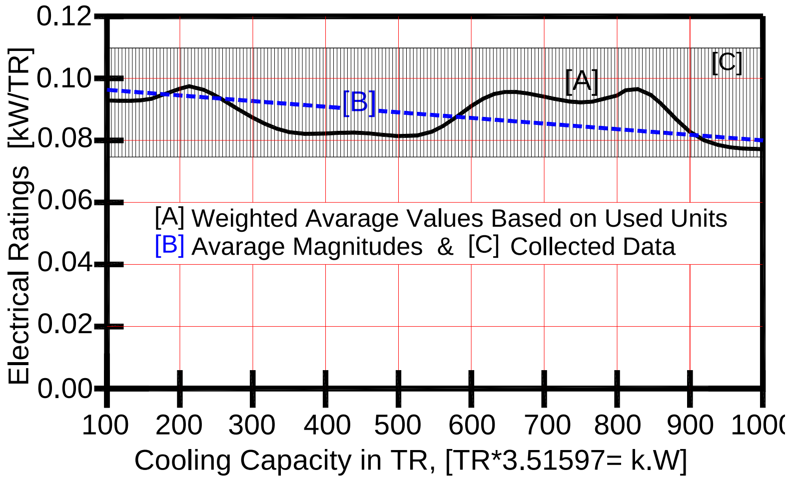

Figure 6, Electrical Approximated Ratings of Wet Cooling Towers

d. Water Pumps are strongly site dependent, the location of the cooling tower relative to cooling unit and the extent of piping, its size and number of bends, valves, etc. between the two units dictates the size of pump required. It is expected that there would be a very wide variation in the sizes and the electrical requirements of pumps for different installations. A review for the past ten years installations revealed a spread in the kW/TR requirement between 0.05 and 0.17 a generalized value for condenser water pump requirement could be within 0.10 kW/TR.

COST ANALYSIS

Capital **C**ost of **A**ir-cooled **U**nits in sizes of 150 KW, 40 TR and higher is shown in Figure 7. The values quoted in the figure are contractor price figures for packaged units delivered to site. The prices are for the packaged units with its air-cooled condenser, electrical panel and cabling. Figure 6 shows that, for chillers with reciprocating compressors in the capacity range of 150-to-530 kW, 40-to-150 TR, the capital cost ranged from 500-to-700 L.E/kW, 1750-to-2500 L.E/TR.

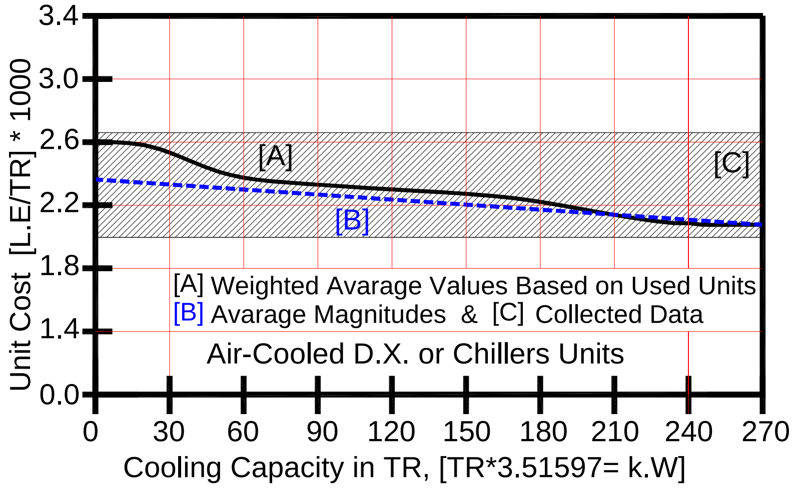

Figure 7, Predicted Cost per Cooling Capacity for Air-Cooled Packaged Unit

Capital **C**ost of **W**ater-Cooled **U**nits figures delivered to site are in Figure 8 The cost figures are for individual packaged units in the capacity range of 150-to-3000 kW, 40-to-850 TR. Installations of large capacities (3000 kW, 850 TR & up) and even those in the mentioned capacity range usually consist of a number of packaged units appropriately chosen to have a competitive L.E/TR cost. It can be observed from Figure 6 that there is a wide scatter in the cost figures, with the average values showing a distinct drop in the L.E/kW cost with the increase in capacity of the packaged unit. This cost starts at around 500 L.E/kW, 1750 L.E/TR for a 350 kW, 80 TR unit and drop off to around 350 L.E/kW, 1225 L.E/TR for units in the 1700-to-3500 kW, 485-to-1000 TR range.

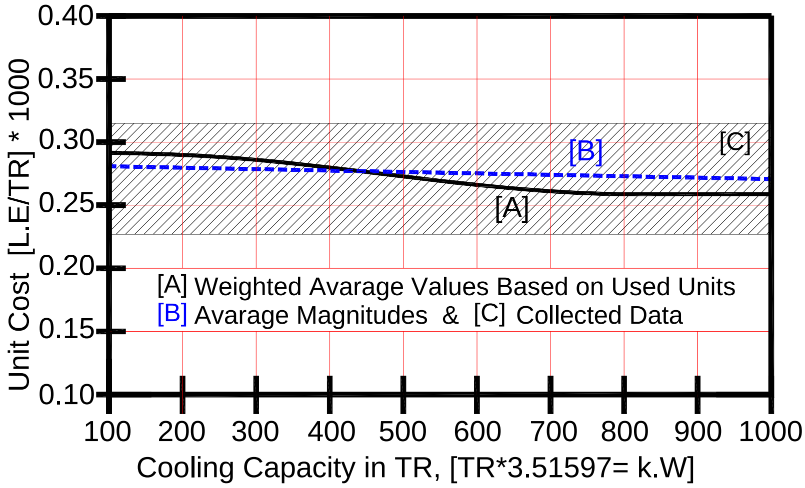

Figure 8, Predicted Cost per Cooling Capacity for Water-Cooled Packaged Unit

Capital **C**ost of **C**ooling **T**ower depends on the type of cooling tower specified for the installation. Capital cost for cooling tower is shown in Figure 9. The figure shows a variation in capital cost from around 125 L.E/kW, 440 L.E/TR for a 350 kW, 100 TR installations to around 100 L.E/kW, 350 L.E/TR for sizes 2800 kW, 800 TR and over. Pump cost varied between 10-to-25 L.E/kW, 35-to-88 L.E/TR, Piping cost has greater variations, from 20-to-65 L.E/kW, 70-to-225 L.E/TR. Other capital equipment costs are water treatment and filtration units that varied around 25 L.E/kW, 88 L.E/TR.

Water-cooled Units require a built-in space as opposed to air-cooled units, which are mounted directly on the roof. This space is normally provided or a separate plant room built, then the expense could add some 30-to-50 L.E/kW, 100-to-175 L.E/TR to the total installation cost of water-cooled units. This cost is not considered in the analysis.

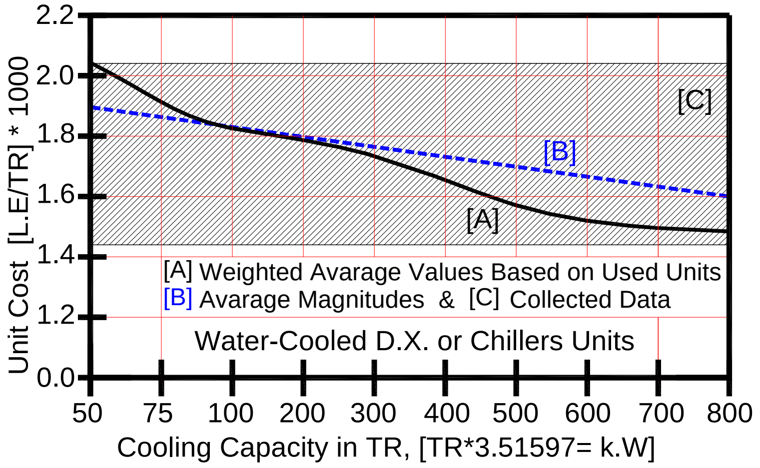

Figure 9, Predicted Cost per Cooling Capacity for Wet Cooling Towers

Installation & commissioning cost is difficult to determine with any certainty and vary among contractors and different installations. This is because some contractors include it within the capital cost or consider it as a certain percentage of the equipment cost. It is deduced that the cost of the installation and commissioning of air-cooled chillers is in the range of 30-to-50 L.E/kW, 100-to-175 L.E/TR. Installation size should have little effects on cost since large capacity installations would be made up a number of packaged units with each unit requiring the same amount of effort in installation and commissioning. A similar analysis for water-cooled units reveals that the cost for installation of all the components of the plant and its commissioning is in the range of 60-to-70 L.E/kW, 210-to-245 L.E/TR for large capacity installations.

POWER AND WATER COST

In Egypt, the energy consumption for residential and commercial private buildings contributed with 35% of the total energy generated [3] the larger portion of this energy was consumed in the cooling processes. Therefore, the residential and tertiary sector is one of the most energy intensive sectors. Nevertheless; the new buildings will be constructed in accordance with new energy efficiency codes. Energy consumed by the older buildings themselves is very high. Heat losses in buildings are caused, not only by low standards building materials and construction, but also by poor insulation of heat and lack of energy efficiency awareness that presents the main reasons for high energy consumptions in residential buildings and describes the traditional methods that have been employed to achieve energy conservation in residential sector. The running cost relating to the electricity and water used by cooling units over a one year period is estimated according national cost. The water consumption in water-cooled units is estimated to be 0.013 m^3/hr/TR for make-up water, while cost of air-cooled units could be similarly obtained, taking into considerations that water cost is subsidized by government.

CONCLUSIONS OF THE RESULTS

Simple analysis of air-cooled units and the effect of various parameters on their respective costs were carried out using simple parametric analysis with the ratio $(CI)_{\text{air cooled}} / (CI)_{\text{water cooled}}$ & $(NI)_{\text{air cooled}} / (NI)_{\text{water cooled}}$. It is observed from analyses that the capital equipment and installation costs represent a small fraction of the CI and NI. Thus, the variation of cost with cooling capacity should have insignificant effect on the analysis as indicated in Figure 10.

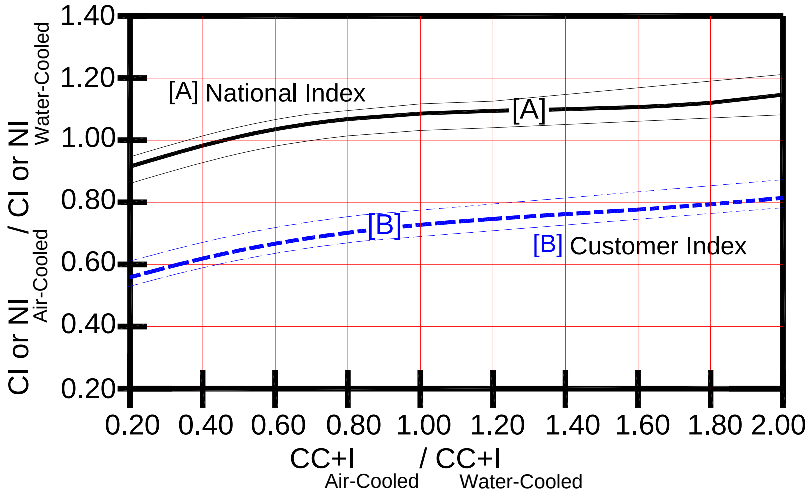

Figure 10, Effects of capital and installation costs on the customer and national index

There is a general consensus among contractors that air-cooled units are cheaper to operate and maintain, there is no clear idea of the actual cost. The figures obtained showed large variations. This is because operation and maintenance contracts are not based on the equipment only but also on site conditions. Also, contracts are usually awarded for the operation and maintenance of the total HVAC system and often these contractors for labor only with the spare parts to be paid by the customer according to use.

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Study on Running Performance of a Split-type Air conditioning System Installed on a University Campus in Suburban Tokyo

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SUMMARY

Split-type air conditioning systems, or heat pump systems with multiple indoor and outdoor units, are becoming very popular for room cooling and heating of small or middle-sized non-residential buildings in Japan. However, their running performance is yet to become known due to the difficulty of measuring actual amount of heat transferred by the system. Mixed irregular flow of vapor and liquid refrigerant has prevented building engineers from obtaining accurate amount heat flow between indoor and outdoor units. The study introduces an alternative method to calculate heat transferred by the system from air volume and enthalpy measured with simple sensors attached to the indoor units. Experimental results on a national university campus in suburban Tokyo showed unexpectedly low COP values both in summer and winter mainly due to the prevailing low partial load factors under 20-30% of the system capacity.

INTRODUCTION

The recent rise in environmental problems and deteriorating state of the earth combined with approaching deadline for public commitments to reduce CO_2 emissions and increasing energy consumption at private sector has intensified the importance of developing and promoting methods for conservation and efficient use of energy.

In this situation, recently, split-type package air conditioners (PACs for short) have been rapidly spreading from the aspect of cost saving and convenience and installed even in the buildings with several tens of thousand square meters of floor area in major Japanese cities. From the aspect of energy efficiency, however, it is suspected that their high rating COP based on the Japanese Industrial standard does not represent actual values in use. It is also suggested that PACs may constitute a primary source of heat island, because unlike centralized air-conditioning systems equipped with cooling towers which primary discharge latent heat, PACs exclusively discharge sensible heat into the urban canyon directly rising its air temperature. PACs efficiency therefore is also a primary interest from the aspect of urban environment. In order to obtain some knowledge to these pending questions, this research has tried to measure energy efficiency of a typical PAC under actual operating environment. The results will help us in designing and installing PACs in a better way.

METHODS

A multiple type package air conditioning unit installed on a national university campus in west suburb of Tokyo has been chosen for measurement. Multi point measurement method is applied to examine its efficiency in daily operation. Summer measurement was conducted with the indoor setting temperature of 27℃, and winter measurement with the temperature of 24℃ (February 14 through 16), 22℃ (17 through 19) and 20℃ (February 20 through 22). Throughout measurement, airflow was set at “strong” mode while wind direction was set “no swing with middle low angle” blowout position. Weather information of measurement periods are provided in table1 and 2.

Table 1. Weather information of summer measurement period at Yokohama weather station (Friday, August 26 through Sunday, September 5, 2005)

	26. Agu.	27. Agu.	28. Agu.	29. Agu.	30. Agu.	31. Agu.	1. Sept.	2. Sept.	3. Sept.	4. Sept.	5. Sept.
Wether											
Maximum temperature(℃)	33.1	31.5	28.2	30.2	30.1	27.4	30.9	31.6	31.2	31.7	25.6
Minimum temperature(℃)	23.8	25.2	22.9	22.5	22.6	22.9	22.5	23.4	23.8	23.8	21.9
Precipitation(mm)	38.5	--	0	0	0.5	0	--	--	--	5	81.5

Table 2. Weather information of winter measurement period at Yokohama weather station (Tuesday, February 14 through Wednesday, February 22, 2006)

	14. Feb.	15. Feb.	16. Feb.	17. Feb.	18. Feb.	19. Feb.	20. Feb.	21. Feb.	22. Feb.
Wether									
Maximum temperature(℃)	18.2	18.5	11.8	9	6.3	9.1	7.4	10.5	15.6
Minimum temperature(℃)	2.2	9.6	6	2.8	0.6	3.8	3.5	3.9	6.7
Precipitation(mm)	--	0	10	0.5	0	--	22	--	--

MEASUREMENT CONTENTS

(1) Indoor thermal environment
In order to measure indoor temperature distribution, 15 representing points were designated each with three vertical measurements making up 45 altogether as shown in Figure 1.

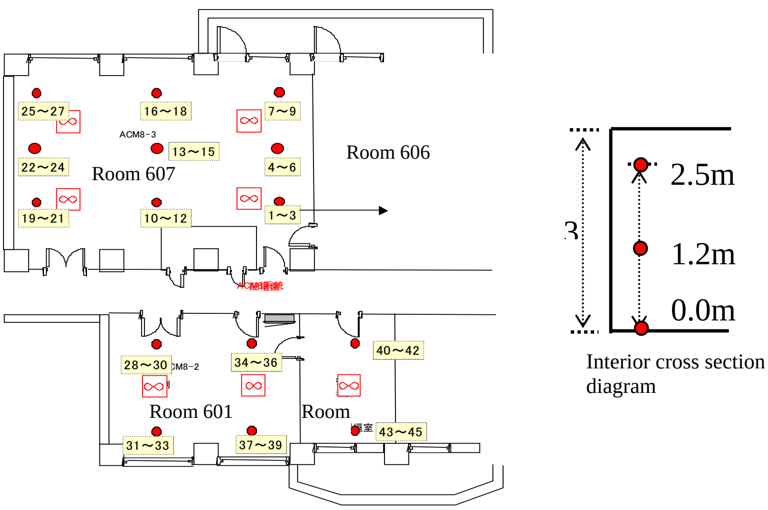
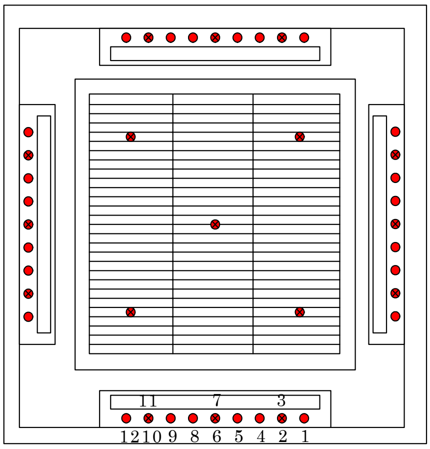


Figure 1. Horizontal and vertical location of thermo-hygrometers installed around indoor units

(2) Thermal flow at Indoor units
Dry bulb and wet-bulb temperature of the inlet and outlet airflow were measured at all of 7 indoor units with thermocouple and data logger (Figure2) while air velocity was measured

with the traverse developed for this study. Heat exchange quantity was then calculated by multiplying inlet and outlet temperature difference and air volume calculated by the air velocity distribution described later. In addition, outdoor temperature, humidity, solar radiation, wind direction, wind velocity, atmospheric pressure and precipitation were measured with a Davis Weather Station.



● Temperature measurement (Thermocouple 0.32T-G)
⊗ Temperature/Humidity measurement (Thermocouple 0.32T-G, Wet gauze)

(air outlet)

(air inlet)

Figure 2. Temperature and humidity measurement points of a indoor unit

Table 3. Measurement specification of indoor units

Position	Item	Sensor	Recital	Point (Summer)	Point (Winter)
Air outlet	Temperature	Thermocouple 0.32T-G		9×4=36	9×4=36
	Humidity	Thermocouple 0.32T-G	The wet-bulb temperature measurement by the moisture gauze	3×4=12	1×4=4
Air inlet	Temperature	Thermocouple 0.32T-G		5	5
	Humidity	Thermocouple 0.32T-G	The wet-bulb temperature measurement by the moisture gauze	5	1

Table 4. Specification of indoor units

Name		ACM 8-1 (1Unit)	ACM 8-2 (2 Unit)	ACM 8-3 (3 Unit)
Electric source		Single phase 200V, 50/60Hz		
Capability	Cooling	4.5kW	9.0kW	7.1kW
	Heating	5.0kW	10.0kW	8.0kW
Electric power consumption	Cooling	0.07/0.08kW	0.11/0.12kW	0.085/0.09kW
	Heating	0.07/0.08kW	0.11/0.12kW	0.085/0.09kW
Power current	Cooling	0.35/0.4A	0.55/0.6A	0.425/0.45A
	Heating	0.35/0.4A	0.55/0.6A	0.425/0.45A
Air volume system		Very Strong(15m3/min), Strong(12), Week(10)	Very Strong(20m3/min), Strong(15), Week(12)	Very Strong(16m3/min), Strong(13), Week(11)

Table 5. Specification of outdoor unit

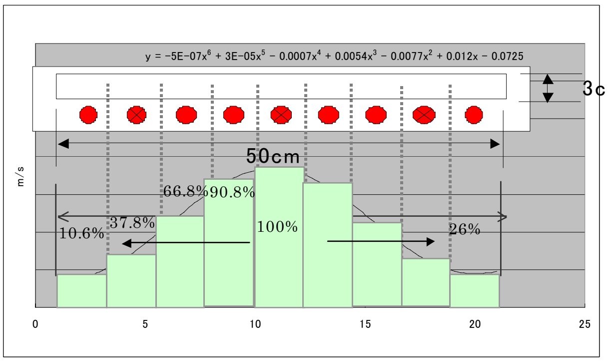
Name		ACM-5,7,8,11
Electric source		Three phase 200V, 50/60Hz
Capability	Cooling	56.0kW
	Heating	53.0kW
striking current		Cooling 168/155A, Heating 166/153A
Electric power consumption	Cooling	21.18/22.48kW
	Heating	18.30/20.10kW
Power current	Cooling	67.6/72.1A
	Heating	59.3/65.1A

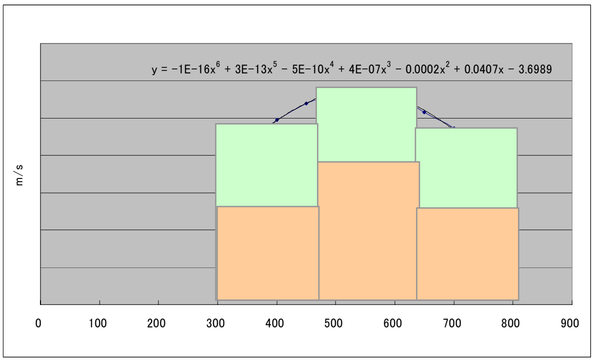
(3) Electricity consumption by outdoor and indoor units
Electricity consumption of 7 indoor units, 4 outdoor units and their control systems were measured every minute by attaching clamp watt meters to the power line.

Specification of measured PAC is shown in table 4 and 5. The rated COP on the specification is 2.64 for cooling and 2.90 for heating.

RESULTS

(1) Airflow calculation
Airflow volume was calculated by airflow distribution model (Figure 3) based on 3 dimensional velocity distribution measured by the traverse unit equipped with measuring devices developed by professor Shigeki Kametani of Tokyo University of Marine Science.





(a) (b)

Figure 3. Wind velocity Ratio of air outlet(a) and inlet(b)

Wind inlet and outlet velocity distribution measured by the traverse were expressed with an approximate curved line which gives a wind velocity ratio at a random point to the central point. Using the curve, mean wind velocity of each measured point and its representing area was calculated and applied to thermal flow calculation. A sum of the calculated airflow volume was adjusted to match the rated volume for “strong wind” mode.

36.5%	60.4%	36.5%
60.4%	100.0%	60.4%
36.5%	60.4%	36.5%

96.9%		96.9%
	100.0%	
96.9%		96.9%

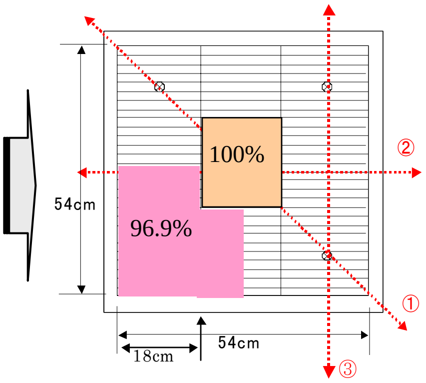


Figure 4. Wind velocity distribution of air inlet

(2) Result of summer measurement
Outside air temperature, humidity and electricity consumption of indoor and outdoor units throughout summer measurement are shown in Figure5 and 6. Room temperature is shown in Figure 7 for reference.

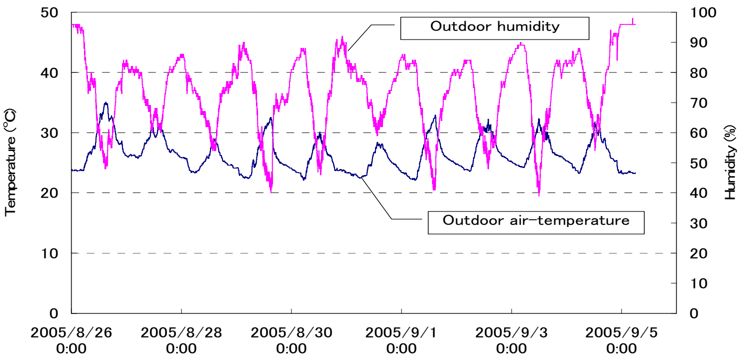


Figure 5. Outside air temperature and humidity (summer measurement)

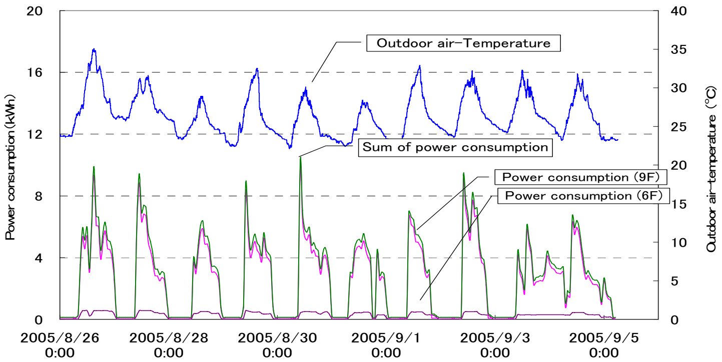


Figure 6. Electricity consumption (summer measurement, indoor: 6F, outdoor: 9F)

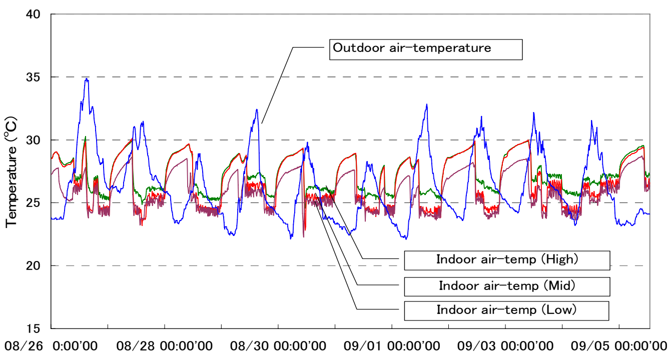


Figure 7. Outside air and room temperature (summer measurement)

The change of COP during operating time (10am-9pm) in summer measurement is shown in Figure 8. Measured COP was considerably lower than rating COP (=2.64).

COP was calculated by the following formula.

$$\mathrm{COP} = \frac{\text{Heat exchange quantity}}{\text{Electric power consumption of indoor and outdoor units}} \qquad (1)$$

= 715.806 [kWh] / 411.382 [kWh] = 1.74 (10am-9pm, August 26-September 5, 2005)

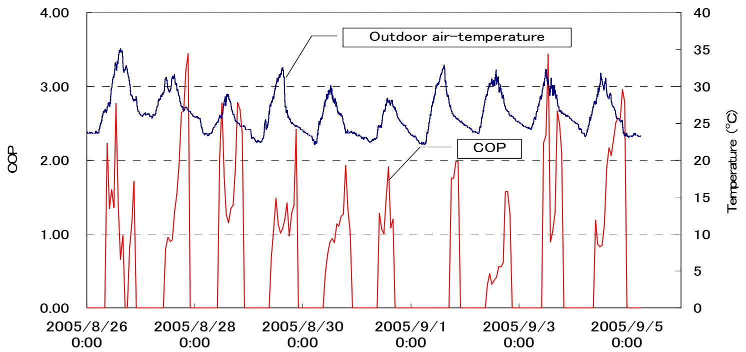


Figure 8. Change of COP (summer measurement)

Figure 9 and 10 show distribution of load factor. Most of the load is concentrated under 30% of PAC's capacity.

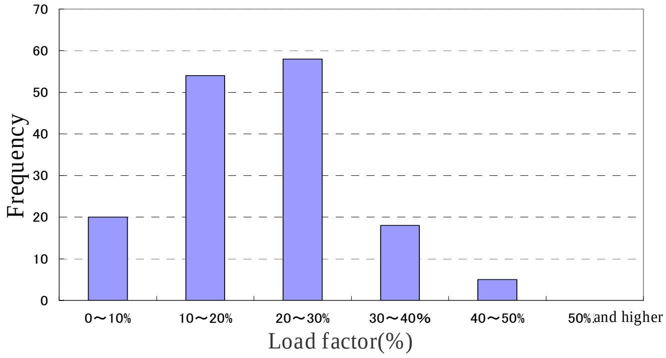


Figure 9. Frequency distribution of load factor (summer measurement)

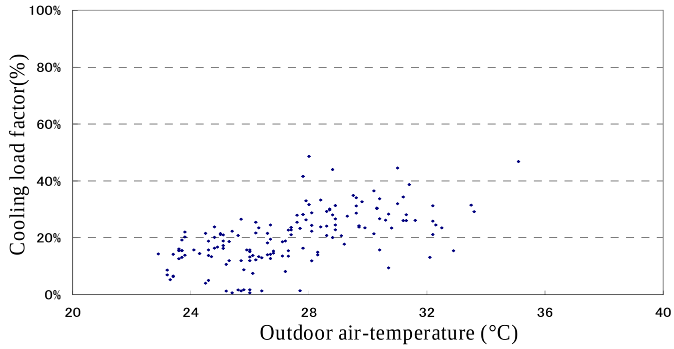


Figure 10. Relation between outside air temperature and load factor

(3) Result of winter measurement

Outside air temperature, humidity and electricity consumption of indoor and outdoor units throughout winter measurement are shown in Figure 11 and 12. Room temperature is shown in Figure 13 for reference.

The change of COP during operating time (10am-9pm) in winter measurement calculated by formula (1) is shown in Figure 14. Measured COP was quite lower than rating COP (=2.90).

COP= 92.700 [kWh] / 153.898 [kWh] = 0.60 (10am-9pm, February 14 - 22, 2006)

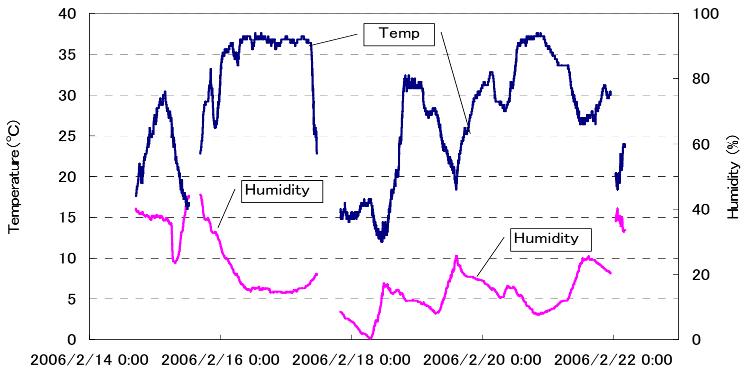


Figure 11. Outdoor air temperature and humidity (winter measurement)

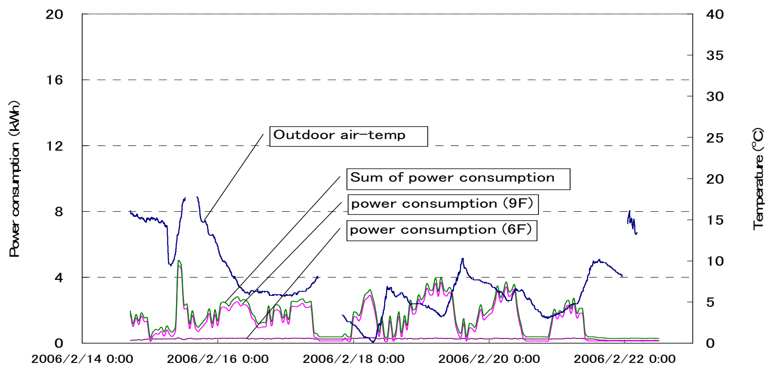


Figure 12. Electricity consumption (winter measurement, Indoor : 6F, outdoor : 9F)

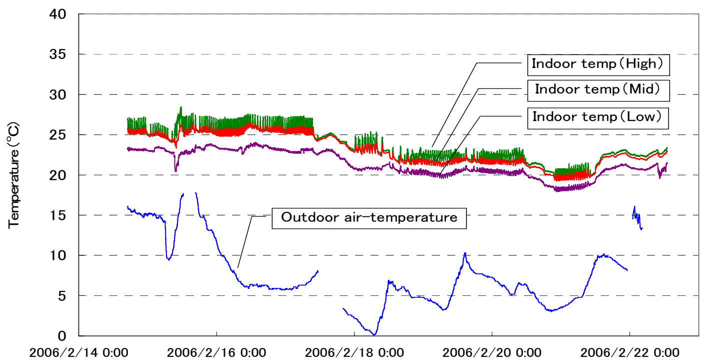


Figure 13. Outdoor air and room temperature (winter measurement)

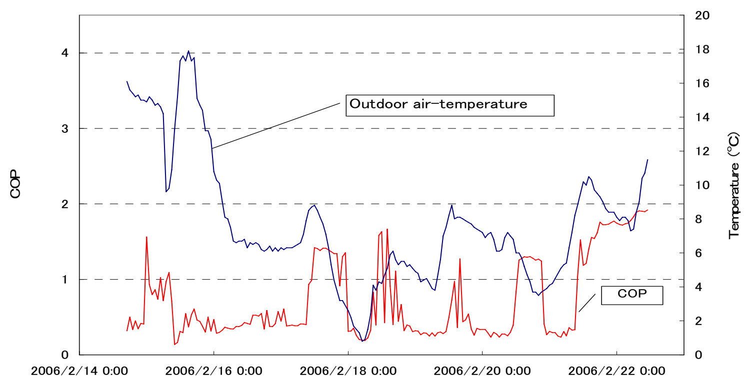


Figure 14. Change of COP (winter measurement)

Distribution of load factor during winter measurement is shown in Figure 15 and 16. Most of the load is concentrated under 20% of PAC's capacity. Load Factor was lower in winter because relatively high outdoor air temperature prevailed during measurement and the inevitable problem of excess heating capacity for a PAC designed to meet maximum summer peak load has come to the surface.

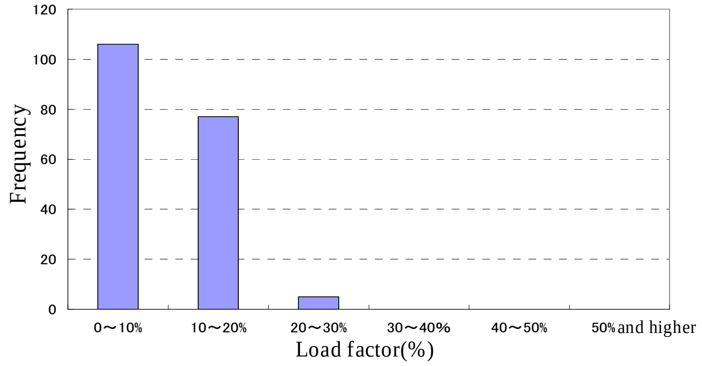


Figure 15. Frequency distribution of load factor (winter measurement)

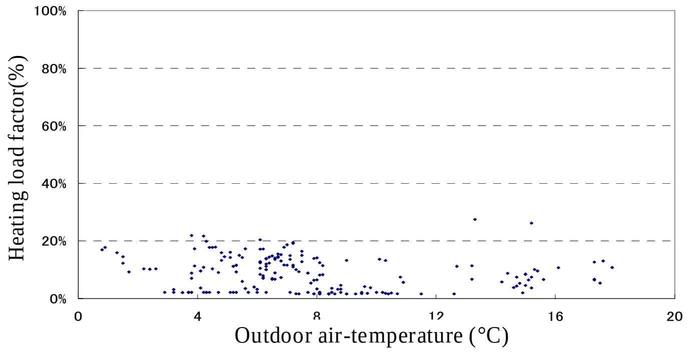


Figure 16. Relation between outdoor air temperature and load factor (winter measurement)

DISCUSSION

In this study energy efficiency of a PAC under actual operating environment was measured by the "multi point measurement method". The result revealed considerably low COP for cooling and very low COP for heating in comparison with their rated values. It also became clear that PAC's capacity far exceeding actual load is causing operation under extremely low load factor which prevents it running at expected efficiency. In addition, if global warming worsens, imbalance between summer and winter load will become greater, creating even lower partial load operation for heating. To solve these problems, accurate load calculation with reliable software is required. And also, a technological breakthrough to integrate PAC's heat load into fewer number of outdoor units to operate them at higher load factor is expected.

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R&D Case: New Holistic Air Conditioning System

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SUMMARY

A new air conditioning system was developed to combine low energy consumption, good indoor conditions and competitive investment costs. Computer simulation was utilized to develop the system operation and dimensioning principles and design tools. The generated HVAC-solution comprises several functions, which are usually separate. It takes care of heating, cooling, ventilation, illumination and electric distribution of the rooms. The integrated system operates with low temperature heating and high temperature cooling principles. Internal heat loads are utilized efficiently in heating. An advanced free cooling process with outdoor air is used in cooling. The system has been build in several office buildings. The measured energy consumptions and indoor conditions fulfill the strict target values. The user satisfaction is high. The work and results showed that simulation is an efficient and reliable tool for system development and design. Low consumption, good conditions and a competitive price level can be combined.

INTRODUCTION

The HVAC, control and electrical systems, which maintain indoor conditions, are usually separate due to historical and commercial reasons. They function typically quite independently. The goal was to develop a new building services system, which integrates various functions and thus improves energy efficiency and indoor conditions of office and public buildings. Further the design methods and practical solutions were to be generated to make the system as a commercial product. The final task was to measure the system performance in a real building and compare it to the theoretical values.

The main objectives of the development project were set already in the year 1996. Good indoor conditions, low energy consumption and competitive investment costs were self-evident goals for the new system. The fourth goal, low environmental impact, was fulfilled through the low energy consumption.

METHODS

System simulation was chosen as the development method due to the large variety of different operating situations during the four climate seasons of the Nordic climate. Scientific university owned and commercial simulation programs were studied. The programs, which were found were either very laborious or too superficial. Finally a simulation environment, which was still under development was chosen. The Royal Institute of Technology in Stockholm was creating a program named IDA [1] [2] with the assistance of Helsinki University of Technology.

The development procedure was straightforward. New system and control principles were innovated. The systems were programmed in IDA and their operation in different outdoor conditions were studied by simulation. Finally simulations throughout the year were carried out. As a whole over one hundred simulations were done before all system properties were defined. A Finnish reference building "HVAC-2000" was used in the models. The building is an imaginary office building designed for research purposes. Several new HVAC-equipment models were created to make the system simulations possible.

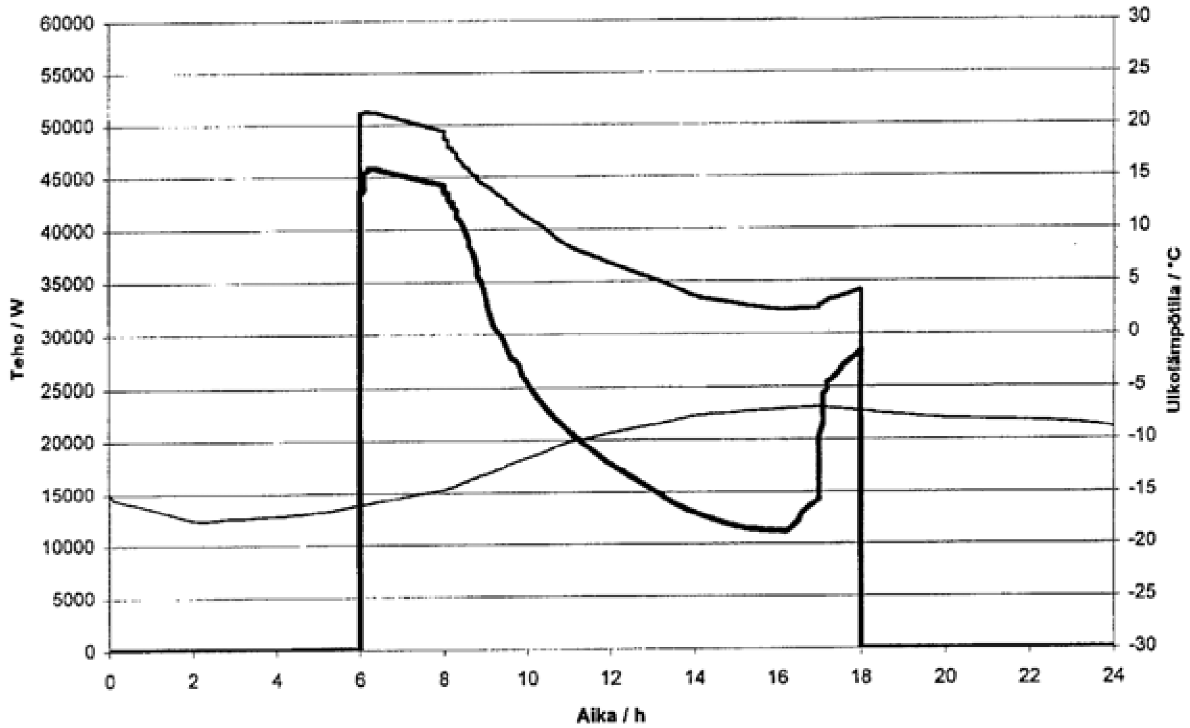

Figure 1. The operation and performance of the new system under development were compared to a conventional HVAC-system by simulation. As an example the heating power need (W) of an air conditioning unit during a winter day is shown. Uppermost curve is a conventional unit and the lower curve is the new system unit. Third curve is outdoor temperature (°C) [3].

The new system operates with low temperature heating – high temperature cooling bases, which is called also low exergy principle [4]. The air heating and cooling coils and fluid heat exchangers operate with very small temperature differences compared to the conventional systems. The behavior of these coils was studied theoretically and with laboratory measurements [3]. The results were programmed in the coil simulation model as initial values.

RESULTS

System principles

Modern office houses require long periods of time simultaneous heating and cooling in cool and moderate climates to maintain comfortable indoor temperatures. Cooling is often needed already in below zero outdoor temperatures. The excess heat from room cooling is discarded

out and at the same time heat is brought in the HVAC-unit and in the rooms, which need heating.
The heating and cooling energy saving principle of the system is simple in the new system. The excess heat from room cooling is not rejected out but brought in the HVAC-unit where heat is needed. The HVAC-unit cools simultaneously the cooling water of the room units with cool outdoor air (free cooling). This reduces the energy consumption of mechanical cooling.

Large heat exchange surfaces are needed in heating and cooling devices of the low exergy systems due to the small temperature differences available. Hence the same devices are used both in heating, free cooling and mechanical cooling in the HVAC-unit and in the room units. The total heat exchange surface is not larger than in conventional systems, which have separate units for heating and cooling. This makes the new system also commercially competitive.

The system has only three pipes to the room units, the heating, cooling and return pipes. Using the common return pipe would increase the energy consumption in conventional systems. In the new system this decreases the consumption, because the return heat is utilized.

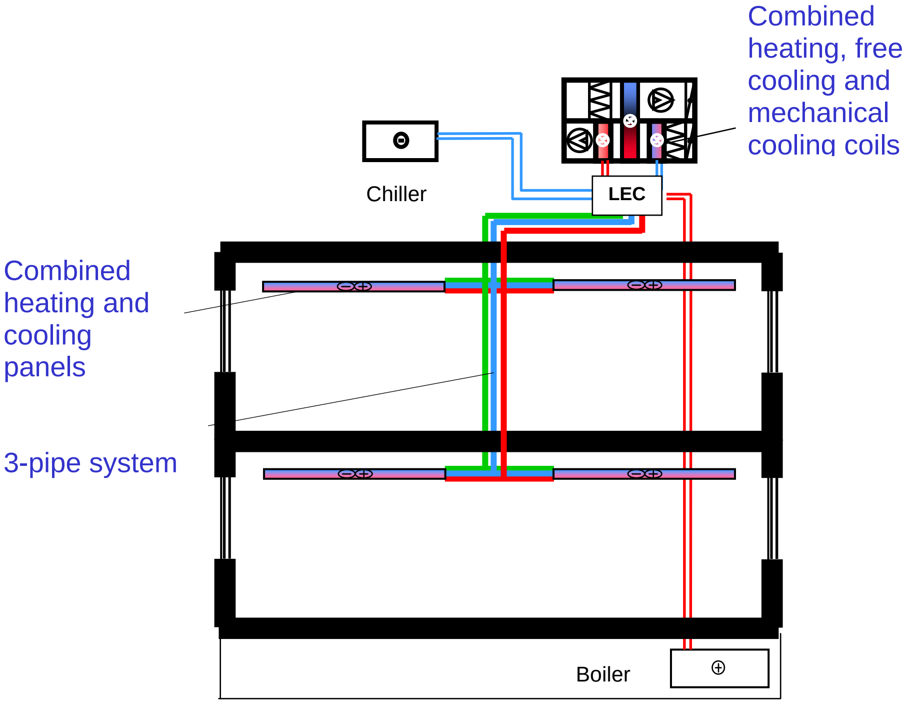

Figure 2. The system has less components than traditional HVAC-systems. The same devices perform several functions. Low energy center LEC transfers heating and cooling energy inside the building reducing the need of external energy [5].

System verification, energy consumption and indoor conditions

The system follow up with the building energy management system show that energy recycling inside the building and free cooling operate practically throughout the whole year. Free cooling is needed in the spaces with high internal heat loads and good thermal insulation even when the outdoor temperature is very low in the winter. On the other hand the new type of free cooling operates nighttime even during the warmest seasons cooling the building structures and reducing the daytime cooling need.

The studies [6, 7] compare the measured and the theoretical system operation with each other in a large pilot building. The studies show that the system operates as designed. The free cooling produces over 70% of the total yearly cooling need.
Measured heating energy consumptions of several buildings equipped with the new system are 25%-40% lower than the average values in buildings of similar type and age (comparison data by Association of Finnish local and regional authorities and by local district heating companies).

The studies [6, 7] show that measured indoor conditions fulfill the design target level S2 of the Finnish indoor classification [8]. Conditions are within the best class S1 about 90% of the time. An international research Hope [9] studied the possibilities and means of combining good indoor environment and low energy consumption. The indoor conditions in a building equipped with the new system was ranked best among eight Finnish office buildings by the users.

DISCUSSION

The project showed, that system integration brings about advances compared to conventional separate systems. Excellent indoor environment, low energy consumption and competitive investment cost is combined in the new holistic HVAC-system.

HVAC-system simulation is often considered mainly as a research implement for scientists. The project proved that simulation is an efficient tool for developing practical solutions and for verifying system function in real use.

Individual HVAC-equipments and units are factory made and tested. Their properties are well known. However total HVAC-systems are designed and build without testing their function as a whole. System dimensioning is made in the extreme steady state design conditions. Operation in other conditions is not examined. Hence system simulation, testing, verifying and follow up in a larger scale would reduce the total energy consumption of the building stock along constructing new and renewing old buildings.

ACKNOWLEDGEMENT

The main development project was done by the HVAC-laboratory of Helsinki University of Technology and Energy laboratory of Lappeenranta University of Technology. The project was funded partly by Tekes.

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Thermal climate requirements and the effects on environmental performance of comfort cooling systems

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SUMMARY

There is a lack of knowledge of how significantly various indoor climate criteria affect the overall environmental performance of air-conditioning systems. The purpose of this study is, therefore, to investigate the relation between environmental performances of four types of air-conditioning systems and five various criteria on indoor thermal climate. Furthermore, the relation between the specific fan power (SFP) factor and the overall environmental performance of the systems studied is investigated. The annual operating energy of each of the systems is modelled for the five indoor thermal climates. The environmental performance is evaluated by the life cycle assessment (LCA) method, including weighting by EPS 2000d.

INTRODUCTION

Elevated requirements on the thermal climate in offices imply increased cooling loads, and to round the year operation of air-conditioning systems even in rather cold climates. This trend leads to an extended energy use, which consequently affects negatively the environmental performance of air-conditioning systems since energy use often is the main contributor to the overall environmental impacts. There is a lack of knowledge of how significantly the indoor climate criteria affect the overall environmental performance of these systems. Heikkilä and Lindholm [1] show no significant difference in environmental performance of a particular air-conditioning system designed for the most usual requirements on thermal climate. However, that study is based on one air-conditioning system operating in one particular office building. The purpose of this study, therefore, is to investigate the relation between environmental performances of several types of air-conditioning systems and various indoor thermal climate criteria. Furthermore, the relation between the specific fan power (SFP) factor and the overall environmental performance of the systems studied is investigated.

METHOD

The analysis is based on different case studies focused on environmental evaluation of four types of air-conditioning systems. For each of the systems, the annual operating energy is calculated for five different indoor thermal climates. The demanded operating energy, as well as the airflow volumes that correspond with the various thermal climates, are estimated by the software application ‘The heat balance of buildings’ [2], known as BV^2, which is a software commonly used in the design practice in Sweden. The different requirements on thermal quality (TQ) are summarized in Table 1. The TQ1, TQ2, and TQ3 criteria represent requirements that are frequently used in Sweden; the more extreme requirements are represented by TQmin (low requirements) and TQmax (high requirements).

Table 1. Criteria for the thermal indoor climate used in the analysis

Criteria for the indoor thermal climate	t_{air}		
	Minimum	Set point	Maximum
TQmax	20 °C	21 °C	22 °C
TQ1	23 °C	24,5 °C	25,5 °C
TQ2	22 °C	24,5 °C	26,5 °C
TQ3	21 °C	24,5 °C	28 °C
TQmin	20 °C	24,5 °C	35 °C

The buildings and the air-conditioning systems

The systems included in this analysis are designated as a bore-hole based system, a traditional system, an air-and-water system, and an all-air system. The bore-hole based system and the traditional one are alternatives designed for the same building designated as *Medicinaberget*. This three-storeyed building, detailed described in Heikkilä [3], has a total net area of 1920 m^2, which is divided into cell offices. The air-and-water and the all-air systems are alternatives for another office building known as *Arendal*. This building is six-storeyed, and the total area of 17 000 m^2 comprises an office landscape with work places of 12 m^2; for more details, see Heikkilä and Fahlén [4]. Both buildings are situated in Göteborg, Sweden.

In the *bore-hole based system*, the heat surplus is lead off from offices by an all-air system that comprises an air-handling unit and an air-distribution system. This system is controlled by a variable airflow volumes (VAV) control strategy, and operates for 11 hours (7 am to 18 pm) each working day, 5 days a week. The cooling energy is supplied by a bore-hole based heat pump system, which supplies 33 % of the cooling energy required. The rest of the cooling energy is supplied by free cooling directly from the ground.

The *traditional system* includes an air-handling unit and an air distribution system, of the same size and composition as the bore-hole based system, but the cooling energy is supplied by a vapour compression chiller. The chiller is designed to cover the whole cooling loads. The operating time and control strategy are the same as in the bore-hole based system.

The *air-and-water system* consists of an air-handling unit, a vapour compression chiller, an air distribution system and a water distribution system. The operating energy is determined for a constant airflow volume (CAV) operation, but with differentiation between the airflow volume during the night and during the day. The CAV strategy is chosen taking into account the limitation of the program that does not enable VAV operation for an air-and-water system.

Similarly, the *all air system* contains an air-handling unit, a vapour compression chiller and an air distribution system. The operating energy is determined for VAV control strategy.

The environmental evaluation

The environmental performance of the systems is evaluated by the life cycle assessment (LCA) method, taking into account the whole life cycle of the system: the production, the user, as well as the disposal stages. The impact assessment is performed by the weighting methodology EPS 2000 default method [5]. For purposes of this analysis, however, the results are normalized with respect to the environmental impact of the all-air system for *Arendal* that affects the environment with the highest environmental burden among the four systems studied.

The environmental performance is studied for two different scenarios:

In *Scenario A*, the specific fan power (SFP) factor is the same for all variations of the indoor thermal criteria; the SFP factor describes the efficiency of operation of fans and is given in kW/(m^3/s). The size of the air-handling unit in this scenario changes with each TQ class due to the various airflow volumes required. In *Scenario B*, the desired indoor thermal climate is met by operation of the same size of air-handling unit originally designed for the TQ2 class; i.e. the SFP factors in this scenario vary with the different airflow volumes required in the various TQ classes. Table 2 summarizes the demanded airflow volumes for the four systems included in this analyses that correspond with the various TQ classes.

Table 2. The airflow volumes of the four different systems that correspond with the various TQ classes

Criteria for the indoor thermal climate	Design air flow volume l/(s.m^2)				
	Bore-hole based system	Traditional system	Air-and-water system		All-air system
			Night	Day	
TQmax	4,5	4,5	1,6	2,3	5,4
TQ1	2,8	2,8	1,6	2,3	3
TQ2	2,5	2,5	1,6	2,3	2,7
TQ3	2,1	2,1	1,6	2,3	2,3
TQmin	1,2	1,2	1,6	2,3	1,2

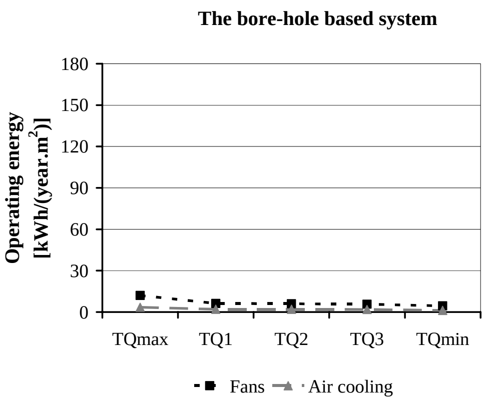
The environmental evaluation of the bore-hole based system and the traditional one is based on results of the LCA study described in Heikkilä [3]. The environmental performance of the air-and-water system and the all-air system refers to the study of Heikkilä [6] which, for the purposes of this analysis, was complemented with environmental impacts related to the disposal stage since the original case does not include this phase. Furthermore, *Scenario A* implies that the size of air-handling system varies for the various TQ criteria. The environmental impacts related to the various sizes are modified (enlarged or reduced) in accordance with the ratio of the actual system to the system from the reference case. The reference size of the air-handling units is that originally designed for the for the TQ2 class.

RESULTS

Operating energy of the various systems

Figure 1 shows the variation of operating energy of the four systems with the different TQ criteria in *Scenario A*; i.e. the SFP factor is the same for all TQ criteria.

Basically, the range of the variations of the total operating energy for fans, air- and water cooling varies with the different systems. For all TQ classes, the total demand of operating energy for systems in *Medicinaberget* is small compared to the systems in *Arendal*, mainly due to the efficiency of fans operation. Fans are the main users of the operating energy in all systems. The systems in *Medicinaberget* work in accordance with SFP of 1,1 kW/(m^3/s), the SFP factor for the systems in *Arendal* is 2,5 kW/(m^3/s). The cooling systems are the minor contributors to the total energy use, and the amounts of the required energy are affected by the specific design of each system. For example, the bore-hole based system requires only 1/3rd of demanded energy due to the possibility to use ‘free cooling’ energy directly from the ground, which is not possible in the traditional system.



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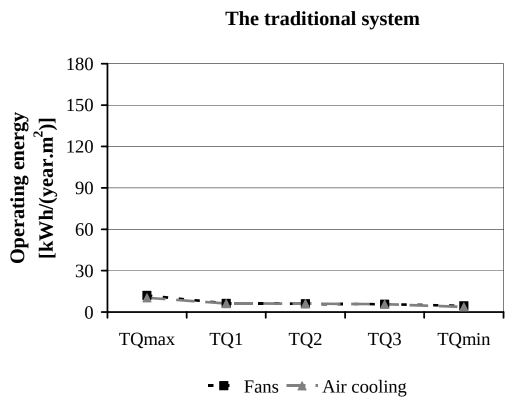

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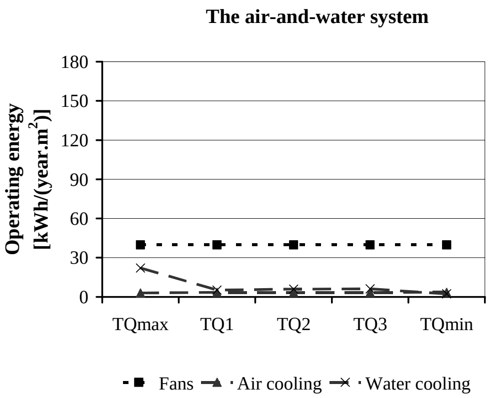

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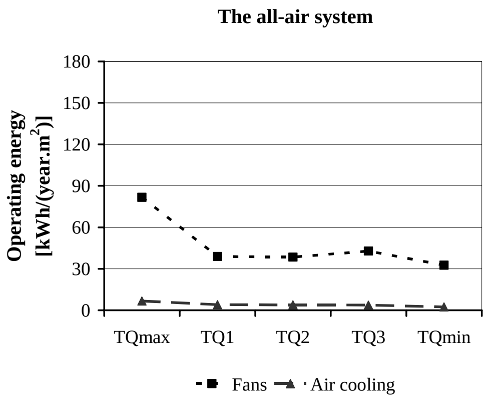

d)

Figure 1. Operating energy of the four air-conditioning systems with the same SFP factor: of 1,1 kW/(m^3/s) for a) the bore-hole based system, and for b) the traditional system; and of 2,5 kW/(m^3/s) for c) the air-and-water system, and for d) the all-air system.

Figure 1 shows, furthermore, the effect of the various thermal climate criteria: the energy demand increases with the higher requirements. The increase is moderate for the usual range of the thermal indoor climate criteria, TQ1 to TQ3, but more marked for the two extreme classes. The most significant increase of energy demand is for operation of fans to achieve TQmax criteria. In *Scenario A*, the SFP factor remains the same for all the TQ classes and systems respectively, therefore, the increased energy amounts are related to the extended operation of the systems, as well as to increased airflow volumes corresponded with the higher TQ criteria (see Table 2). Only the air-and-water system distributes the same amount of airflow volume for all TQ classes since the increasing demand of cooling energy is supplied by the water system. However, the tendency in increase of energy demand with higher TQ classes is different for the all-air system shown in Figure 1d), which requires more energy for operation of fans in TQ3 class compared to TQ2 class. This can be explained by an increased demand of night cooling during the periods of year with the highest outdoor air temperatures, which leads to more extended operation of fans.

Figure 2 presents the operating energy for the different systems calculated in *Scenario B*; i.e. the systems operate with the same size of air-handling unit, which leads to various SFPs. The trend in increase of operating energy with higher requirements on the thermal climate remains also in *Scenario B*, but the increase is more rapid in the TQmax due to the less efficient fan operation. Moreover, the air-and-water system shown in Figure 2 c) operates in both scenarios identically; this system maintains a constant airflow volume in all TQ classes, and the increased cooling demand is supplied by the water cooling system.

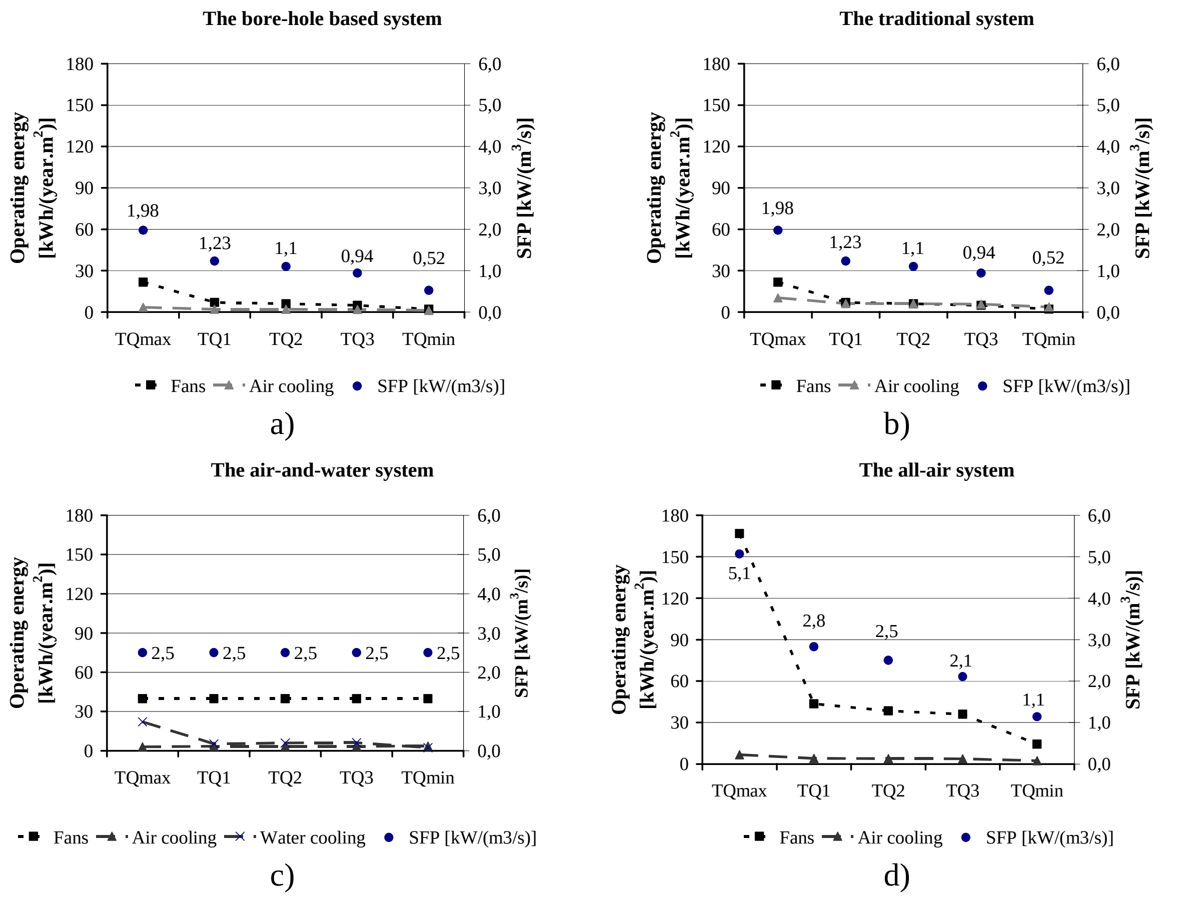

Figure 2. Operating energy of the four air-conditioning systems with various SFPs for a) the bore-hole based system, b) the traditional system, c) the air-and-water system, and for d) the all-air system.

Environmental performance

Figure 3 shows the environmental performances of the four systems in *Scenario A*, i.e. the SFP factor is assumed to be the same for all TQ classes and system respectively. The unbroken line, which is linked to the secondary y-axis, shows the demanded airflow volumes for each of the TQ classes. The total environmental performance of the different systems varies with the type of the air-conditioning system. Also the specific design and the particular preconditions affect the total environmental performance; for example, the total environmental impacts of the alternative systems for *Medicinaberget* is lower compared to the systems for *Arendal*.

For all the systems studied, the total environmental impacts increase with the higher requirements on TQ climate. The increase is moderate within the interval of TQ1 to TQ3 classes, but more rapid for the TQmax and TQmin classes. The operating energy contributes the most to the total environmental impacts as well as to the increasing trend.

The environmental impacts related to the material stage, represented by component materials and EoL treatment, follow the increasing trend for the higher TQ classes. The increase is most obvious for the TQmax class due to the increased airflow volume, with the exception of the air-and-water system that operates with a constant airflow volume for all TQ classes.

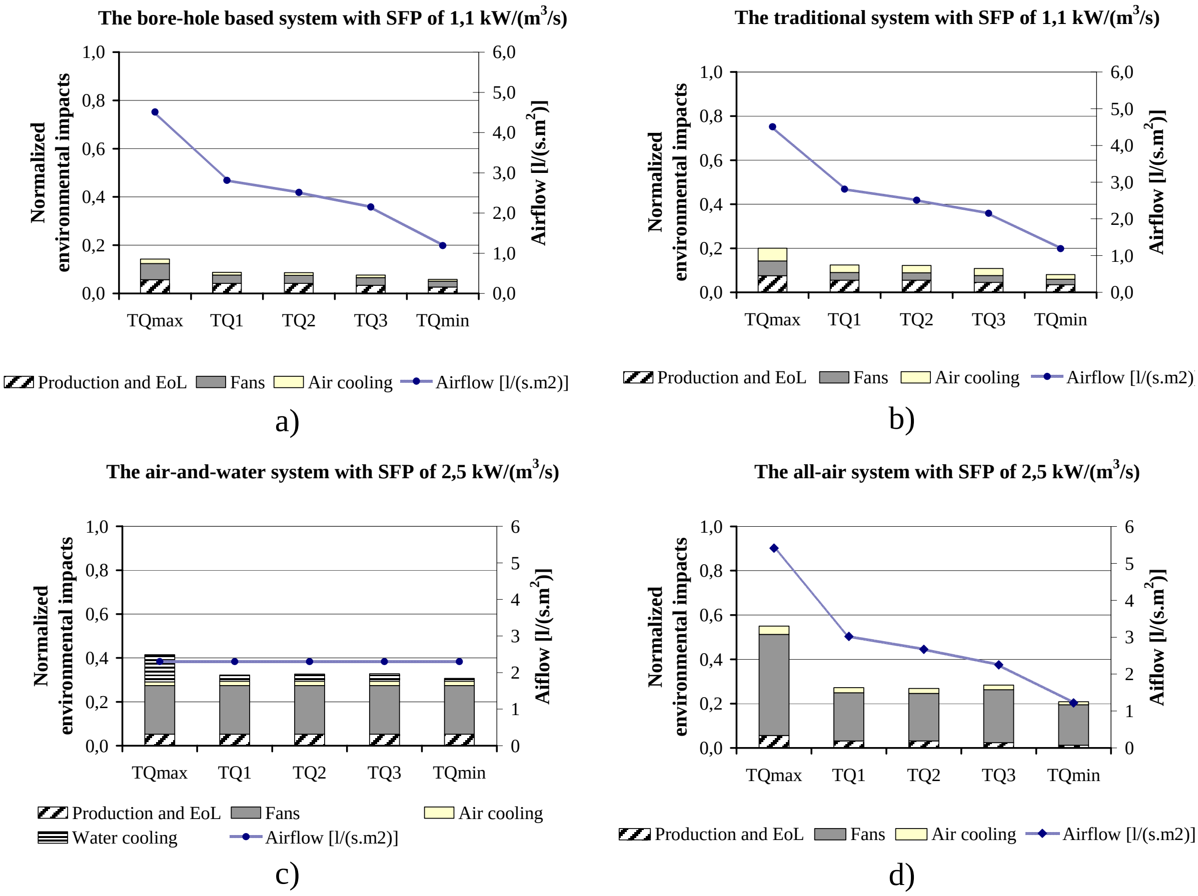

Figure 3. Environmental impacts of the various systems with the same SFP of 1,1 kW/(m^3/s) a) for the bore-hole based system, and for b) the traditional system; and of 2,5 kW/(m^3/s) c) for the air-and-water system, and for d) the all-air system.

However, the proportions between the contribution of the materials stage and the user stage to the total environmental impact differ for the alternative systems in *Medicinaberget* and *Arendal*. In the alternatives for *Medicinaberget*, in the bore-hole based and the traditional systems, the material stage contributes with 38 to 45 % (see Figures 3 a) and b)). These systems are designed taking into account specific requirements on a low energy use to maintain the desired indoor climate; especially, the bore-hole based system can be considered as an exceptionally energy efficient system. The average total environmental impact of these systems is about 50 to 65 % lower compared to the systems designed for *Arendal* which are more traditional air-conditioning systems. The corresponding contribution of the material stage of *Arendal* systems, the air-and-water and of the all-air systems, is 6 to 17 %, see Figure 3 c) and Figure 3 d).

Figure 4 presents the environmental performance of the four systems in *Scenario B*; i.e. the systems operate with the same size of air-handling unit in all TQ classes which leads to different SFPs. Also in this scenario, the trend in increase of environmental impacts with a higher TQ class remains, but it is more marked especially for the all-air system, see Figure 4 d). For all systems, the highest environmental impact is related to TQmax class. The main contributor to the environmental impact is the user stage as a result of a less efficient fan operation; for the SFP factors see Figure 2.

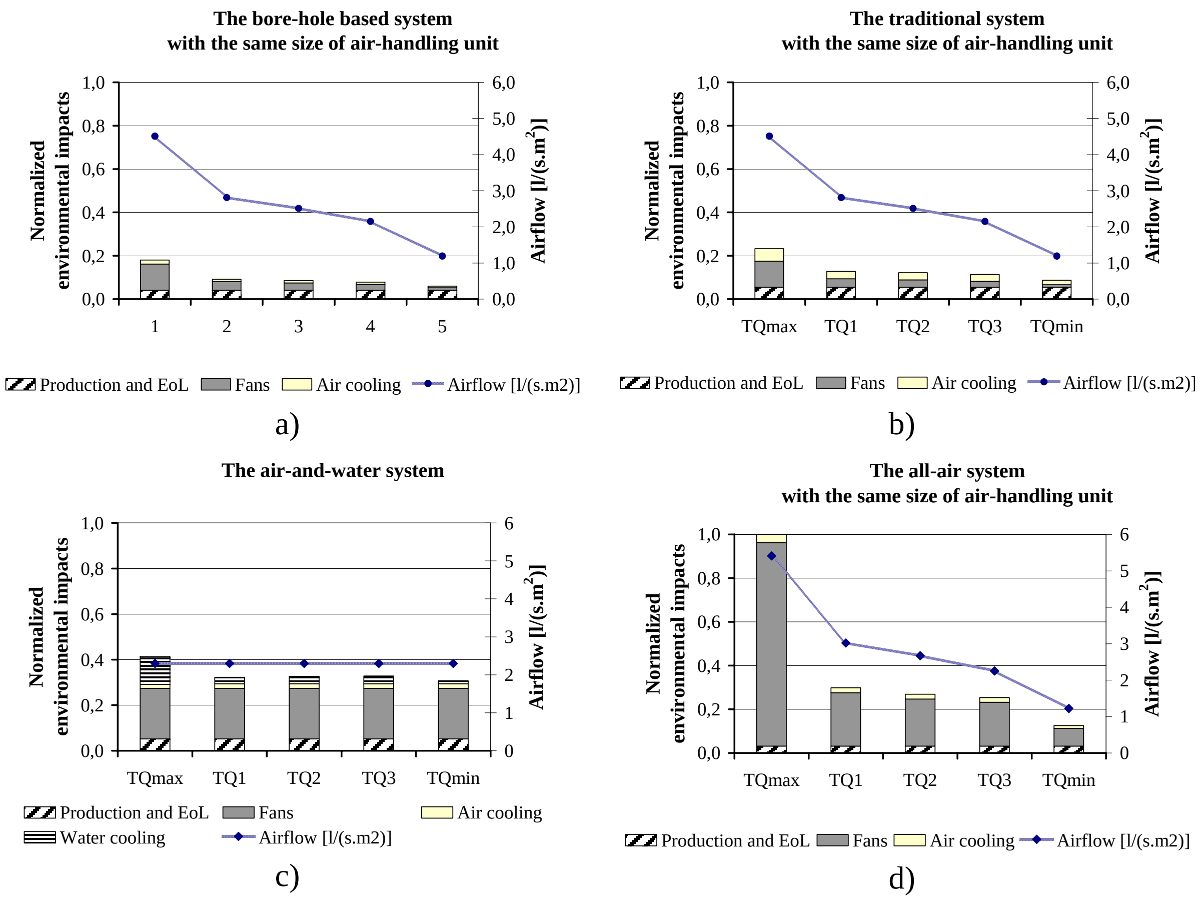

Figure 4. Environmental impacts of the various systems with various SFPs for
a) the bore-hole based system, b) the traditional system;
c) the air-and-water system, and for d) the all-air system.

The difference in contribution of the materials stage to the total environmental impacts between the systems designed for *Medicinaberget* and those for *Arendal* is even more marked in the *Scenario B*. For example, for the TQmin to TQ1 classes, the material stages in *Medicinaberget* systems contribute with 50 % in average; in *Arendal* systems, the corresponding contribution is about 15 %. The environmental performance of the air-and-water system is the same as in *Scenario A*. In both scenarios, this system operates with the same conditions for each of the TQ classes.

DISCUSSION AND CONCLUSIONS

In this study, the environmental performance of four types of air-conditioning systems was investigated, as well as their environmental performance in relation to various indoor thermal climate criteria. Table 3 presents the total normalized environmental impacts (EI_{Norm}) for the various TQ classes, and the corresponding SFPs. The results are based on several assumptions for the estimation of operating energy, as well as of the environmental performance. However, the same assumptions were made in all cases; more refined assumptions would, therefore, have only negligible impact on the proportions of the results.

Based on the results, the following observations can be made: for each of the different systems, there is not a significant difference in environmental impacts within the range of TQ1 to TQ3 criteria. The various TQ criteria affect significantly the amount of the demanded operating energy; the effect on the material stage is moderate. However, each decrease of the environmental impacts related to the material stage, however, contributes positively to the total environmental performance of all systems. The results, moreover, indicate that within the range of TQ1 to TQ3, the SFP factor could be used as an indicator of the predicted environmental performance. As shown in Table 3, the environmental performance within this range does not vary significantly. For TQmin and especially for TQmax, the environmental performance can differ more compared to the range of the most usual TQ requirements. To keep the environmental impacts as low as possible, it is not recommended to use a SFP factor higher than 2,5 kW/(m^3/s).

Table 3. Normalized environmental impacts (EI_{Norm}) of the four systems studied for various TQ classes in *Scenario A* and *Scenario B*

	Bore-hole based system		Traditional system		Air-and-water system		All-air system	
	SFP	EI_{Norm}	SFP	EI_{Norm}	SFP	EI_{Norm}	SFP	EI_{Norm}
	[kW/(m^3/s)]	[-]	[kW/(m^3/s)]	[-]	[kW/(m^3/s)]	[-]	[kW/(m^3/s)]	[-]
Scenario A								
TQmax	1,1	0,14	1,1	0,2	2,5	0,41	2,5	0,55
TQ1	1,1	0,09	1,1	0,12	2,5	0,32	2,5	0,27
TQ2	1,1	0,09	1,1	0,12	2,5	0,33	2,5	0,27
TQ3	1,1	0,08	1,1	0,11	2,5	0,33	2,5	0,28
TQmin	1,1	0,06	1,1	0,08	2,5	0,31	2,5	0,21
Scenario B								
TQmax	1,98	0,18	1,98	0,23	2,5	0,41	5,1	1
TQ1	1,23	0,09	1,23	0,13	2,5	0,32	2,8	0,3
TQ2	1,1	0,09	1,1	0,12	2,5	0,33	2,5	0,27
TQ3	0,94	0,08	0,94	0,11	2,5	0,33	2,1	0,25
TQmin	0,52	0,06	0,52	0,09	2,5	0,31	1,1	0,13

The results presented in this paper do not diverge from the results of previous studies of the author [1]. The highest criterion, TQmax, implies the highest environmental impact. The requirements on the indoor climate should, therefore, be established at appropriate levels to reflect the real demand in each particular case, taking into account the environmental performance of the air-conditioning system. Communication of the environmental consequences of the desired indoor climate to the commissioner of a building project or future owner is also essential.

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Energetic Evaluation of Air Conditioning Systems

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SUMMARY

The paper describes an approach to calculate primary energy demands for air conditioning systems. These values are not attached to specific types of air conditioning systems but to the use of the air conditioned space. In a first step the minimum for the primary energy demand is calculated on the basis of ideal thermodynamic processes. In a second step an analogue calculation is being made for real processes. A comparison between both shows the range of possible primary energy demands. From that a fixed value can be derived for future limitations of the primary energy demands of designed air conditioning systems, which has to be proved during the approval design.

INTRODUCTION

The “Energy Performance of Buildings Directive” (EPBD) requires an energetic evaluation of buildings and their technical systems. In Germany, an “Energy saving directive” exists since 2002 for buildings and heating systems including domestic hot water systems. This directive follows former legal limitations, which were first introduced in 1976.
New aspects in EPBD are the air conditioning systems and the lightning systems.
This will be a legal requirement for the design and building approval. During the approval design of VAC-system a calculation has to be done for the energy demand of the designed system and that then has to be compared with legally given maximum values for the demand. In Germany, these maximum values are given by the new “Energy saving directive”. This regulation is based on the German Standard DIN V 18599. The maximum values insofar are derived by a “reference-building method”.
For the evaluation of the energy demand of VAC-Systems exist so far no appropriate technical rules. In order to force energy conservation, a limitation of the energy requirement for VAC-systems is necessary.
The research project shows by an example a new way to get energetic characteristic values to evaluate the energy demand of VAC-systems.

METHOD

Ideal Thermal Processes

For each case, there is one defined theoretical process for the VAC-system. This process has the minimum energy demand. It is called the ideal process. Figure 1 shows a psychrometric chart with a set point area for the supply air temperature and humidity. This set point area (S) was transformed out of the room conditions, considering the loads.

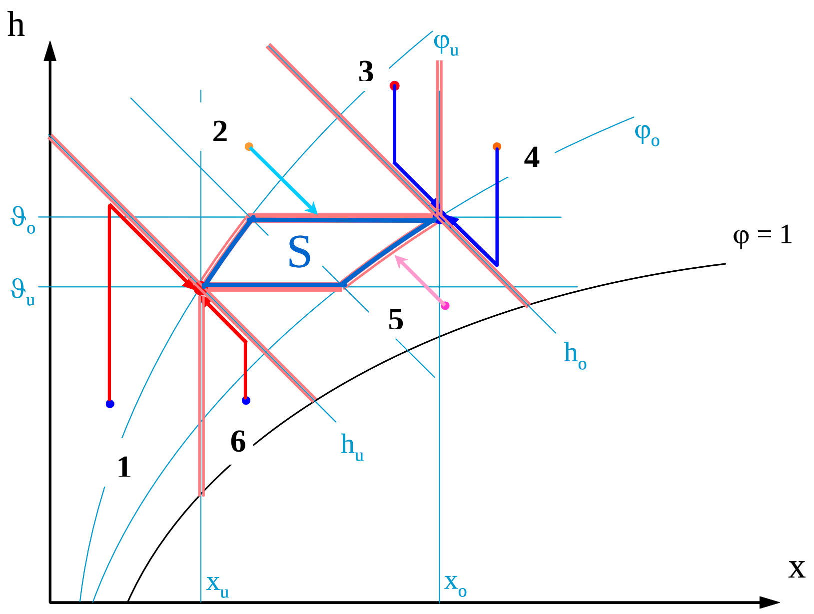

Figure 1: Ideal thermal process

For each outside air condition there is one way to this set point area with minimal energetic effort, neglecting whether this is technically possible nowadays. Within the ideal process, the following thermodynamic processes are possible:

- heating with constant absolute humidity
- cooling with constant absolute humidity
- isenthalpic humidification
- isenthalpic dehumidification.

Using one or a combination of these processes it is possible to get to the boundary of the set point area with a minimal enthalpy difference [1]. These ways are also shown in Figure 1. Each condition is described by:

- air temperature
- relative humidity
- enthalpy
- absolute humidity

where u indexes the lower end of the set point area and o the upper end.

The individual thermodynamic processes have to be applied under the following conditions:

S: no conditioning
1: heating plus isenthalpic humidification
2: isenthalpic humidification
3: cooling plus isenthalpic humidification
4: cooling plus isenthalpic dehumidification
5: isenthalpic dehumidification
6: heating plus isenthalpic dehumidification

For the above conditions, it is possible to calculate the enthalpy difference for each outside condition. The integral of these differences over one year gives the yearly accumulated enthalpy differences H_H and H_K for heating and cooling.

$$H_{H,\,ideal} = \int_a (h_u - h)dt \qquad \forall h < h_u\,, \tag{1}$$

$$H_{K,\,ideal} = \int_a (h - h_o)dt \qquad \forall h > h_o\,. \tag{2}$$

Multiplied with the appropriate primary energy factor f and the air mass flow $\dot{m}$ this gives the yearly primary energy demand

$$Q_{a,th,ideal} = \dot{m}(H_{H,ideal} \times f_H + H_{K,ideal} \times f_K) \tag{3}$$

and related to a floor space A_F

$$q_{a,th,ideal} = \frac{Q_{a,th,ideal}}{A_F}\,. \tag{4}$$

Ideal Air Transport

It is not possible to define a theoretical minimum analogue to the thermal one for the air transport. Therefore assumptions have to be taken. These are shown in Table 1. The assumption is, that we have a minimal pressure drop over the necessary components but no pressure drop in the duct system, resulting in an overall pressure drop Δp .

Table 1: Ideal Air Transport

components	**pressure drop in Pa**	
	supply air	**extract air**
filter	120	120
heating coil	10	-
cooling coil	20	-
humidifier	2	-
attenuator	12	10
heat recovery unit	80	80
sum air handling unit	**244**	**210**
air duct	0	0
sum system	**244**	**210**

Furthermore it is assumed, that the fan has an efficiency η of 1.
The resulting yearly primary energy demand then is

$$Q_{a,el,ideal} = \frac{f_{el}}{\rho}\int_a \left(\frac{\dot{m} \times \Delta p}{\eta}\right)dt \tag{5}$$

and related to the floor space A_F

$$q_{a,el,ideal} = \frac{Q_{a,el,ideal}}{A_F} \tag{6}$$

The total primary energy demand for the ideal process related to the floor space then is

$$q_{a,ideal} = q_{a,th,ideal} + q_{a,el,ideal}\,. \tag{7}$$

Real Thermal Processes

Each case for an air conditioning system can be dealt with by various different types of systems, for example for a system as shown in Figure 2 different heat recovery, different humidification, different control strategy etc. Figure 3 shows one example for such a real process. This real process takes into consideration the enthalpy-humidity-characteristics of the real apparatuses assumed, but each modelled for ideal behaviour. In the shown case it is assumed to have a recuperative heat recovery, an air washer and a direct humidification control.

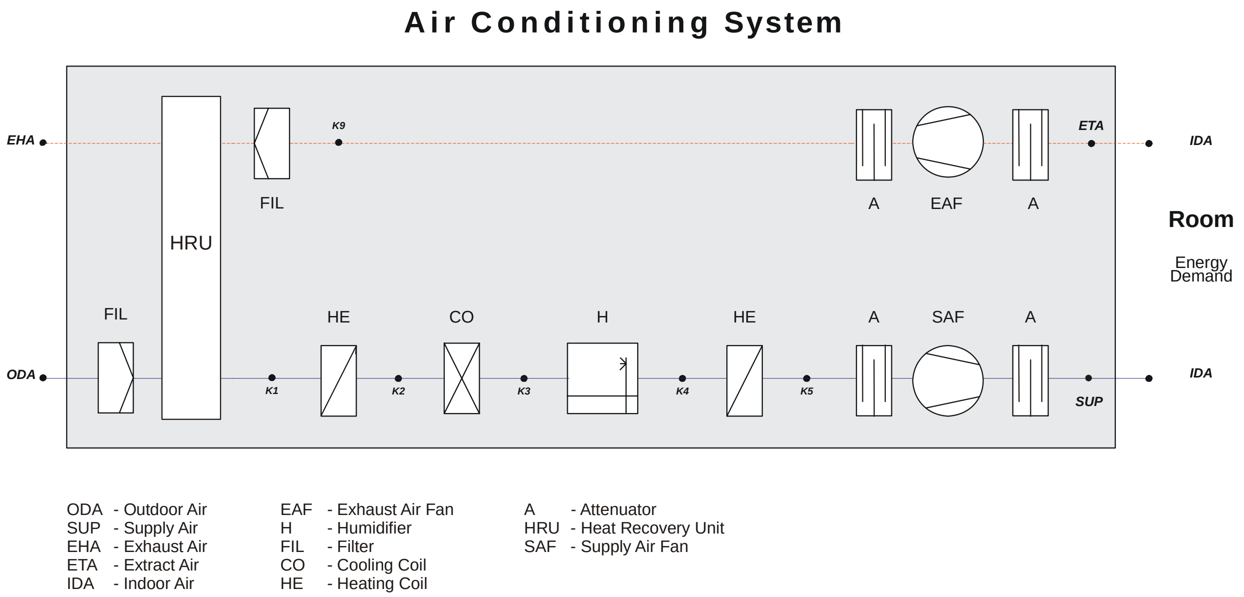

Figure 2: Real Air Conditioning System

Analogue to the calculation for the ideal process, the accumulated enthalpy differences, the absolute and the related primary energy demand can be calculated.

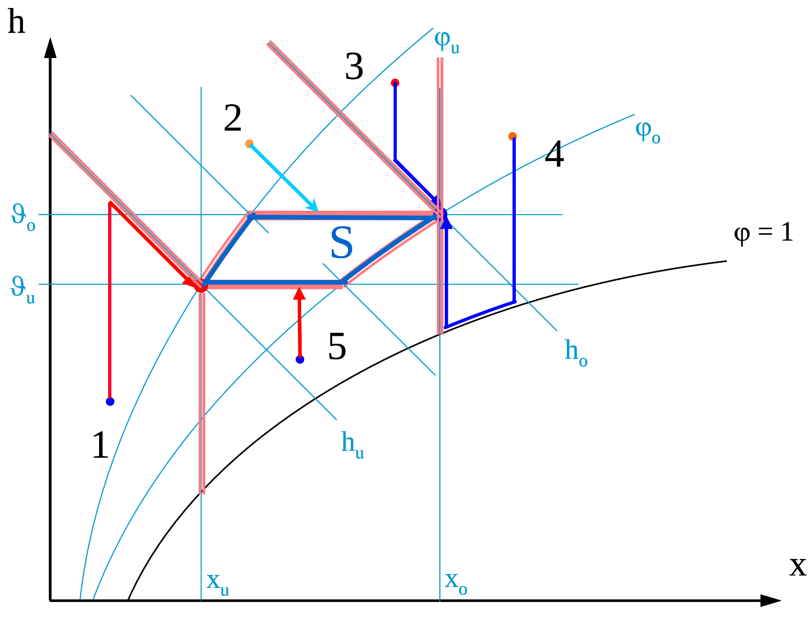

Figure 3: Real Thermal Process

Real Air Transport

Table 2 shows the range for real pressure drops on components and duct systems for typical German VAC-systems. These are the basis for the calculation of the primary energy demand for the air transport of the real systems.

Table 2: Real Air Transport

components	pressure drop in Pa					
	supply air			extract air		
	min.	common	max.	min.	common	max.
filter	120	149	200	120	149	200
heating coil	10	77	160	-	-	-
cooling coil	40	186	400	-	-	-
humidifier	15	134	300	-	-	-
attenuator	40	61	100	40	61	100
heat recovery unit	80	192	370	80	192	370
sum air handling unit	305	799	1530	240	402	670
air duct	250	643	1800	200	607	1500
sum system	555	1442	3330	440	1009	2170

The total energy demand for the real system then is the sum of the thermal plus the air transport demand.

Values for the limited Primary Energy Demand

The calculation of the energy demands for various real systems gives an idea for the range of energy demands for a specific use of a space. The quality coefficient g gives an energetic evaluation for each specific system related to the ideal process:

$$g = \frac{q_{a,\,real}}{q_{a,\,ideal}}. \quad (8)$$

A calculation of this quality coefficient then uncovers one problem. The ideal primary energy demand does not take into account any energy recovery. Because energy recovery is good practice in Germany, it is pragmatic to calculate a quality coefficient g^* on the basis of corrected ideal primary energy demand $q^*_{a,ideal}$ for which we decided to take into account a controlled recuperative heat recovery system, with a maximum recovery rate of 0,85.

$$g^* = \frac{q_{a,\,real}}{q^*_{a,\,ideal}}. \quad (9)$$

The evaluation of air conditioning systems follows the development of requirement, described in German Standard VDI 2067 [2], from the building with its envisaged use and the loads which arise from this and the climate, through benefit transfer (heat, cooling, conditioned air etc.), distribution and to the heating and cooling generation system. Since none of these subsystems – benefit transfer, distribution, generation – can be achieved perfectly, an additional energy effort occurs in each case.
This together with the typical reference energy requirement for the building and its use gives the total energy requirement applicable to the generating system. Figure 4 shows the requirement development in buildings.

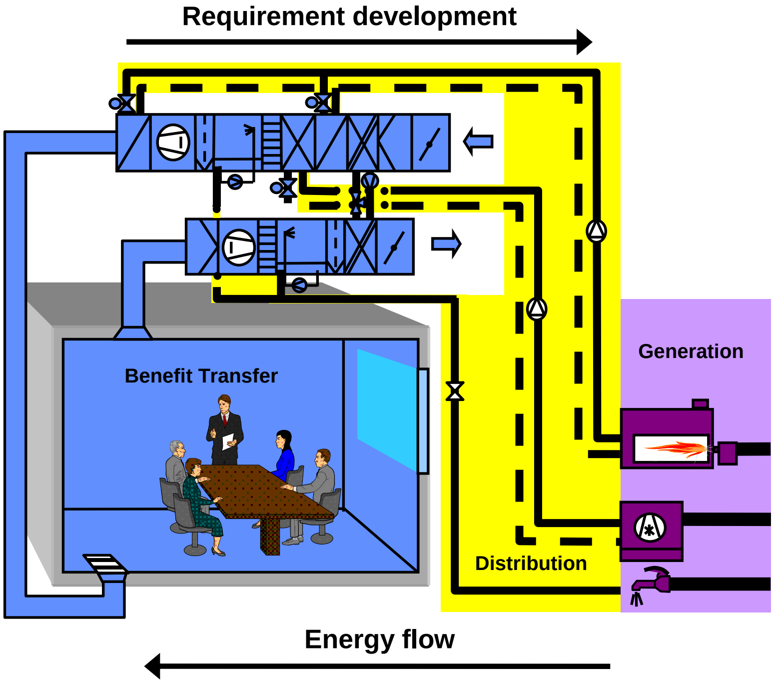

Figure 4: Requirement development (from VDI 2067)

The subsystems of benefit transfer are defined as followed:

- Air handling
- Air transport
- Air conditioning

To evaluate air conditioning systems in this research project only the subsystems air transport and air conditioning are discussed.
In a first step the Reference energy requirement of the room and its use has to be calculated. This value is the input for the requirement development and the energy demand of the benefit transfer than is calculated in a second step. The energy demand here is electrical energy, heating and cooling energy. It has to be evaluated by primary energy factors to compare the different types of energy.
The primary energy demand is a sum of electrical energy for the air transport and thermal energy for the air conditioning like described before.

EXAMPLE AND RESULTS

The paper shows the evaluation method on a typical example for the use of air conditioning systems. We choose here as an example a typical office room with the geometrical details shown in Figure 5. The typical use of the office space is described in Table 3 including all information about set point conditions, weather data, infiltration and the use.

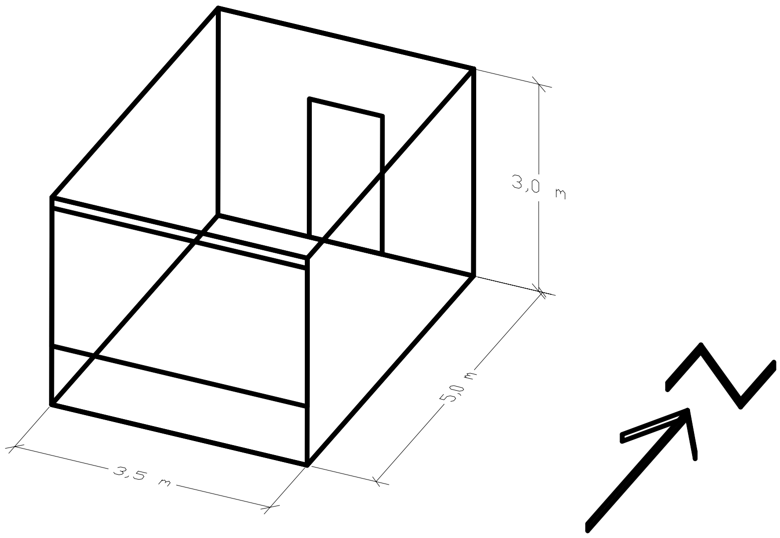

Figure 5: View of the Office Room

Table 3: Example case: Office Room

Utilisation	Standard office from VDI 2067-11, medium-heavy building			
Size	L x W x H = 5m x 3,5m x 3m, A_{window} = 7m²			
Orientation	South, clouding factor b = 0,5			
Weather Data:	Test Reference Year: TRY05			
Utilisation Time	Mo – Fr: 7:00 – 17:00	t_{util} = 2607 h		
Operation Time	Mo – Fr: 6:00 – 18:00	t_{op} = 3129 h		
Requirements	Summer Time:	$\vartheta_{Ra,o} \leq 26°C$	$\varphi_{Ra,o} \leq 65\%$	
	Winter Time:	$\vartheta_{Ra,u} \geq 22°C$	$\varphi_{Ra,u} \geq 45\%$	
Internal Gains		**Heat Gains**		**Substance Gains**
		Convection [W]	Radiation [W]	Humidity [g/h]
	1 Person	50	25	60
	1 Personal Computer	117	23	0
	Ligting (10 W/m²)	70	105	0
	2 Plants	0	0	30
Infiltration	$n = 0{,}2\ 1/h$ is equivalent to $\dot{m}_{inf} = 52{,}5\ m^3 \times 0{,}2\ 1/h = \underline{\underline{10{,}5\ m^3/h}}$			

Figure 6 shows calculated quality coefficients g^* for an office space, with design parameters as given in Table 3.

It can be concluded, that real systems for that purpose show quality coefficients in the range of **2,9…13,2.** The examples shown are pretty good types of systems, because they have up to date energy recovery and direct control. Thereby a decision can be taken to limit future systems to those showing a quality coefficient for example up to **9,5**.
If in the future any new type of system may be invented, it is only necessary to change the maximum quality coefficient for a specific use of a space.

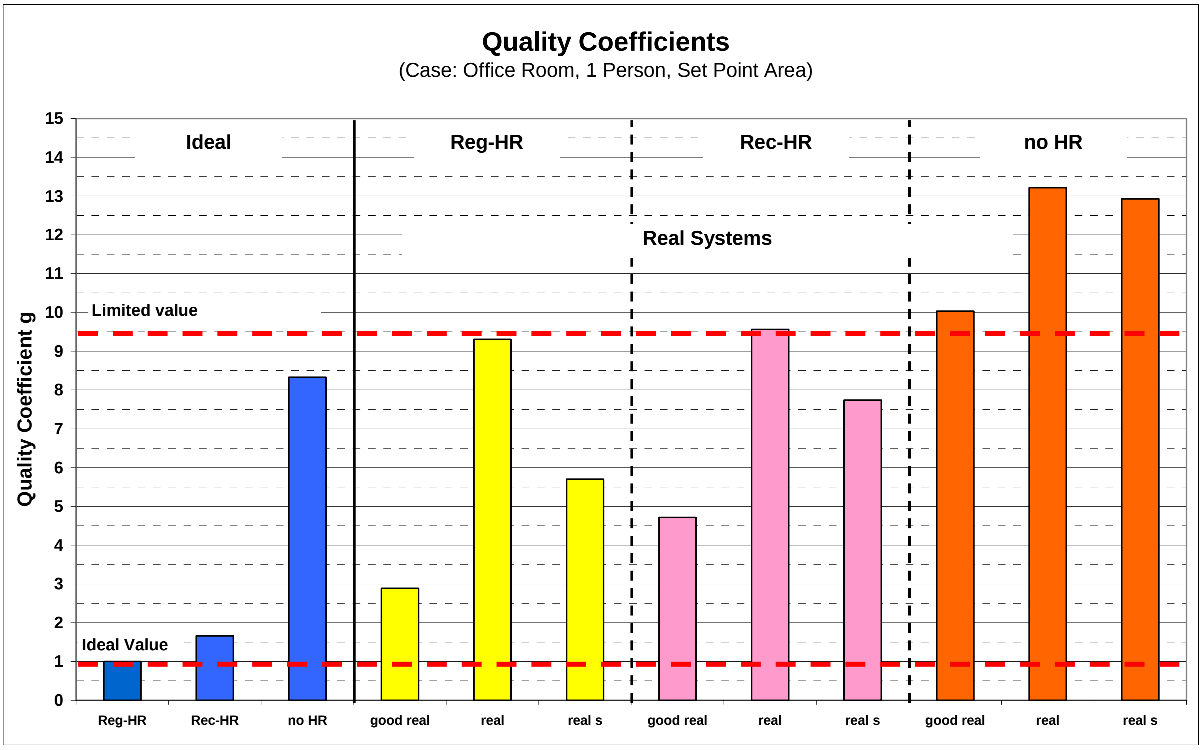

Figure 6: Quality Coefficients for an Office Room with 1 Person

DISCUSSION

It is shown, that by the developed method, values for limited energy demands for VAC-systems can be derived. The method does not take into account any designed specific type of VAC-system, i.e. it does not benchmark a designed system.
The derived limitations are based on the size and the use of the space.
This method should be considered for future application in general to implement a legally given energy demand for VAC-systems.

ACKNOWLEDGEMENT

The project was supported by Arbeitsgemeinschaft industrieller Forschung (AiF-Nr. 13975N). We thank for that.

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Environmental impact and discomfort relief of summer comfort appliances

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SUMMARY

Thermal comfort has always been an ambiguous concept. Any public policy has to understand and respect first the functionalities demanded by the user for equipment and space, and establish the link between those expectations and environmental objectives. Premature judgment would lead to problems when coming to public policies aiming to regulate HVAC. Using fans and moving air in a room is a way people may choose to improve their individual Summer comfort. This study characterized the benefits of comfort fans in term of comfort in summer and investigated if comfort fans could be a real alternative to compressor based systems in France.
By using experimental measurements of air speed along with building simulations with free temperature and humidity evolution, it was made possible to access inside climatic conditions when using a comfort fan. An optimized approach was then defined to describe comfort fan operating and compute the electric consumption. For the purpose of assessing the benefits of fans in terms of comfort and comparing it to air conditioners, a discomfort index based on Fanger indices was defined. It appeared that occupants using comfort fans would face about 10% of discomfort hours during occupation and that comfort fans can suppress about 60% of discomfort by consuming four times less energy than a split system.

INTRODUCTION

In Europe, some environmental agencies and associations consider compressor based air conditioning as useless energy consuming appliances and promote environmentally friendly cooling products without compressors. Considering that public policies have to understand and respect first the functionalities demanded by the user for equipment and space, and establish the link between those expectations and environmental objectives, this study aims to characterize the benefits in term of summer comfort of the air movement created by a comfort fan in French climates. The conditions in which ventilation can provide a reliable level of comfort and can be an alternative to compressor based cooling systems are investigated. After defining and presenting general concepts of comfort fans, a literature review is proposed regarding fans, comfort and energy savings.

Comfort fans

Using fans and moving air in a room is a way people can choose to improve their individual summer comfort. Air moving appliances aiming at cooling people like desk fans, pedestal fans or ceiling fans will be called “comfort fans” in this study. By generating air movement close to the body, comfort fans increase convection and evaporation and by this way individual summer comfort without lowering the room temperature. In fact, the inside temperature is even likely to increase since the comfort fan motor generates heat.

Comfort fans and thermal comfort

Numerous human thermal comfort studies have been carried out to analyze and quantify the positive and negative effects on thermal comfort of air flows and in particular of airflow due to comfort fans. As in ASHRAE standards [1], thermal consequences of air movement in cold ambiances that cause drafts and local discomfort have been often emphasized. This does not form a part of this study which aims at evincing the potential cooling benefits of air flows in Summer conditions, assuming the local discomfort is avoided.

First of all, potential cooling benefits of air flow in summer are presented in several standards. In the ASHRAE and ISO standards [1] [2], the summer thermal comfort zone is extended by increasing the air speed. For instance, the 2004 ASHRAE standard proposes, in summer conditions, to relate the comfort temperature to the air speed and the difference between the mean radiant temperature and ambient temperature [1]. However, air speed is reduced to very low level (0.8 m/s) in order to avoid disturbance like paper flying and drafts and the elevation of air speed must be adjustable and controlled by affected occupants. This kind of approach consisting in expressing the cooling sensation (temperature rise in °C) of uniform air flow according to the air speed can also be found in many research works like for example those of Szocolay [3].

All this is based on comfort indices developed by Fanger [4]: the PMV ("Predicted Mean Vote") and the PPD (Predicted Percentage of Dissatisfied). These indices mainly depend on six parameters, four ambiance ones (air temperature, mean radiant temperature, air velocity, relative humidity) and two concerning people (physical activity, clothing thermal resistance). In order to predict thermal sensations, Fanger assumed that, in comfort conditions, the human thermal balance must be null which leads to a first formula. It means that the individual exactly looses the heat produced by the metabolism. In addition, Fanger realized experiments in climate chambers to correlate metabolism with mean skin temperature and sweating. In more details, the PMV equation assumes that deviations from the human thermal balance vary with thermal comfort vote. A PMV equal to zero represents the optimum comfort when the thermal balance is null and this index can vary from –3 (cold) to 3 (hot). It is also possible to predict the reaction of individuals thanks to the PPD index that aims at calculating the expected number of thermally dissatisfied people in a group according to the PMV. PMV and PPD are used in several standards where comfort zones are often defined by limiting the PMV between –1 and 1 (ie less than 10% of unsatisfied people according to the PPD). In a further study [5], Fanger stated that the quantitative influence of air velocity is in good agreement with PMV and PPD equations.

Regarding comfort fans whose air flow is not uniform several studied have been carried out based on human experiments. Arens [6] showed that the cooling effect of personally controlled fans was approximately 1°C by 0.1 m/s increment of air speed. This study was carried out under specific factors, mainly air temperature from 24°C to 31°C, activity level of 1.2met (sitting in activity) and clothing equivalent to 0.5clo (summer light clothes). Furthermore, Arens [6] compared his results from experiments to results obtained with Fanger's indexes and found that they were close with a 5% difference. Effects of ceiling fans on the summer comfort zone have also been studied by Roshles [7] in experiments gathering eight subjects. They considered that air movement was pleasant up to 1 m/s at 29.5°C and the turbulence of the flow was a beneficial aspect. A study carried out by Konz [8] aimed at comparing fixed fans with oscillating fans. The exposed subjects preferred oscillating fans to fixed ones. A second set of experiments consisted in exposing the subjects to air movements at different angles to the front of the body. It appeared that angle was not a significant parameter in terms of thermal comfort. Another documented effect of air movement is the human response to the power spectrum of the turbulent airflow provided by a comfort fan. In fact, like other receptors contained in the skin, heat receptors are sensitive to some stimulation frequency ranges and have peak response at certain frequencies. Thus, Oelsen [9] determined that human thermal receptors had a significant peak in response around 0.5 Hz in cool ambiances and related velocities at a 0.5Hz gust frequency to equivalent uniform air speed for identical cooling sensations (discomfort in this case). For example, a mean air speed of 0.4m/s at a 0.5Hz frequency will provide the same cooling sensation as a uniform airflow of 0.6m/s. This kind of experiments has been also processed in hot ambiances where the human thermal response to the cooling sensation of air movement has a significant peak around 0.4 Hz [10].

Comfort fans and energy savings
In the United States, efforts are triggered off to quantify the potential energy savings of comfort fans (mainly ceiling fans) and try to improve their efficiency. Regarding energy savings, simulation studies have demonstrated that using ceiling fans can lead to energy savings or wasting in comparison with air conditioners according to set point chosen by end-users. For example, according to James [11], using ceiling fans combined with raising a home's temperature 1.1°C (2° F) will generate about a 14% net savings in annual cooling energy use (subtracting out the ceiling fan energy and accounting for internally released heat). This savings drops to 2.6% with a 0.56°C increase in set point and to a negative 3.7% savings with only a 0.28°C increase in set point. If the thermostat is not adjusted at all for fan use, cooling energy use may increase by 15%. Comfort fans can theoretically extend the natural ventilation season when the air conditioner is not used, allow for higher thermostat set points when the air conditioner is used and therefore should lead to energy savings. However, James [11] mentions that a survey of 400 households in Florida does not indicate cooling energy savings due to ceiling fans because of inappropriate thermostats settings. It was not investigated if this result could be explained by an insufficient level of comfort provided by ceiling fans. Regarding the improvement of fan efficiencies, ceiling fans have been already studied. For example, Schmidt [12] and Parker [13] have presented new designs of ceiling fans based on the improvement of motor and blade efficiencies.

METHODS

This study aims at assessing comfort benefits and electrical consumption of air moving devices and comparing them to other cooling appliances. From the bibliography previously presented, Fanger's indices (PPD and PMV) appear very relevant to assess the global thermal feeling (without drafts and asymmetries) of individuals and notably to take into account air movement benefits. These indices require a full knowledge of indoor conditions along with information regarding people like clothing or metabolic rates. As a result, the first step consists in simulating a house to define typical indoor conditions. Several sets of simulations have been carried out with different climates and different appliances. Available typical indoor conditions along with characteristics of appliances are treated with Matlab in order to provide the PPD index and consumption all over the summer period. An approach has been defined to calculate and determine the number of discomfort and operating hours for fans.

Simulations
From characteristics of the French building stock and from information of the year 2000 regulation defining energy performance requirements for buildings, a typical house has been defined. This house is supposed to be representative of the up-to-date residential building stock. Main geometrical and thermal characteristics are described in Table 1 and Table 2. In addition, assumed occupant usages are specified (occupation profiles, lighting control, installed equipments), an examples is reported in Table 3. Occupant usages are typical of a French family and are not representative of sedentary people usages (offices, rest-homes).

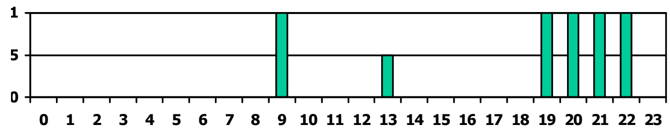
Table 1. Geometrical characteristics

Total area	136m²
Average Height	2.5 m
Windows-to-wall ratio	7.8% of the vertical area
Living room	34m²
Bedrooms	51m²
Kitchen	17m²
Lavatories	9m²
Circulations	25 m²

Table 2. Main thermal characteristics

Window solar factor	0.6
Light transmission rate of windows	0.6
U value for windows	2.45 W/m².K
U value for external walls	0.6 W/m².K
U value for the roof	0.25 W/m².K

Table 3. Example of occupant usages characteristics in the living room

Nominal occupation	Four occupants
Occupation profile From Monday to Friday	

This two storey residential house has been simulated fist with free temperature and humidity evolution and then with an air conditioner with the ConsoClim software. ConsoClim is an energy calculation software that was developed with the aim of reducing the large amount of inputs as much as possible and using input data which are easily available by designers such as data found in manufacturer's catalogs [14]. Consoclim uses hourly weather data to compute hourly energy consumptions and hourly loads. In this study, climatic files of 1999 are used for three cities: Trappes [48.5N-2.2E]; Rennes [48.1N-1.4W]; Pau [43.1N-0.2W].

Determination of devices characteristics

Airflows of a pedestal comfort fans has been measured by means of a hot-wire anemometer in a 30m² room. Since asymmetries are not taken into account in this study, only a global assessment is needed and air speed average has been computed at given distances from the fan. Some experimental values are presented in Table 4. The consumption has been set to 50 W which is the manufacturer value. It is assumed that when the fan operates, the load has to be increased by 50W due to heat production. Furthermore one comfort fan of this type can ventilate up to two occupants and therefore when a room is occupied by more than two people, additional fan is required. When assessing comfort, the potential speed interval that can be provided by this pedestal fan will be between 1.2 and 3.3 m/s. This means that occupants are located between 0.9m and 1.5m from the fan.

Table 4. Pedestal fan characteristics

Type	Distance to measurement	First speed [m/s]	Second speed [m/s]	Third speed [m/s]	Electrical input
Pedestal fan	0.9m in front	1.8	2.5	3.3	50W
	1.5m in front	1.1	1.6	2.3	50W

In France, air conditioners are often present in only one room of the house. In this study, it is presumed that a Split system is installed in the 30m² living room. This air conditioner has been sized and fills the summer load of this room (the set point is 25°C when the room is occupied). Then existing models were looked for and performances given by manufacturers were used for the simulations. As a result, characteristics are different according to the city. The EER varies from 2.63 to 3.27 that can be considered as energy efficient products in the European market. Main characteristics are given in Table 5.

Table 5. Split systems characteristics

Cities	EER	Nominal frigorific power [W]	Nominal electrical input [W]
Trappes	3.27	2750	840
Pau	2.63	3890	1480
Rennes	3.27	2750	840

Data processing:
This subsection aims at explaining how to assess the comfort obtained with comfort fans and air conditioners over French cooling season in a typical French house. The PPD index is used to assess the comfort feeling. For air conditioner, a temperature set point has been chosen and the PPD is then processed for an air speed of 0.4m/s, a clothing rate of 0.7clo and a metabolic rate of 70W/m² that corresponds to a seated subject in activity or a stand subject at rest.
When it comes to calculate the PPD for the pedestal fan, the problem becomes more complicated since the provided comfort is partial and occupants are considered to adjust their clothing and fan speed to maximize the comfort feeling. An approach to reproduce occupant behaviors has been investigated and programmed with Matlab.
First of all, the PPD calculations depend on the metabolic rate that is set at 70W/m² and on several factors that are provided by building simulations with free evolutions of temperature and humidity: pressure, relative humidity, ambient temperature and mean radiant temperature. Other remaining factors (air speed and clothing) are directly under occupant action and as a result occupant behaviors must be defined to access these values and compute the PPD index. From indoor conditions and air speed of 0.1m/s (no air moving device), the PPD is minimized by selecting the most appropriate clothes that are associated to thermal resistances. It means that occupants are assumed to look for the clothes that generate the best thermal feeling. In a second time, if the reached PPD is higher than 10%, occupants face a discomfort feeling and will try to improve it by switching on the pedestal fan. As a result, if the PPD is higher than 10%, a second minimization is computed by selecting the best fan speed (from 1.1 to 3.3m/s). This 10% value has been chosen to distinguish between comfort hours and discomfort hours because this value has been selected in standards ([1],[2]) to define the thermal comfort zone. To conclude, the obtained PPD is not necessary lower than 10% but it corresponds to the best combining between clothing and ventilating. This approach is an optimized approach since the first optimization relies on clothing and that the comfort fan is not used if a cloth modification is enough to reach a PPD lower than 10% (Figure 1).

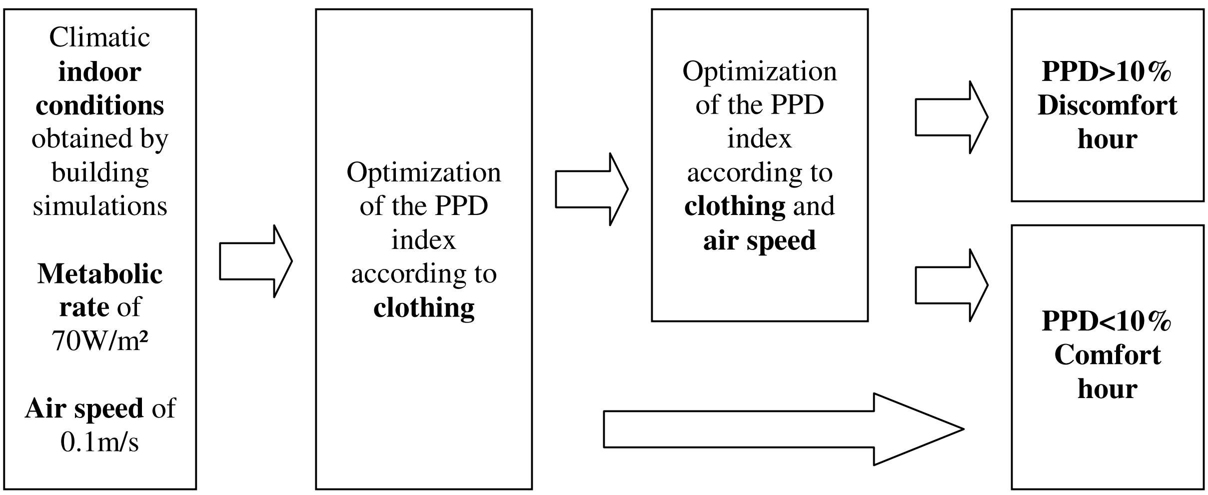
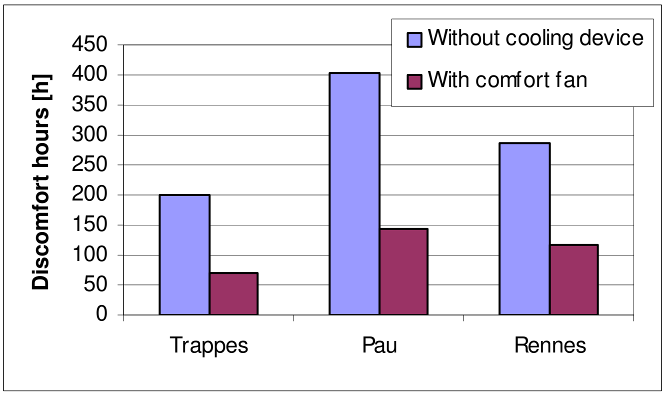


Figure 1. Approach to calculate the PPD index and determine the number of discomfort and operating hours for comfort fans

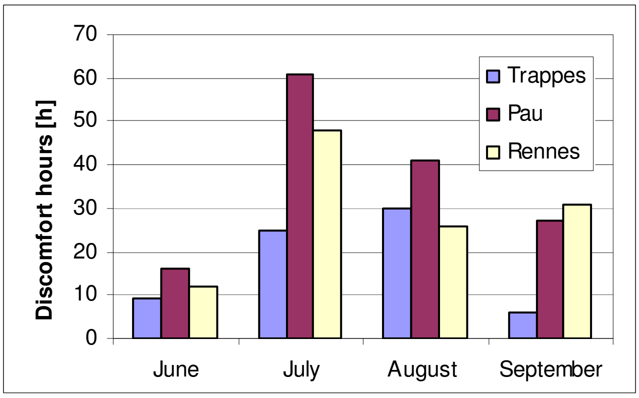
RESULTS

The typical French house previously defined has been simulated for three cities in two configurations: free evolutions of humidity and temperature and air conditioning. Results are given in this section regarding comfort assessment and consumption calculations.

Comfort assessment:
As for international standards, it has been considered that the thermal comfort zone corresponds to a PPD lower than 10% ([1],[2]). Thus, a discomfort hour is defined as an hour during which the calculated PPD is higher than 10%. The number of discomfort hours has been calculated in the three cities (Trappes, Pau and Rennes) for the two types of systems (Split system and pedestal fan) over the summer period from May to September (Figure 2a). When an air conditioner is used, there is no discomfort hour. With a comfort fan, occupants would have to face from 70 discomfort hours in Trappes to 145 in Pau, which correspond respectively to 6.4% and 13.3% of the occupation time. It is also noticed that by using a comfort fan, occupants can reduce the number of discomfort hours by more than two compared with a room without cooling device. With a comfort fan, discomfort hours depend on outdoor conditions and are more present on July and August (Figure2b). Note that French people usually leave for holidays during these two months.



a)



b)

Figure 2. a) Number of discomfort hours over the summer period for every studied city,
b) Number of discomfort hours every month for every studied city by using a fan

Consumption calculations:
The electrical consumption that is often considered as the main environmental impact has been calculated over the summer period for both the pedestal fan and the split system in the three studied cities.

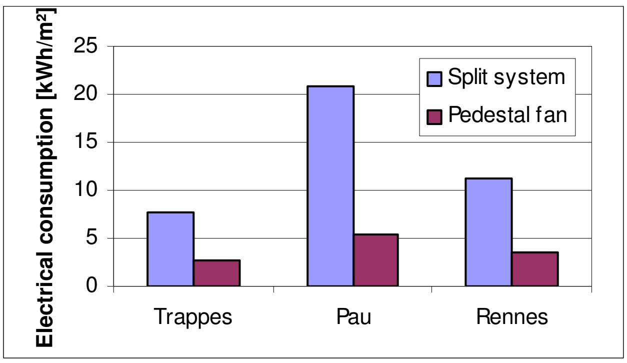


Figure 3. Electric consumptions in the studied cities

The electrical consumption varies from 7.7kW/m² in Trappes to 20.8kW/m² in Pau for air conditioners and from 2.7 kW/m² to 5.5 kW/m² for the studied pedestal fan. In Trappes, a pedestal fan will consume 2.8 times less then a Split system, 3.8 times less in Pau (Figure 3) but both appliances do not provide the same level of comfort.

The benefits of fans in terms of cooling are very dependent on climatic conditions. It turns out that in some cases comfort fans can be considered as good alternatives to air conditioners if occupants are wiling to accept some discomfort hours (less than 15% in the worst cases).

DISCUSSION

The French house previously defined has been simulated for three climates in two ways: free evolutions of humidity and temperature and air conditioning. In order to compare comfort fans to other cooling appliances, the relationship between improvement of summer comfort and consequences in terms of environmental impact is analyzed and a comfort index is proposed for this purpose. Finally, the cost of comfort is used as a second step of unification.

Improvement of summer comfort and consequences in terms of environmental impact

In order to state if comfort fans can be a relevant alternative to compressor based systems for cooling, an analyze of gains in terms of summer comfort and consequences in terms of environmental impact must be done. For this purpose, two different indices must be selected to assess both comfort and environmental impacts. Since the electrical consumption often appears as the main environmental impact, this will be kept as environmental index. In opposition, a unified comfort index is required to compare different cooling solutions. RAC provide total cooling by changing humidity and temperature whereas comfort fans only provide partial cooling by increasing the air speed. An index like the “discomfort degree hours” [15] cannot field this problem since with it a comfort fan is considered as not improving comfort. As a first step of unification, it is proposed to define discomfort as the integral of PPD in respect to a reference value over the summer period (1). In this way, the number of discomfort hours is taken into account along with the amplitude of the discomfort. As for comfort standards, the reference value is fixed to 10%.

$$\mathrm{I} = \sum \int \left[PPD_{occ}(t) - 0.1\right]^{+} .dt \,, \qquad (1)$$

where PPD_{occ} is the PPD index in occupation.

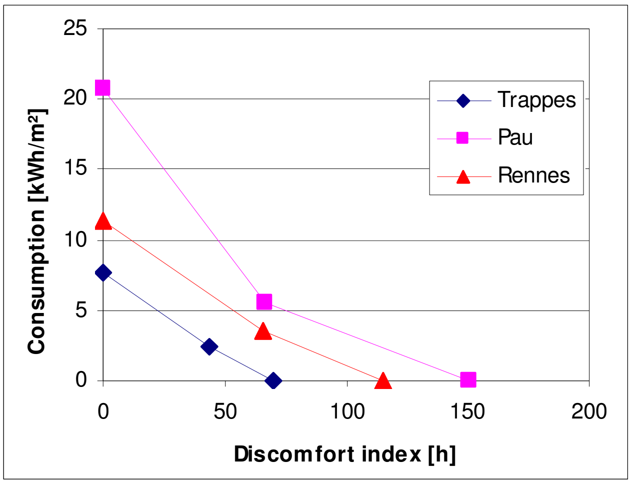


Figure 4. Electric consumption according to the discomfort index in the three studied cities

This index is calculated for three configurations (split system, pedestal fan, no cooling appliance) in the three cities (Figure 4). Results are similar for the three cities, between 40% and 65% of discomfort is suppressed by using a comfort fan. An air conditioner will suppress all of it consuming around 3 or 4 times more energy than a fan.

Comfort cost

As a second step of unification, the cost of comfort has been studied. The concept, presented by Barroso-Krause [15], is based on the acknowledgement that there is a given level of discomfort below which the end-user would try to install or change a cooling system in accordance with what he can afford. As a result, it is relevant to focus on what the costumer has to pay to improve comfort. A raw evaluation consists in only taking into account the purchase cost: 50€ for a fan, 1000€ for a Split system. To assess the gains in comfort, the discomfort index previously defined is used. These gains are defined by device discomfort indices normalized by the index calculated without cooling device. Results are given for the three cities and three configurations (no cooling appliance, comfort fan, air conditioner) in Table 6. Results are similar for the three cities, between 40% and 65% of discomfort is suppressed by spending 50 euros but one have to spend 1000 euros to suppress all of it.

Table 6. Costs according to comfort gains

Cooling devices	None	Comfort fan			Air conditioner
Cities	All	Trappes	Pau	Rennes	All
Gain in comfort [%]	0	37	65	43	100
Cost [€]	0	50	50	50	1000

Methods to compute environmental impact and comfort of fans have been created in such a way that at the end fan performances can be compared to a split system. It appears that occupants using comfort fans would face about 10% of discomfort hours during occupation and that comfort fans can suppress about 60% of discomfort by consuming four times less energy than a split system. In some cases, fans can be considered as good alternatives to air conditioners if occupants are wiling to accept some discomfort hours. Additional simulations should be performed to analyse other cooling appliances (evaporative coolers…) and to study other sectors (office, rest-homes). The discomfort cost should also be computed in a more comprehensive way (electricity bill).

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Measurement, visualization and simulation of air velocity at local air conditioning system

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SUMMARY

For the prototype of the local air conditioning of the working places named PERMICS (Personal Microclimate System) the air velocity and turbulence intensity at horizontal air inlets were analyzed. The velocity field was visualized with smoke. Based on findings from visualization the measurements were made with new technology of the hot-wire anemometer named HW3D-ED. Experimental analysis has been supplemented by a numerical simulation of the velocity field on the air terminal device and in the breathing zone using a commercial CFD package, software PHOENICS. Simulation results using three two-equation turbulence models: standard k-ε, RNG k-ε and Chen k-ε model were compared with measurements results. Findings show complex inlet profile at inlets and small difference between measurements and simulations and also between selected turbulence models. Conclusions from visualization, measurements and simulations are need for further simulations with equivalent diameter of the inlet with uniform velocity and turbulence intensity profile to achieve comparable results of the velocity field from simulations with measurement results.

Keywords**:** personalized ventilation, air distribution, horizontal air jet, turbulence models

INTRODUCTION

What is thermal comfort? Thermal comfort is that condition of mind which express satisfaction with the thermal environment [1]. While this definition has its merits, it cannot be easily translated into physical parameters because thermal comfort depends on many parameters (ambient temperature, clothing, body constitution, health, age of individual, activity, lighting, noise in environment, odours, individual psychological conditions, etc.).

The energy crisis of the 1970's was responsible for an approach which emphasized energy savings. The results of such an approach were decreased volume flow rates of fresh air into building and thus worse indoor air quality than before the crisis. At the same time, more and more interior pollutants were making their appearance. New building materials, furniture, laser printers, photocopiers, detergents, as well as people themselves and their activities (odours, respiration, perfumes…) represent numerous sources of pollutants that are potential health risks. Taken together, this can lead to Sick Building Syndrome (SBS) [2].

Studies carried out in Slovenia [3-5] show that people are dissatisfied with the indoor environment, especially in mechanically ventilated buildings [6]. Another finding shows the significance of psychological factors in the evaluation of indoor environment parameters [7-10]. The conclusions gleaned from these studies indicate the need for new, more qualitative systems of ventilation or air-conditioning of the indoor environment. Such systems should have a local inlet of air with the possibility of local regulation of key parameters of thermal environment. We could achieve satisfactory thermal comfort parameters for any subject with such a system and have better air quality at the same time.

Studies show that displacement ventilation is very useful in achieving good air quality, especially when we have heated sources of contaminants [11]. At the same time it could be problematic from the point of view of draught and a large vertical temperature gradient [12, 13]. Analysis has also shown that up to 50% of occupants may still be dissatisfied with the thermal environment and air quality in buildings using displacement ventilation [14]. Nevertheless, local ventilation or local air-conditioning has numerous advantages and can assure good air quality and thermal comfort in the majority of case [15-21].

The concept and development of a system for local air-conditioning of workplaces PERMICS is based on an analysis of other local air-conditioning systems and on measurements from local air cleaning devices [22, 23]. PERMICS is an individual unit for a single work area which makes air distribution possible at the work area and is designed for use in offices with several employees in one room. The PERMICS concept is analogous to creating smaller, separate "rooms" within each work area with individually regulated microclimate conditions. The system is not designed for heating or cooling large areas due to the draught risk from larger amounts of air or lower air temperatures [23]. A sensor could also be mounted for sensing the presence of a person at the desk and it could then regulate the table lamp and close the air inlets of PERMICS. PERMICS is designed to be connected to the central air-conditioning system. There are several methods of connection. A duct with fresh air in a double floor connected to PERMICS by a flexible duct is one possibility. Elements for air distribution are located within the microclimate zone. Figure 1 shows the scheme of PERMICS. An office desk with dimensions 800 x 1600 mm with height 765 mm was supplemented with a partition wall at a height of 800 mm above desk level. The PERMICS duct connecting the system to the central air duct is located on the back side of the partition wall. The duct is then separated into two parts which both lead to the elements for air distribution. The system is hydraulically balanced with a built in trap.

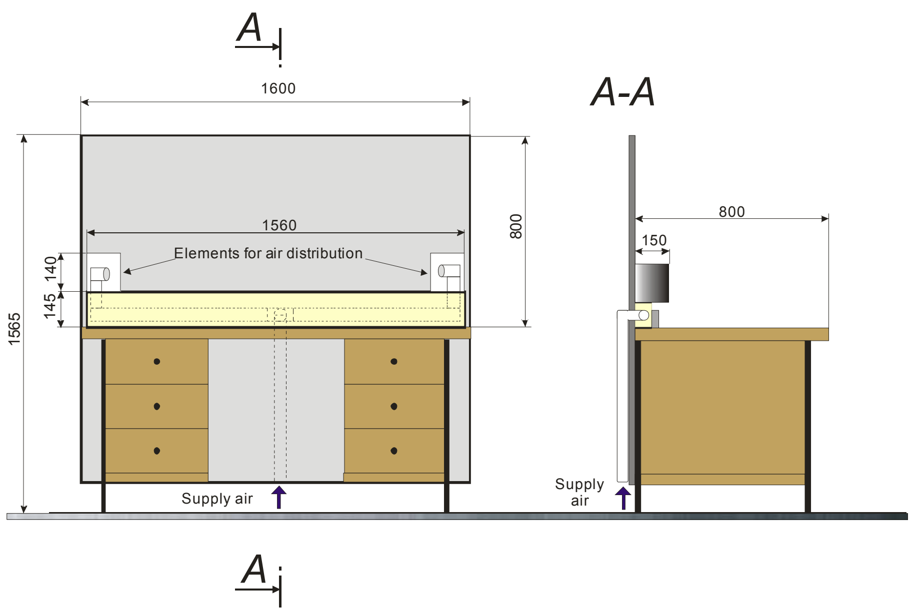


Figure 1: Scheme of PERMICS.

METHODS AND RESULTS

The velocity field at the inlets of PERMICS was visualized with smoke at isothermal inflow of 9.5 l/s of fresh air. Inlet angle of the air distribution element of PERMICS was 45° (Figure 3, right). Line laser was used as source of light. Picture was taken with analogue camera. We found complex inlet velocity profile at the air inlets. Figure 2 shows inlet velocity profile at 6 different times.

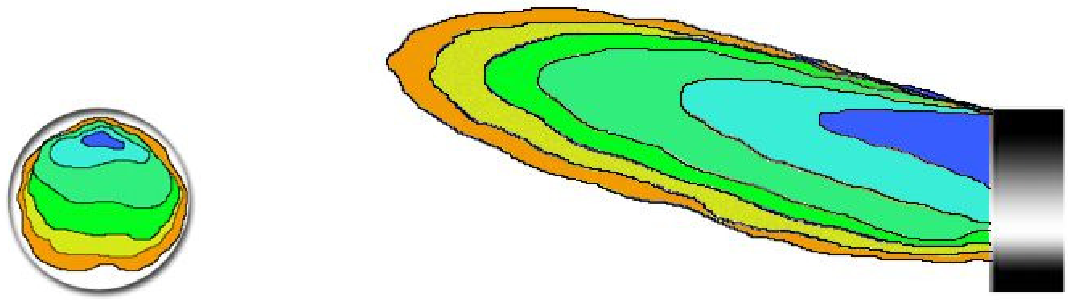
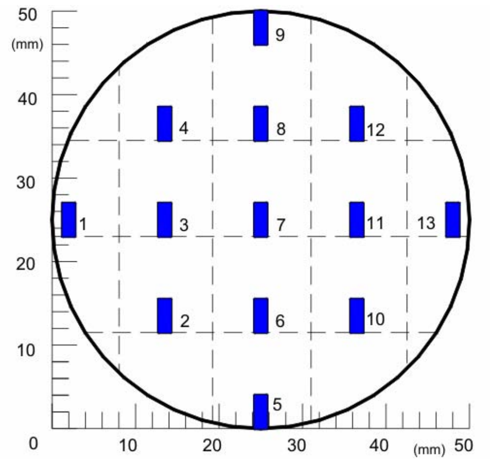


Figure 2: Jet in transverse and in longitudinal section from the visualization.

Based on the findings from the visualization the measurements were made with hot-wire anemometer HW3D-ED. The anemometer is result of our research and development. Figure 3 shows 13 defined measurement points at the inlet of PERMICS. Sensor of the anemometer was positioned at least 0.5 mm accurately. This error is not essential for the visualization of the velocity field with Kriging method [24].



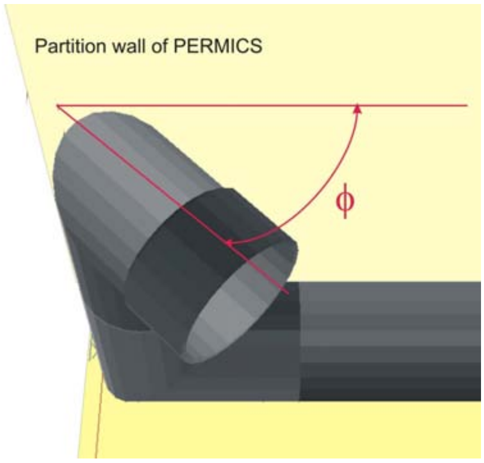


Figure 3: Measurement points at inlet (left) and inlet angle ϕ (right).

Figure 4 (above left) shows measurement results of the air velocity distribution at cross-section of the left air inlet. The results are in accordance with findings from the visualization. We got large gradients of the velocity at the cross-section of the air inlets because of the construction of supply ducts of the PERMICS.

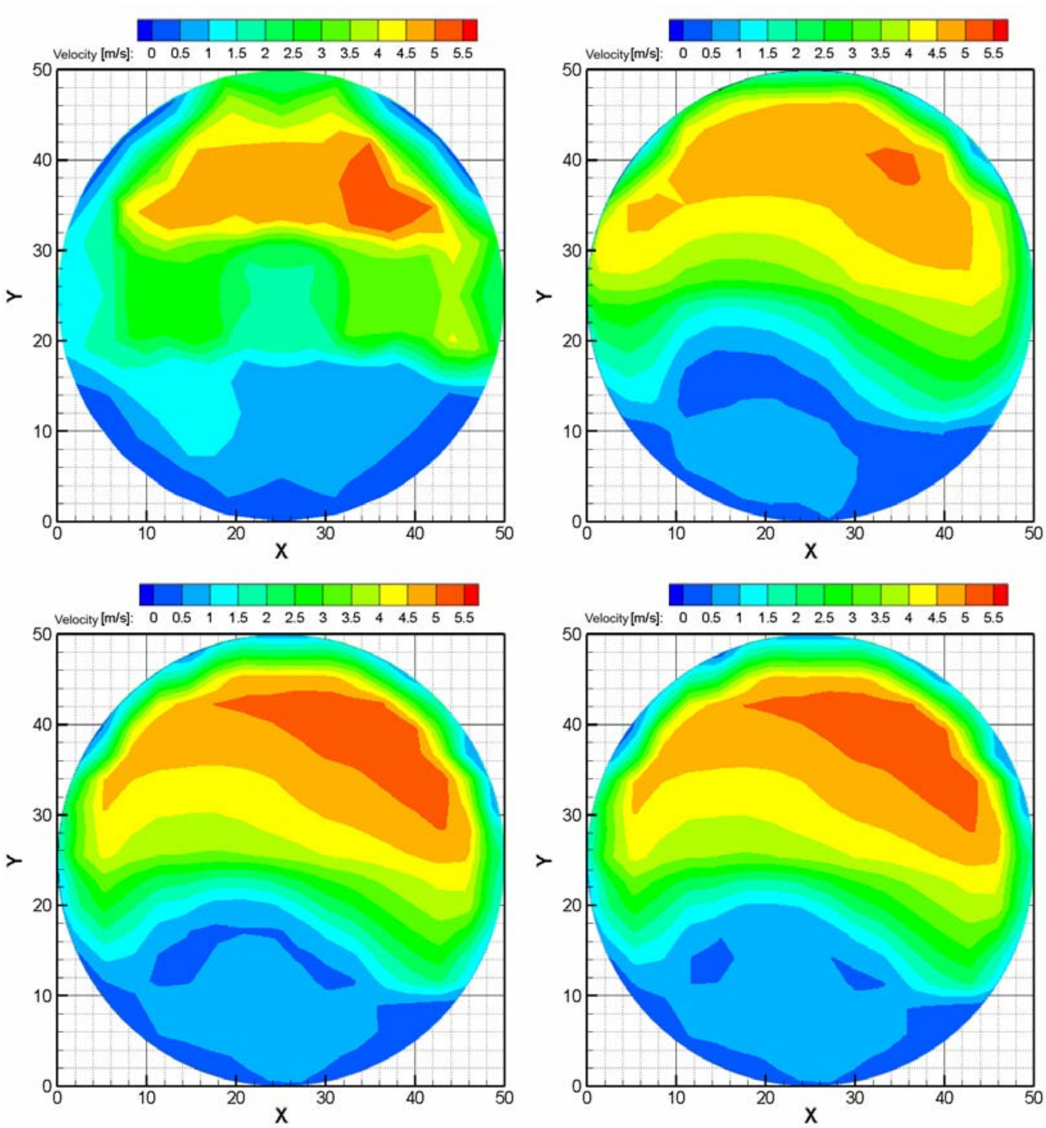


Figure 4: Air velocity distribution at cross-section of the left air distribution element of PERMICS. Above left measurement results, above right results of simulation with standard k-ε model, below left results of simulation with Chen-Kim k-ε model, below right results of simulation with RNG k-ε model.

CFD model was made for the above described setting of the inlets. Adiabatic steady state model of the left air inlet was made with mesh of 76x92x85 cells (Figure 5) and real geometry of the air distribution element and supply duct. Inlet of 4.75 l/s of air with 295K and with 5% turbulence intensity were described at the inlet in the domain (at the straight part of the pipe). Standard k-ε, RNG k-ε and Chen k-ε turbulence model were used for turbulence modeling because of the NOKFOS principle (NObody Knows FOr Sure) [25]. Figure 4 shows measurement and simulation results of the velocity distribution at the left air inlet of PERMICS. There is very accurate prediction of the velocity distribution from the simulation irrespective of chosen turbulence model. Turbulence intensity from simulations was also compared with the measurement results. There are no big differences in prediction between selected models, but all models have higher prediction of turbulence intensity than was measured out. This verification shows that numerical model could be used for optimization of the inlet angle of the air distribution element.

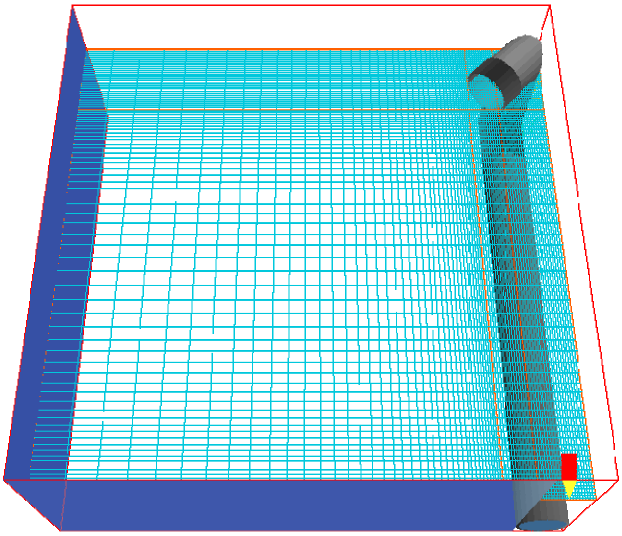


Figure 5: CFD model of the left air distribution element.

CONCLUSIONS

Experimental validation of the simulations shows that we have accurate prediction of the velocity field from the simulations so the CFD model could be used for optimization of the inlet angle of the air distribution element. But we also found very complex inlet profile of the velocity and turbulence intensity at the inlets of the PERMICS, which could not be easily described as the boundary condition for the simulations of the whole microclimate zone of the PERMICS. So the further simulations of PERMICS must be made with equivalent diameter of the inlet with uniform velocity and turbulence intensity profile to achieve comparable results of the velocity field from simulations with measurement results in microclimate zone of user of PERMICS.

ACKNOWLEDGEMENTS

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Analysis of effectiveness of personalized ventilation

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SUMMARY

Analysis of operation of a prototype system for local air-conditioning of working places, named PERMICS-LOS1 (Personal Microclimate System) was made based on measurements of air velocity field, using a hot-wire anemometer and of tracer gas concentrations with decay method. The emphasis of the experimental analysis was on design and effectiveness of the air terminal device in terms of air distribution using also visualization technique. Analysis of system operation was made for different operating regimes of personalized ventilation, where changes of air quantity (volume flow) and direction of supplied air in the breathing zone were applied. Experimental analysis has been supplemented by a numerical simulation of the velocity field on the air terminal device and in the breathing zone using a commercial computational fluid dynamics (CFD) software PHOENICS. Performed analysis of operation of personal ventilation system has verified our expectations in terms of the importance of proper air distribution device design on achieving greater effectiveness of the system.

Keywords: personalized ventilation, air distribution; CFD simulation; vertical air inlet; ventilation effectiveness

INTRODUCTION

In the last years indoor air quality and the importance of providing a healthy and comfortable environment for the occupants of closed spaces has been drawing more and more attention. People have become aware of the term *sick building syndrome* (SBS) [1] and building-related health complaints which represent significant health problems, such as headaches, allergies and fatigue. Room air, which is in majority of ventilated buildings normally supplied by a mixing or displacement ventilation systems, is at the time when it is inhaled by the occupants already polluted by different indoor air contaminants. These are generated by numerous pollutant sources such as building materials and equipment, electrical appliances, detergents, as well as people and their activities themselves. That is the reason for bad indoor air quality which has an economic impact, caused by decreased productivity of employees, absence of work, as well as consequently higher health care costs [2-4].

A new approach for the future air-conditioned environments based on philosophy of excellence was proposed by Fanger [5]. He suggested narrowing the air-conditioned space from the whole room space to a local space of the occupants' breathing zone and proposed a novel personalized air (PA) system which would serve gently a small amount of cool and clean air close or directly into the breathing zone of each individual without causing a draught.

Results of different studies carried out in Slovenia [6-13] were presented in the last research of effectiveness of personal ventilation system [14]. The conclusions gleaned from these studies indicated the need for new, more qualitative systems for ventilation or air-conditioning of indoor environment. In previous research [14] such system with a local inlet of air with a possibility of local regulation of key parameters of thermal environment was analysed. Measurements and simulations of system operation showed that personalized ventilation system is very effective in terms of ventilation efficiency. Furthermore a new index of relative decrease of tracer gas concentration in the first minute of system operation *dC(1)* was defined. The connection of this index with the velocity field and with the ventilation effectiveness parameters was discovered that could be used for direct measurement of ventilation efficiency in a shorter time.

In present paper we report findings of measurements, visualizations and numerical simulations of operation of a new personalized ventilation system for working places, named PERMICS-LOS1. Development of this prototype system was mainly based on a local air-conditioning system PERMICS (Personal Microclimate System), developed in previous research [14,15] as on analysis of other personalized air systems [16-20] and on measurements from local air cleaning devices [21,22]. A novel system is designed as individual unit for distribution of personalized air directly into the breathing zone of the occupant at a single working place and is appropriate for offices with several employees. Therefore it is not designed for heating or cooling of large areas due to possibility of draught from larger amounts of air at lower temperatures. System is designed for use in conjunction with a total-volume ventilation and air-conditioning and makes possible to customize the environment to individuals' preferences by allowing the adjustments of direction, flow rate and temperature of the personalized air. At the stage of development of the system predictions were made to achieve better ventilation efficiencies in the breathing zone with smaller amounts of fresh air than with previous PERMICS system.

METHODS

Air terminal device of PERMICS-LOS1 consists of a desk grille with dimensions of 225mm × 75mm with two freely adjustable grilles, mounted on a horizontal surface of an office desk with dimensions of 800mm × 1600mm and height of 765mm respectively as shown on schemes in Figure 1 and Figure 2. Such position of a desk grille is suitable for working places equipped with personal computer when usual arm position pattern on a keyboard does not interrupt airflow, because a desk grille is located between the arms of sedentary person. Adjustable grilles enable adjustments of direction of personalized air flow which is basically orientated vertical directly to the breathing zone of the occupant or towards the occupant's body. The appropriate inclination of adjustable grilles in respect to vertical position is from 45° to 0° (vertical position) while at greater inclinations from vertical position the air jet is strongly affected by the side pods of desk grille housing.

Air terminal device (desk grille) can be connected with a supply duct for fresh air of an air-conditioning system in several different ways, for example by a flexible duct from the wall or floor at the back side of an office desk as one possibility. System can also be equipped with a sensor for sensing the presence of a person at the desk which turns system on or off.

For the purpose of the analysis a seated manikin behind an office desk was used to simulate a sitting occupant at work. A manikin was during the testing procedure in an upright position with a nose height of 1.10m. In the first stage of experimental analysis the appropriate amount of supply fresh air at isothermal conditions and the proper inclination of adjustable grilles were tested using a visualization technique with a smoke. For the tested specification it was found that the suitable amount of inlet air was between 1.50 and 3.50 l/s and the inclination angle of adjustable grilles between 15° and 35° from vertical position. Inside these operating conditions the system is expected to be working effectively.

Figure 1: Scheme of PERMICS-LOS1.

Based on findings of visualization, measurements of velocity field and turbulence intensity on the desk grille were made for several different operating conditions in the range of proposed operating conditions using a hot-wire anemometer HW3D-ED with the uncertainty ±0.882% of measured value [15]. Hot-wire probe of the anemometer was positioned on inlet surface at different points along the width of the grille between two adjustable grilles as presented in Figure 3 with accuracy of positioning of at least 0.5mm. Furthermore additional velocity and turbulence intensity measurements were performed at the manikin's nose and in the breathing zone above the desk grille.

Figure 2: Horizontal desk grille geometry

As a part of the analysis measurements of ventilation efficiency of PERMICS-LOS1 using a tracer gas technique in the test chamber with a tracer gas carbon dioxide (CO_2) were made.

Used measuring technique was decay, or tracer step-down method [23-25]. CO_2 concentration was measured using a system for CO_2 measurements with infrared absorption sensor at four locations, namely in fresh supply air, at the air terminal device, at the manikin's nose and at the outlet from the test chamber.

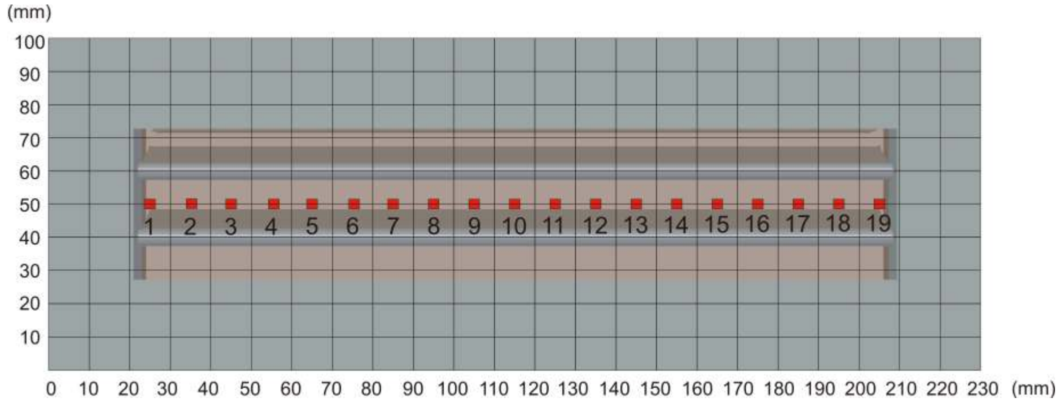

Figure 3: Measurement points of hot-wire probe on a desk grille.

For a described personalized ventilation system two different numerical simulations were performed using a commercial CFD package CHAM PHOENICS. The first one was a 2D simulation of velocity and turbulence intensity field on the horizontal desk grille with mesh of 327×131 cells. This simulation served as a basis for a 3D simulation of velocity field in the breathing zone for a PERMICS-LOS1 positioned in a room with a seating manikin behind the desk with mesh of 71×54×55 cells. For both simulations an adiabatic steady state model was made to satisfy isothermal conditions of air jet similar to those, assured during measurements.

RESULTS

Measured results of the air velocity profile on a surface of a desk grille for maximal proposed amount of 3.50 l/s of air are presented in Figure 4. The velocity gradient at the boundary region is well seen as are slight fluctuations of velocity and consequently of turbulence intensity. These variations in velocity are caused by flow direction lamellas, positioned under both adjustable grilles. The results are in accordance with expectations as the low turbulence intensity at the inlet is assured by an additional element for a laminar flow.

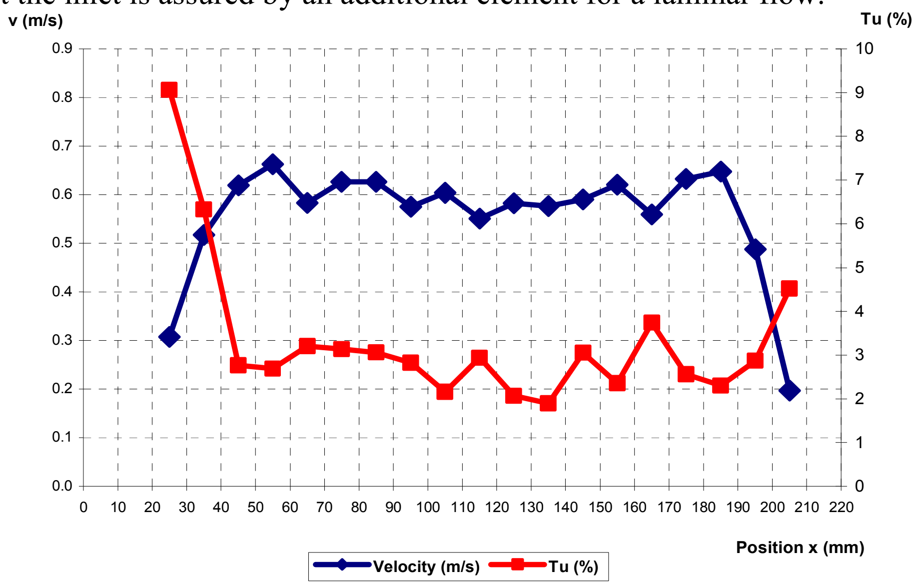

Figure 4: Velocity and turbulence intensity profile in measurement points on desk grille.

In 2D numerical simulation both standard k-ε and Chen-Kim k-ε models were used for turbulence modelling. There were no important differences discovered between two models at the 2D level that would have significant influence on a velocity and turbulence intensity field result. On the other hand for the 3D simulation of velocity field in the breathing zone significant differences were discovered between tested turbulence models. Current findings, similar to previous results of CFD simulations [15,26], confirmed that Chen-Kim k-ε model is the most suitable for this case.

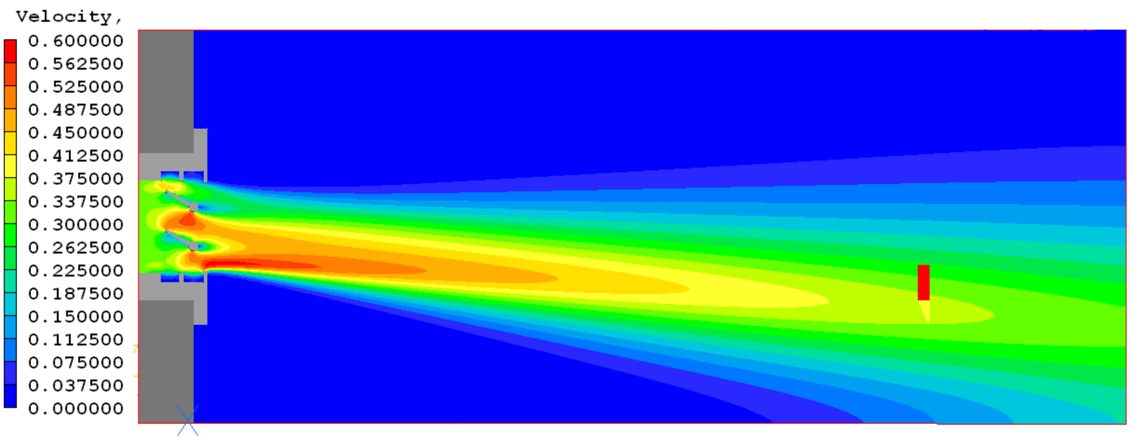

Figure 5: Results of air velocity field simulation on a desk grille.

Figure 5 represents 2D simulation results for an air velocity profile on a desk grille with the usage of Chen-Kim k-ε model for an inlet described with 3.50 l/s of air with 295K and with 5% turbulence intensity with the inclination angle of two adjustable grilles set to 30°. Velocity field at the cross section over simulated air jet at the distance of 0.4m from air inlet on a desk grille is shown in Figure 6.

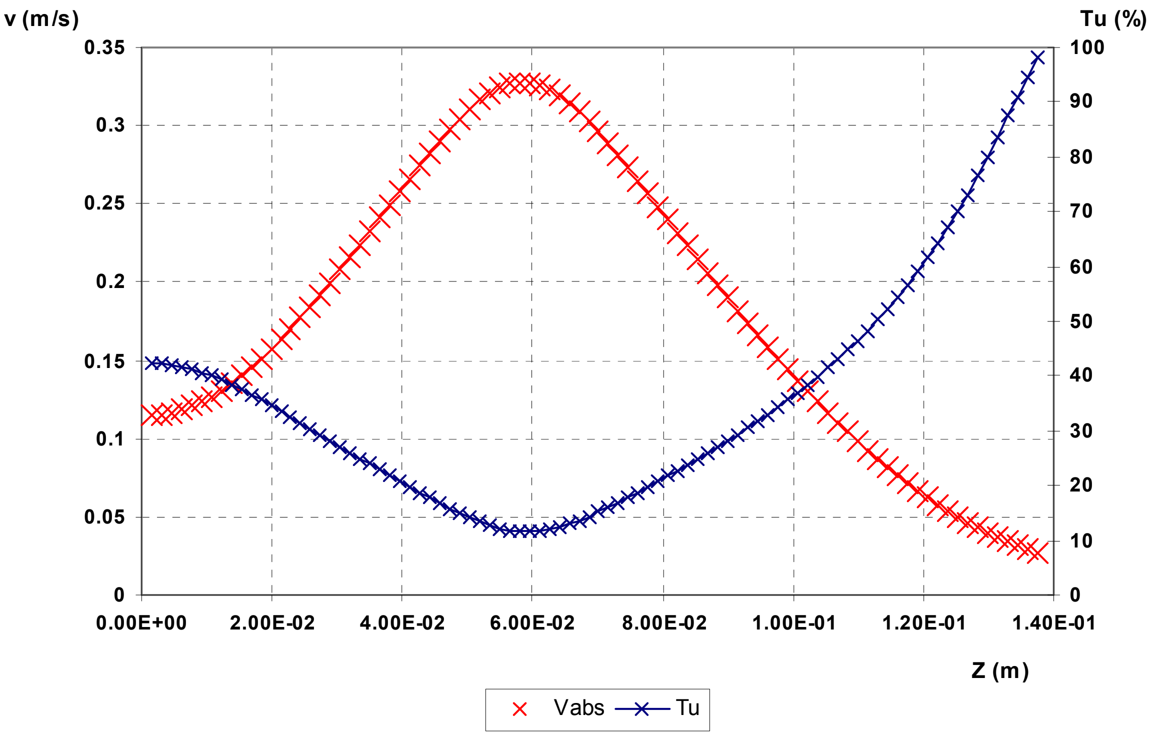

Figure 6: Velocity and turbulence intensity profile at the distance of 0.4m from air inlet on a desk grille.

Velocity field results in the breathing zone of steady-state 3D numerical simulation for isothermal inflow of 3.50 l/s of air with 295K through a desk grille with the angle of inclination set to 30° are presented in Figure 7. Air velocity at the nose level for this case with a maximal proposed amount of air is just under 0.25 m/s, while in other cases with smaller amounts of air, velocities in the breathing zone are consequently lower. Similar is with induction rate of ambient air which is higher at higher amounts of air and vice versa.

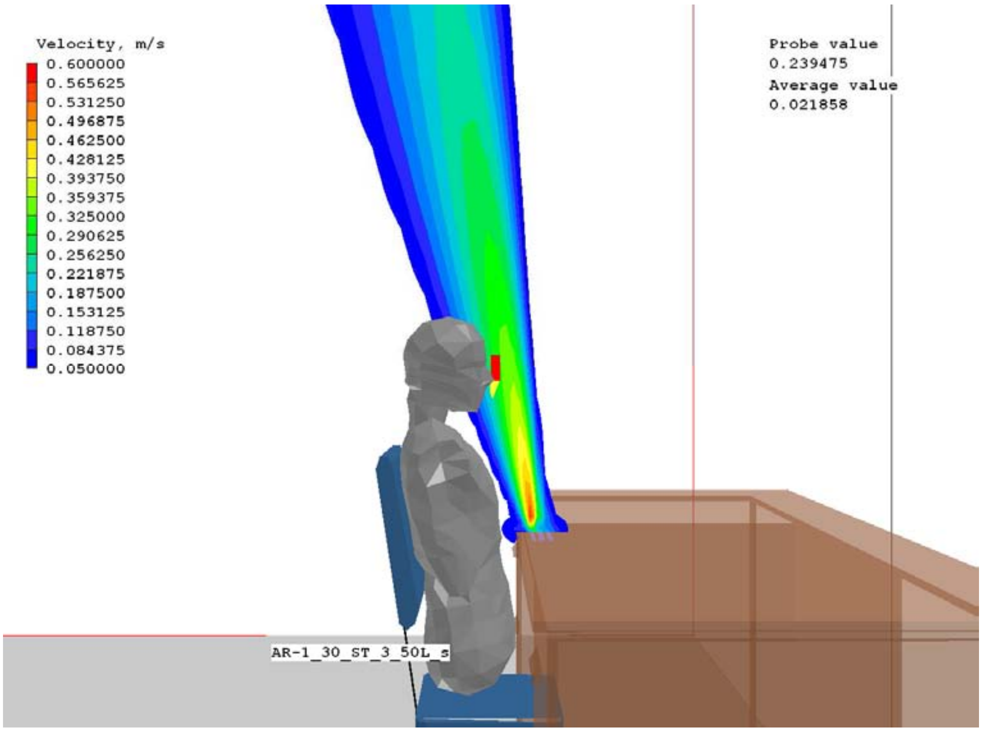

Figure 7: Air velocity distribution in the breathing zone of steady-state 3D simulation for isothermal conditions.

Beside presented results of measurements and simulations of air velocity field on a horizontal desk grill and in the breathing zone of the occupant which were as already mentioned acquired for a case with a maximal amount of air, measurements and simulations were made also for smaller amounts of personalized air in combination with different operating conditions of the system.

Comparison between results of velocity measurements in the breathing zone and at the manikin's nose with results of steady-state 3D simulation for the same operating conditions of PERMICS-LOS1 shows very accurate prediction of air velocity distribution with simulations. Some differences are found between measured and simulated results of turbulence intensity, where there is a higher prediction of turbulence intensity significant for simulation. These findings are important, because they show that it could be possible to optimize the operation of the system not only with measurements but also with simulation tools. However, additional measurements and simulations will be necessary to develop the appropriate numerical model that would be reliable enough for optimization process.

Results of ventilation efficiency measurements show the importance of operating conditions, especially in terms of applying the right air jet orientation and the proper amount of fresh air. It was found that for the same air jet orientation a smaller amount of more laminar air flow assures equal or even higher air quality parameters as a larger amount with higher turbulence level. This is due a reason that there is no actual mixing of air in the laminar flow case as fresh air pushes old air out of the breathing zone and there is less induction rate. The rate of induction is besides dependent also of the width of the air jet at the head level.

CONCLUSIONS

Results of the analysis of ventilation effectiveness of personalized ventilation system PERMICS-LOS1 show that the design of the air terminal device in terms of air distribution is crucial for achieving higher ventilation efficiency in the breathing zone of the occupant. The most influencing parameters are beside the amount of fresh air also the right orientation of air jet and velocity distribution in the breathing zone. Accordingly to measured and simulated data, ventilation efficiency is very dependent of the position of adjustable grilles of the air terminal device and also of the amount and the turbulence intensity of distributed air, while higher amounts of air not necessary assure better air quality in the breathing zone and can also cause a discomfort for the occupant due to draught. Modifications of air terminal device should be considered to achieve wider and thus more uniform air jet in the breathing zone. Further measurements of velocity field and ventilation efficiency, including numerical simulations are necessary for better understanding of the influence of air distribution, especially air jet width on the amount of personalized air in the breathing zone.

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An experimental investigation of a passive chilled beam system in sub-tropical conditions

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SUMMARY

Chilled beam systems are effective in providing good indoor environmental quality (IEQ) in an energy efficient manner. However, chilled beam systems are not common in sub-tropical climates where the requested cooling capacity is much higher than in temperate climates. This paper reports the findings of physical testing undertaken on a simulated office environment when the maximum cooling capacity was 122 W/m². Tests were also undertaken for higher theoretical heat gains (up to 164 W/m²). The mock up test environment included a 3.6m x 3.6m zone, representing a perimeter zone of an office building in a semi sub-tropical southern hemisphere climate. The purpose of the test was to measure the internal temperature distribution and air velocities for different heat gains and also for changes in internal layout. The test results are referenced to recognised standards for occupant comfort, particularly ISO7730. The conducted measures show that the air velocities in the occupied zone were below 0.25 m/s in all tests and the draft rates were below 20 % in nearly all measured points. The location of the workplace also did not have any effect on the air velocities within the space. This shows that passive beam system, including cooled mechanical ventilation, is possible to cover the requested cooling capacity in sub-tropical conditions without any draft.

BACKGROUND

<u>Chilled Beam Application</u>

The application of passive chilled beams in HVAC (Heating Ventilation and Air Conditioning) design for office spaces has been reasonably commonplace in Europe with examples of the application spanning over the last couple of decades. A common design encompasses a combination of ventilation air, distributed by mechanical means to a ceiling (or floor) mounted diffuser, and mechanical cooling via the passive chilled beam element located above a perforated ceiling. The passive beam element (battery type) itself is essentially a series of water tubes with fins attached to increase heat transfer to the air. Cooled water circulates through the beam. Airflow across the beam is driven by the natural buoyancy effects from air cooling through the beam element. Other solutions are active chilled beams where the room ventilation air is integrated into the room unit. This study is focussed on passive chilled beam systems.

Historically the majority of the application of passive chilled beams has been confined to certain room load densities (typically 50 to 80 w /m²). This is likely to be a symptom of the actual design loads encountered in such European regions. Much research has been undertaken in relation to the performance of various combinations and types of chilled beam products for these load scenarios. Zboril et al [1] investigated the air distribution for active beams with internal loads ranging from 50 to 80 w/m². A paper by Fredriksson et al [2] investigates the effect of the location of internal loads on the airflow and finds that the air plume under the beam can be affected by the placement of internal heat sources. It also identifies that air velocities under the beams can fluctuate. The study does not include the presence of mechanical ventilation air.

Other research, including information published by Rehva [3], recommends benchmark limits of capacity for passive chilled beams for “optimum heat loads” to 80 W/m² (watts per square metre of floor area) with a maximum of 120 W /m². These reports equate this to a specific chilled beam capacity of <150 W/m (Watts per linear metre of beam) when the beam is in an occupied zone, and up to < 250 W/m when installed in a non occupied zone. These limits have been established on the basis of air velocities and particularly draft within the space. The guide states that these values should be used as reference values and special attention should be given to room air velocities for higher heat loads. It is noted that these limits are based upon tests with no additional influences from room ventilation systems.

The performance of battery type passive chilled beam systems, in combination with cooled ventilation air and higher internal heat gains has limited research. This is of particular interest in semi tropical climates, including parts of Australia, where internal gains are higher than Europe due primarily to higher ambient temperatures and solar intensity. Perimeter office room sensible design gains can often reach 120 W/m² (or higher), even with relatively high performing double glazed facades.

The application of chilled beam HVAC design for office buildings is gaining popularity in Australia due to its potential of reduced energy consumption in comparison with traditional all-air variable air volume or constant air volume systems. Local regulation and market expectation is also increasing emphasis on energy efficiency in office building designs. Due to the limited application, the performance of chilled beams for higher heat loads (greater than 80 W/m²) is of great interest.

Of course the application of passive chilled beam systems for higher heat load spaces is not limited to spaces with higher fabric loads. Spaces with higher internal gains (e.g. call centres, TV studies etc) could make inference from this study.

<u>Thermal Comfort</u>

An assessment of thermal comfort includes consideration to a number of influences, which can be measured and quantified. These include Dry bulb temperature, humidity (or moisture), radiant temperatures (exposure) and air velocity (or draft). Much study has been undertaken on the interrelation of these items. This study aims to reference standard ISO 7730 standard (1990) which identifies a basic guideline for thermal comfort. This considers the temperature gradient in the room and also the room air velocities as follows (for summer cooling):

Draft rating (DR): < 20 (for category B space. For category C space this is <25)

Vertical air temperature difference: <3°C from foot to head when sitting or standing.

In general terms the differentiation between category B and C relates to the estimated Percentage of People Dissatisfied (PPD), with B targeting less than 10% and C targeting less than 15%. For general office type spaces category B is considered the appropriate target.

The equation for the determination of draft rating can be found later in this report. In general terms the draft rating takes consideration of both the local air turbulence and dry bulb temperature.

METHODS

The experiment was conducted at Halton Research Center. A test room (L *x* W *x* H = 3.6m *x* 3.6m *x* 3.3/2.8m) was used to simulate a real office. In the test room, two passive beams (CPT-105-605-2600/2400) were installed above the perforated false ceiling (3195 mm from the floor level). Three light fittings (each: 2x28 W/T5) and a swirl diffuser (TSR-160) were installed in the false ceiling at the level of 2800 mm from the floor level. The gross free area of the ceiling panels was 37 % and the perforation holes were 5mm. Close to the window wall, there was arranged an additional opening (300 mm) to improve the buoyancy effect, Fig. 1.

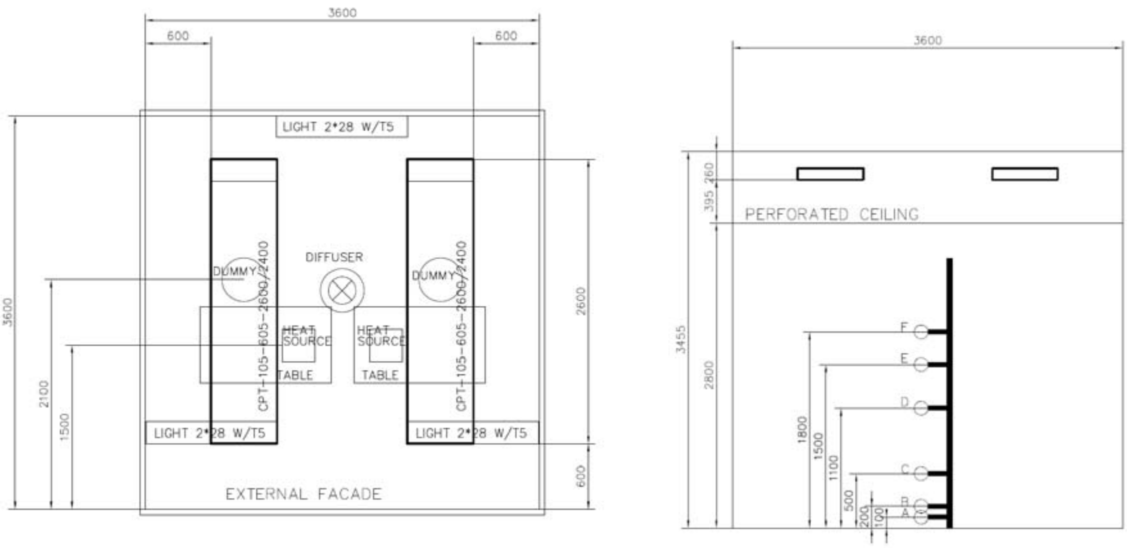

Figure 1. Simulated office (workplace layout 1) and the cross-section of the room space.

In the basic case of 122 W/m^2, the heat loads are: two computers (260 W), two dummies (150 W), light fittings (168 W), warm window surface (35.0 ^{o}C and ~ 600 W) and an electrical foil (400 W) (size of 3.6 m x 1.5 m close to the window) on the floor. The used airflow rate and supply air temperature of the swirl diffuser were 35 L/s (2.7 L/s per m^2) and 13 ^{o}C. The inlet water temperature of the chilled beams was 15 ^{o}C. The requested cooling capacity was arranged by modulating water flow rate of the chilled beams. The room air temperature was in the test cases between 23.5- 24.7 ^{o}C.

Two higher cooling capacity 153 W/m^2 and 164 W/m^2 were also studied. In the case of 153 W/m^2, two extra computers and dummies were installed. The highest cooling capacity (164 W/m^2) was arranged from the case of 153 W/m^2 by increasing the heat gain of the dummies. In this study, there was also analyzed the effect of the location of the workplaces on the performance of the passive chilled beam system. One additional case without false ceiling was measured to analyze the effect of the perforated ceiling panels on the air distribution. The studied cooling capacities and office layout cases are shown in Table 1.

Mean air velocity, air temperature and turbulence intensity were measured within the occupied zone at the locations of 0.1m, 0.2m, 0.5m, 1.1m, 1.5m and 1.8m above the floor. The measurement grid consists altogether 16 locations (altogether 96 points), Fig. 2.

Air flow velocities were measured with velocity sensor type Sensor HT 412 having accuracy ± 1 % of readings. The room air temperature, inlet water temperature, supply and exhaust air temperatures were measured with temperature sensors of type PT100 class A. The water flow rate was measured with Krohne Electromagnetic Flowmeter IFC010 with accuracy less than ± 1% of the readings. The airflow rate was measured with differential pressure transmitter Furness Controls FCO33 with accuracy less than ± 0.5% of the readings.

Table 1. Studied heat load and workplace layout cases.

Case	Room air (^{o}C)	Water flow rate (l/s)	Heat loads (W/m^2)	Workplace (WP) layout:
1	23.5	0.094	122	1:WP in the middle
2	24.1	0.094	122	2:WP close to the side wall
3	24.0	0.094	122	3:WP close to the window
4	23.4	0.094	122	1:WP in the middle (no ceiling panels)
5	24.7	0.196	153	1:WP in the middle
6	24.1	0.250	164	1:WP in the middle

For measuring of velocity, temperature and turbulence intensity 6-channel low-velocity anemometer HT 400 was used. The used 6 omni-directional velocity probes have spherical velocity sensor with a diameter of 2 mm, ensuring a fast response. The temperature sensor is shielded against radiation. Instantaneous values of velocity and temperature are measured simultaneously. Measurements of velocity and temperatures were time averaged over 180 s.

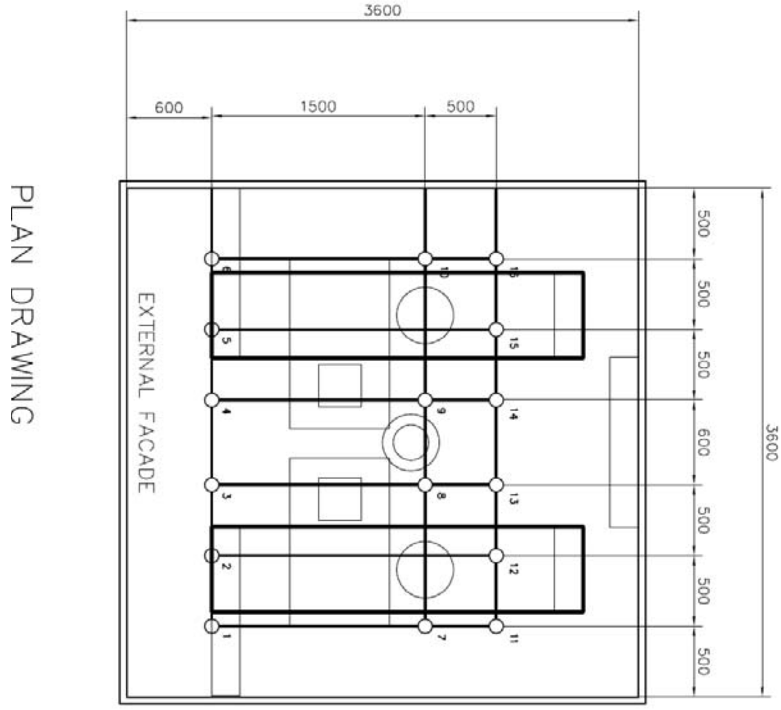

Figure 2. Measurement grid of the conducted experiment when workplaces are in the middle.

RESULTS

In Fig 3, there is presented the measured air velocity, temperature, turbulence intensity and draft rate profiles for test case 1. The measured velocity profile depicts that the air is well mixed over the whole occupied zone. Even with a heat gain of 122 W/m^2, the maximum air velocity was only 0.20 m/s. Also, the draft rate values were acceptable level.

	1				2				3				4				5				6			
Height (m)	v (m/s)	T_a (°C)	Turb. (%)	DR %	v (m/s)	T_a (°C)	Turb. (%)	DR %	v (m/s)	T_a (°C)	Turb. (%)	DR %	v (m/s)	T_a (°C)	Turb. (%)	DR %	v (m/s)	T_a (°C)	Turb. (%)	DR %	v (m/s)	T_a (°C)	Turb. (%)	DR %
1.80	0.09	22.7	41	7	0.07	22.8	41	4	0.06	23.0	45	3	0.06	23.0	60	3	0.07	22.9	46	4	0.09	22.8	53	7
1.50	0.10	22.6	32	8	0.09	22.7	31	6	0.06	22.8	48	3	0.07	22.8	53	4	0.09	22.8	43	7	0.09	22.7	38	7
1.10	0.12	22.6	28	10	0.12	22.6	28	10	0.09	22.6	37	7	0.09	22.7	48	7	0.10	22.7	35	8	0.12	22.7	35	10
0.50	0.09	22.6	39	7	0.13	22.4	33	11	0.12	22.4	35	10	0.12	22.4	35	10	0.11	22.6	40	9	0.10	22.7	45	8
0.20	0.11	22.8	43	10	0.12	22.7	33	10	0.13	22.6	33	11	0.13	22.6	27	11	0.12	22.8	35	10	0.12	22.8	38	10
0.10	0.16	23.1	29	13	0.15	23.0	28	12	0.15	23.0	26	12	0.16	23.0	22	12	0.18	23.2	23	14	0.17	23.2	26	14

	7				8				9				10			
Height (m)	v (m/s)	T_a (°C)	Turb. (%)	DR %	v (m/s)	T_a (°C)	Turb. (%)	DR %	v (m/s)	T_a (°C)	Turb. (%)	DR %	v (m/s)	T_a (°C)	Turb. (%)	DR %
1.80	0.15	22.2	40	15	0.10	22.4	41	8	0.11	22.2	40	10	0.14	22.1	46	15
1.50	0.13	22.2	40	12	0.12	22.3	35	11	0.13	22.1	35	12	0.14	22.1	43	14
1.10	0.10	22.5	46	9	0.14	22.3	34	13	0.14	22.1	38	14	0.15	22.1	38	15
0.50	0.13	22.4	38	12	0.19	22.1	22	16	0.17	22.1	30	16	0.16	22.0	33	16
0.20	0.16	22.1	28	15	0.16	22.1	21	13	0.13	22.2	30	11	0.14	22.1	28	12
0.10	0.20	22.1	22	18	0.18	22.1	22	15	0.15	22.2	31	14	0.15	22.1	29	14

	11				12				13				14				15				16			
Height (m)	v (m/s)	T_a (°C)	Turb. (%)	DR %	v (m/s)	T_a (°C)	Turb. (%)	DR %	v (m/s)	T_a (°C)	Turb. (%)	DR %	v (m/s)	T_a (°C)	Turb. (%)	DR %	v (m/s)	T_a (°C)	Turb. (%)	DR %	v (m/s)	T_a (°C)	Turb. (%)	DR %
1.80	0.14	22.0	43	14	0.09	22.0	49	8	0.09	22.3	44	7	0.10	22.2	44	9	0.17	21.8	47	20	0.19	21.8	38	21
1.50	0.12	22.0	47	12	0.10	22.0	36	8	0.08	22.2	46	6	0.13	22.1	45	13	0.15	21.8	40	16	0.15	21.8	43	16
1.10	0.14	22.0	40	14	0.12	22.0	39	11	0.09	22.1	42	7	0.14	22.0	35	13	0.14	21.9	38	14	0.15	21.8	47	17
0.50	0.13	22.0	39	13	0.13	22.1	38	12	0.12	22.1	34	11	0.13	22.2	37	12	0.14	21.9	34	13	0.12	21.9	40	11
0.20	0.09	22.1	42	7	0.09	22.2	39	7	0.10	22.1	38	8	0.11	22.1	34	9	0.11	22.0	42	10	0.09	22.1	41	7
0.10	0.11	22.1	36	10	0.09	22.2	36	7	0.11	22.2	37	10	0.11	22.1	33	9	0.10	22.0	37	8	0.08	22.1	41	6

Figure 3. Measured air velocity, temperature, turbulence intensity and draft rate in case 1.

Changes in office layout did not have any effect of the performance of the air distribution. This is seen in case 2 &3. The maximum air velocity was 0.20 m/s when the workplaces were located in the middle of the room , close to the side wall or close to the window. Also when the false ceiling was removed, the air velocities did not increased. In fact, the maximum velocity slightly reduced.

When the cooling capacity was increased from 122 W/m^2 to 153-164 W/m^2, the maximum air velocity was increased to value of 0.23 m/s. Also, the draft rate values were acceptable level: only in the couple of points DR- index was higher than 15 %. It should be noted that the typical limit of air velocity is set to 0.25 m/s during cooling season. Thus, the studied chilled beam concept fulfills this international standard threshold for the air velocity even when the cooling capacity was very high 150-160 W/m^2.

Table 2. Measured air velocities and draft rate (DR) in the test cases.

Case	**1**	**2**	**3**	**4**	**5**	**6**
Heat gain (W/m^2)	122	122	122	122	153	164
Workplace location	Middle	Side	Window	Middle	Middle	Middle
v_{max} (m/s)	0.20	0.20	0.20	0.17	0.23	0.23
v_{avg} (m/s)	0.12	0.10	0.09	0.12	0.13	0.13
Line number of the max. velocity	7	2	7	9	10	7
DR_{max} (%)	21	19	20	19	20	20
DR_{avg} (%)	11	8	8	11	11	11
Number of point DR> 15 %	8	2	7	6	16	15
Number of lines DR> 15 %	5	1	3	4	7	6

DISCUSSION AND CONCLUSIONS

Several tests were performed in order to establish the draft and heat gain performance of a mock up passive chilled beam installation under a number of scenarios. In summary these scenarios included:

- Adjustment of internal heat gains from 120 W/m² through to 164 W/m² (all cases);
- Adjustment to the internal layout of equipment, desks and people (case 2&3);
- Adjustment to the water flow rates delivered to the passive chilled beams in order test for increased capacity;
- Removal of perforated ceiling (case 4);

The results identify that within the range of these tests the draft rate (DR) could be maintained below 20 (with one reading at 21 for case 1) which would indicate compliance with ISO7730 For a category B building. More specifically the following was observed.

The removal of the perforated ceiling produced minimal difference to the draft and temperature performance for the internal heat gain loads at 120 W/m². Similarly, increasing the internal gains to 164 W/m² (case 6) resulted in broadly equivalent performance in terms of draft. At first glance this appears to be in contradiction to established limits of performance for battery type passive beam products. However further review indicates that the influence of the mechanically introduced air supply via swirl outlet diffusers has a profound effect on the air flow within the space. This can be illustrated by the smoke test demonstrations where it is apparent that the air movement is driven from these diffusers.

Similarly, adjustments to the internal layout of equipment produced little effect on the air movement. This could be attributed to the relatively small load as a percentage of the simulated fabric and lighting gains, which were fixed.

In all cases, the whole volume is fully-mixed as demonstrated by the measured velocities and from smoke tests. There is not any location in the occupied zone that exhibited excessively high velocities. Also the maximum velocity is approximately 2-times higher that average velocity in all cases.

Temperature distribution across the height of the room in all cases is significantly less than the 3°C target. Again the fully mixed environment is reason for this.

In conclusion it is apparent that the total performance is a combination of the influence of the cooling elements, air diffusion and thermal plumes of heat gains (although in this instance the affect of thermal plume was limited to changes in internal layout, which resulted in negligible affect). Focusing on the performance of a chilled beam element in empty space will not give the total view. All elements should be analysed concurrently. In cases where heat gains are high (> 80 W/m²) , this is required. Analysis could be undertaken using full-scale mock-up or with CFD simulation. However special attention should be made to the boundary conditions and modelling of heat gains to achieve an accurate result.

These tests found that under certain conditions a combination of cooled air supply through swirl outlet diffusers, and battery type passive chilled beams can provide room sensible cooling capacities of 120 W/m² and potentially up to 164 W/m² without exceeding the requirements of ISO7730 for occupant comfort for category B office buildings. It is believed that the mechanically introduced air supply plays a significant role in the resultant air velocities within the space. The extent of influence of this is likely to be dependant upon the diffuser selection and airflow rate and this should be given due consideration. The tests presented did not compare different diffuser types so quantifying the effect of this may be the subject of further study.

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Study on the Distribution Law of Human Adjacent Environmental Parameters and Prediction on Human Thermal Comfort

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SUMMARY

Task/ambient air-conditioning (TAC) system allows a user to control his microenvironment flexibly by setting the local supply air outlet and control device near his working location. Present researches show that this system can improve human thermal comfort remarkably. With the help of numerical simulation, the paper mainly studies the temperature and PMV (Predicted Mean Vote) values both in working and ambient area. The results show that the air temperature in front of body is lower than in the back of body, and the lowest temperature is in the head region. And PMV values both in working and ambient area can be kept within the range of -1~+1 by proper combinations of supply air temperature and volume. Finally some detail suggestions on local supply air parameters are given to keep PMV values in the whole working area within -1~+1, and those in the ambient area below +2, and also the temperature difference between the head and the feet levels in the ambient area less than 3℃ under the background air temperature at 28℃. It would be fascinating to further study on the mechanism of asymmetrical air parameters for human thermal comfort.

INTRODUCTION

TAC system is an air-conditioning system that has the air outlets and control devices set together near working stations so that workers in working area could control their individual environment flexibly. Current researches on TAC system show that it not only can improve human thermal comfort and indoor air quality, but also has the advantage and potential of lowering energy consumptions [1-3]. Desktop TAC system can supply air directly to breathing zone by air outlet mounted on desktops and provides each user with control devices for their local environment adjustment. In recently ten years, many scholars have been dedicated their effort to the TAC technology, mainly focusing on human thermal comfort and air quality with much achievement obtained [2-5]. One of the differences between TAC system and traditional system is that parameters such as temperature and air velocity around a human body are non-uniform. So characteristics of environmental parameters distribution of TAC system should be well studied in order to solve its thermal comfort problem. In this paper, the room airflow and environmental parameters distribution within work stations under different supply air parameters of TAC system have been studied using the CFD software of PHOENICS.

METHODS

I. Air-Conditioning System Arrangement

Eight work stations isolated by separation boards are set in the air-conditioned room dimensions, 7.5m×5.6m×3.6m. The overhead air delivery background air-conditioning system is adopted with ceiling mounted return air grilles far away from supply air outlets whose dimensions are 12cm×14cm.

Since the cooling load comes from the indoor facilities and human bodies in an inner zone of a large office building, only the indoor heat sources are considered in this study. The cooling load includes eight computers, eight workers and four fluorescent lamps where the heat capacity of a computer is 300w, a human body 60w, and a light 40w. And a human body is simplified to a cube. The supply air temperature of the background air-conditioning system is set at 20℃ and the amount of air supply is calculated by maintaining the indoor air temperature at 28℃ according to the following formula:

$$G = \frac{Q}{\rho c_p (t_n - t_o)} \quad (1)$$

where G is supply air volume flow rate, m^3/h; Q is cooling load, W; ρ is air density, kg/m^3; c_p is specific heat, $kJ/(kg\cdot K)$; t_n is the calculation temperature of indoor air, ℃; t_o is the supply air temperature, ℃.

II. The Basic Air Turbulence Model[6-7]

Usually, control equations describing air motion can be written in the following universal pattern:

$$\frac{\partial(\rho\phi)}{\partial t} + \nabla\cdot\left[(\rho V\phi) - \Gamma_\phi grad(\phi)\right] = S_\phi , \quad (2)$$

where Φ is dependent variable; ρ is air density; V is the vector of air speed; Γ_Φ is the coefficient of air diffusion; S_Φ is the source term of Φ; t is air temperature.

Continuation equations, momentum equations and energy equations can be gotten for the different meaning of Φ in the study.

For continuation equations:

$$\Phi=1; S_\Phi=0; \Gamma_\Phi=0 \quad (3)$$

For momentum equations:

$$\Phi=u \quad (4)$$

$$S = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial u}{\partial x}\right) + \frac{\partial}{\partial y}\left(\mu_{eff}\frac{\partial v}{\partial x}\right) + \frac{\partial}{\partial z}\left(\mu_{eff}\frac{\partial w}{\partial x}\right) \quad (5)$$

For energy equations:

$$\Phi=T \quad (6)$$

$$\Gamma_{\phi} = \frac{\mu}{P_r} + \frac{\mu_t}{\sigma_T} \quad (7)$$

$$S_{\Phi}=0 \quad (8)$$

In the above equations, P_r is for air pressure; μ_{eff} is for equivalent viscosity; μ is for laminar air dynamic viscosity; μ_t is for turbulent air dynamic viscosity; σ_T is for turbulent Prandtl number.

It is usually impossible to solve the above partial differential equations directly except for some very simple cases. So numerical methods have been used, which include finite difference method, finite element method, border element method and finite analysis method where the last three methods have gained much progress by dissolving some difficult problems on fluid flow and heat convection. As the finite difference method still prevails over the others because of the mature level, easiness of putting into practice and broadness of application fields, the finite volume method—one of finite difference method—is adopted in the study.

III. The Meshing Model

Much more nodes and smaller meshes should be arranged around human bodies, computers, and air supply grilles due to the relatively sharper gradient of temperature and velocity around them. The total number of calculating meshes to 130,340 at the end of the model meshing stage.

IV. Working Modes of Simulation

All twenty five working modes are selected for simulation, in which there are five air supply volumes in combination with each of five different temperatures. All the working modes are listed and numbered in table 1.

Table 1. Different working modes for simulation

Air supply temperature (℃)	Air supply volume (m^3/h)				
	20	40	60	80	100
14	A_1	A_2	A_3	A_4	A_5
16	B_1	B_2	B_3	B_4	B_5
18	C_1	C_2	C_3	C_4	C_5
20	D_1	D_2	D_3	D_4	D_5
22	E_1	E_2	E_3	E_4	E_5

RESULTS

I. Comparison of Simulation and Field Test

A TAC system is set up in an environment controlled chamber according to simulation conditions. Twelve testing points are set within the working area, and four points are set in the ambient area at different heights for the air temperature and velocity measurement.

Multi-functional instruments are used to measure air temperatures and velocities. The temperature measurement range is -20℃~+70℃ in accuracy of ±0.4℃, and velocity measurement range is 0~5m/s in accuracy of 0.03m/s+4%m.v. Figure 1~3 show the testing and simulating results only at the height of one meter due to the restriction of the paper length. It can be seen from the figures that the temperature and velocity distribution between testing and simulating results are similar at the largest temperature difference of 4.05% and the biggest velocity deviation of 0.07m/s.[9]

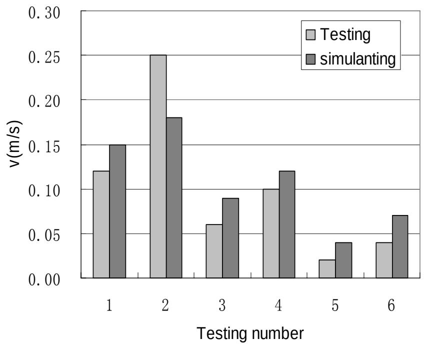

Fig.1. Air velocity comparison in working area

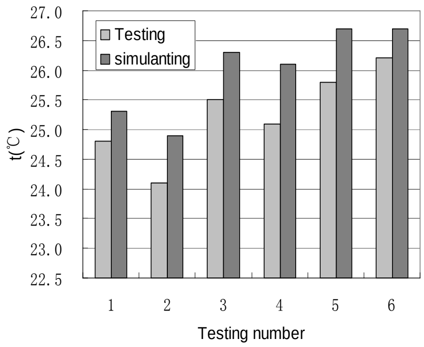

Fig.2. Air temperature comparison in working area

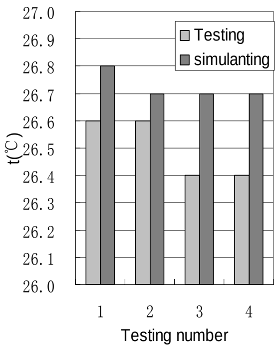

Fig.3. Air velocity comparison in ambient area

II. Temperature Distribution around a Human Body

Comparisons have been made between the mean air temperature in front of body (at the height of 0.6m~1.2m) and that in the ambient area (below the height of 2.0m) under different working modes by means of numerical simulation method. Refer to figures 4-7. [9]

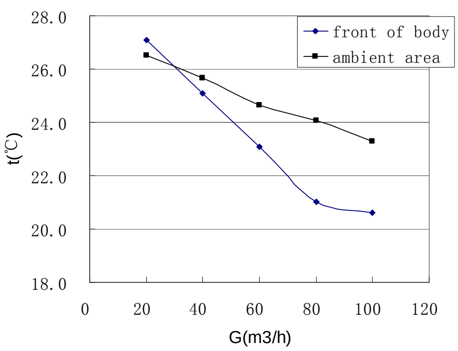

Fig.4. Mean air temperature variation with different air supply volumes at 16℃

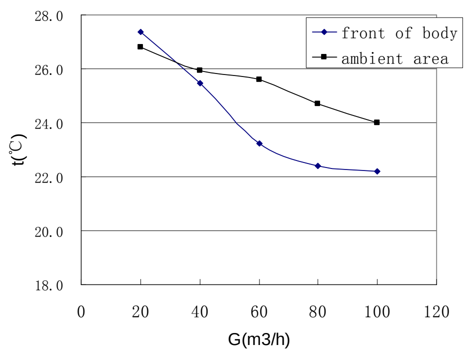

Fig.5. Mean air temperature variation with different air supply volumes at 18℃

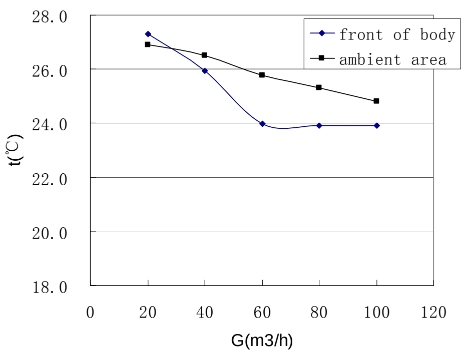

Fig.6. Mean air temperature variation with different air supply volumes at 20℃

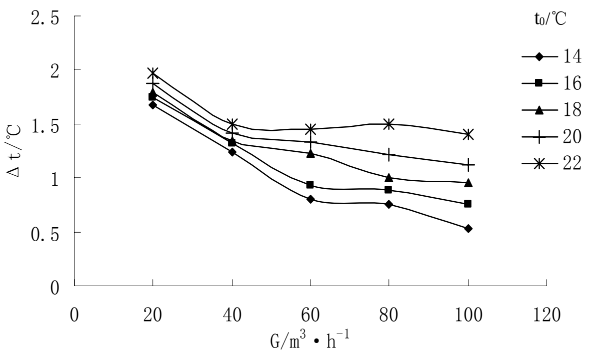

Fig.7. Mean air temperature differences at head and feet levels in ambient area at different working modes

It can be seen from figure 4~6 that air temperatures in front of body are slightly higher than in ambient area when the supply air volume flow rate is 20m^3/h because of the heat flow coming from heat sources which mainly located in the working area. As the supply air volume reaches 40m^3/h, the air temperatures in front of body become lower than those in the ambient area. Moreover when the supply air temperature is at 14℃ or 16℃ and the supply air volume is below 80m^3/h, the air temperatures in front of body will drop dramatically as the supply air volume increases. And the tendency that the air temperature in front of body lowers as the supply air volume increases will be slowed down when the supply air volume is over 80m^3/h. But when the supply air temperature is at 18℃, 20℃ or 22℃, for turning point of the tendency, the supply air volume flow rate is about 60m^3/h. It is also found that the lower the supply air temperature, the lower the mean air temperature in the working and ambient area under the same supply air volume. And when the supply air temperature is set at 14℃, the mean air temperature in front of body will drop to below 20℃, but when the supply air temperature is raised up to 22℃, the mean air temperature in front of body will only drop to 25℃ under the same supply air volume. Though the mean air temperature difference between the front body and the ambient environment can reach as high as 3℃ at a relatively lower supply air temperature, this temperature difference will drop to some 1℃ at a higher supply air temperature.

Figure 7 shows the air temperature differences between the head and the feet levels in the ambient area with different supply air volumes under the five supply air temperatures. It can be seen from figure 4~7 that all the air temperature differences between the head and the feet levels in the ambient area under the twenty five working modes are kept within 2℃. So under calculation conditions, thermal discomfort can be avoided in spite of the large vertical temperature gradient in the ambient area. It can also be found that the air temperature differences between the head and the feet levels will lower as the supply air volume increases at the same supply air temperature, but after the supply air volume reaches 60m^3/h, the temperature difference basically remains unvaried. [9]

III. PMV Distribution

PMV is a thermal environment index established by P.O. Fanger on the basis of thermal comfort equation after he collected thermal sensation data of 1396 American and Danish subjects. The difference between PMV and other thermal environment indexes is that the PMV value is the occupants' mean vote for a thermal environment based on ASHRAE's classification on thermal feeling instead of a certain equivalent temperature. [10] The index values of the ASHRAE's classification on thermal feeling are from 1 to 7, and these values minus 4 are the PMV index values, that is from -3 to 3, in which zero means neutral feeling (thermal comfort), values below zero stand for cool feeling, and values above zero stand for warm feeling.

It is confirmed that human thermal comfort can be improved by modification of the supply air temperature or volume or both from the simulation calculation in the working area. As can be seen from figure 8 and 9 that although the background temperature remains at 28℃, a slightly cool feeling (PMV<0) can still be felt at relatively low supply air temperature with large amount of supply air, but thermal feeling will become warm (PMV>0) when the supply air temperature is raised up to 22℃. So it can be predicted that in a relatively high ambient air temperature, human thermal comfort can be improved effectively by environmental parameters difference around a human body produced by a local air supply system.

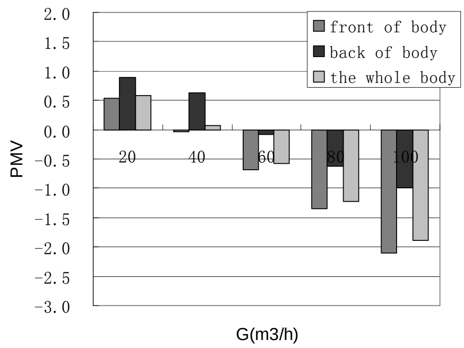

Fig.8. Human thermal comfort at supply air temperature of 16℃

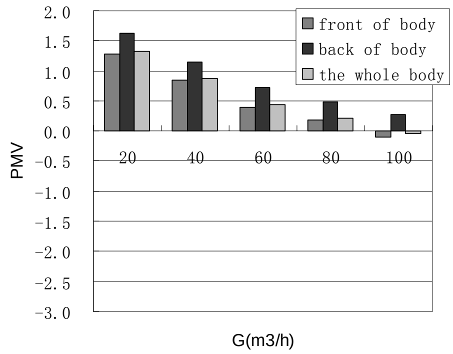

Fig.9. Human thermal comfort at supply air temperature of 22℃

The relationships among the thermal feeling of the whole body in the working and ambient area and the local supply air temperature and volume in all the twenty five working modes are drawn in figure 10 and 11 according to simulation calculations. It is found that PMV values can be kept from -1.0 to +1.0 in most working modes, and the highest PMV value is +1.4 while the lowest is -0.9 in the ambient area at all working modes. So a person will feel neither too cool nor too warm within working area or ambient area.

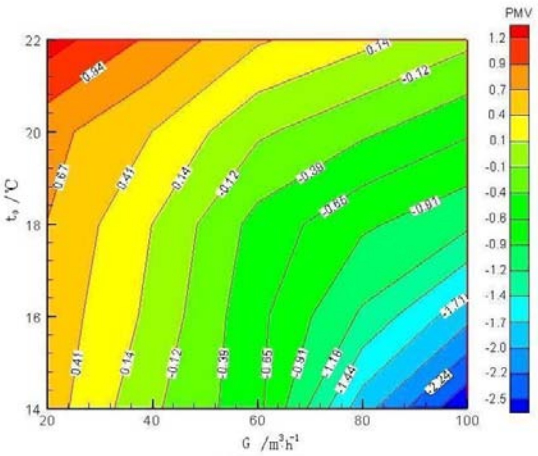

Fig.10. Thermal feeling of the whole body in working area with different supply air temperatures and volumes

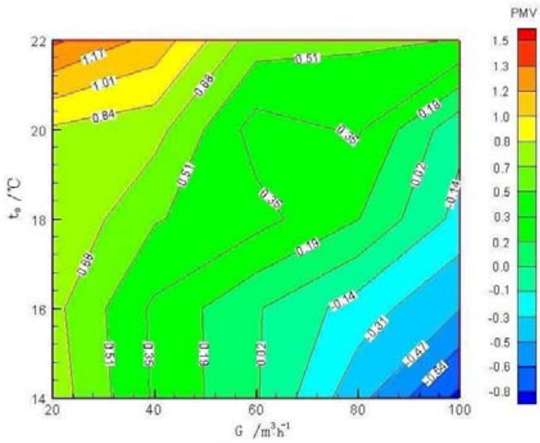

Fig.11. Thermal feeling of the whole body in ambient area with different supply air temperatures and volumes

DISCUSSION

I. The air temperature in the head region is apparently lower than in other areas in the desktop TAC system, but at the lower level of the room, the air temperatures around a human body have no distinct differences from that in ambient area.

II. The maximum mean air temperature difference between the front body and the ambient area is 3℃ under different supply air temperatures and volumes in all working modes.

III. The PMV values can be kept in a range of -1 ~ +1, in working and ambient area under different combinations of supply air temperatures and volumes. The prediction could be made that even at a relatively high ambient air temperature, human thermal comfort will be improved effectively by changing the environmental parameters around the human body through produced a local air supply system.

IV. The following suggestions on supply air parameters of desktop TAC system should be achieved in order to keep the PMV values in the working area between -1 and +1, and the ambient area below +2, and also the temperature differences between the head and the feet levels in the ambient area is less than 3℃ when the indoor air calculation temperature of the ambient air-conditioning system is 28℃: The supply air volume should be 20~60m^3/h if the supply air temperature is 14℃ or 16℃; The supply air volume should be 20~80m^3/h if the supply air temperature is 18℃; The supply air volume flow rate should be 20~100m^3/h if the supply air temperature is 20℃; And the supply air volume should be 40~100m^3/h if the supply air temperature is 22℃. It would be better to set the supply air volume at 40~60m^3/h for energy efficiency.

ACKNOWLEGEMENT

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Evaluation and rating of building performance

A Systematic Method for Improving Indoor Environment Quality through Occupant Satisfaction Surveys

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SUMMARY

A systematic method for assessing and improving indoor environment quality (IEQ) is developed for existing and occupied office buildings. The method begins with an occupant satisfaction survey that is directed to everyone working in a building. The structure of the questions follows a pattern that offers valuable information of the technical reasons leading to dissatisfaction. The questions assess satisfaction with the following IEQ areas: office layout, office furnishings, thermal comfort, indoor air quality, lighting, acoustics, and building cleanliness and maintenance.

A database of 25 office buildings in Southern Finland has been collected. The average value of all indoor environment factors in these 25 buildings is 0,85 in scale from –3 to +3. This average value can be used as a benchmark value to compare workplaces to each other. Office layout, office furnishings, lighting and cleaning have received higher scores than thermal comfort, air quality and acoustics. Also the variation between buildings has been bigger in thermal comfort, air quality and acoustics. These are also the areas where most of the technical problems have been found. In each building the occupant perception map lays the foundation for auditing and evaluating the indoor environment quality and for locating the possible technical problems.

INTRODUCTION

Well-being of people and environmental issues are some of the major concerns of our European societies today. Several hundreds of thousands of people die too early due to environment in Europe each year. Increased hospital admissions, extra medication and millions of lost working days are not only a financial issue but above all they influence the quality of our everyday life. [1]

People in modern societies spend most of their time (90%) in indoor spaces such as at home, work, school and in vehicles. Improving the quality of indoor environment enhances well-being and productivity of people. Prevention of dissatisfaction created by poor indoor environment and disoperation of technical systems needs to be our aim.

There are many effects of inappropriate indoor environment quality, like increased need to keep breaks during work, decreased concentration or fatigue. In extreme cases problems can lead to absence from work due to permanent or temporary health effects like headache, eye, skin, throat or nose irritation, thermal stress, allergy or asthma. [2]

Productivity of work is one of the most important factors influencing the profitability of business. Often other factors like cost of work or material are determined by external reasons. Good indoor environment improves productivity quicker than changes in working habits or skills, especially if indoor environment has not been in focus before. In a typical company operating in office environment 90 % of total company costs consist of employees' salaries. The rest of the costs include workplace related investment and running costs. International research has proven a development of 2-10 % in workers' productivity, when indoor environment has been improved. [3]

Indoor environment can create dissatisfaction and health problems for users due to various reasons. Often the reason is the performance of technical systems. In many cases technical problems are local and that is why it is difficult to find and locate them. Occupants complain about too hot or too cold temperatures during summer, too hot or too cold temperatures during winter, draughts from ventilation system, stuffy or stale indoor air or lack of sound privacy in open plan offices. At the same time average physical measures (e.g. room air temperature) and average occupant satisfaction may show high comfort.

When occupant satisfaction for thermal environment is known, the indoor climate category can be specified based on CEN Standard prEN 15251 (draft) "Criteria for the Indoor Environment including thermal, indoor air quality, light and noise", published in 2005 [4]. This standard specifies the three categories of indoor environment that shall be selected for a space to be conditioned (see Table 1 for recommended criteria and categories for the thermal environment).

Table 1. Recommended criteria and categories for the thermal environment according to CEN prEN 15251

Category	Thermal State of the body as a whole		Local Discomfort			
	Predicted percentage of dissatisfied PPD [%]	**Predicted Mean Vote PMV [-]**	**Percentage of dissatisfied due to draft DR [%]**	**Percentage of dissatisfied due to temperature difference [%]**	**Percentage of dissatisfied due to warm of cool floor [%]**	**Percentage of dissatisfied due to radiant asymmetry [%]**
A	**< 6**	**-0,2 …0,2**	**< 15**	**< 3**	**< 10**	**< 5**
B	**< 10**	**-0,5 …0,5**	**< 20**	**< 5**	**< 10**	**< 5**
C	**< 15**	**-0,7 …0,7**	**< 25**	**< 10**	**< 15**	**< 10**

Occupant satisfaction with the average thermal conditions can be translated to performance of workers (Fig. 1). That could be used as a tool to estimate the potential productivity improvement and furthermore to calculate the total investment potential.

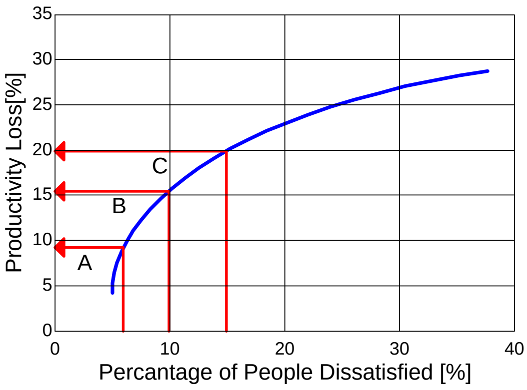

Figure 1. Linkage between percentage of dissatisfied people for thermal comfort (%) and productivity loss (%) [5].

METHODS

A systematic method for assessing and improving indoor environment quality (IEQ) is developed for existing and occupied office buildings. The method begins with an occupant satisfaction survey [6] that is directed to everyone working in a building. The structure of the questions follows a pattern that offers valuable information of the technical reasons leading to dissatisfaction. Questions focus on the human experience of indoor environment, not leading respondents directly to problems or symptoms. By using the occupant satisfaction survey all comments, also the silent comments, can be collected and taken into account when deciding on improvement actions. The questions assess satisfaction with the following IEQ areas: office layout, office furnishings, thermal comfort, indoor air quality, lighting, acoustics, and building cleanliness and maintenance.

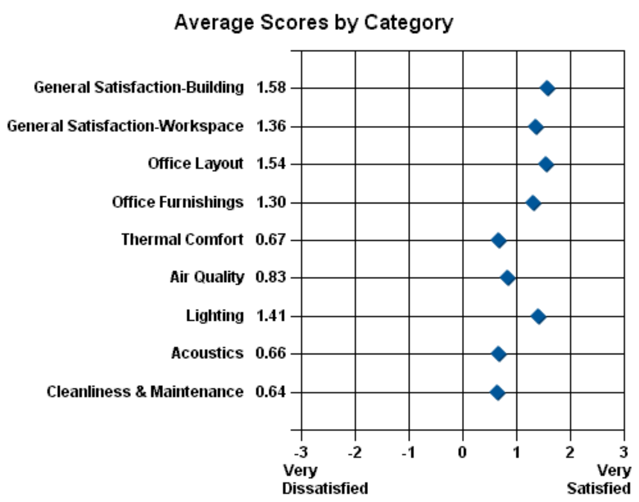

Figure 2. Average occupant satisfaction survey results normally show good values in many areas of indoor environment. [6]

Average satisfaction values between different evaluation areas (Fig. 2) do not vary as much as the local values between different areas of a building. This is why it is important to analyze results not only at building level, but also to locate the possible dissatisfaction more exactly into floors and facades. This helps to allocate actions to specific areas where they are needed. Furthermore, if the exact problems have been localized it is often possible to optimize the

HVAC-system so that both the energy usage of the system and the dissatisfaction of users can be reduced at the same time (e.g. when users complain too cold room air during summer). Table 2 presents an example of an occupant perception map.

Table 2. Occupant perception map (% of dissatisfied people) on different floors and facades helps to identify and locate indoor environment problems.

		All	2nd floor	3rd floor	4th floor	5th floor	6th floor	North	South
General satisfaction	Building	3%	3%	0%	0%	13%	7%	3%	0%
	Workplace	5%	3%	6%	0%	29%	0%	0%	14%
Office furnishings	Comfort	10%	15%	12%	8%	0%	6%	7%	0%
	Adjustment	10%	9%	15%	9%	0%	13%	10%	14%
Thermal comfort		26%	36%	12%	58%	0%	35%	19%	43%
Air quality		20%	39%	12%	8%	25%	12%	10%	43%
Lighting	Quantity	9%	15%	6%	25%	0%	0%	0%	29%
	Quality	12%	12%	12%	17%	13%	12%	16%	0%
Acoustic quality	Sound level	16%	15%	21%	33%	13%	6%	10%	43%
	Acoustic privacy	32%	21%	36%	58%	25%	35%	16%	57%

As an example, we can calculate the productivity improvement potential of e.g. thermal environment presented in Table 2. There 26 % of the personnel are dissatisfied or very dissatisfied with the thermal environment. 25 % of the personnel also think that the present conditions interfere significantly with their ability to get their job done. If the quality level of indoor environment is improved to class A (CEN prEN 15251), the improvement in performance according to scientific studies is 5.8 %. Total monetary benefit for increasing the productivity of 100 people is € 290.000 per year with annual labor cost of € 50.000 per person. Total possible investment for improving thermal conditions is € 1.680.900 with five years of repayment time and 3.0 % of interest rate.

A technical analysis requiring multidisciplinary knowledge is conducted in building areas where more than 30 % of respondents are dissatisfied. Systems creating and maintaining indoor environments consist of many components that are usually provided by different companies. Cross-scientific team of people representing various aspects of technical services conducts the technical analysis in the building. Improvement actions are determined based on the results of the survey and the technical analysis.

Sometimes technical problems are more complicated to solve, especially if there are any doubts of local thermal conditions e.g. drought. In those cases a full-scale mock-up can be built into laboratory conditions to find a technical solution. Another option is to use CFD simulations, but then it is critical to know the real boundary conditions of the specific terminal unit and the heat loads in the room space.

RESULTS

A database of 25 office buildings occupying 5200 people in Southern Finland has been collected. The smallest building was with 30 occupants and the largest building occupied 1200 people. The average value of all indoor environment factors of all 25 buildings is 0,85 in scale from –3 to +3. The highest average value has been 1,24 and the lowest 0,42. This average value can be used as a benchmark value to compare workplaces to each other (Fig. 3).

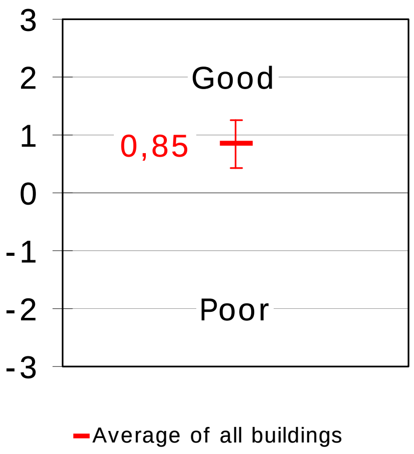

Figure 3. The average value of all indoor environment areas of all 25 studied buildings can be used as a benchmark value to compare workplaces to each other.

When searching technical improvement areas in buildings it is more valuable to present each indoor environment area separately. The evaluation of a building does not necessarily describe the magnitude of problems, because in occupants' evaluation the scale of different elements varies. Office layout, office furnishings, lighting and cleaning have received higher scores than thermal comfort, air quality and acoustics. The average values of each indoor environment area in the database of 25 buildings are in descending order:

- Lighting 1,25
- Building in general 1,23
- Office furnishings 1,22
- Office layout 1,20
- Workspace in general 1,12
- Cleaning 1,03
- Technical maintenance 0,97
- Indoor air quality 0,83
- Thermal comfort 0,49
- Acoustics 0,18

Also the variation is bigger in thermal comfort, indoor air quality and acoustics (Fig. 4). These are also the areas where most of the technical problems have been found. Occupants in office buildings complain mostly about:

- Too cold temperatures,
- Draughts from ventilation system,
- Stuffy or stale indoor air,
- Cold radiation of windows,
- Lack of sound privacy in open plan offices and
- Poor quality of cleaning service.

The typical technical reasons have been:

- Non-balanced ductworks or pipe works,
- Diffusers, which do not have a proper throw pattern
- Chilled beams, which are out of optimum operation range
- Lack of planning in room acoustics

- Operation of building management system.

All the analyzed buildings have typical Scandinavian room systems: chilled beams and mixed flow systems.

Sometimes reason has been a device failure, but too often it is just a change in office layout or change in usage of space. Changes have been completed in furniture and wall layout, but no adaptation of technical systems has been made. Also the combination of air diffusion, warm or cold window surface and layout of office furnishings may create unexpected airflows in the space.

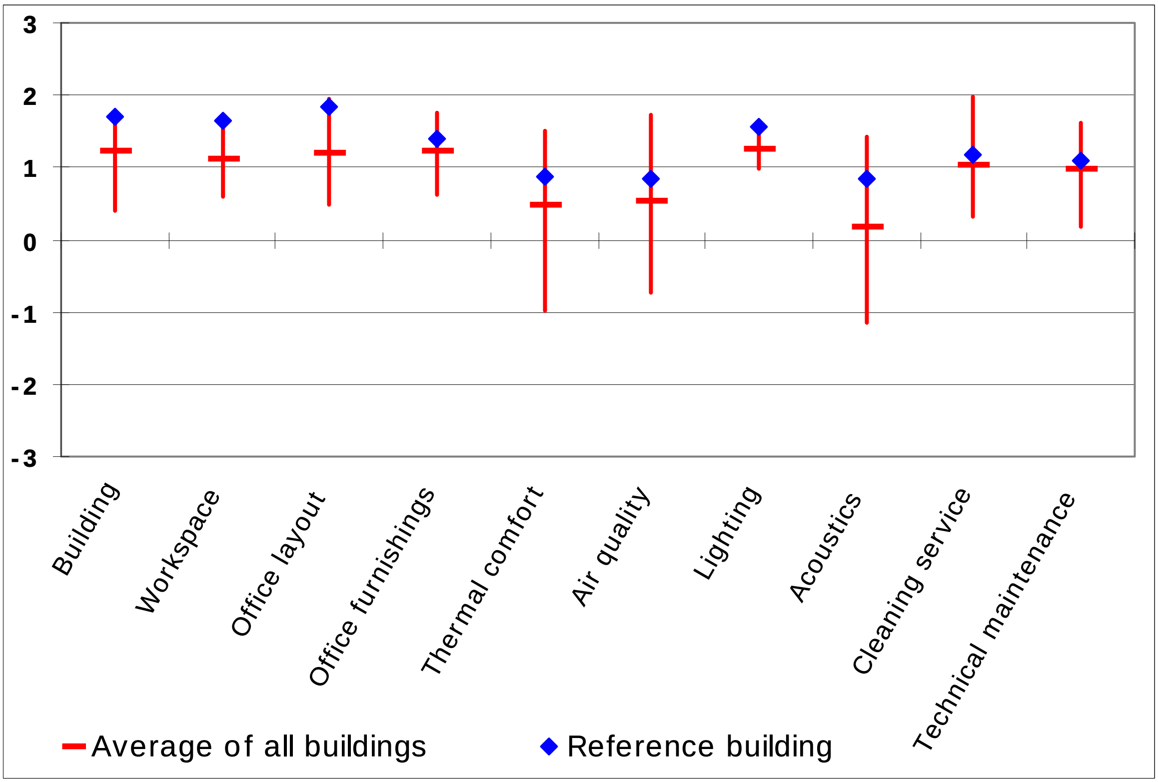

Figure 4. The average values and variations of each indoor environment area concerning 25 studied buildings.

In each building the benchmark values (Fig. 4) in association with occupant perception map (Table 2) lay the foundation for auditing and evaluating the indoor environment quality and for locating the possible technical problems.

DISCUSSION

The average value of occupant satisfaction survey (mean value of all indoor environment areas) can be used as a benchmark value when comparing workplaces to each other. It could also be the basis of indoor environment rating of buildings in the future and become a part of a holistic building labeling, where the mandatory part is energy labeling (based on Energy Performance of Buildings Directive). However, it is challenging to draw detailed conclusions of technical problems from the average occupant satisfaction only. There are several buildings, where serious problems related to indoor environment have been found in some parts of building even when the average satisfaction has shown excellent conditions.

The reason why office layout, office furnishings, lighting and cleaning get generally speaking higher scores than thermal comfort, air quality and acoustics can not be specified based on this material. Maybe people are more aware of thermal comfort, air quality and acoustics and that is why they are more critical about them. Or the other more likely option is, that the quality of thermal comfort, air quality and acoustics are poor in office environment. Technical problems found in all cases, where dissatisfaction is high, favors the latter assumption. However more research is needed in this area before any final conclusions can be drawn.

CONCLUSIONS

Occupants are an important source of information about indoor environment quality. Even though physical measures describe (numerically accurately) parts of the indoor environment only the perceived quality determines the functionality of the space to workers. People create the outcome of the work and therefore indoor environment's only purpose is to support the human performance and wellbeing. It is essential to measure the perception of people towards different indoor environment factors and in different parts of a building in order to find and correct the problems that affect human performance most. Even when the average satisfaction shows high scores there may exist major problems in some parts of the building. Occupant perception map is a beneficial tool to systematically solve the indoor environment related problems.

Many different vendors create thermal comfort, air quality and acoustics and therefore the end result is nobody's responsibility. There are no companies providing integration services to enable a satisfactory indoor environment within thermal conditions, air quality and acoustics. A more holistic approach of indoor environment development and maintenance is needed in order to provide satisfactory workplaces for people to perform at their best.

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Building Energy and Environmental Performance tool BEEP

Development of a method to compare the true energy efficiency of buildings

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SUMMARY

Regulatory devices for the energy efficiency of buildings currently in use, including the new EU "Directive on the Energy Performance of Buildings" [1] and especially the methods currently proposed in the various member states to determine and judge the energy performance of buildings as required by this directive deal only with energy demand and not with energy efficiency. This paper proposes a method which allows the true energy efficiency of a building design to be determined and thus a real comparison of various building design options. Energy efficiency is understood here as the relationship between the quality of the internal thermal environment in a building and the quantity of energy consumption required to maintain this environment. The proposed method takes into account the interrelationship between energy demand and internal environment and the calculated BEEP value is an indicator for total Building Energy and Environmental Performance.

INTRODUCTION

The true meaning of energy efficiency must take into account the internal environmental conditions as well as the energy demand required to maintain them. In fact, I propose that it is the relationship between the quality of the internal environment in a building and the quantity of energy consumption required to maintain this, which defines the energy efficiency of a building, at least in a thermal sense. The economic importance of the relationship between thermal comfort and productivity is becoming increasingly recognized [2]. The real challenge in energy efficiency is achieving a good indoor environment with a low energy demand. Prospective tenants and buyers of buildings should know what they are getting – a certificate with an energy demand rating means little if information about the quality of the associated internal environment is not made available. The current one-sided approach with concentration on energy consumption can lead to situations whereby seemingly high energy efficiency is only being achieved on paper. If the indoor environment is not acceptable, systems will be adjusted or new systems added to achieve a better environment at the cost of higher energy consumption. A method is needed which allows both energy demand and indoor environment to be quantitatively appraised to allow a real comparison between various options.

METHODS

The goal of the study was to develop a chart which allows the energy efficiency of various building designs to be plotted and thus compared. In order to measure the energy efficiency of the considered options it is necessary to relate the quality of the internal environment to the energy use necessary to maintain this. It is proposed here that the quality of the internal environment be indicated by the number of hours whereby comfortable conditions are not achieved; this in turn is measured by determining how many hours the "Predicted Percentage Dissatisfied" (PPD) is greater than 10%. PPD is the percentage of people likely to be dissatisfied with the thermal environment [3]. In the ISO 7730 a PPD value of less than approx. 10% is recommended [4].

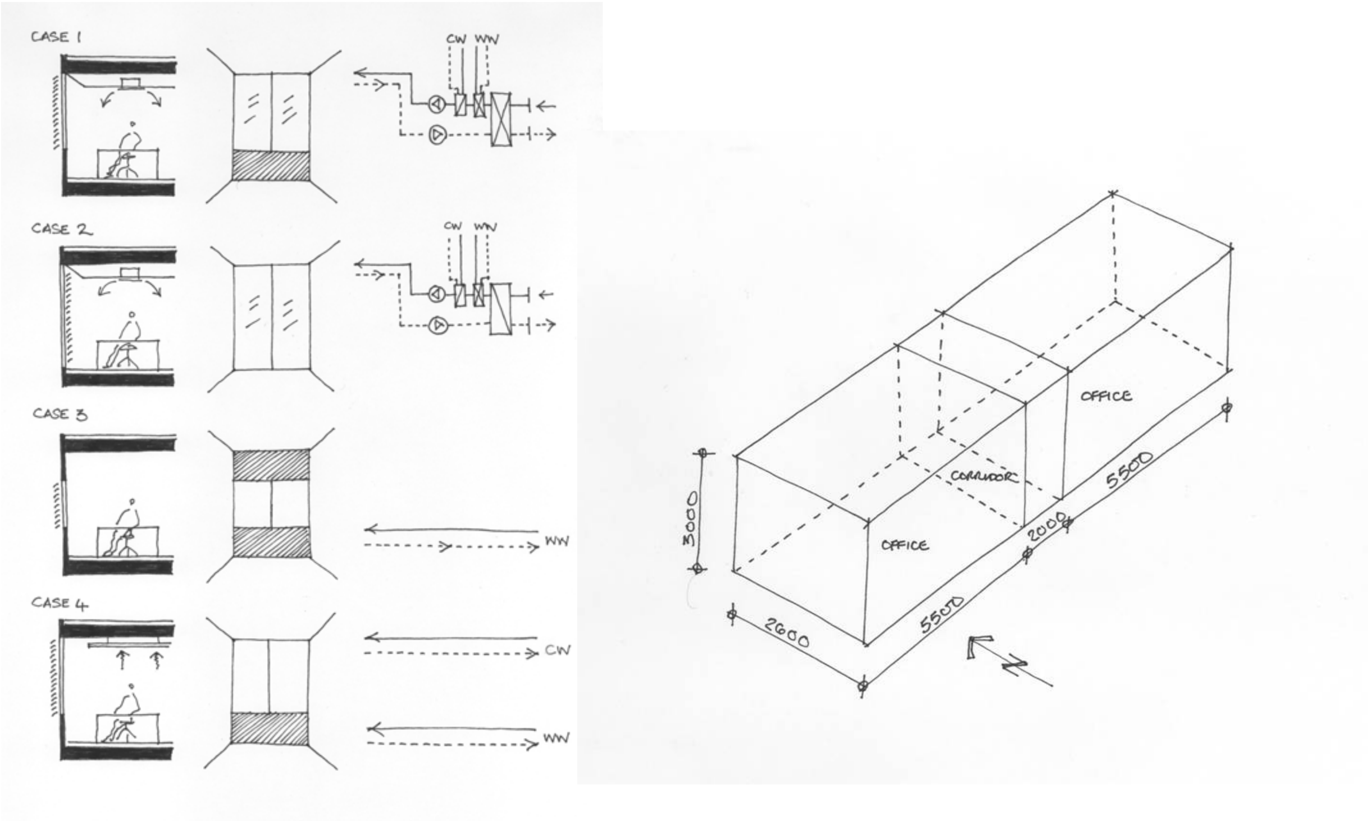

Figure 1. Design options and model geometry

In a second step four design options for a hypothetical office building were examined using dynamic thermal simulation and plotted on the chart as a means of testing the suitability of the proposed method. The options examined are various design alternatives for a hypothetical office building in Vienna city orientated with main facades east and west (see figure 1). The first design option has facades with approx. 70% window area and external blinds and is air-conditioned with non-operable windows. The second is fully glazed with a highly selective solar control glass and internal shading devices. It is also air-conditioned with non-operable windows. The third building has 40% window area, external blinds, operable windows and is heated only. Exposed concrete slabs and night time ventilation are used to limit summertime temperatures. The fourth building has façades with approx. 70% window area and external blinds, exposed concrete slabs, night time ventilation, comfort cooling and natural ventilation. For the thermal simulations a representative slice of the building as shown in figure 1 was used. The following assumptions were made:

- office hours from 9 am to 5 pm
- room setpoint in summer 24°C
- room setpoint in winter 22°C (reduced to 16°C at night)
- normal office internal loads (1 Person per 14 m², 15 W/m² machines, 15 W/m² lights)
- window frame 10%
- no humidification in winter
- supply air condition 20°C all-year round

- no ventilation in the corridor
- ventilation system in operation from 8 am to 6 pm
- nighttime ventilation from 12 pm to 6 am, May to September, assumed constant at 1.5 ac/h
- a constant air change rate of 2 ac/h is assumed for the natural ventilation options
- % window to wall is based on internal wall area (seen from office)
- internal walls are assumed adiabatic

To assume a constant air change rate of 2 ac/h for the natural ventilation options is a simplification and could potentially lead to overestimation of both heating demand in winter and summertime overheating. However cross-checking of the results with previous detailed studies into natural ventilation of offices [5] shows that the results correlate well with previous results based on more complex models. Due to the uncertainties associated with occupant behavior in naturally ventilated buildings it was felt that this approach was adequate for the present purpose. For details on the various design parameters used for the four options see table 1.

Table 1. Parameters

	Alternative 1	Alternative 2	Alternative 3	Alternative 4
Window Area as % of indoor wall area	73%	100%	40%	73%
Window U-Value	1.3	1.5	1.5	1.3
Glass g-Value	62%	36%	62%	62%
Glass light transmission	80%	66%	80%	80%
Operable windows	no	no	yes	yes
Solar Shading Position	external	internal	external	external
Solar Shading Control Value	200 W/m²	200 W/m²	200 W/m²	200 W/m²
Wall U-Value	0.3		0.5	0.3
Suspended ceiling	yes	yes	no	no
Cooling	yes	yes	no	yes
Ventilation	mechanical	mechanical	natural	natural
Air change rate	2.5 ac/h	2.5 ac/h	2 ac/h	2 ac/h
Dehumidification	yes	yes	no	no
Heat Recovery efficiency	70%	50%	none	none
Heating Unit Capacity (W/m²)	70	70	70	70
Cooling Unit Capacity (W/m²)	50	50	0	50
Nighttime ventilation	no	no	yes	yes

Heating and cooling loads were calculated using dynamic thermal simulation software. Fan energy was calculated based on 3 W per l/s of supply air, typical for conventional ventilation systems currently installed in European buildings. Lighting energy was estimated using a simple method developed by the author, which uses calculated daylight factors and annual external light availability data to roughly estimate the hours when electrical lighting can be expected to be in use. The primary energy demand thus calculated represents roughly 90% of the total primary energy demand of an average office building (excluding computers, office machines etc.). Items such as domestic hot water, pumps and lifts are not included.

RESULTS

Figure 2 shows the BEEP chart which was developed to allow the energy efficiency of various building designs to be plotted and thus compared. The x-coordinate of a given point represents the primary energy demand of the building design for heating, cooling, lighting and fans and the y-coordinate the percentage of occupied hours in a year with PPD > 10% as an indication of the comfort level achieved.

It is proposed that energy efficiency is expressed as the relationship of the number of hours with comfortable conditions to the primary energy demand necessary to maintain this condition. This is called the BEEP value and is calculated thus:

$$BEEP = \frac{NOH(100 - N)}{100PED} \qquad (1)$$

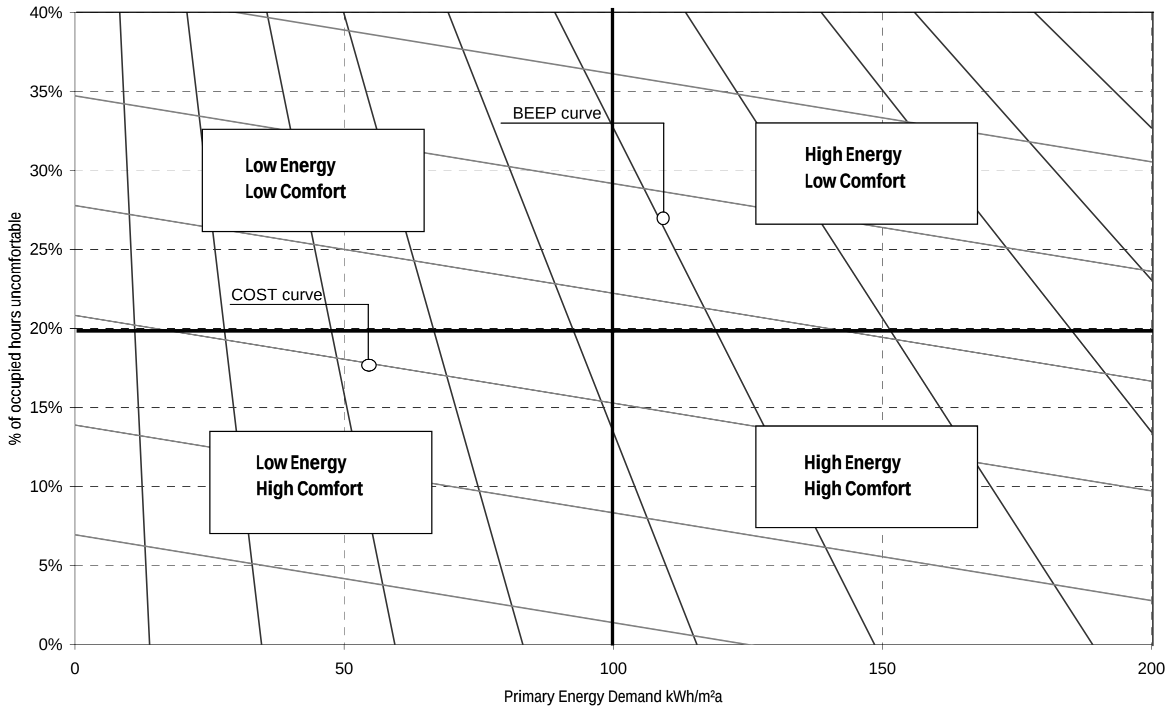

Figure 2. BEEP Chart

where BEEP is the Building Energy and Environmental Performance in hours comfortable per kWh/m²a, NOH is the number of occupied hours in a year, here assumed to be 2080, PED is the primary energy demand for heating, cooling, fans and lighting in kWh/m²a and N is the % of occupied hours with non-acceptable internal environmental conditions (PPD > 10%). To allow a comparison of the economic or financial implication of the considered options in operation it is proposed to sum the energy costs and the effect of the loss of productivity due to poor internal conditions and use this as an indicator of the economic cost of the various options.

$$COST = (EC)(PED) + (PL)\frac{(SC)(N)}{10000} \qquad (2)$$

where COST is measured in €, EC is the cost of a unit of primary energy in EUR/kWh, here assumed to be 0.04 €/kWh, SC are the staff costs and are assumed here to be EUR 7200/ m²a (€12 000 (Salary+overhead+profit) x 12 months / 20 m²) and PL is the assumed loss in productivity in % for the time when PPD is higher than 10%. It is assumed here that a 1% loss in productivity occurs whenever the PPD is greater than 10%. It should be noted that this is probably a gross underestimation of the effect of poor comfort on productivity. The assumption that the loss remains at 1% regardless of how high the PPD is, is also of course a simplification and probably an underestimation of the effect. Roelofsen states values of approx. 3% productivity loss at 10% PPD rising to nearly 20% at 60% PPD [6]. However for the purpose of providing an indicator such as proposed here for use in the comparison of options, it was felt that it may be better to initially underestimate this effect. As the results will show, even a constant value of 1% weighs in heavily when compared to energy costs. Further research should be undertaken into the application of the available data on productivity loss dependent on comfort in a method such as proposed here.

The resulting BEEP chart can be thought of as comprising four areas as shown in figure 2. The goal of building designers should of course be the bottom left area. Often decisions are between the bottom right and the top left areas. Obviously the top right area is to be avoided. Lines connecting BEEP points with equal values are called BEEP curves. Buildings which lie on the same BEEP curve may be said to be equally energy efficient. Lines connecting COST points with equal values are called COST curves. Buildings which lie on the same COST curves may be thought of as having similar economic implications in operation. The higher the BEEP value is, the higher the energy efficiency of the solution. A lower COST value means lower costs in operation. 100 kWh/m²a was taken as the boundary value between high and low energy solutions. This is the value used to define a low energy building in a governmental support program for energy efficient building in Germany [7]. A value of 20% of the occupied hours uncomfortable was chosen as the boundary value between high and low comfort. While this may seem high, one must remember that a large proportion of office buildings in Europe have no passive or active cooling and will thus be probably uncomfortable for large portions of the summer. Obviously these values are to some extent by nature arbitrary and will need to be adjusted after further research work. Table 2 shows the results of the dynamic thermal simulation of the four design options described above, including the BEEP values, which were calculated as described above. The primary energy value obtained from the simulation of the representational slice was multiplied by a factor of 0.85 to convert the value to kWh per m² total floor area of the building. This factor is based on typical values for the relationship between office and total floor areas and between energy demand in the office areas and in the other areas of the building.

Table 2. Results

	Energy Demand	**Comfort**		**BEEP**
	Primary Energy kWh/m²a	Hours PPD>10%	% of occupied hours	Value
Alternative 1	110	13.0	0.6%	19
Alternative 2	150	615.0	29.6%	10
Alternative 3	100	564.0	27.1%	15
Alternative 4	87	174.0	8.4%	22

Figure 3 shows the break-down of the primary energy demand for the various options. In figure 4 the four alternative solutions are plotted on the BEEP chart. When the alternative solutions are ranked in terms of both BEEP value and COST values the results are surprising when compared with a ranking done on intuition. Interestingly, alternative 1 can be seen to have a higher BEEP value and therefore a higher total energy and environmental performance than alternative 3, which may not have been apparent before being plotted on the BEEP chart. From an economic point of view alternative 1 can be seen to be by far the most efficient, also not immediately apparent before plotting on the BEEP chart. Before finalizing a decision, capital costs and the costs of system maintenance etc. would of course need to be considered. Alternative 2 is the worst type of solution; high energy and low comfort. In terms of costs in operation alternative 2 and 3 are similar, also an unexpected result. Of course the capital costs associated with alternative 2 will be higher. Alternative 4 is a low energy high comfort solution and has the highest BEEP value. Considering that alternative 1 will also be more expensive in terms of initial costs than alternative 4 and that the absolute energy demand of alternative 4 is considerably lower than alternative 1, alternative 4 may be judged to be the best solution. The relatively large percentage of time however, whereby the internal environment is not comfortable, needs to be considered. On the other hand, many solutions exist for low energy high comfort buildings which may be expected to perform considerably better than alternative 4. Note also that if natural ventilation was reduced in winter and increased in summer the comfort performance of alternative 4 (and alternative 3) could be expected to improve (see assumption above for natural ventilation). In other words there is potential with alternatives 3 and 4 to improve comfort and energy performance by optimizing natural ventilation strategies. Note that whilst the energy demand from the alternatives 3 and 4 are similar, the total performance of alternative 4 as measured by the BEEP value is significantly higher.

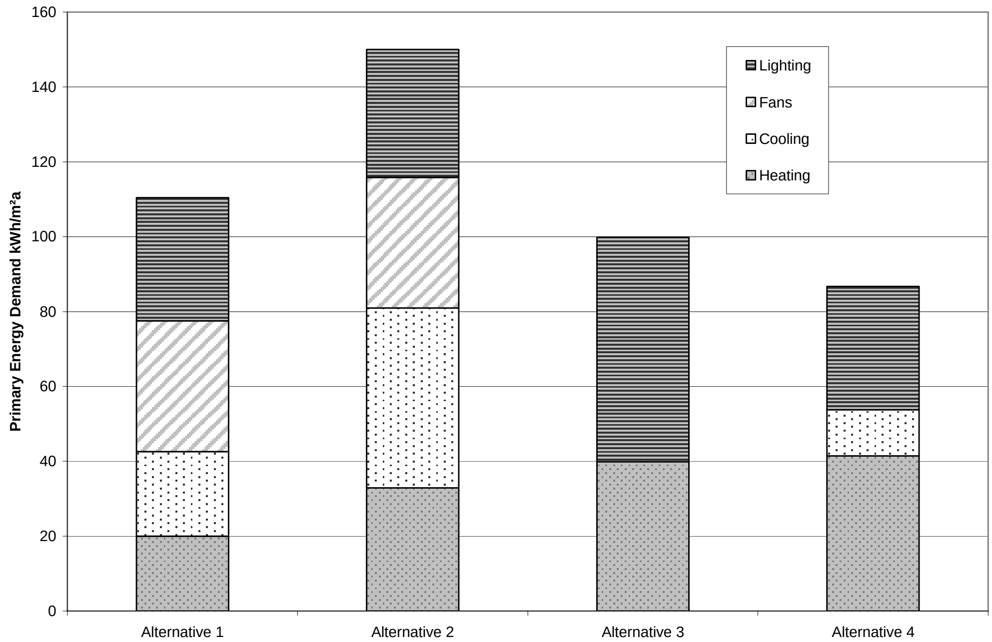

Figure 3. Primary energy demand of the various options

An interesting question poses itself with regard to alternative 3. What would happen, if the building was upgraded at a later stage with a cooling system to improve comfort? This option was also simulated and the result is shown in figure 4 as point 3´. The cooling system is the same as that used in alternative 4. It can be observed that although comfort is significantly improved and the BEEP value is also improved (slope of the improvement curve is steeper than the BEEP curves), the total energy and environmental performance is still less than for alternatives 1 and 4. The energy demand of the upgraded alternative 3 is very similar to 1, yet the attained comfort level significantly lower. This result demonstrates that the BEEP value in such a case should be calculated for both the present condition and the condition after an upgrading to include systems which provide a certain minimum level of comfort.

DISCUSSION AND CONCLUSIONS

The method presented here outlines an approach which could form the basis on which further work could be carried out with the ultimate goal of developing a comprehensive way of comparing the energy efficiency of building design options. The examples here were used as a means of initially testing the method. The results obtained are not intended in any way to provide conclusions on the appropriateness of the various design solutions studied. The results do indicate however that we need to compare options comprehensively and understand energy efficiency not as energy use alone but as the relationship between energy use and value in terms of the quality of the internal climate achieved. This should be part of a total approach in which capital and running costs, functionality and architectural quality etc. of the various options are also compared with one another. The economic curves are primarily displayed to indicate tendencies and to show the vast difference between the energy and economic efficiencies (demonstrated by the different slope of the curves). An increase in energy cost would change the slope of the economic curve and it is interesting to note, that based on the data considered here, the energy price would need to increase by a factor of approx. 15 before the slopes would roughly coincide and decision making based on total energy and environmental performance and decision making based on cost would lead to the same solution. It must be remembered that capital costs have not been factored-in. The goal of any design should be a high BEEP value positioned in the bottom left area. Note that the higher the BEEP number, the steeper the curve; i.e. large differences in comfort and small differences in energy demand.

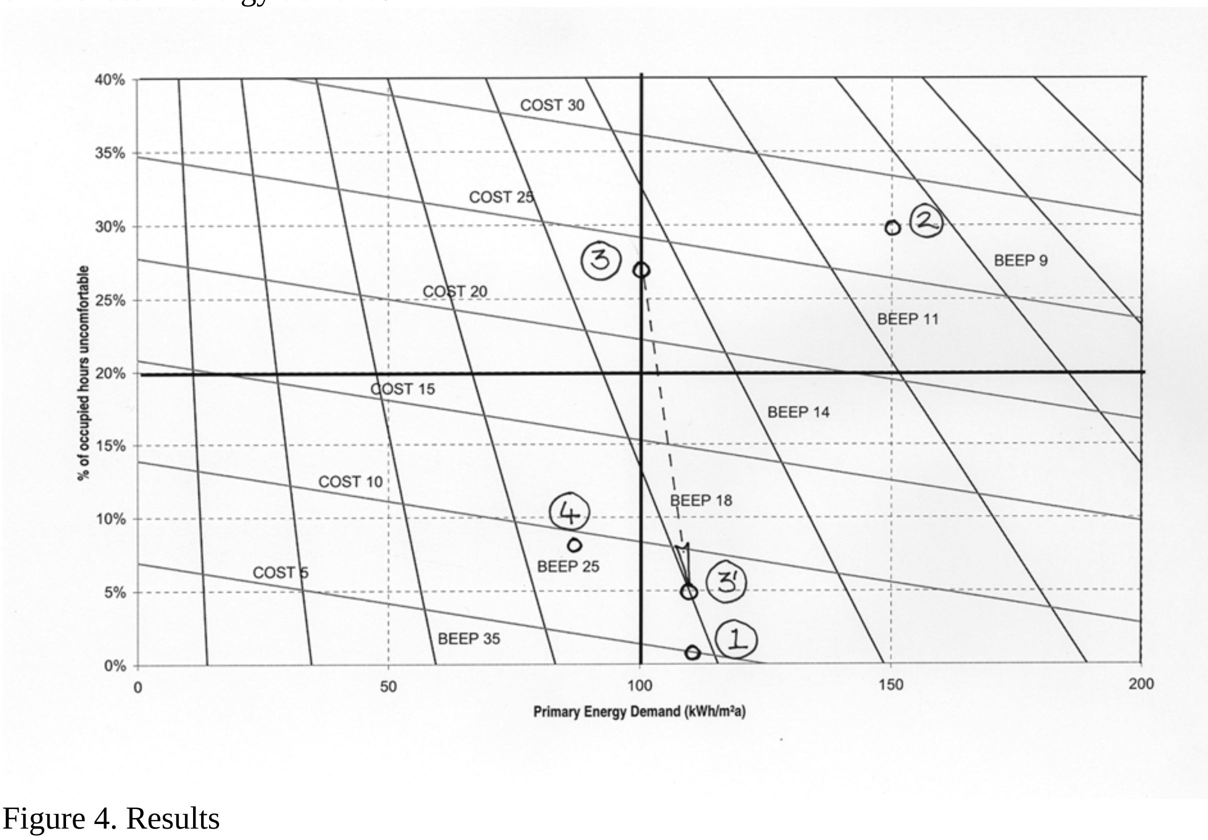

Figure 4. Results

The proposed tool could be developed to be put in use not only as a means of demonstrating the energy efficiency of building designs at the planning permission stage (similar to the energy certificate as required by the new EU directive) but also as a design tool to compare various options during the design stage. Items such as energy production (photovoltaic, wind energy, solar cooling) and system configurations such as underground fresh air ducts are not considered in the illustrated case studies but could and should be included in the final planning instrument. A more complex instrument can be imagined whereby the x-axis is represented by total primary energy demand over the total life cycle of the building (including embodied energy etc.) with the y-axis representing the total internal environment achieved including factors such as air quality, lighting levels and acoustics but also psychological issues such as daylight, operable windows etc. The capital costs could be factored into the economic curves.

Contemporary engineering design firms use dynamic simulation to design buildings as standard procedure. It seems questionable, whether the right approach is to develop spreadsheets for energy calculations for the production of energy certificates, which is what is happening all over Europe right now, instead of accrediting commercial dynamic simulation software programs. Further research is necessary to determine the range of appropriate values in the BEEP chart and to produce BEEP charts for different applications. A series of charts could be produced for different building uses (office, apartments etc.) for various climatic regions. Further research should also look at ways to measure comfort which would be appropriate to be used in such an approach. The simulation software used in the study here calculates the PPD value for the centre point of the room. It should be investigated further whether this relatively primitive indicator is adequate for the intended purpose or alternatively whether a better one could be employed. Further research on the exact nature of the relationship between PPD and performance would also be valuable. Future work could also improve the complexity of the natural ventilation model (see above). Note also that research has shown that comfort perceptions in naturally and mechanically ventilated buildings are different [8]. This aspect has not been considered yet in the approach used here.

The upgrading of alternative 3 with cooling shows the danger of the current methods concentrating only on energy demand; a seemingly low energy building may after upgrading to rectify comfort problems be less energy efficient than a relatively conventional fully air conditioned building. In terms of conserving energy or possibly even from an ecological point of view, concentrating on reducing energy demand is possibly a legitimate approach but is it really sustainable? Achieving sustainability is complex and consideration of the economic and social aspects may mean that conserving energy at the expense of a lower quality of internal environment is not the most sustainable approach.

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The AUDITAC Customer Advising Tool (CAT) to assist the Inspection and Audit of Air Conditioning Systems in Buildings

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SUMMARY

This paper describes the production of the Customer Advising Tool (CAT), a piece of software to help assess the potential for reducing the cooling demand of Office buildings, as required by Article 9 of the EPBD. It is a practical tool aimed at use by building owners, inspectors and auditors as part of the Inspection and Audit process. The tool was developed by the authors as part of the IEE AUDITAC project.

The tool is based on a mixture of monitored and modelled data, and this paper presents the tool's development, concluding with how the inputs and outputs are simplified down to those required to provide a focus to the on-site inspection and audit.

It is anticipated that many Member States will refer to the tool as part of their support for implementing Article 9. The tool can be downloaded from, or used at, the following website: http://www.cardiff.ac.uk/archi/research/auditac/advice_tool.html

INTRODUCTION

Article 9 of the European Energy Performance of Buildings Directive (EPBD) is concerned with reducing the energy consumption of Air Conditioning systems in buildings. The second paragraph of Article 9 states *"This inspection shall include an assessment of the air-conditioning efficiency and the sizing compared to the cooling requirements of the building. Appropriate advice shall be provided to the users on possible improvement or replacement of the air-conditioning system and on alternative solutions"*.

The practical reality for the auditor/assessor in meeting Article 9 is that they have a limited timescale in which to assess/estimate the performance of both a building and its A/C systems. Within this timescale it is likely that the assessment of the A/C systems will occupy the majority of the time. A quick and easy method of assessing the effect that the building and its usage is having on the A/C system cooling demand is therefore required to minimise the time and effort spent on this task.

The purpose of this paper is to provide details of the recently published AUDITAC Customer Advising Tool (CAT) [1]. This tool has been designed to provide information, in a form accessible to a building owner or assessor, on which aspects of the Building Fabric and Operation are likely to be having the largest effect on the cooling energy demand imposed on the building's Air Conditioning systems, and therefore which areas are most likely to provide the greatest potential for reducing this demand. The tool also provides an indication of the percentage change in cooling demand that would occur through variation of aspects of the building design or operation. It is important to note that the tool is a **holistic** design tool, i.e. it considers the overall cooling demand resulting from the **interactions** between all the criteria and parameters specified.

It is important to note that the CAT has been derived using data from Offices, but it is believed that the basic findings should be generally applicable to any situation that matches those described in the CAT input sections.

The nature of cooling demands in buildings
Modeling, monitoring and other studies on cooling in buildings and building services by the authors [2,3,4,5,6,7,8,9,10,11,12,13,14,15,16,17] and other researchers, have shown that those factors which have a significant effect on a building's heating and cooling demands are relatively few.

The heat transfer mechanisms within buildings are the familiar ones of radiation, convection and conduction. These mechanisms will transfer energy into or out of a building depending on the following main factors:

- Time of day, week, year.
- Ambient weather conditions.
- Design of the building, its services, and its location with respect to overshadowing, exposure, etc.
- Building use, i.e. occupancy type, hours of use, layout.
- Internal conditions provided in the building.
- Effectiveness of building services controls

The CAT considers these aspects in terms of their effects on the cooling demand of buildings. The data presented by the CAT is obtained purely from modeling, but the results have been compared with previous measurements and monitoring to provide a few points of reference. The CAT presents the findings from the work in a manner that is of use to the assessor in meeting the requirements of Article 9.

The design of the CAT uses the extensive experience of the authors in assessing the energy performance of buildings and A/C systems to minimise the information an assessor needs to obtain to use the tool. It does this by asking only for information that can be used to estimate, and make improvements to, the building cooling demand.

Within the context of the CAT, the factors noted above which affect the cooling and heating demand in buildings can therefore be simplified into various criteria and parameters that the assessor can establish for each building.

Not all of the criteria and parameters identified as important were able to be assessed through modeling, primarily due to time and resource constraints. Details are given later in this paper. The other limitation of the CAT is that it has not been possible to model all the possible permutations of the criteria and parameters. This is again due to the impossibility of achieving this in the time with the computing resources available. Therefore the CAT modeling has used a technique of varying some of the individual **parameters** around an **average** value that is considered by the authors to represent the best compromise of all the possibilities available for that parameter. Nearly all the possible permutations of the **criteria** assessed have been modeled however.

Use of the Customer Advising Tool

The authors consider that the CAT will be used primarily at the pre-audit stage of an inspection. At this stage the CAT can usefully identify the main aspects of the building and its operation which might be further studied onsite by the assessor in their search for potential means in reducing the building cooling demand.

While the CAT cannot consider the detail of the individual buildings to be inspected, its ability to holistically consider the interaction of some of the main contributors to building cooling demands is a major assistance in writing Inspection Reports.

DETAILS OF THE PARAMETERS AND CRITERIA USED IN THE CAT MODELING

The basis of the CAT modeling is an open-plan cube of dimensions 10m x 10m x 2.4m. This cube is shown in Figure 1 as part of a 9 cube square. Once each layout is generated, inter-zonal adjacencies between each cell are calculated, making it possible to discern for each side whether it is adjacent to another cell or exposed to external conditions. Windows of the appropriate size are then automatically inserted into each exposed facade element.

The apparently low ceiling height reflects the authors' desire to approximately model the effect of occupancy on the reduction of internal volume in the space (taken as about 20%), and hence the reduction in ventilation air needed to achieve the required air change rate.

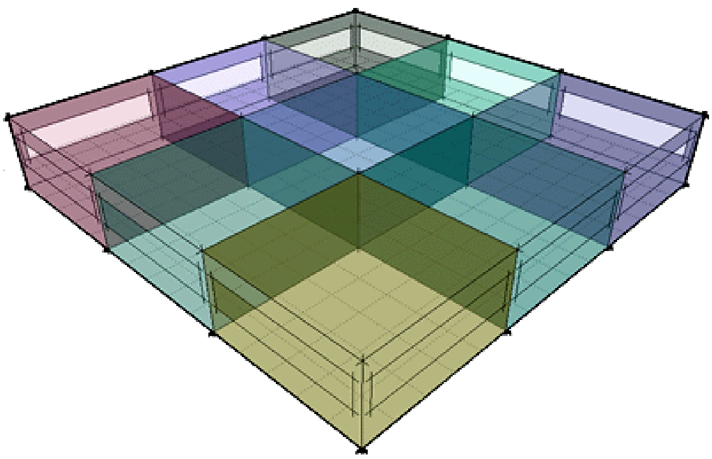


Figure 1. Basic cube model, shown as the deep plan 30m x 30m layout.

The various combinations of the basic cubes and their properties were modeled in the ECOTECT software programme [18] and exported to the EnergyPlus building energy modeling software [19] to be run with the various combinations of criteria and parameters required to meet the purposes of the CAT.

Descriptions of the individual criteria and parameters modeled which are applied to the permutations of this basic cell model are discussed in more detail below, along with the ranges over which each criteria or parameter was modeled. The descriptions also consider how an assessor might go about classifying each for a building.

CAT criteria

For the CAT, **criteria** are deemed to be those elements of the building design or operation that are known to have an influence on the heating and cooling demands, but are realistically unlikely to be altered simply to try and reduce heating and cooling demands.

***Plan depth*: The ratio of floor area to total external facade area.** A shallow plan will normally allow daylight and natural ventilation to be used over the entire floor area. A deep plan will usually be fully mechanically ventilated. The choices available and modeled for this criterion are shallow plan (10m x 10m); medium plan (20m x 20m); and deep plan (30m x 30m)

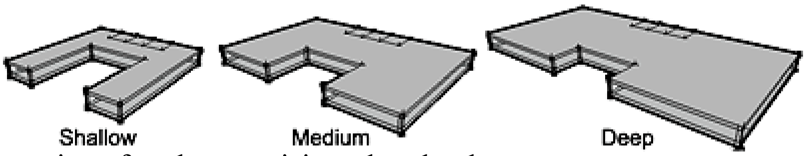


Figure 2 - Three options for characterising plan depth.

Glazing ratio: The glazing ratio represents the percentage of the exposed external vertical facade area of the building that is considered to be a solar aperture. This is typically a glazed area having some transparency to both visible light and solar radiation. Variations in glazing ratio were modeled using windows of the same sill and height but with different widths relative to each wall surface such that the ratio of glazed area to total vertical façade area represented 20%, 40%, 60%, 80% and 90%.

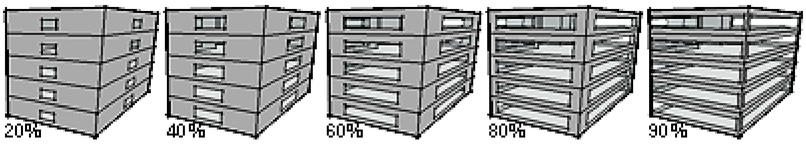
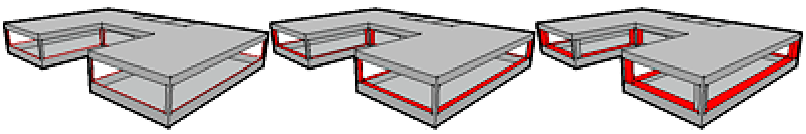


Figure 3 - Values for characterising facade glazing ratio.

Thermal mass: In order to consider the effect of thermal inertia on building performance, it is necessary to define this criterion which is based on the amount of internally exposed thermal mass. A building with significant amounts of masonry or concrete visible would usually be classed as heavyweight, whilst a building which is predominantly suspended ceilings and lightweight partition walls would usually be classed as lightweight. The choices available and modeled for this criterion are lightweight, mediumweight and heavyweight.



Lightweight *Medium weight* *Heavyweight*

Figure 4 - Three levels of internally exposed thermal mass**.**

Weather locations: The city location/region modeled by the CAT which is most likely to reflect the weather conditions to which the audit building will be subjected. Regional variations in climate were considered by using average annual weather data sets for a range of locations throughout Europe in the calculations. The choices available and modeled for this criterion are Vienna (Austria), Berlin (Germany), Madrid (Spain), Paris (France), London (UK), Athens (Greece), Rome (Italy), Lisbon (Portugal) and Stockholm (Sweden). These were modeled to allow a choice of climatic zones.

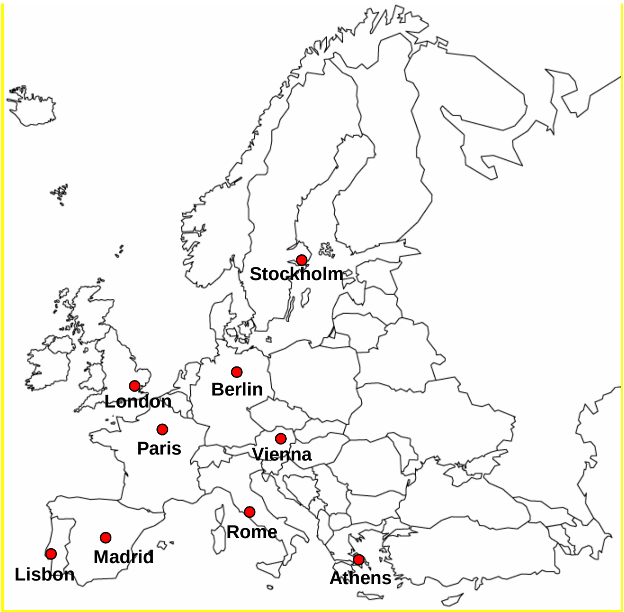


Figure 5 – Map of European weather locations modeled in the CAT.

***Internal layout*:** This reflects whether the building is predominantly cellular offices or open plan. *The CAT currently considers the building to be comprised of 10m x 10m open-plan cells as noted earlier. These options are not available in the model yet.*

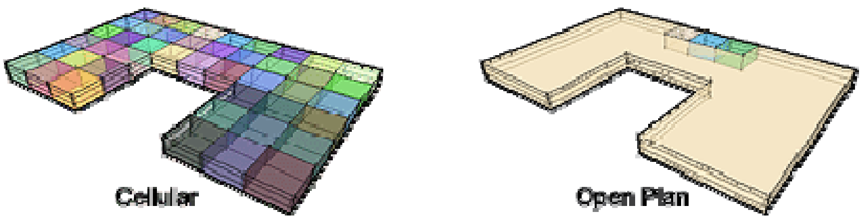


Figure 6 – The two extremes of internal layout.

***Building form*:** This reflects the effect of building form on issues such as self-shading and ventilation rates. *Only the square form has been modeled for this criterion.*

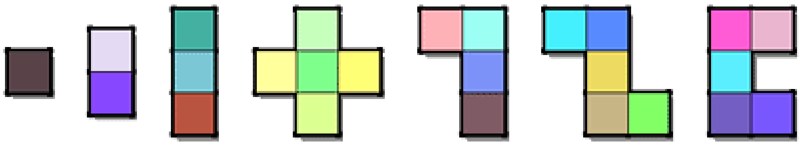


Figure 7 – Examples of simple cell layouts.

***Multiple storeys*:** Due to the number of runs involved, work to include the effects of multiple storeys has not yet been included in the CAT. However, the strategy to do so has been carefully considered. To represent buildings with any number of storeys, it is necessary to simulate ground floors (which may be underground or in direct contact with foundations) separate from top floors (which may have an exposed roof or be in direct with a ventilated plant room directly above) separately from mid-level floors (for which thermal transfer through the floors and ceilings is likely to be negligible).

Moreover, self shading of windows on lower floors may be quite pronounced in some of the plan layouts making it necessary to model the floors above. Whilst it would be possible to generate and test a series of multi-storey versions of each plan layout, as shown in Figure 8, the resulting number of runs becomes prohibitive.

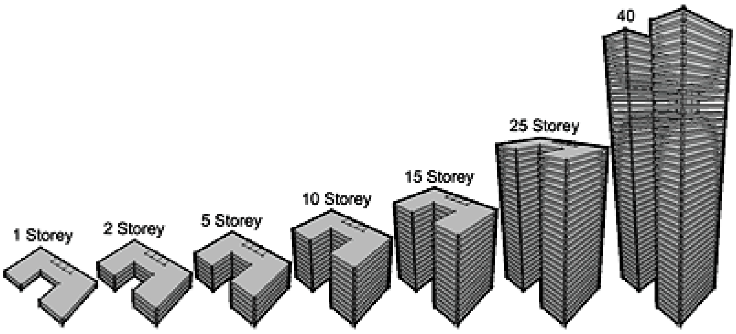
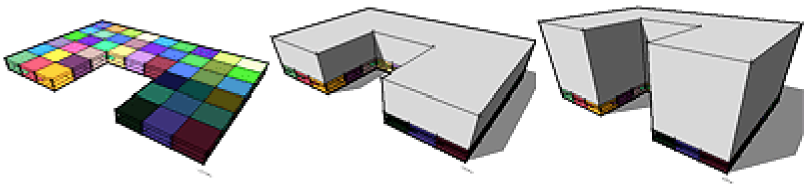


Figure 8 - Simulations should allow choice of any number of storeys.

A proposed solution was to generate 3 models for each plan layout. The first, representing a top storey, would be assigned an adiabatic floor (to simulate insignificant heat flow to the zones below) and an exposed roof. The second, representing a mid-level storey, would be assigned both an adiabatic floor and an adiabatic ceiling (to simulate insignificant heat flow to zones both above and below) with an additional non-thermal zone generated above to simulate any self-shading effects. The third model, representing the ground storey, would be assigned a ground-connected floor and an adiabatic ceiling, as well as a much larger non-thermal zone above. The different models are illustrated in Figure 9 below.



Top floor *Mid-level floor* *Ground-floor*

Figure 9 - The three models required to simulate multi-storeys

This way, by simply performing three runs, it is possible to represent any number of storeys by adding the energy demand of the ground and top floors to a number of mid-level floors equal to [storeys-2]. This assumes that the self shading effect on the mid-level floor represents the average over all floors. Figure 10 illustrates the theoretical assembly of this building.

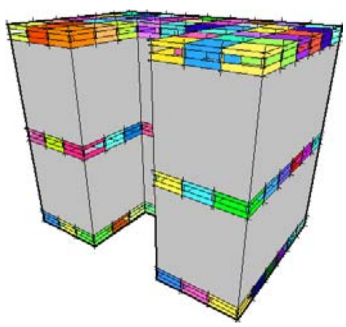


Figure 10 - Assembling a multi-storey building from the 3 models.

Single storey buildings, with both an exposed ceiling and a ground-connected floor must be represented by an additional and separate fourth run based on a model with both a ground-connected floor and an exposed ceiling. *Only this single storey building has been modeled for this criterion in the CAT to date.*

CAT Parameters

Parameters are defined as those elements of the building design or operation that are known to have an influence on the heating and cooling demands, and are potentially able to be altered by the assessor or building owner.

***Fabric U-value*:** This refers to the overall surface-averaged thermal conductance of the opaque elements in the external building envelope. This includes the roof as well as facades. The effects of this parameter have been modeled between 0.1 and 4.0 W/m^{2}K. The average value used for this parameter is 0.5 W/m^{2}K.

***Window U-value*:** This refers to the overall surface-averaged thermal conductance of the windows and transparent elements in the external building envelope. The effects of this parameter have been modeled between 0.5 and 6.0 W/m^{2}K, where the most insulating transparent element available would have a U-value of around 0.5 and single glazing would have a U-value of around 6.0. The average value used for this parameter is 3.0 W/m^{2}K.

***Solar Heat Gain Coefficient (SHGC)*:** This is the fraction of incident beam (direct) solar radiation that enters the building through the transparent elements, such as windows. This includes the transmitted solar radiation and the inward flowing heat from the solar radiation that is absorbed by the glazing. The effects of this parameter have been modeled for SHGC's ranging from 0.1 to 0.9, where 0.1 is highly shaded, (i.e. little solar gain), and 0.9 means no effective shading. The reduction from 1.0 is simply due to the normal properties of glass. The average value used for this parameter is 0.5.

***Air change rate (infiltration only)*:** This refers to the uncontrolled exchange of air between internal spaces and outside air due to gaps in the building fabric linked to design details. This value is given as the number of complete air changes per hour. The effects of this parameter have been modeled between 0.1 and 4.0 ac/hr, where a well detailed, well-sealed building would be around 0.1 and a very leaky, shallow plan building in an exposed and windy location would be around 4.0 ac/hr. The average value used for this parameter is 0.5 ac/hr.

The tool does not specifically include a parameter for varying the **ventilation** air change rates, as it was deemed that this would be set by the building owner to meet the ventilation and cooling demands of the building. However, in tightly sealed buildings, the effects of modifying the ventilation rate can be approximated by adding the infiltration and ventilation rates together. This sensitivity analysis could be useful if a change from using air as the heating and cooling transport medium to recirculation A/C systems with separate fresh air supply were being considered.

***Internal Gains (W/m^2)*:** This refers to the contribution of small power, lighting and people loads per metre squared floor area. The effects of this parameter have been modeled between 10 and 160 W/m^2, where 10 W/m^2 represents a very low density of occupation and equipment use, and 160 W/m^2 would represent a very high density of equipment and occupancy, such as a business call centre. The average value used for this parameter is 40 W/m^2.

***Hours of use of the heating and cooling systems*:** This parameter refers to the effect of running the systems continuously or over the working week only (assumed 08:00 to 18:00 in this system). *The effects of optimum start/stop have not been assessed.*

***Variation in heating and cooling setpoints*:** The CAT assumes a heating setpoint of 21°C and allows the cooling setpoint to be varied between 21 to 25°C in 1°C intervals. *It does not allow the effect of varying the heating setpoints to be assessed.*

***Variation in occupancy, equipment and lighting schedules*:** This parameter refers to the effect of intermittent or continuous occupation, equipment or lighting schedules. The CAT assumes an 08:00 to 18:00 schedule for all these parameters including the HVAC equipment. *The effects of various combinations of continuous and intermittent schedules of these parameters have not been assessed.*

***Spatial distribution of cooling loads*:** This reflects the possibility of moving the cooling loads within the space to minimize the cooling demands on the A/C system. *Only evenly distributed loads are currently considered by the CAT.*

System Parameters

In order to accurately compare heating and cooling energy, as well as calculating potential carbon emissions savings, the relative efficiency of the plant and equipment used to supply the building's heating and cooling demand must be specified or estimated. These values are usually very difficult to obtain with any accuracy so it is necessary to test the likely range in the building being assessed to estimate the sensitivity of any recommendations to reasonable variations in these figures. These are defined in this work as *system parameters* and include the following for both the heating and cooling systems if present in the building.

Heating System Efficiency (SCOP)

This defines the Seasonal Coefficient Of Performance (SCOP) for the equipment used to provide space heating within the building. This is given as the annual average ratio of total heat output compared to total energy input to the equipment. A good condensing boiler based system would expect to achieve a SCOP of around 0.9, whereas an old cast-iron boiler based system might struggle to reach 0.6.

***Cooling System Overall Efficiency*:**

This is the overall annual ratio of the cooling energy demand met by the installed cooling system, divided by the total energy input to the cooling system. It includes auxiliary energy consumption as well as the chiller energy consumption. It is important to note that this is NOT the Energy Efficiency Ratio (EER) for the equipment used to provide comfort cooling within the building. A low rating (0.3 - 0.6) would apply to an inefficient system, and a high rating (>2.0) to an efficient system. An average performance is likely to be 1.0 - 1.5.

Countries with high cooling demands are likely to see better overall efficiencies than countries with low cooling demands, as the chiller energy demand in hot countries will be a greater proportion of the overall cooling system energy consumption.

Fuel Carbon Emission Factor

The amount of carbon dioxide emitted through the consumption of 1 kilo-Watt hour of fuel used. This is given as a value between 0.0 for totally renewable energy and 0.43 kgCO2/kWh for delivered grid electricity - although allowance is made in the CAT for values up to 0.60.

For countries other than the UK the appropriate Grid delivered electricity factor should be used. Refer to the International Energy Agency (http://www.iea.org) for further information.

DISCUSSION

Interpreting the outputs from the cat

Once the building assessor has entered values for each of the above criteria and parameters (or accepted the default values offered) the CAT then searches its database to find the output graphs that match these values.

Figure 11 shows an example output graph.

Output graphs present the PERCENTAGE change in heating and cooling demands that would occur for a given change in an individual parameter, along with an estimated overall change in carbon emissions. The value assigned to each parameter is shown by the vertical green line on each graph, and the percentage changes in the Y-axis are relative to this value.

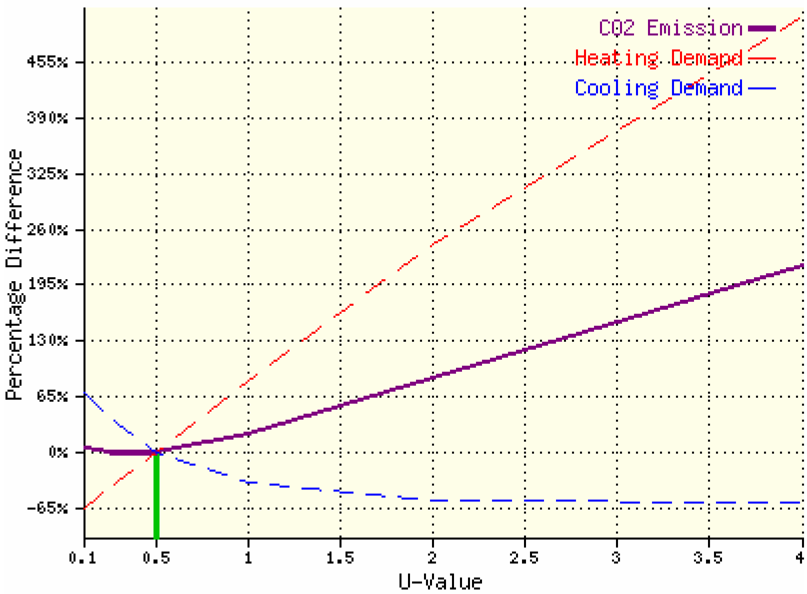


Figure 11. Example output graph showing the percentage change in heating and cooling demand achievable by varying the building fabric U-value from the input value (0.5).

A positive percentage increase shows an increasing demand for heating or cooling, and increasing carbon emissions. A negative percentage shows a reduction in these values.

The carbon emissions line is used as a proxy for assessing whether the proposed parameter changes might adversely affect the heating demands. The carbon emissions estimates are derived using the heating and cooling system parameters entered on the input screens, so it is important these are as accurate as possible, but they are still only estimates as the values entered will vary with time of year and actual demands on the systems.

In this example the graph shows that from a carbon viewpoint the current fabric U-value is probably the optimum for the combination of criteria and parameters chosen. Had we increased the U-value to 2.0 then from the graph we would have potentially achieved around a 60% reduction in the cooling demand, but the heating demand would have risen 250% and the overall carbon emissions around 100%.

The assessor would therefore not need to consider this as a potential area for improvement in the on site survey. This graph also shows the importance of considering the building and occupancy holistically. Had the heating impact not been considered then it would have appeared that we would need a less insulating U-value than currently installed.

The output graphs are viewable as both annual and monthly graphs for each parameter, so that the seasonal variation of the loads due to a given parameter can be assessed. At present the CAT does not allow the assessment of dynamic parameter changes in one run, e.g. using moveable shading to alter the SHGC only through the months requiring cooling. However the assessor can run a series of different parameter values which would give him the same information.

CONCLUSION

This paper has provided an overview of the derivation of the CAT, a tool which shows how a building's basic design and operation might affect the cooling demands placed on an A/C system used in the building, and how these demands might potentially be reduced through altering aspects of the buildings design or operational parameters.

ACKNOWLEDGMENTS

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The sole responsibility for the content of this paper lies with the authors. It does not represent the opinion of the European Communities. The European Commission is not responsible for any use that may be made of the information contained therein.

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MINERGIE-P® – A Building Standard of the Future

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SUMMARY

Already 20 years ago, Gro Harlem Brundlandt stated «I believe the time has come for higher expectations, […] for an increased political will to address our common future» ([1], p. 13). Indeed reading the newspapers one could get the notion that finally politicians (at least in Europe) are re-thinking those lines.
This paper presents the low-energy label for buildings MINERGIE-P® launched in Switzerland end of 2002. MINERGIE-P® demands a decrease of the heat demand by 80%. Solutions are shown of how to comply easily with these demands that appear severe on first sight. Furthermore the historical and political necessity to act is outlined and put into perspective.
Of utmost importance regarding MINERGIE-P® is not only the massively decreased heat demand. The used technologies and appliances furthermore guarantee a high indoor air quality and the long-term conservation of building value. This is easily communicated and in the meanwhile broadly accepted by banks offering better conditions for real estate mortgages.

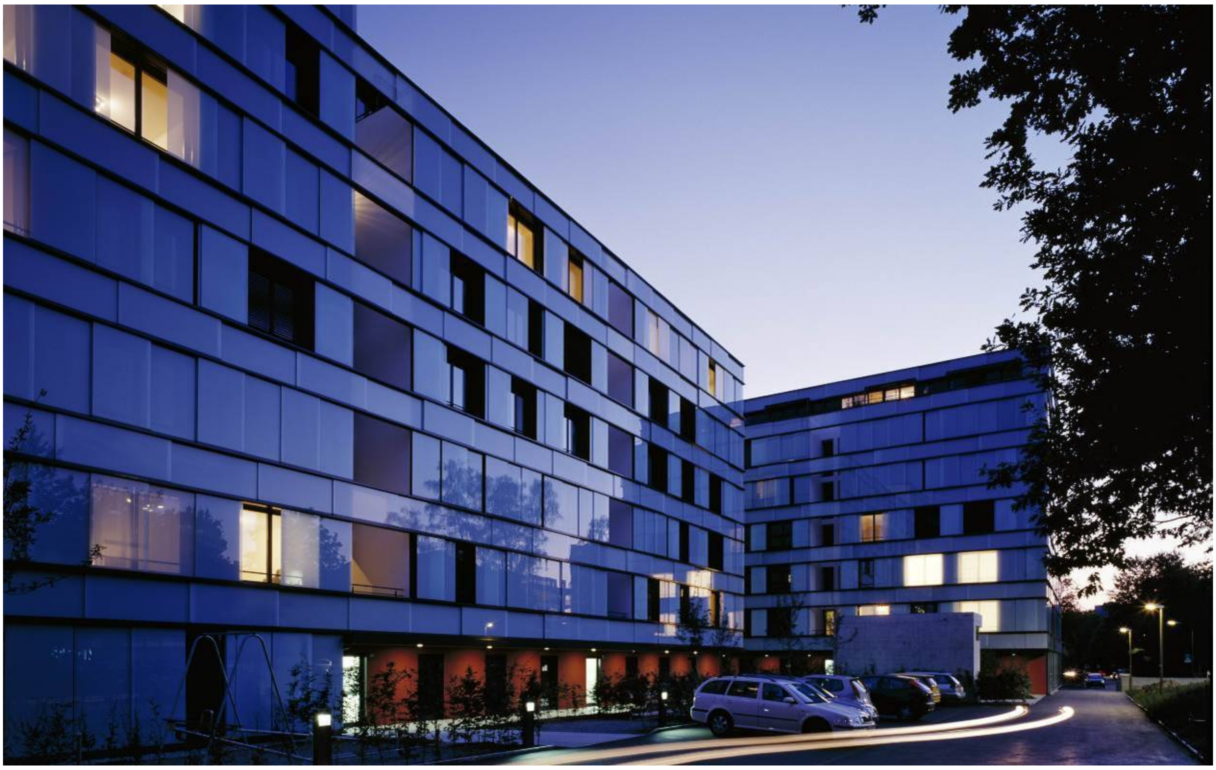

Figure 1. Multiple-family dwelling offering 90 apartments close to Zurich designed to comply with the demands of MINERGIE-P®. (Picture courtesy of Senn BPM, Switzerland)

INTRODUCTION

At the moment of writing, energy consumption and cutting down on CO_2 output are highly pressing matters (and hotly debated at least in the European Community). One of the central points to get the matters at hand under control is of course the augmentation of today's energy efficiency. This is especially true for buildings. Nowadays it is possible using existing technology to reduce consumption for heating by 80%.

In Switzerland energy calculations are standardised since 1988. More and more the legally allowed energy consumption is reduced according to the energy label MINERGIE and the still more rigorous label MINERGIE-P (harmonised with the German *Passivhaus* of Dr. Wolfgang Feist). In the paper presented the historical and political background of MINERGIE-P as well as the demands imposed by MINERGIE-P are illuminated. The paper presents both precise values of energy consumption and the consequences for low-energy buildings.

MATERIALS

Not only the European Performance of Buildings Directive (EPBD) postulates steps towards a more efficient way of handling our resources. NOVATLANTIS in cooperation with the SIA (Swiss Society of Engineers and Architects) and the BfE (Swiss Federal Office of Energy) map out in [3] the need to realise the so-called "2000-Watt society". The thought which stands behind this programme is the realisation that the worldwide average energy consumption in 1990 equalled 17,5MWh/y which amounts to a constant power consumption of 2000W per person. As could be shown in [4], 2000W in the long run guarantee a sustainable development covering all three aspects of sustainability. The authors postulate a "window of energy consumption". Thus it shall be ensured that we are not «compromising the ability of future generations to meet their own needs» ([1], p. 24).

This "window of energy consumption" takes the energetic poverty line of economic development to define the minimal possible energy consumption whereas the maximum is defined by the amount still compatible with our planet's ecological equilibrium. Lastly the distribution of consumption among consumers serves as an indicator of social stability. The industrialised countries obviously still need to take large steps to make this idea come true whereas the developing countries mostly lie well below the consumption of 2000 Watt per capita (Fig. 2).

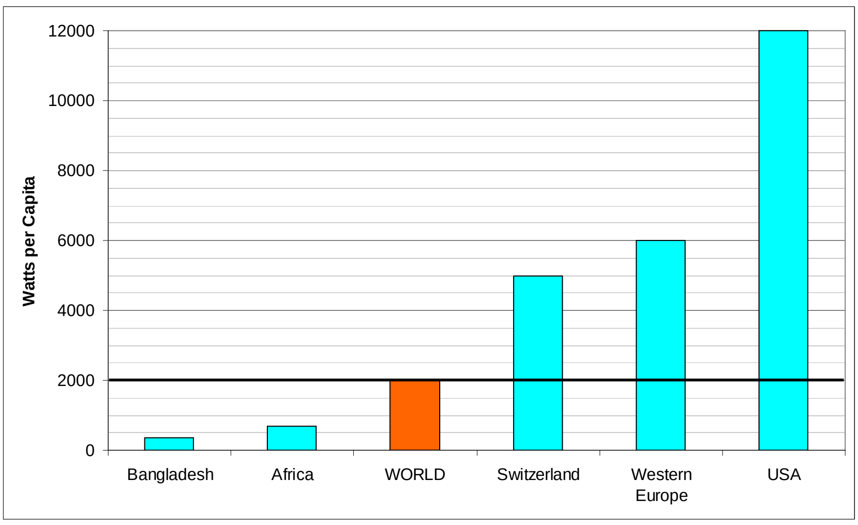

Figure 2. Comparison of the energy use per-capita in selected areas. Adapted from [5].

The association MINERGIE®, which is financially sponsored by the Swiss Federal Government as well as by the Cantons (cf. Federal States), took up those ideas in 2002, and the standard complies with them. It generated the building energy standard MINERGIE-P® which adopts mainly the ideas of *Passivhaus* elaborated by Dr. Wolfgang Feist during the 1990's in Darmstadt/D [6 and 7]. The differences between *Passivhaus* and MINERGIE-P derive mainly from the different standardisation in Germany and Switzerland. In addition, there are always different (political) points of view on the conceptual design of the optimal low-energy houses, of course. Here MINERGIE pursued perspicuously the following ideas:

- High living comfort due to surfaces of uniform temperatures,
- High indoor air quality due to mechanical ventilation,
- Better noise control since it is no longer necessary to open windows,
- Low energy consumption (not first but equal among others, see below),
- Long-term conservation of building value due to less structural damages such as caused by mould,
- Viable with a minimum of extra cost (less than 10% to 15%),
- Feasible with technology that is proven and existing.

It is of utmost importance to follow the idea of MINERGIE closely. Quite obviously the name derives from MINimal enERGIE (spelled with "ie" instead of "y" in German). The central focus of the MINERGIE agenda is not so much the absolute minimal use of energy but providing the comfort expected today at the lowest possible energy consumption. Otherwise one could just lower the room temperature to 16°C and thus generate energy savings of up to 25%! Another important point concerns the reasonable use of proven technology. It is not the goal to provide a playground for prospective technologies but to define a standard that can be met easily using existing technology and appliances.

Yearly the Swiss Federal Office of Energy (BfE) provides up-to-date data on energy usage [8]. Fig. 3 shows clearly the sectors featuring the highest potential for energy saving being traffic and housing. At the time, housing in particular offers a realistic fulcrum to start cutting down energy usage. This is the starting point for the definition of MINERGIE-P.

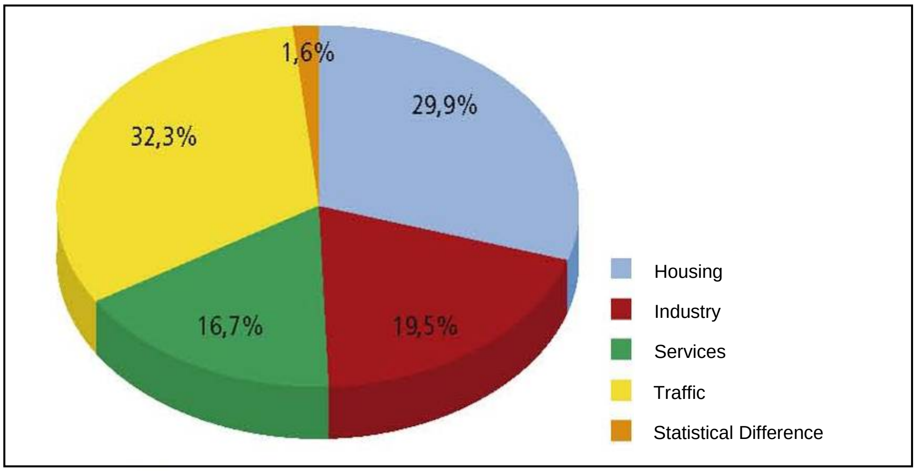

Figure 3. Energy use divided into sectors. From [8] p. 5.

In 1988 the SIA published its normative paper SIA 380/1 [9]. For the first time a standardised mathematical method was created to provide information on the predicted energy use of houses and moreover to demand a certain upper limit of energy use (as has recently been imposed by the EPBD). Thus the energy flows for the heating period were characterised and divided into heat flow due to transmittance (loss), heat flow due to airing and leakage (loss), heat flow due to persons and electricity (internal gain), and heat flow due to solar radiation (external gain), whereupon the difference between total loss and total gain, of course, needs to be provided by an additional heating system as shown in Fig. 4.

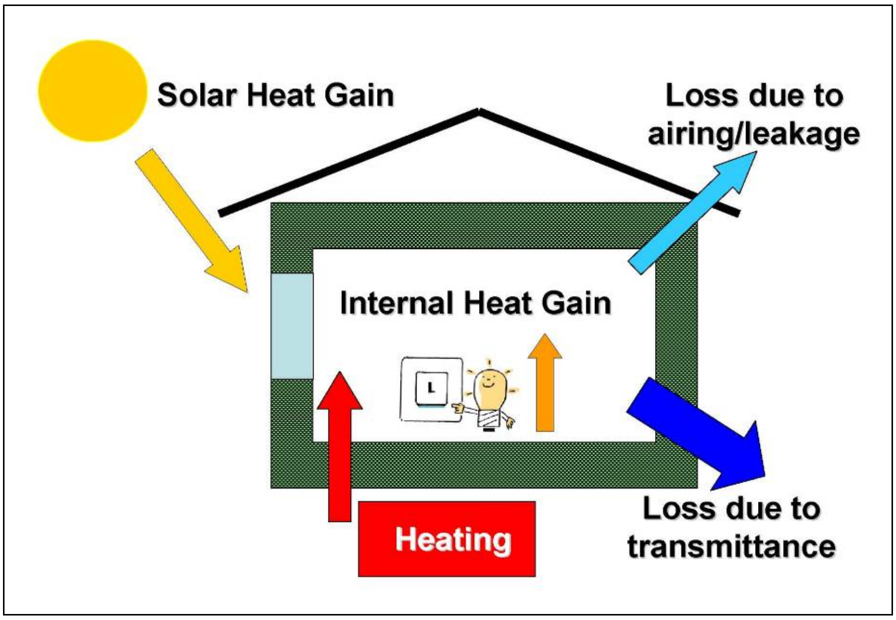

Figure 4. Energy flows following [9].

Obviously, one attempts to minimise losses and maximise gains for buildings that should consume as little energy as possible. Consequently the package of measures includes:

- Very well insulated walls (U-values ≤ 0.15W/m^{2}K, tending towards 0.10W/m^{2}K),
- Very good windows, consisting of three panes (U-value transparent ≤ 0.6W/m^{2}K, U-value including frame ≤ 0.9W/m^{2}K, total solar energy transmittance ≥ 0.5),
- Minimised thermal bridges and therefore a homogenous heat insulation of the building envelope,
- Highly airtight outer shell ($n_{50} \leq 0.6h^{-1}$ – air change rate at 50Pa pressure difference),
- Controlled ventilation using high performance heat recovery.

Moreover the building's main façade should face southeast to southwest featuring a minimum of 30% transparency of the outer shell. These orientations provide massive solar gains even in winter so that windows facing these directions contribute to the reduced heat demand. Windows in façades to other orientations and above all to the northwest to northeast produce losses and should therefore be quite small. (These statements are true for average Swiss meteorological conditions as in [10].)

Very often architects and energy engineers underestimate the importance of the realisable gains and concentrate on minimising the losses. This will never suffice to comply with the label MINERGIE-P. Indeed one could state that the "P" stands for "passive solar gains". Every factor that reduces those gains in any way (shading, skyline, and regulation) increases the heat demand considerably. The second most and often ignored point concerns thermal bridges. Common houses may feature a considerable amount of thermal bridges with losses of almost 5% of the heat demand. After having eradicated a good part of losses due to transmittance, thermal bridges (even in very well constructed and thought-through buildings) add up to 10%. If proper construction to avoid thermal bridges is neglected, the associated loss can increase considerably! Thus the ambitious goal of saving a very good part of the heat demand compared to standard houses may be missed.

RESULTS

To give the reader an idea of the dimensions we are talking about, we take an ordinary Swiss single-family detached house of $200m^2$ total area. The area in Switzerland is always measured grossly therefore including structural components as well. All the statistical energy values are based on this so-called energy reference area (ERA) and looked at over a whole year. The outer surface of the house amounts to $400m^2$ thus resulting in a ratio of thermal exposed area to ERA of 2 ([11] describes the ERA as "conditioned space" and does not explicitly specify a ratio of ERA to enveloping, thermally exposed surface). The annual energy consumption allowed by law amounts to $75kWh/m^2_{ERA}$. MINERGIE-P allows 20% of the legal energy consumption. This means that only $15kWh/m^2_{ERA}$ per year might be used as maximum heat demand. This is the primary of four conditions. The German energy label *Passivhaus* also allows $15kWh/m^2$ but measures the area differently (net instead of gross).

Second, the demand of energy including hot water production, auxiliary energy for pumps and ventilation including weighting factors must not exceed $30kWh/m^2_{ERA}$. Weighting factors always hold a political component and shall not be discussed here. Table 1 shows the weighting factors of MINERGIE together with those of *Passivhaus* and SIA to give an idea of the different points of view. All those weighting factors try to take into account the conversion between secondary energy (i.e. the amount of energy delivered to the property) and primary energy (i.e. the energy including losses due to extraction, transport, and the like).

Table 1. Weighting factors to express the conversion from primary to secondary energy

Heat producer	*Passivhaus* (D)	MINERGIE (CH)	SIA (CH)
Heating oil	1.08	1.0	1.1
Natural gas	1.07	1.0	1.1
Wood (any form)	1.01	0.5	0.1
Electricity	2.97	2.0	2.9

The Swiss mathematical method to calculate the energy demand standardises not only the internal gains, ventilation rates, and room air temperatures but also the demand of heat for production of hot water for sanitary purposes. The amount is referred to the ERA. Thus the examined single-family house needs $13.9kWh/m^2_{ERA}$ or a total of 2778kWh per year. This is enough to heat 130 litres of water from 10°C to 60°C per day. Assuming typical values for ventilation ($150m^3/h$) and auxiliary purposes ($1.5kWh/m^2_{ERA}$), Table 2 provides an overview of the energy demand so far.

Table 2. Overview to the energy demand

Energy demand by	Demand [kWh/m^2_{ERA}]	Form of energy	Weighted energy
Heating	15.0	Heat (wood)	7.5
Sanitary hot water	13.9	Heat (wood)	7.0
Ventilation	3.0	Electricity	6.0
Auxiliary energy	1.5	Electricity	3.0
Total			23.5

Of course, the exact values may vary as well as the way to produce the energy for heating and hot water. At the same time the presented values represent the typical case of a MINERGIE-P single-family house.

The second to last condition involves that the outer shell must be very airtight. Air tightness is determined by applying a pressure difference of 50Pa and measuring the air change rate.

MINERGIE-P requires an air change rate below $0.6h^{-1}$ at 50Pa (average of both overpressure and vacuum). Taking into account the consumption of electricity, the last condition demands that all domestic appliances hold the European energy label A, and A^{+} for freezers, respectively.

Statistical interpretation of 100 Swiss MINERGIE-P houses

As a matter of course the statistical interpretation of all realised MINERGIE-P houses are of interest. The following figures show the evolution of the label, which was launched at the end of 2002 (Fig. 5). Furthermore the heat production systems (Fig. 6) as well as the heat distribution systems (Fig. 7) are shown and give a distinct idea of the most common solutions to both challenges. All values are based on the last statistical review of mid-February 2007.

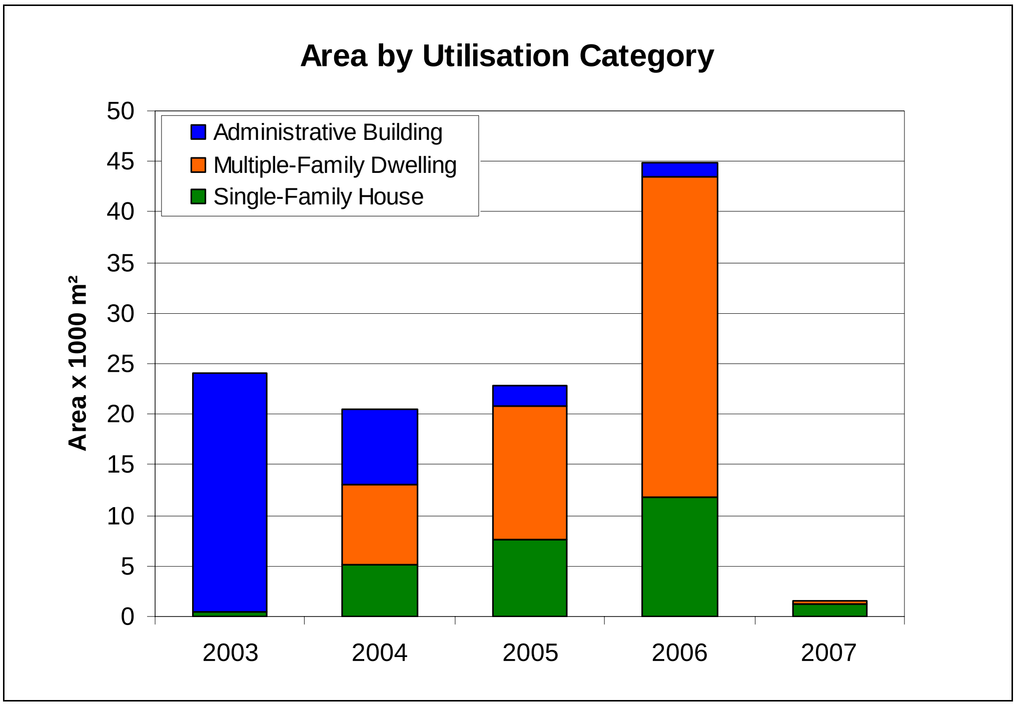

Figure 5. Evolution of area certified each year as MINERGIE-P.

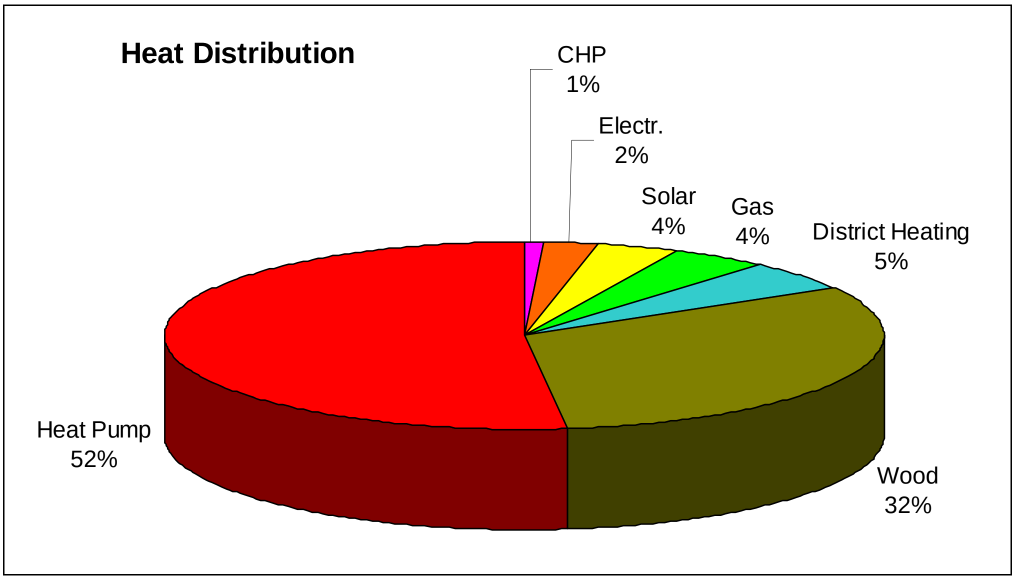

Figure 6. Statistical interpretation of used heat production systems.

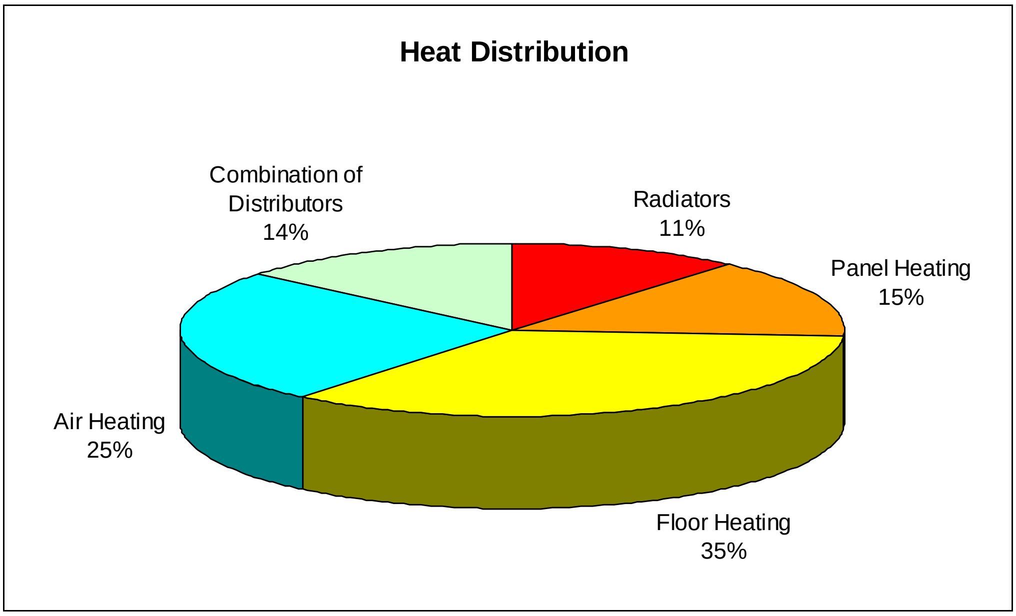

Figure 7. Statistical interpretation of used heat distribution systems.

DISCUSSION

The energy label MINERGIE-P presents a valuable way of classifying energy-efficient buildings. The perspective lies both on energy saving and, in particular, presenting an excellent comfort providing high indoor air quality and minimising the risk of structural damages due to mould and the like.

It could be shown that using proven technologies and appliances of today's state of the art one can decrease the heat demand by 80% without taking any risks. Furthermore, the importance of realising feasible energy savings could be shown. In particular the housing sector, which consumes roughly one third of the energy in Switzerland, still holds a huge potential to reduce the massive output of CO_2.

Experience shows that the presented standard can easily be communicated not only to energy-interested pioneers or idealists but to the broad public as well. The astonishing evolution of the still young label is demonstrated by the increasing rate of newly built living area with the high MINERGIE-P standard. Recently the better conservation of building value even convinced several banks to offer better conditions for home mortgages.

The complete success of all MINERGIE energy labels presents a prime example of an effective marketing strategy. (To date there are four energy labels assigned by MINERGIE – compare [12].) MINERGIE as a brand has consistently been built up over the last years. Without the tireless work to establish the vision of energy saving buildings in the minds of both the public and politicians, in Switzerland population and building industry would probably still be waiting for «an increased political will to address our common future» ([1], p.13).

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BGP index: an approach to the certification of building global performance

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SUMMARY

In order to improve life quality into buildings, the present paper proposes a certification model for indoor total comfort.

The first part of the paper concerns energy certification of buildings, with attention to energy efficiency, renewable sources, heating and air-conditioning. In the second part of the paper certificate concept is extended to factors which contribute to ideal requirements into buildings: acoustics and lighting, use of water, safety and technology are studied according to standards. Finally, an overall building quality certificate is proposed, based on indexes concerning specific performances. Every index is assigned a weight to characterize different contributions to Building Global Performance (BGP) index, which expresses total indoor quality. Weights are estimated according to objective criteria. Measurements and provisional models allow to analyse and classify a private flat as a case study. A reliable approach is shown to improve indoor environments and to support comfortable and sustainable buildings.

INTRODUCTION

Human activities impose a long time stay into buildings, where high comfort must be guaranteed: such a condition comes from a combination of different elements, such as thermal, hygrometric, acoustic, lighting, safety and technological factors.

According to Directive 2002/91/EC [1] and to national or regional procedures, energy certification of buildings assumes remarkable importance in order to rationalize the use of energy sources and to reduce costs and pollution.

The improvement of indoor life quality is due to many other factors too, which contribute to global well-being. In the present paper noise protection is studied according to Italian decree DPCM 5 December 1997 [2], which concerns building systems and acoustic insulation of façades, walls and floors. Suitable artificial lighting and natural light into buildings enhance indoor comfort; at present, Italian standard UNI EN 12464-1 [3] disciplines lighting of work places, while UNI 10840 [4] defines criteria for natural light into schools. Attention must be paid also to rational use of water: the installation of specific equipments can reduce considerably domestic water consumption, thus improving building performances. As concerns electrical, gas and water systems, they must be provided with protection devices and automatic control stop in case of leaks, according to Italian law 5 March 1990, n. 46 [5] and respective decrees; safety of persons, instead, can be upgraded by means of control and alarm devices. Finally, technology optimizes functionality and well-being into buildings with automation and communication systems.

The aim of the present paper is to propose a model of certification extended to building global performance, concerning all main aspects in an independent and objective way. Experimental measurements were carried out in order to analyse a private flat as a case study, considering use of rooms, building components and consumption bills.

METHODS

Indoor global performance

Quality of buildings means an appraisal of overall indoor comfort. Assessment indicators of single contributions were proposed and a global index was defined to classify new and existing buildings, referring to total performance: the model of building global certification was then applied to a case study.

ENERGY CERTIFICATION OF BUILDINGS

Building needs energy to condition indoor climate. Requirements based on primary energy consumption allow to have a coherent evaluation of energy efficiency of adopted systems. The energy performance of a building must be estimated with a methodology based on a general framework, which was defined by Directive 2002/91/EC [1]. Construction, sale and rent of a building need a certificate of energy performance for owner and purchaser: the purposes of certification are to estimate and to compare buildings, in order to make energy performance part of the appraisal, together with dimensional and aesthetic considerations.

Therefore, as household-electrics, also buildings must be provided with a coloured label, that identifies 7 classes of energy performance, where "A" means low consumptions/high efficiency while "G" means high consumptions/low efficiency. The aim is to estimate all energy contributions (heating and cooling, hot water, ventilation, lighting systems and renewable sources) thus pursuing high quality buildings. "Building Energy" index is defined according to energy certification: each class of energy performance (from G to A, Table 1) is assigned a value from 1 to 7.

Table 1. Classification of energy performance for residential buildings.

Annual specific primary energy consumption (kWh/m^2year)	Energy performance class	"Building Energy"
< 50	A	**7**
50-70	B	**6**
70-90	C	**5**
90-110	D	**4**
110-130	E	**3**
130-150	F	**2**
> 150	G	**1**

Energy certification procedures

Building energy requirements can be simulated by implementing calculation codes. General and climatic data, composition of opaque structures (walls and floors), transparent elements (windows) and exposure of building must be defined. Energy performance was studied for 3 different Italian cities (Perugia, Milan and Bolzano) which defined energy certificates, in order to evaluate the energy efficiency of the same building in such different places (Table 2).

Table 2. Main differences among examined energy certification procedures.

Energy performance	Perugia	Milan	Bolzano
Building shell	Yes	Yes	Yes
Heating system	Only 75%	Yes	No
Hot water	Yes	Yes	No
Plant efficiency	Only for hot water	Both for heating and hot water	No

Acoustic insulation

Noise sources can determine uncomfortable acoustic conditions into buildings. Partitions obstacle noise transmission, due to material mass. Mass increment doesn't represent a feasible method to achieve high acoustic well-being into rooms, so it is necessary to study building acoustics. In order to assess indoor acoustic quality, "Building Acoustics" index is proposed: it is evaluated by means of 7 classes of acoustic performances for 5 acoustic requirements into buildings, according to Italian decree D.P.C.M. 5 Dicember 1997 [2]. As concerns existing residential buildings, the limits imposed by decree for each parameter are assumed as central values of the scale, that was obtained by steps of 3 dB (+ 3 dB is equivalent to doubling of sources) both in increasing and decreasing sense: each acoustic requirements is assigned a score from 1 to 7, as specified in Table 3. Regarding noise insulation indexes R'_w (partitions) and $D_{2m,nT,w}$ (façades), acoustic comfort improves for increasing values, while the very contrary is the case of acoustic pressure levels L'_{nw} (floors), $L_{AS,max}$ and L_{Aeq} (systems); "Building Acoustics" index is calculated as arithmetic mean of obtained values.

Table 3. Classification of acoustic requirements for existing residential buildings.

	1	2	3	4	5	6	7
R'_w	< 41	41-44	44-47	47-50	50-53	53-56	> 56
$D_{2m,nT,w}$	< 31	31-34	34-37	37-40	40-43	43-46	> 46
L'_{nw}	> 72	69-72	66-69	63-66	60-63	57-60	< 57
$L_{AS,max}$	> 44	44-41	41-38	38-35	35-32	32-29	< 29
L_{Aeq}	> 44	44-41	41-38	38-35	35-32	32-29	< 29

Artificial lighting and natural light

Ranges of medium illuminance *E* according to visual tasks, limits for direct dazzle of light sources and suitable chromatic features of lamps are important to have comfortable lighting conditions. Natural light into building depends on site, exposure, windows, adjacent buildings and landscape natural elements. Correct exposure to sun is necessary to use natural lighting into buildings. Day lighting medium factor FLD_m is defined according to (1):

$$FLD_m = \frac{E}{E_0}, \qquad (1)$$

where E is room medium illuminance and E_0 is sky medium illuminance (5000 lux).

"Building Lighting" index concerns indoor artificial lighting and natural light. The assessment of artificial lighting is carried out by identifying 7 classes, for typical rooms into residential buildings (bedroom, kitchen, bathroom, living-room/access area) based on E medium values according to use. Maximum and minimum values (in lux unit) recommended by technical literature and standards are assumed as extreme values and the central range is divided into equal parts (Table 4). Natural lighting, instead, is evaluated by identifying 7 classes of FLD_m, for each room. The minimum value imposed by Perugia Building Regulations (2%) is assumed as inferior limit, while, on the basis of experimental measurements and technical literature advices, superior limit is fixed equal to 22%. The scale is obtained by dividing the range between 2% and 22% in equal parts and each class is assigned a score from 1 to 7 (Table 5). "Building Lighting" index is calculated as the arithmetic mean of the indicators, concerning artificial lighting system and natural light, into each room. If rooms of the same kind are present (for example, 2 bedrooms), the lowest value is considered cautiously.

Table 4. Classification of artificial lighting performance for residential buildings.

	1	*2*	*3*	*4*	*5*	*6*	*7*
Bedroom	< 50	50-70	70-90	90-110	110-130	130-150	> 150
Kitchen	< 200	200-260	260-320	320-380	380-440	440-500	> 500
Bathroom	< 50	50-70	70-90	90-110	110-130	130-150	> 150
Access area	< 50	50-70	70-90	90-110	110-130	130-150	> 150

Table 5. Classification of natural light performance for residential buildings.

	1	*2*	*3*	*4*	*5*	*6*	*7*
Bedroom	< 2%	2-6%	6-10%	10-14%	14-18%	18-22%	> 22%
Kitchen	< 2%	2-6%	6-10%	10-14%	14-18%	18-22%	> 22%
Bathroom	< 2%	2-6%	6-10%	10-14%	14-18%	18-22%	> 22%
Access area	< 2%	2-6%	6-10%	10-14%	14-18%	18-22%	> 22%

Rational use of water

Waste of water, for washing and other uses, can be reduced by specific devices or effective options, such as systems for the recovery of rain water. "Building Water" index evaluates the rational use of water taking into account the presence of water saving equipments listed in Table 6. Index may varies from 0 to 7 and each device increases the value by 1 point, except toilet with double key or stop key flush (2 points), because it guarantees higher water savings.

Table 6. Classification of water saving performance.

Rational use of water	Presence	Value
Water saving tap	Y/N	1/0
Flow controller tap	Y/N	1/0
Flow controller shower	Y/N	1/0
Low water consumption toilet	Y/N	1/0
Double key/Stop key flush toilet	Y/N	2/0
Rainwater tanks	Y/N	1/0

Safety into buildings

Safety of systems is basic in the appraisal of indoor quality: a building can't be comfortable if first of all it is unable to protect people from the risks caused by dangerous or obsolete plants. Safety is also protection from outside, so alarm and video-surveillance systems may prevent to enter someone's property illegally and contribute to satisfy the growing need of safe buildings. "Building Safety" index estimates safety into buildings, both of systems and people; it can be equal to 0 or assume a value between 4 and 7. The safety of electrical, gas and water systems, disciplined by Italian law 5 March 1990, n. 46 [5] and respective decrees, was assumed as absolutely necessary: if minimum law prescriptions are respected, the index assumes value 4 (medium safety degree). The presence of safety devices to protect people increases index value by 1 point (Table 7).

Table 7. Classification of safety performance.

Safety parameter	Presence	Value
Properly done system (certificate) + IMQ, CEI or equivalent brand + Magnetothermal switch + Differential switch + Earthed system + Aeration hole + Gas/Steam aspiration hood	Y/N	4/0
Video screen entryphone	Y/N	1/0
Alarm system	Y/N	1/0
Signalling to Police force or private vigilance service	Y/N	1/0

Technological devices

High quality habitability and functionality of buildings depends more and more on available technological systems. Integration of services affects positively indoor well-being. "Building Technology" index estimates the technological level into rooms and can assume values from 0 to 7. Each device listed in Table 8 increases by 1 point the index minimum value.

Table 8. Classification of technological performance.

Technological devices	Presence	Value
PC/Internet/ADSL	Y/N	1/0
Sensors	Y/N	1/0
Automatic lighting/curtains	Y/N	1/0
Parabolic antenna	Y/N	1/0
Hi-Fi/Wi-Fi	Y/N	1/0
Auxiliary devices (transducers, interface circuits)	Y/N	1/0
Residential gateway	Y/N	1/0

Building Global Performance index

"Building Global Performance" (BGP) index is defined by giving each indicator a weight. In order to attribute the weight fitting every index, different criteria were examined and the one based on costs was preferred, which seems to assure the greatest objectivity; therefore weights are calculated on the basis of cost analysis. Average costs to be supported in order to equip new or existing buildings with materials and devices which can guarantee high performance, quality and comfort are estimated in about 54% of the total costs of construction works (Table 9), according to Italian market general data. The weights given to the indexes were obtained by comparing single contributions to total percentage 54% (Table 9).

Table 9. Index weights according to Building Global Performance (BGP).

Index	Contribution	Percentage on total costs	Weight
Building Energy	Thermal insulation/energy performance	20%	37%
Building Acoustics	Acoustic insulation	15%	28%
Building Lighting	Natural light/artificial lighting system	5%	9%
Building Water	Water saving equipments	7%	13%
Building Safety	Safety devices	3%	6%
Building Technology	Technological devices	4%	7%
BGP	*Total*	*54%*	*100%*

Case study

A private flat in Perugia (Umbria Region, Central Italy) was assumed as a case study and an example of global certification was given. The examined flat is sited in a residential building that was built in the early Seventies (Figure 1). The occupants are 2 students. Independent heating system is fed by a boiler, having nominal thermal power of 24 kW; fuel is natural gas. Indoor temperature can be regulated by a thermostat. The system works both for heating and production of hot water. The flat has a volume of 161,7 m^3; the area is equal to 56,7 m^2. External walls are 29 cm thick (1,5 cm external and internal plaster, 2 layers of 12 cm bricks separated by 2 cm of mortar). The walls that separate the examined flat from the adjacent ones are 15 cm thick (12 cm bricks and 1,5 cm coat of plaster). The floor thickness is 30 cm and dividing walls into the flat are 10 cm thick.

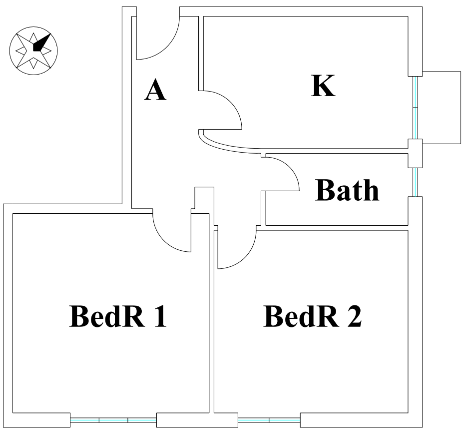

Figure 1. Plan of examined flat.

RESULTS

Building energy performance

According to calculations, the flat was assigned “E” class, with annual specific consumption of primary energy equal to 126,13 kWh/m^2year (86,90 for heating; 0,00 contribution from renewable sources; 39,23 for hot water). The examined procedures classify the flat in the same category: differences among obtained values of energy consumption, however, is due mainly to the methodology of calculation, the contributions to be considered and the climatic zones (average temperatures). In order to compare the specific energy performance of the building shell, the values to be considered are 115,87 kWh/m^2year for Perugia, 102,72 kWh/m^2year for Milan and 106,80 kWh/m^2year for Bolzano. Finally, the value obtained as simulated energy consumption of the case study has a good agreement with the study of real energy consumptions of the same flat as concerns the bills (126,75 kWh/m^2year).

Case study “Building Energy” index, defined by “E” class, obtained 3 points (Figure 2).

Building acoustic performance

Italian and international standards UNI EN ISO 140 [6] and 717 [7] were applied to carry out measurements and calculations (Table 10). According to Table 11, “Building Acoustics” index for the examined flat obtained 2,0 points (medium; Figure 2).

Table 10. Case study acoustic performance.

Acoustic parameter	Measured value	Unit
R_w	37	dB
$D_{2m,nT,w}$	33	dB
$L'_{n,w}$	74	dB
L_{ASmax}	47	dB(A)
L_{Aeq}	33	dB(A)

Table 11. Classification of case study acoustic performance.

R'_w	1						
$D_{2m,nT,w}$	1						
L'_{nw}		2					
$L_{AS,max}$	1						
L_{Aeq}					5		

Building lighting performance

According to standards UNI EN 12464-1 [3] and UNI 10840 [4], artificial lighting and natural light were measured (Tables 12 and 13). "Building Lighting" index for the case study equals 3,8 points (medium-low), as shown in Figure 2.

Table 12. Case study lighting performance.

Room	Natural light	Artificial lighting	Unit
Bedroom	1030	109	lux
Kitchen	1235	168	lux
Bathroom	783	83	lux
Access area	145	66	lux

Table 13. Classification of case study lighting performance.

Artificial lighting	Bedroom				4			
	Kitchen	1						
	Bathroom			3				
	Access area		2					
Natural light	Bedroom						6	
	Kitchen							7
	Bathroom					5		
	Access area		2					

Other building performances

The flat obtained a low value of "Building Water" index, equal to 2 (only water saving tap and flow controller tap are present). Also "Building Technology" index obtained 2 points (flat equipped with PC/Internet/ADSL and sensors/thermostat). Finally, as alarm and signalling devices are unavailable, case study has "Building Safety" index equal to 4 (Figure 2).

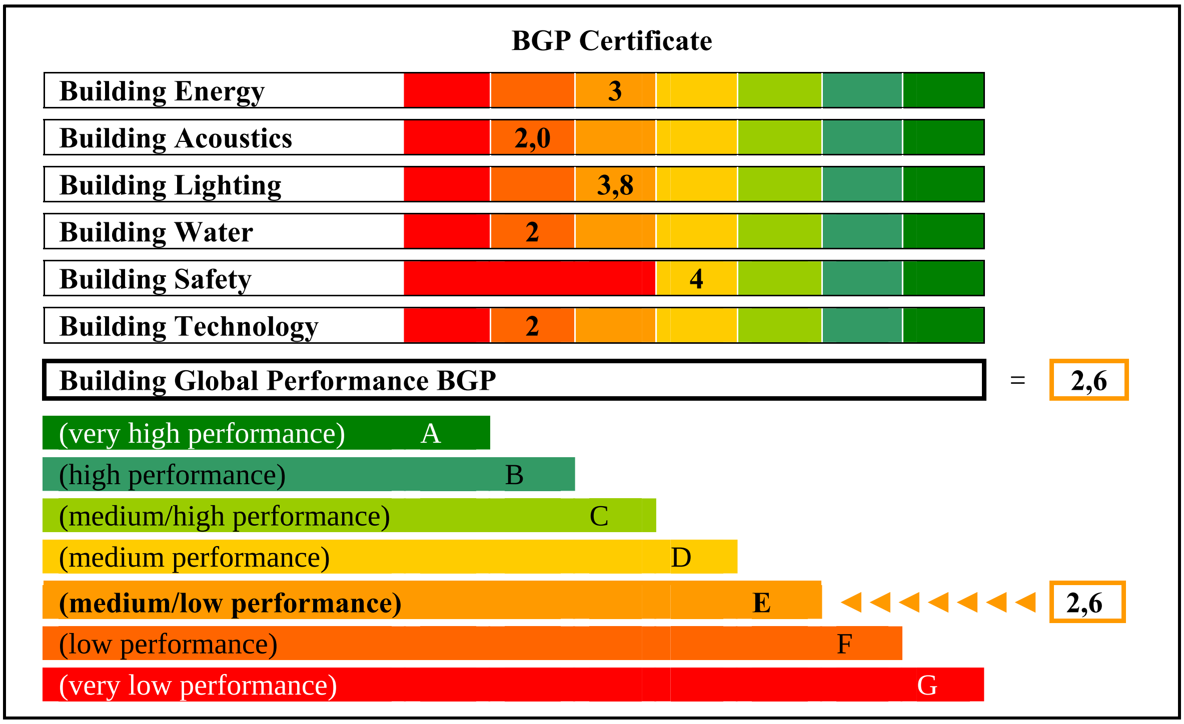

Figure 2. BGP certificate: results of application to the case study.

Building Global Performance index

All required input data were processed to calculate BGP index: the calculation gives back the values of all indicators as a certificate (Figure 2). Case study obtained BGP = 2,6 (low-medium performance), in tune with the appraisal of the occupants.

DISCUSSION

A new instrument for indoor quality certification was proposed: “Building Energy”, “Building Acoustics”, “Building Lighting”, “Building Water”, “Building Safety” and “Building Technology” indexes were defined to determine “Building Global Performance” BGP.

As a validation, the assessment of a private flat in Perugia (Umbria Region, Central Italy) was carried out. Energy certification was analysed according to different methodologies: total energy consumptions of the flat, calculated according to Perugia Building Regulations, are equal to the ones obtained by studying actual energy consumptions, thus demonstrating the reliability of such a procedure. Weights were assigned to indexes on the basis of cost analysis; actually costs were estimated with approximation, so a combination of mentioned analysis and further assessments, even subjective evaluations (such as statistical preferences of people), may improve the model. Being comfort a personal feeling, there are different perceptions, but it must be attempted to satisfy the greatest number of persons. The assessment could be integrated with other peculiar features, such as architectural quality of buildings. BGP index could be measured to test existing buildings or it could be predicted to evaluate new buildings, according to standards.

The purpose of the present paper is to put in evidence the advantages that would be obtained by pursuing building overall quality. High performances are expensive, however the increment of building cost is amortized thanks to lower consumption and optimal comfort.

Quality and saving must be pursued by builders and planners in order to guarantee high indoor habitability, functionality and efficiency and to perform sustainable buildings.

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Using fractional experimental designs to establish multi-parametric expressions for energy consumption of commercial buildings

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SUMMARY

The large advancement in the market of building materials and of HVAC systems leads to a wide variety of technical solutions that can be employed to build an efficient low energy commercial building.

The comparison between different combinations of parameters representing the main elements of the envelope, lighting management and solar blinds, office automation, plug consumptions and efficiency of the HVAC system is always of special interest to building owners and HVAC design engineers; however, such a study is rarely totally accomplished because of the time required.

The paper describes a fast method for organizing simulations through the use of fractional factorial designs. The method permits reduction of the number of simulations needed to cover all combinations of influent parameters.

The reduction result shows that it is feasible and quick to determine mutli-parametric models for a large range of building types. As a result, consumption can be calculated bypassing steps of projects description, sizing of components and running the simulations. The precision of such multi-parametric models is established by comparison with full simulation.

INTRODUCTION

The purpose of the project is to build a large library of analytical multi-parametric models giving the consumption as a function of key parameters of typical building/system.

The project is addressed mainly to engineers and commercial buildings design departments who aim to obtain optimized solutions quickly and to identify the impact of main choices on total yearly energy consumption.

In a first time, the description of French commercial sector, mainly for offices and hospitals, has been established. It permits to draw up a representative typology of these buildings [1] [2]. On the basis of this typology, geometrical description and different values or scenarios necessary for accomplishing the simulation are defined [3][4]. Simulations are then carried out with ConsoClim [5], a building energy simulation software package. Its particularity is that systems models [6] require simple and available parametric inputs and are coupled to the building model.

The variables predominantly affecting energy consumption are then listed. The high number of possible simulations needed to cover all the variation of variables in addition to the organization of a huge quantity of results was initially the main reasons for introducing experimental design notion.

METHODS

The aim of the method is to obtain maximum information with a minimum number of simulations. For a defined problem, various solutions can be tested, evaluated and compared. A systematic organization of simulations helps to analyze the results and decreases the risk of errors at the same time.

The experimental designs are essentially based on multi factorial experiments and on the use of multiple regressions besides variance analysis while treating the results. Their application supposes a good knowledge of the studied phenomenon before realizing the tests. This has to be accompanied by the ability to detect the parameters most likely to influence the system operation [7]. It is necessary to represent the diversity of powerful energetic technical solutions already present on the market, then to be able to establish a correspondence between the chosen solutions and a level of the cluster parameter representing the variety of these solutions.

If the number of considered parameters and the number of their levels are very high, the solution consists in reducing the number of levels and/or factors followed by control of the precision of obtained results. When defining the design of experiments, a preliminary step is to study the linearity (or not) of variables in order to set the number of levels.

Figure 1. Choice of number of levels: 2 levels on the left (linear evolution), 3 levels at least on the right (non linear evolution)

COMPLETE FACTORIAL DESIGN

Factorial design or complete factorial design is to experiment all the possible combinations between the levels of the variables. The number of combinations is a product of the numbers of levels of the factors. [8]
In the study we considered 13 variables for office building application, they are:

Table 1. Low and high values of the considered variables – building/system

	Variables	Level	
		Level (-1)	Level (+1)
U_{op}	Insulation of ceiling and walls (opaque)	0.2, 0.4 W/m^2/K	0.3, 0.6 W/m^2/K
U_{bay}	Characteristics of glazed surfaces	2 W/m^2/K	3 W/m^2/K
Ori	Orientation	Glazed facades N/S	Glazed facades E/W
MSP	Management of solar protection as a function of natural lighting	Occupant not so reactive	Occupant very reactive
Off_Aut	Mode of management and effeciency of office automation equipments	15 W/m²	7.5 W/m²

Light	Management and efficiency lighting	Interrupter 18 W/m²	Interrupter + graduator 10 W/m²
Iner	Inertia level	light	heavy
Perm	Permeability	1.2 $m^3/(h.m^2)$ under 4 Pa	2.4 $m^3/(h.m^2)$ under 4 Pa
Vent	Ventilation	Reduction in flow rate during occupancy -30%	Normal flow rate during occupancy
Aux_Eff	Fans and pumps efficiency	0.87	0.52
Boil_Eff	Boiler efficiency	0.98 (70°C), 1.08 (40 °C)	0.89 (70°C), 0.88 (40 °C)
Dist_Ins	Insulation of distribution network	0.14 W/m.K	0.28 W/m.K
EER	EER Chillers - Split ,VRV (EER/COP)	3.3 - 4.2/4.5	1.9 - 1.8/1.95

This paper presents an application of the method to a small area office building (1000 m^2) existing in the industrial suburban zones and simulated in the climatic region of Paris; for other cases see [11]. The selected system for this article is made of a chiller for cooling, gas boiler for heating, air-handling unit for fresh air, and 4 tubes fan coil units (domestic hot water is not taken into consideration). The HVAC system is described from market real devices after sizing all the needed elements in cooperation with engineer design departments. Simulations are realized to check the consumption tendency between the levels of each variable (the linear response justifies keeping only two levels). The variables used in the experimental design are centered and reduced. To pass from a variable, A, to the reduced centered variable, x, the formula is:

$$x = (A-A_0)/\Delta_A \quad (1)$$

Where A_0 is the central value of the interval [- 1, +1], expressed in current unit, Δ_A being the variation between the average position of the variable and the domain extremity.

Figure 2a (left) gives consumption while passing from the basic case (all variables at level –1) to another case – each time one variable passes to level 0 (central value of the interval [- 1, +1]) then on the level (+1). The value obtained is compared with the calculated value at level (0) obtained from a linear regression.

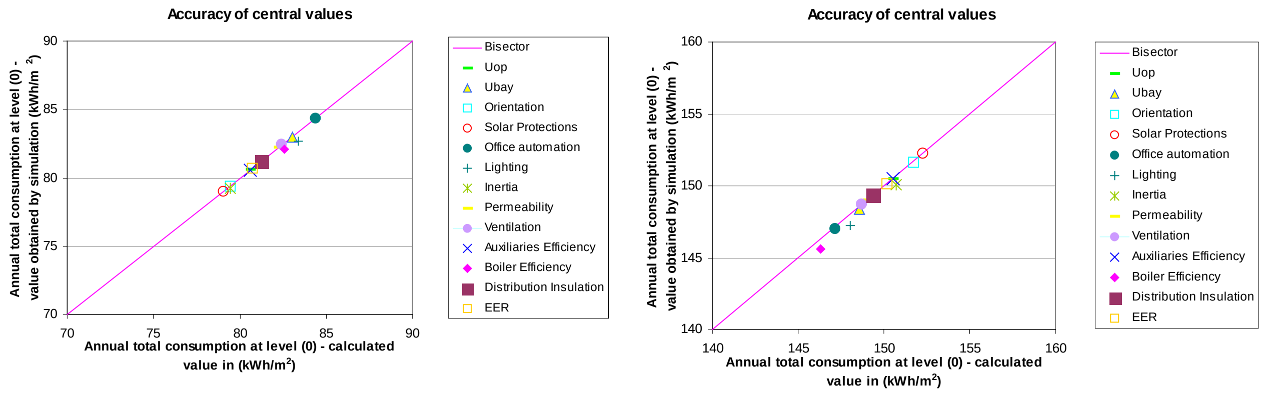


Figure 2. Precision of the assumption of the answer (consumption) linearity

Figure 2b (right) is obtained by the same principle, this time starting from a basic case where all the variables are at the high level (+1). For each simulation, a variable passes to the level (-1) then to the level (0). The variation obtained with the calculated values is negligible.

To carry out a 2^{13} complete factorial design, 2^{13} simulations have to be done. Each simulation gives an answer y_i. The answer is represented by a matrix-column [Y] including 2^{13} lines. The mathematical model associated to the 2^{13} design contains 2^{13} unknowns: one constant, 13 principal effects and 2^{13} - 13 – 1 interactions.

The effects are represented by a matrix-column [A] constituted of 2^{13} lines. The matrix of calculation of coefficients X contains a number of lines equal to the number of answers y_i that is 2^{13}, and as many columns as there are unknown factors i.e. 2^{13}.

Matrix X is thus a square matrix (2^{13}, 2^{13}). It is reversible, we can write:

$$Y\,(2^{13}, 1) = X(2^{13}, 2^{13})\,A\,(2^{13},1) \qquad (2)$$
$$A = X^{-1}Y \qquad (3)$$

The researched model is constituted from 13 parameters of Table 1. The matrix representation usually employed in experimental designs is replaced by YATE notation [13]. The variable is multiplied by the level it takes (-1 or +1); the interaction between two variables or more is represented by the product of their levels. So the simplified writing of the model becomes (Refer to table 1 for nomenclature):

C_{tot} (kWh/m^2) = M_{tot} + U_{op} + U_{bay} + Ori + MSP + Off_Aut + Light + Iner + Perm + Vent + Aux_Eff + Boil_Eff + Dist_Ins + EER + all interactions of the 1st order (= 78 interactions) (4)

M_{tot} is the average and C_{tot} is the total yearly consumption. This model considers that the interactions of 2nd order and above are negligible, which is validated by comparing the results of the model to the "real" answer: that of the simulation.

In this study, the energy ratios refer to total energy consumption (lighting, heating, cooling, plug loads and auxiliaries) without considering specific areas of the building such as data processing centers. Conversion into primary energy is given by applying the French rules [10] ratios (2.58 for electrical energy and 1 for gas energy).

The number of simulations needed to accomplish such a model is: 2^{13} (13 variables with 2 levels each) or 8192 simulations. This number is huge in terms of results and organization. As mentioned in the introduction, the methodology is to be applied to two main commercial sectors: offices and hospitals represented by five types of office buildings and two types of health buildings. The building/system matrix is representative of the real park in France and each system is sized per climatic region. With this model of the park, just for the small area building analyzed in this paper we have: 5 HVAC systems, 2 climatic regions i.e. 10 times 8192 simulations. The time for a series of 8192 simulations is around 820 hours with a relatively powerful computer. These figures show that complete factorial design is not a realistic solution. For us, it is only a reference set of results to compare with the fractional factorial design or fractional design presented below.

FRACTIONAL DESIGN

The fractional design is a "fraction" of the factorial design; it is an orthogonal subset of combinations of the complete factorial design. Its main advantage is revealed in the significant reduction of the number of simulations and therefore rapidity in giving reduced models.
Nevertheless, the reduced number of experiments makes it impossible to determine all coefficients of the multi-polynomial model related to the complete factorial design. The

reason is that the number of coefficients is higher than the number of experiments. In fact, the fractional design makes it possible to determine the sum of certain coefficients identified by the complete factorial design; these coefficients are called “aliases”.

A half-design is less expensive in terms of simulation number than a complete one, but this profit is offset by an ambiguity in the estimation of effects [13]. However, when building fractional designs, it is necessary to alias as much as possible the principal effects with interactions of a higher order, these latter being in general negligible.

To simplify the application of fractional designs, we use the Taguchi method or Taguchi tables [15]. The construction of an experimental design according to the method of Taguchi is based on two principles [9]:

1- the use of tables, facilitated thanks to linear graphs, provides the mode for filling the matrix.
2- considering the actions of 2nd order negligible except in particular cases.

Some interactions are immediately neglected when physical reasons deem it possible; for example, the interaction between the boiler efficiency and the EER is null. On the other hand, there are some interactions that cannot be neglected:

- interactions between the thermal transmittance of the glazing and the internal contributions or lighting for a largely glazed building (thus strong internal contributions and lighting),
- interaction between the boiler efficiency and the insulation of the distribution for a significant length of the network; etc.

The reduction of the number of interactions considered leads to a reduction of the degree of freedom of the model and consequently the number of experiments to be realized. The first order interactions, which are considered significant, are kept. The influence of this choice is then evaluated.

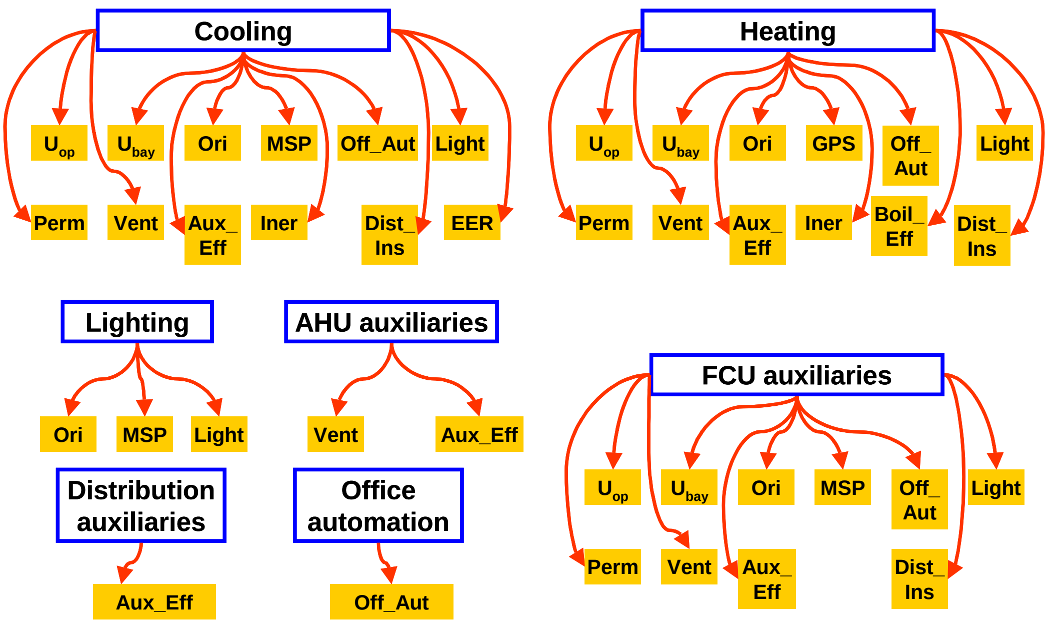


Figure 3. Variables that have an effect on the consumption of each sub model

Selection of the most significant interactions requires advance experience in the building/system simulation. To avoid this expertise requirement, we suggest the establishment of different models representing categories of annual consumption (figure 3). The addition of these elementary models of subparts constitutes the total consumption. The variables that influence each category of consumption are the following:

The global model is then composed of sub-models (Refer to table 1 for nomenclature):

$$C_{cool}\ (kWh/m^2) = M_{cool} + U_{op} + U_{bay} + Ori + MSP + Off\_Aut + Light + Iner + Perm + Vent + Aux\_Eff + Dist\_Ins + EER.Perm + EER.U_{bay} + Perm.U_{bay} + EER.Off\_Aut + Perm.Off\_Aut + U_{bay}.Off\_Aut + EER.Aux\_Eff + EER.Light + Perm.Light + U_{bay}.Light + EER.MSP + Off\_Aut.Light + EER.Ori + EER.Uop + EER.Iner \quad (5)$$

$$C_{heat}\ (kWh/m^2) = M_{heat} + U_{op} + U_{bay} + Ori + MSP + Off\_Aut + Light + Iner + Perm + Vent + Aux\_Eff + Dist\_Ins + Boil\_Eff.U_{op} + Off\_Aut.U_{bay} + Ubay.Light + U_{bay}.Perm + Boil\_Eff.U_{bay} + Off\_Aut.Light + Off\_Aut.Perm + Boil\_Eff.Off\_Aut + Light.Perm + Boil\_Eff.Light + Boil\_Eff.Iner + Boil\_Eff.Perm + Boil\_Eff.Vent + Boil\_Eff.Aux\_Eff + Boil\_Eff.Dist\_Ins \quad (6)$$

$$C_{Light}\ (kWh/m^2) = M_{Light} + Light + MSP + Ori + Light.MSP + Light.Ori + MSP.Ori \quad (7)$$

$$C_{Aux\_AHU}\ (kWh/m^2) = M_{Aux\_AHU} + Vent + Aux\_Eff + Vent.Aux\_Eff \quad (8)$$

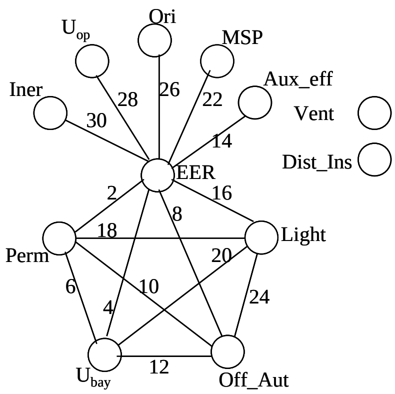
$$C_{Aux\_FCU}\ (kWh/m^2) = M_{Aux\_FCU} + U_{op} + U_{bay} + Ori + MSP + Off\_Aut + Light + Iner + Perm + Vent + Aux\_Eff + Dist\_Ins + U_{op}.Aux\_Eff + Iner.U_{bay} + Perm.U_{bay} + U_{bay}.Aux\_Eff + Iner.Off\_Aut + OFF\_AUT.Aux\_Eff + Light.Aux\_Eff + Iner.Aux\_Eff + Perm.Aux\_Eff + Vent.Aux\_Eff + Aux\_Eff.Dist\_Ins \quad (9)$$

$$C_{Aux\_Dist}\ (kWh/m^2) = MA_{ux\_Dist} + Aux\_Eff \quad (10)$$

$$C_{Off\_Aut}\ (kWh/m^2) = M_{Off\_Aut} + Off\_Aut \quad (11)$$

To find the table of Taguchi adapted to the model, two criteria must be validated [11]: criterion of the degrees of freedom and criterion of orthogonality.

We give an example for the cooling sub-model. The number of degrees of freedom is 28 (12 variables, 15 interactions, 1 for average). The number of experiments should be equal to or higher than 28. The PPCM between disjoined orthogonal actions is (4,8,16) = 16. The Taguchi table $L_{32}(2^{31})$ is then chosen - 32 simulations, 31 columns that can be assigned to variables or interactions [14] - with a resolution plan IV, i.e. variables of order 1 are aliases of interactions of order 3, and interactions of order 2 are aliases with other interactions of order 2.
The associated linear graph given by Taguchi has been adapted to our model [11]. The most significant variable in the model of cooling is obviously the EER.



RESULTS

The total consumption C_{tot} is deduced from the sub models distinguishing gas and electricity.

$$C_{tot}\,(kWh/m^2) = C_{cool}\,(kWh/m^2) + C_{heat}\,(kWh/m^2) + C_{Light}\,(kWh/m^2) + C_{Aux\_AHU}\,(kWh/m^2) + C_{Aux\_FCU}\,(kWh/m^2) + C_{Aux\_Dist}\,(kWh/m^2) + C_{Off\_Aut}\,(kWh/m^2) \quad (12)$$

The results of the reduced model have been compared with the reference simulations (32 in the case of cooling). The precision is in the interval of $\pm$ 0.5%.
Neglecting some interactions may influence the results precision; this latter improves systematically when more interactions are taken into consideration. In the selected example, we also compared the results of the additive model (12) with the results of a complete factorial design (4).
In other words, we compare the model obtained by 112 simulations (32 for cooling, 32 for heating, 8 for lighting, 38 auxiliaries and office automation (without counting identical simulation) with the model obtained by 8192 simulations. The precision is shown in figure 4.

Table 1. Average deviation (with respect to the complete model) is 0.66 (kWh/m^2)

	-1 % ≤ Deviation ≤ 1 %	-3 % ≤ Dev.< -1 % 1 % < Dev.≤ 3 %	-5 % ≤ Dev. < -3 % 3 % < Dev. ≤ 5 %
N° of simulations (total = 8192)	6840	1330	22
%	83.50 %	16.24 %	0.26 %

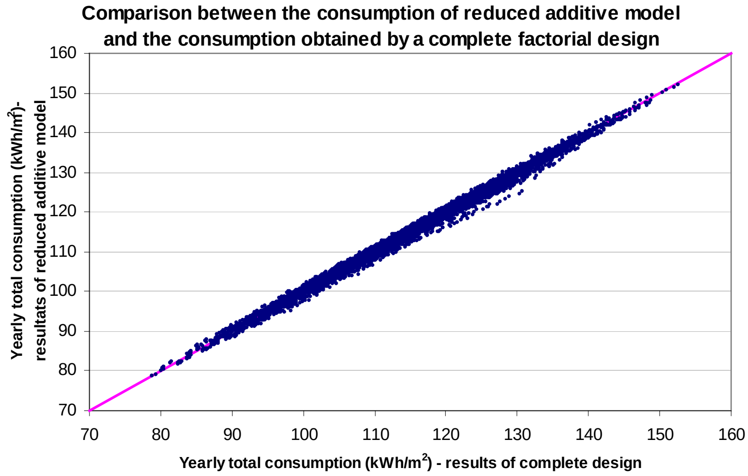
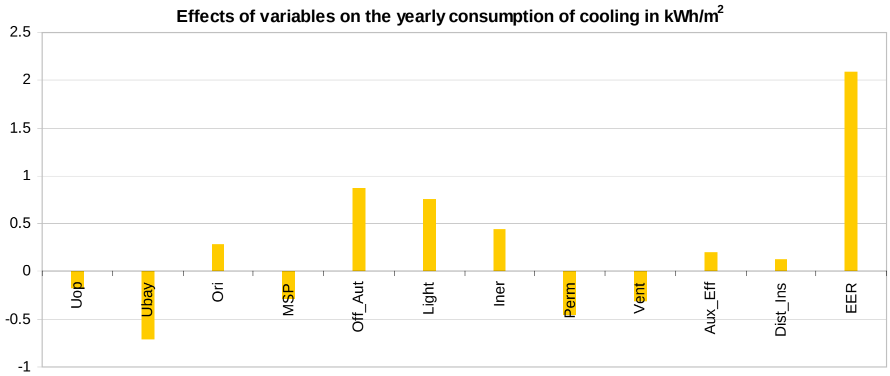


Figure 4. Precision of reduced model with respect to complete model

Typical results like effects of the elements are deduced for each model, we give that of cooling as an example.



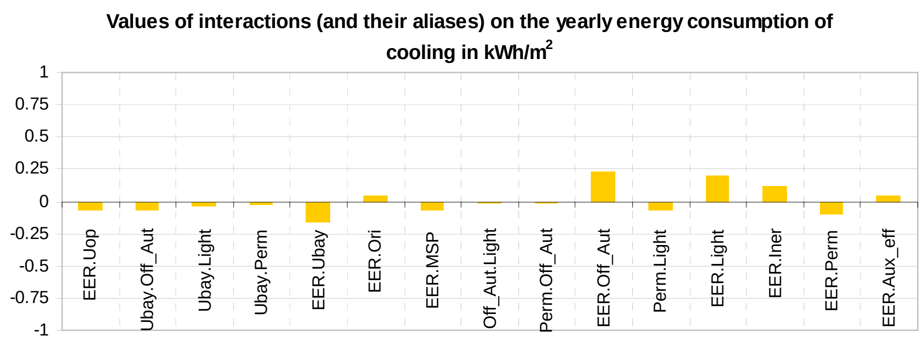


Figure 5. Example of effects and interactions values for cooling model, average = 7.8 kWh/m^2

DISCUSSION

It has been shown that the reduction of the number of simulations does not have an important influence on the quality of the answer, granted precaution with significant variables and interactions. The time requirements for running and, more importantly, preparation of simulations are greatly reduced when compared up to the factorial design. Evidently, the preparation of the fractional plan must be carried out with firm knowledge of the method. The user must be attentive while launching simulations of validation. This method is applicable to all types of buildings and systems in all sectors. The multi-parametric model is useful for an economical optimization when disposing of cost libraries of different technical solutions.

ACKNOWLEDGEMENT

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Environmental Assessment of a Low-Energy House

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SUMMARY

The paper shows the life cycle of a low-energy detached house from the viewpoint of CO_2-emissions. It compares the energy use in the course of its service life in relation to energy input necessary for its assembly and manufacturing of single building products. The service life is described using both the standardized calculation methods for energy balance of buildings and the open software tools, e.g. esp-r, as well. The input data related to embodied energy are based on information originating from selected recent works on life cycle analysis. The paper discusses the limits of the building envelope improvements in terms of thermal insulation thickness and the sense of the building environmental assessment.

INTRODUCTION

The European countries, the economy of which is based on export of industrial products and services and completely dependent on imports of fossil fuels, systematically support the improvement of energy efficiency of buildings. They do it in two basic ways:

- Normatively and legislatively, using restrictions in order to ensure the basic quality of buildings from the viewpoint of energy effectiveness, e.g. by requiring more and more improved and detailed investigation of the future energy demand for heating and hot water preparation and by suitable systems of criteria,
- Motivating, using various state and communal programs, usually the aim of which is the effective use of energy from fossil fuels and the development of alternative and ecological energy sources (solar radiation, water, wind).

These two basic instruments focus almost entirely on building performance after its assembly on the building site. Explained in terms of the life cycle of a building, the mentioned policy does not take into consideration the energy needed either for production of the building materials or for assembly of a building or for its dismantling. The main argument for this exclusive concentration on the service life of a building is that 40% of the total energy consumption is caused by operation of the buildings. The remaining 60% fall on industry and transportation, whereas 20% out these 60% are supposed to be caused by production of building materials, building processes, renovation and dismantling of the buildings. The following case study wants to show that this argumentation is no longer valid for buildings built in compliance with existing standards or even in a low-energy way. As the energy supply needed for the operation of such buildings is quite low, a significant rise of the building industry portion within this imaginary scheme should be a consequence. In this context several questions occur, e.g.:

- Whether the perception of buildings as power plants using renewable sources of energy is justified in relation to energy inputs and CO_2-emissions,
- Whether the currently used energy demand calculations should not be complemented by information on energy inputs needed for building assembly and manufacturing,

- Whether alternative and lasting (sustainable) building materials would be a kind of solution and what is the optimal insulation of the building envelope,
- Whether the energy savings by building industry in the course of manufacturing could lead to other architectural solutions than so called “low-energy design”,

and

- Whether it is possible in the design phase to consider the questions of CO_2-emissions due to the life cycle of buildings at all?

The presented study tries to find answers to the above questions. However it does not reserve the right either on completeness or absolute correctness. It is far more an attempt to realize to which extent the so called low-energy design (in the sense of highly insulated building envelope) is meaningful and where is the limit to the overflow of material – and related production energy and financial resources.

METHODOLOGY

The crucial problem of the presented comparison of CO_2-emissions due to the expected building operation on the one side and due to the built-in energy on the other side was the gathering of reliable data regarding the CO_2-emissions due to the production of building materials. Usually, the building industry does not record information on kilograms or tons of CO_2-emissions per building product, e.g. brick or window. Under circumstances these values could have been derived from the annual reports of single companies, if they had been at our disposal and had included the CO_2-emissions and the number of products per year. Unfortunately this was not the case. Therefore, some research in the libraries and on the internet was necessary. This effort showed that there had been serious research on this topic done, particularly in the German speaking countries. From a number of serious sources first of all the use of information in the Austrian Ökologischer Bauteilkatalog [1], the German Ökologische Bilanzierung von Baustoffen und Gebäuden [2] and on the website www.architektur.tu-darmstadt.de/ee [3] was made. The majority of the relevant values in all three cited publications are identical. Where differences occur (mostly steel, iron and aluminium based products) the newer information was taken. The necessary energy input for the production of building materials used for the assembly of the case study building and the related CO2-emissions were calculated using the PEI (Primary Energy Input [MJ per unit of material or product]) and the GWP (Global Warming Potential [kgCO_2-eq. per unit of material or product]) values in the cited publications. Unlike the notion PEI, which is quite comprehensible, the notion GWP should be explained at this point. The GWP represents not only a production of single CO_2-emissions but also other greenhouse gases that contribute to the global warming as well and have the same impact as comparable amount of CO_2-emissions, e.g. methane, NO_x or particles. Hence, it is more the measure of all relevant greenhouse gases converted and added up to CO_2-equivalent emissions [1].
The energy demand for heating (operation) of the case study building was calculated using the software esp-r - a kind of open software for integrated building simulation [4]. Just for the verification purposes also a calculation according to the German energy conservation decree EnEV2004 [5] was performed – primarily in order to get a plausible initial range of heating power for esp-r model and as a kind of reference. The calculations were based on climate data for Berlin. The resulting calculated annual energy demand was converted into primary energy demand and following CO_2-emissions using the conversion table published in “Der österreichische Gebäude-Energieausweis – Energiepassport” written by Professor Panzhauser et all [6]. However, the same conversion coefficients are introduced in esp-r, too. Of course, only the fossil-fuels-based CO_2-emissions were traced. As a primary energy source for heating the natural gas delivery was taken into account.

CASE STUDY

The construction of detached houses (up to 120 m^2) and apartment buildings (having flats with up to 80 m^2) is in Germany and many other countries, e.g. Slovakia very often supported by the state under the fulfilment of certain conditions. One of these conditions represents the compliance of the project with the building regulations, e.g. the minimum thermal resistances of the main building envelope components, the highest allowed transmission heat loss (thermal-coupling coefficient (H_T')) of the entire building envelope and maximum heat or even primary energy demand. The figs. 1 and 2 show the floor plans, the cross section and the characteristic elevation of the family house in consideration, the basic versions of which meet the above mentioned standard requirements both for Germany and for Slovakia as well. The basic versions were the heavy one (brick masonry combined with thermal insulation) and the lightweight one (thermally insulated timber frame-work). Each of the basic versions was then modelled and simulated with four other combinations of the thermal insulation thicknesses according to the table 1.

Tab.1. Combinations of building envelope elements modelled

		Brick house					**Lightweight house**				
Mean U-Value:	[W /(m2K)]	0.41	0.32	0.30	0.29	0.28	0.41	0.32	0.30	0.29	0.28
U-values of single parts of the building envelope [W /(m2K)]	Base plate	0.76	0.48	0.48	0.48	0.48	0.76	0.48	0.48	0.48	0.48
	Walls	0.25	0.19	0.15	0.13	0.11	0.25	0.19	0.15	0.13	0.11
	Roof	0.17	0.15	0.14	0.14	0.12	0.17	0.15	0.14	0.14	0.12
	Windows	1.09	1.09	1.09	1.09	1.09	1.09	1.09	1.09	1.09	1.09
	Entrance door	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50	1.50
Thermal insulation thickness [m]	Base plate	0.020	0.050	0.050	0.050	0.050	0.020	0.050	0.050	0.050	0.050
	Walls	0.100	0.150	0.200	0.250	0.300	0.140	0.190	0.240	0.280	0.330
	Roof	0.220	0.250	0.275	0.275	0.300	0.220	0.250	0.275	0.275	0.300

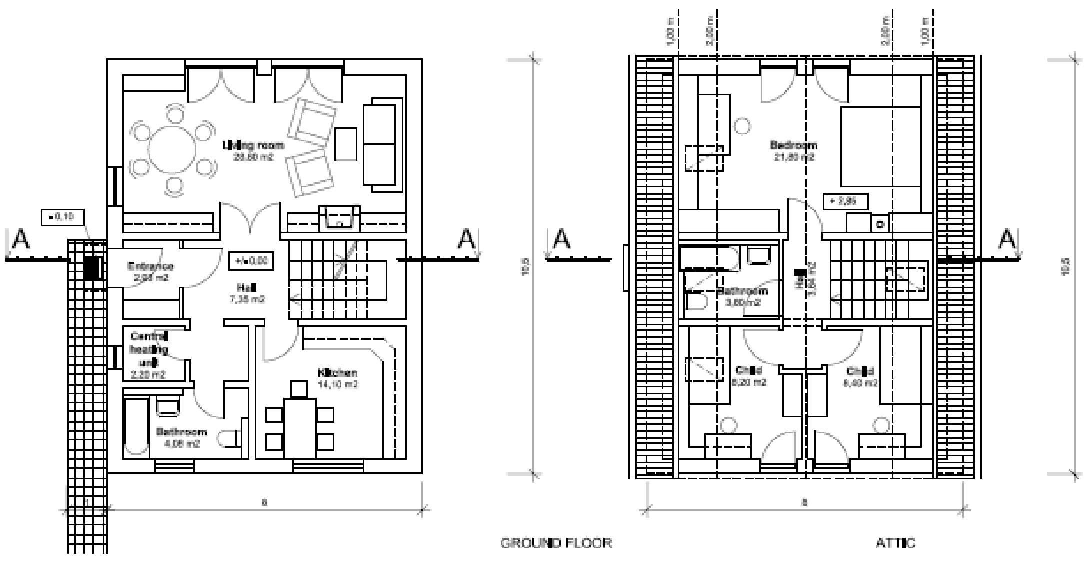

Fig.1. Floor plans of the investigated detached house

The U-values of the windows remained in all calculations the same. The simulations were performed using esp-r. The fig. 3 introduces pictures of the one zone model of the case study house generated by the Radiance [7] module within esp-r. The ventilation heat losses were set to 0.5 air change per hour. The heat gains are represented by occupant sensible ones only (fig. 4). They stand also for heat gains due to equipment and light in order to keep the model as simple as possible. The required indoor air temperature was set to 20°C and an ideal zone heat control was chosen.

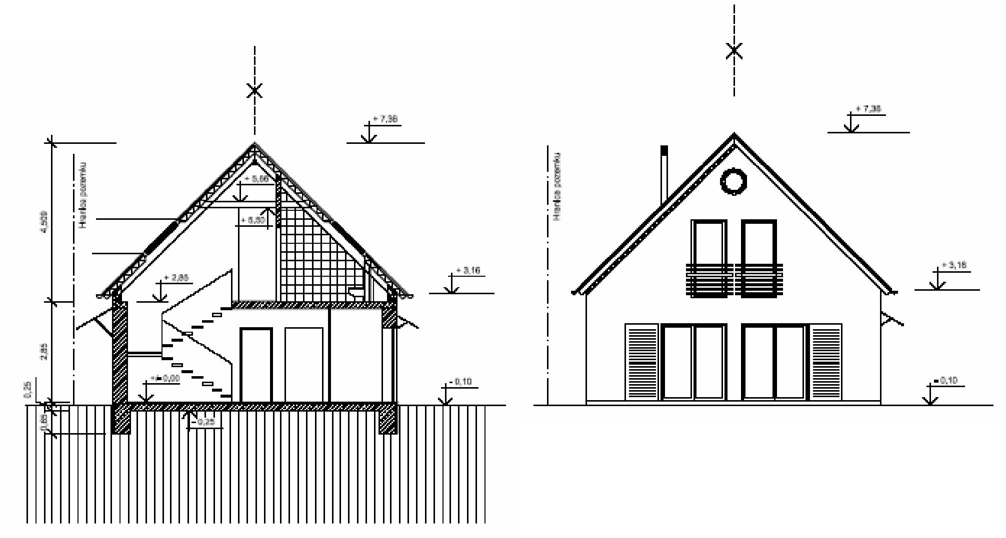

Fig.2. An elevation and the cross-section of the investigated detached house

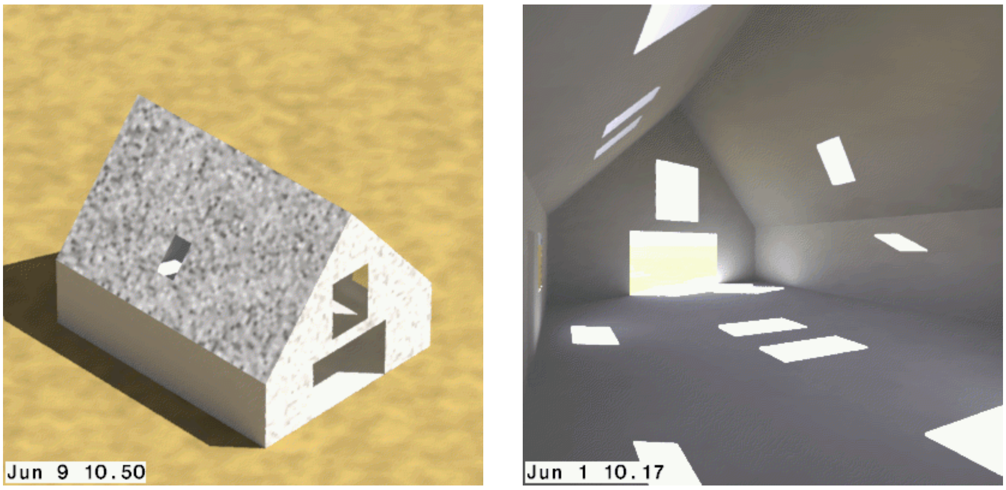

Fig.3. Visualization of the one zone model of the case study house

In addition to the simulation of the heat energy demand an ecological balance model (fig.5) for each of the 10 combinations (5 for each basic version) has been set up and the PEI and GWP values calculated. In the balance models the transportation of final products from the factory / the selling place to the client and the processes at the building site, e.g. the production of shuttering, the formworks or the use of machines were not considered.

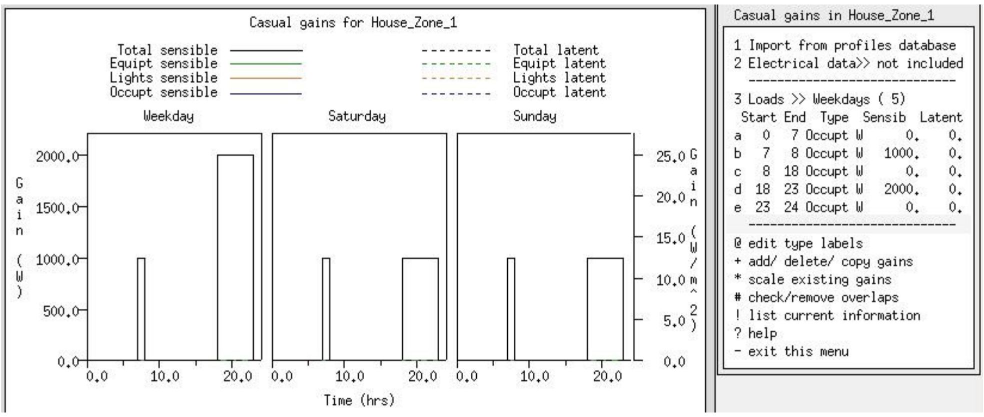

Fig.4. Heat gains distribution over the week

ECOLOGICAL DATA PROFILE:											
Envelope element / Layer	Thickness	Area	Volume	Weigth	Density	Density	PEI/unit	PEI / kg	PEI	GWP 100 (1994)	GWP 100 (1994)
	d	A	V		ρ (D)	ρ	D	material	Total	kg CO_2 eq. / unit (D)	kg CO_2 eq. / kg mat.
	[m]	[m²]	[m³]	[kg]	[kg/m³]	[kg/m³]		[MJ/kg]	[MJ]	[kg CO_2-eq./kg]	[kg CO_2-eq./kg]
Foundations											
thermal insulation horizontal (Rockwool)	0.02000	66.600	1.665		85.000	85.000	17.500	17.500	2476.688	1.200	1.200
thermal insulation vertical (XPS)	0.05000	50.400	2.520		37.000	37.000	101.000	101.000	9417.240	3.600	3.600
waterproofing (PVC,0,002 m; 1kg/m2)		80.000		80.000	1500.000	1500.000	63.000	53.000	4240.000	2.200	2.200
antiradon layer (aluminium,0%Rec.,0,24 kg/m2)		80.000		19.200	2700.000	2700.000	200.000	714.000	13708.800	13.000	36.000
normal concrete (slab + foundations)			42.000		2000.000	2000.000	0.800	0.800	67200.000	13.000	0.130
normal concrete - floor (8 x 10 x 0,03 m)			2.400		2000.000	2000.000	0.800	0.800	3840.000	13.000	0.130
gravel			6.000		2500.000	1800.000	1.640	0.100	1080.000	0.002	0.000
Total:									101962.728		
Walls											
exterior plaster (Silikatputz)		160.000	0.480		1500.000	1800.000	1.783	1.500	1296.000	0.299	0.200
thermal insulation (Rockwool)	0.10000	160.000	16.000		149.000	149.000	17.000	17.000	40528.000	1.400	1.400
30 cm bricks	0.30000		27.800		750.000	750.000	2.600	2.600	54210.000	1.300	0.130
25 cm bricks	0.25000		8.860		750.000	750.000	2.600	2.600	17277.000	1.300	0.130
17,5 cm bricks	0.17500		3.520		750.000	750.000	2.600	2.600	6864.000	1.300	0.130
premade brick carrier ((0,12 x 0,065)			0.257		1887.000	1887.000	3.500	3.500	1699.998	0.300	0.300
above windows and doors)											
normal concrete			1.000		2000.000	2000.000	0.800	0.800	1600.000	0.130	0.130
reinforced concrete			1.110		2400.000	2400.000	0.800	0.800	2131.200	0.130	0.130
reinforcing steel				100.000	7850.000		24.000	36.000	3600.000	1.700	2.400
thermal insulation (XPS, d = 0,05 m)	0.05000	5.000	0.250		37.000	37.000	101.000	101.000	934.250	3.600	3.600
chimney (75,1 kg/m²)		1.88		140.81			338	338.000	633.75	27.200	27.200
interior plaster (Kalkputz, d = 0.002m)		350.000	0.700		1300.000	1200.000	1.136	1.500	1260.000	0.136	0.200
Total:									132034.198		
Ceiling											
Ceiling facing bricks			5.760			750.000		2.600	11232.000		0.130
ceramic ceiling bricks [100 kg/m²]		70.000		7000.000				2.700	18900.000		0.300
ceramic ceiling carrier [16 kg/m] x 108 m				1728.000				3.500	6048.000		0.300
reinforced concrete			0.530			2400.000		0.800	1017.600		0.130
normal concrete (8 x 10 x 0,05 m)			4			2000.000		0.800	6400.000		0.130
reinforcing steel				370.000	7850.000		24.000	36.000	13320.000	1.700	2.400
Total:									56917.600		

Fig.5. Ecological balance model of one of the 10 combinations in tabular form

INTERPRETATION OF RESULTS

The diagram 1 shows the comparison of the CO_2-emissions due the energy demand for heating between the heavy (also called "brick house") and lightweight variant of the case study house (the compared bars are always for the same mean U-Values of the building envelope). The brick house performs slightly better due to the higher thermal capacity. This difference would be probably higher if the ceiling between ground floor and attic was simulated. Unlike this information, which is already quite known, the diagram 2 explains the relationship between the improvements of the mean U-values by increasing the heat insulation thicknesses on one side and the reduction of energy demand for heating on the other side. It is quite obvious that the linear reduction of the heat energy demand is achieved by the geometrical increase of the heat insulation thickness. Somewhere between the mean U-Values 0.30 and 0.32 W/(m^{2}K), which corresponds to approx. 17.5 cm of thermal insulation of the brick house and approx. 21 cm of the lightweight house, stops then the rational increase of the thermal insulation thickness. Just to compare: for the improvement of the mean U-Value from 0.41 to 0.32 W/(m^{2}K) and the reduction of heat energy demand in the range of 1500 kWh/a 5 cm of an additional thermal insulation is necessary. But the next increase of the thermal insulation by 5 cm brings about the improvement of the U-Value by just 0.02 W/(m^{2}K) and the reduction of heat energy demand in the range of 300 – 400 kWh/a. This leads to the conclusion that the efficiency of the increase of the thermal insulation thickness has its economical and also ecological limits as it will be obvious from further diagrams. They can be quite well established using the computer simulation tools.

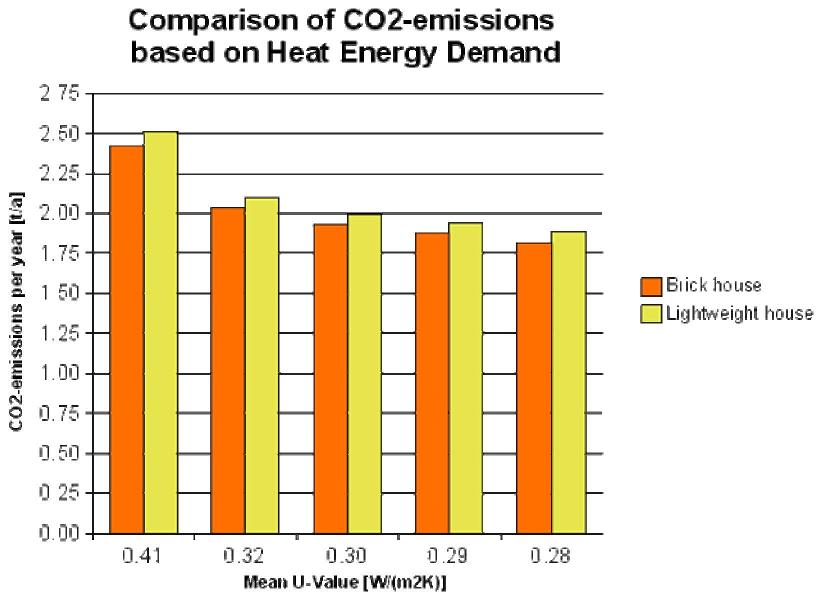

Diagram 1. Comparison of the CO_2-emissions based on energy demand for heating against the mean U-value of the heavy ("brick house") and lightweight variant of the building envelope.

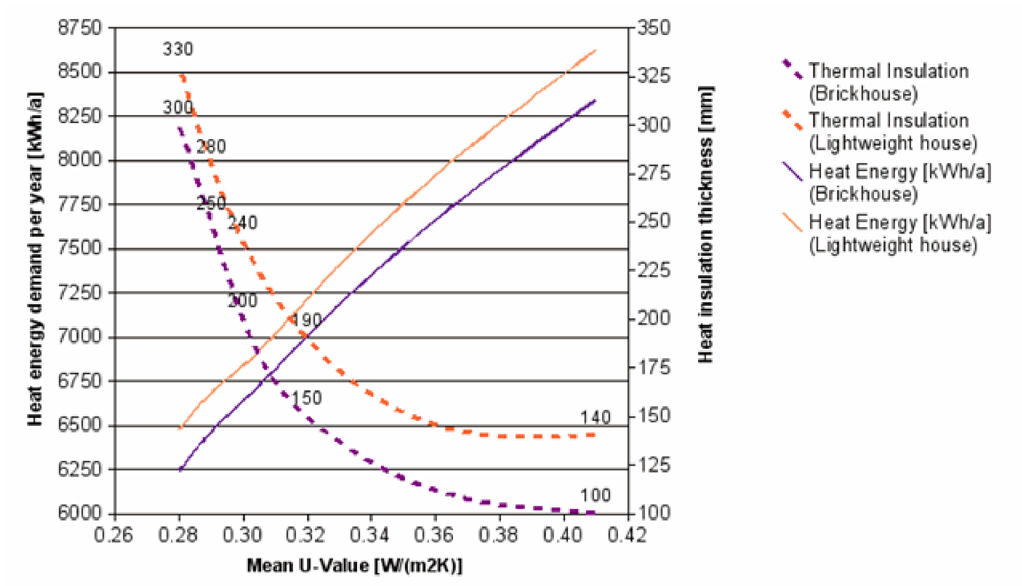

Diagram 2. The effect of the mean U-value improvement in dependence on heat insulation thickness upon the heat energy demand reduction.

Diagram 3 is a summary of the performed study and depicts the increase of the CO_2-emissions due to the building and operation of the case study house for both the heavy (B – brick house) and the lightweight (LW) variant as well. For each variant 5 modifications of the thermal insulation thickness (in diagram represented by the mean U-values) were examined. The starting values of the lines are the calculated GWP values due to the fabrication of the construction material and building products. The inclination of each line is then caused by the annual CO_2 output due to the building operation.

Looking at it two basic information can be deduced:

– The improvement of the mean U-Value, and hence the thermal insulation thickness, under 0.32 W/(m^{2}K) does not have significant effect on CO_2 reduction within the first 25 years of operation. On the other side the GWP value due to the fabrication of construction material and building products can increase in case of heavy construction by up to 16 % and in case of lightweight construction by about 30 % for the U-Value of 0.28 W/(m^{2}K) against base variant (U-Value = 0.41 W/(m^{2}K)). Provided that the building operation is based on renewable energy sources these lines would be shallower and closer to constant value. Therefore it might have sense to introduce a reasonable system of limitations on the initial GWP values due to the fabrication of construction material and building products. This would likely have positive effects on:
  - Larger use of environmentally friendlier construction material and building products,
  - Increasing the moral obligation to use renewable energy for building operation.

– The lightweight variant seems to perform environmentally better than the heavy one. It is then purely the question of expected building service life and material durability that is to consider primarily when doing the decision between heavy and lightweight variant.

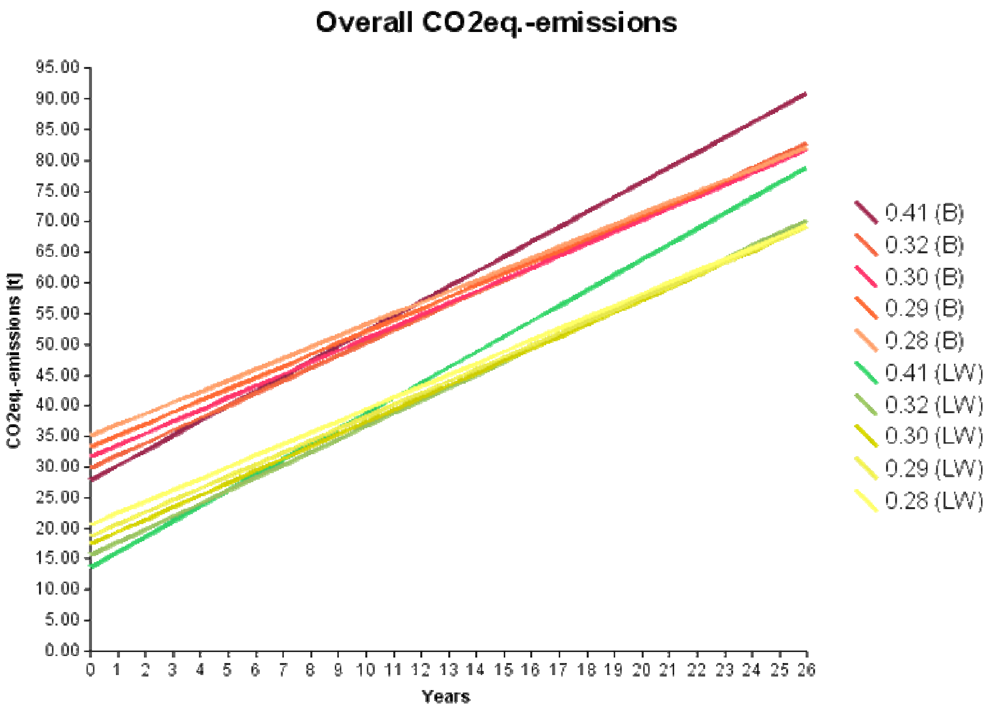

Diagram 3. The CO_2-emissions due to manufacturing and operation of the case study house calculated in dependence on time for both the heavy (B – brick house) and the lightweight (LW) modifications of the building envelope (represented by the mean U-values).

CONCLUSIONS

The current exclusive focusing on the energy efficiency of the building operation leads to heavy insulated building envelopes and to the utilization of alternative renewable energy sources (mostly on a decentralized basis). In principle this trend is right as the good insulated building envelope is a basic precondition for efficient use of energy, regardless if it comes from conventional or renewable sources. However, as the above case study tried to show the increasing the thickness of thermal insulation and the improvement of the mean U-value are effective to certain extent only. Then it becomes a kind of useless overflow that just cost investment money. Providing that the building envelope is optimized and the use of renewable energy sources is made, the third way to improve the environmental harmlessness of the buildings is to use materials and products the production of which is as much CO_2 free as possible. In order to do this a kind of environmental product declaration for buildings (EPD) should be introduced as a complementary measure to energy passports of buildings that are being prepared at the present. In fact the work and first discussions on EPDs for buildings have already started, too [8]. As the energy passports are quite complex matter and a subject to strong criticism the author's opinion is that the EPD's should exclusively cover the period of the building's life cycle prior to its commissioning. The other option would be an EPD that would consist of the energy passport and of some other document that would describe the environmental quality of the building manufacturing before its commissioning. Whatever the decision will be, there is a lot of reliable information on the environmental quality of building products and materials already available. It can help progressive architects to assess and optimize the design quality from both points of view - the building operation and the building manufacturing as well. Perhaps, as a consequence, a new architectural style based on optimally insulated buildings supplied from central green power plants could originate.

ACKNOWLEDGEMENTS

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A voluntary scheme for certification of indoor environment and energy use

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SUMMARY

Specialist researchers, property owners, builders and building managers have together developed a quality assurance (QA) management scheme that considers indoor environment and energy use. The primary objective of quality assurance is to work towards continuing improvements and to encourage those concerned to perform measures that otherwise would not have been considered, and to ensure that energy improvements are not introduced at the expense of indoor environment conditions. The QA scheme therefore aims to be flexible so that it can achieve the primary objective independently of the building's category or the management organisation. The new QA scheme's flexibility has successfully been tested in three different buildings; a school building, an office and an area of multi-family houses.

INTRODUCTION

To achieve the intended results of building, managing and using a property requires knowledge, continuity and communication, which can be assured by a dynamic and flexible quality assurance management scheme. Such a voluntary scheme focusing on high quality indoor environment has been developed during the 1990s and been successfully applied to schools, offices and multi-family houses [1]. Clients have been very satisfied with the scheme and its results in terms of an improved indoor environment, with fewer complaints from the building users [2, 3].

However, new emphasis on energy conservation (such as in the European Energy Performance in Buildings Directive [4], has added new demands for energy improvements as well. On the other hand, a reduction in energy use is appropriate only if it does not adversely affect the indoor environment. In order to avoid a one-sided focus on either good indoor environment or energy efficiency that might result in mutually adverse effects, the building sector requested that the QA scheme should be extended to consider energy use as well.

Specialist researchers, property owners, builders and building managers have therefore jointly developed the QA scheme with the objective of including energy efficiency assurance [5]. The scheme has been extended to a labelling scheme for the total building performance of both indoor environment and energy use. It includes methods and routines to control the indoor environment and energy use by using occupant questionnaires, the building monitoring systems or other methods during operation. A third party certifies the energy and indoor environment and makes annual inspections. To ensure that the scheme's rules are accepted, and that they are needed by the building sector, the scheme has been approved by a committee consisting of representatives of private and municipality property owners. However, the main target group of the end results is the occupants. It is important to occupants to know that their potential home or workplace building has a healthy indoor environment with minimum use of energy. The new QA scheme for both indoor environment and energy use is now ready to be applied in practice.

The system covers the planning, design, construction, commissioning and operation phases, and it would be natural to perform a performance analysis of all phases of the extended scheme. However, to do so, considering both the indoor environment and energy use, would have required a very long trials period. In order to obtain a first relatively quick evaluation, the extended scheme was applied to buildings which formed a special case in that their indoor environments had already been certified with the QA scheme. They therefore needed only the additional element of QA of their energy schemes. The pilot buildings were chosen in order to represent a wide range of building categories and property manager organisations. The pilot projects are a school building, an office and an area of multifamily houses.

DESCRIPTION OF THE QUALITY ASSURANCE SCHEME

The primary objective of the scheme is to work towards continuing improvements and to encourage clients, builders, architects, administrators and occupants to perform measures that otherwise would not have been considered. This requires quantified and measurable goals, action plans for measures, and management systems during operation.

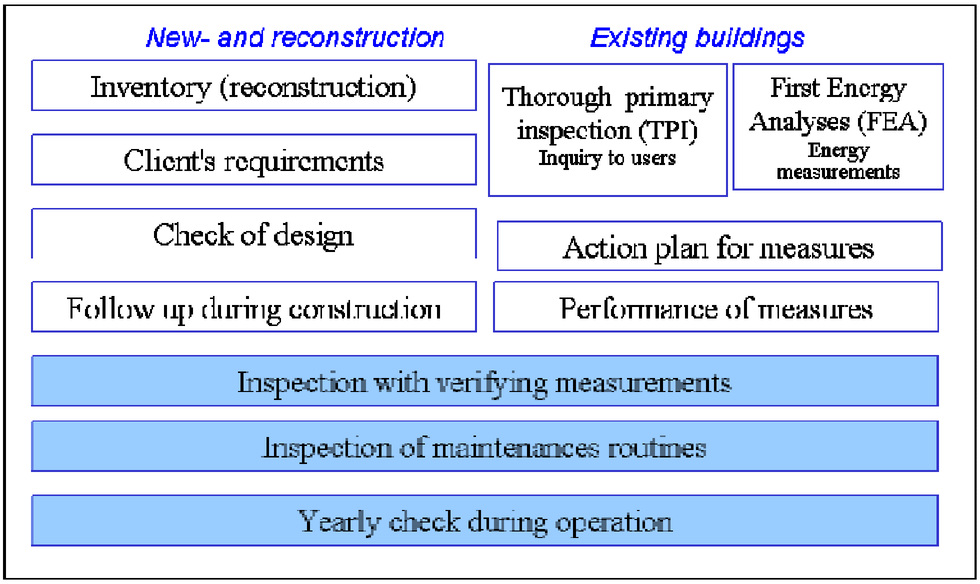

Figure 1. Illustration of procedures for quality assurance of indoor environment and energy use.

Cooperation between all parties, from scientists and public authorities to designers, contractors, managers and users, is important for the end results. All need to listen to, and to learn from, each other. Good indoor environment and efficient use of energy can be achieved by creativity, planning and layout design, choice of materials, general designs and detailed and overall designs of systems for heating, ventilation, electricity and water supply. This QA scheme makes sure that the requirements set out in legislation, standards or common codes of practice are fulfilled as intended. An independent third party supervises, evaluates and checks that the requirements are fulfilled. Measurements show that the performance requirements have been met. Occupants' perceptions of the indoor environment are evaluated with the help of questionnaires, while energy use is evaluated with energy measurements or energy bills. The QA system includes new building and reconstruction work, as well as improvements of existing buildings, and covers the entire process, from planning and design, through the construction stage to final use and operation. The certification is based on ISO 9000 procedures, is illustrated in Figure 1 and described in SPCR 114E [6].

Additional elements for energy use

Certification work for existing buildings begins with a first energy analysis (FEA), which consists of an inventory of the actual property with its actual energy status, energy aspects and energy performance [7]. This can be done by examining construction drawings, operational follow-up programs, control systems or other documentation; inspections, interviews with staff and additional measurements. It is recommended that this should be carried out in conjunction with the TPI (Thorough Primary Inspection) of the indoor environment, particularly with respect to visual inspection and interviews with staff.

The results of the FEA are then used to set objectives to be achieved, including a performance measurement specification of how the comprehensive targets should be measured and checked. The QA scheme considers the fact that each building project is unique, and therefore the annual energy use target will be set on the basis of the building's current condition and its associated limitations, rather than on the basis of a specific predefined figure. For new construction, or major reconstruction, the primary limitations relate to building use and climate, while minor reconstruction and existing buildings also have limitations due to design.

The next step is to draw up an action plan for measures and carry them out in order to reach the set targets. Experience shows that successful energy efficiency in a building will be maintained only if the building is efficiently managed, operated and maintained, with all parties steadily improving their performance and with the results regularly monitored. This means that the energy target must be regularly monitored and reviewed, and the QA scheme is therefore based on a management system modelled on a Swedish standard (SS 62 77 50, [8]). The standard includes comprehensive routines for energy management for any organisation and has therefore been refined and customised to fit the building sector.

DESCRIPTION OF THE PILOT PROJECTS

The pilot projects are a school building, an office and an area of multifamily houses.

The Sjöbo School

The main classroom building of Sjöbo School (originally built in 1959) was rebuilt in 2001 and 2002, complemented by new building of the gymnasium, kindergarten, teachers' staff rooms etc. In total, the school consists of six buildings, with a floor area of 6673 m^2. The indoor environment element of the QA scheme was employed during the planning, design, construction and commissioning stages, and after some months of operation the school indoor environment was certified. As far as energy use is concerned, the school should perform quite well, since all buildings were rebuilt or newly built, with new building envelopes and HVAC systems. However, the focus of this project was to introduce the extended QA scheme in order to ensure retention of good building performance rather than (primarily) to improve energy use.

Figure 2. The Sjöbo school.

The Elektra Office

The Elektra building was built in 1894 for Borå's steam engine-driven power station. In 1970, it was converted to office premises. After the last refurbishment in 1990, the building presents a modern impression, while at the same time reminding us of older days with its large windows and small chimney towers. The 2100 m^2 interior was certified according to the indoor environment QA scheme in 2003. The owner and property manger is the town's district heating company, with good knowledge of energy efficiency measures and therefore already having clear ideas about how to improve the energy performance, with respect to the building's age, during the extension of the QA scheme to cover energy use.

Figure 3. The Elektra office.

The Högsbohöjd area with multifamily houses

The Högsbohöjd district consists of 14 multi-family houses with 940 apartments and a total area of 57 817 m^2. The district was built during 1959 to 1961 and was totally renovated in 1992. Part of the buildings were certified according to the indoor environment QA scheme in 2001, and the last building in 2006.


Figure 4. The Högsbohöjd district with multifamily houses.

TARGET DETERMINATION AND SPECIAL CONSIDERATION FOR EACH PILOT PROJECT

The three pilot projects are very different in their character regarding building categories and property manager organisations. They also had different additional aims with the introduction of the QA scheme and ways of working during the introduction.

The Sjöbo School

The property manager, based in the municipality and responsible for management of the Sjöbo school, worked very hard with getting all parties involved in the management process with several meetings between the principal, the operation staff at the school (caretaker, cleaner, etc.), teachers' and occupants' representatives and the municipality property manager. Besides this series of regular meetings with standing agendas, two consultants had been engaged to help with computerisation of the building management system, installation of new sensors and instrumentation equipment and commissioning the building control system. One of the main measures that was introduced was time-control and demand-control of the ventilation system in the kindergarten and the gymnasium. This required special consideration on how the ventilation should be controlled while the indoor environment was kept at a high level, which is described in Wahlström et al. [9].

The FEA showed that the annual energy use of district heating was 121 kWh/m^2, and that of electricity was 59 kWh/m^2. The target was a 10 % reduction during the first year, and 15 % after two years, while the use of electricity should not exceed the previous level. After one year, the first target was met for district heating, while electricity use had also been reduced by a few percentage points.

The Elektra Office

The property management structure for the Elektra office is a very flat organisation for management of the building, and it was simple to introduce a management system with rules and responsibilities since there was already an environmental management system. At the same time, it was very easy to monitor the energy performance, since the office consisted of only one building. The building owner had, however, an additional aim with the introduction of the management system, which was to analyse how energy use could be controlled using the company's own internet-based statistical monitoring tool. Monthly and yearly energy performance was evaluated with the internet tool, in the form of graphs and alarms for easy follow-up, during the introduction of the QA scheme.

The work involved in the Elektra building consisted of replacement of the building control system, modifications to the ventilation, heating and cooling control system, and calibration of sensors. The target for energy performance was set at a 15 % reduction in heating energy, a 7 % reduction in electricity use, and a 10 % reduction in cooling energy use during the first year.

The Högsbohöjd district with multifamily houses

The property manager for the Högsbohöjd district wanted to work in a different way with both organisational and target set-up. The main approach with their energy reduction work was to first consider and work with the buildings that had the highest energy use, and thus also the highest potential for energy improvements. This enabled the current year's investment costs to be used where they gave the most benefits. Targets were therefore set up for all 14 buildings together, instead of an individual target for each building, but with the limitation that the energy use was not allowed to increase in any building. An action plan for work was set up with the following main features; adjustment of heating, replacement of windows, insulation of windows, insulation of attics, hot water metering and improving the airtightness of the building envelope. However, what was not defined was in which building the various measures should be performed. Each year, a few buildings with the highest energy demand will be examined in order to decide which energy measures would be of most benefit to that particular building.

Today, the Högsbohöjd district has an average specific district heating demand of 169 kWh/m^2, and has set a target of 149 kWh/m^2 for the whole area by 2010. Electricity use is 21 kWh/m^2, and the target here is prevent any increase by 2010.

The organisation for management of the Högsbohöjd district consists of a chief property manager, based in the head office, and staff known as local landlords who deal directly with daily maintenance within the buildings. If anything acute happens that need to be dealt with immediately, the owner has a contract with a service company. The local landlords check the energy performance every month and report deviations directly to the chief property manager. At yearly evaluations by the chief property manager, buildings with high energy use are selected for further investigation by energy auditors. This can also be the case if a local landlord has recognised a large monthly deviation. The energy auditors examine the selected buildings and give suggestions for necessary improvement or remedial work.

RESULTS AND DISCUSSIONS

New demands for energy improvements must not be allowed to draw attention away from the indoor environment. The preliminary results from the pilot projects show that this can be done by using a practical and flexible quality assurance scheme, intended to design and maintain a good indoor environment and efficient energy use.

The primary objective with the QA scheme is to work towards continuing improvements and encourage property managers, administrators and occupants to perform measures that otherwise would not have been considered. The QA scheme aims to be flexible so that it can be used for different building categories, different organisational structures and different parts of the building process. This report describes application of the QA scheme to buildings that already employ the indoor environment management scheme, but in three different kinds of pilot projects.

The results show that the QA scheme is really flexible. For each pilot project it was possible to introduce quantified and measurable goals, action plans for measures and management systems during operation with authorities, responsibilities and awareness for all actors within the process. All pilot projects have already shown, with their new targets and action plans for work, that they are moving towards improved energy performance. The projects have also led to a significant improvement in the systematic work related to the controlling energy use, while at the same time maintaining a good indoor environment.

The main target group of the end results is the occupants. It is important to occupants to know that their potential home or workplace building has a healthy indoor environment with minimum use of energy.

The next step is to test the flexibility of the QA scheme during planning, design, construction and commissioning of different building categories.

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Simplified methods to evaluate energy use for space cooling in the energy certification

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SUMMARY

Both on European and national scale, standards are now being drawn to extend to summer air conditioning the evaluation of energy requirement in building-plant systems. To this aim different simplified methods have been taken into account, anyway all similar to the utilization factor method already used for the winter season.
The results obtained by the application of these methods in the case of typical buildings are therefore compared here with those from a dynamic simulation of building-plant system by means of comprehensive computer programs like Energy-Plus. The analysis points out the possibilities but also the limits of these simplified methods.

INTRODUCTION

Comprehensive software programs to simulate building energy performances in dynamic conditions are available today for professional purpose. But the use of simplified methods is still diffuse and recommended or sometimes even compulsory in various national standard. Also in the new European standard [1], in draft , for the calculation of energy use for space heating and cooling are foreseen both quasi-steady state methods and dynamic methods. At national level it may be decided which of these methods is or are allowed to be used. Eventually depending on the purpose of the calculation and the complexity of building. With regard summer air conditioning, the actual tendency is to elaborate a simplified approach similar to the one used for heating requirements calculation and based on the utilization factor method. For example this method is already present in the standard EN 832 [2]. Naturally the application of this technique to the more complicated summer case is destined to increase the problems, the doubts and the criticisms which have appeared over the winter calculation. The possible application of the utilization factor method for the cooling calculation is therefore discussed here on the basis of the comparison with the results obtainable with the EnergyPlus program [3] . This is a dynamic simulation program of the building-plant system based on the most popular features and capabilities of BLAST [4] and DOE-2 [5] programs. Its development is supported by the US Department of Energy and actually it can be considered one of the most reliable software codes for building energy simulation.

THE APPLICATION CASES

For our investigations about summer requirement evaluations some typical application cases have been analysed . As the fundamental aim is to investigate the influence of the heat gains on the simplified calculation of the cooling needs, we have concentrated our attention about office buildings normally characterized by greater heat fluxes than dwellings. Three buildings with different shapes have been considered: a single unit, an horizontal serie of units and a vertical serie. The three buildings are sketched in figure 1. One of the largest façades is always South oriented. The conditioned volume is respectively 432 , 10368, 9720 m^3 for the three buildings. Typical envelope structures have been assumed , for example brick walls well insulated as required by the current Italian laws. Double glazing 6+13+6 mm uncoated, clear and air filled with a thermal transmittance of 2.71W/m^2K and a SHGC=0.70. Further details can be found in [6]. Three different amounts of external glazing have been considered: 10%, 50% and 100% of the external surfaces for the single unit, 20%, 50% and 100% for the horizontal and vertical volume. The air conditioning can be continuous or intermittent, from 7 am to 7 pm. (12h).

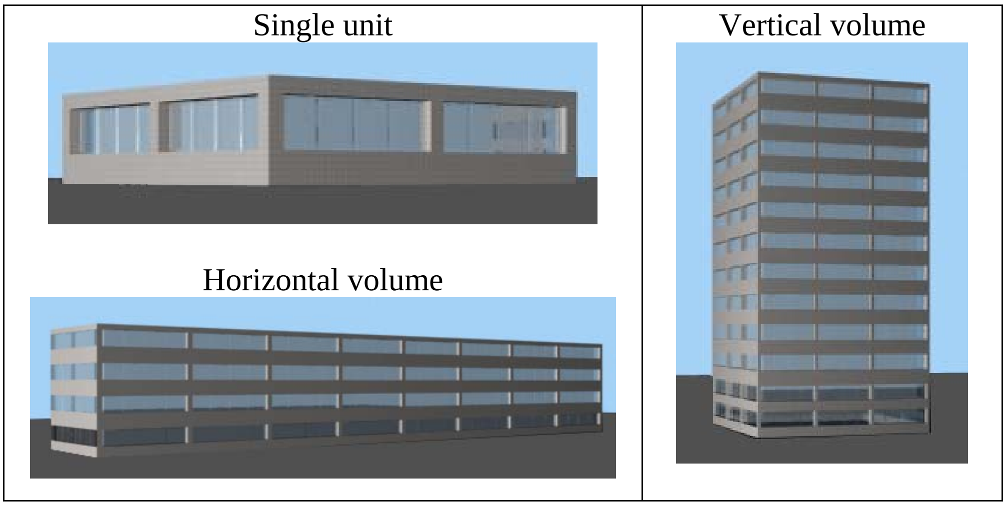

Figure 1 A view of the analysed buildings

Forced ventilation equal to 2 vol h^{-1} is present only in the period from 7 am to 7 pm. Internal heat gains for persons (maximum 1 person 10 m^{-2}), lighting (maximum 20 W/m^2, fluorescent) and different maximum electric machine gains (computers): 12, 48 W/m^2. Typical daily

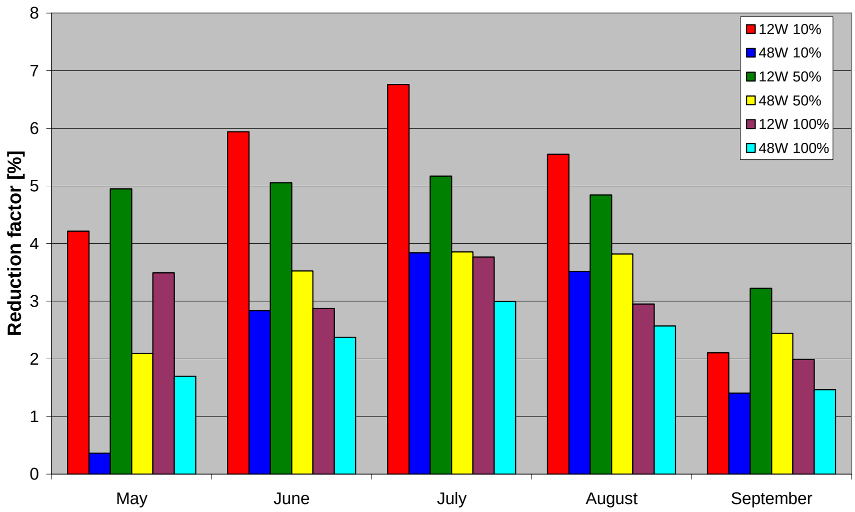

Figure 2 Percentage reduction of the cooling needs with intermittent instead of continuous air conditioning for the single unit in Venice with various electric gains and glazed quota.

scheduling for office buildings are foreseen. The indoor air design temperature is 26°C, a standard value in Italy. The air conditioning period analysed is from May to September (5 months). The procedure to calculate energy requirement for humidity control will not be discussed here even if latent loads normally have an important effect on overall plant performance. In fact hygrometric load calculation is normally easier to do by a mass balance on the indoor air. The following considerations will refer only to sensible load estimation whose determination is undoubtedly more critical. For the simulation the weather of two different localities in Italy: Venice and Palermo have been utilized. They are assumed representative of the climate in the central and southern part of Europe.

In figure 2 the percentage reductions of the seasonal cooling needs with the intermittent instead of continuous air conditioning are presented in the case of single building with different internal electric gains and external glazed surface ratios. Unlike the winter case, the passage from continuous to intermittent plant operation does not cause noticeable differences in the seasonal requirement because the heat gains are normally concentrated on daytime. In this first phase of the analysis therefore, we have concentrated the attention on the plant continuous working case. All the results presented are referred to the continuous cooling.

THE UTILIZATION FACTOR METHOD

The monthly heating requirement for each zone Q_h of a building is calculated using the following correlation:

$$Q_h = Q_l - \eta_u \cdot (Q_{si} + Q_i) \qquad (1)$$

where:
Q_l is the total heat loss, sum of transmission and ventilation losses (J)
Q_{si} is the solar gain through fenestrations (J)
Q_i is the internal gain (J)
η_u is the utilization factor of the heat gains
The utilization factor is a reduction factor for the heat gain introduced to take into account of the dynamic behaviour of the building. The parameters having the greatest influence on the utilization factor are:
- the gain/loss ratio γ which is defined as:

$$\gamma = \frac{Q_{si} + Q_i}{Q_l} \qquad (2)$$

- the time constant τ which characterizes the internal thermal inertia of the heated space:

$$\tau = \frac{C}{H} \qquad (3)$$

where C is the effective internal thermal capacity (J/K).
The utilization factor η_u is then calculated with the following equations:

$$\eta_u = \frac{1-\gamma^{\alpha}}{1-\gamma^{\alpha+1}} \qquad \text{if } \gamma \neq 1 \qquad (4)$$

$$\eta_u = \frac{\alpha}{\alpha + 1} \qquad \text{if } \gamma = 1 \qquad (5)$$

where α is a numerical parameter depending on the time constant τ.

$$\alpha = \alpha_0 + \frac{\tau}{\tau_0} \qquad (6)$$

The values of α_0 and τ_0 are provided by the standard.

THE APPLICATION TO AIR CONDITIONING

In analogy with winter calculation the review of the standard EN 13790 proposes the evaluation of the cooling requirement Q_c. In the summer period the heat gains become the fundamental cause of the air conditioning needs, instead the quantity Q_l due to transmission and ventilation normally reduces the cooling load. In fact, in the European climate the average monthly temperature of the outdoor air is lower than the indoor design temperature which is for example in Italy equal to 26°C. Therefore for each zone and in the case of continuous air conditioning, at the present time the standard suggests the following equation:

$$Q_c = (Q_{si} + Q_i) - \eta_u \cdot Q_l \qquad (7)$$

where η_u becomes an utilization factor for heat losses and it is calculated again as a function of γ and τ with different values α_0 and τ_0.

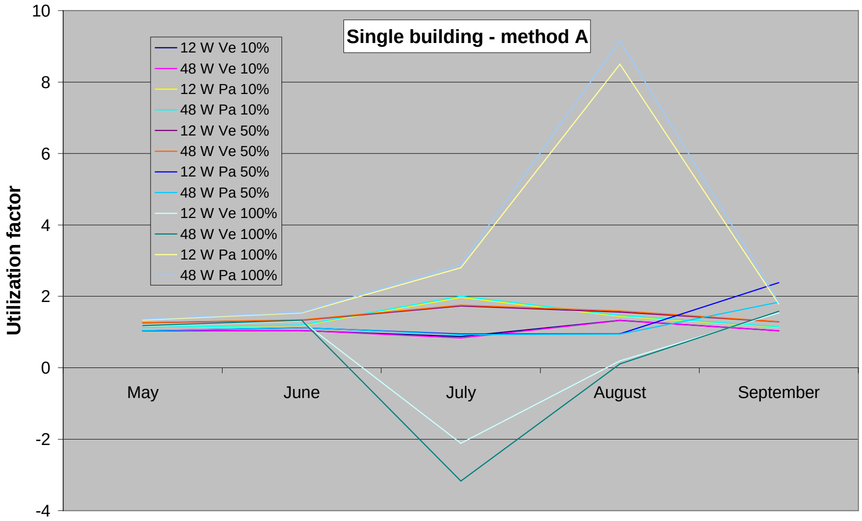

Figure 3 Utilization factors with method A for the single unit in the various cases

If these losses are negative (γ <0) the utilization factor will have the value one and consequently the negative losses are totally added as gains. By this procedure η_u can be less o equal one. An investigation on this procedure, here called method A, is carried on, starting, on monthly basis, from the cooling demand Q_c , heat losses Q_l and heat gains (Q_i + Q_{si}) obtained by dynamic simulation with EnergyPlus program. In this way it is possible to calculate the corresponding utilization factor necessary with method A in order to obtain the same Q_c . In the figures 3, 4, 5 this utilization factor is reported for the three considered buildings respectively and for the different localities, internal gains and glazed quota.

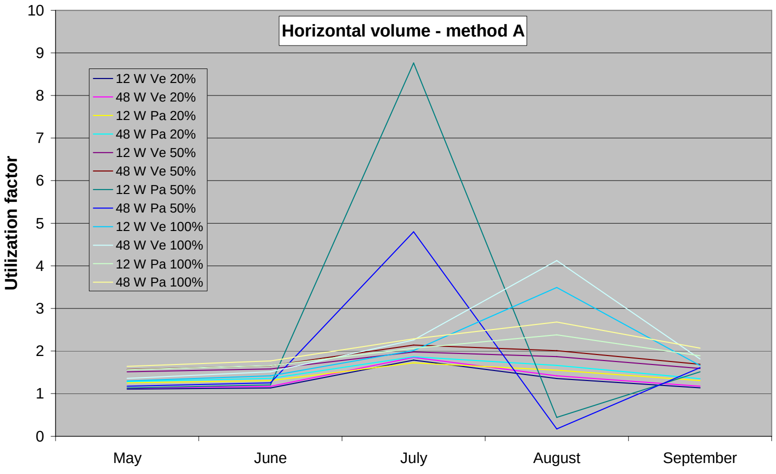

Figure 4 Utilization factors with method A for the horizontal volume in the various cases.

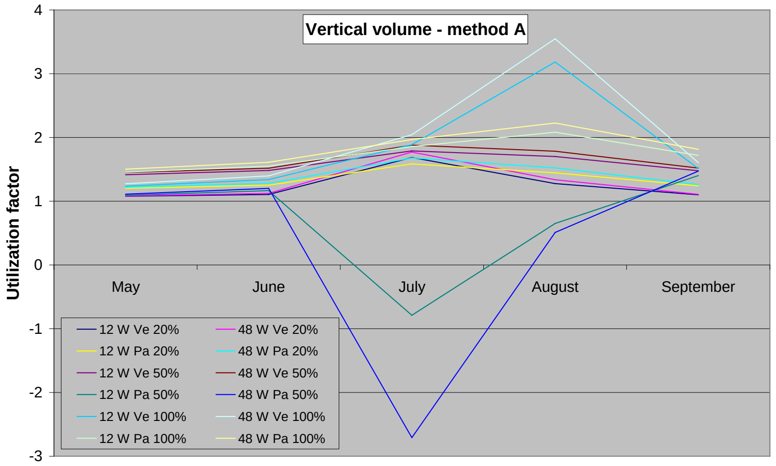

Figure 5 Utilization factors with method A for the vertical volume in the various cases.

An extreme variability of the utilization factor can be noted and this fact involves the difficult to introduce a simple correlation able to calculate a correct value for the equation (7). Besides it is often strongly greater than the unit. This is because normally only a fraction of the heat gains contributes to the cooling requirements while a considerable quota is lost to the surroundings. Indeed this conclusion is not a great novelty for people familiar with cooling load calculations by dinamic simulation. The transfer function method from ASHRAE [6] foresees reduction coefficients for the various types of heat gains. The necessity to reduce also the heat gains is the reason of the utilization factors greater than one. In the same way when sometimes the heat losses are negative (i.e. in Palermo) the procedure can need negative values of $\boldsymbol{\eta_u}$. We can conclude that especially in presence of remarkable solar contributions and smaller effect of free cooling connected with heat losses there is the necessity to introduce an utilization factor for the heat gain. Another method (here called method B) is then proposed and based on the following equation:

$$Q_c = \eta_u \cdot (Q_{si} + Q_i) - Q_l \qquad (8)$$

where η_u becomes again an utilization factor for heat gains. For sake of simplicity an utilization factor of the heat losses is here neglected. The effort is to maintain a simple correlation like in the winter procedure. A verification of the method B is presented in the figures 6,7 and 8 in the various cases. We can appreciate in this case the moderate variability of the values. In this way it is justified to consider eventual approximations or errors anyway of limited amount. The use of only one utilization factor for the heat gains seems to be able to

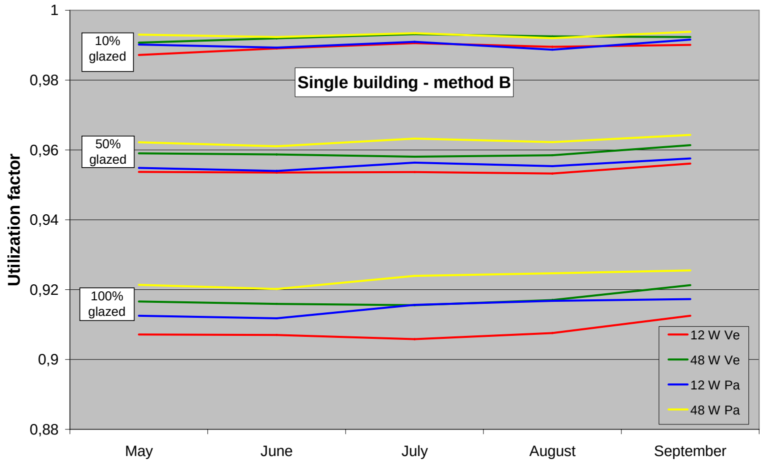

Figure 6 Utilization factors with method B for the single unit in the various cases.

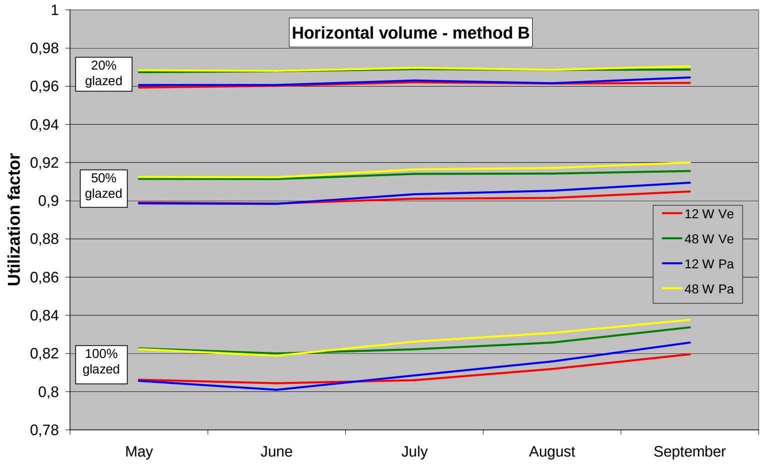

Figure 7 Utilization factors with method B for the horizontal volume in the various cases.

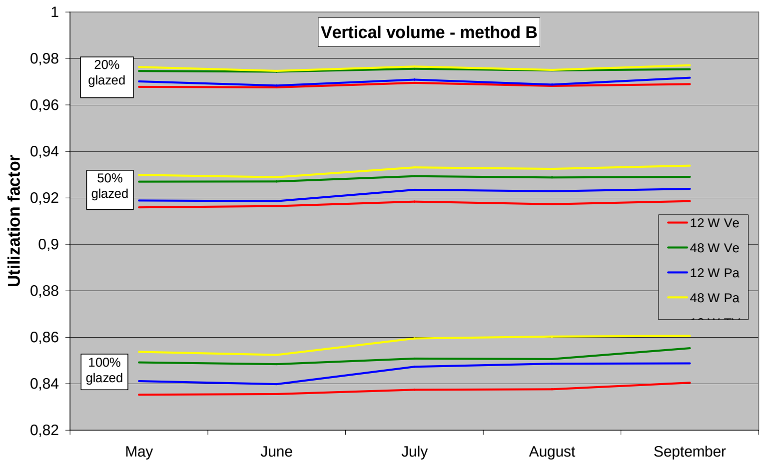

Figure 8 Utilization factors with method B for the vertical volume in the various cases.

control in a correct way the effect of both the heat fluxes and contributions. In addition the figures show a greater influence of the percentage glazed quota than the internal gains on the values of utilization factors. It means that the solar contributions deeply influence the η_u In particular we find, as foreseen, a significant reduction of the utilization factor by increasing the glazed quota i.e. the solar flux. On the contrary it is quite logical that an internal electric gain (computer, electric machines) becomes immediately a convective cooling load and then it is completely added to the cooling needs. The variability of η_u regards therefore especially the solar gain. For this reason a third method (here called method C) is here analysed where the utilization factor is referred only to the solar contribution like presented in the following formula:

$$Q_c = \eta_u \cdot Q_{si} + Q_i - Q_l \qquad (9)$$

A verification of the method C is presented in the figures 9, 10 and 11 in the various cases.

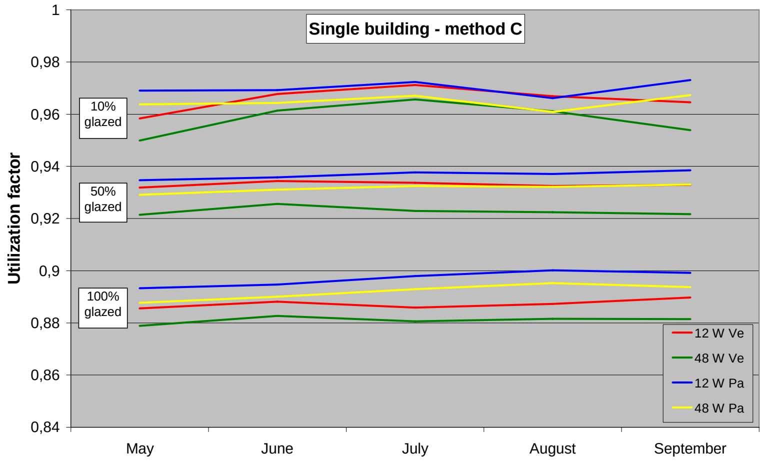

Figure 9 Utilization factors with method C for the single unit in the various cases.

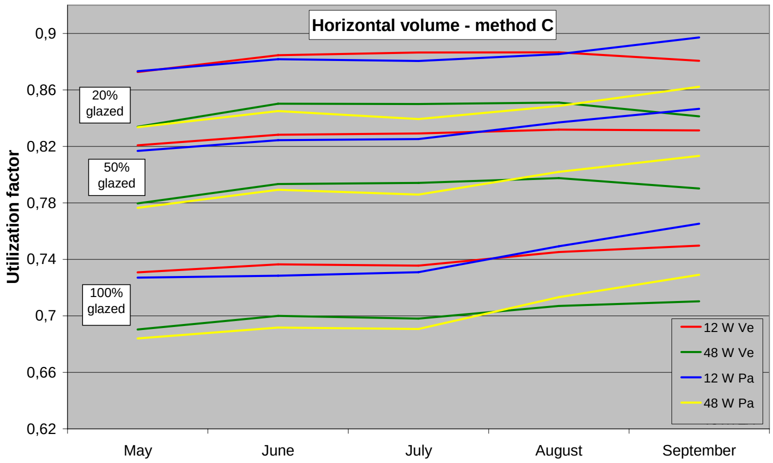

Figure 10 Utilization factors with method C for the horizontal volume in the various cases.

The application of the method C points out a better sensibility of η_u to the influence of the amount of glazed quota and therefore to the parameter γ . This fact confirms the correct interpretation of a reduction of the utilization of the heat gain only referred to the solar contribution which is absorbed first by the internal surfaces and later partially transferred to the internal air as a cooling load. This mechanism is influenced by the thermal inertia of the structures and the heat loss coefficient of the room. Besides, for the same glazed quota, the figures show that the utilization factor increases lightly in presence of less internal gains. Also with method C therefore, it is correct the research of a correlation to calculate η_u where the ratio γ still contains all the heat gains.

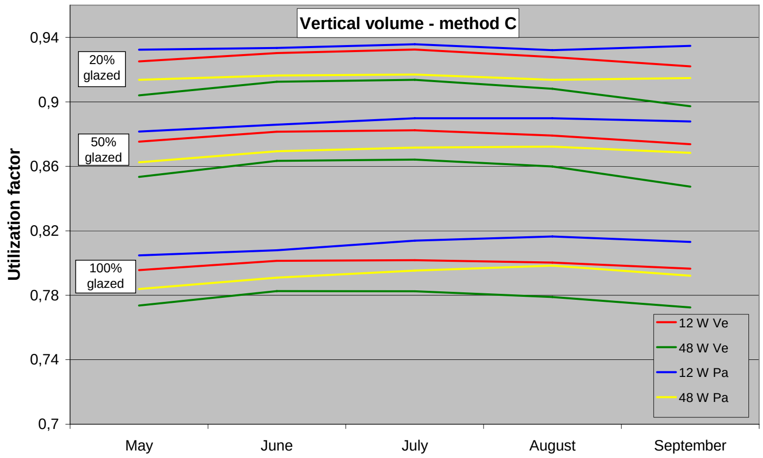

Figure 11 Utilization factors with method C for the vertical volume in the various cases.

DISCUSSION

The analysis confirms the possibility of a simplified estimation of building cooling requirement based on the utilization factor method likewise the procedure to calculate heating energy use. But in the cooling case the utilization factor must be referred to the free heat gains which are the fundamental heat fluxes which determine the air conditioning need. Instead in summer heat losses permit often a free cooling, but they are normally modest. Therefore a reduction coefficient of their influence on the cooling load is not the best way to take into account also the more significant reduction of the heat gains. The results of the dynamic simulation suggest the introduction of an utilization efficiency referred only to the solar contribution. It seems to be possible to evaluate it by an algorithm equal to that one already used for heating calculation as a function of the parameters γ and τ. But new correlations to calculate them will have to be elaborated and verified by an extended data base of simulation results with different building characteristics, use and climatic conditions.

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Planning Healthy Buildings – A Swiss Label that Insures Quality

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SUMMARY

The following article presents a method that outlines how to effectively construct buildings that emit only minimal concentrations of harmful airborne substances. The conception and implementation of a label that certifies the ambient air quality of newly constructed and renovated buildings by carrying out accredited, post-construction air quality evaluations, will touched upon as well. In addition, a certified building of Swiss Life, Zurich will also be profiled.

INTRODUCTION

Even in Switzerland today, it is common for health complaints to arise in newly constructed buildings due to excessive levels of ecological contaminants. Sometimes these new buildings must be shut down for decontamination work. This was the case of a newly renovated school building in Zurich that was found to have formaldehyde values ranging between 190 and 311 $\mu g/m^3$ in no fewer than 10 of its classrooms. In an even more recent case, stemming from 2006, a school building of wood construction was found to be emitting maximum formaldehyde concentrations of 381 $\mu g/m^3$ [2]. In each of these cases, adverse health effects resulted. Pupils and teaching staff alike complained of eye and throat irritations. The publication Innenraumklima - which was produced in collaboration with the Swiss Federal Office of Public Health and a number of Swiss building authorities – documents 18 separate cases of complaints, complete with detailed laboratory readings and materials data [1]. As a direct result of these cases, the Planungsleistung Innenraumklima (Indoor Air Quality Service IAQS) was developed [3]. The IAQS has established concrete performance levels (in accordance with performance model SIA 112) [4], that are to be adhered to throughout each planning phase in an effort to optimise indoor air quality. In addition: performance targets, decision making protocol and impending documents of the building developer as well as a performance itinerary for the planner are precisely outlined. With the help of spreadsheets which, as templates, can also be presented electronically, relevant influences on indoor air quality can be determined, controlled and thus minimised. In addition, an agreement could be made concerning the target values to be achieved by construction's end and consequences if not achieved. Requirements on ambient air quality driven by target values are also included. The label, GI Gutes Innenraumklima, is the logical development and further enhancement of such target requirements. A two year test phase was initiated to show whether and how such target values might be achieved, and thus promoted, in an actual construction situation. The present work will illustrate the findings of our investigations both into building projects that followed the protocol of the IAQS as well as projects that took no special measures to optimise indoor air quality. Furthermore, the certification of one building with the GI Gutes Innenraumklima will be presented.

METHODS

For VOC (volatile organic compounds) test measurements, the substance TENAX TA with accompanying thermodesorption was used as a carrier material/substrate. Identification and quantification were done via gas chromatography/high resolution mass spectrometry (GC/MS) in accordance with VDI 4300, page 6 [5] und ISO 16000-6 [6]. VOC encompass all TENAX TA absorbable and identified substances within the given retention periods between n-Hexane and n-Icosane. The identified substances that register above the maximum, per substance stipulated value are accounted for (2-10 $\mu g/m^3$). Any unidentified signals registering above 10 $\mu g/m^3$ are also accounted for. Through the application of gas chromatic separation, the concentration of individual compounds, each found within the parameters of its own compound range and each exhibiting its own characteristic marker (response factor) will be evaluated through external standards. Moreover, the peak area of each unidentifiable compound will be taken under observation. Through the use of a characteristic marker of the reference compound, Toluene, equivalent concentrations of the as-yet-unidentified compound can be calculated. The target values for individual substance classes cannot be superseded, even if the exigency TVOC $\leq$ 1000 $\mu g/m^3$ has been met. Substances that have neither been identified nor ordered into a specific class cannot exceed 15% of the TVOC. If the stated final concentration does exceed 15%, then the reason for this must be clarified. Substances which are found but are not yet on the list must be individually studied and evaluated. In this regard, protocol must be conducted in analogy to the derivation of the target values.

Aldehyde Measurements

DNPH cartridges were used as aldehyde carriers. Analysis was done according to VDI 3484, page 3 [7]. Identified and listed substances above the limit of determination of 5 $\mu g/m^3$ were taken into account.

Particulate Matter Readings

Measurements were taken using a particle measuring device that operates on the basis of scattered light measurements, a laser and photodiodes or a system of measurement capable of achieving the same values. Measurements were conducted long enough for a stable reading of each measurement point to be achieved - at least 10 minutes per reading. A stable reading was defined as the average value of the last five minutes of the measurement. The measurement was conducted under isokinetic conditions.

Germs Measurements and Identification

Measurements were done using an impacter device with a mass flow meter. In the air delivery stream there is an impaction volume of at least 100 litres per minute (in exceptional cases, where the air delivery stream is weak - <100 litres per minute - the impact volume may be reduced to 50 litres). TSA, DG18, MEA and DRBC were used as culture mediums. The measurement was conducted under isokinetic conditions. Identification was done to at least the genus level. The specification of results was done in cfu/m^3 of ambient air. Moreover, individual petri cultures are to be in cfu/m^3 of ambient air and the average germ count of the cultures DG 18, MEA and DRBC in cfu/m^3 must be presented.
The label GI Gutes Innenraumklima also defines the standard building inspection procedure for conducting air quality evaluations. In office and large residential buildings, at least 2 of a construction project's building units or at least 10 % of all building units up to a total of 100

units must be inspected. For office, industry, administration, school and health services buildings, the surface area of a building unit ranges in size from 10 m^2 to a maximum of 200 m^2 of net space. For buildings consisting of more than 100 units, the number of units to be measured will be established according to a gradation scheme determined by the certification authority. In office and residential buildings, all interior surface materials must be considered. Evaluation readings collected from the inspected building space must comply with the established target values for TVOCs as well as every other individual substance and class of substance. If individual readings are too high, the building unit in question may be reassessed within 30-100 days after the final day of construction. If air handling systems are present, the stream of supply-air serves as the point of measurement: the number of supply-air readings is determined by the number of listed hazardous chemical and organic substances. Of these supply-air readings, 20 % must check for germs and particulate matter. The stream of supply-air from every building segment that makes up a building unit must be tested at least once during the evaluation.

Target Values

Target values had to be determined for harmful chemical substances as well as airborne germs and particulate matter found in the supply-air of existing air handling systems. For reasons of practicability and general acceptance in the field, the general scope of investigation has been limited to these three category groupings. Reliable target values for harmful chemical substances cannot be derived strictly from statistical target values, as was done by Schleibinger [8]. Weeks and months after a building's completion, measured values are much higher than statistical values reported by Schleibinger, making these as resulting target values wholly impracticable. It was important then to select and focus on target values that would help minimise the risk of complaints from both a respiratory hygiene as well as a toxicological viewpoint; and, those values needed to be both relevant and attainable for building developers. In the work at hand, we have gravitated toward the pragmatic method of using already existing and established values. Above all, existing RW-I-values (reference value) [9] or other established indoor air guidelines like WHO guidelines will be drawn from. In the event that no data was available, 1/100 of the Maximale Arbeitsplatz Konzentration (MAK) values, respective to 1/100 threshold limit values (TLV) were selected. In the case of a few compounds, low values of LCI (lowest concentration of interest) were also chosen as orientation markers in order to establish target values. Furthermore, where available, odour threshold limits have also been taken into account. This was especially the case with regard to the higher aldehydes (C_7-C_{10}), for which no RW-I-values exist. Because odour threshold values tend to differ widely depending on the literature consulted, a number of different sources have been drawn from and the respective plausibility of each source cross-examined. It is well understood by the authors that odour threshold limits are to be used with the greatest caution. Furthermore, existent testing criteria, guidelines and odour threshold limits have been a useful source - where available - for the establishment of target values. Because odour threshold values tend to differ widely depending on the literature consulted, a number of different sources have been drawn from. Wolkoff suggests, for instance, that with new olfactory methods, odour threshold values for many VOCs could well be placed lower than they presently are [10]. This would also have an effect on the relation of target values to the compounds which are ultimately based upon odour threshold limits. And certainly not least of all, there exists the problem that substance mixtures will indeed exhibit different olfactory characteristics from those same substances when analysed in their pure, individual forms. Yet, in practical experience, in spite of all of these difficulties, one must rely upon orientation values that pertain to ambient air which enable one to evaluate the myriad complexities of building construction and at the same time to make meaningful statements about ambient air quality upon construction's end. In an effort to achieve this goal, and with regard to [11] for the various substance classes, maximum values were defined. Highly reliable data has been tabulated and presented (see tables 4 and 5). From the individual representatives of these substance classes, the target value was deduced as being half of this value when no lower RW-I-values, respective of other values, were available (see above). And thus, the target value for Toluene is 300 $\mu g/m^3$, and that of Naphthalene: 5 $\mu g/m^3$ (RW-I-value 2 $\mu g/m^3$). Bearing in mind that complaints of noxious air quality and building-related health problems occur with the greatest frequency within the first few months of occupants settling into a newly constructed building and that concentrations of malodorous chemical compounds are most intense during this initial period, it should be made clear that it is necessary to achieve threshold limits within the 30 to 100 day period following the last day of a building's construction (including all final touch ups). If tests are conducted beyond this 100 day period, certification can no longer be granted. Under normal conditions - barring some kind of serious and unexpected ecological mishap - the measured values will attenuate further in the ensuing months. In buildings with low energy consumption (i.e. Minergie-Standard) [12], it is now common to install mechanical handling systems to insure adequate levels of air exchange. It is not altogether unusual for these systems to prove insufficient at the outset putting a negative strain on the supply-air. In addition to the standard inspection for chemically harmful substances, it becomes necessary in such cases to test for particulate matter as well as the level of airborne germs in the supply-air. Established target values for biological air pollutants delimit the concentration densities for bacterial, as well as fungal spores present in the stream of supply-air. These values are based, in part, on the results of the German ProKlima-Projekt [13] as well as on our own appraisals of 25 properly functioning air handling units (ACUs).

Table 1. TV (target values) for particulate matter and airborne fungal spores for the label

Parameter	TV	Unit
Bacteria	≤190	cfu/m^3
Thermoactinomyces	0	cfu/m^3
Molds	≤120	cfu/m^3
Particle class		
Particulate matter >2 µm	≤10	particles/litre
Particulate matter >0.75 µm	≤150	particles/litre

Target values for aldehydes and VOCs are listed in Tables 4 and 5.

Concerning particle concentrations, target values will be specified for two relevant particle sizes. Achieving the prescribed target readings can only be accomplished if the air is being handled by a fully operational ACU that has been equipped and maintained in accordance with SIA Norm 382/1 [14] filters. Our own readings, taken from 20 ACU systems, show that these target values are practicable. If the filter performance is too low or if its performance is compromised by an influx of additional particles to the supply-air stream, the stipulated values will be exceeded.

Certification

The label GI Gutes Innenraumklima certificates ambient air quality after standard building inspection procedure evaluations. It defines the requirements of the ambient air quality including germs and particulate matter concentrations of the supply-air stream. The certification authority defines the requirements of the measuring points and also the scheduling and realisation of the inspection evaluations. The label will be awarded by the swiss certification authority for construction materials S-Cert AG [15].

RESULTS

Aldehyde readings show (Table 2) that the detected values in buildings in which there was no IAQS implemented are significantly higher - in the case of almost every listed compound - than in those buildings where there was IAQS implemented during construction. Tests were conducted 30-100 days after the last construction date in buildings where consultation was used and 1-30 weeks in buildings without IAQS. In the case of VOC measured readings in buildings without IAQS were found to be markedly higher than in those structures that had undergone IAQS (table 3). The median of the 37 buildings (n=48) without building consultation was found to be 1302 µg/m^3 for TVOC (95 Percentile 4512 µg/m^3). These results are considerably higher than those found in buildings with IAQS (Median 698 µg/m^3, 95 Percentile 2338 µg/m^3).

Table 2. Aldehyde readings without/with building consultation in new and renovated buildings, total 45 buildings (n=91), 37 buildings (n=57) without building consultation, 8 buildings (n=34) with building consultation. Data readings in µg/m^3. (SD): Standard deviation, (%): Percentage of frequency, n: number of measurements.

Compound	50.P without/with		95.P without/with		Max without/with		SD without/with		% without/with	
Formaldehyde	45	25	138	40	179	57	40	10.0	94.7	100
Acetaldehyde	39	25	355	76	475	142	108	28	94.7	94.1
Propanal	12	6	66	16	126	22	27	6	66.7	58.8
2-Propenal	0	0	0	0	0	0	0	0	0	0
Benzaldehyde	7	0	28	23	40	28	11	8	56.1	41.2
n-Butanal	0	0	44	16	52	28	15	7	42.1	47.1
2-Butenal	0	0	0	0	537	0	94	0	3.5	0
3-Methyl-butanal	0	0	19	8	36	9	8	3	22.8	11.8
n-Pentanal	10	0	177	11	375	12	74	4	61.4	23.5
1,5-Pentanedial	0	0	7	15	60	19	8	6	8.8	44.1
Hexaldehyde	39	24	419	53	644	63	142	16	89.5	91.2

Table 3. VOC evaluation readings in new and renovated buildings, total buildings 45 (n=87), 37 buildings (n=48) without building consultation, 8 buildings (n=39) with building consultation. Data readings in µg/m^3. (SD): Standard deviation; (%): Percentage of frequency.

Compound class	50.P without/with		95.P without/with		Max without/with		SD without/with		% without/with	
Aldehydes/ Ketones	136	66	449	257	575	400	142	92	95.8	85.6
Aliphatic HC	55	10	1318	321	2372	982	491	190	83.3	56.4
Amines/ Amides	0	0	56	18	92	49	19	11	16.7	12.8
Aromatic HC	279	145	1023	796	1820	1299	378	299	97.9	92.3
Carbon acids, Alc., Ether	76	29	260	130	360	168	98	42	75.0	71.8
Chlorinated HC	0	0	0	0	6	5	1	1	2.1	2.6
Cycloalkanes	0	0	14	34	261	172	38	34	8.3	7.7
Esters	108	86	621	515	2330	1117	366	222	91.7	69.2
Glycols/ derivates	52	11	401	190	490	335	122	75	75.0	56.4
Siloxanes	74	20	385	265	488	459	122	92	91.7	66.7
Terpenes	145	7	541	85	1422	228	247	43	89.6	51.3
TVOC	1302	698	4512	2338	4813	2450	1335	660	100	100

(HC): hydrocarbons, (TVOC): total volatile organic compounds, (Alc.): Alcohols

Case Study

The headquarters Swiss Life office building in down town Zurich, consisting of 9’589m^2 of office space on each floor, was completely renovated. The building was first stripped down to a bare, skeletal structure. After that, the floor work, non-supporting partition walls, suspended ceilings, all interior surfaces and numerous installations were newly reconstructed. The work was optimized with the help of the IAQS and the use of on-site, ecological consultants was made.
In the project development phase, materials and construction methods were scrutinized to evaluate the potential adverse impact of emission levels. This was made possible through the early selection of building material types. Adequate airing-out periods were also made an integral part of the construction time-frame.
During the construction phase, the implementation and inspection of ecological planning specifications to be adhered to by the contractor played a crucial role. Wherever possible, materials were selected or prescribed on the basis of the relevancy of their emission labelling. Contractor’s bids were carefully considered and building products and on-site construction methods were regularly controlled. Eight weeks after the final day of construction, readings were taken. During that eight-week interval, offices not equipped with air handling equipment were manually aired out once daily in strict accordance with a pre-arranged building ventilation plan. The air ventilation equipment on the executive floor was kept operational until one day before readings were taken. In the following table, the readings for each individual room are presented. For a complete and successful evaluation, each room had to fulfil the target values. The maximum shows that this was indeed the case for every room.

Table 4. Final measurements for aldehyde, 8 weeks after the last day of construction. 13 rooms were analysed, measurements in µg/m^3. SD: standard deviation, TV = Target Value

Compound	50.P	95.P	Max	SD	TV
Formaldehyde	18	27	28	7.4	60
Acetaldehyde	15	38	41	13.9	200
Propanal	0	6	6	2.8	20
2-Propenal	n.n.	n.n.	n.n.	-	5
Benzaldehyde	0	8	9	3.2	50
Butanal	0	9	12	3.7	10
2-Butenal	n.n.	n.n.	n.n.	-	5
3-Methyl-butenal	n.n.	n.n.	n.n.	-	20
n-Pentanal	n.n.	n.n.	n.n.	-	20
1,5-Pentanedial	6	16	19	6.6	20
Hexaldehyde	22	41	52	15.1	60

Table 5. Final measurements of VOC 8 weeks after last day of construction. 13 rooms were analysed. Results in µg/m^3. TV = Target Value, SD: standard deviation

Compound class	50.P	95.P	Max	SD	TV
Aliphatic HC	32	55	57	22.6	500
Aromatic HC	79	232	234	83.8	500
Carbon acids, Alcohols, Ether	29	85	123	31.9	300
Chlorinated HC	0	2	5	1.4	10
Esters	147	203	222	69	300
Glycoles/ Derivates	26	46	46	13.3	300
Siloxanes	20	63	80	23.5	500
Terpenes	0	33	62	17.2	300
TVOC	485	800	818	240.8	1000

DISCUSSION

The VOC and aldehyde target values listed in tables 4 and 5 are derived from real and pragmatic test measurements that have been carried out on actual buildings. They should not, however, be taken as universal examples of ambient air quality, as they refer to newly constructed buildings which in general give off higher levels of harmful contaminants. With today's building techniques, it is not only possible to achieve these target values but to reduce them to even safer levels through lending careful attention to relevant IAQS criteria during the planning such as the use of low-emission building materials, air handling systems and ecologically sound construction methods. Meeting or improving upon derived target values incites us to consider more deeply the care and attention we should be granting the use of low-emission materials both in the planning and execution phases of a building project. Success in meeting the required certification standards helps minimise the risk of complaints or lawsuits from building occupants. This work demonstrates that through clear and careful planning and implementation, achieving target values is feasible. The selection of to-be-tested VOCs was made after close scrutiny of the compounds presented in the VDI of 4300 listed compounds [5]. This list was supplemented further by a number of additional compounds: (n-1-Methyl-2-pyrrolidinone (TV: 40 $\mu g/m^3$), Decamethylcyclopentasiloxane (TV: 250 $\mu g/m^3$), 2-Butanone oxime (TV: 100 $\mu g/m^3$), n-Decylaldehyde (TV: 10$\mu g/m^3$) 2-Decenal (TV: 2 $\mu g/m^3$). The list of aldehydes selected for testing, is identical with those compounds enumerated in table 4. An extended list of target values and measured VOCs is published in [16]. In an academic effort to assess the hygienic air quality of newly constructed buildings, a more theoretical path is sometimes taken by which hyper-efficient, ecological show rooms are set up using emissions-certified materials [17]. Our practically-minded intentions have taken us in a different direction; because in the end, the completed building with all of its shortcomings and imperfections is the more relevant object and it is the one requiring evaluation and certification. The advantage of swaying in favour of the practical side is that the inevitable imponderabilities of the building industry: such as frequent changes of contractor, additional subcontractors - who may avoid the use of approved materials - do not end up taking you completely by surprise. Experience shows that the optimisation of the indoor air quality the evaluation and certification thereof is well within the realm of feasibility without the need for legally imposed limit values of hazardous compounds. Moreover, new product formulations alter the spectrum of hazardous substances trapped in our non-ambient air and fresh discoveries concerning photochemical reactions in indoor space make it periodically necessary to reshuffle the selection of compounds that are to be verified.

The weight of experience has shown that by making and communicating the timely decision to have the indoor air quality of your building project evaluated and to certify it with a label has positive effects on the entire planning process. By relying on indoor air quality planning with regard to both newly-constructed buildings as well as renovation projects, it is possible to construct buildings that give off very low concentrations of hazardous substances even directly after the last day of construction. The budgetary implications of ecological consultation with respect to the illustrated case studies - including the end-of-construction building evaluations - runs modestly in the neighbourhood of 0.1 to 0.2 percent of the total building costs. In the case of the Swiss Life building, this amounted to <60 Cents/m^3 of building volume according to SIA 416 [18].

We have noted further that every effort made and communicated on the building developer's behalf to improve the living quality of future occupants produces in the latter a courteous attitude which is good at the best of times and invaluable in those potentially difficult times when relations between renter and rentee become strained. As a direct result of this constructive phenomenon, the Swiss Reinsurance Group has made it their standard practice -

with positive effect- to include the label of certification as part of each new occupant's building reference documentation package.

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Optimization of refurbishment choices of a building by the use of the experimental design concepts

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SUMMARY

In order to reduce the energy consumptions, it is urgent to be focused on existing buildings. The answer is how to determine priorities of interventions on a building to be renovated. The traditional method consists in carrying out a great number of tests and quantifying profits obtained with each solution or combinations of solutions. Our object is to optimize the number of tests to be realized by use of the method of the experimental design concepts which is applied to draw up decisional flow charts of interventions on a building built before 1975. Software Nemrod [1] is used for the experimental design concepts. This stage carried out, the plans are applied by the means of the software Trnsys [2] giving as output the requirements.

INTRODUCTION

Existing buildings consumption is estimated at 60% more than the new buildings.
To determine rehabilitation works on the building, it's possible to vary a lot of parameters. Nevertheless, a mathematical method making it possible to reduce the number of simulations for what is strictly necessary to make a decision, thus allowing a saving of considerable time, was selected in order to draw up a decisional flow chart in term of interventions on the building. It is the methodology of the experimental designs. These experiments determine the influence of such or such factor of the building on the performances of the latest. Two experimental design concepts must be carried out separately according to the weather for two localisations. Identified elements are the following: windows, frontages, roof, ventilation and floor.
On a reference building built before 1975, the method of the experimental design concepts is applied to draw up decisional flow charts of interventions on the existing buildings according to a share of the typology of the building but also of the climatic conditions.
Thermal air flow modelling of building is integrating in TRNSYS via TRNFLOW [3] which is the integration of the multi-zone air flow model COMIS (Conjunction of Multi-zone Infiltrations Specialists) into the thermal building module of TRNSYS (Type 56).
The required objective is the reduction of a factor four of the requirements in winter.

METHODS

The reference building is described first, to know requirements in winter in Type 56. On this building the method of the experimental design concepts is applied by supposing that the lower limits correspond to the building not renovated. Responses are given for requirements. On those responses, the weight of elements is studied for two weathers.

The period of heating considered is spread out from October 1 to April 31.

Reference building

Reference building built in 1966 (Figure 1) is given by [4]. It's made of three floors: Basement, Ground floor (Dining room, services rooms, Rooms 1 at 3) and a stage (rooms 4 and 5, attics 1 and 2 and cupboards) on 120m² of liveable area with single glasses. The roof slope's is 45°. Building description is given in Tables 1 and 2.

Figure 1. General view of the 1966's residential building

Table 1. Layers description. Layers are defined outside towards the interior of the zone

	Materials	**Thickness (cm)**	**U (W/ (m².K))**
FLOOR_BASEMENT	Concrete_stone	20	3,297
WALL_OUT_BAS	Breezeblock _full	20	2,183
DOOR	Wood_heavy	5	2,585
FLOOR_LOW	Hourd20_concrete	20	3,069
	Tiles	1	
WALL_OUT_FAÇADE	Brick_Hollow_22	22,5	1,858
WALL_IN	Breezeblock _10	12	3,6
PARTITION_PLASTER	Breezeblock_50	5	3,242
DOOR_ISOPLANE	Wood_heavy	3,4	3,15
ROOF	Slate	1	0,717
	Mineral wool	5	

Table 2. Dimensions and orientations of windows

Rooms	Orientations	Area [m²]	H/ ground [m]
Basement	North	0,5	1,3
	South	0,5	1,3
Living room + Bedroom 1	South	4,515	2
	South	1,75	3,05
	West	1,75	3,05
Bedroom 2	South	1,75	3,05
Bedroom 3	North	1,75	3,05
Services rooms	North	3,5	3,05
Bedroom 4	West	1,75	5,55
Bedroom 5	East	1,75	5,55

Assumptions of simulation

The building is divided in several thermal zones in Type 56 as saw above (Figures 1 and 2):

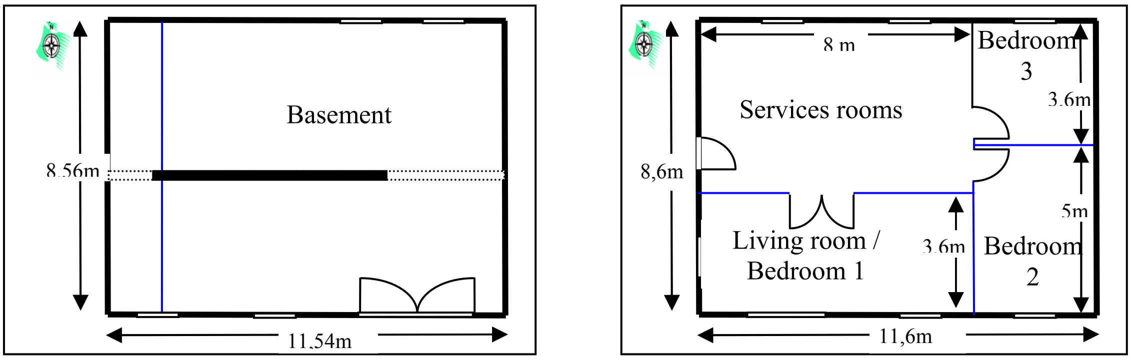

Figure 2: Basement and ground floor plans

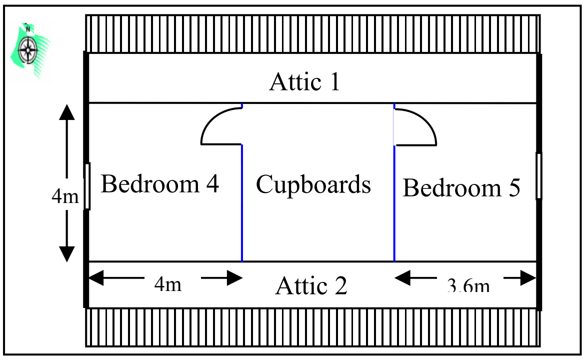

Figure 3: Stage plan

Choice of convective heat transfer coefficients: Inside walls convective heat transfer coefficients are calculated by TRNSYS software in Type 56. For other vertical walls, default values are retained: 3 W/m².K for front wall and 17,8 W/m².K for back wall.
For slope surfaces like roof, result is carried out via an input related to an equation computed convective heat transfer coefficients described above. Brau [5] allows calculating them for outdoor surfaces and shows what h_{out} can express (1):

$$h_{out} = 3{,}89 \cdot \left(\frac{v_m}{l}\right)^{\frac{1}{2}} \quad [\mathrm{W/(m^2.°C)}] \qquad (1)$$

where v_m is the wind speed [m/s] and l the length considered [m].
For front surfaces, convective heat transfer coefficients for slope walls are the same as the one proposed by TRNSYS in roof Type (usable only with the mono-zone building) result from ASHRAE Handbook of Fundamentals : $h_{i,45}$ = 9 W/m².K.

Ground Temperature: In order to approach reality as well as possible, the temperature of the ground is modelled by a specific type (Type 77) which represents the sinusoidal evolution of the temperature over the year. A parameter of this model is the average temperature of surface which is typically the annual average of the temperature of the air to the place considered: it's 10,15°C for Lille et 14,8°C for Marseille.

Internal loads: Internal loads results of the Concerto study were retained [6] for the summer and for the winter they are based on French thermal requirement RT2000 (4 W/m²).

Air flow model: Wind pressure coefficients, cp, are defined for a building reference high of 7,5m. Wind prevailing is supposed North direction, rather frequent situation in France, for a wind profile: "Open terrain with isolated obstacles" [3].

Orme, Liddament M., Wilson A. [7] provide cp for ratios width/height of building equal to 1 or 2. This report/ratio is worth approximately 1,35 for the building of reference. We thus carry out a linear interpolation in way as well as possible of bringing us closer reality.
TRNFLOW takes the difference between the data weather and the data of site into account and uses a law of power to express the wind speed on the building level at the reference height. In TRNFLOW, it is necessary to define a factor of opening equal to 0 for a door or a closed window and to 1 for a door or a completely open window. It thus proves useful to differentiate the strategies from natural ventilation by the opening of the windows in winter and summer. The interior doors are supposed to be half open with during all the period of simulation. A strategy of opening is defined for each period for the windows giving on the outside: 10 min per day during the cold season.

Existing building diagnostic

Thermal air flow modelling of the not renovated building gives results of requirements for various weathers (Lille and Marseilles). Heated area is 120m². Attics, basement and cupboard are not heated.
Heat requirements found simulating this existing building are 320kWh/m².year for Lille and 183kWh/m² year for Marseille.

According to climatic zones, priorities of interventions will not be the same ones. In winter, one will seek to reduce the requirements in heating in the zones located in North of France. In France the announced target value for obtaining the factor four is 80kWh/m² for water heat and heating. The order of magnitude of the calculated requirements corresponds to reality. This factor results in approximately 80kWh/m² at Lille and 50kWh/m² at Marseille.
By which skew is it possible to reach these values?

EXPERIMENTAL DESIGNS CONCEPTS ON THE BUILDING BUILT IN 1966

Experimental designs concepts help to draw up interventions priorities on the building by the analysis of the weights of the various building elements.
It's a mathematical method allowing reducing simulations number for what is strictly necessary to make a decision, saving of considerable time. It is selected in order to draw up a decisional flow chart in term of interventions on the building. For further information on methodology of the experimental designs, the reader will refer to lesson of Fürbringer [8].
An experimental design is carried out for a given reference building, in a fixed configuration. It is clear that for similar buildings (morphology, situation) the results found could be exploited. Nevertheless, the main object of the step is to propose a methodological approach replicable on any building.
Elements building influence is now studied on the following answers: heating requirements in winter.
It is necessary to carry out different experimental designs according to the period considered (hot period or cold period). Thus the analyzed answers are distinguished. In the same way, the studied factors intervals will be indicated according to the desired answer. However, intervals corresponding to the not renovated building will be identical. Cold period is considered in this study.
Seven factors are analyzed corresponding to 5 building elements. Thus, for the window the insulation and the capacity to generate heat are studied.
High factors selected levels summary correspond at low energy label of France [9].

In an experimental design, the *sifting* makes it possible to know the weights of the levels of the factors on the answers. This study type makes it possible to find the few factors active, quantitative or qualitative, among a great number of factors in a limited number of experiments. The factor weight represents the importance of the answer variation due to a change of factor level.
The experiment matrix is indicated and the twelve following experiments (Figure 4) are carried out, Xi are the various factors:

N° Exp	X1	X2	X3	X4	X5	X6	X7
1	1	1	1	-1	1	-1	-1
2	-1	1	1	1	-1	1	-1
3	-1	-1	1	1	1	-1	1
4	1	-1	-1	1	1	1	-1
5	-1	1	-1	-1	1	1	1
6	1	-1	1	-1	-1	1	1
7	1	1	-1	1	-1	-1	1
8	-1	1	-1	-1	-1	-1	-1

Figure 4. Experiment matrix

Next, answers vectors are indicated in the experiment matrix. At the end of this stage Software Nemrod provides several types of exits of which the graphic study which makes it possible to analyze various factors effects on 1966's building.

Experimental designs in cold period

In cold period, the transmission coefficient and solar gains will seek to be increased. For ventilation, the lower limit corresponds to the not renovated house with all its defects of sealing. In the old buildings, one supposes indeed that ventilation is carried out by the defects of sealing. The upper limit represents here a system with recovery of heat. This parameter corresponds at infiltrations. For low level, infiltrations maximum are indicated in Table 3 for all zones and null for high level (completely blocked up building).

Table 3. Definition of ventilation in experimental design concepts

Period	Old building: Level 0		Renovated building: Level 1	
	Lille	Marseille	Lille	Marseille
Cold period	188 kg/h	142 kg/h	0 kg/h	0 kg/h

For walls solar absorptance, accent was related to the choice of the coating in particular on the importance of surfaces colour and texture (Table 4).

Table 4. Solar absorptance of walls in experimental design concepts

	Not renovated building: Interval down		**Renovated building: Interval up**	
	Front	**Back**	**Front**	**Back**
FLOOR_BASEMENT	0,4	0,8	**0,9**	0,8
WALL_OUT_BAS	0,6	0,6	**0,2**	**0,2**
DOOR	0,7	0,7	0,7	0,7
FLOOR_LOW	0,5	0,7	**0,9**	**0,2**
WALL_OUT_FAÇADE	0,45	0,45	**0,2**	**0,2**
WALL_IN	0,45	0,45	**0,2**	**0,2**
PARTITION_PLASTER	0,45	0,45	**0,2**	**0,2**
DOOR_ISOPLANE	0,7	0,7	0,7	0,7
ROOF	0,7	0,89	**0,2**	0,89

Table 5. Factors selected levels summary in cold period

	Physical parameters	Symbol	Unit	Range of variation	
				Lower limit	Upper limit
Window	Thermal transmission coefficient	U	W/m².K	**5,74** (Single glass)	**1,5** (Double Low E Argon : 1,42)
	Solar factor	g		**0,1** (Shading device)	**0,87** (Single glass)
Walls	Thermal transmission coefficient	U	W/m².K	Wall_out_bas: 2,183 Wall_out_façade: 1,858 } **2** (Not renovated walls)	Wall_out_bas: 0,243 Wall_out_façade:0,238 } **0,24** (Without thermal bridge, 15 cm insulation λ=0,04W/m.K)
Roof	Thermal transmission coefficient	U	W/m².K	**0,713**	**0,133** (25 cm insulation λ=0,04W/m.K without thermal bridge)
Ventilation				**Level 0**	**Level 1**
Floor	Thermal transmission coefficient	U	W/m².K	Floor basement not isolated: 3,297	0,251 15cm mineral wool
Absorptance	Abs. factor	α		Level 0	Level 1

RESULTS

Various factors weights are given for cold period for Lille and Marseille weathers.
In the figures analyzing the effects of the factors on the answers, the latter are indicated as follows:

b1: Thermal transmission of windows
b2: Solar factor of windows
b3: Thermal transmission of walls
b4: Thermal transmission of roof
b5: Thermal transmission of basement floor
b6: Ventilation
b7: Absorptance of walls.

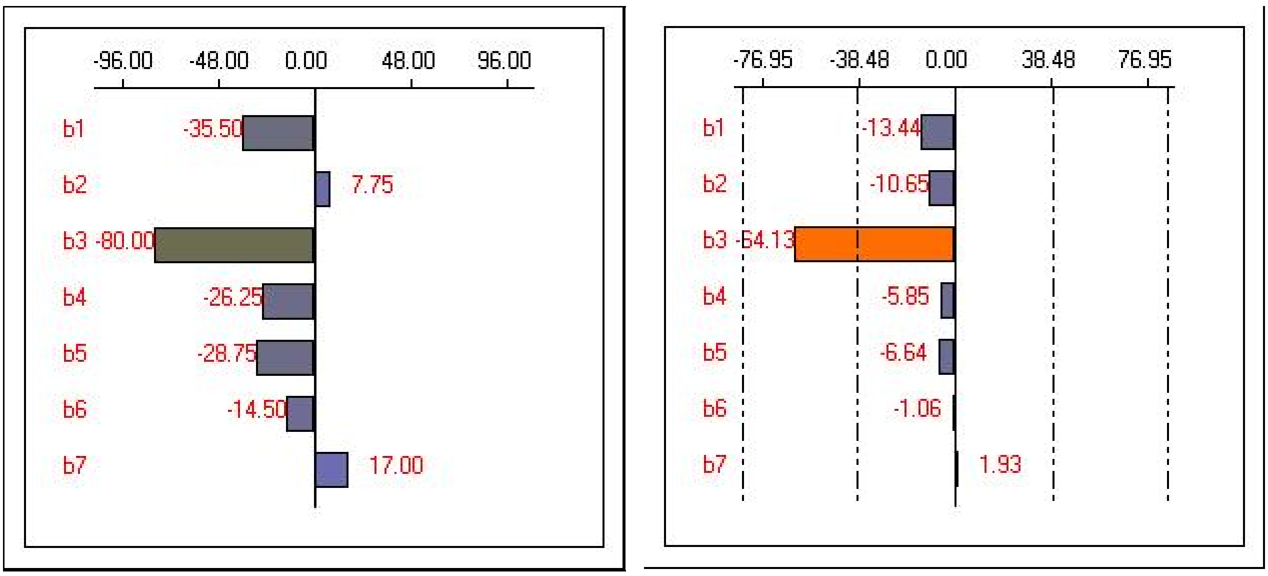

Figure 6. Effects Graphic study of the response "Heating requirements" in Lille (left) and Marseille (right)

DISCUSSIONS

Experimental design result analysis give decisional flow charts in terms of interventions on the building below:

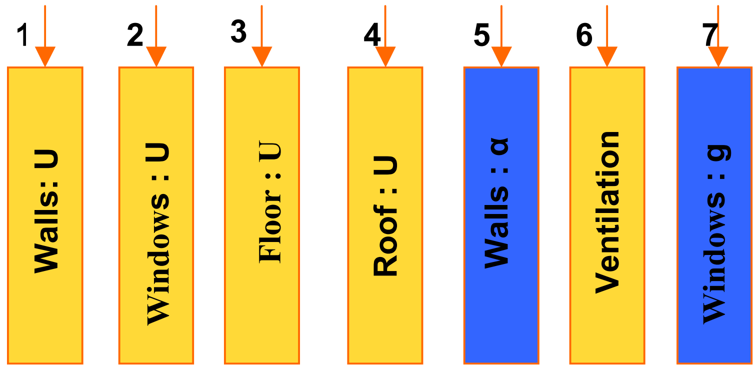

Figure 7. Decisional flow chart in terms of interventions for Lille

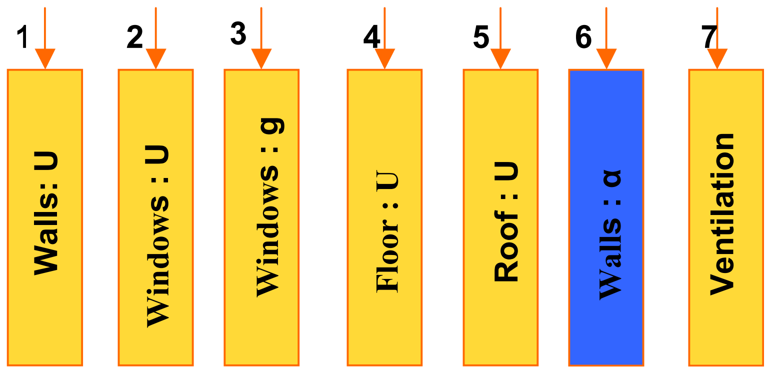

Figure 8. Decisional flow chart in terms of interventions for Marseille

Decisional flow charts, in terms of interventions in the building, are different according to the geographical localization.
For Lille and Marseille, the main priority is the improvement of wall insulation followed by thermal transmission of windows (U).
For Marseille, the other factors have a negligible influence.
In both cases, the wall absorption operates negatively on requirements. This result is not surprising since when this coefficient decreases (vertical surfaces are brighter when walls are painted), solar radiation is absorbed less. The high limit corresponds rather to an improvement of comfort during summer.
Ventilation improvement has more impact in Lille (cold city) than Marseille (Mediterranean city).

In the two cities, the floor insulation decreases the heat requirements with a higher impact in Lille where the ground temperature is lower (approximately 10°C against 15°C in Marseille).
A similar influence appears for the roof. Note that this element is often cited as a main intervention priority on the building. In our study, the reference building has 5 cm of roof insulation, which explains its position in the decisional flow chart.
The solar factor has a negligible influence in Lille, while in Marseilles it is the third parameter after the wall and window insulation.
The singularity of the method lies in the fact that, by defining an existing building through a mathematical methodology, it is possible to determine the intervention priorities which are not always those recommended in literature [10]. Each old building is a unique case due to the

building materials used for its construction and its geographical location and thus must be regarded as such.
At this stage, it would be interesting to reflect on what the influence of these factors in terms of summer comfort improvement would be?

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11 June 2007 at 12:30 - 14:00

B02
Building components and double skin facades

Thermal Effect of 'Changing Clothes Function' of Building with Changeable Thermal Property of Wall Surfaces

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SUMMARY

To realize a desirable building environment by saving energy and low global emission, the author has been investigating passive systems in buildings where the environment is controlled biomimetically and autonomously by simulating the physiological functions of human and other organisms. An environmental harmonized "Biomimetic Building", which simulates the environment physiology mechanism of a human body and human wisdom for environment symbiosis, has been developed. This paper describes the regional variation of thermal effect estimated theoretically of the "Changing Clothes Building (CCB)" whose roof and wall surfaces can, depending on the season, autonomously change their thermal properties, such as absoptivity and emissivity. The simulation results showed the remarkable thermal effectiveness and the desirable changing mode of the thermal properties of the "CCB" for various regions in Japan. The saving energy effects due to the "changing clothes function" in the typical detached house in Japan were 7 to 32% in house sensible heat load and 6 to 24% in house total (sensible and latent) heat load as compared to the "Non Changing Clothes Building (NCCB)". The effect in hot weather regions became remarkably bigger than the one in cold weather regions.

INTRODUCTION

To realize a desirable building environment by saving energy and low global emission, it is important to learn more about physiological function of organisms and human wisdom for environmental symbiosis and control. Our idea of "Biomimetic (creature imitative) Building", where the environment physiology mechanism of the human body and other creatures are compared and applied to the environment symbiosis and control, has been developed. This paper describes the regional variation estimated theoretically, in summer and winter, of the thermal effectiveness of the "Changing Clothes Building" whose roof and wall surfaces autonomously change their thermal properties, such as absorptivity and emissivity.

THE "CHANGING CLOTHES BUILDING"

The "Changing Clothes Building (CCB)" changes autonomously; the thermal properties of the building surface become low solar absorptivity with high emissivity (selective emission surface) in summer and become opposite (selective absorption surface) in winter. According to the building surface temperature, these functions are fulfilled by the following ways. One way is transforming the "thermo sense transformation material" or "bimetal" with different thermal properties on both sides of the fin plates, one side of which is a selective emission surface and the other selective absorption, set to the building. The other way is using the torque of the shape-memory alloy to rotate the fin plates with different thermal properties on both sides or the cylinder with different thermal properties on each half of the circumference set to the building. The walls and roofs are now under development and the thermal effects are examined experimentally.

SIMULATION OF THERMAL EFFECT

The room temperature and heat loads of the "CCB" are theoretically estimated and its thermal effect is studied by comparing the results between the "Changing Clothes Mode (CCM)" and "Non Changing Clothes (usual) Mode (NCCM)" of the "CCB".

Analysis method and algorithm

The method adopted is of calculating multi room temperature, heat load and ventilation rate with the use of simultaneous non linear equations of room heat balance, room air rate balance, wall outer and inner surface heat balance, with room temperature, room pressure, wall outer and inner surface temperatures for each room set as unknown quantities, which is shown in literature [1,2]. As for the heat conduction calculation of the wall, the finite difference method (implicit scheme) is used.

Simulation model

A detached house was taken up as a simulation model. The plan and elevation views are shown in Figure 1, the building material specifications in Figure 2, and the schedules of various items in Figure 3. The areas taken up were Sapporo(43°4′N, 141°21′E) as a cold weather region, Tokyo(35°39′N, 139°41′E) and Osaka(34°41′N, 135°30′E) as mild weather regions, and Naha(26°12′N, 127°41′E) as a hot weather region, and the Standard year of expanded AMeDAS weather data 1981-2000 for each area were used. The "flat plate thermo sense transformation material" or some others set to the wall surface will express the change of surface thermal properties. In the simulation, 8 ideally changing modes were set as shown in Figure 4. Mode A is "Non Changing Clothes", which means thermal properties do not change through the year; solar absorptivity(a) is set at 0.9, emissivity(ε) at 0.9.

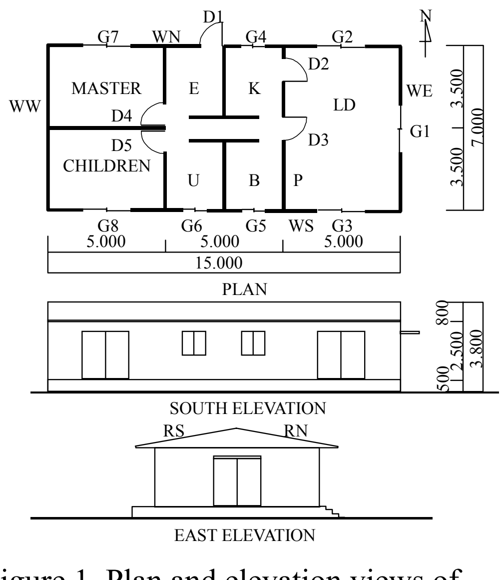


Figure 1. Plan and elevation views of detached house

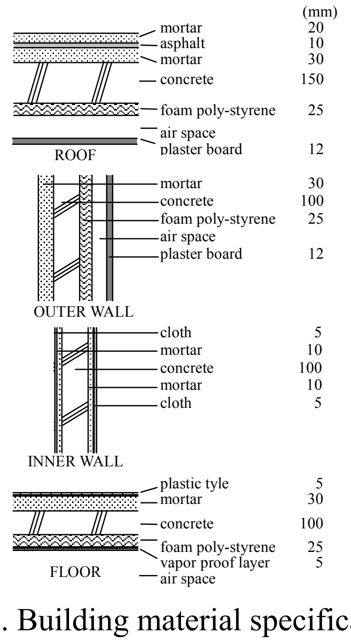


Figure 2. Building material specifications

The modes from B to H are "Changing Clothes", where the thermal characteristics are assumed to vary linearly between set two temperatures high and low. The properties of materials considered, solar absorptivity(a) is set at 0.2-0.9, emissivity(ε) at 0.2-0.9. From mode B to D, the set high and low temperatures become higher gradually. Mode E has a wider range of set temperature. The variation of the surface thermal properties can be fulfilled by rotating the cylinder and by changing a sunward surface ratio of selective emission to selective absorption (Figure 5 a)). The modes from F to H are also "CCM", where the thermal properties are assumed to change at a certain temperature, which can be fulfilled by rotating and turning over the flat plates with different properties between front and back surfaces (Figure 5 b)). From mode F to H, the set temperature becomes gradually higher. Both air-conditioning case (all rooms air-conditioned) shown in Figure 3 and un-air-conditioning case (all rooms un-air-conditioned, i.e. natural room temperature calculation case) were investigated, and in the former case, the room set temperature and humidity were

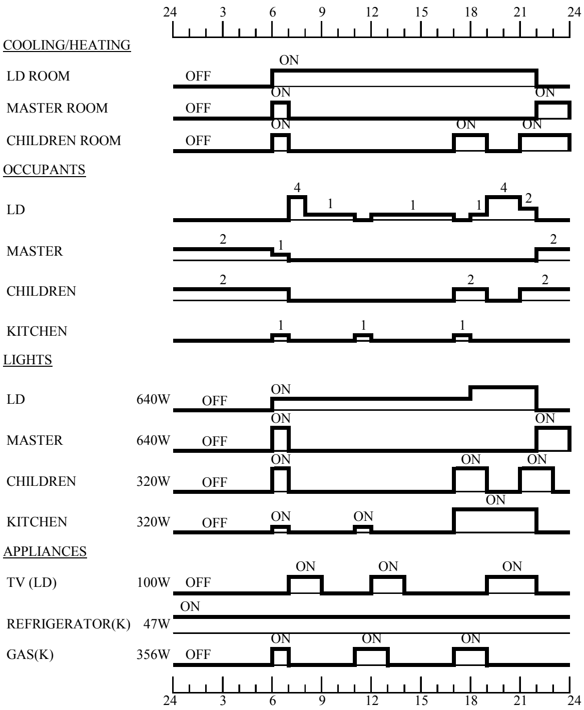


Figure 3. Schedule of items

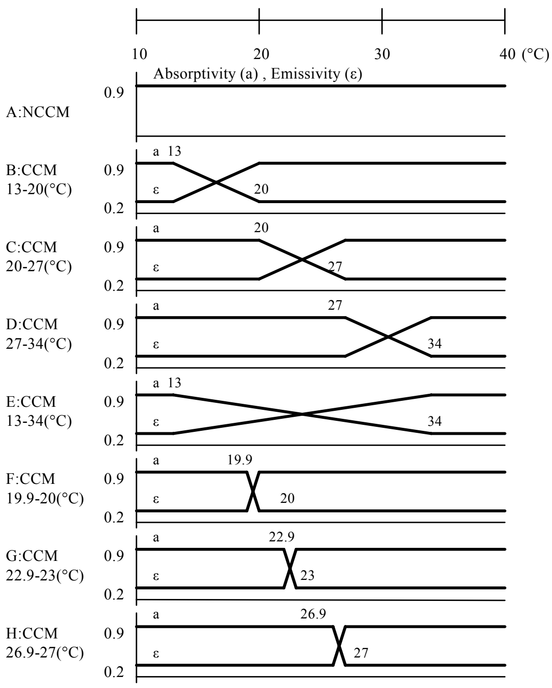
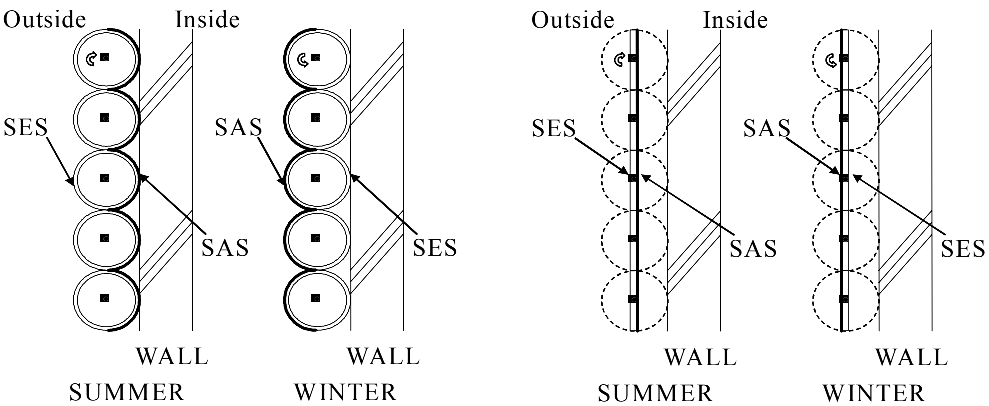


Figure 4. Variation of surface thermal properties



a) Cylinder type   b) Flat plate type

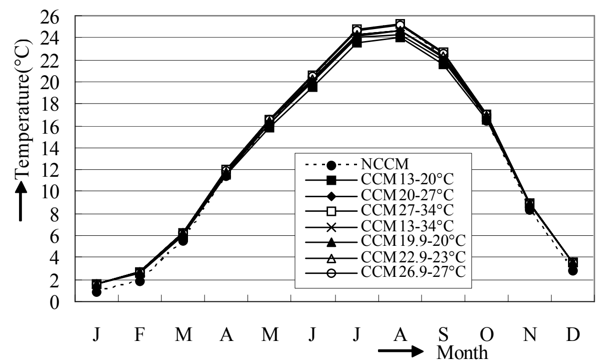
[SAS:Selective Absorption Surface   SES:Selective Emission Surface]

Figure 5. System examples to fulfill "changing clothes function"

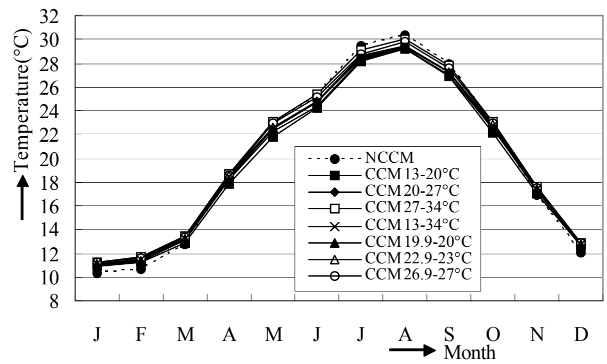
27ºC, 60% in the cooling period, and 20 ºC, 40% in the heating period. In the heat load calculation, the cooling period is set to be July-September in Sapporo, June-September in Tokyo and Osaka, and May-October in Naha, while the heating period November-April in Sapporo, December-March in Tokyo and Osaka, and December-February in Naha.

Simulation results

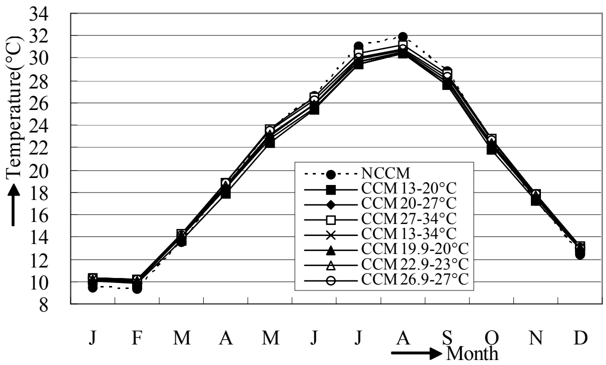
As one example of the examination results in each area, Figure 6 shows the monthly average natural room temperature in LD room in each mode. Table 1 shows the difference in the average natural room temperature between the “Non Changing Clothes Mode (NCCM)” and each “Changing Clothes Mode (CCM)” where the difference of room temperature is expressed; $\text{temperature}_{NCCM}$ – temperature_{CCM} in the cooling period and temperature_{CCM}– $\text{temperature}_{NCCM}$ in the heating period. In every room for all the regions, the difference in the average natural room temperatures between the “CCM” and “NCCM” is the greatest in mode B (in the cooling period) and mode D (in the heating period), meaning the “Changing Clothes” effect is significant, while mode D is the smallest in the cooling period, and mode B in the heating period. Table 2 shows the saving energy effect of the house total annual heat load (cooling load + |heating load|) in the air-conditioning case in each “CCM”, where the saving energy effect is expressed; (heat load_{NCCM} – heat load_{CCM})/ heat load_{NCCM}. Figure 7 shows the house total monthly and annual heat load in mode A, “Non Changing Clothes” and in the minimum mode of total heat load of the “CCB”, mode G in Sapporo and Tokyo, mode F in Osaka and Naha. Table 2 shows the saving energy effect in the cooling period becomes the most evident in mode B (84.4% in sensible heat load (SH)) and mode F (62.3% in total heat load (TH)) in Sapporo, in B (36.4% in SH) and F (27.7% in TH) in Tokyo, in B (35.8% in SH) and F (30.9% in TH) in Osaka, and in B (31.2% in SH) and F (25.6% in TH) in Naha.



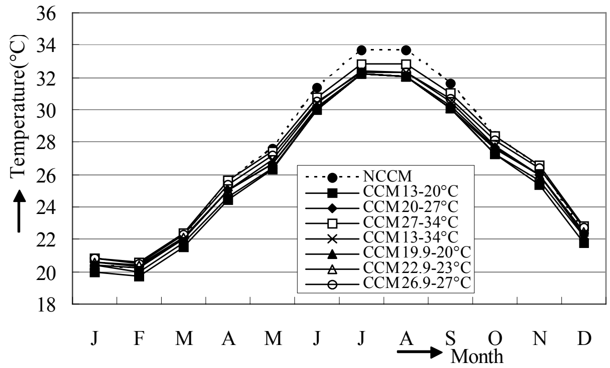
a) Sapporo



b) Tokyo



c) Osaka



d) Naha

Figure 6. Monthly average natural room temperature in LD room for each mode

Table 1. Differences of term average natural room temperature between “CCM” and NCCM”

[CP:Cooling Period HP:Heating Period]

a) Sapporo

(°C)

Effect	LD				K+B				E+U				MASTER				CHILDREN			
	CP		HP		CP		HP		CP		HP		CP		HP		CP		HP	
big	B	0.96	D	0.74	B	1.17	D	0.83	B	1.20	D	0.86	B	1.20	D	0.88	B	1.27	D	0.89
	F	0.60	H	0.72	F	0.67	H	0.81	F	0.66	H	0.84	F	0.70	H	0.85	F	0.77	H	0.87
	E	0.33	C	0.67	E	0.47	C	0.76	E	0.50	C	0.79	E	0.46	C	0.78	E	0.50	C	0.82
	C	0.26	G	0.67	C	0.40	G	0.76	C	0.40	G	0.79	C	0.40	G	0.78	C	0.43	G	0.82
	G	0.26	E	0.64	G	0.33	E	0.71	G	0.33	E	0.74	G	0.34	F	0.77	G	0.37	E	0.75
	H	-0.10	F	0.64	H	0.00	F	0.70	H	-0.04	F	0.72	H	0.04	E	0.76	H	0.03	F	0.74
small	D	-0.24	B	0.54	D	-0.20	B	0.58	D	-0.20	B	0.62	D	-0.20	B	0.65	D	-0.20	B	0.65

b) Tokyo

(°C)

Effect	LD				K+B				E+U				MASTER				CHILDREN			
	CP		HP		CP		HP		CP		HP		CP		HP		CP		HP	
big	B	1.15	D	0.88	B	1.32	D	0.93	B	1.40	D	0.98	B	1.40	D	1.03	B	1.38	D	1.03
	F	1.05	H	0.80	F	1.20	H	0.83	F	1.27	H	0.88	F	1.27	H	0.93	F	1.28	H	0.93
	G	0.85	C	0.70	C	0.97	C	0.75	C	1.00	C	0.83	G	1.00	C	0.80	C	1.00	C	0.85
	C	0.83	E	0.68	G	0.95	G	0.75	G	1.00	G	0.80	C	0.97	G	0.78	G	0.98	G	0.85
	E	0.73	G	0.68	E	0.87	E	0.68	E	0.90	E	0.73	E	0.95	E	0.70	E	0.88	E	0.75
	H	0.43	F	0.63	H	0.50	F	0.65	H	0.50	F	0.68	H	0.52	F	0.68	H	0.53	F	0.70
small	D	0.20	B	0.45	D	0.25	B	0.48	D	0.27	B	0.50	D	0.30	B	0.48	D	0.28	B	0.48

c) Osaka

(°C)

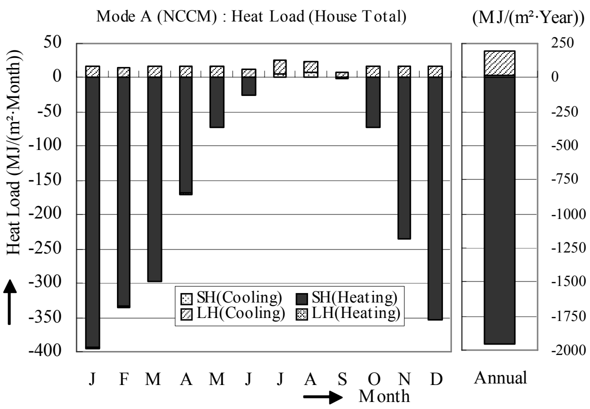
Effect	LD				K+B				E+U				MASTER				CHILDREN			
	CP		HP		CP		HP		CP		HP		CP		HP		CP		HP	
big	B	1.35	D	0.78	B	1.53	D	0.88	B	1.65	D	0.88	B	1.60	D	0.88	B	1.63	D	0.95
	F	1.33	H	0.73	F	1.48	H	0.85	F	1.58	H	0.83	F	1.55	H	0.83	F	1.58	H	0.88
	G	1.10	C	0.65	C	1.23	C	0.75	C	1.30	C	0.73	C	1.25	C	0.73	C	1.33	C	0.77
	C	1.08	G	0.65	G	1.20	G	0.75	G	1.30	G	0.73	G	1.25	G	0.73	G	1.33	G	0.77
	E	0.93	E	0.58	E	1.05	F	0.65	E	1.18	E	0.65	E	1.18	E	0.63	E	1.20	E	0.65
	H	0.75	F	0.58	H	0.83	E	0.62	H	0.90	F	0.63	H	0.88	F	0.57	H	0.95	F	0.63
small	D	0.40	B	0.42	D	0.45	B	0.45	D	0.53	B	0.40	D	0.53	B	0.40	D	0.55	B	0.45

d) Naha

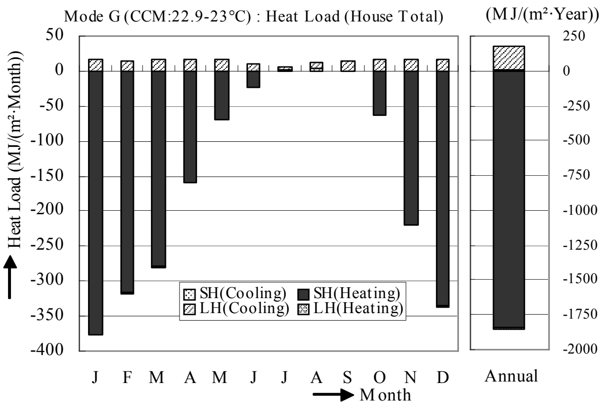
(°C)

Effect	LD				K+B				E+U				MASTER				CHILDREN			
	CP		HP		CP		HP		CP		HP		CP		HP		CP		HP	
big	B	1.40	D	0.43	B	1.58	D	0.46	B	1.70	D	0.46	B	1.70	D	0.50	B	1.73	D	0.53
	F	1.39	H	0.36	F	1.55	H	0.43	F	1.69	H	0.40	F	1.67	H	0.37	F	1.71	H	0.37
	G	1.27	G	0.20	G	1.40	G	0.23	G	1.54	G	0.23	G	1.53	G	0.24	G	1.56	G	0.20
	C	1.20	C	0.16	C	1.37	C	0.20	C	1.45	C	0.20	C	1.48	C	0.17	C	1.51	C	0.17
	E	1.02	E	0.00	E	1.17	E	0.00	E	1.27	E	0.00	E	1.33	F	0.00	E	1.30	E	-0.03
	H	0.89	F	-0.10	H	1.00	F	-0.04	H	1.09	F	-0.04	H	1.10	E	-0.03	H	1.13	F	-0.07
small	D	0.54	B	-0.47	D	0.60	B	-0.54	D	0.65	B	-0.54	D	0.68	B	-0.53	D	0.70	B	-0.67

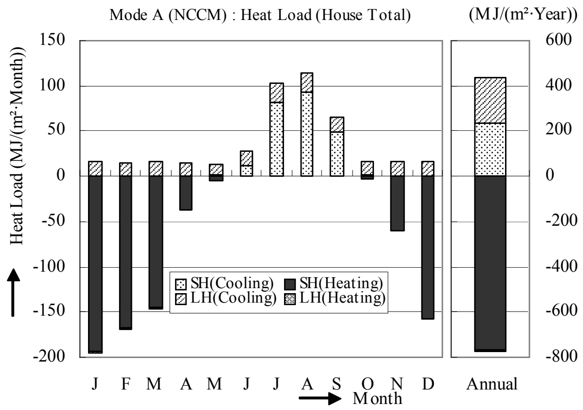
It also shows the saving energy effect in the heating period becomes the most evident in mode D (5.7% in SH) and mode D (5.7% in TH) in Sapporo, in D (14.7% in SH) and D (11.6% in TH) in Tokyo, in D (10.3% in SH) and D (10.3% in TH) in Osaka, and in D (37.7% in SH) and D (37.6% in TH) in Naha. On the contrary, in the cooling period, the effect becomes the least in mode H (in SH) and mode D (in TH) in Sapporo, in D (in both SH and TH) in Tokyo, in D (in both SH and TH) in Osaka, and in D (in both SH and TH) in Naha, while in the heating period, in mode B (in both SH and TH) in Sapporo, in B (in both SH and TH) in Tokyo, in B (in both SH and TH) in Osaka, and in B (in both SH and TH) in Naha. As the result, the most effective mode in saving energy all through the year is mode D (7.2% in SH) and mode G (6.3% in TH) in Sapporo, G (15.8% in SH) and G (13.9% in TH) in Tokyo, G (16.2% in SH) and F (14.0% in TH) in Osaka, and F (31.9% in SH) and F (23.5% in TH) in Naha. On the other hand, the least effective mode all through the year is mode B (in both SH and TH) in Sapporo, B (in both SH and TH) in Tokyo, B (in both SH and TH) in Osaka, and D (in both SH and TH) in Naha.



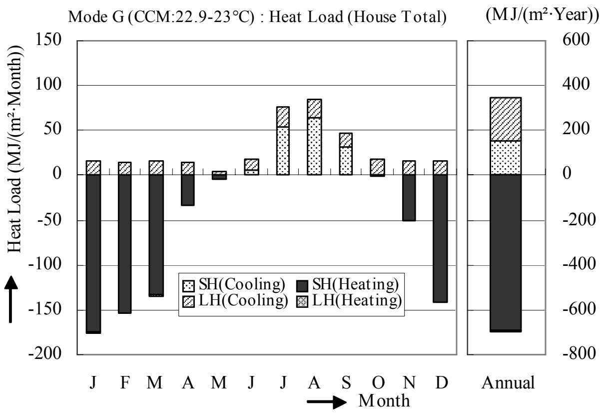
a) Sapporo(mode A)



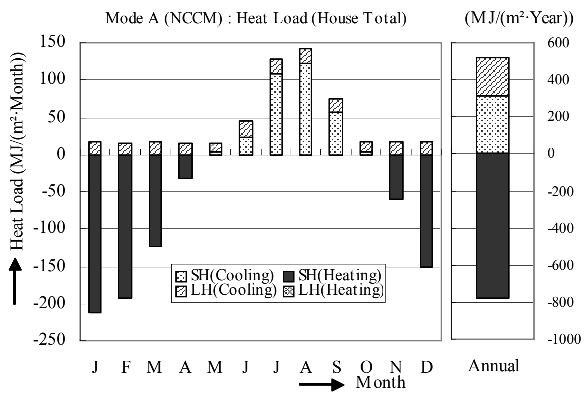
b) Sapporo(mode G)



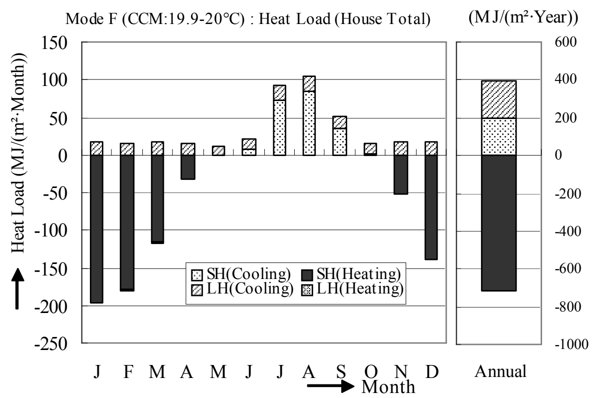
c) Tokyo (mode A)



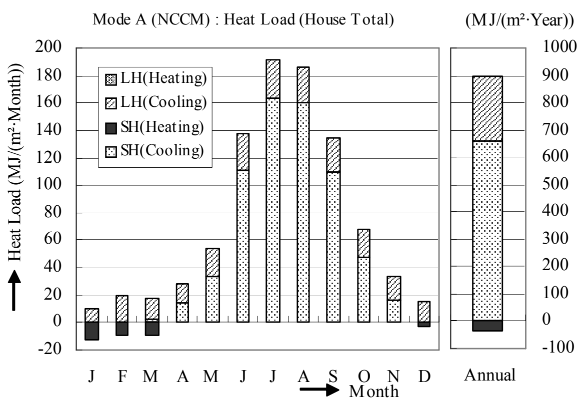
d) Tokyo (mode G)



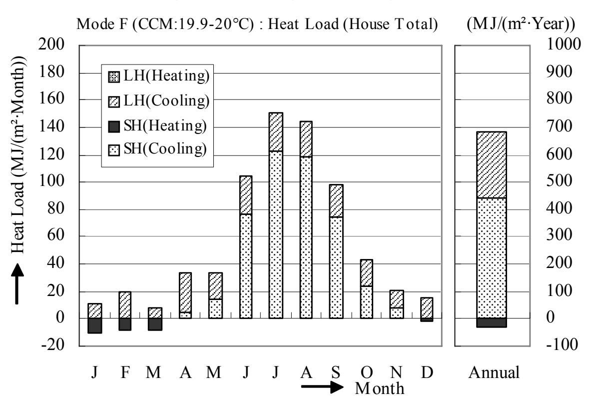
e) Osaka (mode A)



f) Osaka (mode F)



g) Naha(mode A)



h) Naha(mode F)

Figure 7. Monthly and annual heat load

Table 2. Energy saving effect

[SH:Sensible Heat Load TH:Total Heat Load]

a) Sapporo

(%)

Effect	Cooling Period				Heating Period				Annual			
	SH		TH		SH		TH		SH		TH	
big	B	84.4	F	62.3	D	5.7	D	5.7	D	7.2	G	6.3
	F	78.0	B	54.8	H	5.6	H	5.6	H	6.6	H	6.3
	C	62.6	E	43.1	C	5.3	C	5.3	G	6.0	C	6.2
	E	60.6	G	42.7	G	5.3	G	5.3	C	6.0	D	6.2
	G	60.3	C	41.5	E	4.9	E	4.9	E	5.3	F	5.9
	D	37.0	H	16.2	F	4.9	F	4.9	F	5.2	E	5.6
small	H	34.0	D	-6.0	B	4.2	B	4.2	B	3.6	B	4.1

b) Tokyo

(%)

Effect	Cooling Period				Heating Period				Annual			
	SH		TH		SH		TH		SH		TH	
big	B	36.4	B	27.7	D	14.7	D	11.6	G	15.8	G	13.9
	F	36.1	F	27.2	H	10.5	H	10.5	C	15.3	C	13.5
	G	34.3	G	26.8	C	9.5	C	9.4	F	14.9	H	12.6
	C	32.0	C	25.0	G	9.3	G	9.2	H	14.3	F	12.3
	E	27.6	E	21.6	E	8.3	E	8.3	D	13.4	D	11.8
	H	22.7	H	17.2	F	8.1	F	8.0	E	13.0	E	11.6
small	D	14.7	D	11.1	B	5.4	B	5.3	B	11.7	B	9.5

c) Osaka

(%)

Effect	Cooling Period				Heating Period				Annual			
	SH		TH		SH		TH		SH		TH	
big	B	35.8	F	30.9	D	10.3	D	10.3	G	16.2	F	14.0
	F	35.7	B	29.6	H	9.7	H	9.6	C	15.9	G	13.6
	G	33.1	G	26.5	C	8.6	C	8.6	F	15.8	C	13.5
	C	32.2	C	25.9	G	8.6	G	8.5	H	14.8	H	12.5
	E	28.2	E	22.7	E	7.3	E	7.3	E	13.5	E	11.4
	H	24.7	H	19.6	F	7.3	F	7.3	D	13.0	D	11.0
small	D	16.4	D	12.0	B	4.7	B	4.7	B	12.9	B	10.9

d) Naha

(%)

Effect	Cooling Period				Heating Period				Annual			
	SH		TH		SH		TH		SH		TH	
big	B	31.2	B	25.6	D	37.7	D	37.6	F	31.9	F	23.5
	F	31.2	F	25.6	H	33.8	H	33.2	G	30.6	G	22.9
	G	30.1	G	24.6	G	27.0	G	27.0	B	30.4	B	22.9
	C	28.9	C	23.6	C	25.3	C	25.3	C	29.5	C	22.1
	E	24.9	E	20.3	F	18.1	F	18.1	E	25.1	E	18.8
	H	23.6	H	19.2	E	12.7	E	12.7	H	23.2	H	17.2
small	D	14.8	D	11.9	B	-21.0	B	-20.9	D	15.3	D	11.2

Consideration

As for the "Changing Clothes Mode", the biggest saving energy effect all through the year was mode D and G in Sapporo, mode G in Tokyo, mode G and F in Osaka, and mode F in Naha. Mode B and F have a great effect in the cooling period; solar absorptivity is small while emissivity is big even from low surface temperature. Mode D is effective in the heating period; solar absorptivity is big while emissivity is small from high surface temperature. The annually effective modes are, on the whole, mode D in the cold weather, mode G in the mild weather and mode F in the hot weather, which means, in cold weather regions in Japan, the most effective mode in the heating period is considered as annually effective, while in hot weather regions, the effective one in the cooling period is.

DISCUSSION

This paper describes the thermal effect of the "changing clothes function" of the "Changing Clothes Building" whose roof and wall surfaces can autonomously change their thermal properties, such as solar absorptivity and emissivity. The regional variation of thermal effect and the desirable changing mode of the thermal properties of the building wall surfaces has been demonstrated, by estimating room temperature and heat load when the thermal properties vary according to various modes changed and by comparing the results obtained from the modes non-changed ("Non Changing Clothes Mode"). The results showed the remarkable thermal effectiveness of the "CCB". The annually effective modes in Japan were, on the whole, mode D in cold regions, mode G in mild regions and mode F in hot regions. The saving energy effect due to the "changing clothes function" in the typical detached house in Japan was 7 to 32% in house sensible heat load and 6 to 24% in house total (sensible and latent) heat load as compared to the "Non Changing Clothes". The effect in hot weather regions became remarkably bigger than the one in cold weather regions.

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Multiple Film Based Daylight Control System

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SUMMARY

A multiple film based daylight control system for window has been developed to maintain the illuminance level at task plane. The developed system consists of three films with a visual transmittivity (Tvis) range of 0.159 - 0.015 and a 2x55W dimmable compact fluorescent lamp (CFL) fixture. The system works on the principle of feedback control, which records the changes in light intensity with the help of photosensor placed on the ceiling and facing the task plane and accordingly adjusts the film-position as well as the control voltage of dimmable CFL to maintain the task plane illuminance within a tolerance of -10%, +20% of setpoint. Optimized use of solar illuminance saves energy used for the task lighting for the period from 8:00 to 19:00, which was found to be about 70%. The paper describes the design, implementation of the proposed system and the preliminary results for task light illuminance and energy savings achieved in laboratory setup.

INTRODUCTION

Window in a building provides daylight, ventilation and the outside view. A well designed fenestration system usually provides sufficient daylight during the day but it suffers with the problems of glare and heat. Usual glare control strategies deploy blinds or curtains which are not automated and require frequent manual intervention, hence most of the time these systems are kept closed [1]. Moreover, these systems block the outside view and cause lighting energy consumption when they are controlling the excess daylight [2]. Every household and firm spends about 20% of its annual energy budget for lighting purpose [3], [4]. Thus a need arises to develop smart light-controllers [5] for windows which not only maintain a constant light intensity in the room within the comfort levels of the occupants but also decrease glare and maintain outside view for the room. Field studies for such control systems have been done as discussed in [6], [7]. Emerging technologies such as electrochromic windows [8] satisfy the purpose of glare protection and clarity of outside view but they are expensive. The proposed system is not only cost-effective compared to electrochromic windows, but also provides the constant task lighting with the tolerance of -10% to +25% of setpoint with minimized glare.

The controller system was built and installed at the Center for IT in Building Science at International Institute of Information Technology, Hyderabad. The setup consists of a single room with a door, three windows and three twin lamp fluorescent fixtures along with a 2x55W dimmable CFL fixture. The system consists of three films of varying visual transmittivity, joined with each other at ends forming a long sheet and rolled over two rollers. By rolling both the rollers, the system can cover the window with different type of films and thus controlling the amount of daylight entering into the room. If required illumination at task plane is not being satisfied by sun light even after placing the most transparent film then the system can top up the task plane illuminance by operating dimmable CFLs. A graphical user interface has been provided on mobile phone, using which user can set the comfort value of task plane illuminance.

This paper describes the design and implementation of proposed system and the preliminary results of energy saving as well as the clarity of outside view achieved in laboratory setup.

METHODS

Laboratory setup

A room of the dimensions 5m x 10m with ceiling height of 4m was instrumented and configured to test the multiple film based daylight control system. The room consists of three windows W1, W2 and W3; and three fluorescent fixtures T1, T2 and T3 and a 2x55W dimmable CFL fixture with a control voltage range of 0-10V shown in figure 1. The CFL fixture has been placed at a distance 1.3m from ceiling and 1.7m from task table above W2 at a distance of 0.305m from the vertical wall containing W2.

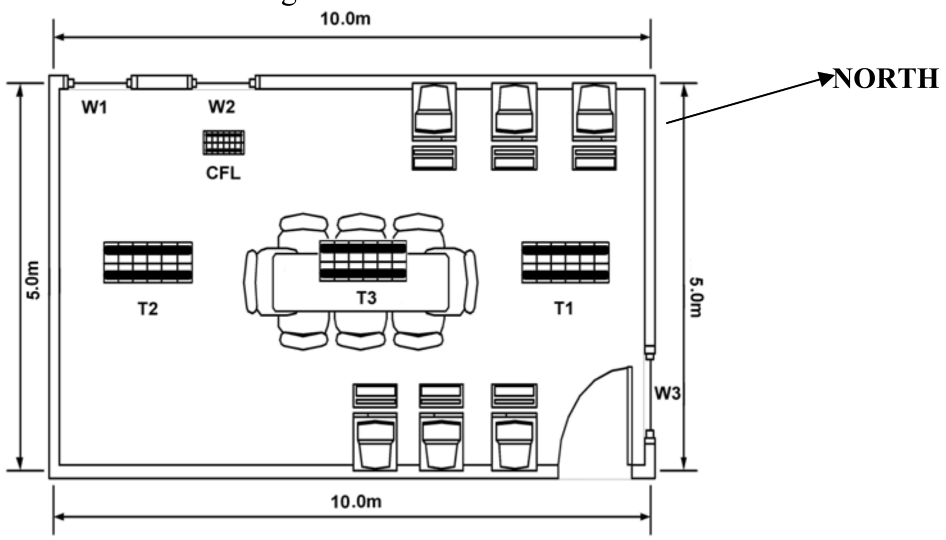


Figure1: Experiment room layout

Hardware setup

A National Instrument's data acquisition card (DAQ) PCI-6229 [9] was used for the setup, which has 32 analog input, 4 analog output channels and 4 digital ports.

The sheet of three films was rolled on two rollers. These rollers were connected to gears whose direction and duration of rotation was controlled by stepper motors. The sheet consisted of three tinted films each of varying transmittivity. The darker film was attached below the lighter one to provide less sun light at level of eyes [10].

A lux meter was placed at the centre of the task table which was used to monitor the actual illuminance on the table. A photosensor was installed in between two dimmable CFLs. The downward-facing, shielded photosensor sends out a proportional signal in response to the illuminance level within its field of view. The sheet used for daylighting control consisted of three different films each of different Tvis. Transmittivity of the three films comprising the controller are 0.159 (Film A), 0.055 (Film B), 0.015(Film C).

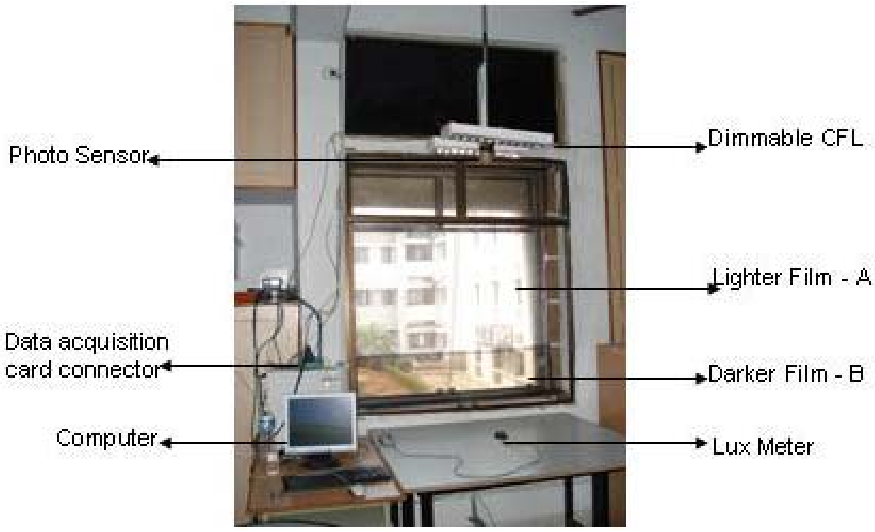


Figure 2: Photograph of installed system

Control Algorithm

System is designed to maintain the total illuminance from the artificial lights and daylight within the setpoint illuminance range and to block direct sun by adjusting the sheet position and control voltage of dimmable CFL.

Four correlations were required to determine the daylight and electric lighting contributions to the task plane illuminance in the space, where the average of illuminance at four points on task plane at a distance of 0.30m, 0.60m, 0.90m, 1.20 from window W2, I_{avg}, was used as the benchmark as suggested in [11]:

1. Correlation between lighting power consumption (W) and I_{avg}, electric lighting only.
2. Correlation between photosensor signals (V) to I_{avg}, electric lighting only.
3. Correlation between photosensor signals (V) to I_{avg}, daylight only.
4. The ratio of measured task plane illuminance to measured illuminance by photosensor as M factor [12] for any given instant in time for all three films.

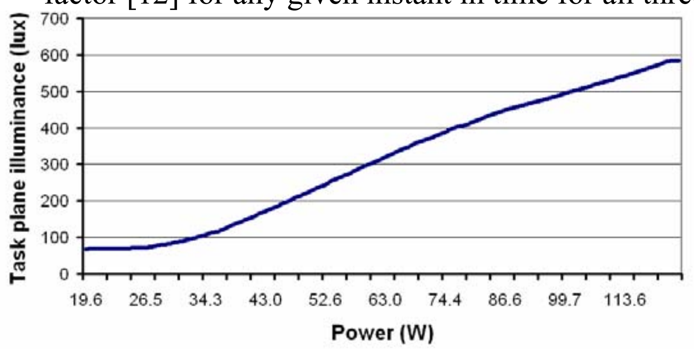


Figure 3: The correlation between electrical power consumption (W) and I_{avg}, electric lighting only. This shows the average task plane illuminance measured at night by varying control voltage of dimmable CFL.

Formula (1) calculated from night calibration is used to calculate the illuminance caused by artificial lighting during the experiment by using the control voltage given to the dimmer

$$y = 0.1711x^4 - 4.2481x^3 + 34.541x^2 - 39.846x + 78.923 \quad R2 = 0.9999,$$
$$y = \text{Task plane illuminance, } x = \text{Control voltage to dimmer of CFL} \tag{1}$$

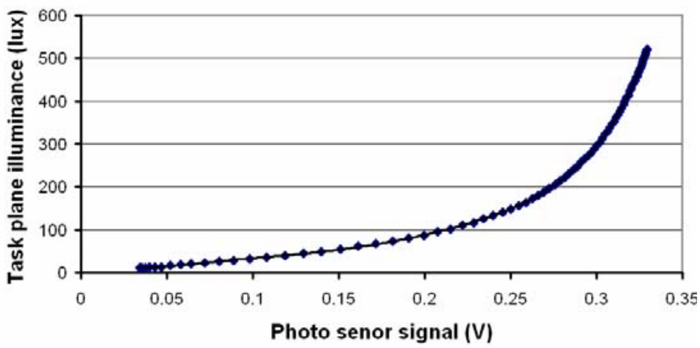


Figure 4: Correlation between photosensor (V) to I_{avg}, electric lighting only, with photosensor on top and lux meter on table.

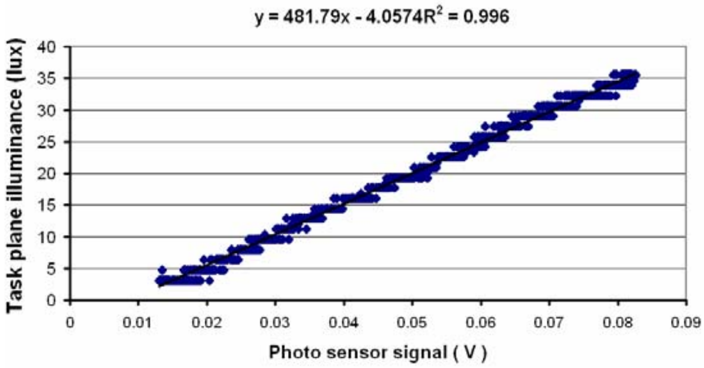


Figure 5: Correlation between photosensor (V) to I_{avg}, electric lighting only, with both the sensors on top. The correlation shown in Fig. 4 has photosensor output in the range from 0V to 0.35V and gives a non-linear relationship with task plane illuminance. But it was observed that during the experiment photosensor output remains in the range of 0V – 0.1V. To get more accurate readings in this range both photosensor and sensor of lux meter were kept on the top, downward facing. This correlation gives the formula (2), which is used to convert photosensor output to Lux within this range.

$$\text{Lux} = 481.79^{*}\text{voltage} - 4.0574 \tag{2}$$

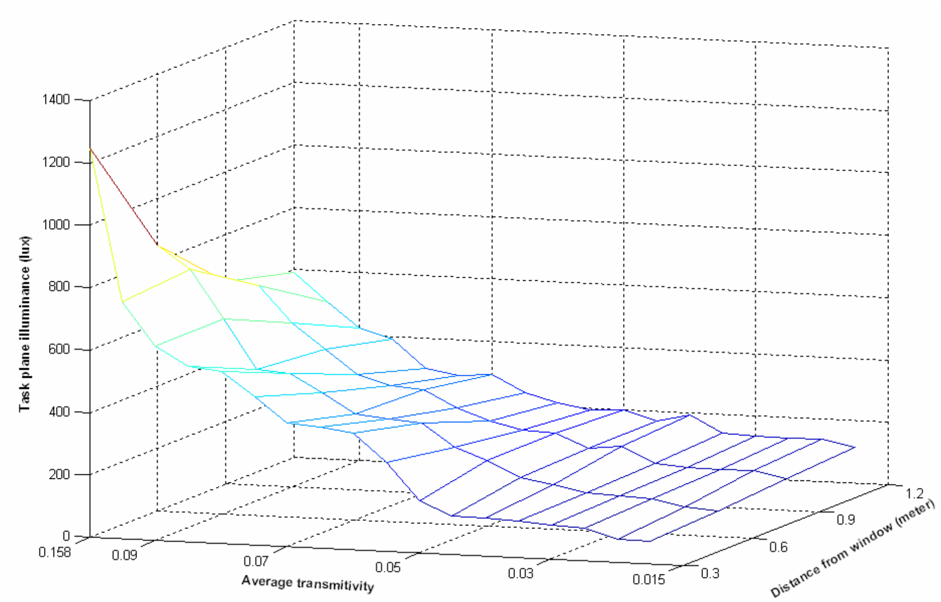


Figure 6: Correlation between photosensor signals (V) to Iavg, daylight only. This shows that the light distribution is uneven as lower film is darker than upper film, hence average light distribution has been considered.

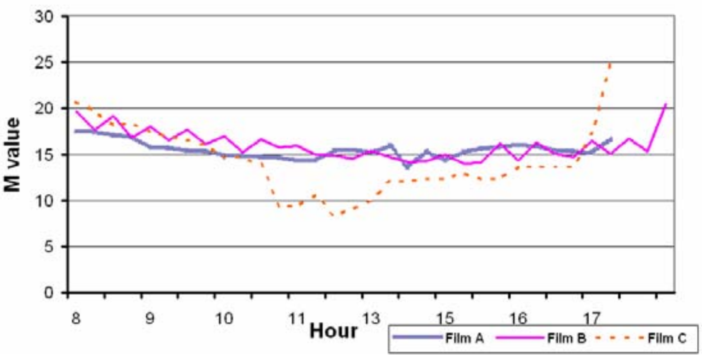


Figure 7: The ratio of measured task plane illuminance to measured illuminance by photosensor considered as M factor for any given time instance for all three films. This shows how M factor changes for different films at different times of the day. During the experiment the task plane illuminance was calculated by multiplying the lux output calculated from formula (2) with this M factor for Film A, which changes every hour.

The algorithm which has been implemented using in Lab VIEW is as follows:

1. Output of photosensor in voltage is converted to Lux using formula (2).
2. Multiply photosensor reading of illuminance by M factor for that time of the day to get task plane illuminance.
3. Calculated task plane illuminance is compare with user setpoint. If it is not in the tolerance range of -10%, +20% of setpoint then appropriate signals are sent to rotate motor and to change the signal voltage of dimmable CFLs.

The main objective of the controller is to attain the setpoint as quickly as possible. The illuminance output of dimmable CFL can be maneuvered quickly and with much ease as compared to the tinted film positioning. Thus by varying the signal voltage by 0.2V, the CFL output is set as the primary light source till the task plane illuminance comes in the tolerance range of setpoint and till this point of time, the film is the secondary light source. The signal voltage is changed with a small value so as to avoid the fluctuations. Once the tolerance range has been attained, the film is made the primary light source by slowly changing dimmer voltage and repositioning the film until end of sheet comes. The end of sheet has been detected by the amount of time for which film is rotating in a direction. If end of sheet is reached on either side of the film and the instantaneous light intensity is not within the tolerance range, then the dimmer voltage is adjusted to provide the light intensity within the range of setpoint.

Graphical user interface on mobile phone

A graphical user interface has been provided on the mobile phone using which user can change the setpoint of task plane illuminance as shown below in Fig. 8. The GUI has been developed for phones supporting Symbian[13] series by using symbian C++. User can eneter the value of setpoint either using key pad or value can be incremented or decremented with help of joystick. After enetring the setpoint when user presses the "Done" key, the value of setpoint gets tranfered to computer via bluetooth.

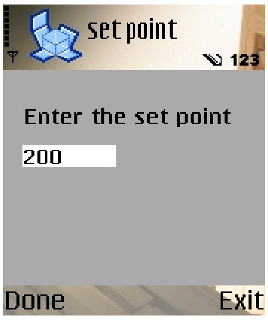


Figure 8: Graphical user interface on Mobile Phone to change the setpoint

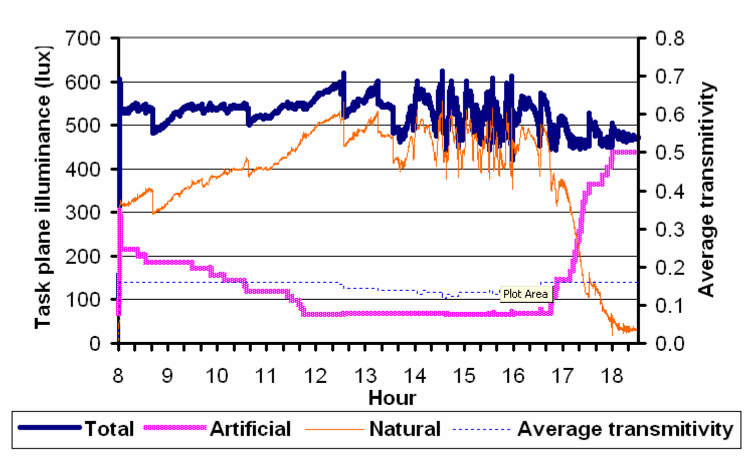
RESULTS

Reference Case

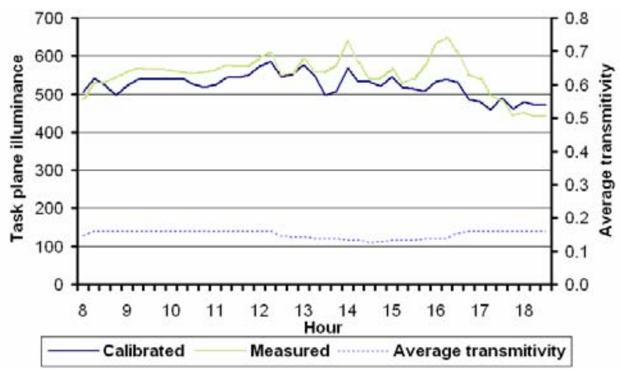
In the Reference case, the control voltage of dimmable CFL was at its maximum to achieve the setpoint of 500 lux, which consumes 124W power. The other three fluorescent tubes were also in their ON state and the W1, W2 and W3 were kept covered with opaque blinds to avoid the glare caused by the daylight and to determine the power consumption by the artificial light sources.

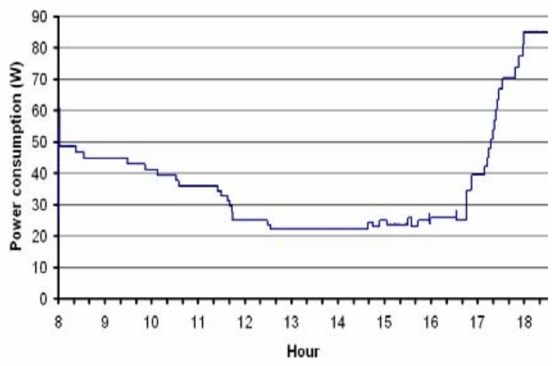
Monitored Data

The objective of the controller was to maintain the light intensity of the room within the tolerance range. The test value of the illuminance was set at 500 lux which maintained task light in the range 450-600 lux. During the test three fluorescent fixtures T1, T2 and T3 were kept ON and windowsW1 and W3 were kept covered with opaque vertical blind. It was found that the controller was able to maintain the task plane illuminance within the designated range for over 99% of the test time.



a)





b) c)

Figure 9: (Plotted with reading at every 10 seconds)

(a) Task plane illuminance for different timings during the experiment on a typical day
(b) Difference between calibrated and measured task plane illuminance
(c) Power consumption during the experiment by dimmable CFL. The average power consumption was 36.7W and reference case power consumption was 124W. Hence about 70% task light saving was achieved.

DISCUSSION

Controller Performance

The controller was tested on three aspects. First, how effectively can it maintain the room intensity within the tolerance range of setpoint, second, how much task lighting energy was saved and finally, the controller was evaluated by the user feedback as to whether it caused any type of discomfort. It was observed that the system was able to maintain task plane illuminance within the tolerance range for over 99% of the test duration irrespective of the local and global solar radiation intensity. Secondly, as explained in Fig. 9(c), about 70% task light saving was achieved with the base case of CFL being lit at the maximum intensity. Finally, user feedback was taken on comfort aspect achieved by the controller. It was found that the initial algorithm maneuvered the dimmer voltage in the large steps and this caused user some amount of discomfort. The algorithm was then refined to employ a ramp function for the dimmer voltage. The second point of discomfort was caused by the noise of the rollers maneuvering the sheet.

This noise was minimized by keeping the sheet and rollers in between the two pans of the double glazed window.

Conclusion

Multiple film based automated day lighting control system is a cost-effective solution for smart fenestration systems which not only provides dynamic sunlight control with glare reduction but also saves energy. As the sheet is in between the two pans of the double glazed window by using reflective films, the amount of heat entering the room can be reduced. The corresponding tests have to be performed to give the savings of air-conditioning in the room. The technology has potential to be used commercially. Pilot studies will help refine the technology.

AKNOLEDMENT

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Daylighting and thermal analysis of an obstructed double skin façade in hot arid areas

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SUMMARY

In recent years, there has been an increasing number of publications on the thermal and daylight performance of double skin facades in moderate climates. However, there is a scarcity of research on how different configurations of the cavity and its height can lead to an impact on internal cooling/ heating loads and the availability of daylight indoors in hot arid climates. This paper looks at comparing surface temperatures and daylight distribution between a continuous double skin façade over six floors and a corridor double skin façade (dividing the cavity by floor height) in a hot arid climate (Cairo-Egypt). A CFD code is used to predict the behaviour of air flows. RADIANCE software is used to test indoor luminance levels. Results indicate that there are no major differences in terms of surface temperatures between the two configurations, luminance levels are different in distribution but in both cases average luminance levels seem to provide adequate daylight levels indoors.

INTRODUCTION

Glass is a material with a huge innovative potential, offering the possibility of a completely transparent building envelope. Glass facades are seen not only as a symbol of financial prosperity but also as a reflection of organizational transparency. Although glass has not yet achieved its optimum thermal performance compared to traditional heavy insulated opaque technologies, its daylight transmittance and its effect on human well being can not be ignored.
The transfer of façade technologies to the Middle East can be traced back to 1888 [1] observing that in Egypt 'Today people have abandoned old ways in construction in favour of European style because of its more pleasant appearance and reduced building costs'. Since the 1970's it is observed that many major cities in the Middle East have aspired to increasing the glazed areas on facades. In Cairo, Dubai, Bahrain and Jeddah, although in hot arid climates, it is accepted that glazed facades would lead to increased cooling loads and energy consumption. The availability of air conditioning systems coupled with cheap electricity supply, and difficulty of cleaning and maintaining traditional systems of shading (wooden shutters or concrete louvers) drove design away from vernacular forms of providing solar shading.
Within this context it is prudent to look into the possibility of applying double skin façades as an architectural technology in hot arid climates before it is transferred -from European and North American countries to the Middle East, without an in-depth study of its impact on cooling load demand, daylight and human well being.
Reviews of simulation studies on double skin facades in hot arid climates [2],[3] and [4] and in moderate climates [5],[6],[7] [8] claim that the exterior leaf acts as a first line of defence against direct solar radiation passing through the internal leaf into rooms; trapped heat in the

double skin cavity is expected to induce natural buoyancy as a mean to reduce elevated air temperatures away from the inner building skin. This may result in additional reduction of conductive heat gains through the inner façade layers into the occupied space. Kim etal [9] suggested a double skin façade with fixed blinds as a refurbishment option for an existing single skin façade to reduce summer cooling loads and provide a more visually comfortable indoor environment in Seoul Korea. Hamza [10] predicted, that compared to a single skin façade with a 40% window-wall ratio and clear glazing, an 18% reduction in cooling loads maybe achieved if an absorptive tinted glazing is used on the outer leaf of a continuous cavity double skin façade configuration. Further analysis using CFD, validated the effect of buoyancy and the solar shading properties of a continuous double skin façade with a tinted glazing on the outer skins in a hot arid area [11]. This work departs from previous attempts by studying the thermal and daylight performance of a corridor double skin façade configuration where a walkway, inlet and outlets air vents at each floor level are included. In this paper, the term obstructed cavity is also used to express the configuration of a corridor double skin façade whenever reference to the interruption to the air flow, due to the walkways, within the cavity is made.

METHOD

In this work, two software packages are used to test the thermal (CFD) and daylight (RADIANCE) performance of a corridor double skin façade. The three dimensional CFD modelling provides a framework to study the performance of a double skin cavity as a full conjugate problem of the heat transfer. This conjugate problem involves coupling of ray tracing and radiation models in conjunction with the conduction-convection coupling through the domain. The three dimensional CFD modelling is used to study the thermal performance and the behaviour of the buoyancy driven flow in a corridor double skin façade, where the air flow is obstructed at each floor level by a walk way.

The metal walkway consists of longitudinal slots that run the length of the façade with 50% openings. The walkway is used for maintenance and cleaning but is also expected to reflect daylight deeper into the room while allowing air flow by buoyancy between floors. An inlet and outlet is provided at each floor level to augment the airflow in the cavity. Figure 1, explains the configuration of the continuous and obstructed double skin facade used in both software analysis.

The purpose of using a CFD code is to analyze the solar radiation through the intermediate air cavity in which buoyancy-driven flows augment turbulent mixed convection. The temperature of the air in the cavity is influenced by many interlinked factors such as solar radiation, outside temperature, wind speed and direction, type of glazing, and time. A finite volume discretization method of standard k-ε model using body-fitted coordinates and a SIMPLEC-based solution algorithm of the velocity-pressure coupling were used with a segregated solver. The momentum and energy equations were solved by the power-law and second order upwind scheme respectively. The PERSTO scheme was used in pressure interpolation. A ray-traced solar model is coupled with long wave radiation tracing to solve the complete solar and radiation fields as well as the convection and conduction. The contribution presented here is a conjugate heat transfer modelling used to simulate as near to actual conditions to predict the thermal performance of a corridor double skin façade configuration on an East and West orientation. The three dimensional model domains are cast over three computational zones; wind (external) zone with solar radiation entering the outer skin of tinted glass; buoyancy-driven air cavity zone with convection and transmitted solar radiation; and an internal zone with a brick solid wall and clear glazed windows of 40% window to wall ratio.

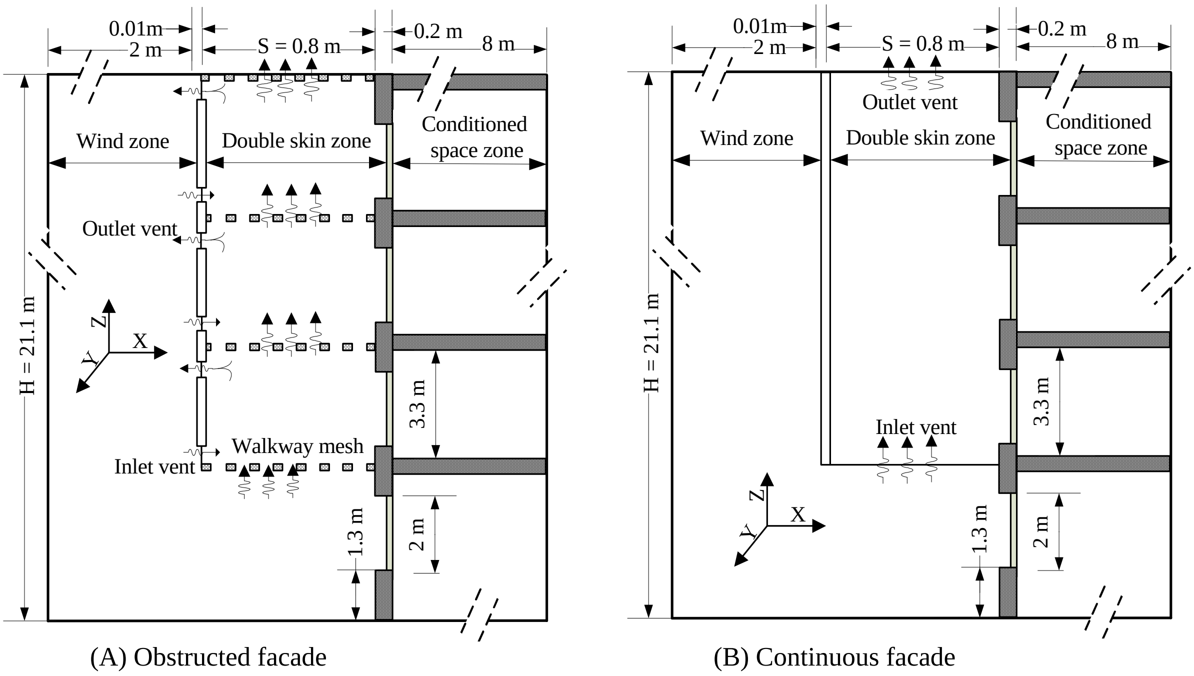


Figure (1) Schematic diagram of the computational domain fields

The heat gain due to direct solar, heat radiation and convection through the inner glass layer over time is presented. The length of the cavity is assumed to be 30m and of height 21.1m. The walkway is 50% solid, the voids are to maintain a vertical air flow between floors (Figure 1-A)

The boundary condition of the outside air temperature, wind speed and solar inputs over the time are adapted according to field data of ASHRAE. After some initial refinements to the mesh strategies to obtain reasonable accuracy, the grid system of 93*68*105 (674,340 grid points) is chosen. The values of the convergence criteria were less than 10^{-5} for all parameters. The governing equations are solved using FLUENT-6.2.16 CFD code. The turbulent flow through the double skin cavity is governed by mass, momentum and heat transfer equations. Steady state natural convection flow in the double skin cavity can be represent by the following equations;

$$\frac{\partial}{\partial x_i}\left(\rho U_i \varphi\right)=\frac{\partial}{\partial x_i}\left(\Gamma_\varphi \frac{\partial \varphi}{\partial x_i}\right)+S_\varphi \qquad (1)$$

Where φ represents the flow variable such as velocity, pressure, temperature and turbulent parameters. Boussinesq approximation model was adapted to obtain faster convergence criteria in solving the free convection governing equation, in which the Boussinesq approximation of $\rho=\rho_1\left(1-\beta\Delta T\right)$ is accurate for such kind of applications.

SOLAR RAY TRACING AND RADITION MODELS

The solar ray tracing model is used to calculate radiation effects from the sun's rays that enter a computational domain. Solar calculator was used to compute solar beam direction and irradiation for a given time, date, and position. These values are used as inputs to the solar ray tracing algorithm in relation to the sun direction vector, direct normal solar irradiation at

earth's surface, diffuse solar irradiation on vertical and horizontal surface and ground reflected (diffuse) solar irradiation on vertical surface.
The direct normal solar irradiation is computed using the ASHRAE Fair Weather Conditions method. At the earth's surface, the direct solar irradiance E_{DN}, is represented by

$$E_{DN} = \frac{A}{e^{\left(B/\sin\theta\right)}} \qquad (2)$$

The values of *A* and *B* vary during the year due to seasonal changes in the dust and water vapour content of the atmosphere [12]. The solar ray tracing algorithm is used to predict the direct energy source that results from incident solar radiation. The resulting heat flux that is computed by the solar ray tracing algorithm is coupled with the calculation via a source term in the energy equation. A two-band spectral model is used for direct solar illumination and accounts for separate material properties in the visible and infrared bands. A single-band hemispherical-averaged spectral model is used for diffuse radiation. Opaque materials are characterized in terms of two-band absorptivities. A transparent material requires specification of absorptivity and transmissivity in the visible and infrared portions of the spectrum. A P1-radiation model (equation 3) is coupled with the solar ray tracing model to deal with emission from surfaces, and the reflecting component of the primary incident load through the computational domain;

$$q_r = \frac{1}{3\left(a+\sigma_s\right)-C\sigma_s}\nabla G \qquad (3)$$

BOUNDAY CONDITIONS

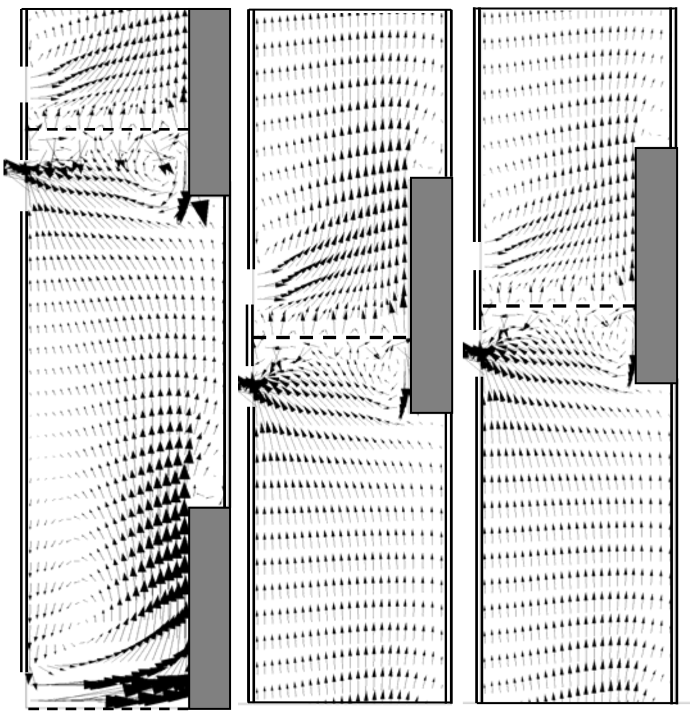
Cairo is located on latitude (30.13^o), longitude (31.4^o), time zone relative to GMT (GMT+2). The double skin façade configuration is studied on the East and West orientation. The ambient temperature and the wind speed are applied according to the local climate field data of ASHRAE [13]. The buoyancy effect was activated and the gravity is set to -9.81m/s^2 with z-direction.

In order to minimize the solar radiation through the double skin cavity, the outer glass skin is absorbing glass (0.01 m thickness) of transmissivity (τ = 0.45) and absorptivity (α = 0.49) at zero incident radiation angle. The inner skin glass (windows covering 40% of the inner facade) is clear glass (0.006 m thickness) of transmissivity (τ = 0.78) and absorptivity (α = 0.15) at zero incident radiation angle. Glazing optical properties are dependent on incident angle, which were varied as the incident angle increases from 0^o to 90^o. The conditioned space is assumed at constant temperature of 22^{o}C.

RESULTS AND DISCUSSION

The flow field of the obstructed double skin cavity is complicated, where in each floor there are two perpendicular inlets (walkway mesh and the side inlet vent) and two perpendicular outlets (walkway mesh and the side outlet vent). A further complication is the expansion and contraction of the flow field due to changes in the cavity's cross section. The double skin façade is built starting from the 1st floor leaving the ground floor as a single skin configuration; this is to reduce the possibility of fine dust rising into the cavity. . The flow field of the obstructed double skin cavity for the 1st, 2nd and 3rd floors on the 21 of June is illustrated in Fig. (2). Air is induced into the cavity from two inlets on the 1st floor (walkway mesh and the side inlet vent), this explains the higher mass flow rate at the bottom of the cavity. The CFD simulations indicate that the air flow through the inlets forces the air into the cavity horizontally till it meets the brick wall then it moves vertically into the cavity. At the zone where the walkway, inlet and outlets are in close proximity in intermediate floor levels (2^{nd} to 6^{th} floors) a turbulent flow is created with a reverse flow nearer to the inside brick wall. CFD simulations predict that the mass flow rate decreases within the cavity height, this is expected due to the presence of the walkway which affects the airflow between floors. The local tracing of the air velocity and temperature through the cavity at different level is illustrated in Fig. (3). At the walkway, the flow velocity is small (0.2 m/s) and then increases to a maximum (0.4 m/s) aided by the inlet flow. As the flow moves vertically on each floor level it expands to the full depth of the cavity in front of the glass decreasing the air velocity at mid floor level height. The flow then contracts as the cavity depth decreases which leads to increases in velocity aided by buoyancy and the outflow vent. However, the variations in air velocity are small. CFD simulations predict that air velocities in the continuous cavity reach a maximum of 0.8 m/sec [2]. It could be concluded that although the air velocity in the cavity is slightly slowed down by the presence of the walkways, this effect has been offset by the introduction of the inlet and outlets at each floor level.

The prediction of the heat fluxes passing through the inner skin (windows) which impact the cooling load of the building for both the West and East orientation are illustrated in Figures (4 -5). These heat fluxes are the transmitted part of the sun ray passing indoors (direct solar). The longwave radiation exchange between the inner surface of the inner skin and the interior of the conditioned space is generated due to the temperature difference (ranged from 8 ^oC at 6 am to 20 ^oC at 4 pm). The convective heat transfer is generated between the conditioned air and the inner surface of the inner skin.



(A) 1^{st} floor **(B) 3^{th} floor** **(C) 4^{th} floor**

Figure (2) Flow vector through the double skin cavity

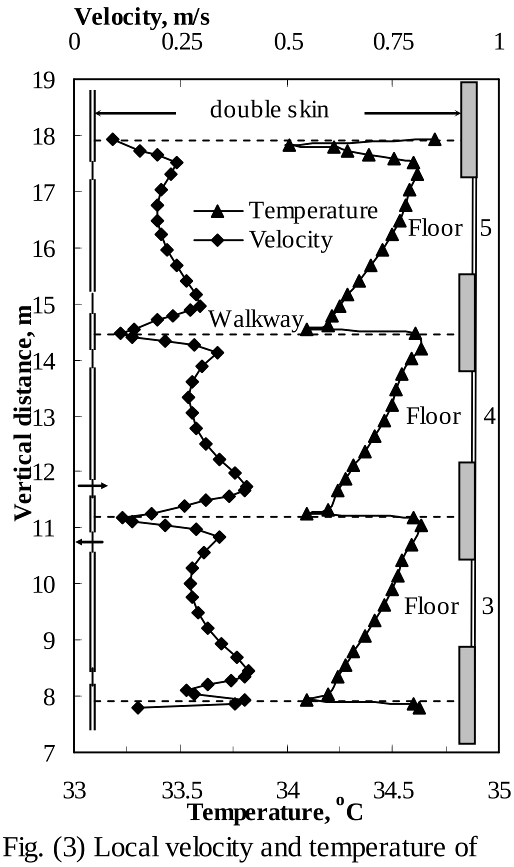


Fig. (3) Local velocity and temperature of floors No. 3, 4, 5 at mid of x and y axes

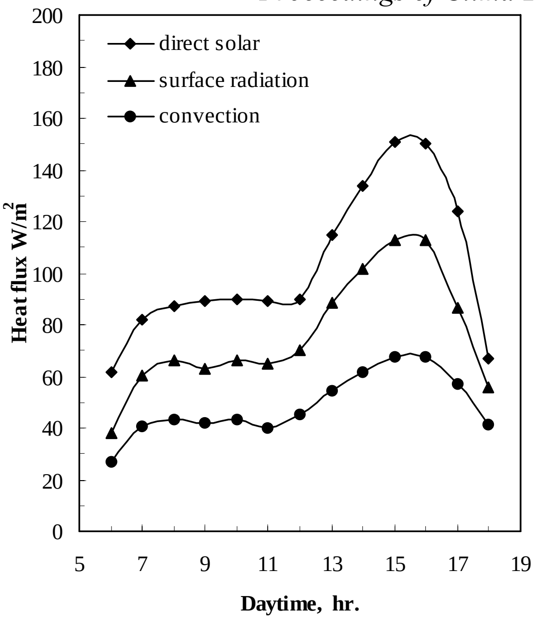


Figure (4) Radiation, convection and direct solar heat fluxes passing through the inner skin that impact the cooling load of the west building

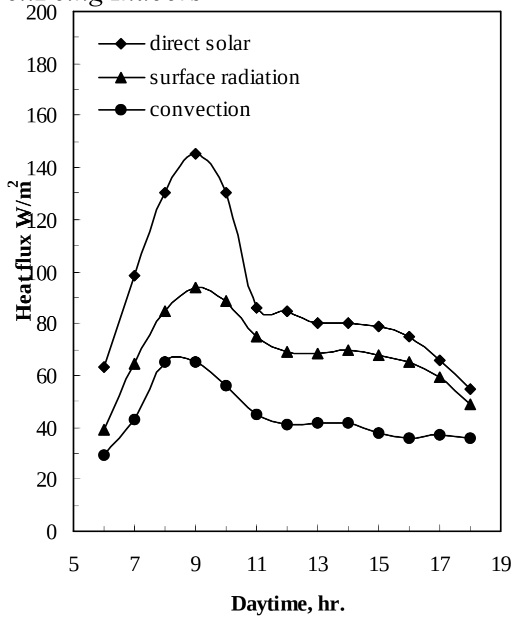
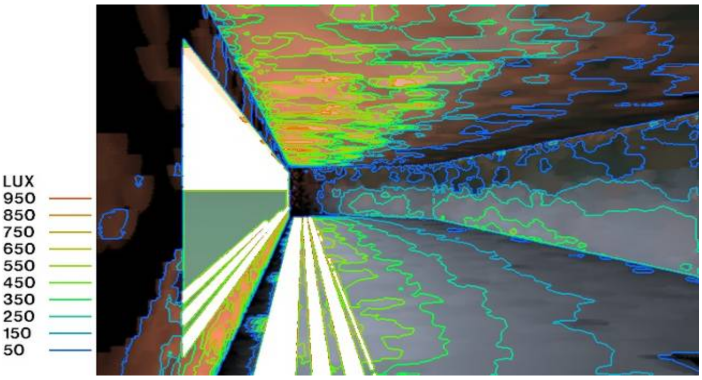


Figure (5) Radiation, convection and direct solar heat fluxes passing through the inner skin that impact the cooling load of the east building

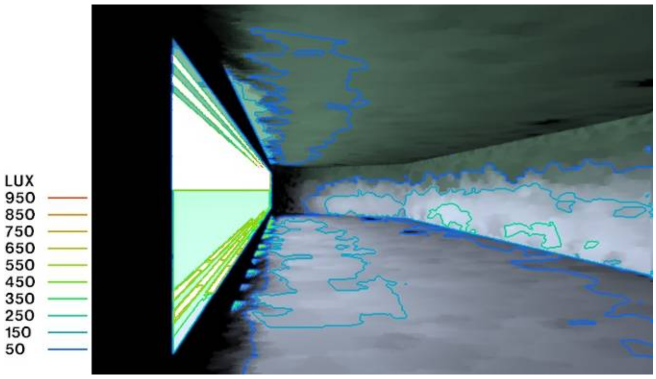
Results indicates that compared with surface radiation and convection through the inner façade, direct solar radiation dominates the heat gain. For example, at 4 pm on the West building orientation, the direct solar heat flux = 150 W/m^2 while surface radiation heat flux = 113 W/m^2 and the convection heat flux = 68 W/m$^{2.}$ Direct solar heat flux is 25 % and 62 % higher than the radiation and convection heat fluxes indoors respectively.

Daylight simulations on the East and West orientations are simulated using RADIANCE software Figure (6). The view point is taken at the corner of the room as one of the poor locations in terms of daylight. It is observed that the luminance levels on the back wall tend to be similar in the case of the obstructed cavity and the continuous cavity. This is due to the deep room configuration of the room where only reflections from the other room surfaces would illuminate this surface. Luminance levels are tested on both orientations at direct and diffuse solar radiation incidence. Luminance levels and the distribution of daylight indicate minor value differences for both orientations and therefore the Lux grid is only presented for the West orientation.

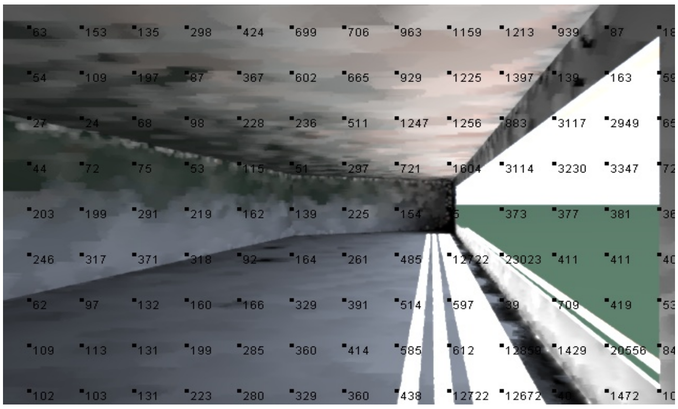
During direct solar radiation hours for both façade configurations luminance levels beside the window area exceed 10,000 Lux which will require the use of blinds to control excessive glare on the work surfaces. Reflectance from the walkway leads to stripes of sharp contrast between light and shade in the area nearer to the window. On the floor level the walkways decrease the luminance levels than the continuous cavity leading to a darker indoor environment with an average of 150 Lux on the working planes during diffuse hours. This may lead to offsetting any benefits from the walkways decreasing surface temperatures within the cavity, as it necessitates the use of artificial lighting to provide the required maintained luminance levels. However, the walkways reflect higher daylight onto the ceilings than those in the case of the continuous cavity, this may give a better psychological effect. It is understood that changing the depth of the room and the position of the walkways and their material may lead to different results; therefore this is an area for future research.



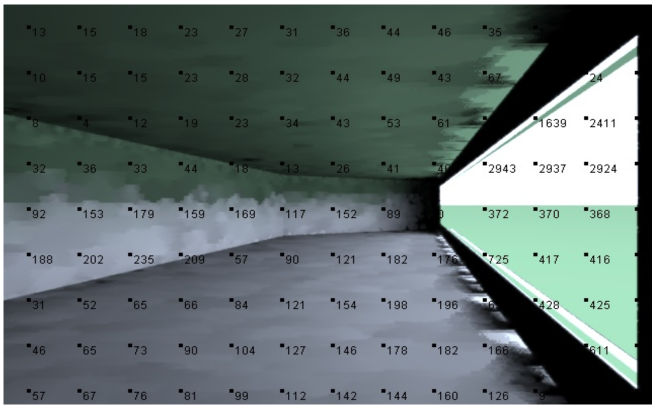
Corridor DSF cavity: East luminance levels for glass transmittance RGB=0.55 at 10am



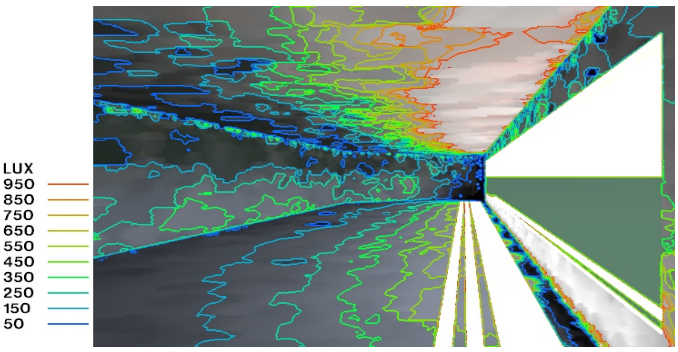
Corridor DSF cavity: East luminance levels for glass transmittance RGB=0.55 at 2pm



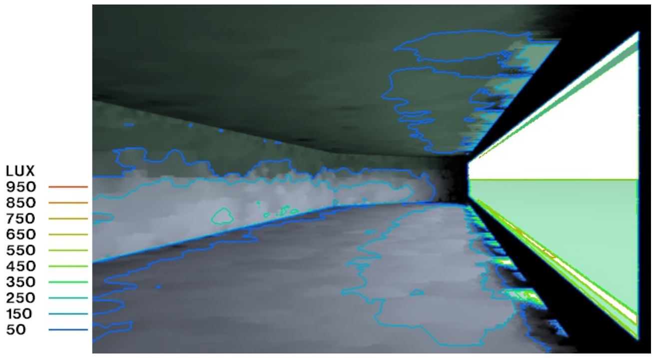
Obstructed DSF cavity: West daylight transmittance for RGB=0.55 at 10a.m.



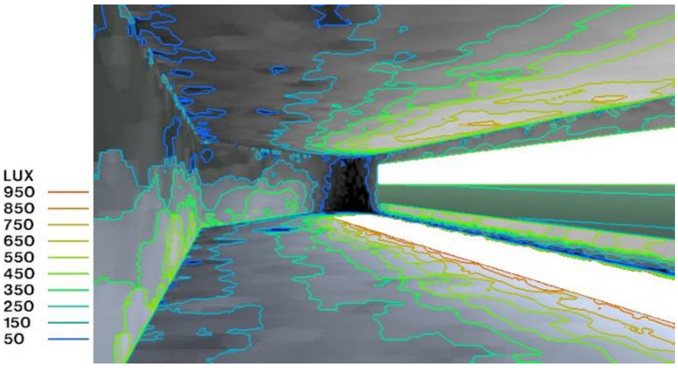
Obstructed DSF cavity: West daylight transmittance for RGB=0.55 at 2pm



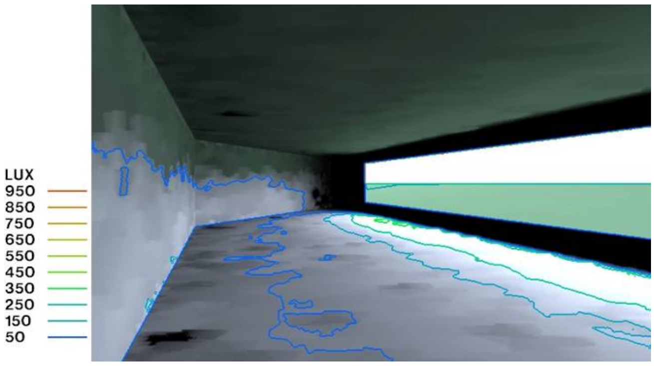
Obstructed DSF cavity: West daylight transmittance for RGB=0.55 at 10a.m.



Obstructed DSF cavity: West daylight transmittance for RGB=0.55 at 2pm



Continuous DSF cavity: West daylight transmittance for RGB=0.55 at 10a.m.



Continuous DSF cavity: West daylight transmittance for RGB=0.55 at 2pm

Figure 6: A comparison of illuminance levels on East and West orientation between a continuous and obstructed double skin façade configuration

Figure (7) compares the variations of temperatures between a continuous and an obstructed façade at mid of X-Z and X-Y planes at 2p.m on the East orientation on the fourth floor, the graph indicates that the air temperature of the continuous cavity is about 1.5 ^{o}C higher than that of the obstructed cavity. Compared to the continuous cavity, the obstructed double skin façade experiences a turbulent flow at outlet and inlet levels. The ambient air which induced through inlet vents leads to a minor reduction of temperatures in the cavity and on the surface temperatures of the brick. The absence of this cooling effect in the continuous double skin configuration leads to increasing the surface temperature of the glass facing the cavity by about 1.5^{o}C. Figure (8), explains the higher surface radiation and convection heat gain of the continuous type generated from the inner surface of the inner skin, compared to the obstructed configuration.

In the case of the obstructed double skin cavity, the effective factor affecting the flow rate is the wind pressure and its direction aided by the buoyancy effect due to temperatures in the cavity being higher than ambient. In the case of the continuous double skin, the mass flow rate is higher due to trapped heat over the height of the cavity assisting the buoyancy effect as higher temperature differences between the cavity and ambient temperatures.

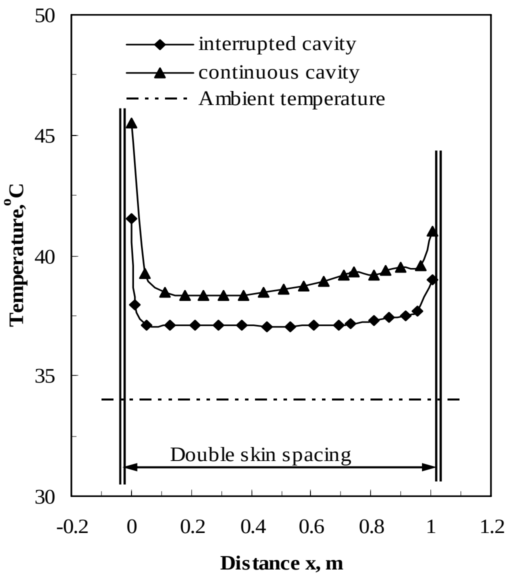


Figure (7) Temperature variation through double skin width of the continuous and obstructed cavity

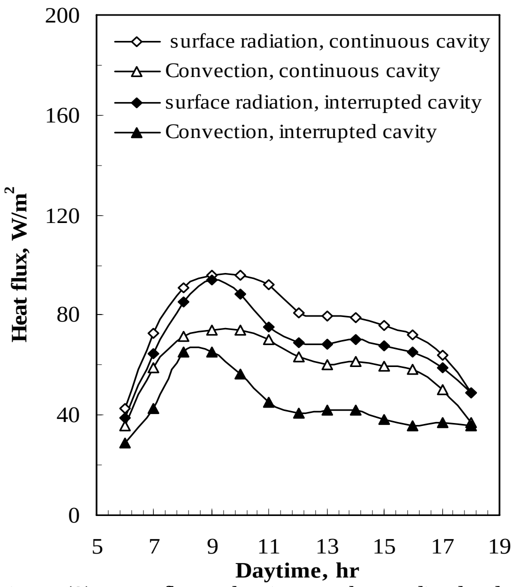


Figure (8) Heat fluxes that impact the cooling load through the inner skin of the continuous and obstructed double skin cavity

CONCLUSIONS:

In this work a comparison between the thermal and daylight performance of a continuous and an obstructed cavity of a double skin façade configuration is undertaken.

Both double skin façade configurations seem to perform similarly in reducing direct solar radiation indoors. Direct solar radiation in hot arid climates is the main contributor to increasing cooling loads. From a thermal point of view, CFD results indicate a turbulent air flow at the inlet, walkway and outlet zone. This turbulence leads to a minor back flow within the cavity that helps in reducing inner surface temperatures by about 1.5^{0}C, the significance of this reduction on cooling loads needs to be researched in future studies.
From a daylight perspective, during direct solar radiation hours, very high luminance levels in the window area will necessitate the use of blinds indoors. However, during diffuse solar

radiation, at view points away from the main daylight area, it is predicted that the obstructed double skin façade will further decrease the daylight levels on the work plane to an average of 150Lux (compared to a 250 Lux in continuous facades). The reduction in day light levels will lead to switching on of electrical lights to provide the minimum of 300Lux required for office work [15]. Simulation results in both façade configurations predicts more than adequate luminance levels [300 Lux] in the main daylight area (6m depth).Further research is needed to test the impact of using different glazing types on the exterior façade and optimizing daylight performance in the obstructed double skin façade.

NOMENCLATURE

a	Absorption coefficient	ΔT	Temperature difference, K
A, B	Constants	U_i	*Velocity vector*
C	Anisotropic phase function coefficient	V	*Air velocity,* m/s
E_{DN}	Solar irradiance, W	Γ_{φ}	Diffusion coefficient, $N\ s/m^2$
G	Incident radiation, W	α	Absorptance of the glass
H	Building height, m	β	Thermal expansion coefficient, $1/K$
L	Office depth, m	θ	Incident angle, degree
q	Heat flux, W/m^2	σ	Scattering coefficient
S	Cavity width, m	τ	Transmittance of the glass
S_{φ}	Source term.	ρ	Density, kg/m^3
T	Temperature, K	ρ_1	Initial density, kg/m^3

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Modeling the Heat Gain of a Window With an Interior Shade, How Much Energy Really Gets In?

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ABSTRACT

Not long ago the ASHRAE Technical Committee on Load Calculation, TC 4.1, had a "bake off" of sorts between different peak air-conditioning load calculation schemes and programs. One of the outcomes of this exercise was the realization that practitioners and software developers make largely different assumptions about how solar energy absorbed by window glass and by window shades contributes to the room solar heat gain. For unshaded glass windows there is general agreement [ASHRAE Handbook of Fundamentals, 2005]. For a shade however, there were two extremes in the models—one assumes that the shade rejects most of the solar energy that it does not transmit, logical if the shade is highly reflective and the glass highly transmissive, the other assumes that all radiation absorbed by the shade is immediately convected into the room.

Suffice it to say that we are not the first to derive and present the fundamental equations of this heat transfer problem. What we have done is to avoid any simplifying assumptions in formulating the problem while allowing that some physical constants, convection coefficients in particular, are not well known and need to be parameterized. We whet the readers' appetite by revealing that for a glass/shade system where the glass was 22% transmissive and the shade 52% transmissive, the total heat gain to the room from this window assembly was nearly half of the incident radiation. Of course "It all depends!"

INTRODUCTION

Not long ago the ASHRAE Technical Committee on Load Calculation, TC 4.1, had a "bake off" of sorts between different peak air-conditioning load calculation schemes and programs. One of the outcomes of this exercise was the realization that practitioners and software developers make largely different assumptions about how solar energy absorbed by window glass and by window shades contributes to the room solar heat gain. For unshaded glass windows there is general agreement [ASHRAE Handbook of Fundamentals, 2005]. For a shade however, there were two extremes in the models—one assumes that the shade rejects most of the solar energy that it does not transmit, logical if the shade is highly reflective and the glass highly transmissive, the other assumes that all radiation absorbed by the shade is immediately convected into the room.

Suffice it to say that we are not the first to derive and present the fundamental equations of this heat transfer problem (see [McCluney and Mills, 1993, and Duffie and Beckman, 1991]). What we have done is to avoid any simplifying assumptions in formulating the problem while allowing that some physical constants, convection coefficients in particular, are not well known and need to be parameterized. We whet the readers' appetite by revealing that for a glass/shade system where the glass was 22% transmissive and the shade 52% transmissive, the total heat gain to the room from this window assembly was nearly half of the incident radiation. Of course “It all depends!”

OBJECTIVE

The objective of this study was to model the true solar heat gain of glass and shade. Older, rough calculations have assumed that a 50% opaque shade will reduce the solar heat gain through the glazing by something just less than 50%. One of the less well understood parts of these estimates is the energy convected from the glass and shade into the room.

The model developed under this study provides an accurate and complete calculation scheme to support the performance of a shade having various transmissivities and absorptivities. The absorptivities are important for the calculation of solar interaction between the inside surface of the glass and shade and to determine the temperature of the shade facing the occupants in a room. More importantly, inside and outside surface convection coefficients can be varied to establish limits on how they impact solar heat gain.

BACKGROUND

We know that if a porous shade is suspended between the inside surface of the glass and the adjoining room, the solar radiation will be reduced. However a detailed model is required to address several issues. These include: How much energy is reflected between the outer surface of the shade and the inner surface of the glass and what happens to this energy? How much energy is re-radiated back into the room through the shade? How much energy is convected from the glass and shade into the room.

THE MATHMATICAL CALCULATIONS

The model developed permits the parametric evaluation of the window/shading characteristics including: The effect of different transmissivities of the shade. The surface temperature of the glass with the shade in place. The surface temperature of the shade in the cavity. Convection coefficients, both inside and outside.

Note that in the model the effects of the sash and frame have been excluded. However, ASHRAE Standards 90.1 and 90.2 and national Energy Codes require that fenestration ratings be determined in accordance with NFRC (National Fenestration Rating Council) procedures. These procedures (NFRC 200 for solar heat gain coefficient) explicitly rate the entire fenestration product including the frame, sash, and glass. See the NFRC website at www.nfrc.org for further information.

Modeling Incident solar radiation (this section is presented for the sake of completeness)

The solar constant, G_{sc}, is the energy from the sun, per unit time, received on a unit area of surface perpendicular to the direction of propagation of the radiation, at mean earth-sun distance, outside of the atmosphere. Measurements of the constant were made with high altitude aircraft flights, balloons, and spacecraft [Thekaedara et.al. 1971 and 1976]. Later, spacecraft and rocket flight data were reported [see Hickey et.al, Wilson et.al., and Duncan et al.] and led to the adoption of the value of 1367 W/m^2 for the solar constant by the World Radiation Center. We will use this value to calculate the beam, diffuse and reflected radiation on the exterior of a window.

The distance from the earth to the sun varies with time of year. Hence the extraterrestrial radiation, G_{on}, measured on the plane normal to the radiation on the *n*th day of the year is given by:

$$G_{on} = G_{sc}(1 + 0.033\cos\frac{360n}{365}) \qquad \text{Eq.[1]}$$

Clear sky radiation on the earth's surface can be estimated by calculating the atmospheric transmittance for beam radiation, $\tau_{batmos,}$ from a method presented by Hottel [Hottel, 1976].

$$\tau_{batmos} = a_o + a_1 \exp(-k/\cos\theta_z) \qquad \text{Eq[2]}$$

θ_Z = solar zenith angle, a_o=0.3887, a_1=0.5496, k =0.3176, (for mid-latitude winter)
Liu and Jordan, 1960 formulated an empirical relationship for diffuse radiation for clear days:

$$\tau_{datmos} = \frac{Gd}{Go} = 0.271 - 0.294\tau_b \qquad \text{Eq[3]}$$

where G_o=$G_{on}(\cos(\Theta_z))$.

As an example, for a clear day on December 21st in Los Angeles[1], the above formulations yield: τ_{batmoa}=0.6938 –atmospheric transmittance beam radiation, τ_{datmos}=0.0670 –atmospheric transmittance diffuse radiation, G_{on}=1411 W/m^2 –extraterrestrial normal beam radiation, G_{cnb}=979.3 W/m^2 –normal beam radiation at the earth's surface, G_o=761.5 W/m^2 --extraterrestrial beam radiation on a horizontal plane, G_{cd}=94.59 W/m^2 –diffuse radiation at the earth's surface.

Beam and Diffuse Radiation Transmitted and Absorbed

Figure 1 shows a schematic of the glass and porous shade system. The arrows are a symbolic representation of the short wave energy, accounting for radiation, transmitted, reflected, and absorbed. The diagram is for beam radiation but a similar accounting applies to diffuse radiation.

"Abs" denotes the fraction of absorbed short wave energy for the glass or shade on each successive reflection. The portion of the incident beam short wave energy that is ultimately absorbed by the shade, Abs_{bss}, is:

$$Abs_{bss} = \tau_{gb}\alpha_{ss} \sum_{n=0}^{\infty} [\rho_{ss}\rho_{gd}]^n = \frac{\tau_{gb}\alpha_{ss}}{1-\rho_{ss}\rho_{gd}} \qquad \text{Eq[4]}$$

τ_{gb}=short wave transmittance of the glass (a function of incidence angle, θ), α_{ss}=short wave absorptance of the shade, ρ_{gd}=diffuse reflectance of the glass[2]

---

[1] We use this example location and date throughout the paper a representing a hot winter climate when south sun will be at a maximum. Also, accounting for cloudy weather is possible using the model provided but since the paper is motivated by and concerned with peak load calculation we have not considered cloudy days in our example.

[2]Brandemuehl and Beckman have shown that the effective incidence angle for isotropic diffuse radiation is approximately 60 degrees. [Brandemuehl and Beckman, 1980]. integration performed using manufacturer's data shown in the next section produced results that agree to within 1% of the Brandemuehl/Beckman results.

The incident radiation on the inside of the glass is diffuse radiation reflected from the shade. Optical properties for diffuse radiation for the glass can be determined for isotropic radiation by integration over all angles of incidence. This was done using glass properties from a manufacturer and shown in the next section of this report.

Similarly, the short wave beam radiation fraction absorbed by the glass, Abs_{bg}, is:

$$Abs_{bg} = \alpha_{gb} + \tau_{gb}\alpha_{gd}\sum_{n=0}^{\infty}[\rho_{ss}^{n+1}\rho_{gd}^{n}] = \alpha_{gb} + \frac{\tau_{gb}\alpha_{gd}\rho_{ss}}{1-\rho_{ss}\rho_{gd}} \qquad \text{Eq[5]}$$

where variables are as defined before and α_{gb}= beam absorptance of the glass (a function of incidence angle, θ), α_{gd}= diffuse absorptance of glass. Equations 4 and 5 also apply for diffuse radiation Abs_{dss} and Abs_{dg}, but τ_{gb} is replaced with τ_{gd} and α_{gb} is replaced with α_{gd} (see footnote 7).

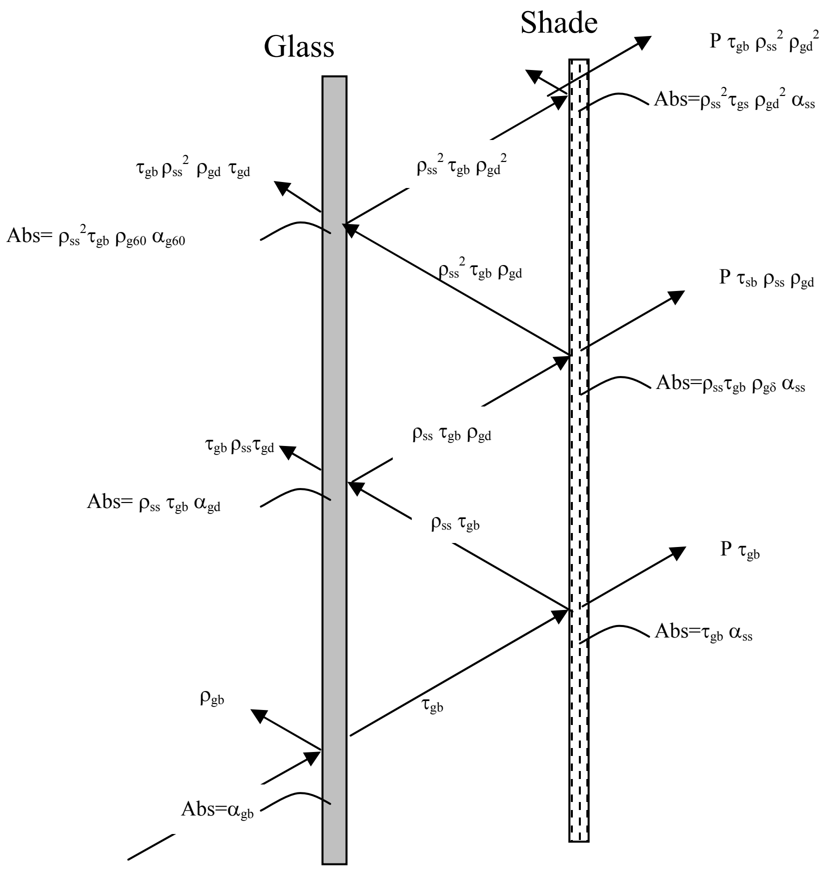

Figure 1. Short wave energy flows.

We can now do a short wave energy balance:

$$\dot{Q}_{gs} = \left[G_{cnb}\cos\theta Abs_{bg} + G_{cd}Abs_{dg}\right]^{3} \qquad \text{Eq[6]}$$

$$\dot{Q}_{ss} = \left[G_{cnb}\cos\theta Abs_{bss} + G_{cd}Abs_{dss}\right] \qquad \text{Eq[7]}$$

[3] We assume a uniform diffuse short wave field. This is accurate except for the lower floors where the diffuse radiation is a mix of diffuse sky radiation and ground reflected radiation. We have ignored ground reflected radiation in this analysis but its inclusion would be straight forward.

P = porosity or open fraction of the shade (this is the transmittance of the shade), $\dot{Q}_{gs}$ and $\dot{Q}_{ss}$ are the short wave radiation absorbed by the glass and shade respectively. Some of the radiation reflected from the shade is reflected again by the glass and then transmitted through the shade to the room. Referring to Figure 3 the quantity is:

$$\dot{Q}_{refin} = G_{cnb}\cos\theta\left[\frac{(P * \tau_{gs})}{(1 - \rho_{ss} * \rho_{sd})} - P * \tau_{gs}\right] + G_{cd}\left[\frac{(P * \tau_{gs})}{(1 - \rho_{ss} * \rho_{sd})} - P * \tau_{gs}\right] \qquad \text{Eq[8]}$$

Glass Properties

Data has been obtained for a heat absorbing glass. This product represents an application where total heat gain might be significantly higher than intuition may suggest because its transmissivity is so low. We will begin our analysis with this product but will move on to others. Figure 2 shows the transmittance, absorptance, and reflectance for the glass as a function of incident angle. In general, these properties can be characterized as functions of the cosine of the incidence angle, Θ. An incidence angle modifier can be defined as the ratio of the property at angle of incidence Θ to the property at normal incidence.[4] For example:

$$K_{transmittance} = \frac{\tau_{\theta}}{\tau_{n}} \qquad \text{Eq[9]}$$

Plots of incidence angle modifiers for transmittance, absorptance, and reflectance versus $\frac{1}{\cos(\theta)} - 1$ are shown in Figure 3. Curve fits are shown for transmittance and reflectance – the fits are generally excellent. The equations for the trend lines can be used to calculate transmittance and reflectance for any angle Θ less than 80 degrees and, since $\tau+\alpha+\rho=1$, the absorptance can be calculated. For a winter design day at noon in Los Angeles, Θ is 22.6 degrees.

[4] The incidence angle modifier (IAM) has its roots in the experimental characterization of solar collectors. The definition used here is a slight modification of the IAM used with solar collectors.

Figure 2. Optical properties of heat absorbing glass.

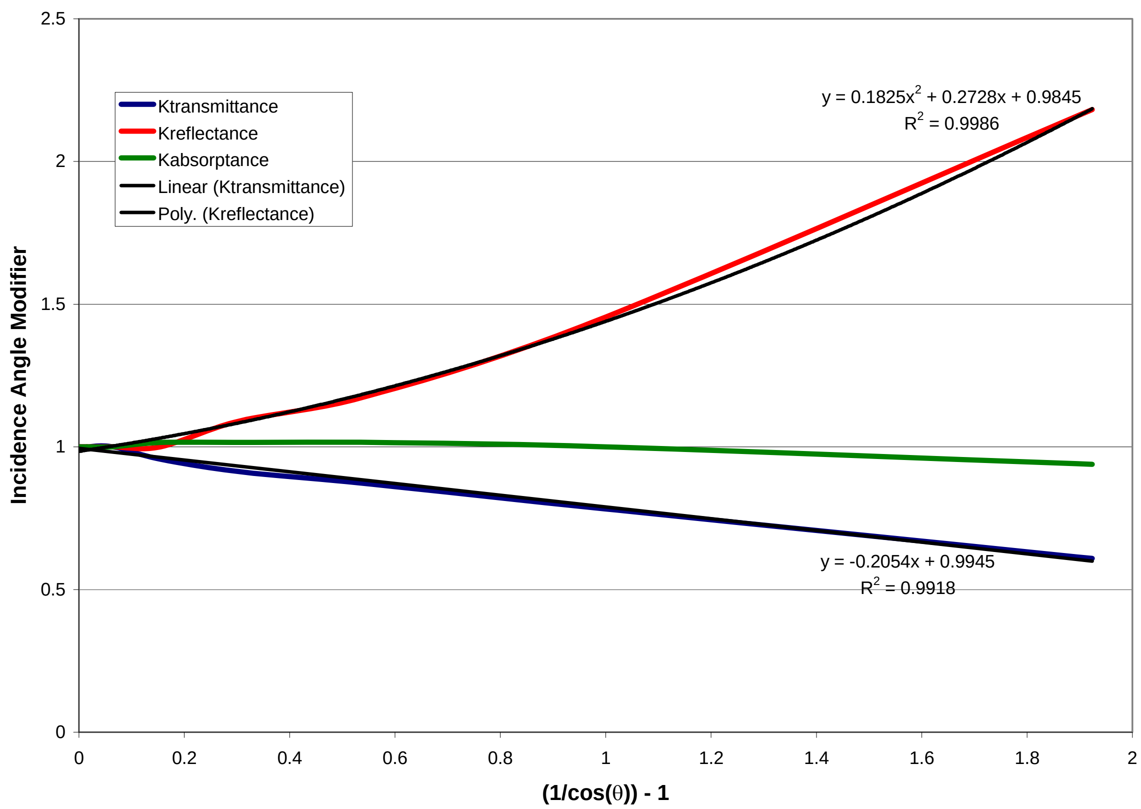

Figure 3. Incidence angle modifiers versus (1/cos(θ)) – 1)

Long Wave Radiation Exchange

The long wave radiation exchange between the glass and the shade can be characterized by treating the problem as one of gray body radiation exchange with essentially the same geometric interpretation as for short wave radiation. The interior is assumed to behave like a cavity and hence a black body. The long wave radiation heat exchange between the glass and shade can then be written as:

$$\dot{Q}_{glsl} = \frac{\varepsilon_{ssl}(1-P)\varepsilon_{gl}\sigma(T_g^4 - T_s^4)}{1-(1-\varepsilon_{gl})(1-P)(1-\varepsilon_{ssl})} \quad W/m^2 \text{ into the mesh} \qquad \text{Eq[10]}$$

where σ =Boltzman constant = 5.670E-08 $W/m^2\text{-}K^4$ and T_g and T_s are the glass and shade temperatures in degrees Kelvin. Note that the conductivity of the glass and shade are sufficiently high that their temperatures can be assumed to be locally uniform. Similarly,

$$\dot{Q}_{slgl} = -\dot{Q}_{glsl} \quad W/m^2 \text{ into the glass} \qquad \text{Eq[11]}$$

The long wave radiation from the interior to the glass is:

$$\dot{Q}_{igl} = \frac{P\sigma(T_i^4 - T_g^4)}{1-(1-\varepsilon_{gl})(1-P)(1-\varepsilon_{sl})} \quad W/m^2 \text{ into the glass} \qquad \text{Eq[12]}$$

The glass also exchanges long wave radiation with the outdoors. Assuming that all the surroundings are at the outdoor temperature, the long wave radiation received by the glass from the outdoor environment is:

$$\dot{Q}_{ogl} = \varepsilon_{gl}\sigma(T_o^4 - T_g^4) \qquad \text{Eq[13]}$$

Similarly, the radiation from the interior to the shade can be written as:

$$\dot{Q}_{isl} = \frac{P(1-\varepsilon_{gl})(1-P)\varepsilon_{sl}\sigma(T_i^4 - T_g^4)}{1-(1-\varepsilon_{gl})(1-P)(1-\varepsilon_{sl})} + (1-P)\varepsilon_{sl}\sigma(T_i^4 - T_s^4) \qquad \text{Eq[14]}$$

Hence the total incoming long wave radiation for the glass is:

$$\dot{Q}_{gl} = \dot{Q}_{ogl} + \dot{Q}_{slgl} + \dot{Q}_{igl} \qquad \text{Eq[15]}$$

and for the shade:

$$\dot{Q}_{sl} = \dot{Q}_{glsl} + \dot{Q}_{isl} \qquad \text{Eq[16]}$$

Convection

Heat is transferred from or to the glass and shade by convection to the interior air and outdoor air. For the glass, heat flow in by convection is:

$$\dot{Q}_{gc} = h_i * (T_i - T_g) + h_o * (T_o - T_g) \qquad \text{Eq[17]}$$

where h_i=interior convective heat transfer coefficient W/m^2-K (5 was used initially)[5], h_o=exterior convective heat transfer coefficient W/m^2-K (15 was used initially). For the shade:

$$\dot{Q}_{sc} = h_i 2(1 - P)(T_i - T_s) \qquad \text{Eq[18]}$$

Total Heat Balance

We now note that the heat flows for long and short wave radiation and for convection have been defined as being into the glass and shade. Since the sum of the energy into the glass and into the shade has to be zero in steady state, some flows will have a negative sign indicating that the heat flow is out of the material. The overall grand total heat balance for the glass is:

$$\dot{Q}_g = \dot{Q}_{gs} + \dot{Q}_{gl} + \dot{Q}_{gc} = 0 \qquad \text{Eq[19]}$$

For the shade:

$$\dot{Q}_s = \dot{Q}_{ss} + \dot{Q}_{sl} + \dot{Q}_{sc} = 0 \qquad \text{Eq[20]}$$

(As a reminder, all flows and heat balances are based on a unit area of the exterior façade, i.e. one square meter of glass adjacent to one square meter of shade)

The total radiant heat flow transmitted and reradiated into the room is:

$$\dot{Q}_{gainrad} = -\dot{Q}_{isl} - \dot{Q}_{igl} + P(G_{cnb}\cos(\theta)\tau_{gs} + G_{cd}\tau_{gd}) + \dot{Q}_{refin} \qquad \text{Eq[21]}$$

The total heat flow into the room caused by radiation and convection from/to the glass and shade is:

$$\dot{Q}_{gaintot} = \dot{Q}_{gainrad} - h_i(T_i - T_g) - \dot{Q}_{sc} \qquad \text{Eq[22]}$$

Engineering Equation Solver

The model, as defined by the above equations, was implemented in Engineering Equation Solver (EES), a tool for solving simultaneous equations, especially those involving solar energy relations and thermodynamic properties [Engineering Equation Solver, 2005].

GENERAL RESULTS

The results are from a simulation of a south facing room on a hot winter day in Los Angeles (outside temperature 35 ^{o}C, 95 ^{o}F). This illustrates how the model can be used. The glass in this example was assumed to be oriented vertically and facing due south. However, any tilt or azimuth angle can be explored using the EES model. Energy flows for the glass/shade ensemble were enumerated for this day and are shown in Figure 4 below.

The figure shows that about 45% of the incident radiation becomes a heat gain to the room. This occurs even though the transmittance of the glass is about 0.22 and the transmittance of the shade is 0.53. The long wave radiation and energy convected into the room from the glass and shade contribute significantly to the heat gain (in addition to what is transmitted directly). Multiple reflections between the shade and glass, sometimes not considered in simple models, also impact heat gain, both directly and as they affect the surface temperatures of the glass and shade.

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[5] Convection coefficients from the ASHRAE Handbook of Fundamentals were used as starting points but the ASHRAE values are higher than those used because they include equivalent radiant transfer, which is handled explicitly in this model. See ASHRAE Handbook of Fundamentals. Even taking this into account both these coefficients may be high and we will return to this issue later.

This study also evaluated the affects of shade "porosity," the fraction of open area of the shade or its transmissivity. Their impact was evaluated using parametric tables. Figure 5 shows the variation of glass and shade temperature with variation of porosity. For the shade, more energy is absorbed with decreasing porosity but there is a simultaneous increase in energy convected off of the shade due to the increased surface area of the shade material. The glass, on the other hand, receives more reflected radiation from the shade at low porosity causing a very slight increase in glass temperature. However, the glass temperature is most strongly influenced by its high absorptance in the solar spectrum.

Figure 6 shows the total heat gain to the room and the radiant heat gain versus shade porosity. The relationship is very nearly a straight line ranging from about 415 to 520 W/m^2 (130 Btu/hr-ft^2 to 166 Btu/hr-ft^2). At the lower end, heat transfer to the room is dominated by convection from the glass and shade and by long wave radiation transfer. At the upper end, direct solar gain plays a more important role. Long wave radiation from the glass to the room also increases as the glass "sees" more of the room and less of the shade. The glass absorbs a large fraction of the incoming radiation and hence is hot, causing long wave radiation to go into the room and promoting convective transfer from the glass to the room.

Some observations are in order. First it is important to say that the heat absorbing glass used in our model represents a rather extreme case – it really is a solar collector. That said, we note from figure 4 that 45% of the heat gain to the room is convected from the glass and shade. Further more, we used table 22 of chapter 31 of the 2005 ASHRAE Handbook of Fundamentals to find the solar heat gain coefficient and the interior attenuation coefficient and to calculate the fraction of incident radiation that would become heat gain. For a shade transmissivity of 0.2 the result was 0.29. The model described above predicts 0.43.

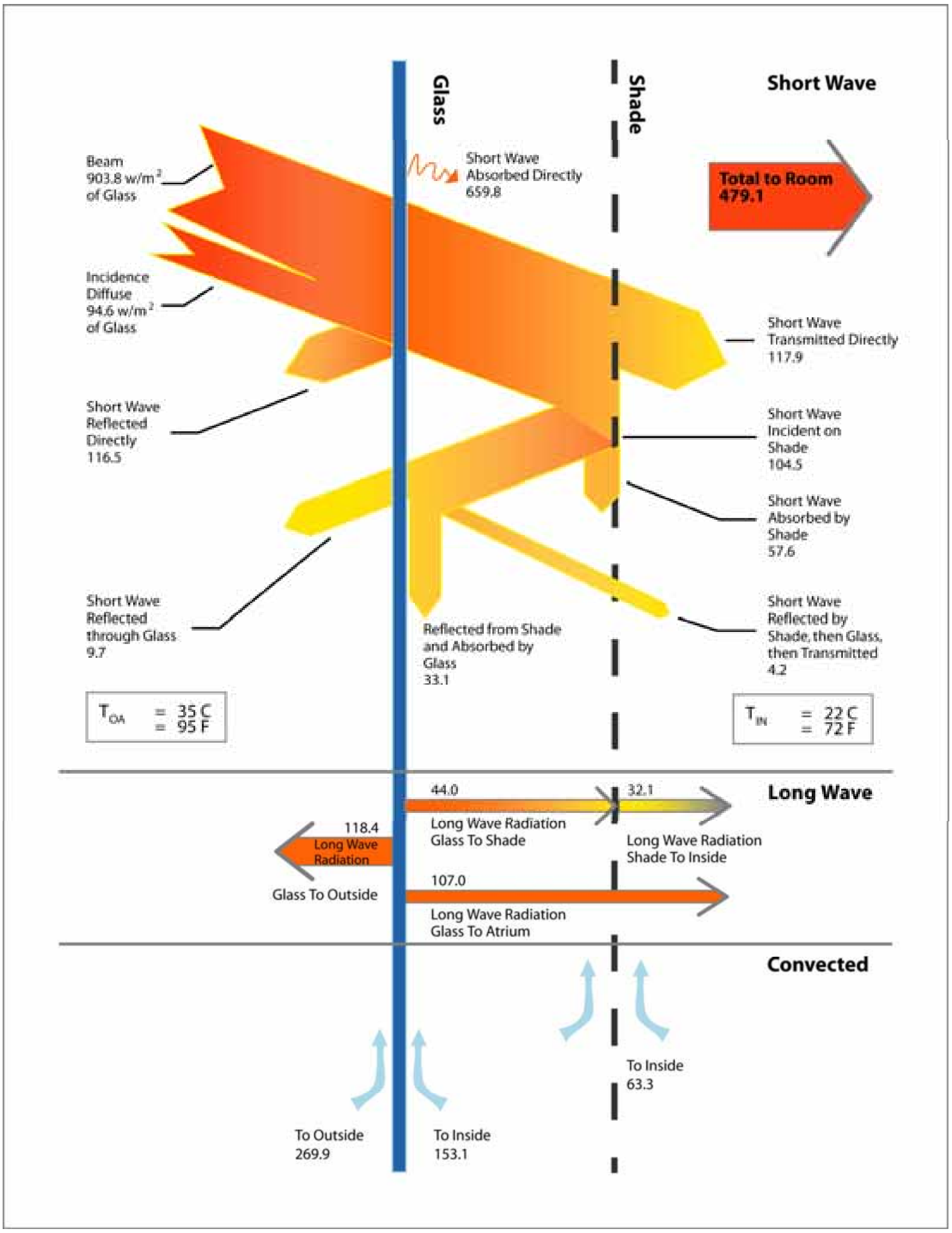

Figure 4. Energy flows occurring in the glass/shade system.

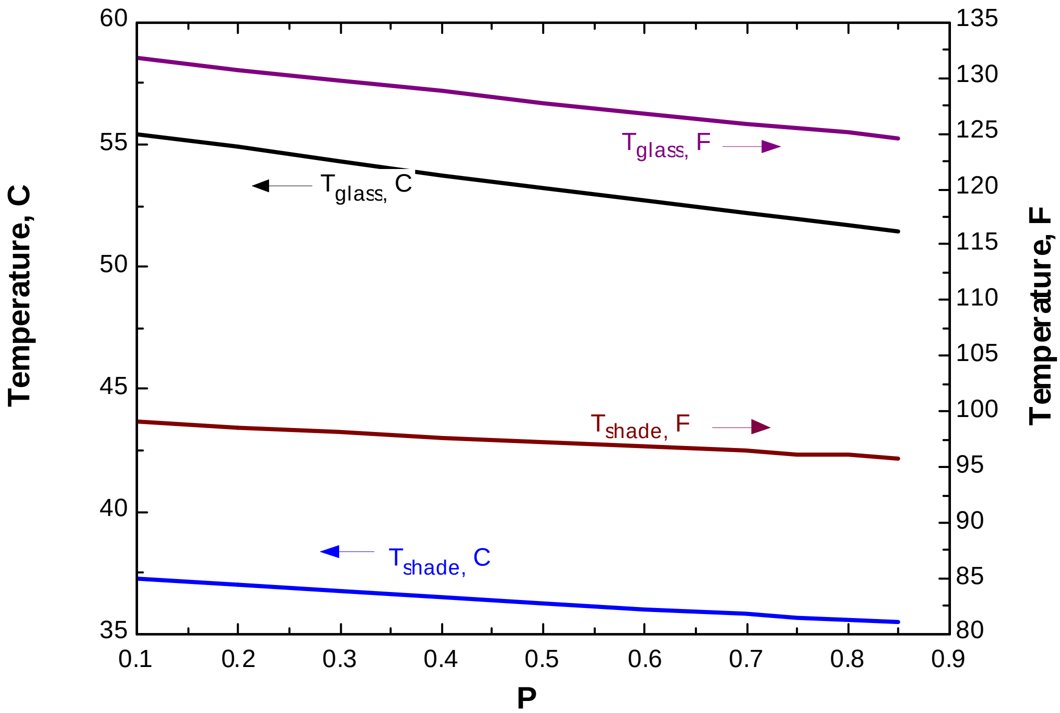

Figure 5. Glass and shade temperature versus shade porosity (transmittance).

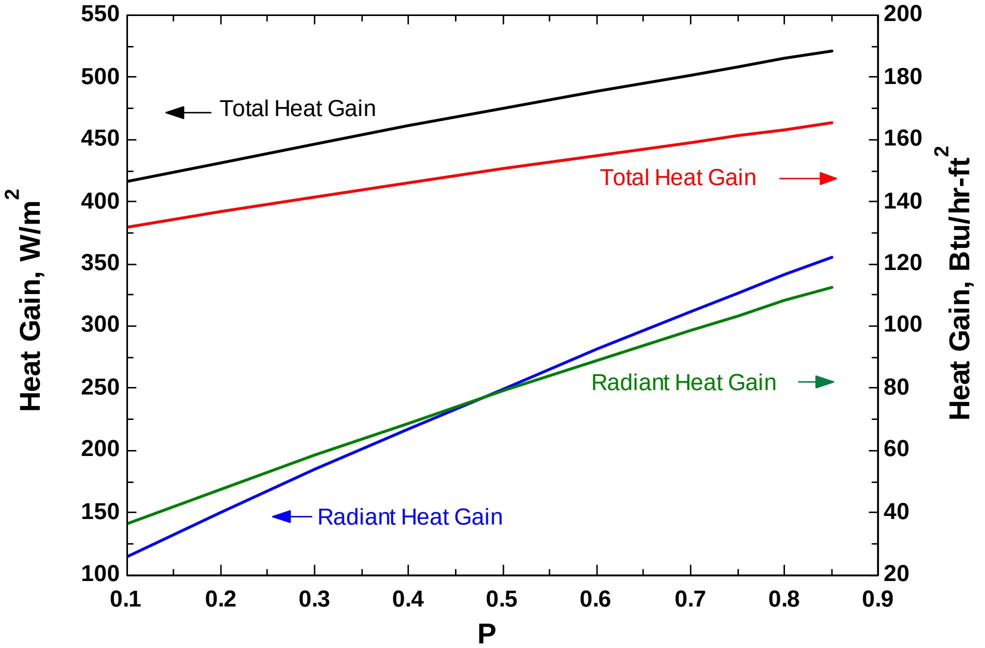

Figure 6. Total heat gain through façade versus shade porosity (transmittance).

Convection coefficients

We next explore the impact of convection coefficients on total solar heat gain. As a reminder, the properties of the glass and shade are:

	Glass	Shade
Transmissivity	0.23	0.53
Reflectivity	0.11	0.22
Absorptivity	0.66	0.25

We deal first with the outside convection coefficient. Yazdanian and Klems [1994] measured the convection coefficient on their test facility for a first floor window and determined that it varied from 2-20 W/m^2-K over 0-10 m/s (0-45 miles per hour). Keeping 5 W/m^2-K for the inside convection coefficient, the variation in heat gain due to variation in outside convection coefficient can be calculated and is shown in figure 7.

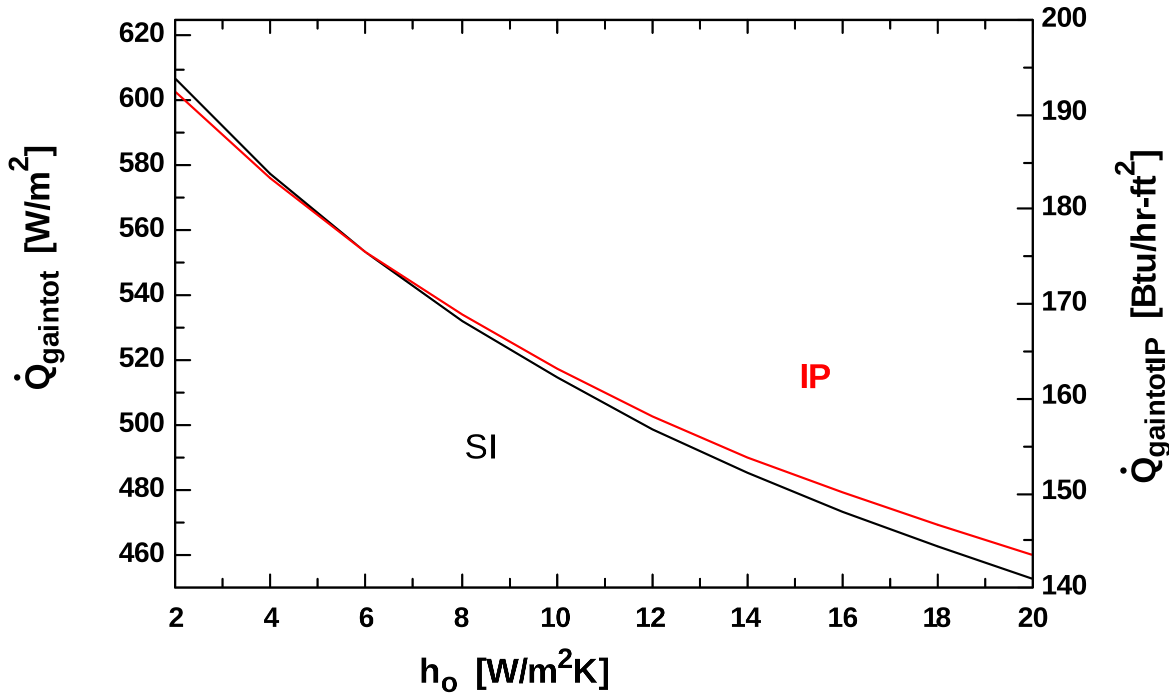

Figure 6. Heat gain versus outside convection coefficient.

How does figure 7 illuminate us. Firstly, we are reminded that the model we have developed is for *total* heat gain through a glass/shade system. In the dark, at night, there would be heat gain from a hot outdoors to a cool room and a low convection coefficient would reduce the heat gain. However, when the glass is hotter than the surroundings, wind (and a high outside convection coefficient) helps reduce the heat gain to the room even when it is hot outdoors and cool indoors.

In this example with heat absorbing glass, the worst case scenario is not when the exterior film coefficient is high (i. e. a high speed wind) but when the air outside the glass is still. Methods that try to separate conduction/convective heat gain due to temperature difference through a window from solar heat gain seemed doomed or at least impractical. Ditto for schemes that try to combine convection and radiation at the inside or outside surface.

Figure 7 shows that heat gain to a room caused by the sun can vary by 50% depending on how much energy absorbed by the glass is swept to the outdoors. How about convection off the inside of the glass and shade? Fisher and Pederson [1995] and Spitler in referenced work show that the inside convection coefficient off interior walls ranges between 2 and 5 W/m^2-K. If, for the sake of consistency, we assume that the outside convection coefficient is 15 W/m^2-K, what happens when we vary the inside coefficient? Figure 8 below shows the result.We varied the inside coefficient from 2 to 10 W/m^2-K because the glass and shade could be hotter than other room surfaces, promoting convection. The variation is 45%.

We now compare heat absorbing glass with 1/8 inch clear glass in the figures below. Figure 9 shows the total heat gain to the room for each glass/shade system. The clear glass is more transmissive and the glass/shade system admits more heat to the room. It is not quite as simple as it may seem however as figure 10 reveals the temperature of the glass and shade for the two glass/shade systems. For the heat absorbing glass/shade system, the glass is hotter than the shade. For the clear glass/shade system the opposite is true, the shade is hotter than the glass. From the point of view of convective heat gain, most comes from the glass in the first case most comes from the shade in the second case.

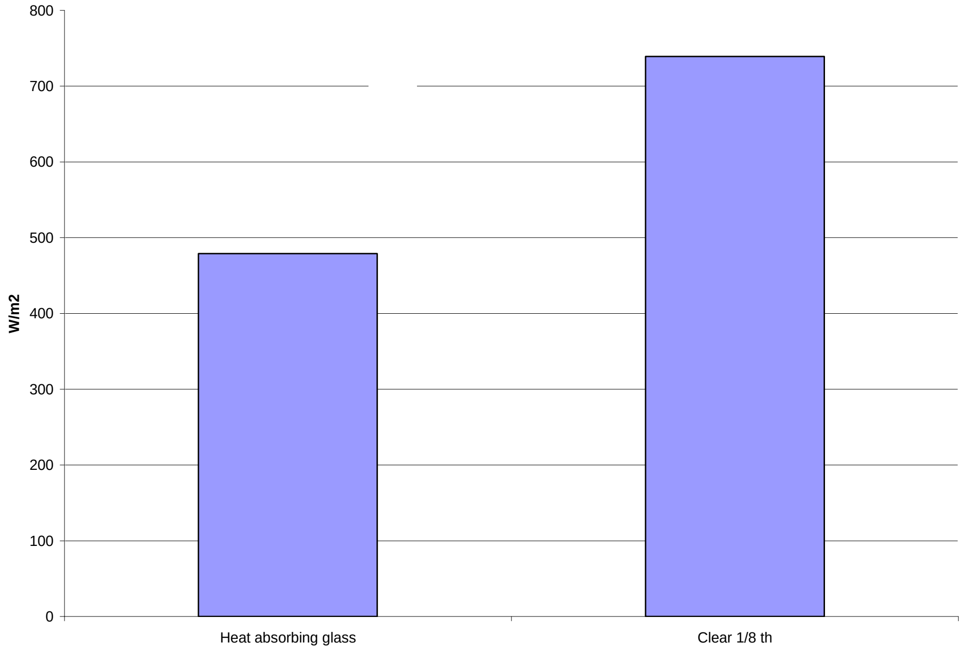

Figure 7. Window heat gain for heat absorbing and clear glass

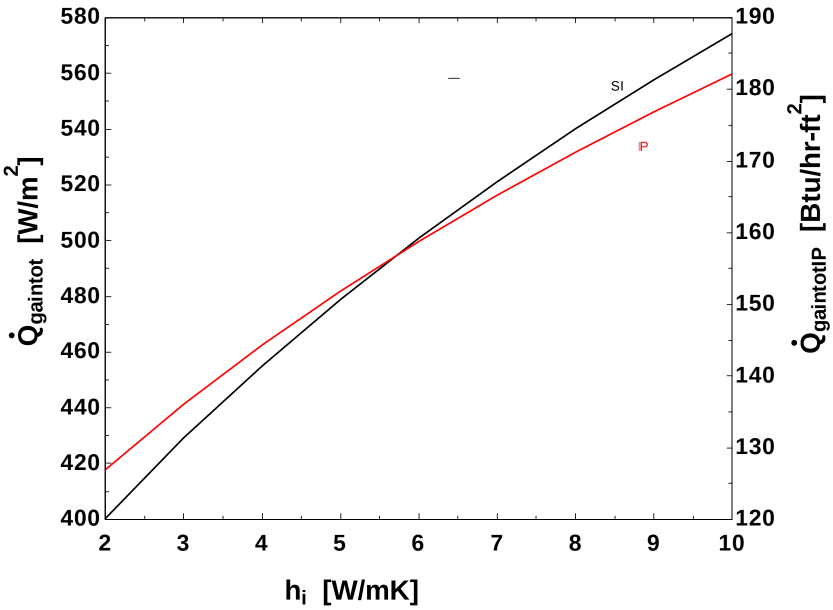

Figure 8. Window heat gain versus inside convection coefficient.

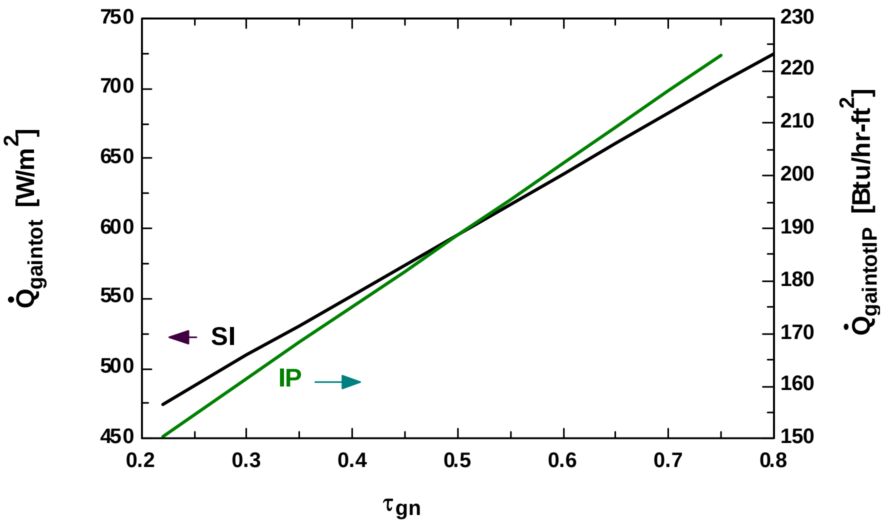

Figure 11. Solar heat gain versus glass transmissivity, reflectivity of 0.11, shade transmissivity of 0.53.

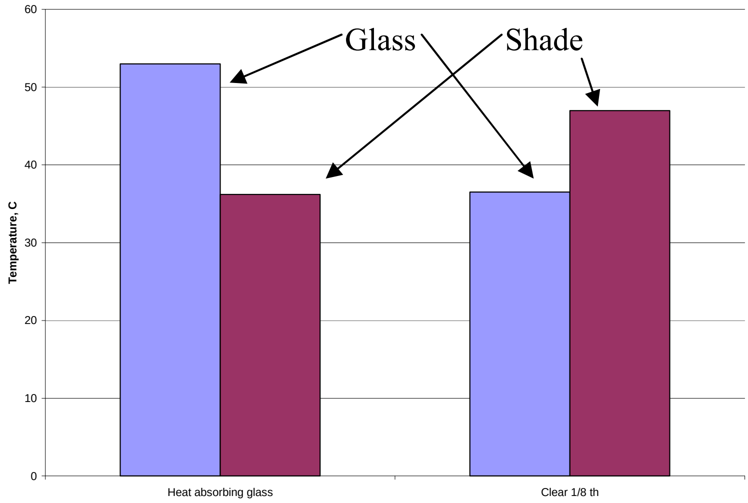

Figure 8. Temperature of glass/shade systems.

Finally, figure 11 shows the impact of the variation of transmissivity of the glass from 0.23 to 0.80, keeping the reflectivity of the glass at 0.11 and the shade tranmissivity at 0.53. The relationship is linear.

CONCLUSIONS

Modeling results lead to the following conclusions:

For the baseline glass/shade system studied here, the temperature of the glass and shade vary with shade transmisivity, the glass varying from 51.4 to 55.5 C (124.6 to 131.8 F) and the shade changing from 34.4 to 37.6 C (95.8-99 F).

Again for the baseline glass/shade system, the heat gain through the south façade varies almost linearly with shade transmisivity, ranging from about 416 W/m^2 (132 Btu/hr-ft^2) to about 521 W/m^2 (165 Btu/hr-ft^2) as the transmissivity varies from 0.1 to 0.85.

The transmissivity of glass and a shading device do not, by them selves, define the solar heat gain into the room. The reflectance of the glass is equally important. Long wave radiant exchange and convection from the glass and shade to the room are significant components of heat gain.

For the hot winter condition studied, the *solar heat gain* is predicted to be 480 W/m^2 (152 $Btu/hr\text{-}ft^2$). The contribution of this heat gain to the *cooling load* should be calculated using the load calculation methods described in the ASHRAE Handbook of Fundamentals, 2001.

FURTHER WORK

ASHRAE is currently sponsoring work to develop more complete models and verify them experimentally.

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The Influence of Blinds on Temperatures and Air Flows within Ventilated Double-Skin Façades

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ABSTRACT

Ventilated façades have become an increasingly employed feature in the design of low energy buildings over recent years in that they offer the attractive features of a conventional glass façade but without the thermal disadvantages. These façades consist of a double skin surface, the outer layer of which is of toughened glass, and the inner layer of which usually comprises conventional double-glazing, behind which is the occupied space.

The cavity formed by the outer and inner layers is ventilated, and frequently contains a blind. This blind, together with the cavity ventilation, provides a means to control the heat transfer across the façade, in terms of solar gain transmission and recovery of heat lost from the interior.

A three-year project, funded by the UK's Engineering and Physical Sciences Research Council (EPSRC), has investigated the thermal and airflow performance of ventilated façades. A series of parametric experiments have been performed using the Large Scale Solar Simulator at Loughborough University. Results from these experiments have been used to validate models of airflow and thermal behaviour developed at De Montfort University. Advice on practical application and industrial practice has been provided by Arup Research and Development and IT Power. The result of the research is an improved understanding of the thermal and air flow behaviour of such ventilated double skin façades.

The effects of external conditions, solar irradiation and exterior air temperature, on double skin façades with differing internal characteristics are presented and analysed in this paper. In particular, the effect of the blind blade angle on cavity temperatures and ventilation air flows will be reported, together with an outline of the guidance that is now emerging to assist designers of such façades.

INTRODUCTION

Detailed measurements of the thermal performance of building elements are essential in the validation of models used for performance prediction. A collaborative project funded by the UK's Engineering & Physical Sciences Research Council (EPSRC) has recently been completed. The focus of this project was a detailed investigation of the performance of double skin façades, which are an increasing feature of many commercial and public buildings. Such façades typically consist of an outer layer of toughened glass, a double glazed inner layer and a ventilated cavity that frequently includes a solar control device, typically a venetian blind [1]. An objective of this research was to measure and provide a comprehensive data set detailing air and surface temperatures, together with air flow rates, so as to provide a practical resource for further research and for use by building designers.

Facilities for providing detailed measurements of such building façade elements under controlled conditions are hard to find. Most testing to date has been conducted outdoors where there is little or no control of the key environmental factors [2, 3]. Further development and characterisation of double skin façades requires good control of the test conditions, not only of the solar radiation striking the façade surface, but also of the thermal environments that affect the façade surfaces both externally ('outdoors') and internally ('indoors'. Such a facility is available at Loughborough, and has been used to provide the experimental results reported here. Key characteristics of the specially designed Large Scale Solar Simulator, are also presented.

Prior to carrying out the detailed tests, wide ranging consultation with the designers and manufacturers of building façades was undertaken. As a result, the more conventional façade element comprising a single storey 'box-window' was selected for investigation [1]. In this respect, a full sized double skin façade incorporating a sun-shading blind was constructed and installed in the solar simulator for investigation of its performance . As the majority of practical installations of the double skin façade utilise natural ventilation, the work has concentrated on this more challenging situation of buoyancy-driven flow in the façade cavity.

This paper describes details of the experimental conditions, together with the characteristics of the test facility and the tested façade element. Test results showing façade temperatures and airflow profiles and how they depend on the climatic conditions and the blind blade angles are presented.

THE LARGE SCALE SOLAR SIMULATOR AND THE DOUBLE SKIN FAÇADE

Large Scale Solar Simulator

Numerous details of the simulator have been reported in Mei et al[4], but are given again here for convenience. The large scale solar simulator installed at Loughborough University essentially provides a source of artificial solar radiation of acceptable spectrum and uniformity for irradiating a test element surface. In addition, the simulator provides for two separately conditioned thermal environments on either side of the test element. In this way, the thermal behaviour of building-integrated test elements, as if bounded by 'outdoor' and 'indoor' environments, can be effectively addressed. The solar radiation is provided by fifteen Sol 1200 lamps manufactured by Honle UV(UK) Ltd. The lamps are capable of producing a natural sunlight spectrum to D65 standard, at an air mass ratio of 1.5, and can deliver an irradiance upon a test surface of up to 1000W/m² with a uniformity to within ±10%. A combination of mesh attachments can be used to reduce the 1000W/m² peak irradiance value in steps of 200W/m² down to a minimum of approximately 200W/m², without spectral alteration.

The solar simulator is designed to test elements up to 2.5m in length and 1.5m in width. It is also possible to ventilate the test element both on its front face (to simulate wind) or through an internal cavity within the element, as required. This can consist of natural or mechanical convection, depending upon the nature of the element being tested.

The environment that adjoins the front irradiated surface of the test element corresponds to the 'outdoor' ambient condition. Air conditioning of the test laboratory space was used to maintain the required outdoor ambient environment within the temperature range 12°C to 30°C. The environment that adjoins the rear of the tested element corresponds to the 'indoor' ambient condition, such as a room or office. A rectangular shaped enclosure attached to the rear of the test element was used to produce the rear environmental conditions. The air and mean radiant temperatures within the rear enclosure are mostly maintained equal in the tests reported here but can be independently controlled if required. Air can also flow through the rear enclosure if required so as to generate known heat transfer conditions.

The solar simulator facility has been designed specifically for testing multifunctional façade elements but can also be used for other, more conventional, components such as solar thermal collectors, curtain walls, roof lights and for PV applications. Fig 1 shows the arrangements of the lamps in the solar simulator.

Fig. 1 Large Scale Solar Simulator, showing lamp array.

THE TEST FAÇADE AND ITS COMPONENTS

The test façade element was custom-built using materials and dimensions that closely resembled a typical single-storey ('box-window') commercial façade design. To facilitate experimental testing and access, both the inner and outer skins of the façade were constructed as door assemblies. The outer skin of the designed double skin façade is a single glass door which is 144cm wide and 206cm high comprising an aluminium frame and 12mm thick toughened clear glass. The glass area is 128cm × 191cm. The outer skin of the façade was designed to be opened and closed for easily installing the measurement devices and for changing the blind positions. Air leakage was minimised by sealing the façade doors with rubber strip seals. Both the air intake and exhaust of the double skin façade are designed as a commercial grille arrangement to permit air flow through the façade cavity. The grilles are of height 24cm and width 145cm. Each grille has three spaces for air ingress and egress, each space being 4.5cm high. In commercial façades, wire meshes are installed horizontally at the bottom and top of the façade cavity to prevent birds getting into the façade. These are included in the customised test section for authenticity of design. The size of the mesh 'hole' is 2.5cm square.

The façade inner skin consists of another door, fitted with double glazing housed within an aluminium frame. The glazing cross section comprises two panes of low-e toughened clear glass each 6mm thick, separated by an air cavity of width 16mm. The dimensions of the inner skin are 138cm × 200cm and the glass area is 122cm × 185cm. Fig 2 shows photographs of the constructed double skin façade where the outer skin can be opened for installing measurement devices.


a)


b)

Fig 2 The Tested Double Skin façade: a) outer door closed; b) outer door open

SUN-SHADING BLINDS

The sun-shading devices commonly used in practice and installed for these experiments are Venetian type blinds, These were purchased from the Krülland Company in Germany. This type of blinds can be motor driven. The blinds tested are made of aluminium and the blade width is 8cm (total blind dimensions 2.1 m high by 1.45 m wide). The colours of blinds selected for testing were white (reflectance = 0.762), and dark brown (reflectance = 0.079). In Fig 3. both colours of blinds can be seen in the cavity of the double skin façade.

a) b)

Fig. 3 The blinds installed in the façade cavity:
(a) dark brown ('spartbrown'); (b) white

MEASUREMENTS

TSI air velocity transducers, based on anemometry, were selected to measure air velocities within the ventilated cavity. The TSI Omni directional Model 8475 offers accurate measurements at low velocities from 0.05 to 0.5m/s and is suitable for unknown or varying flow direction. The accuracy of Model 8475 is ±3.0% of reading over the temperature range 20°C to 26°C, outside this range, and within temperature compensation range, an additional reading error of 0.5% per °C must be added to the measured values.
Cable to the connector

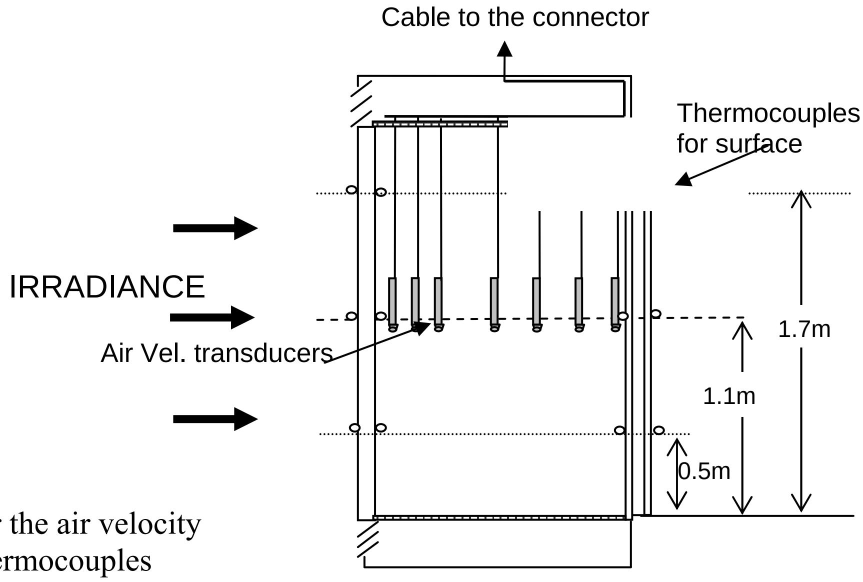


Fig. 4 Positions for the air velocity transducers and thermocouples (side cross-sectional view of cavity)

Seven such air velocity transducers were used for the measurements, and to each velocity transducer a calibrated type-T thermocouple was attached to provide co-located airflow and air temperature data. The transducers were suspended vertically in the cavity with the probes facing the upward air flow. For most of the experiments, the transducers were located at the middle height of the façade cavity (1.1m from the bottom of the façade). The distribution of air velocity transducers and thermocouples can be seen in Fig. 4.

In total, twenty two T-type thermocouples were used to measure the façade surface temperatures and the cavity air temperatures. For measuring the surface temperatures, twelve thermocouples were attached to the glass and covered with thermocouple pads to shade the thermocouples from the direct solar radiation. Pad emissivities matched that of the surrounding surface. For measuring the cavity air temperature, seven thermocouples, were attached to the air velocity transducers; the thermocouples were located about one centimetre distance downstream of the velocity transducer probe head.

A Kipp & Zonen CM3 pyranometer (0-2000W/m² = 0-0.5V) connected to the data logging system was used to measure the radiation on the façade outer skin surface. Table 1 gives the measured irradiance values corresponding to the nominal irradiance values which can be selected by choice of mesh in front of the lamps.

Table 1 Measured irradiance corresponding to nominal irradiance.

Nominal	800W/m²	600W/m²	400W/m²	200W/m²
Measured	715W/m²	540W/m²	360W/m²	187W/m²

In the experiments, the nominal irradiances were varied from 200W/m² to 800W/m² by changing the mesh attachments. However, in all results, the measured irradiance values were used and quoted.

The data acquisition and logging (DAQ) system for the Large Scale Solar Simulator is PC based and employs National Instruments (NI) Labview software, with two internally fitted NI data acquisition boards.

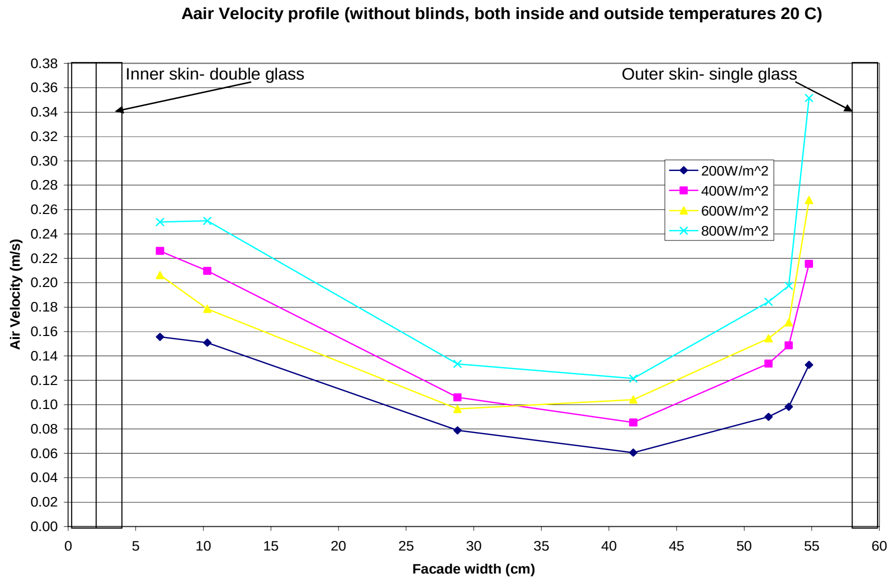
EXPERIMENTAL RESULTS

Test without blinds

Testing of the double skin façade without blinds was performed over a range of controlled climate conditions.

Tests have been carried out under the following measured irradiance values: 187W/m², 360W/m², 540W/m² and 715W/m² For each irradiance, three values for the test room air temperatures (‘outdoor’ air temperature or T_{room}) were set: 12°C, 20°C and 30°C, respectively. A total of twelve environmental conditions were tested for the double skin façade. The plenum air temperature (‘indoor’ air temperature, or T_{box}) was set to 20°C for all the experiments. The cavity width was fixed at 550mm for all experiments. Typical results are shown in Fig. 5 for profiles of the air velocity and façade temperatures (surfaces and cavity air) at four irradiances and at 20°C for both 'indoor' and 'outdoor' conditions. It is evident that the cavity air velocity and façade temperatures increase with increasing values of the irradiance. It is also clear that the air speed is faster nearer the surfaces of the glazing that bound the cavity.

In order to better see the impacts of varying conditions, air velocity and façade temperature were plotted as a function of the irradiance for each data set as shown in Fig. 6. It appears that the increase of the air velocity and the local façade temperatures are more or less directly proportional to the irradiance. For both air velocity and temperature, each set of data gives slightly different gradients.



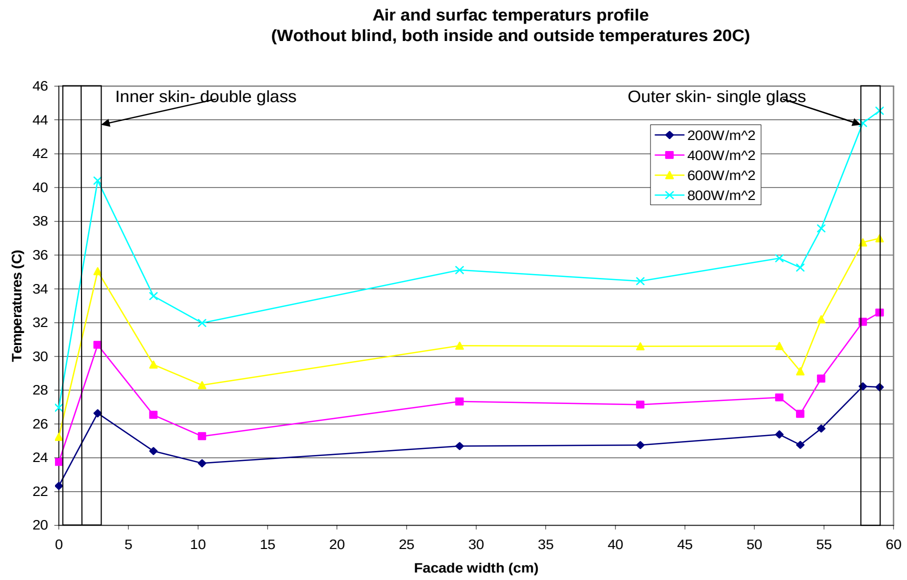


Fig. 5 The profile of the air velocity and the façade temperatures across the façade cavity

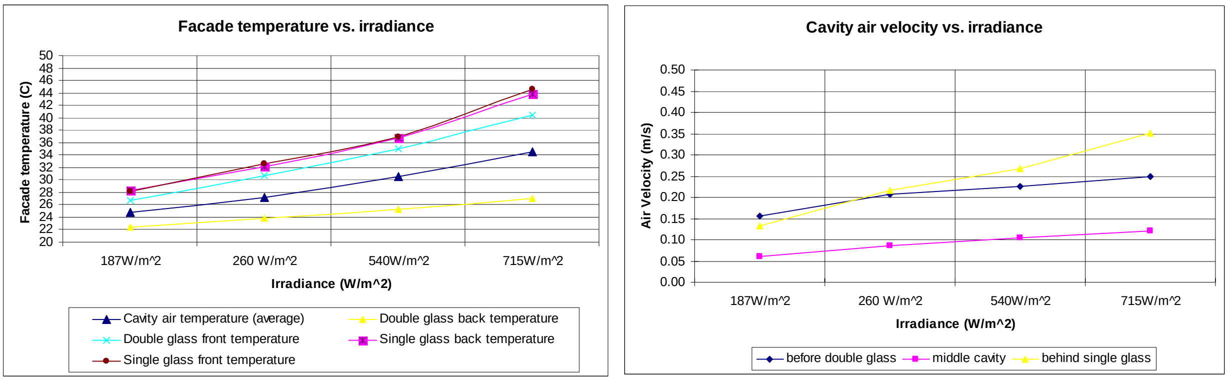


Fig. 6 Façade temperature and cavity air velocity increase vs. solar irradiance; Both 'outdoor' and 'indoor' air temperatures are fixed at 20°C.

In a similar manner, the effect of 'outdoor' temperature on the cavity air velocity and façade temperature is shown in Fig. 7. In this graph, the comparison of the air velocities in the cavity and the temperatures of the façade (surfaces and cavity air) for three different temperatures of the 'outdoor' air at a fixed irradiance of 715W/m² are presented. Façade air velocity and façade temperatures appear to be proportional to the 'outdoor' air temperature.

It is noticed that the double glass back temperature (inner skin back) was dominated by the controlled 'indoor' temperature which was at 20°C. Therefore, for the effects of either irradiance or outdoor air temperature, the double glass back temperature tends to be low with only small gradients evident. From Figs. 6 and 7, it also can be seen that the cavity air velocity was affected strongly by the irradiance rather than the outdoor air temperature.

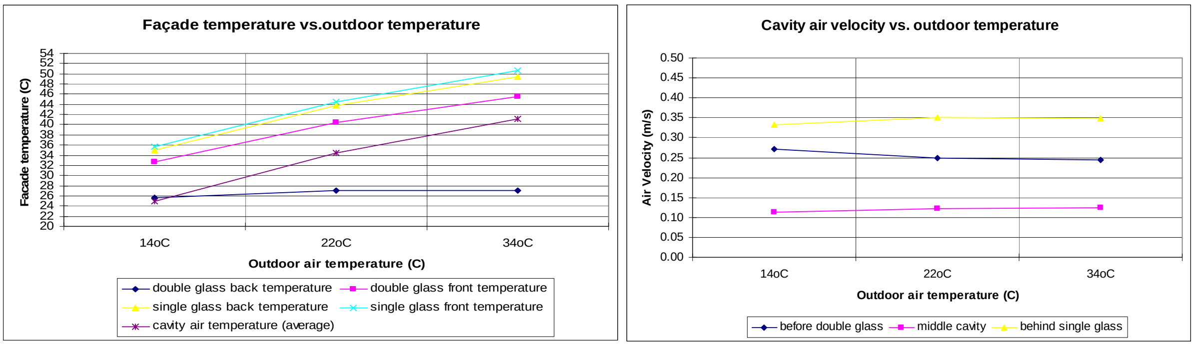
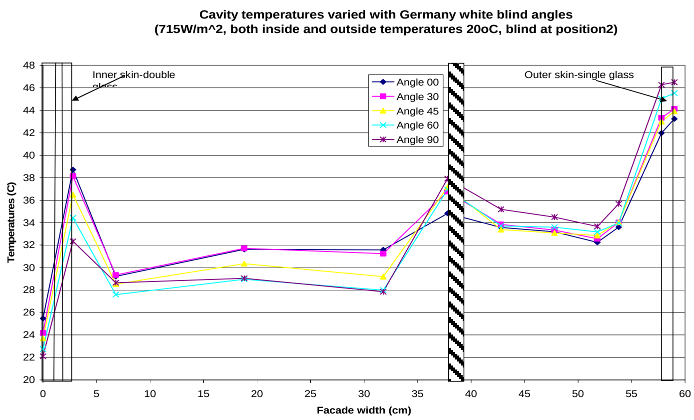


Fig. 7 Façade temperature and cavity air velocity increase vs. outdoor temperature; the irradiance is set at 715W/ m²

TESTS WITH SUN-SHADING DEVICE (BLINDS)

Tests with the blinds within the cavity of the double skin façade were performed to investigate several effects, including the effects of blind blade angle on cavity airflow and thermal behaviour. The angle of the blind blade is an important factor for the heat transfer through the façade cavity, and data on observed behaviour is needed based on parametric testing in controlled conditions. These data are presented in this paper.

The tests for the effects of blind blade angles were carried out at the fixed conditions of 715W/m² irradiance and 20°C temperature for both 'indoor' and 'outdoor' situations. The blind colour is white and the blind was located at one third of the cavity width as measured from the outer skin. The blade angles were set at 0, 30, 45, 60 and 90 degrees, where 0 and 90 degrees relate to the blinds being fully open or fully closed, respectively. In Fig. 8, the measured air velocities of the cavity and the façade temperatures are displayed as profiles across the cavity width for the blind blade angles. In Fig. 9, the cavity air velocities and the façade temperatures are plotted as a function of the blind blade angles. As the blind effectively separates the façade cavity into two vertical chambers, the effect on the cavity air velocity and the façade temperatures will be considered as two parts: in front of the blind and behind the blind. From both Fig. 8 and 9, it can be seen that the temperatures of the façade surface and cavity air in front of the blind increase as blind blade angle increases until the fully closed position is reached (90 degrees), as well as the surface temperature of the blinds themselves. Behind the blind, the temperatures for the surface and the cavity air decrease as blind blade angle increases. It is clear that with the blade at 0 degrees (fully open), more radiation energy transfers to the back chamber and causes the higher temperature in the chamber behind the blind. When the blade is at 90 degrees (fully closed), the blind absorbs more radiation energy and its surface temperature rises. At the same time, the blind reflects more energy to the front chamber and the single glass (outer skin) to increase the temperature of both the air and the surfaces.



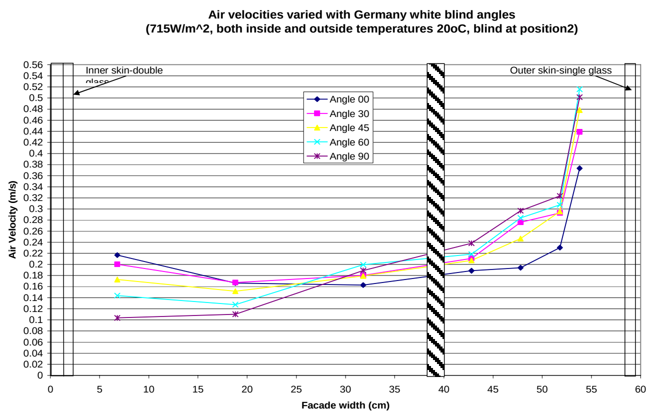


Fig. 8 The profile of air velocity and the façade temperatures crossing the façade cavity

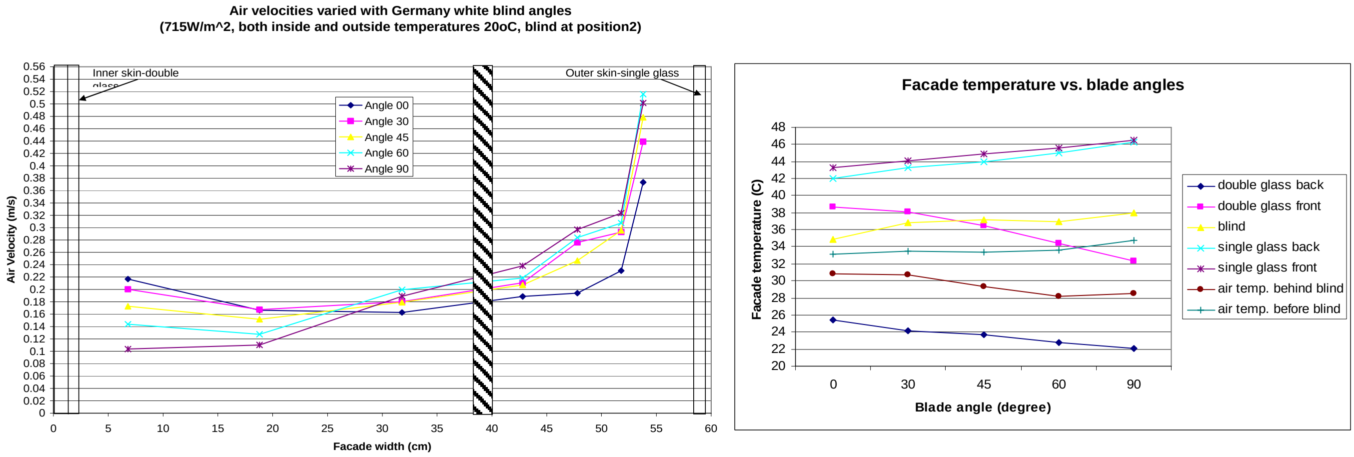


Fig 9. Façade temperatures and cavity air velocity vs. blind blade angles

CONCLUSIONS

An outline description has been presented of a new facility installed at Loughborough University for the testing and development of building ventilated façade elements. A series of parametric experiments have been carried out for testing the performance of these façades under controlled conditions using a large scale solar simulator.

One series of tests involved a section of double skin façade with a common type of solar shading device (venetian blinds) located between the two skins. Test results for the air flow and façade temperatures have been presented. The effects of the irradiance and outdoor air temperature were shown to affect the air velocity and façade temperatures, and thus the façade thermal performance. Furthermore, the effect of the blade angles of the shading device located in the naturally ventilated cavity was presented . The presence of the shading device within the cavity can be considered to separate the cavity into two vertical chambers, in front of and behind the blinds. It is concluded that the blinds have a significant influence on the thermal and airflow performance of the façade. If the sun-shading device is fully closed, the 'front chamber' of the cavity and the shading element itself will have higher temperatures than if it is opened. In contrast, the temperatures behind the sun-shading device will be higher if the sun-shading device is fully opened. Whilst these findings might at first sight appear intuitive and unremarkable, it is important to note that the behaviour has been related parametrically to specific changes in conditions, has revealed trends in performance, and thus constitute a useful data resource, as well as being an important aid to designers. These findings were obtained under well-controlled conditions in an experimental facility that incorporates a representative section of double-skin façade. This facility remains available for further testing across a wide range of parameters (blind colour and position within cavity, cavity width), and these will be reported in due course. It is further concluded that the large scale solar simulator is a valuable facility for testing the solar thermal performance of building components and can provide realistic and meaningful results.

ACKNOWLEDGMENTS

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Full-scale experimental investigation of room wall containing phase change materials wallboard

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SUMMARY

Because we spend most of our time in enclosed spaces, demand in thermal comfort of buildings rose increasingly and then energy consumption correspondingly is increased, aggravating the pollution of natural environment. Integrating phase change materials (PCM) into building walls is a potential method of reducing energy consumption in passively designed buildings. A wallboard new PCM material is experimentally investigated in this paper to enhance the thermal behaviour of light weight building internal partition wall. The experiments are carried out in a full-scale test room MINIBAT (3.10m x 3.10m x 2.50m) which is completely thermally controlled. The external temperature and radiative flux from spotlights dynamically simulate a summer and mid-season repetitive days. The tests concern walls with and without PCM material under the same external conditions. The PCM allows reducing the room air temperature fluctuations, in particular when overheating occurs.

INTRODUCTION

Because we spend most of our time in enclosed spaces, demand in thermal comfort of buildings rose increasingly and then energy consumption correspondingly is increased, aggravating the pollution of natural environment. Integrating phase change materials (PCM) into building walls is a potential method of reducing energy consumption in passively designed buildings. This tendency is confirmed by the numerous papers which a review can be found in [1]. However, there is a strong need for experimental data to evaluate the capacity of PCM to stabilize the internal environment (and thus minimise energy consumption) when there are external temperature changes and solar radiations in case of light weight buildings: that is the purpose of our article.

The use of PCM materials in building envelop is subject to considerable interest in the last decade. Their main interest is that they can store latent heat energy, as well as sensible energy. As the temperature increases, the material change phase from solid to liquid. The chemical reaction being endothermic, the PCM absorbs heat. Similarly, when the temperature decreases, the material change phase from liquid to solid. The reaction being exothermic, the PCM desorbs heat.

The main disadvantage of light weight buildings is their low thermal mass. Obviously, they tend to large temperature fluctuations due to external cooling or heating loads. Using PCM material in such buildings can decrease the temperature fluctuations, particularly in case of solar radiations loads. This have been proved in several numerical studies [2], [3], [4], [5].

A new product has been achieved by the DuPont de Nemours Society: “Energain ™”it is constituted of 60% of PCM, of which the temperature of fusion has been chosen to 22°C. The figure 1 shows the experimental heat capacity measured by differential scanning calorimetry

system. The figure 2 presents the experimental thermal conductivity. This product is like a polymeric membrane, relatively flexible, of 5 mm thickness.
The use of PCM in the building is a relatively old concept that could be ever exploited really because of the inherent difficulties of setting such materials. The novelty in this case is constituted by encapsulation of an important quantity of active matter in a thermoplastic polymer that, after transformation in a relatively thin membrane, permits a practical installation in all type of envelope of the building.
In order to show the efficiency of this PCM made of a 5 mm thickness, we have made a comparative measurement in our experimental cell MINIBAT. Very few experimental data under controlled external conditions exist. These one make it possible to be used as reference for numerical simulations, mainly in the case of solar radiations.

The first part of our article presents the experimental test cell. The experimental set-up simulates a complete day with dynamical effects, and the climatic conditions can be reproduced. The second part presents the results of a summer and a mid-season day and a comparison is made between the case with PCM and the case without PCM. The results show a good improvement in thermal comfort for the case with PCM wallboards.

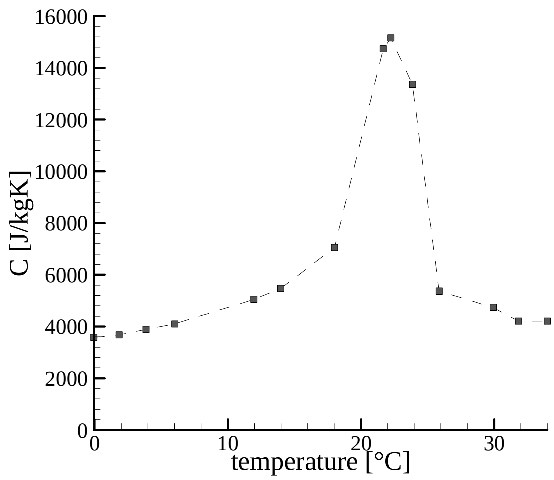


Figure 1: Experimental PCM material heat capacity measured for several temperatures

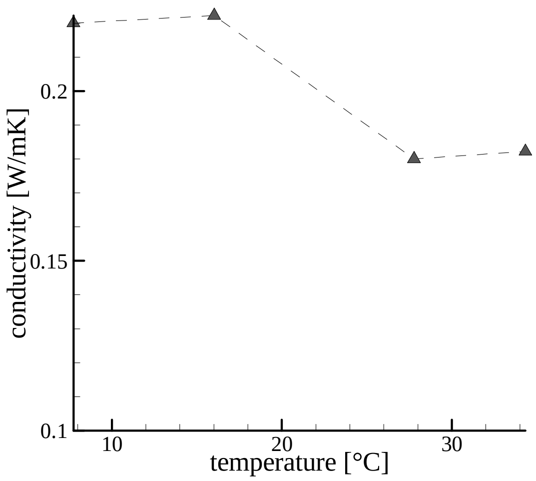
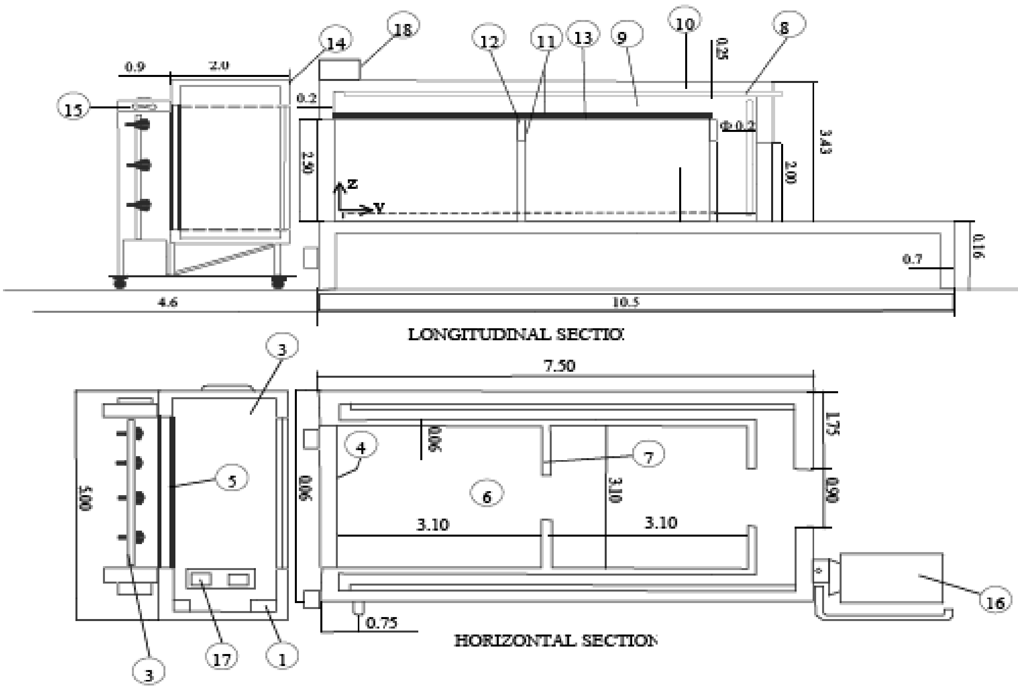


Figure 2: Experimental PCM material thermal conductivity measured for several temperatures

PRESENTATION OF TEST CELL MINIBAT

The experimental cell MINIBAT, presented figure 3, is composed of two identical parts which dimensions are 3.10mx3.10x2.50m respectively to (x, y, z). We will use in our study only the cell 1 which will be called experimental test cell. In our studies, cell 2 is not distinct from the thermal guard because the communication door between these two entities is open. The composition of the walls is given in table I. Three of the walls (which location are given figure 4) have been covered with PCM material for the tests. The glazed façade separates the cell from a climatic chamber whose temperature is controlled and can vary between -10°C and 40°C. A thermal guard allows keeping the five others faces at a constant temperature, which can vary between 5°C and 30°C. During our tests, the thermal guard temperature was kept constant at 19°C.

A battery of 12 spotlights, of 1000W each one, makes it possible to simulate an artificial sunning (gas-discharge lamps with metal halides which spectrum is similar to the sun one). The spotlights are placed on 3 horizontal lines (see figure 4), each line being tilted of an angle α: for line A α=0°, for the line B α=25° and for the line C α=50°. The radiative flux thus created penetrates in the cell via the glazed wall. The control makes it possible to dynamically control the temperature of the climatic chamber as well as the entering level of radiative flux, by the means of the number of lit spotlights.



(1) cold conditioner; (2) spotlights; (3) climatic chamber;
(4) simple glass; (5) double glass; (6) experimental room; (7) door;
(8) air pipe; (9) thermal guard; (10) insulated concrete;
(11) plaster plate ; (12) wood plate; (13) wood plate;
(14) insulated chamber; (15) light ventilator;
(16) thermal guard air treatment system;
(17) climatic chamber air treatment system; (18) test room air treatment system

Figure 3: Experimental test cell MINIBAT

All the faces temperatures are measured using thermocouples of resolution ±0.4°C, each face with 9 thermocouples. The temperatures of the climatic chamber and the various parts of the thermal guard are measured using Pt100 probes of which the resolution is of ±0.3°C. The globe temperature is measured using a thermocouple inside a black ball centred in the experimental cell. The temperature of the air is measured using shielded Pt100 probes: the first one is positioned at the middle of the room and at a height of 85cm (the soil being at height 0cm); the second one is at a height of 170cm. The various levels of radiative fluxes are measured using a pyranometer.

Table I: Composition of the existing walls

Wall	Material	Thickness (mm)
Floor	concrete	200
Wall	plaster plate wood plate	10 50
Ceiling	plaster plate insulated material wood	10 55 25
Glazed facade	glass	10

The acquisition of the various parameters is done by the means of a multiplexer-multimeter connected to a PC. The control of the whole of the apparatuses, except control, is made using software LABVIEW. The time step chosen between two series of measurement is 10mn and the duration of each test is three days.
The tests presented in this article correspond to a summer test, for which the temperature of the climatic chamber varies between 15°C and 30°C and a mid-season test, with climatic chamber temperature varying between 10°C and 20°C. The climatic chamber temperature (T_{cl}) and the radiative fluxes (E) are shown for the two test cases on figures 5. We can notice the good repeatability of the controls of the climatic chamber temperature and lighting of the projectors.

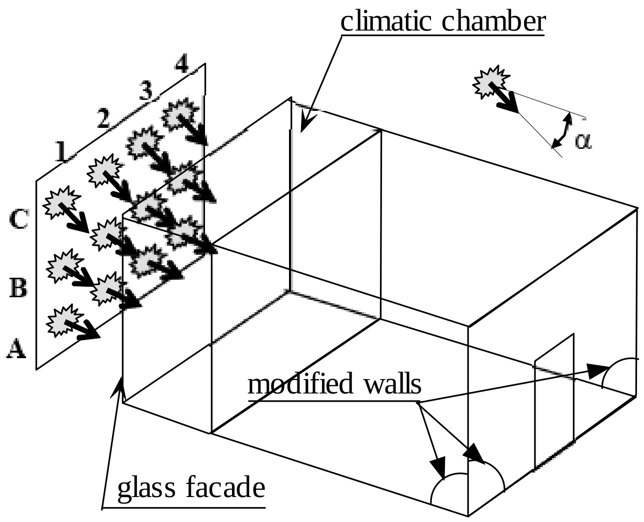


Figure 4: Isometric diagram of the climatic chamber with the projectors and location of the 3 modified walls

In conclusion, the experimental methodology permits us to obtain a complete boundary conditions description and the temperature inside the test cell, all the values being dynamically measured.

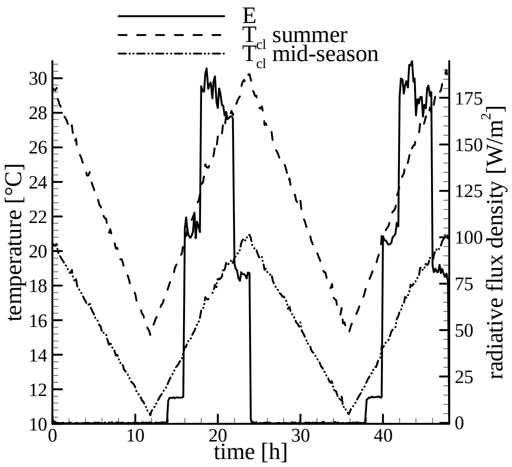


Figure 5: Experimental conditions for the summer and mid-season cases

RESULTS

Summer case

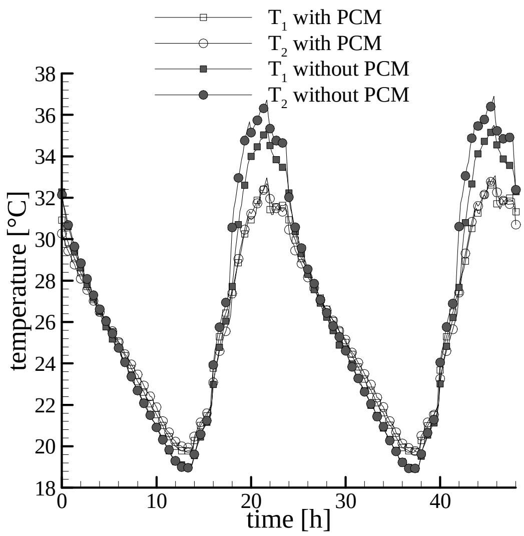


Figure 6: Room air temperatures T1 and T2 measured respectively at a height of 85 cm and 170 cm – Summer case

The results concern the temperatures mesured during the two experiments with and without PCM. The figure 6 shows the evolution of the air temperatures for the two different height positions (85cm and 170cm). The differences between the two cases tested concern mainly the temperature evolution maximum and minimum. With PCM, the maximum temperature maximum is about 32°C when without PCM this value is about 36°C. Concerning the minimum temperature minimum, the differences between the two cases is about 1°C, the PCM case being higher than without PCM (20°C with PCM and 19°C without PCM). The gain concerning the air temperature is clearly shown for the summer case. It is interesting to note that a thermal stratification exists in the case without PCM (a difference of 1°C for the temperature maximum between the two probes) which doesn't happen for the PCM case. This is mainly due to higher natural convection effects because of the lower vertical walls temperatures. This improves the thermal comfort and has never been observed before.

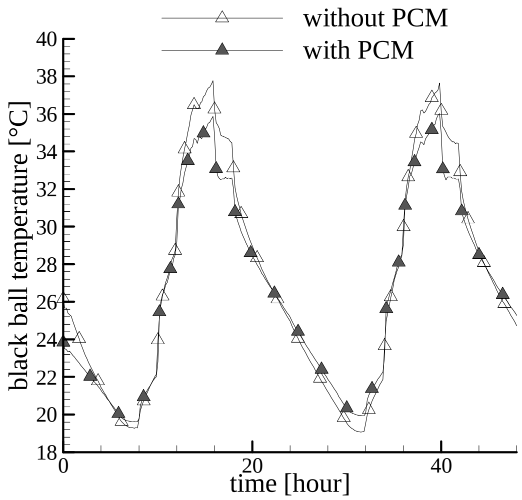


Figure 7: Globe temperatures for the cases tested – Summer case

The figure 7 shows the evolution of the globe temperature measured with a thermocouple in a black ball. The main difference concerns the maximum temperature maximum which is lower of about 2°C for the PCM case. This measure shows that the PCM material enhanced the thermal comfort for which the globe temperature is an indicator.

Concerning the time delay between the cases with and without PCM material, the figures 6 and 7 shows a phase difference (measured at the temperatures profiles maimum and minimum) of about 1 hour. The PCM material can stock-destock energy (the effect being the decreasing of temperatures fluctuations) but canot prevent the radiative effects more than 1 hour.

Mid-season case

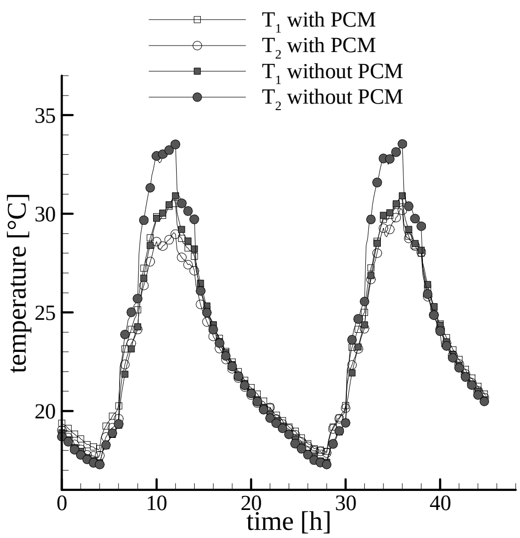


Figure 8: Room air temperatures T1 and T2 measured respectively at a height of 85 cm and 170 cm – Mid-season case

The figure 8 shows the evolution of the air temperatures for the two different height positions (85cm and 170cm. With PCM, the temperature maximum is about 28°C when without PCM this value is between 31°C and 34°C. The gain concerning the air temperature is obvious for the mid-season case. It is interesting to note that the thermal stratification is more important than for the summer case and once more, this stratification doesn't exist for the room with PCM wallboards.

The figure 9 shows the evolution of the globe temperature measured with a thermocouple in a black ball. The main difference concerns the maximum temperature maximum which is lower of about 2°C for the PCM case

Concerning the time delay between the cases with and without PCM material, the figures 8 and 9 show a phase difference (measured at the temperatures profiles maximum and minimum) of about 1 hour. The PCM material can stock-destock energy even during mid-season.

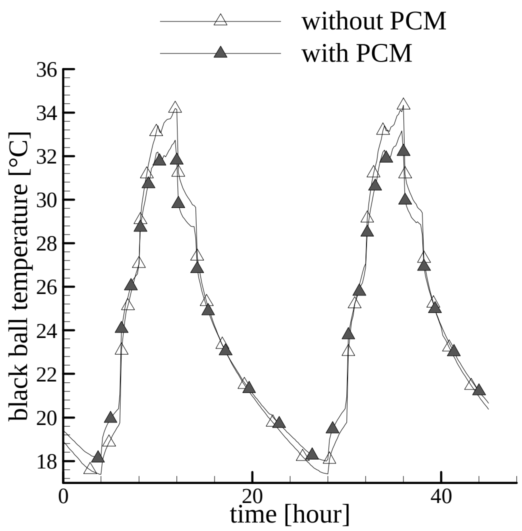


Figure 9: Globe temperatures for the cases tested – Mid-season case

DISCUSSION

In this article experiments are carried out in a test cell and in controlled external conditions with solar contributions which is new in the literature.

According to the comparisons presented in this paper, the PCM composite walls allow to enhance the thermal comfort of the room in the summer case. The maximum air temperature is decreased by about 4°C and the minimum increased by about 1°C. This free regulation is obtained only by using PCM material. Another notable effect is the natural convection enhancement, allowing to have a better mixing concerning the air in the room and decreasing the thermal stratification.

The globe temperature evolution confirms the enhancement of the thermal comfort because of the PCM used.

Further investigations are needed in order to have more results concerning various external conditions. The winter case is actually our subject of interest. For that case, we hope a reduction of the heating energy consumption with the use of PCM materials. These results will be used to validate numerical codes for the prediction of energy consumptions of such systems.

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Optimization Of Double Skin Facades For Buildings: An Office Building Example In Ankara-Turkey

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ABSTRACT

There are about 16 millions building in Turkey. The total energy consumption of these buildings is approximately 3.48×10^5 GJ. The energy consumption of the office buildings having HVAC systems can be stated as 150 kWh/m^2.year.Turkey is in the warm climate band and the day time outside air temperature in summer (May ~ September) is about 30°C. Parallel to the increasing comfort demands, energy consumption also increases. These factors make the necessity for increasing energy efficiency of buildings apparent.

In this study, a life cycle analysis is introduced for an office building having double skin obtained by dressing a second glace façade to the two main façades and for the situation of natural ventilating and night cooling in summer. In according to the results from the analysis, approximately 45% energy saving is possible by this application. Consequently the energy use will be reduced from 203 kWh / m^2.year to 106 kWh / m^2 year.

Keywords: Existing office building, natural ventilating, optimization, simulation, thermal comfort

1. INTRODUCTION

Starting from the year of 1970, increasing researches in the world on the energy efficiency have resulted in a number of new standards and regulations. In addition, Kyoto protocol against global warming and ozone layer depletion is taking effect more seriously all over the world. At this connection, both the energy efficient design of new buildings and increasing the energy efficiency in existing building turned to the one of the leading issues of construction sector. The retrofit projects having the harmony between renovation and energy efficiency are more attractive than those having these processes separately [1]. In this study a life cycle analysis has been made for an existing office building situated in a complex of a public institution. The two principal façades (back and front façades) of this building is planned to be dressed by two glass skins and a cost-benefit analysis has been conducted for the double skin construction obtained, for the situation that natural ventilating and night-time-cooling was used in summer.

2. EXISTING CHARACTERISTICS OF THE BUILDING

2.1. Architectural Characteristics

a) Building working areas: Normal floors of higher blocks: 900 m^2/floor (14 floors exist), Ground floors (ground, 1st and 2nd floors) : 2.800 m^2/floor, Basement floors (3 floors): 4.300 m^2/floor,	b) Net usable area: 34.000 m^2.

The external view and the normal floor plan in higher block of the building are shown in Figure 1 and 2.

Figure 1.External view of the building.

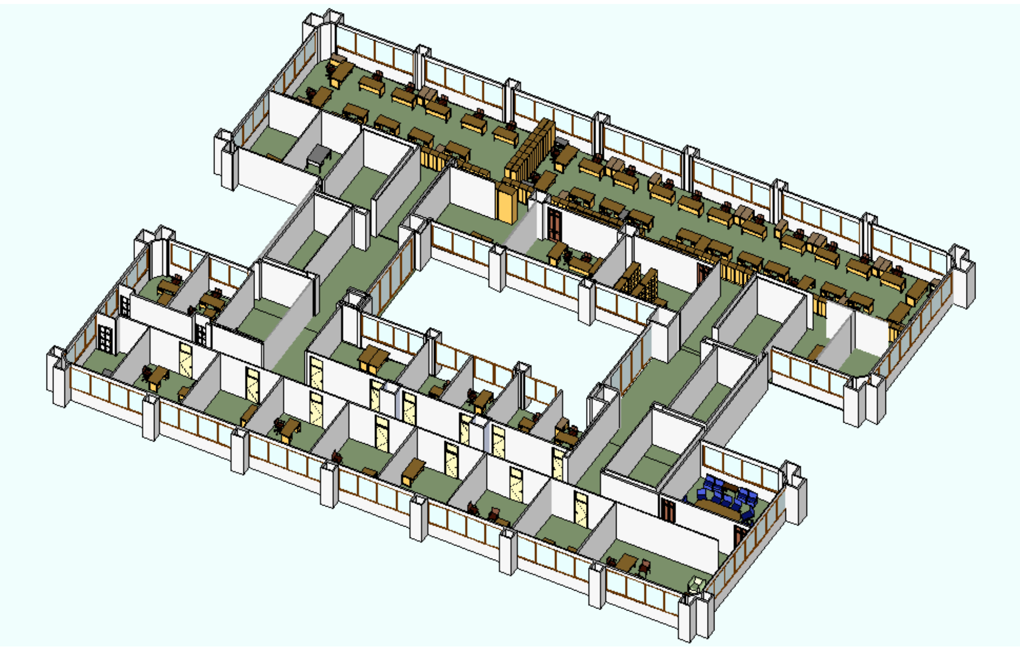


Figure 2 the sample floor plan for upper sections of the building.

As it can be seen in Figure 2, the floors have been oriented as open and closed offices. The natural ventilating is considered as single-sided ventilating in the offices facing to the principle façades.

2.2. Building Façade

The building has been oriented in southwest-northeast axis. The basic façades for heat gains areas are given in Table 1:

Table1 The external surface areas basis for heat gains.

Directions	Glass area (m^2)		Cement(wall) area (m^2)	
	Per Floor	Total	Per Floor	Total
SW	80	1120	120	1680
NE	80	1120	120	1680
NW	55	770	65	910
SE	55	770	65	910
TOTAL		3780		5180

The façade of the existing building has been constituted by clear glasses (solar transmittance ε=%100 and over all U coefficient= 4.0 W/m^2K) with alumina frame. In this case, the heat gains in winter and the heat losses in summer are high.

2.3. HVAC Systems

Since, the building has been got into operation on 1988 and HVAC systems had been based on the old and out-dated technologies; these systems have completed their life-cycles. The second and lower floors of the building, are heated/cooled by the air, supplied by the air-handling units. These systems are the constant flow, variable temperature, single zone systems. Consequently they create inefficient and uncomfortable spaces. The 4th and upper 14 floors; are heated / cooled by a two-pipe fan-coil system and there is no possibility of outside air supply. For this reason, there always are internal air quality problems in the building. Furthermore, the two-pipe fan-coil system provides insufficient comfort for transient seasons.

In sunny winter days and particularly in southwest façades, the temperatures, rise in the extent that it can disturb the comfort levels in the offices. Also in summer; the electrical energy consumed by the fan-coil system is much more than normally required. This is because the façades are not protected against solar radiation.

2.4. Annual Energy Consumption

The energy consumption (readings from the meters) to meet the heat gains/losses and their monetary values are given in Table 2 (as the average of the years between 2002~2005).

Table 2a The actual annual utility (natural gas, electricity) consumption rates of the building.

Item	Type of Consumption	Consumption Rate	Unit Price (EU)	Total (EU)
1	Electricity	8.000.000 kWh/year	0.11 EU/KWh	880.000
2	Natural gas	635.000 m^3/yıl	0.30 EU/ m^3	190.500
TOTAL				1.070.500

40% of natural gas and 30% of electrical energy in Table 2a, are used in the buildings other than this building. Thus the natural gas consumption of the building is (0.6x635.000=) 381.000 m^3/year and the consumption of electrical energy is (0.7x8.000.000=) 5.600.000 kWh/year. The distribution of energy consumption is determined, checking the nominal powers, numbers, operating times, and annual energy consumption of the devices using electricity. The data given below have been based on;

			Remarks
Annual heating period	=	1600 hrs	
Annual cooling period	=	1600 hrs	(for comfort purposes)
Heat value of natural gas	=	11 kWh/m^3	
Energy equivalent of natural gas consumed in boilers	=	4.191.000 kWh	(381.00 m^3x11 kWh/m^3 =)
Annual electrical energy consumption for auxiliary heating equipments	=	425.000 kWh/year	burners, fans, pumps etc
Total energy consumption for heating	=	4.616.000 kWh/year	(4.616.000 + 425.000 =)
Unit energy consumption for heating	=	136 kWh / m^2	(4.616.000/ 34 000 =)

The electrical energy consumption of the building (rounded values) is given in Table 2b.These values are calculated taking into consideration of metered values, the powers and the operating times of the equipments. As apparent from the Table 2b, 2.282.000 kWh/year energy is consumed by comfort cooling systems (chilled water cooling units, circulating pumps, cooling towers etc). This is approximately 40% of the total energy use in the building.

The total heat gains from solar radiation and the internal loads is 3.880.000 kWh/year. Then, the overall COP of the cooling system can be estimated as (3.880.0000/2.282.000 =) 1.7. And the unit energy consumption for cooling will be (2.282.000 kWh/yıl/34.000 m^2 =) 67 kWh/m^2yıl. Thus, the total unit energy per m^2 for heating and cooling is (136+67=) 203 kWh/m^2. Since the uncontrolled, single glass fenestration with alumina frames having no shading features and inefficiencies in HVAC systems, this figure of unit energy consumption is significantly higher.

Table 2b Distribution of electrical energy consumption by sources.

System	Electrical energy consumption (kWh/year)	(%) of total
Comfort cooling system	2.282.000	40.7
Continually operating cooling system (EFT, DATA Execution etc)	1.133.000	20.2
Heating systems (electrical)	425.000	7.6
Plumbing systems	170.000	3.1
Illumination	340.000	6.1
Elevators	250.000	4.4
Aspirators	500.000	8.9
BIM, PC, copy, printer, fax, telephone centrals	400.000	7.2
Others	100.000	1.8
TOTAL	5.600.000	100

3. Basic DATA For The Calculations

3.1. Climatic Characteristics

The outside temperatures between May-September through which natural ventilating is possible, are shown in Figure 3 and the wind velocities (m/s) for the same period in Figure 4 [2]. As can be seen, Ankara in summer is hot and dry in day times and cool in night times. For these characteristics of the city, night-cooling and natural ventilating seems to be possible.

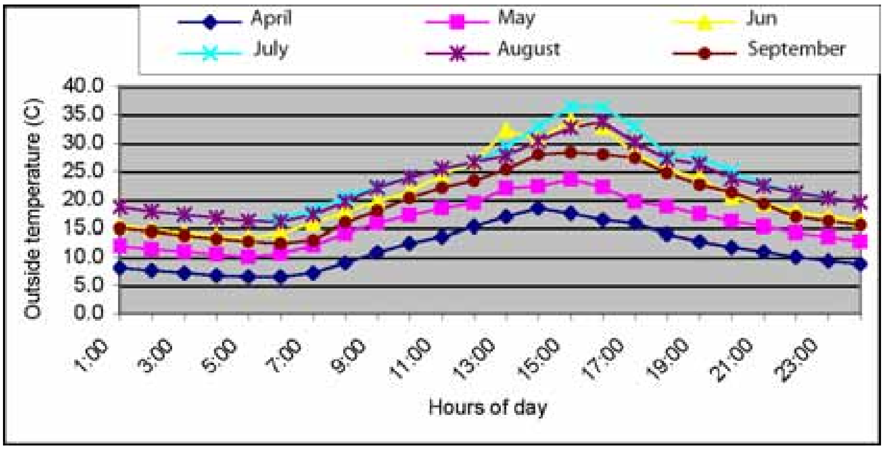


Figure 3 The average dry bulb temperatures between April~September [2].

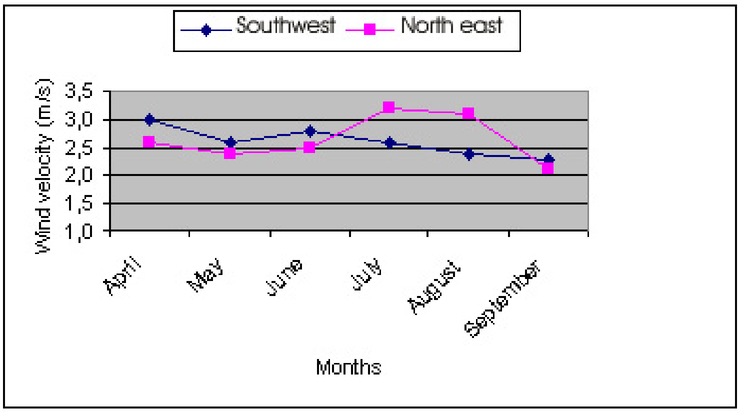


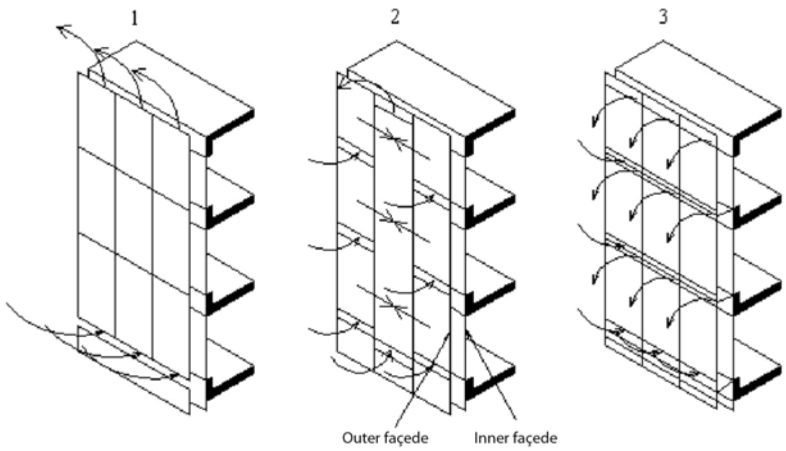
Figure 4 The average wind velocity between April~September [2].

3.2. Heat Losses/Gains And Ventilating

In the calculation of heat losses and gains the glass characteristics, the U values, inside (23°C) and outside temperatures given in Table 3 are used. The wind velocities from Figure 4 and 10 m^3/hm^2 natural ventilating air are considered. In addition, 12 W/m^2 for peoples, 10 W/m^2 for illumination and 8 W/m^2 for electrical devices heat gains are assumed [3, 4].

4. Double Skin Façade Systems

To reduce the energy consumption, it is considered to dress the secondary glass façade for the two main façades of the building. For the natural ventilating, mainly three alternatives shown in Figure 5, are taken into considerations. For the first alternative, the air flow is provided along the entire façade. For the second alternative; the air is taken from the openings arranged for the individual floors and exhausted from the upper floor. For the third alternative, the air is introduced in each floor separately and exhausted from the same floor, again separately. In all of the alternatives, to reduce the mechanical cooling system operation time, in addition to natural ventilating the dampers on the glass façade will be leave open all night long for night-time cooling. In winter, the air between the two façades will be trapped and an additional insulation will be obtained. Furthermore, in suitable weather conditions, natural ventilating will also be possible during day-times. [5].



Type 1 *Type 2* *Type 3*

Figure 5 Alternatives for ventilating in two skin building [5].

4.1. Dressing A Second Glass-Skin To The Building

Mainly two measures have been considered for the building: (1) changing the glasses in the façades of the building, and (2) substitution of the old HVAC systems with new and more efficient systems. But, since the building is occupied and there are peoples working in the offices, it will be very difficult to renew the glasses with the low-e fenestration and insulated frames. For this reason and since this application will not be as effective as desired, instead of changing the glasses, it has been preferred to dress another glass skin to the main facades. Thus, the building will have an external inoperable glass skin and an operable inner skin (double skin façade). The glass characteristics of additional Low-e glass skin shown in Table 3.

Table 3- Physical characteristics for alternative glasses [6].

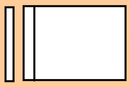
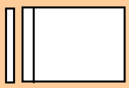
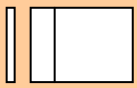
Type	Total transmittance	Shading coefficient	Heat transfer coefficient (U) EN 673 (W/m^2K)	Unit costs (EU/m^2)
6+12+6* mm clean heat-glass (tempered)	0.69	0.81	2.8	50,00
6+12+6 mm low-E haet glass (tempered)	0.59	0.63	1.8	65,00
6+12+6 mm low-E comfort glass (tempered)	0.40	0.56	1.7	80,00
* 6+12+6= glass tichkness+ air gap+glass tickhness				

The unit prices for the glasses include carrier profiles (frames) and mounting workmanship. These additional costs are constant for all alternatives and valued as 150 EU/m^2.

4.2. Second Façade Options

The second skin that can be dressed into the this building may be 10 configurations as shown in Figure 5. Theese options are listed in Table 4. The total glass area of the two façades of the building is 3780 m^2 and approximated costs of the system including frames also, shown in the table. Since the number of dampers and automatic controllers etc is lesser, the type 1 is least expensive Type 2 is moderately expensive and type 3 is most expensive system (Figure 5).

Table 4. The façades options used in life-cycle analysis.

Option number	**Façade type**	**Arrangement of double skin elements**	**Shading coefficient (SHGC)**	**Heat transfer coefficient (W/m^2K)**	**Estimated investment cost of second façade (EU)**
1	Existing situation (no second skin)		1.00	4.00	0,00
2	Second skin: 6+12+6 mm clear glass *(Figure 5, type 1)*		0.81	2.80	756.000,00
3	Second skin: 6+12+6 mm clear glass *(Figure 5, type 2)*				770.000,00
4	Second skin: 6+12+6 mm clear glass *(Figure 5, type 3)*				795.000,00
5	Second skin: 6+12+6 mm insulated glass low-E *(Figure 5, type 1)*		0.63	1.80	775.000,00
6	Second skin: 6+12+6 mm low-e insulated glass *(Figure 5, tip 2)*				792.000,00
7	Second skin: 6+12+6 mm low-e insulated glass *(Figure 5, type 3)*				826.000,00
8	Second skin: 6+12+6 mm low-e comfortal glass *(Figure 5 ,type 1)*		0.56	1.70	791.000,00
9	Second skin: 6+12+6 mm low-e comfortal glass *(Figure 5, type 2)*				817.000,00
10	Second skin: 6+12+6 mm low-e comfortal glass *(Figure 5 ,type 3)*				835.000,00

5. Analysis for Natural Ventilating

The performance of natural ventilating is dependent to the outside air conditions. Not to cause discomfort and energy losses, the dampers will be controlled by BAS's (Building automation systems). The calculations related with natural ventilating are as below.

5.1.1. Effective Pressures

5.1.1.1. Pressure Differences Caused By The Wind

The average air velocity (m/s) on building façade can be estimated by [7];

$$U_h = U_{met}\left(\frac{\delta_{met}}{H_{met}}\right)^{a_{met}}\left(\frac{H}{\delta}\right)^{a} \qquad (1)$$

Where;
U_{met} is the meteorological air velocity (m/s), H_{met}= 35 (mid plane of the building); the meteorological exponent α_{met}=0.33 (from ASHRAE), flow exponent α_{met}=0.22, meteorological wind depth δ_{met}=460m, wind depth δ=850m (assumed) . The pressure caused by wind velocity can be estimated by;

$$\Delta P_{wind} = \rho_a C_p \frac{U_h^{\,2}}{2} \qquad (2)$$

Where ρ_a is the outside air density (kg/m^3), C_p is the local wind coefficient given by ASHRA as;

$$C_p = C_s - C_{\text{int}er} \qquad (3)$$

C_s is taken as -0.3 and C_{inter} is assumed to be -0.2 [7].

5.1.1.2. Pressure Differences Caused By Temperature Differences (Chimney Effect)

There will be a pressure difference originated by the density difference between the hot and cold air. The difference caused, by density differences of the air, can be estimated by the equation below [7];

$$\Delta P_s = \rho_i g T_{\text{int}} (H - H_{neut}) C_d \frac{T_{\text{int}} - T_{out}}{T_{out}} \qquad (4)$$

Where ρ_i *i*nternal air density (kg/m^3), H_{neut} height in reference to the neutral axis of the building (m), H *a*verage height of the building (m), T_{int} inside air temperature (°C), T_{out} outside air temperature (°C), C_d total wind coefficient. It is between 0.63 ~0.82 and here taken as 0.63.

5.1.1.3. Total Pressure Difference

This pressure difference is obtained by adding the dynamic pressure difference and heat affected pressure differences each other and can be stated as;

$$AP_t = \Delta P_{wind} + \Delta P_s \qquad (5)$$

5.2. Air Flow Rate

The air flow rate in single sided ventilating through the windows can be estimated by the formula below [8];

$$V_i = C_d A_i \left(\frac{2\Delta P_t}{\rho_a} \right)^{1/2} \qquad (6)$$

Where V_i air flow rate (m^3/s), *i* number of openings, A_i opening area (m^2), ΔP_t total pressure drop (Pa), ρ_a density of outside air (kg/m^3).

The total air flow rate (m^3/h) can be estimated as;

$$V_{i-tot} = \sum_{i=1}^{n} V_i \qquad (7)$$

Air change rates for offices can be estimated dividing the air flow rates to the total space volume.

$$n = \frac{V_{i-tot}}{V_{tot,space}} \qquad (8)$$

Where n air change rate (1/h), V_i-t_{ot} total air flow rate (m^3/h),
$V_{tot.space}$ total volume of spaces (m^3).

6. Heat Transfer Between The Surfaces Of The Spaces And The Ventilating Air

During natural ventilating, heat transfer is between the surfaces of hot internal space and cold outside air. In other words, ‘cold’ is stored in the space walls and elements. This stored ‘cold’ is transferred to the warm air in the space and thus, less operation time for the mechanical cooling system is achieved. The heat energy transferable to the ventilating air can be estimated by the equation below;

$$Q = hA(T_S - T_{out}) \qquad (9)$$

Where A the area of the surfaces swept by the ventilating air (m^2), h overall heat transfer coefficient (W/m^2K), T_s surface temperature of space elements (°C), T_{out} temperature of outside air (°C).

When the ventilating commences, since the space temperature is gradually reduced; T_s is a time dependant characteristic. These temperatures for the months of natural ventilating and night-cooling are given in Figure 3.   It is difficult to determine the effects of the temperature of the elements such as tables, chairs, etc and an accurate h value for the space components.

The only solution for the issue seems to model via CFD techniques and use approximate values for the coefficients mentioned above. For most of the engineering purposes, the approximate value of h is sufficiently acceptable. For example, in free convection; h value can be evaluated as 6~30 W/(m^2.K) [9]. Despite that there is a wind and/or chimney driven ventilating; h value has been taken as 10 W/m^2K as a safely approach.

7. Life Cycle Analysis

An optimization process is needed for minimizing the life cycle for options in Table 4. This optimization process will aim to reach a, minimal total cost value which is the addition of investment costs and operation costs. The target function used for the minimization of life-cycle costs (20 years) is;

$$C_T = C_{fi} x CRF + (C_{en} + C_{maint}) = \min \qquad (10.1)$$

The limitations are;

$$E < E_1$$
$$PPD < \%10 \qquad (10.2)$$

where ;
Where C_T present value of annual life-cycle cost (EU/year), C_{fi} initial cost of the façades (EU), E annual energy consumption (kWh/ year), E_1 annual allowable energy consumption (kWh/m^2year), C_{en} present value of annual energy cost for life cycle period (EU/year), C_{maint} present value of annual maintenance cost (EU/year), PPD Predicted Percentage Dissatisfied (%), CFR capital recovery rate for life-cycle of the building. CRF can be estimated as follows;

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} \qquad (10.3)$$

Where i average annual interest rate (%), N life expectancy of the system (year).

The influence of the glass and façade types on investment and energy costs has been defined; considering different options for glasses. Similarly influence of different combinations has also determined. During the optimization process, it is considered that the PPD have to be lower than 10%. Investment cost of each option includes; material per m^2 (fenestration, frames, grilles, automatic control system etc), workmanship, transportation, auxiliary equipments etc. To define the energy costs during life-cycle period; it is needed calculating the energy costs for the heat losses/gains and converting them to present values. Annual energy costs of the options have been calculated as the product of the unit energy price and the annual energy consumptions estimated by Carrier E-20 (Hourly Analysis Program) software. After the investment and operating costs of the options defined, the optimal one has been found using a PC software based on Revise Simplex method [10].

8. Conclusions

In the Equation (10.3) assuming the annual interest rate (i) 4% and the life-cycle period 20 years. CRf is obtained from the Equation (10.3) as 0.0726. Then Table 7 has been designed solving the Equation (10.1) for every option. From the Table, it is apparent that the most suitable option is number 8 having 4.7 years recovery period for the investment. While natural ventilating improves inside air quality; night-cooling provides energy conservation reducing wall temperatures. As a consequence of these measures, the energy consumption of the building is reduced from 203 to 106 kWh/(m^2.year). In fiscal terms, this means 228.000 EU/year (485.000-257.000=) and for the life-cycle of 20 years 4.560.000 EU/year [values rounded]. The value that has to be invested, calculated as 791.000 EU. On the other hand air change is calculated as 3.5 change/hour during the natural ventilating season.

Table 7 Costs from the life-cycle cost analysis.

Alternative no	**Façade type**	**Initial cost of second skin (EU) (1)**	**Annual operating cost (EU/yıl) (2)**	**Annual total cost (EU/year) (3=1xcrf+2)**	**Simple recovery period (Year) (4=1/(485.000 – 3)**
1	Existing situation (no second skin)	0,00	485.000,00	485.000,00	-
2	Second skin: 6+12+6 mm clear glass *(Figure 5, type 1)*	756.000,00	311.000,00	366.640,00	6.4
3	Second skin: 6+12+6 mm clear glass *(Figure 5, type 2)*	770.000,00	335.000,00	391.670,00	8.3
4	Second skin: 6+12+6 mm clear glass *(Figure 5, type 3)*	795.000,00	350.000,00	408.510,00	10.3
5	Second skin: 6+12+6 mm insulated glass low-e *(Figure 5, type 1)*	775.000,00	288.000,00	345.040,00	5.5
6	Second skin: 6+12+6 mm low-E insulated glass *(Figure 5, type 2)*	792.000,00	298.000,00	356.290,00	6.2
7	Second skin: 6+12+6 mm low-E insulated glass *(Figure 5, type 3)*	826.000,00	322.000,00	382.790,00	8.1
8	Second skin: 6+12+6 mm low-e comfortal glass *(Figure 5, type 1)*	791.000,00	257.000,00	315.220,00	4.7

Alternative no	**Façade type**	**Initial cost of second skin (EU) (1)**	**Annual operating cost (EU/yıl) (2)**	**Annual total cost (EU/year) (3=1xcrf+2)**	**Simple recovery period (Year) (4=1/(485.000 – 3)**
9	Second skin: 6+12+6 mm low-e comfortal glass *(Figure 5, type 2)*	817.000,00	283.000,00	343.130,00	5.75
10	Second skin: 6+12+6 mm low-e comfortal glass *(Figure 5, type 3)*	835.000,00	305.000,00	366.450,00	7.0

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An Analysis of Environmental Performance and Improvement of the Envelope for High-Rise Residential Buildings

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SUMMARY

Due to high-rise residential buildings and extension of balcony, to resolve discomfort of indoor-environment and the problem which energy consumption increases, high-rise residential buildings, coming natural ventilation and decreasing expense of an air-conditioning system., enable envelope system to be developed.

The object of this study is to present the improvement on envelope of high-rise residential buildings to reduce heating and cooling load. To improve the environmental performance of envelope, it is necessary to modify envelope vent system and ensure intermediate space.

In this study, the shape of vent in high-rise residential buildings has been inspected, and the survey about the environmental conditions related to the envelope of high-rise residential building has been conducted. Environmental measurement has been done to examine out door air quality for natural ventilation.

INTRODUCTION

The envelope of building is, horizontally and in section, one of project factors different from other building around on surface and mainly decoration and also from a variety of weather conditions such as solar radiation, warm and cold internally protects residential environment of house so that it is evaluated as one of the important function maintaining uniform house-environment

In Korea, balcony section of high-rise residential buildings is apt to be extended as front-glass on case of high-rise residential building. While the window glass provides not only, unit residents of natural lighting and view, but also is essential for psychological aspect, attaching to outdoor. It increases heating and cooling load for much heating acquisition in summer caused by high overall heat transmission and much heat losses in winter

Also it is reared up that the problem of comfort for a resident and window opening in case of high-rise building which high window pressure exists. Especially the seriousness comparatively appears on the side of natural ventilation in the building being so for mechanical ventilation. The most problem is difficult to open window because of high window pressure outside in case of high-rise building.

Not only most of high-rise buildings to resolve that ventilation problem have the equipment of mechanical ventilation, but also come to minimize window size as possible as opening. But the induction of the mechanical ventilation increase needs for mechanical-cooling as decreasing the probability of natural ventilation through window frame.

Now even though the envelope of high-rise residential building comparatively is less than architecture regulation related to building height and structure, as the curtain wall of older high-rise office is selected without any other improvement, For matching residence that the envelope performance needs to be improved, through those above, it is supposed to think of indoor-environmental performance of high-rise residential buildings.
In this study, on the basically step of improvement project of envelop about building in order to reduce the energy of high-rise residential building, To catch on problems of high-rise building now we had posed a survey about residents of older high-rise residential building. Also practicing natural ventilation through outdoor-air, as a plan of reducing heating and cooling load, it had measured environmentally whether natural ventilation of high-rise residential building now brought in or not.

PROPOSED APPROACH

The survey of preexistence envelope of high-rise residential buildings

The curtain wall involved in the department of envelope about high-rise residential building is largely classified by the department of window and wall having glass. As the performance of those two parts, it is decided that the performance of curtain wall about two parts above is.
It is practiced that the survey at Youido, Yongsan-gu, Gangnam-gu, Yangcheon-gu in Seoul in the country ranged in high-rise residential buildings, total 15 areas is. After practicing survey of envelop pattern, survey target building have applied most of the curtain wall. A part of curtain area divided frame have been using.
In order to resolve the uncomfortable problem of indoor-environment caused by high-rise tendency of residential building, balcony extension and the problem of increasing energy consumption, we had focused shape of vent in the purpose of improving envelope performance of high-rise residential building as easy as natural ventilation and saving heating and cooling expense.
Vent pattern as well as vent area is major one of factors deciding vent quantity, due to that, it is decided that induction quantity of opening air is. Therefore having investigated vent opening-pattern of high-rise residential building's envelope, we have gained the actual condition of opening-air induction.
Shape of vent is applied by most of top-hinged out swing window type except low-floor department of T- high-rise residential building at Youido and that is adopted by casement type, T-low-floor department at Youido.

The simulation of preexistence envelope of high-rise residential buildings

Even though shape of vent is important not to be able to ventilate directly caused center core structure, as we have known that the area and opening-angle which ventilation window occupies are small, It is evaluated that this is to reduce the risk of the accident caused the fall and excessive wind-pressure because of wide opening-area.
In case of top-hinged out swing window type, as ventilation volume is 2600 m^3/h such as simulation conclusion, a chart above, while it satisfies 1240 m^3/h, encouragement ventilation volume, because the ventilation volume is comparatively smaller to be lower than different opening-method, natural ventilation to the cooling of the open air is enough carried out. In order to resolve that, in case of top-hinged out swing window type, transforming the tall and width of window, it can correct faults. But we have known imperfection of ventilation performance that opening-angle is not big. (Figure 1, Figure 2)

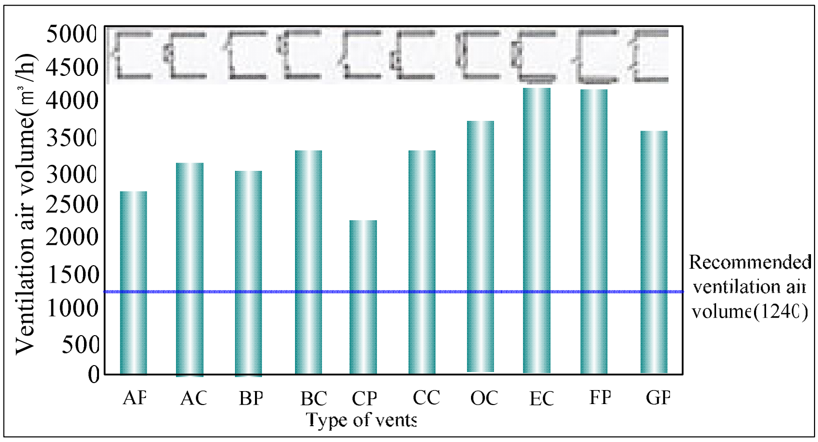


Figure 1. Amount of ventilation simulation by vent form

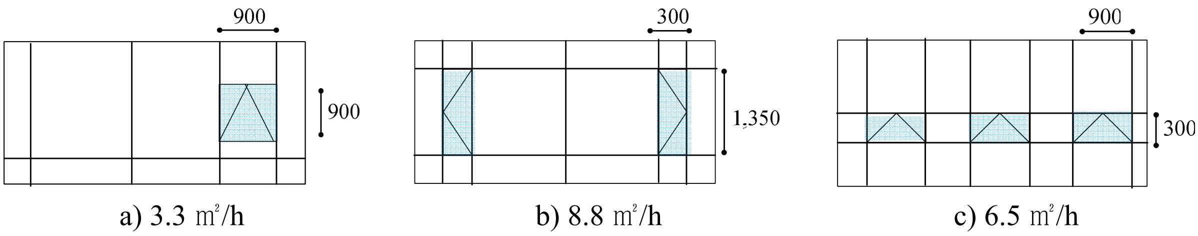


Figure 2. Amount of ventilation according to window area and type (unit:mm)

FIELD SURVEY

Survey outline

To get gist of envelope of high-rise residential building influences indoor-environment performance we had posed survey and deep interview related to environment performance for the envelope of high-rise residential building complex. Through it, after catching on environment performance how the resident in the high-rise building complex feels in their life, eliciting the problem, we got the improvement outline.

The building, which is surveyed in this study, is a complex in Seoul, Korea. It consists of the lower podium for commercial shops and higher stories for residential building. Although it is a complex, the commercial zone and the residential zone are separated from each other. This study focuses mainly on the higher stories for residential building. To increase the reliability of survey, people of the survey is selected that in the building the height and window of a household gives a steady distribution. The survey is carried out to a high-rise residential building in Seoul. To increase the reliability of survey, people of the survey is selected that in the building the height and window of a household gives a steady distribution. The residential building has 37 floors above ground and 6 underground, which located in the metropolitan Seoul .The envelope is composed of AL.Curtain wall +24mm Lowe pair glass and the general characteristics are like Table 1. Fig. 3, Fig. 4 and, Fig. 5 is the typical. The items of survey are composed of factors for thermal, indoor; lighting, sound environment and the details are like Table 2

Table 1. Building description

Location	Seoul, KOREA
No. of floor	37 floors
Height	111.63M
Use	Residential
Envelope	AL.Curtain wall +24mm Low–e pair glass

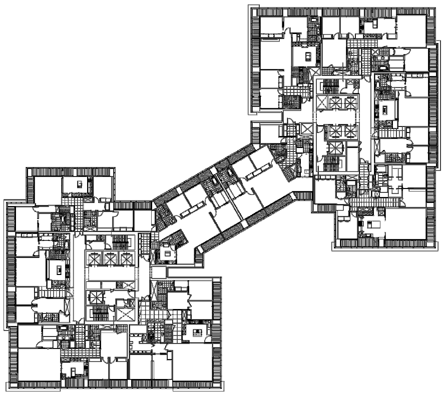


Figure 3. Typical floor plan of the survey building

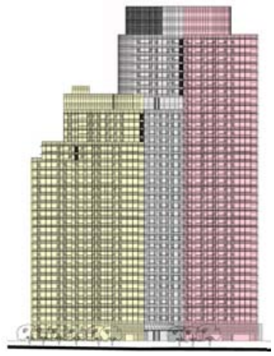
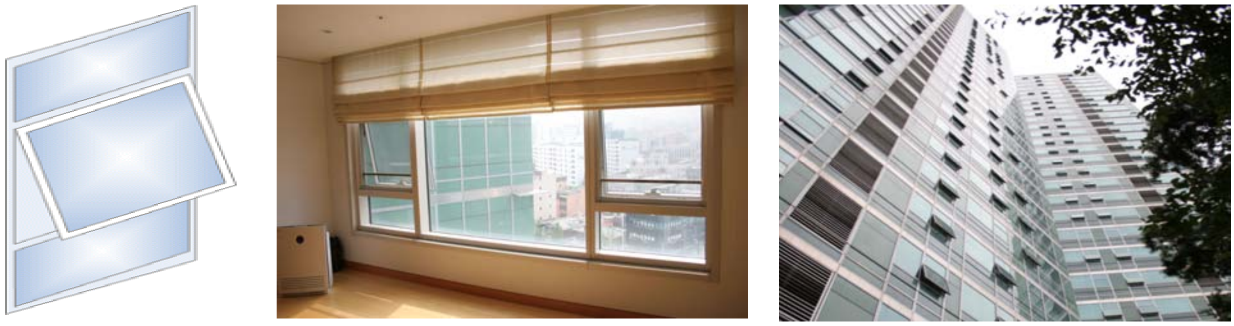


Figure4. Elevation of the survey building



a) Window type b) Interior Envelope c) Exterior Envelope

Figure 5. Envelope of the survey building

Table 2. Composition of resident question item

Classification	Item
Thermal environment	Thermal ,Humidity, Condensation ,Type of sunshade, Purpose of sunshade Frequency sunshade, operation time, Heating and Cooling expense
Indoor air environment	Draft and effect of outside wind, To use availability of ventilation facilities, opening method of windows and doors and opening -area, Frequency of natural ventilation, Satisfaction of natural ventilation, Satisfaction about IAQ
Lighting Environment	Type of sunshade, purpose of sunshade, frequency to use shade, vibration of sunshade, appearance of glare and effect
Sound Environment	Outdoor noise influence, Sound insulation performance of window

Survey result and discussion

Because of the survey conclusion about a resident, the air tightness drops of envelope. heat and environment problem such as water leak and infiltration occurrence, airflow speed and the deficiency of natural ventilation, comparatively smaller of opening area of window, so people feels psychological stuffy. The arrangement of the problem at a part of envelop of high-rise residential building is like Table 3.

1) Natural ventilation performance through outdoor air

Taking a complex of a tendency of high building, cooling load have increased, the increase of cooling load has been aggravated by envelope utility of high curtain wall which the rate of glass area is high. But because of the problem of wind pressure and security, decreasing Opening area of ventilation window, ventilation volume has decreased.

Table 3. Classification and composition of resident question item

Classification	phenomenon	Diagnosis
Thermal environment	Heating and Cooling Load	· Glass area rain because is high excessive inflow of one thing · Buffer zone delete by balcony extension
	Water leakages and infiltration	· Envelop airtightness decline
	Cold draft	· Heat-loss occurrence caused envelop in winter · Envelop airtightness decline from balcony extension · Disability of heating in extended space of balcony
Indoor Air environment	Decrease of air speed	· Small of opening square of ventilation window · Difficult to feel air sense of a resident · Even though more than standard of average ventilation volume, feeling deficiency · When the speed of air speed outside is up, window is suddenly locked
	Deficiency of natural ventilation	· Disability to ventilate directly as a center core structure · Decrease of cooling effect of open air as opening area of decreased ventilation window · Cooling load and period increase caused natural ventilation decrease
	Deficiency of psychological opening sense	· Small of visible opening area · Psychologically stuffy and discomfort not to feel air-flow · Unpleasant of top-hinged out swing window type and sliding type about the apartment of older finest shape
Lighting environment	Ineffectively operation of shade	· Do not operate after establish shading device · Shading device e does not warmer ability
Sound environment	Satisfaction of need condition for a resident	· Sound interception by tree on lower floor · Do not problem caused distance attenuation effect of noise on higher-floor

In that case, not to get cooling effect of open air, lengthening cooling period, it badly influences directly to energy utility quantity. Therefore, it is required that in the middle, until open-air cooling comes enough, the design of open-mouse to be able to make enough ventilation quantity is.

2) Increasing air speed

A resident of high-rise building complex appears to want to feel air sense through induction of open air. This tendency was more prominent to the resident who had lived in plate-type block having experience of cross ventilation before living in a high-rising complex. Therefore, need introduce outdoor air at the higher air current speed than present within scope that do not cause the uncomfortable. On the other hand, low stories part and upper floor part can do opening area measure differently because outside wind pressure gives and is difference in the air current speed.

3) Psychological openness

The residents wanted to feel well ventilated psychologically unconcerned in amount of ventilation. The high-rise residential building that was resident question target was measured

was satisfying required amount of ventilation and indoor air quality.. However, a window was not opened widely compared with a window of plate-type block and the size of the window was quite small so the residents might feel uncomfortable psychologically
Instead of installing the top-hinged out swing window system in the building, careful consideration of various type and combination of ventilation window system which may give a resident visible wide open space would be required to solve out this problem.

4) The Insulation

Since the space of balcony has been renovated as one of residential space after the extending the balcony was legalized by a law, it gives an effect to temperature conservation of interior space because of the sunshine through the window has bad insulation ability.
In winter season, it may cause discomfort for residents because of cold draft and the damage of interior material because of a condensation around of the window. In the case of summer, it may cause the outstanding air conditioning cost because of the room temperature increase and gives a problem with envelope with weak thermally.
Because the existing balcony acted like intermediate space environmentally, we could check plan that reduce heat road by introducing intermediate space at curtain wall of high-rising residential building. Besides, the insulation ability of envelope is decided by a mixture of glass during manufacture process, the type of interior window and exterior window, the width of intermediate space, the way of unlocking the ventilation window and area of those, and so on. Consequently, a choice of envelope would be determined in many ways and totally considered in any conditions.

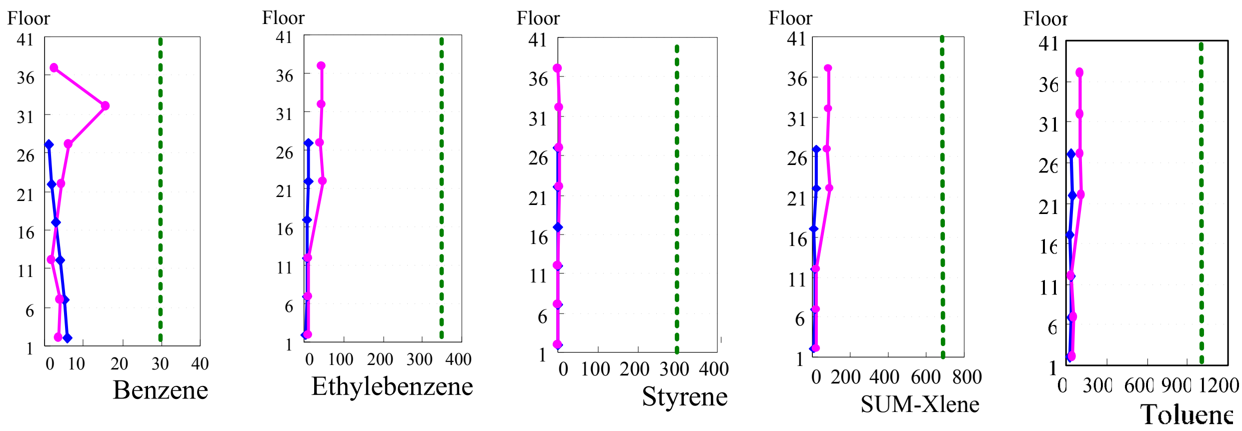
MEASUREMENT OF THE OUTDOOR AIR

Measurement outline

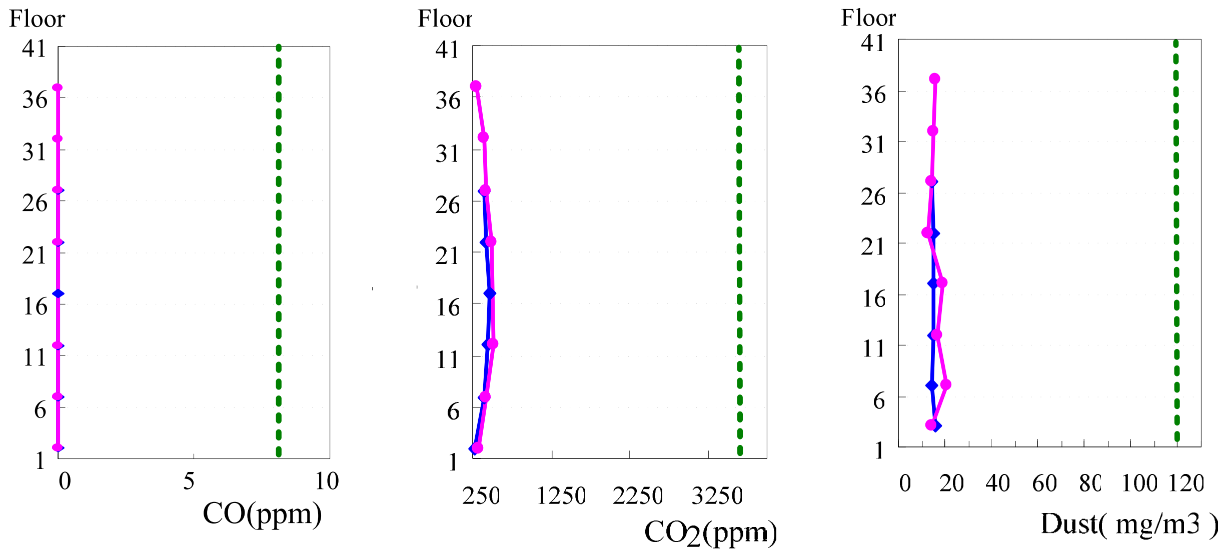
Need to examine propriety of outdoor-air introduction so that can reduce air conditioning cooling load by natural ventilation extension application, and I measured cleanliness of outdoor air for this.
Stand-by status by height measured each about 40heights of story (120m) air contaminant concentrations from same native place vent 5 floors interval with west exposure of A high-rise residential building two coppers located downtown in Seoul

Measurement result and discussion

Pollution level of outside spot results of measurements outdoor air of high-rise residential building estimates that it is possible to appears by thing which do not reach greatly and introduces outdoor air into indoor without special device for natural ventilation do acceptable level.
Since it had lower degree than legal air pollution level around both high stories and low stories of the building, the air pollution doesn't matter with the height-level of building.



a) VOCs Concentrations



b) CO, CO_2. Dust Concentrations

Figure 6. Outdoor-air contaminant concentrations measured data by heights

CONCLUSION

In this study, it includes the contents, such as the survey of residential people and extensive interview with the purpose of drawing up a plan in architectural environmental approach for prospective problem with envelope system which can be occurred by increasing the number of the high-rise residential building.

It aims to bring out proper choice of envelope material for the high-rise residential building which needs to maintain good insulation system especially in this country which has distinctive four seasons, and to give a suggestion to improve the insulation system which can economize in energy in this high population density city. Considering the plan that introduce double-skin façade system in the high-rise residential building which can do natural ventilation efficiently as reduce wind pressure and solve buffer zone by balcony extension.

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Studies on Energy Storage Capacity of a Spherical Encapsulated PCM Using Eutectic Salt as Phase Change Material

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SUMMARY

The need for higher productivity to match the growing competition has forced employers to look for better Indoor Work Environments. Since most of the buildings are being air conditioned, the Heating Ventilating and Air Conditioning (HVAC) systems account for nearly 40% of total building energy consumption. To reduce this energy consumption Thermal energy storage systems (TES) were developed. The Latent heat thermal energy storage (LHTES) systems using water/ice are commonly used for peak shifts in electrical demand. To maintain a temperature below 0°C, the water/ice storage system is not suitable, but for a HVAC application the chiller requires a temperature below 0°C. In TES systems, phase change materials (PCMs) have attracted a great interest as thermal storage materials because the thermal energy is stored in them with high density and it is also possible to maintain below 0°C. This research uses Sodium nitrate along with nucleating agent (borax), freeze point depressant (ethylene glycol) and thickening agent (propylene) as a eutectic mixture in PCM. The percentage of composition of these selected phase change materials are Sodium nitrate 33%, ethylene glycol 27%, borax 22% and propylene 18 %. The numerical studies are conducted to determine the maximum storage capacity of the eutectic mixture and also the influence of the nodule size on TES.

INTRODUCTION

Thermal energy storage systems are used during periods of low cooling demand. To store the cooling energy, the chill storage media such as water, ice, or a phase change material are used. Thermal storage systems using latent heat of phase change material has the advantages of high storage density and heat retrieval at constant temperature during phase change. The storage application involves a 24-hour or alternatively, weekly or seasonal storage cycle depending on the system design requirements. It is proved that the cool storage technology is an effective means of shifting peak electrical loads as part of the strategy for energy management in buildings [1]. It is also considered as a useful tool to reduce the size of refrigeration machinery and air conditioning by means of spreading the daytime load over a 24-hour period. PCMs are used for various heat storage applications since 1800s, but they are recently used as cool storage media. Most of the PCMs for cool storage are inorganic salt hydrates and mixtures of salt hydrates. They are employed due to their high latent heats of transition, high densities and low cost. The experimental results using stearic acid as phase change material showed that the melting stability of PCM is better in the radial direction than in the axial direction. [2]. Ze-shao et al [3] introduced a new style cool storage scheme with high temperature water for air-conditioning. The capital cost of the cool storage system with paraffin waxes as phase change material indicates that the cool storage system not only saves energy and other operating costs but also saves a significant fraction of the initial capital costs

[4]. The manganese (II) nitrate hexahydrate, with melting point from 15°C to 25°C is used as a new PCM for the TES cooling system [5]. It was also found that the modulation of melting point and reduction of super cooling can be made by dissolving small amounts of salts in the material.

METHODOLOGY

To determine the maximum storage capacity of the eutectic mixture, this research uses sodium nitrate along with additives as suitable PCM, because of their desirable properties such as lesser degree of super cooling, high density, large heat of fusion and stability for repetition of melting and solidification. Additives such as nucleating agents, freeze point depressants and thickening agents are added along with sodium nitrate to attain the required temperature. Nucleating agent, borax is used to stabilize the refreeze temperature. The freeze point depressant, ethylene glycol is used to provide greater cooling capacity and the thickening agent, propylene is used to overcome separation and degrading problems. The eutectic mixture is considered in nodular form for this analysis and the heat storage capacities of PCM nodules of different size are compared.

SYSTEM DESCRIPTION

An air-conditioning system is simulated with a chiller of capacity 1.7 kW and heater of 1.8 kW, connected to the storage system of height 0.52m, diameter 0.484m, and volume 0.0956 m^3 is shown in figure 1. The cold fluid (ethylene glycol) and hot fluid (hot water) are circulated to transfer stored thermal energy in the system. The coriolis flow meter and RTDs are to monitor the mass flow rate and temperature of cold and hot fluids respectively. To pump the cold and hot fluids into the storage tank, 1.5 kW capacity pumps are connected across the chiller and heater units. Cold fluid of -6°C from the chiller enters into the storage tank and absorb heat from the hot fluid and leaves at -3°C. Hot fluid at 15°C from heater rejects heat to the cold fluid and leaves at 11°C. In the storage tank 60% of volume is filled with spherical nodules, where the PCMs are packed and remaining 40 % is filled with a heat transfer fluid (ethylene glycol).

NUMERICAL ANALYSIS

The simulation is fast becoming a substitute to real time experimentation trends where the stakes involved are too high to implement. For this model the type of analysis is thermal transient non-linear. The material properties such as conductivity, density and enthalpy are specified for the PCM, ethylene glycol, and water .Figure 2 shows the volume of the tank considered for analysis. In the storage tank, 60 % of the volume is filled with spherical nodules and remaining 40 % is filled with heat transfer fluid. The fluid acts as medium to accommodate the expansion and compression of phase change material during heat transfer. The modelled storage tank is symmetrical about x-y plane; hence the heat transfer characteristics are similar in the plane. To reduce the complexity, a portion of the storage tank is considered for analysis.

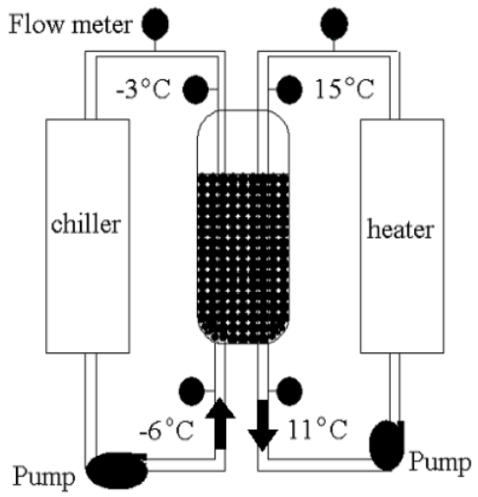

Figure 1. Experimental setup.

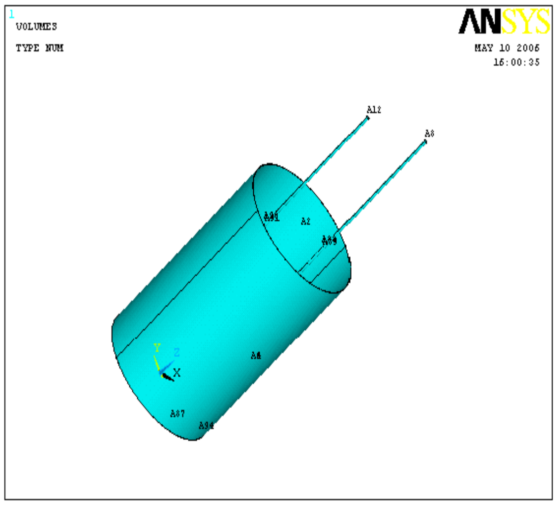

Figure 2. Volume of the tank considered for analysis.

RESULTS AND DISCUSSION

The characteristics of thermal energy storage system like heat transfer capacity of the nodules in the storage tank and the time required to attain steady state phase change temperatures are examined to predict the influence of nodule size. The various nodules considered are of diameters 77mm, 88mm, and 98 mm. Figure 3 shows the temperature profile of the storage tank with nodules of size 77mm. It shows that the temperature of the nodule of diameter 77mm and the heat transfer fluids are of uniform temperature of -3.8°C. To attain a required steady state temperature, the system requires duration of 3100 seconds, which is less than the available commercial eutectic salts.

The time required to obtain a steady temperature is shown in figure 4. It shows for the first 3100s the temperature of the PCM is gradually reduced and attains a steady state temperature. Further increase in cooling does not have any effect in PCM; hence it attains a complete phase change, i.e. it solidifies completely. Figure 5 shows the breaking point where the complete phase change occurs for the particular mixture of PCM. It shows that the temperature drops continuously until a complete phase change occurs.

Similarly the analysis is performed for the other two nodules of diameters 88 mm, 98 mm. The temperature contour plots for the nodule diameters 88 mm, 99mm are shown in figure 6 and figure 9 respectively. The time taken to attain complete phase change in nodule of size 88 mm is 12000 seconds as shown in figure 7 and figure 8. In the case of nodule of size 98 mm the time taken to solidify completely is 23000 seconds as shown figure 10 and figure 11.

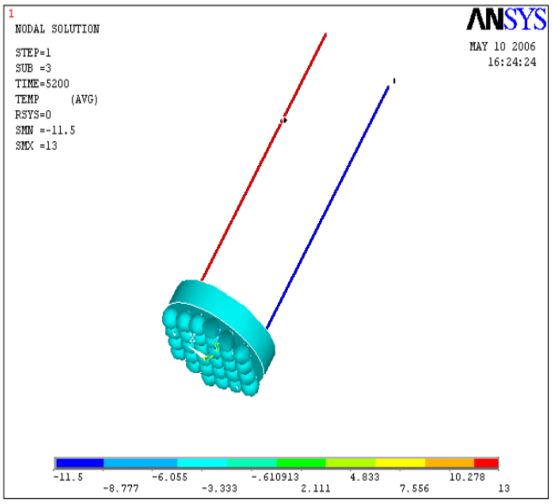

Figure 3. Temperature profile for the nodule of size 77mm.

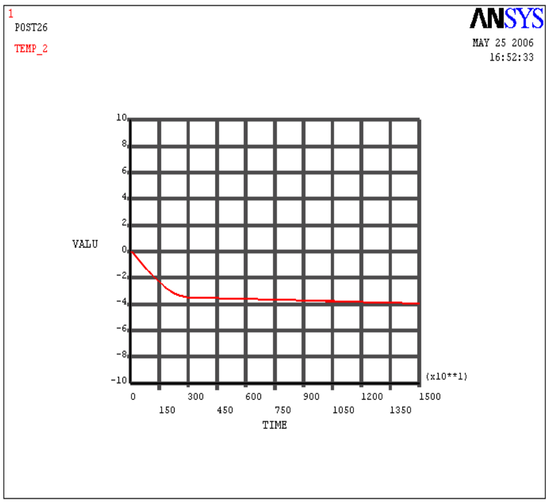

Figure 4. Steady temperature for the nodule of size 77mm.

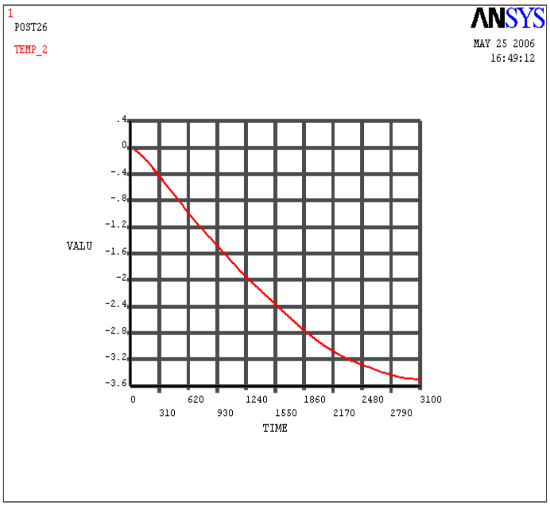

Figure 5. Phase change breaking point for the nodule of size 77mm.

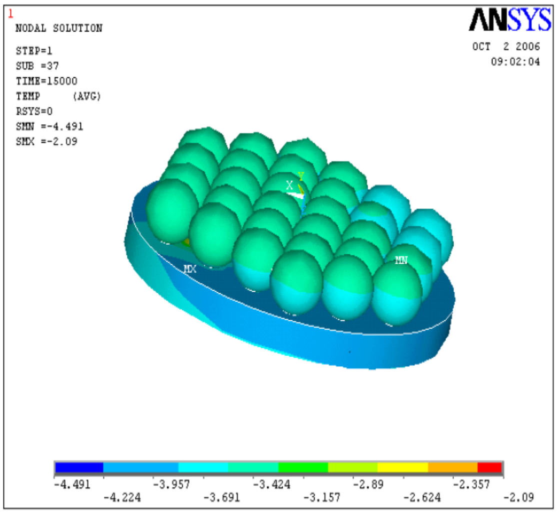

Figure 6. Temperature profile for the nodule of size 88mm.

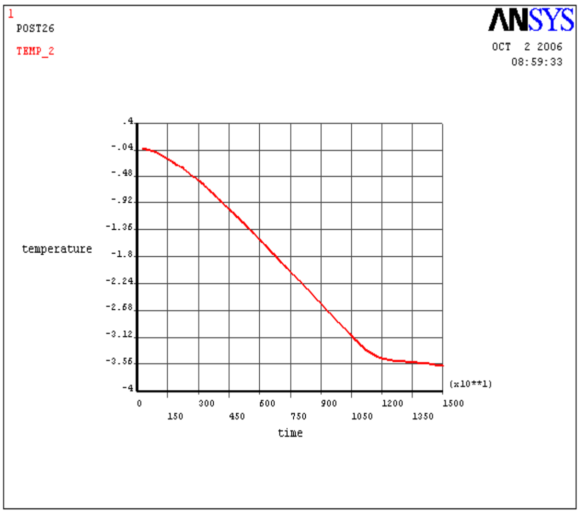

Figure 7. Steady temperature for the nodule of size 88mm

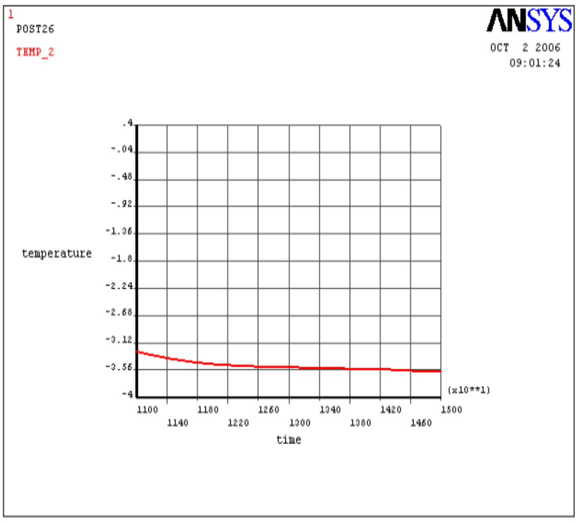

Figure 8.Phase change breaking point for the nodule of size 88mm

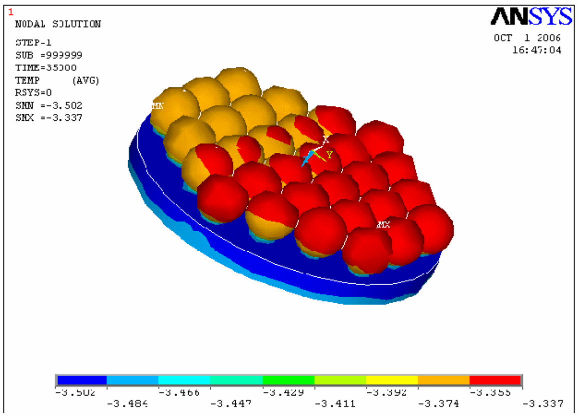

Figure 9. Temperature profile for the nodule of size 98mm.

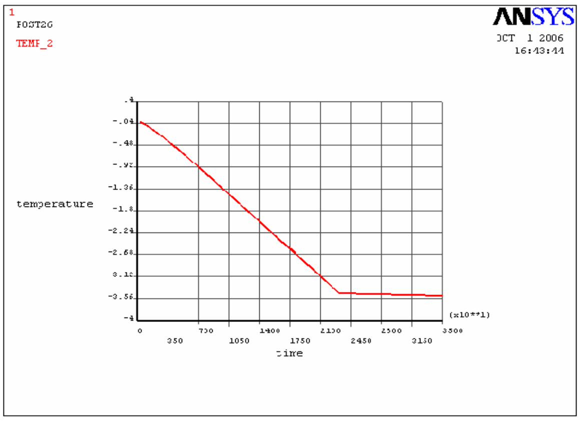

Figure 10.Steady temperature for the nodule of size 98mm.

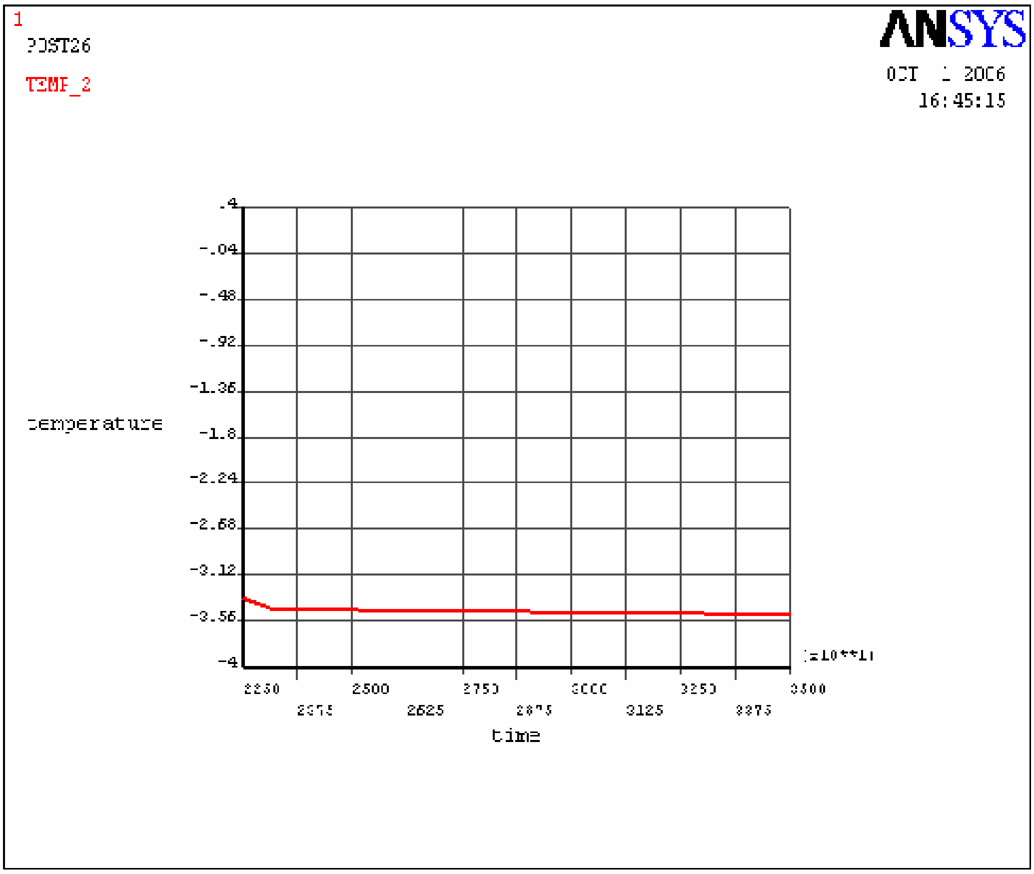

Figure 11. Phase change breaking point nodule of size 98mm.

CONCLUSIONS

The study on thermal energy systems depicts that to obtain a steady state temperature of -3.8°C, this eutectic mixture requires continuous cooling for 3100 seconds, 12000 seconds, and 23000 seconds for nodules of diameter 77 mm, 88 mm, and 98 mm respectively. The heat transfer capacity of nodules of diameter 77 mm, 88 mm and 98mm are 47.66 kWh, 45.369 kWh and 45.01 kWh respectively. Hence the results indicate that the reduction in nodule diameter leads to lesser time for attaining a steady state. It also indicates that heat transfer decreases with increasing nodule diameter.

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An Experimental Study for the Evaluation of the Environmental Performance by the Application of the Automated Venetian Blind

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SUMMARY

Venetian blind can have a major impact on building energy use and occupant's comfort. But manual or motorized Venetian blind has limitations in meeting occupant's needs and in reducing energy consumption. These limitations can be overcome by the automatic control of the Venetian blind. This study aims to analyze the control method of commercially used automated Venetian blind and to evaluate the environmental performance of this type of blind. The environmental performance of an automated Venetian blind was evaluated by a real-scale experiment and occupant's response in summer. The enhancement of environmental performance by the application of an automated Venetian blind was confirmed.

1. INTRODUCTION

Recently, cooling loads through the building envelope has greatly increased with the increase of glass area in office buildings in Korea. Windows provide occupants with daylight, direct sunlight, visual contact with the outside and a feeling of openness. While it is desirable to introduce sunlight for natural lighting over a given constant level, the radiation heat of the sun has to be determined to allow its entrance or not according to the weather conditions. The radiation heat from the sun reduces the heating load in the winter season but increases the cooling load during the summer season. Due to the characteristics of solar energy, which is comprised of light and heat, solar energy is not easy to control; that is, daylight introduction and excessive heat isolation have to be considered at the same time. In addition, excessive sunlight will cause glare problems in an office space, which are a major complaint by occupants. These glare problems must also be taken into consideration.
In office buildings, since the reduction of the opening ratio and the design of the windows orientation are restricted from the design perspective, Venetian blinds are generally used to control the incoming solar radiation. Venetian blinds are better than roll blinds because they can be controlled to slat angle as well as to occlusion index.
Venetian blinds can greatly affect building energy use and occupant comfort, so it is important to control Venetian blinds properly according to the change of the external weather condition for enhancement of indoor environment. But previous studies have shown that in reality, occupants rarely change the position of the manual or motorized Venetian blind [1] [2] [3] [4] [5] [6] [7]. The manual or motorized Venetian blind has limitations in meeting occupant's needs and in reducing energy consumption. To overcome these limitations, the Venetian blind must be controlled automatically. The use of the automated Venetian blind,

controlled automatically by sensors according to the external weather condition, would be more effective than the manual or motorized Venetian blind. During times of peak solar gain, such a blind can reduce cooling loads and overheating. Under cloudy conditions, or in winter, it can be withdrawn to allow daylight and useful solar gains to enter the building, so that the building can reduce its dependence on electric lighting and heating requirement. To achieve this condition, the blind should be properly controlled. Otherwise, unwanted solar gain may enter the building and increase the cooling load. Occupants may experience glare from the sun and be unable to operate the shading to alleviate it.

This study aims to analyze the control method of the commercially used automated Venetian blind and to evaluate its environmental performance in comparison to the manual or motorized Venetian blind. The environmental performance of the automated Venetian blind was evaluated by a real-scale experiment and occupant's response in summer. Finally, the potential of energy savings and comfort enhancement by use of the automatic control was confirmed.

2. BLIND TYPE AND OPERATION SURVEY

A blind can be categorized into manual, motorized and automated according to its control method. The manual blind is the simplest type of blind without a motorized device. It is controlled directly by the occupant and used generally in modern buildings. But the occupants control the manual blind only when they feel discomfort [8]. In fact, manual blind control has problems because occupants do not control the blind until they feel discomfort such as glare and solar radiation. The motorized blind that can be controlled by a remote controller, and in some buildings, computer are used to control motorized blinds. Though the motorized blind can be controlled easier than the manual blind, it is operated inefficiently in real situations. According to the operation survey of actual motorized blinds, though it is able to control the motorized blind in central control room, central control is not used by the occupants need and 82.6% of all motorized blinds do not move, as shown in Tables 1 and 2. Also, the occlusion index of the stopped blind is usually, almost opened (0 ~ 20%) or closed (80 ~ 100%), as shown in Table 3. An automated blind is automatically controlled by sensors according to the external weather condition to enhance environmental performance.

A motorized blind requires higher initial cost than a manual blind but does not produce better environmental performance. Therefore, the application of automated blind is necessary.

Table 1. Outline of survey building

City, County	Number of floors	Sky condition	
Seoul, Korea	22	Cleary sky(2 days)	Overcast sky(2 days)

Table 2. Frequency of blind operation. (South orientation)

Frequency	Number	Percentage(%)
0	213	82.6
1	31	12.0
2	14	5.4
Total	258	100.0

Table 3. Occlusion index of blind. (Frequency 0)

Occlusion index(%)	Number	Percentage(%)
0 ~ 20	207	39.8
21 ~ 79	141	27.1
80 ~ 100	172	33.1
Total	520	100.0

3. EVALUATION EXPERIMENT ON ENVIRONMEMTAL PERFORMANCE BY AUTOMATED BLIND

3.1 Outline

The effects of commercially used automated blind on thermal and visual environment performance and comfort are compared with those of conventional manually operated blinds. The product specification of an automated blind used in this experiment is presented in Tables 4 ~ 6. In the test room, located on the top floor of a building in Seoul National University, Korea, evaluation experiments were conducted to measure the energy saving effect and occupant comfort enhancement for the month of August 2006.
Two test rooms of dimensions 5.8m x 4.8m x 2.7m with the same architectural planning were used to reproduce the same conditions. Each room was equipped with an internal Venetian blind of same material and color, and the blind in one room was manually controlled and that in the other room automatically controlled, as shown in Figure 1 and 2. Various instruments were installed inside and outside the rooms to measure and analyze the various environments, as presented in Table 7.

Table 4. Product specification of the automated blind

Blind	Sensor	Operation mode
Aluminum(gray) Slat width : 50mm	Sun sensor (Outdoor vertical illuminance sensing)	Energy saving mode(see Table 5) Comfort mode(see Table 6)

Table 5. Operation sequence of energy saving mode.

Control time	Control conditions				Operation at ON		Operation at OFF
	ON		OFF				
	Exterior Illuminance	delay	Exterior Illuminance	delay	Occlusion index	Slat angle	
09:00 ~ 18:00	≥16 klux	3 min.	≤15 klux	15 min	100%	90°	Completely open

Table 6. Operation sequence of comfort mode.

Control time	Control conditions				Operation at ON		Operation at OFF
	ON		OFF				
	Exterior Illuminance	delay	Exterior Illuminance	delay	Occlusion index	Slat angle	
09:00 ~ 10:30	≥25 klux	3 min.	≤15 klux	15 min	100%	45°	Completely open
10:30 ~ 12:00	≥25 klux	3 min.	≤15 klux	15 min	100%	30°	
12:00 ~ 13:00	≥16 klux	3 min.	≤15 klux	15 min	100%	90°	
13:00 ~ 13:45	≥25 klux	3 min.	≤15 klux	15 min	100%	30°	
13:45 ~ 18:00	≥25 klux	3 min.	≤15 klux	15 min	100%	45°	

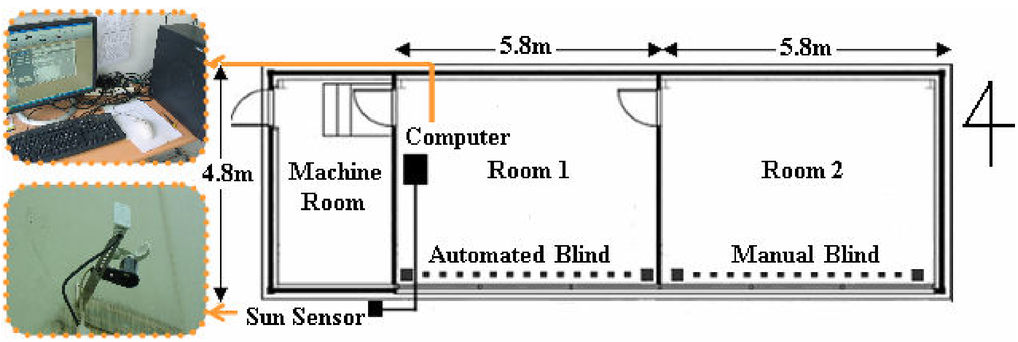


Figure 1. Plan view of test rooms.

Figure 2. External view of test rooms.

Table 7. Measuring equipments.

	Equipment	Position
Interior illuminance	Daylight factor meter, Lux meter	Room 1, Room 2
Room temperature	T-type thermocouple	Room 1, Room 2
Outdoor temperature	T-type thermocouple	Outside
Solar radiation	Irradiance meter	Roof(Outside)
Exterior illuminance	Exterior lux meter	Roof(Outside)

3.2 Method

In order to investigate the effect of the automated blind on the environmental performances of the rooms in the summer season, experiments were conducted for four cases, as presented in Table 8. In test room 1, the energy saving mode and comfort mode, which were controlled by an outdoor illuminance sensor, were selected. When in the energy saving mode, it was set up to shut out the solar radiation at the maximum level within the control range (see Table 5), and in the comfort mode, the slat angle was controlled to cut off direct daylight (although allowing some daylight to enter), in consideration of the working hours of the occupants and the comfort factor (see Table 6).

Table 8. Experiment cases.

Case	Test room 1	Test room 2	Air-conditioned	Remarks
1	Energy saving mode	Completely open	x	Evaluate energy consumption
2		Completely close		
3	Comfort mode	Completely open	○	Evaluate comfort
4		Motorized*		

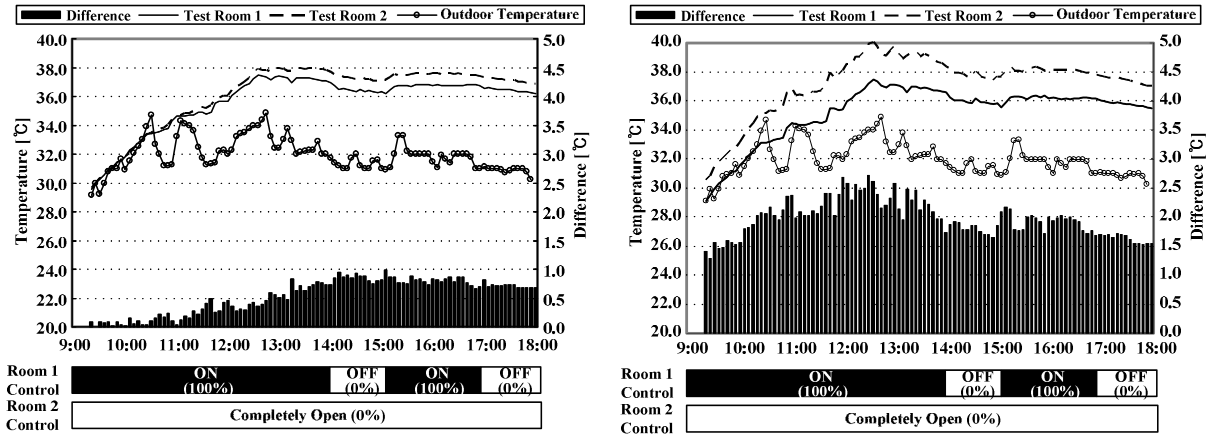
* Occlusion index 75%, Slat angle: 90°

For the set up of the compared blind, three model cases were selected, in which the blind was completely open (occlusion index: 0%), the blind was completely close (occlusion index: 100%, Slat angle: 90°), and the blind was motorized (occlusion index: 75%, Slat angle: 90°), respectively. The latter case represented the results of the blind operation survey. In cases 1 and 2, the characteristics of automatic control in terms of energy consumption in comparison with those of the extreme case of manual control were verified by comparing temperature and workplace illuminance. In cases 3 and 4, the conditions (with and without automatic control) were compared by PMV measurement and workplace illuminance, and a questionnaire was conducted four times per day(10:00, 12:00, 14:00, 16:00) to investigate their characteristics with regard to comfort. For each case, the experiments were conducted from 09:00 AM to 18:00 PM, without artificial lighting.

4. EXPERIMENT RESULT AND DISCUSSION

4.1 Case 1 : energy saving mode vs. completely open

Case 1 was conducted on August 12th. On this day, the sky was fairly clear but became rather cloudy from 13:00 ~ 14:00. The temperature of the interior zone of the test room is shown in Figure 3 a).
The temperature in test room 2 was higher than that of test room 1. Because solar radiation was effectively shut out in the energy saving mode, the temperature in test room 1 was lower than test room 2 by a maximum of 1.0℃. For the perimeter zone, which is directly subject to solar radiation, the temperature difference was greater. In Figure 3 b), the perimeter zone temperature of test room 1 was lower than that of test room 2 by a maximum of 2.7℃.



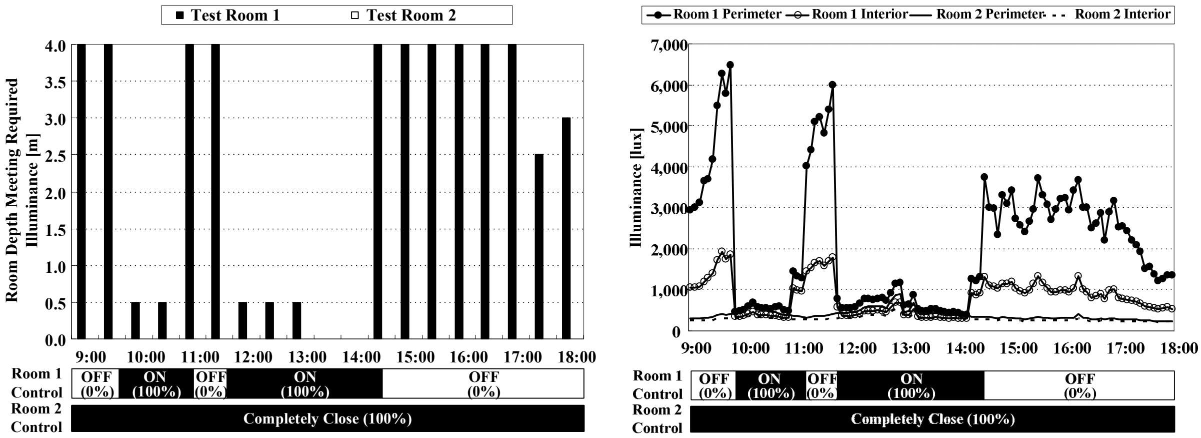
a) Interior zone b) Perimeter zone

Figure 3. Temperature profile of case 1.

In conclusion, the rise in room temperature was effectively reduced by the automated blind. In particular, in the perimeter zone, the automated blind effectively shut out solar radiation to reduce temperature rise. Especially, on clear summer day, the cooling load by the excessive solar radiation can be minimized through the appropriate automatic control of blind.

4.2 Case 2 : energy saving mode vs. completely close

Case 2 was conducted on August 16th. The sky was cloudy all day. There was little temperature difference between the two test rooms. Therefore, though the blind was fully opened a few times during the experimental period in response to the external weather condition, the cooling load of test room 1 was slightly higher than that of test room 2, but the lighting energy of test room 1 could be saved by introducing daylight.
The room depth meeting the required illuminance profile at the workplace in the two test rooms are shown in Figure 4. Compared to test room 2(completely close), which needed artificial lighting “on” whole day, test room 1 could satisfy the illuminance criteria(≥500lux) during most of the experiment period without artificial lighting because daylight was introduced. So the lighting energy of test room 2 was saved.
Consequently, the overall energy consumption, including cooling and lighting energy consumption, could be reduced by using the automated blind. In addition, the automated blind provided an open field of vision and a view of the outside scenery.



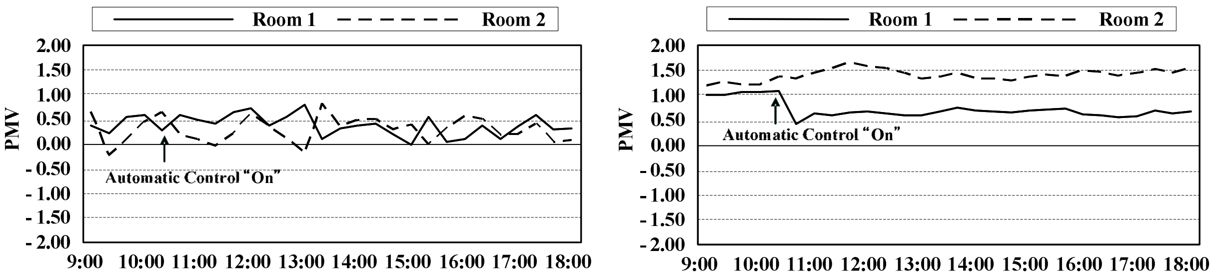
a) Room depth within satisfied range b) Workplace illuminance profile

Figure 4. Analysis of the satisfied range and measured workplace illuminance of case 2.

4.3 Case 3 : comfort mode vs. completely open

Case 3 was conducted in August. The sky was clear.

A representative comfort indicator PMV was measured in two test rooms to determine the degrees of comfort of the two test rooms, as shown in Figure 5. In the interior zone, since both rooms were air-conditioned continuously and the PMV meter was installed away from the influence of solar radiation, the difference in the measured values was insignificant, and largely satisfied with the comfort criteria (-0.5 < PMV < +0.5) of the ISO. However, in the perimeter zone, which is directly subject to the influence of solar radiation, the PMV measured in test room 2 remained constant between 1.2 and 1.6, which indicated a discomfort environment. However, the PMV in test room 1 fell by about 0.5 at 10:30 when the blind began to be controlled, and as a result, enhanced the degree of comfort.

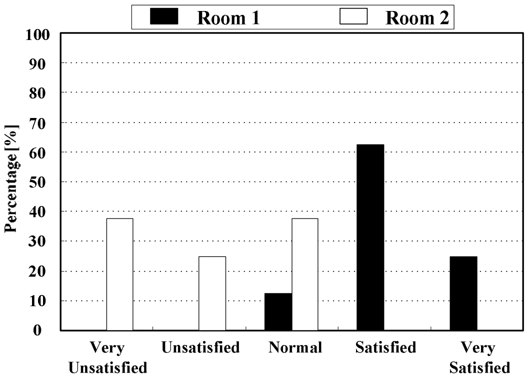


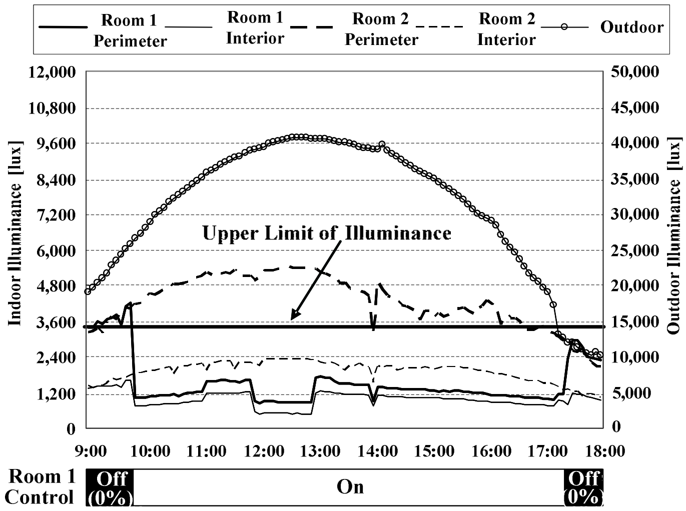
a) Interior zone b) Perimeter zone

Figure 5. PMV profile of case 3.

With respect to the overall operation of the blind, 87.5% of the subjects in test room 1 were satisfied, while 62.5% of those in test room 2 were dissatisfied due to the rise in temperature resulting from solar radiation as shown in Figure 6 a). With respect to workplace illuminance, both test rooms showed illuminance of higher than 500lux during entire experiment period, but in test room 2, the illuminance exceeded the recommended upper limit (3,340lux [9]) during most of the experiment period as shown in Figure 6 b). Test room 1, which was operated in the comfort mode, was maintained within the recommended value because slat angle was controlled to cut off direct sunlight but allow daylight to flow in.

Since the experiment had been conducted during a period of hot weather and clear sky, solar radiation must have had great impact on the result. Consequently, the degree of direct radiation and direct light should be appropriately controlled with a blind to protect the occupants from glare and to improve comfort.





a) Overall satisfaction b) Illuminance profile

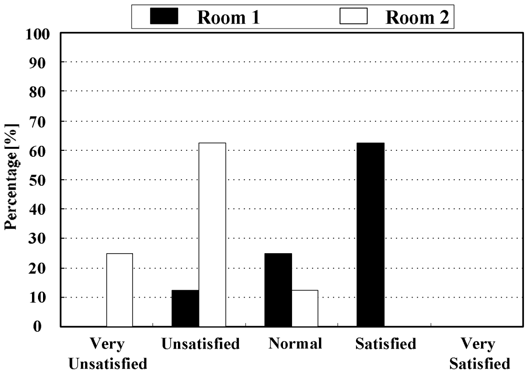
Figure 6. Subject response and illuminance profile at workplace of case 3.

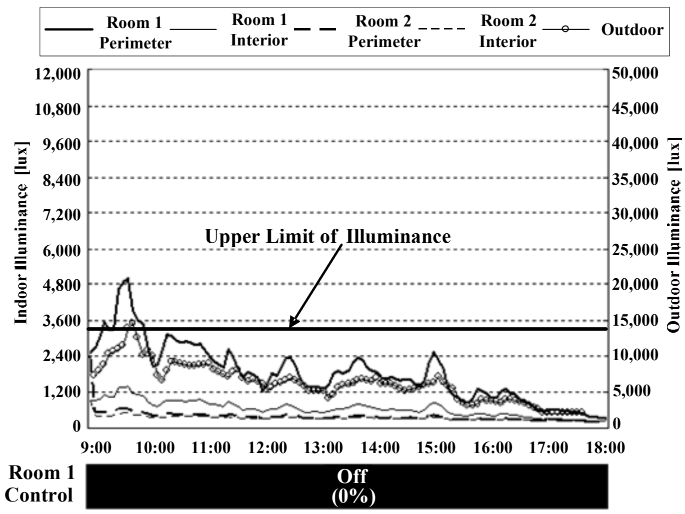
4.4 Case 4 : comfort mode vs. motorized

The experiment was conducted on August 29th, a cloudy day with rainfall starting from the late afternoon. Motorized blind condition was determined based on the operation survey results. Since the weather was cloudy and no significant influence from solar radiation was likely to be felt, no significant difference of PMV was observed due to the constant air conditioning, and the ISO comfort criteria were largely satisfied in both interior and perimeter zones.

With respect to the overall operation of the blind, subjects in test room 1 were satisfied since optimal illuminance could be obtained by automatic control, while subjects in test room 2 were dissatisfied with the brightness, since the blind kept shutting out the daylight greatly during the cloudy weather as shown in Figure 7 a).

With respect to workplace illuminance, test room 1 showed illuminance of higher than 500lux during most of the experimental period , but test room 2 under 500lux, as shown in Figure 7 b). Test room 1 was able to introduce more daylight by automatic control of the blind reflecting the outdoor weather condition.





a) Overall satisfaction b) Illuminance profile

Figure 7. Subject response and illuminance profile at workplace of case 4.

Based on the questionnaire and measurements, visual comfort could be improved without additional lighting by introducing daylight selectively with the automated blind in accordance with the prevailing weather conditions. Additionally, it was judged that an open field of vision could be provided by automatic control.

5. CONCLUSION

In this study, control methods of commercially used automated Venetian blind were analyzed and improvement effects of environmental performance were proven through the experiments on energy saving and comfort.

In the energy saving mode, cooling load was reduced more than that with the fully opened window because blind blocked solar radiation according to established conditions under the clear sky condition. In overcast sky, the blind allows daylight to enter according to established conditions and so it can reduce the lighting energy and minimize the cooling load increase, simultaneously. In the energy saving mode, in which the blind is only fully opened or closed, the modification of the occlusion index and slat angle is not considered. Therefore, we need a control method that can consider the lighting and cooling energy saving simultaneously by proper control of the occlusion index and slat angle.

In the comfort mode, the slat angle is controlled to cut off direct sunlight and this reduces the discomfort from excessive solar radiation and direct sunlight. Also, in this mode, daylight can be introduced to supply a feeling of openness to the occupant. Because modification of the occlusion index is not considered and modification of slat angle is limited to two times per day in the comfort mode, a more detailed and delicate control method which can consider the control of occlusion index is necessary.

The control method discussed in this paper is operated only through external weather condition sensing. But the purpose of blind control is to improve indoor environment, so a more effective and enhanced control method which can reflect indoor environment needs to be developed by sensing internal factors such as workplace illuminance, indoor temperature etc.

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The Influence Of Window Type And Orientation On Energy-Saving In Buildings – Application To A Single Family Dwelling

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ABSTRACT

The trend to reduce energy losses through enclosures (walls and windows) is an increasingly important part of improving the energy efficiency of a building. Adequate window design and orientation can reduce the heating and cooling requirements of a building.

An optimization process was undertaken for a single-zone building envelope, based on the climate of the Basque Country and using a parametric study of annual heating requirements, according to glazing type and orientation. Two software were used in the analysis: WINDOW 5.2 for window thermal performance and TRNSYS 16 for building thermal simulation.

The results of the study were applied to the case of a single-family dwelling to predict energy savings for heating, based on the recommended window type for each orientation.

INTRODUCTION

Windows play a key role in the energy efficiency of both residential and commercial buildings. At least one-fourth of domestic heating requirements in OECD countries (Muneer T. et al, 2000) are considered to be the result of energy losses through windows. In recent years, major efforts have been made to improve the thermal behavior of windows using various technologies, such as low-emissivity (low-E) coatings, inert gas fill, insulating edge spacers, low-conductivity frames, etc. This approach has led to windows with a thermal transmittance as good as a wall.

Energy consumption in buildings depends largely on the type of building, whether residential or commercial. In the latter case, 90% of the heating and cooling is linked to losses and gains through the façade, particularly through windows, which are the most sensitive to heat flows. Energy requirements are a function of various factors such as climate, building location, season of the year, and operational and functional conditions.

This study analyzes the influence of window type on heating requirements for the climate of the Basque Country. To accomplish the objective, two simulation tools were used: WINDOW 5.2 to determine the thermal performance of window systems and TRNSYS 16 to simulate the heating requirements of the building envelope.

NOMENCLATURE	
A	Area (m^2)
c_p	Specific heat (J/kg-K)
g	Solar heat gain coefficient
R	Thermal resistance (m^2K/W)
R_{fsol}	Solar reflectance of the exterior-facing surface of the glazing system
R_{bsol}	Solar reflectance of the interior-facing surface of the glazing system
R_{fvis}	Visible reflectance of the exterior-facing surface of the glazing system
R_{bvis}	Visible reflectance of the interior-facing surface of the glazing system
T_{sol}	Solar transmittance
T_{vis}	Visible transmittance
U	Thermal transmittance or U-factor (W/m^2K)
α_1	Absorptance of the first pane in the glazing system, counting from the exterior-facing surface
α_2	Absorptance of the second pane in the glazing system, counting from the exterior-facing surface
α_3	Absorptance of the third pane in the glazing system, counting from the exterior-facing surface
ρ	Density (kg/m^3)

DESCRIPTION OF BUILDING ENVELOPE AND STANDARD WINDOWS

The building envelope considered is similar to the one described in ANSI/ASHRAE Standard 140/2004, consisting of a single-zone enclosure of 8.0 m x 6.0 m x 2.7 m, fitted with two windows of 3.0 m x 2.4 m, as shown in Figure 1. The thermophysical properties of the materials used for the model are listed in Table 1, which correspond to the configuration of the high-mass enclosure described in the ANSI/ASHRAE Standard 140/2004 mentioned above.

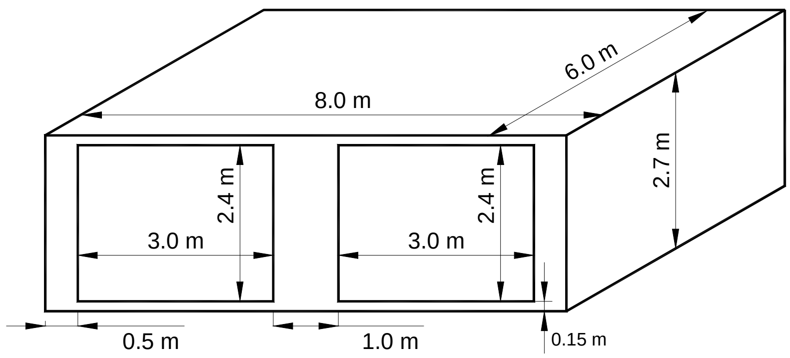


Figure 1. Simulated building envelope for determination of heating requirements

Tables 2 to 4 show the characteristics of the various types of windows and glazing used in the study. The coating position is indicated by numbering the pane surfaces from exterior-facing to interior-facing. The effect of varying the glazing was analyzed, while using the same PVC frame for all windows. Double-pane glazing (Type 1) was taken as a reference. The window Type 2 is a double-pane glazing with a low-E coating and a high solar gain (0.72). Type 3 is a double-pane glazing with low-E coating and a moderate solar gain(0.61) and Type 4 is a double glazing with low-E coating and a low solar gain (0.40), spectrally selective. When compared to tinted glass, this glazing system has the property of greater visible transmittance (in this case, 0.70). A common measure of the efficiency of spectrally selective glass is given by the light-to-solar ratio. This is the visible light transmission divided by the solar heat gain coefficient for the glazing system.(in this case, 1.76, with a maximum possible ratio of 2).

Low-E coatings reduce radiative heat transfer. If a lower conductive-convective heat transfer in the air-gap is also desired, then gases with a lower thermal conductivity than air can be used, as occurs with windows Type 5. If heat flow through the system is to be further decreased, triple glazing with two low-E coatings and a low-conductance gas can be used; this is the configuration chosen for the window Type 6 .

Table 1. Physical properties of the enclosure

	Element	**K (w/mK)**	**Thickness (m)**	**U (W/m^2K)**	**R (m^2K/W)**	**ρ (kg/m^3)**	**c_p (J/kgK)**
Exterior wall	Int. surface coeff.			8.30	0.12		
	Concrete block	0.510	0.100	5.10	0.196	1400	1000
	Foam insulation	0.040	0.0615	0.65	1.54	10	1400
	Wood	0.140	0.009	15.56	0.06	530	900
	Ext. surface coeff.			29.30	0.03		
Floor	Int. surface coeff.			8.29	0.12		
	Concrete slab	1,130	0,080	14.12	0.07	1400	1000
	Insulation	0.040	1.007	0.04	25.17		
Ceiling	Int. surface coeff.			8.29	0.12		
	Plasterboard	0.160	0.010	16.00	0.06	950	840
	Fiberglass	0.040	0.118	0.36	2.79	12	840
	Roof deck	0.140	0.019	7.37	0.14	530	900

Summary		
Element	*Area (m^2)*	*A.U. (W/K)*
Wall	63.600	32.58
Floor	48.000	1.89
Ceiling	48.000	15.25

Table 2. Types of glazings

	Number of panes	**Glass/air-gap thickness (mm)**	**Coating position**	**Coating**	**Gas fill**
Type 1	Double	3.9/12.7/3.9		None	Air
Type 2	Double	3.9/12.7/3.9	Surface 3	Low-E, 0.155	Air
Type 3	Double	3.9/12.7/3.8	Surface 3	Low-E, 0.066	Air
Type 4	Double	3.8/12.7/3.9	Surface 2	Low-E, 0.027	Air
Type 5	Double	3.9/12.7/3.8	Surface 3	Low-E, 0.066	Argon
Type 6	Triple	3.8/12.7/3.0/12.7/3.8	Surface 2, Surface 5	Low-E, 0.066	Argon

Table 3. Visible and solar energy parameters of the window

Types of windows	U (W/m^2K)	g	Tvis
Type 1	2.757	0.712	0.729
Type 2	1.921	0.660	0.667
Type 3	1.749	0.558	0.710
Type 4	1.666	0.370	0.630
Type 5	1.467	0.559	0.710
Type 6	0.974	0.447	0.627

Table 4. Optical properties of glazing system for a normal incident angle

	Tsol	Rfsol	Rbsol	Tvis	Rfvis	Rbvis	α1	α2	α3	g
Type 1	0.720	0.128	0.128	0.812	0.145	0.145	0.086	0.066		0.780
Type2	0.579	0.160	0.141	0.742	0.172	0.158	0.089	0.172		0.722
Type 3	0.525	0.277	0.282	0.791	0.115	0.115	0.108	0.090		0.609
Type 4	0.350	0.336	0.381	0.702	0.097	0.112	0.295	0.018		0.399
Type 5	0.525	0.277	0.282	0.791	0.115	0.115	0.108	0.090		0.610
Type 6	0.394	0.324	0.324	0.699	0.144	0.144	0.163	0.062	0.058	0.485

BUILDING ENVELOPE MODELING AND SIMULATION TOOL

The energy analysis of the building envelope was performed using TRNSYS 16, a transient simulation software of thermal systems created by the Solar Energy Laboratory of the University of Wisconsin. The TYPE 56 multi-zone model was used for both the model envelope and the single-family dwelling.

The most important heat flows considered in this TYPE are a long-wave radiative exchange with the exterior environment, a convective heat exchange and a fraction of absorbed solar radiation. There is also a transient heat exchange through the wall layers, modeled using transfer functions (Mitalas and Stephenson, 1971). Long-wave radiation heat exchange between the interior surfaces of the building envelope are estimated using the star network method implemented by Seem (1987).

The TRNSYS software allows importing a file with the optical data and thermal and solar properties of windows from the WINDOW (LBNL) software. The WINDOW calculation procedure (Finlayson et al., 1993) used to obtain an overall window property is based on computing an area-weighted average of the properties of each component of the window (ISO 15099). The total heat flow through a glazing system can be divided into the heat loss flow depending only on temperature difference and the heat flow depending on the intensity of the short-wave radiation.

SIMULATION OF MODEL ENVELOPE

The simulation was performed using climatic data from two Basque cities: Bilbao (43.30º N, 2.93º W) and Vitoria (42.51º N, 2.40º W). The climate of Bilbao is a temperate oceanic climate, whereas Vitoria has a temperate continental climate. The mean annual ambient temperature is 14.3ºC for Bilbao and 11.2ºC for Vitoria, whereas the mean annual irradiance of global horizontal radiation is 146 W/m^2 and 164 W/m^2, respectively. Since the TRNSYS building model requires hourly data to estimate energy requirements, Meteonorm 5.1 (Meteotest, Edition 2003) was used to generate files with typical meteorological year data for the cities under consideration.

The building envelope operating conditions described in ANSI/ASHRAE Standard 140-2004 were used, i.e. continual infiltration throughout the year of 0.5 air changes per hour, an internal gain of constant sensitive heat of 200 W, with a 60% radiant fraction and 40% convective fraction. A temperature setting of 20ºC for heating was used as a control strategy to estimate energy requirements.

Based on the results obtained, an analysis was then carried out to analyze the sensitivity of annual energy requirements to the following factors: type of glazing and orientation. The effect of window orientation was analyzed by turning the building envelope in steps of 90º, such that the only wall with windows was oriented toward the north, south, east and west. The heating season is considered to be from October to May. The heating requirements for the cases studied according to orientation are shown in Tables 5 and 6.

Table 5. Heating requirements according to orientation and window type (kW.h) for Bilbao

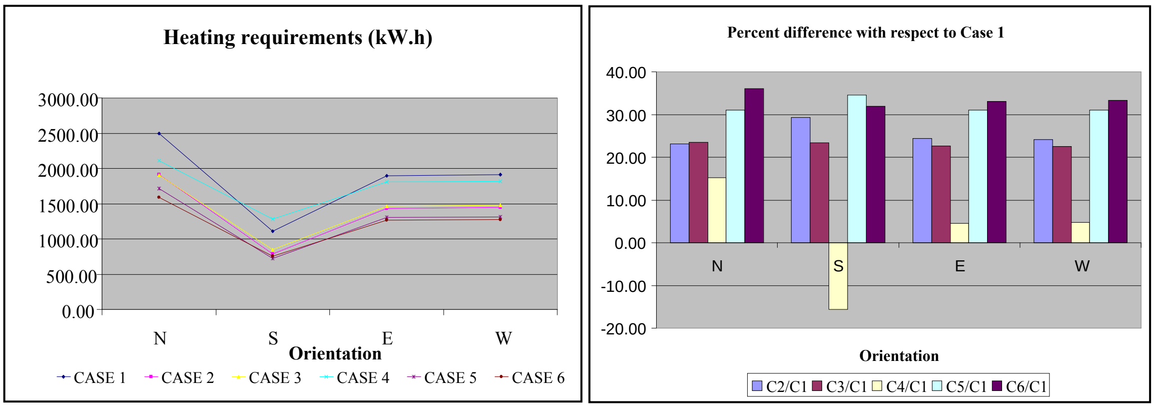
	Type 1	**Type 2**	**Type 3**	**Type 4**	**Type 5**	**Type 6**
N	2493.28	1915.78	1908.07	2112.7	1718.94	1594.17
S	1109.83	783.95	850.7	1283.1	726.40	755.07
E	1898.60	1435.11	1468.2	1812.7	1309.08	1270.85
O	1910.06	1449.53	1480.95	1819.6	1315.58	1272.89

Table 6. Heating requirements according to orientation and window type (kW.h) for Vitoria

	Type 1	**Type 2**	**Type 3**	**Type 4**	**Type 5**	**Type 6**
N	4026.20	3202.44	3196.80	3466.45	2893.81	2696.41
S	1904.77	1346.13	1485.83	2234.19	1274.49	1353.61
E	3072.65	2376.32	2450.00	2934.62	2206.13	2152.26
O	3117.02	2404.50	2472.00	2954.25	2227.40	2158.40

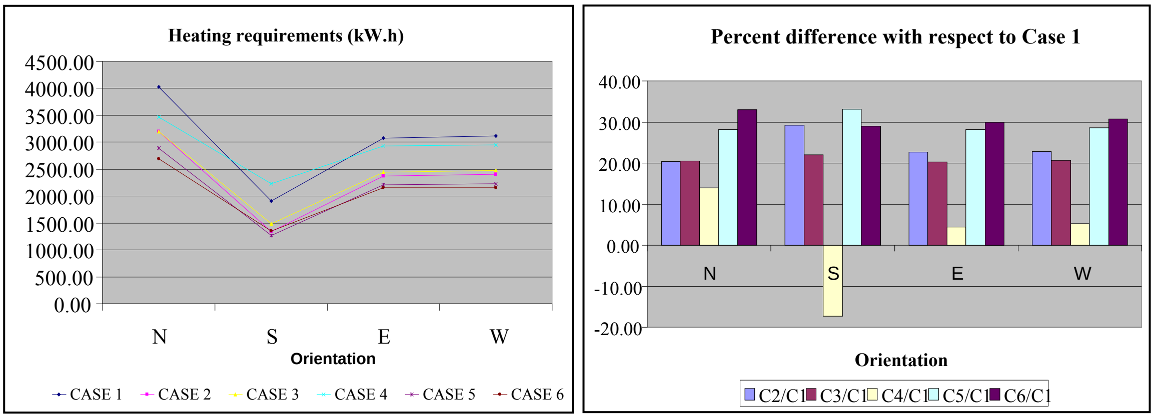
Type 1, which corresponds to an uncoated double-pane glazing, was taken as the reference case, and the heating energy savings obtained by replacing this by other glazing types were then analyzed. The conclusions obtained for the Basque cities were similar, since their climates are not much different. Figures 2 and 3 show the results obtained for Bilbao and Vitoria, respectively.

When replacing the reference glazing with low-E double glazing Type 2, the heating requirement decreases in all directions, particularly the southern direction, with a 29% decrease for Bilbao. Therefore, the glazing Type 2 is appropriate for the climatic zone of the Basque Country. In the southern direction, this pane works well in winter, since it is low-E glass with high solar heat gain. Its high longwave infrared reflectivity bounces the thermal radiation, which originates in the space, back into the space. This also true for glazing Type 3.



(a) (b)

Figure 2. For Bilbao: (a) heating consumption for each window type, (b) % differences with respect to Type 1.



(a) (b)

Figure 3. Vitoria: (a) heating consumption for each window type, (b) % differences with respect to Type 1.

COMMENTS

If low-E double glazing Type 3 is used, the heating requirements are lower in all directions considered, making this an appealing option. However, in the southern direction, the effect is less noticeable than with glazing Type 2, because Type 3 has a moderate solar gain (0.61) with a lower solar transmittance (0.53). For both Bilbao and Vitoria, the heating requirements are lower with windows Type 2 than Type 3, except on the northern side. Therefore, glazings Type 2 are recommended for all orientations, and glazings Type 3 are strongly recommended for the northern direction.

If a double-pane glazing is replaced by a spectrally selective glazing (Type 4), somewhat higher heating requirements are obtained on the southern direction and the savings are minimal on the other sides, since the solar heat gain (solar factor of 0.40) is lower. This type of glazing is adequate, particularly in climates where cooling is required. Spectrally selective coatings, which are not only highly reflective in the far infrared, but also in the near, are not recommended for these locations, which require heating during part of the year.

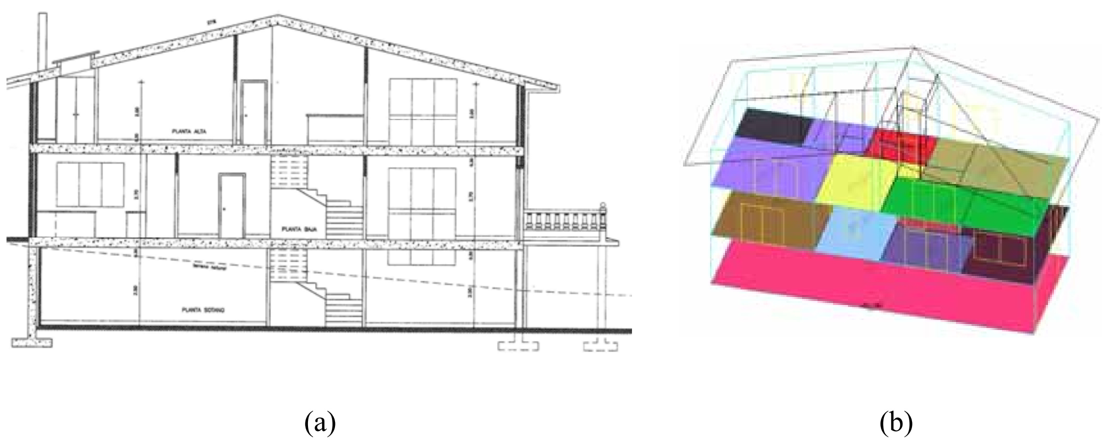
The glazing Type 5 is a version of glazing Type 3 in which the air gap is replaced by a gas of a lower conductivity. In addition to reducing radiation heat exchange, this approach reduces heat transfer by conduction-convection with the low-E coating. This type of glazing results in considerably lower heating requirements than those achieved with uncoated double glazing and similar to those obtained with triple glazing with low-E coatings. In particular, 31% savings is achieved for Bilbao in the northern, eastern and western directions, and 34% in the southern direction.

Triple glazing Type 6 represents a good option from the energy point of view, since it decreases the heating requirements considerably, achieving 36% savings in the northern direction and 32% in the southern for Bilbao for the model envelope.

ENERGY SAVINGS IN A SINGLE-FAMILY DWELLING

A heating consumption study of a single-family dwelling located in Bilbao was conducted by varying the glazing type. Figure 4 shows the single-family dwelling that was simulated. The window with glazing Type 1 was taken as a reference. Considering a mean system efficiency of 70%, the heating requirements were determined by performing the following simulations: with windows Type 1, with windows type 3 for northern orientation and type 2 for all other orientations and with windows Type 5 for all orientations

The results are shown in Table 7. The savings obtained when replacing the reference window with others are lower than when doing so with the ASHRAE envelope. This is primarily due to the fact that the model envelope has large windows and its walls are better insulated. In addition, whereas in the physical model, the effect of glazing surfaces has been analyzed for each orientation separately, in the actual case, a combination of glazing areas in the various orientations was used. Replacing Type 1 by a combination of Type 2 and 3 according to the orientation, 9.1% savings in heating requirements were obtained. When Type 5 was used, similar savings were achieved.



(a) (b)

Figure 4. (a) Cross-section of the simulated home, (b) thermal areas defined for Trnsys

Table 7. Results obtained when varying window types in a single-dwelling home.

	Heating consumption, kW.h	**%**
Type 1	16984.04	
Type 2/3	15440.26	9.09
Type 5	15432.61	9.13

CONCLUSIONS

Double glazings Type 4 are not interesting from the heating viewpoint. In south-facing windows, there is a loss with respect to the glazing taken as the reference. In other directions, the savings that would be produced are insignificant compared to other low-E windows.

Cases of interest for the climate of the Basque Country are glazings Type 2 and 3, e.g., low-E glass of 0.155 and 0.066, with high and moderate solar gain, respectively. With these panes, the heating requirements decrease for all orientations, making them truly of interest for our climate conditions. In the southern, eastern and western directions, glazing Type 2 is prefered, whereas Type 3 is prefered for the northern direction, due to higher longwave reflectivity. Therefore, these types of glazing are suitable for residential buildings. In this study, 9% savings was obtained by using a combination of glazings Type 2 and 3.

Type 5 is similar to Type 3, in which the air-gap has been replaced with argon, thus decreasing the conductive-convective exchange. This represents a slight improvement over Types 2 and 3, but may be important in the case of large windows, depending also on the type of enclosure.

The application of the results to the actual case confirms the trends observed with the model envelope. The extent of percent savings will depend on the specific characteristics of the building, in particular, the relative area of glazing surface and the inertia and insulation of the enclosure.

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Natural Convection Heat Transfer in a Saltbox Roof with Eave in Winter Day Conditions

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SUMMARY

In this study, a numerical analysis has been performed to examine the natural convection heat transfer and flow field inside a saltbox roof with eave in winter day conditions. This analysis is important for applications since it shows the effective parameters on natural convection heat transfer. The governing equations of natural convection in streamfunction-vorticity form were solved using central difference method to obtain flow and temperature fields inside the roof. Also, the Successive Under Relaxation (SUR) technique was used to solve linear algebraic equations. Results are presented by streamlines, isotherms and local and mean Nusselt number by using effective parameters as Rayleigh number, aspect ratio of the roof, and length of eave. The results indicated that both eave length and aspect ratio can be used as control parameter for heat transfer.

Keywords: *natural convection, saltbox roof, eave*

INTRODUCTION

Natural convection is formed in some parts of buildings due to temperature difference between cold and hot surfaces. The roof is an important part of the building because it protects the building from the environmental effects. Natural convection heat transfer occurred inside the roof due to temperature difference which depends on the climate such as winter or summer.

Natural convection heat transfer analyzed in differentially heated square or rectangular enclosures in earlier studies due to simplicity of numerical solution [1]. However, the number of study about partially heated enclosures is very limited and they investigated mostly for square cross sectional geometries [2-4].

Researchers the natural convection in roofs using different numerical techniques for triangular cross-sectioned roofs (Gable roofs) were investigated by different researchers. Asan and Namli [5] investigated the laminar natural convection heat transfer inside the triangular roof with different aspect ratio in winter day boundary conditions. They observed that, both aspect ratio and Rayleigh number affect the heat transfer inside the roof. Tzeng et al. [6], Akinsete and Coleman [7], Holtzman et al. [8] and, recently, Ridouane et al. [9] investigated the laminar natural convection inside the triangular cross-section enclosure which is applicable for roof geometry of building at different constant temperature or constant heat flux boundary conditions. On the other hand, Varol et al. [10,11] investigated the natural convection in different roof shapes including Gambrel and Saltbox roofs. They observed that natural convection flow fields strongly depend on the geometrical shape and thermal boundary conditions of the roof. Moukalled and Acharya [12] analyzed the natural convection heat transfer in a trapezoidal roof with baffles that located at the top and bottom of the wall. They indicated that baffles are effective parameters on thermal field inside the roof.

The main purpose of the present study is to analyze the flow and thermal fields inside the saltbox roof with eave in winter day conditions. According to knowledge of the authors and literature review given above, laminar natural convection heat transfer has not been investigated yet for partially heated roofs. Thus, the present study will be the first attempt in that area and it will help the designers and constructers from the energy saving point of view.

The photo and physical model of the saltbox roof with eave and the boundary conditions is shown in Figure 1 with coordinate system. It is considered that the inclined surface and eave have constant cold temperature. Also, room ceiling (bottom surface of the roof) has constant hot temperature to simulate winter day boundary conditions. The vertical boundary has adiabatic with height, H. Thus, total length of the bottom of roof (L) can be calculated by adding the length of room ceiling (B) and the length of eave (E). An aspect ratio can be defined as the ratio of the height of the roof to the length of bottom wall (AR=H/L) which can be changed according to climatical conditions.

Figure 1a. Photo of saltbox roof from Ref. [14].

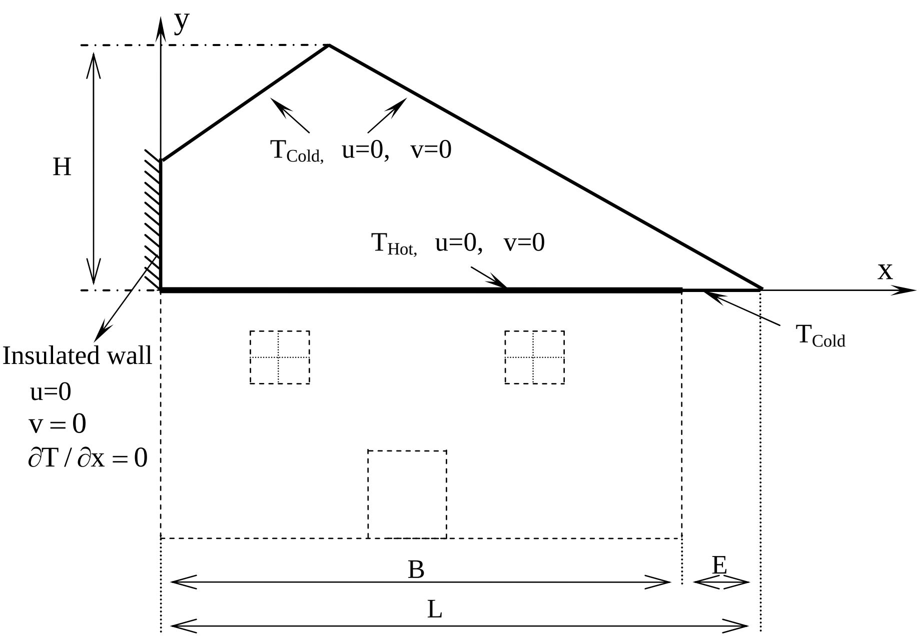


Figure 1b. Physical model

GOVERNING EQUATIONS AND NUMERICAL SOLUTIONS

The governing equations of natural convection (Eqs. 1-3) are written in streamfunction-vorticity form for laminar regime in two-dimensional form for steady, incompressible, and Newtonian fluid with Boussinesq approximation. It is assumed that radiation heat exchange is negligible according to other modes of heat transfer and the gravity acts in vertical direction.

$$-\Omega = \frac{\partial^2 \Psi}{\partial X^2} + \frac{\partial^2 \Psi}{\partial Y^2} \tag{1}$$

$$\frac{\partial^2 \Omega}{\partial X^2} + \frac{\partial^2 \Omega}{\partial Y^2} = \frac{1}{Pr}\left(\frac{\partial \Psi}{\partial Y}\frac{\partial \Omega}{\partial X} - \frac{\partial \Psi}{\partial X}\frac{\partial \Omega}{\partial Y}\right) - Ra\left(\frac{\partial \theta}{\partial X}\right) \tag{2}$$

$$\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} = \frac{\partial \Psi}{\partial Y}\frac{\partial \theta}{\partial X} - \frac{\partial \Psi}{\partial X}\frac{\partial \theta}{\partial Y} \tag{3}$$

The employed non-dimensional variables are given as

$$X = \frac{x}{L},\quad Y = \frac{y}{L},\quad \Psi = \frac{\psi\,Pr}{\upsilon},\quad \Omega = \frac{\omega (L)^2\,Pr}{\upsilon},\quad \theta = \frac{T - T_{cold}}{T_{hot} - T_{cold}}, \tag{4}$$

$$u = \frac{\partial \psi}{\partial y},\quad v = -\frac{\partial \psi}{\partial x},\quad \omega = \left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right),\ Ra = \frac{\beta g (T_{hot} - T_{cold}) L^3\,Pr}{\upsilon^2},\ Pr = \frac{\upsilon}{\alpha}. \tag{5}$$

Boundary conditions for the considered model are depicted on the physical model (Figure 1(b)). In this model, boundary conditions as

On all solid walls, $u = v = 0$

On the adiabatic wall, $\frac{\partial T}{\partial x} = 0$

On the bottom wall, $0 < x < B$, $T=T_{Hot}$, $B \le x \le L$, $T= T_{Cold}$

On the inclined wall, $T= T_{Cold}$

Local and mean Nusselt numbers are calculated via Eq. 6 a and b, respectively.

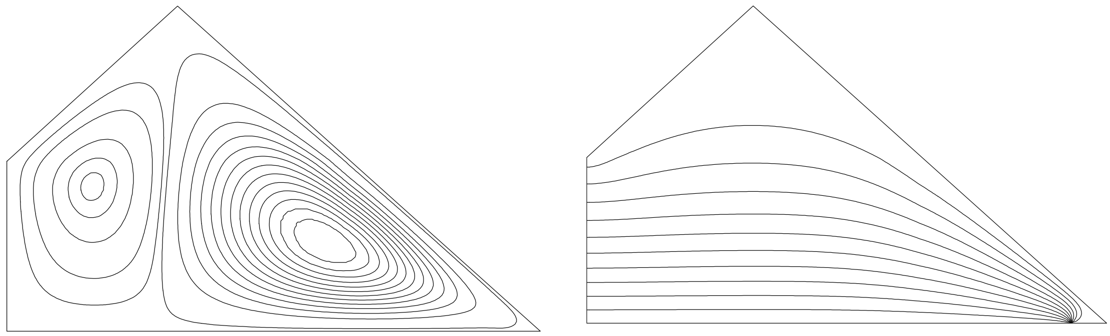
$$Nu_x = \left(-\frac{\partial \theta}{\partial Y}\right)_{Y=0}, \qquad Nu = \int_0^B Nu_x\,dx \tag{6 a.b}$$

Governing equations in streamline-vorticity form (Eqs. 1-3) are solved through finite difference method. Algebraic equations are obtained via Taylor series and they solved using Successive Under Relaxation (SUR) technique, iteratively. The central difference method is used for discretization procedure. The detailed solution technique is well described in the literature [13]. The convergence criterion, 10^{-4}, is chosen for all depended variables and value of 0.1 is taken for under-relaxation parameter. Some grid tests are made between 34x23 and 352x235 to obtain optimum grid dimension. The test results showed that 214x143 grid dimension is enough for calculations.

RESULTS AND DISCUSSION

A numerical simulation is performed for a saltbox attic with eave for winter season. Different parameters such as Ra number, eave length and aspect ratio (AR) of the attic were tested to see the effects of these parameters on natural convection.

Figure 2 shows flow field (by streamline, on the left) and temperature distribution (by isotherms, on the right) for different Rayleigh numbers and E=6%, AR=0.6. Both AR and eave length represent real values which is used in architecture. In the case of small Ra number ($Ra=10^4$), conduction mode of heat transfer is dominant to convection and almost parallel temperature distribution is formed. For both $Ra=10^4$ and 10^5, two circulation cells are formed in different rotation directions. However, multiple cells are obtained for the highest Ra number, namely $Ra=10^6$ (Figure 2c). At this value of Ra number, plumlike temperature distribution is observed. However, for all values of Ra number, at the intersection point of bottom wall and eave, isotherms are stored due to higher temperature difference.



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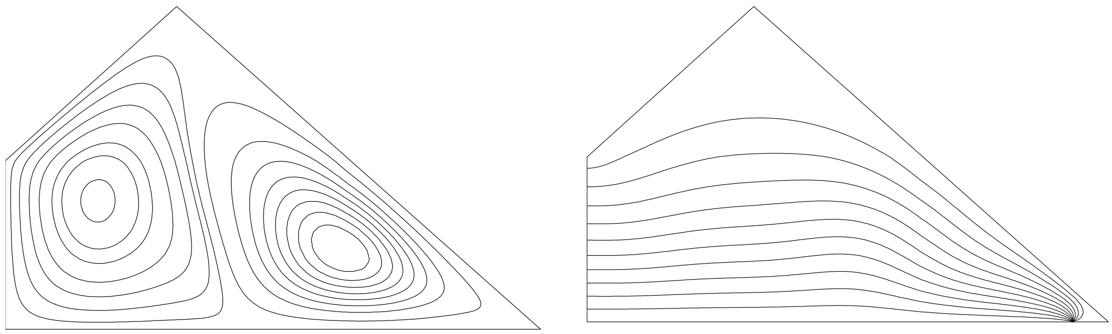

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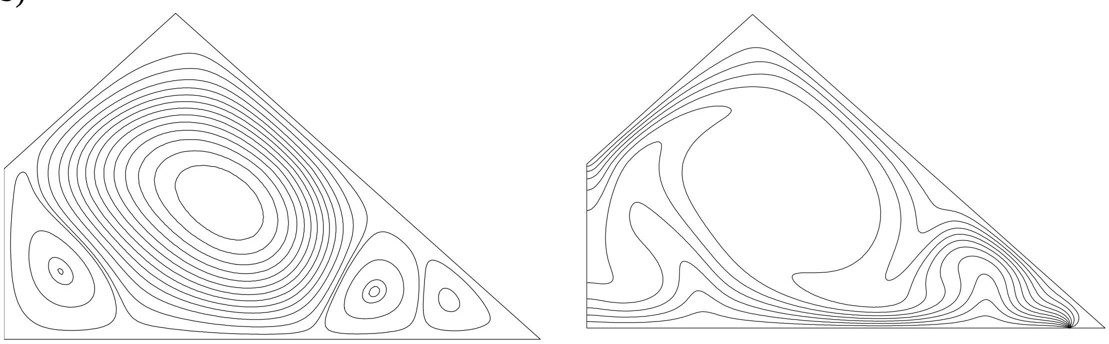
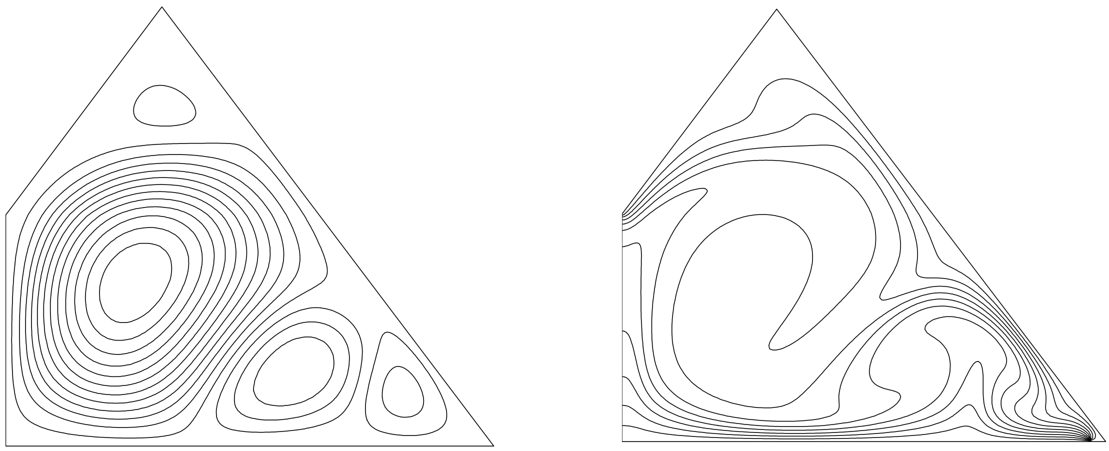


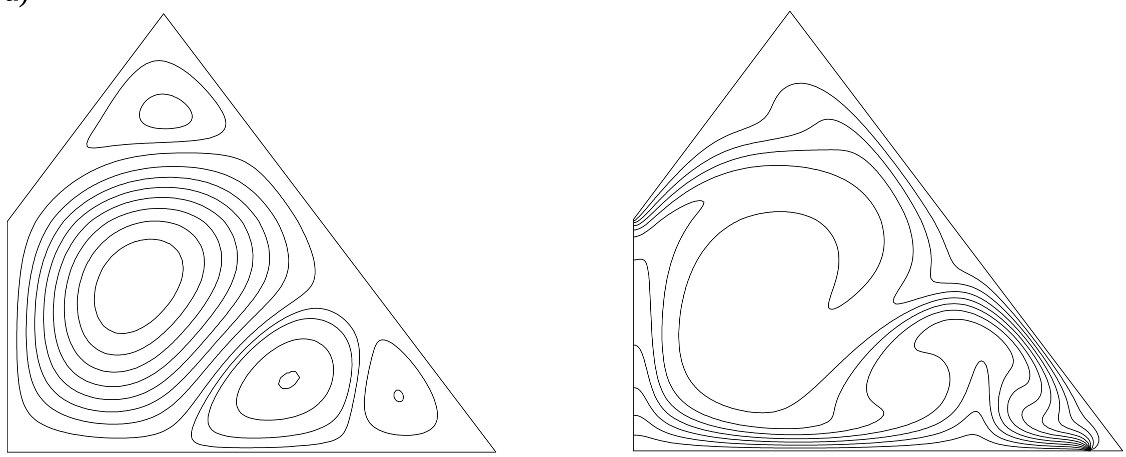
c)

Figure 2. Streamlines (on the left) and isotherms (on the right) at E=%6, AR=0.6, a) $Ra=10^4$, b) $Ra=10^5$, c) $Ra=10^6$

Figure 3 is presents the flow and temperature distribution for AR=0.9 and Ra=10^6 to show the effects of eave length. As can be seen from the figure, length of eave affects the isotherms and streamlines due to length of heated and cooled part of the bottom wall are changed. Strong plumelike distributions are observed for isotherm. Even, they show mushroom shaped distribution near the right corner. It means that strong convection is occurred. For the highest value of eave, two cells are formed which one of them locates to the top of the triangle like region and the other forms almost at the middle of enclosure (Figure 3(c)).



a)



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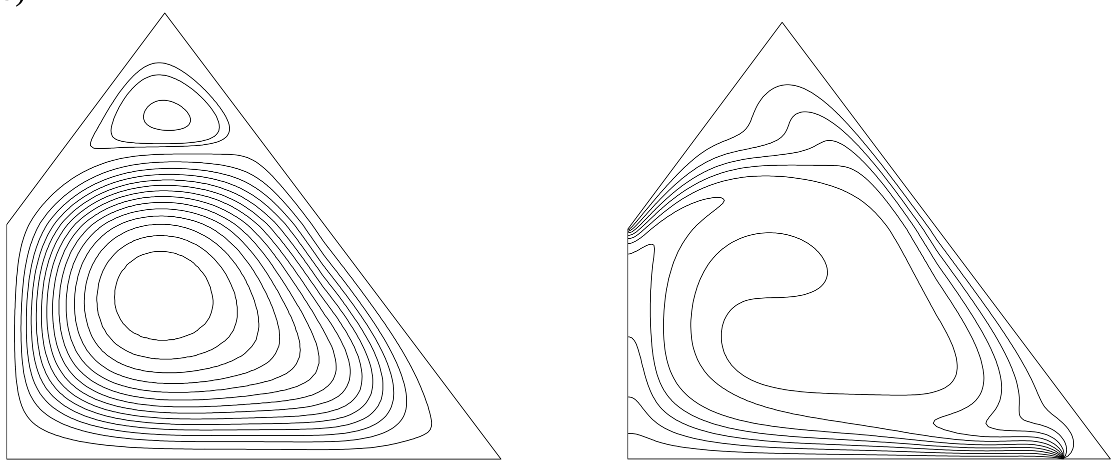
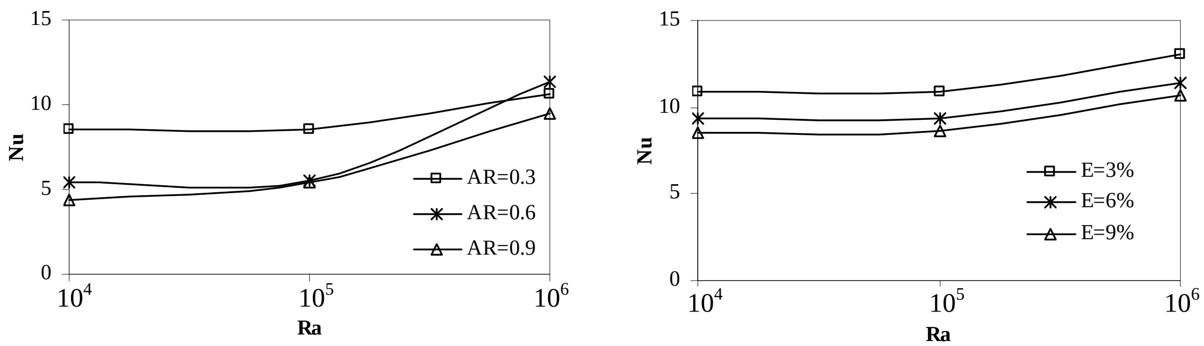


c)

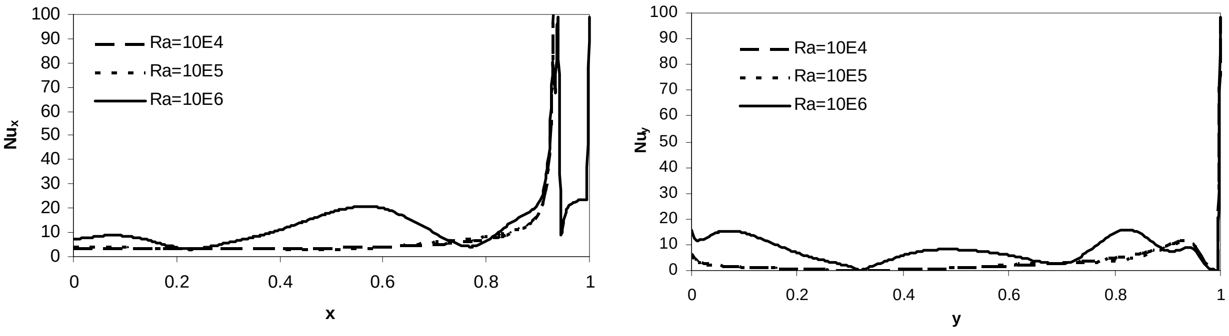
Figure 3. Streamlines (on the left) and isotherms (on the right) at AR=0.9, Ra=10^6, a) E=3%, b) E=6%, c) E=9%

The results of mean Nusselt number are shown in Figures 4 (a) and (b) for different aspect ratios and eave lengths, respectively. Figure 4(a) shows that heat transfer decreases with increasing AR. However, conduction mode of heat transfer is dominant to convection at lower Ra number ($Ra<10^4$). It enhances with increasing Ra number due to domination of convection mode of heat transfer. Mean Nusselt number decreases with increasing eave length due to decreasing heater surface. It means that eave makes negative effect on heat transfer inside the roof. Variation of local Nusselt number with Rayleigh number and eave length is shown in Figure 5 and 6, respectively. In these figures, results are presented for both inclined and bottom surfaces. As can be seen from the Figure 5 (a) that results of local Nusselt numbers are close to each other due to domination of conduction mode of heat transfer along the bottom wall. However, higher values were obtained for $Ra=10^6$. At this value of Rayleigh number, local Nu number shows wavy variation. The highest Nu number was formed at the intersection point of eave and ceiling due to high temperature difference. Local Nusselt number shows similar variation along the inclined surfaces except intersection point of eave and ceiling. The variation of local Nusselt number for different eave length is shown in Figure 6 (a) (along the bottom surface) and 6(b) (along the inclined surface). As can be illustrated in Figures, length of eave directly affects the peak point of local Nusselt number due to changing of location of maximum temperature difference. Higher values were obtained due to increasing of heater surfaces.



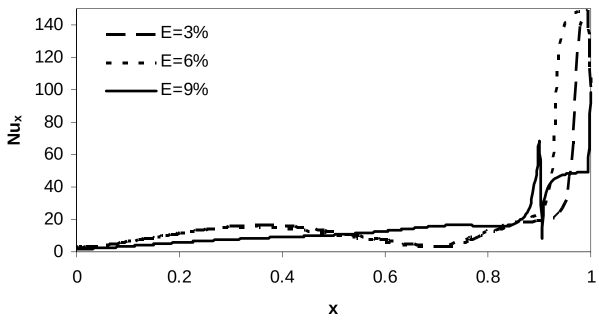
a) b)

Figure 4. Variation of mean Nusselt number with Rayleigh number, a) at different ARs, b) at different eave lengths



a) b)

Figure 5. Variation of local Nusselt number for different Rayleigh numbers a) along the bottom wall, b) along the inclined wall



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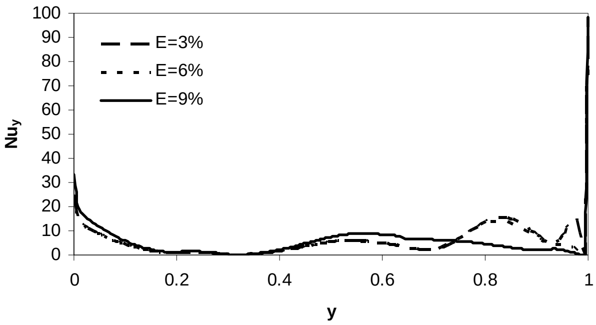

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Figure 6. Variation of local Nusselt number for different eave lengths a) along the bottom wall, b) along the inclined wall,

CONCLUSIONS

Steady state flow field by natural convection in a saltbox roof with eave for different aspect ratios has been numerically studied. Based on the findings in this study, we conclude that heat transfer regime is conduction for lower Rayleigh numbers. Heat transfer increases with increasing of Rayleigh number with domination of convection mode of heat transfer. Due to geometrical shape of the roof at least double circulation number was formed. It is observed that number of cells is strongly depending on eave length and Rayleigh number. Higher heat transfer was formed for lower aspect ratios. Similarly, smaller eave length enhances the heat transfer.

NOMENCLATURE

AR	aspect ratio, AR=H/L
B	length of room ceiling (m)
E	eave length (m)
g	gravitational acceleration (ms^{-2})
Gr	Grashof number
H	height of roof (m)
L	length of bottom wall (m)
Nu	Nusselt number
Pr	Prandtl number
Ra	Rayleigh number
T	temperature (K)
u, v	axial and radial velocities (ms^{-1})
x,y	cartesian coordinates (m)
X, Y	non-dimensional coordinates
Greek Letters	
υ	kinematic viscosity (m^2s^{-1})
Ω	non-dimensional vorticity
θ	non-dimensional temperature
β	thermal expansion coefficient (K^{-1})
α	thermal diffusivity (m^2s^{-1})
Ψ	non-dimensional streamfunction

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Effects of Geometrical Shape of Roofs on Natural Convection for Winter Conditions

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SUMMARY

In the practical applications, roofs of buildings can be in different shapes depending on architectural design of building or climate. Some of these building roofs can be classified as gambrel, saltbox and gable roofs. In the present study, we investigated the natural convection heat transfer and fluid flow inside the gambrel, gable and saltbox roofs for winter boundary conditions. With this aim, the identified roofs are compared with each other from the heat transfer and flow field point of view. Effects of Rayleigh number also tested in each type. Results are presented with streamlines, isotherms, local and mean Nusselt numbers. It is aimed that this study will help for designer to show more efficient ways of saving energy.

Keywords: *natural convection, roof, winter conditions*

INTRODUCTION

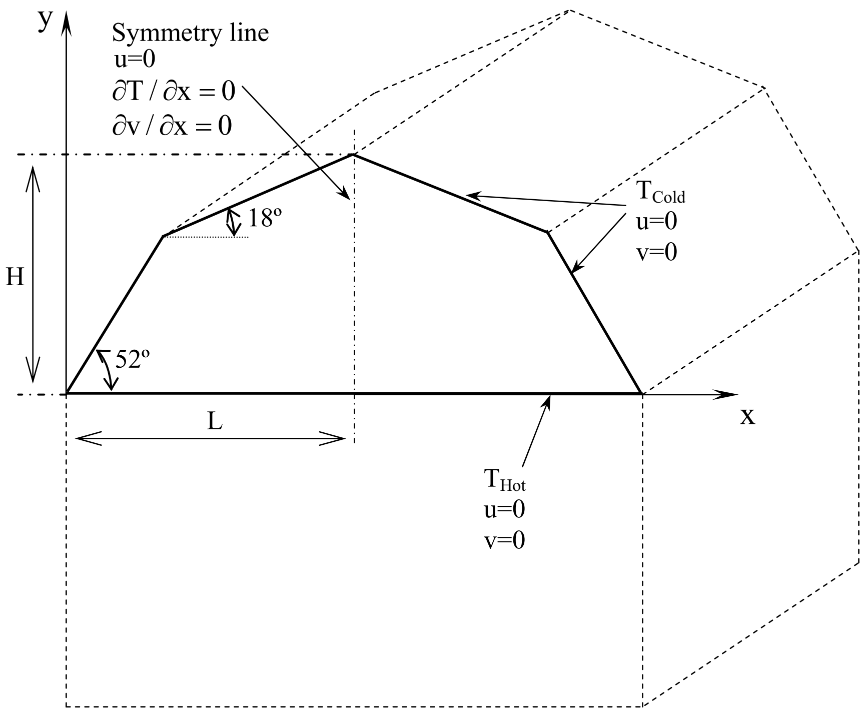
Roofs protect the building from environmental effects such as rain, snow and wind or particle matter in air. It also prevents the heat losses from environment to the room and vise versa. One can see different types of roofs as gambrel, gable, saltbox, and shed roofs etc. in urban or rural areas. These types of the roofs are chosen according to decorative view or type of the building. Both winter and summer day conditions natural convection heat transfer occurs inside the roof due to buoyancy forces and temperature difference between ceiling of the room and sloping wall of the roof.

Investigation of the natural convection heat transfer and fluid flow inside the roof is important to save energy and reduce to heating or cooling load of the building. Thus, cost of energy of the building will be decrease. Natural convection in roofs is analyzed mainly for gable roofs in the literature. Asan and Namli [1,2] investigated the natural convection heat transfer in gable roofs for both summer and winter day boundary conditions using finite difference techniques. They indicated that both aspect ratios of the roof and temperature boundary conditions are important parameters on temperature and flow fields inside the roof. Varol et al. [3] made a numerical solution for gambrel roofs for both summer and winter conditions. The same method applied the solution of natural convection inside the Saltbox roofs [4] and shed roofs with or without eave in summer conditions [5]. Natural convection in trapezoidal enclosure with offset baffles investigated by Moukalled and Acharya [6]. They tested the effects of baffle on buoyancy-induced flow and thermal fields in the trapezoidal shaped attics. They observed that conjugate baffles make important effect on natural convection and can be used control parameter for heat transfer.

The main purpose of this study is to examine the natural convection heat transfer in the different types of the roof. These roofs are chosen as gambrel, gable, saltbox and compared with each other from the heat transfer point of view. The bottom area of the roofs is equal in each type to compare occurred heat transfer in the roof.

DEFINITION OF TREATED ROOFS MODELS

Three different roofs models were chosen to compare the effects of roof shape on temperature and flow field of natural convection. Figure 1 shows physical models for treated roof types. Inclination angle is the most important parameter for a roof which is chosen based on meteorological data of the region. To compare roof types in this study, inclination angle of roofs are chosen according to climate of the East Anatolian region of Turkey. Thus, main inclination angles are chosen 52^o and 18^o for gambrel, gable and saltbox roofs, respectively. Comparisons are performed depends on winter conditions. Thus bottom wall (ceiling of the room) behaves as heater and temperature of the inclined wall is colder than that of bottom wall.



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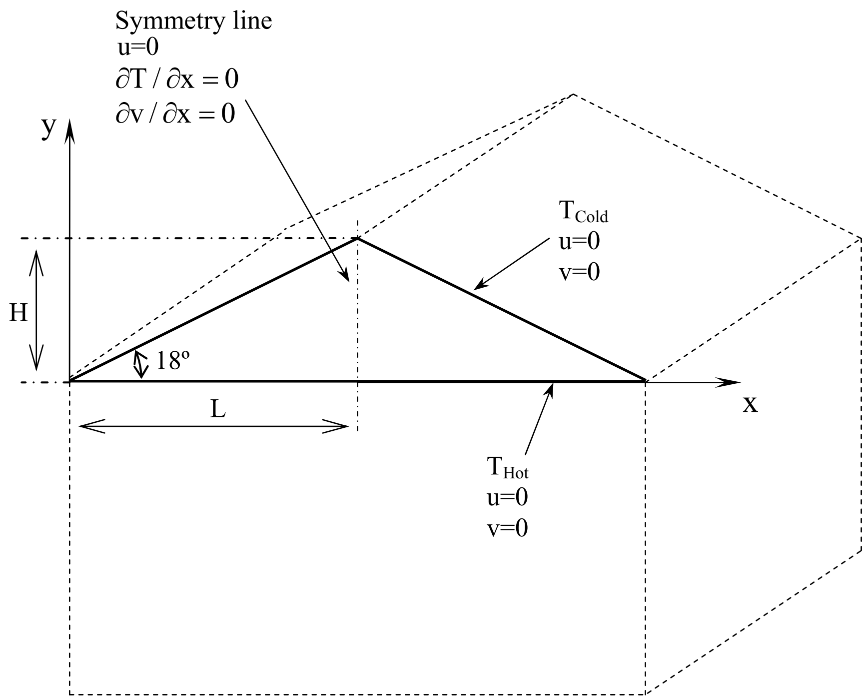

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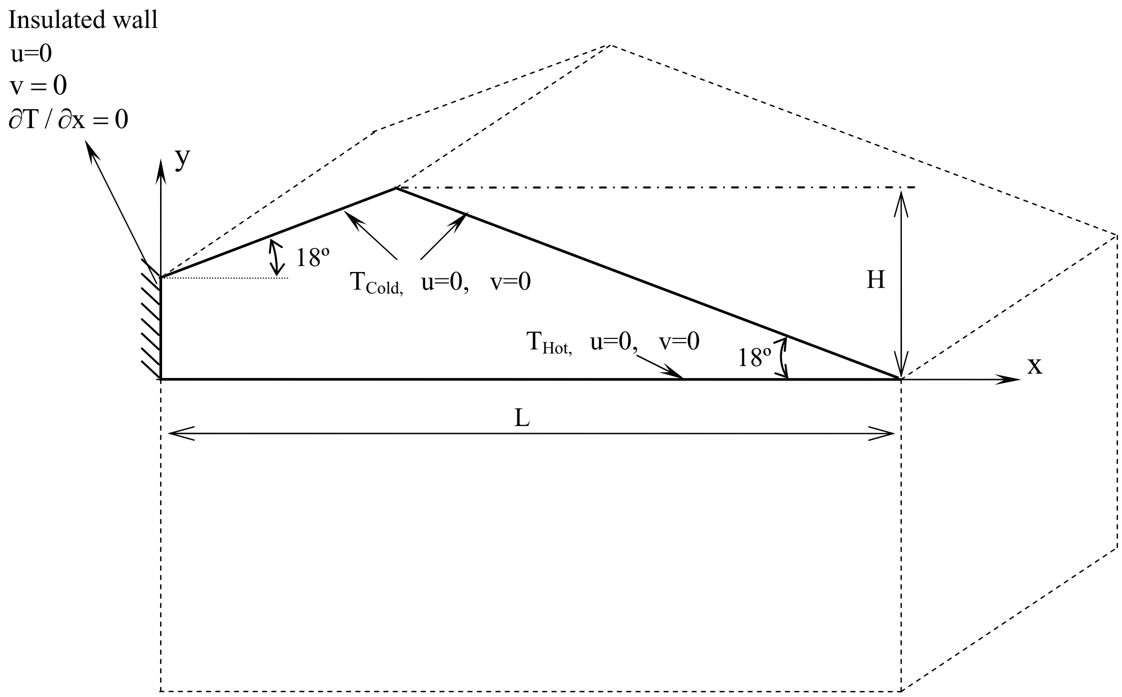

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Figure 1. Different types of roofs a) Gambrel roof, b) Gable roof, c) Saltbox roof

Left part of the roof according to symmetry line was chosen as computational domain in each type except saltbox roof. Two-dimensional, steady-state and laminar solution were accepted. Further assumption was made that flow is incompressible and Newtonian. Thus, governing equations can be written as follows:

$$X = \frac{x}{L}, \quad Y = \frac{y}{L}, \quad \Psi = \frac{\psi \, Pr}{\upsilon}, \quad \Omega = \frac{\omega (L)^2 \, Pr}{\upsilon}, \quad \theta = \frac{T - T_c}{T_h - T_c} \tag{1}$$

$$u = \frac{\partial \psi}{\partial y}, \quad v = -\frac{\partial \psi}{\partial x}, \quad \omega = \left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \right), \quad Ra = \frac{\beta g (T_h - T_c) L^3 \, Pr}{\upsilon^2}, \quad Pr = \frac{\upsilon}{\alpha}. \tag{2}$$

Based on the dimensionless variables above governing equations (stream function, vorticity and energy equations) can be written as

$$-\Omega = \frac{\partial^2 \Psi}{\partial X^2} + \frac{\partial^2 \Psi}{\partial Y^2} \tag{3}$$

$$\frac{\partial^2 \Omega}{\partial X^2} + \frac{\partial^2 \Omega}{\partial Y^2} = \frac{1}{Pr} \left(\frac{\partial \Psi}{\partial Y} \frac{\partial \Omega}{\partial X} - \frac{\partial \Psi}{\partial X} \frac{\partial \Omega}{\partial Y} \right) - Ra \left(\frac{\partial \theta}{\partial X} \right) \tag{4}$$

$$\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} = \frac{\partial \Psi}{\partial Y} \frac{\partial \theta}{\partial X} - \frac{\partial \Psi}{\partial X} \frac{\partial \theta}{\partial Y} \tag{5}$$

The numerical method used in the present study is based on finite difference method to discretize the governing equations (Eqs. 3-5) and the set of algebraic equations are solved using Successive Under Relaxation (SUR) technique. The solution technique is well described in the literature [7,8] and has been widely used to solve natural convection equations. The convergence criterion 10^{-4} is chosen for all depended variables and 0.1 is taken for under-relaxation parameter. The present solutions are compared with the known results for triangular shaped roofs from the open literature. As indicated in the Figure 2, it was observed that these results of the present code are in very good agreement with the literature [2,9,10].

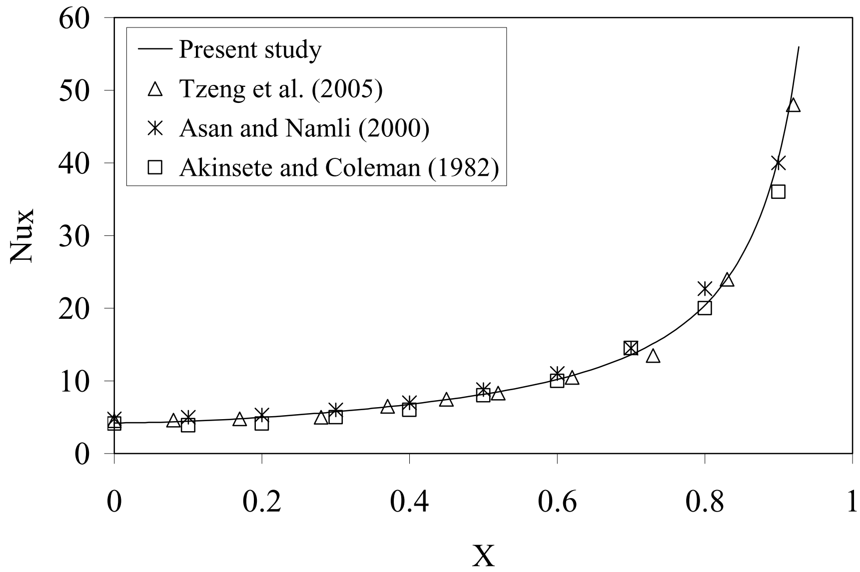


Figure 2. Comparison of obtained results with literature.

As indicated above, the problem is defined for winter boundary conditions. The physical boundary conditions are illustrated in the physical model (Figure 1) and they can be defined as follows:

On the inclined surfaces, u=v=0, $T_U = T_{cold}$ (6)

On the bottom wall, u=v=0, $T_B=T_{hot}$ (7)

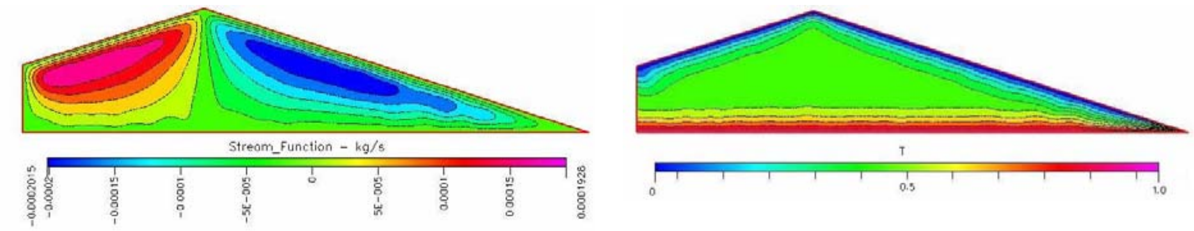
On the symmetry line, u=0, vertical wall, u=0, $\partial T/\partial x = 0, \partial T/\partial x = 0$ (8)

Calculation of the local Nusselt number was performed by

$$Nu_x = -\left.\frac{\partial \theta}{\partial Y}\right|_{y=0} \tag{9}$$

RESULTS AND DISCUSSION

Numerical analyses of natural convection temperature and flow fields for different types of the roofs have been performed in this study for different Rayleigh numbers. Streamlines and isotherms are shown in Figure 3 for different type of roofs in winter conditions at $Ra=10^6$. Thus, the bottom area of the roofs is chosen as equal. As can be seen from the figure, double circulation cells were formed in type of saltbox roof. However, in other types of the roofs single cell was obtained.



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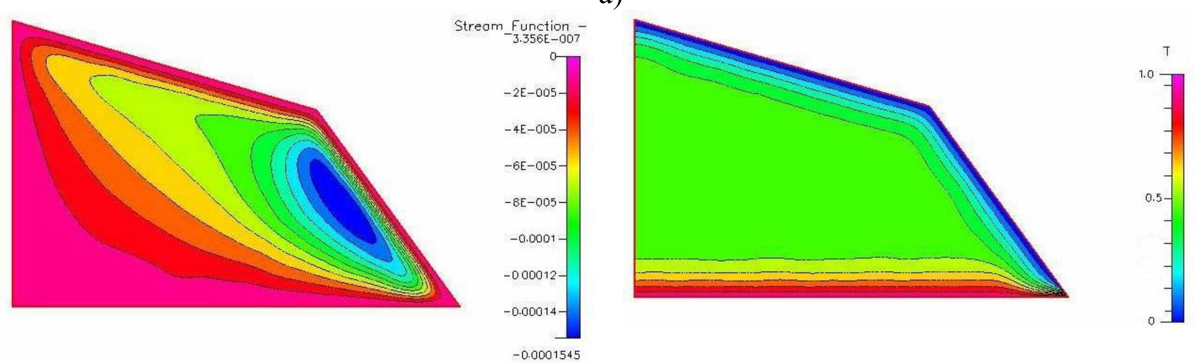

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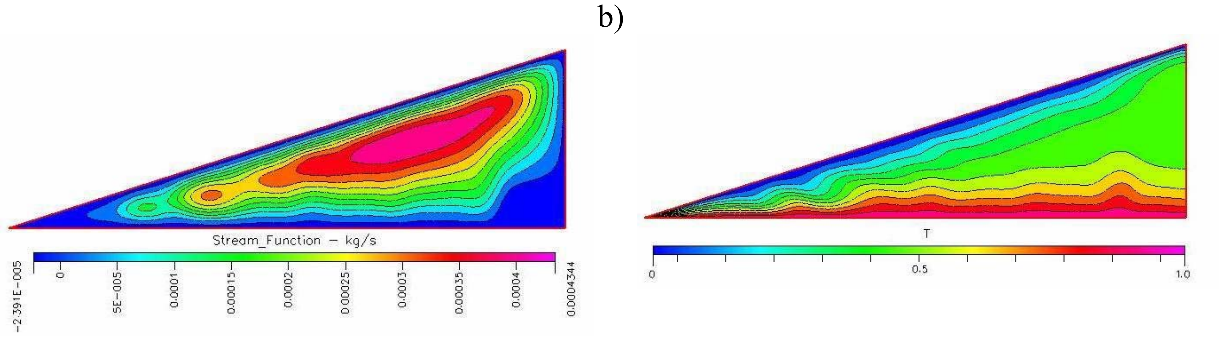

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Figure 3. Flow field by streamline (On the left) and temperature distribution by isotherms for winter day conditions at $Ra=10^6$ a) Saltbox, b) Gambrel, c) Gable

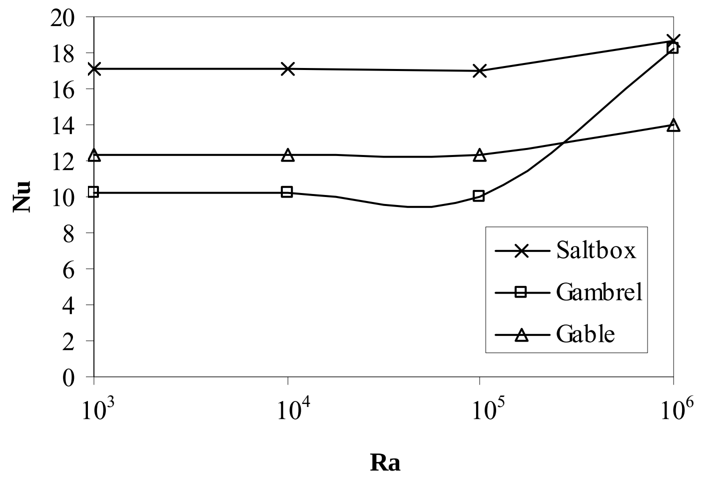

Figure 4. Variation of mean Nusselt number with Rayleigh number for different types of the roof

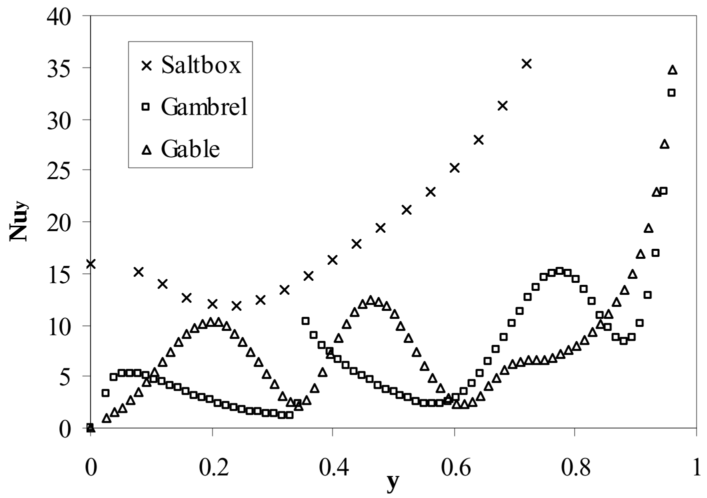

Figure 5. Variation of local Nusselt number along the heated wall at $Ra=10^6$

Figure 4 shows the mean Nusselt numbers which defines the overall heat transfer. They have given for different types and Rayleigh number. As can be seen from the figure, heat transfer mode is mainly conductive at low Rayleigh numbers. Thus, mean Nusselt number becomes constant with increasing of Rayleigh number up to 10^5. For higher Ra numbers, mean Nusselt number increases with increasing of Ra number due to domination of convection mode of heat transfer. The figure shows that, the highest mean Nusselt number is obtained when saltbox roof is chosen due to strong flow strength. It is interesting result that when gambrel roof was chosen same values of mean Nusselt number was obtained between saltbox and gambrel roof at the highest Rayleigh number. Figure 5 presents the local Nusselt number over heated wall of roofs. Thus, variation of local Nusselt number can be comparable for different types of the roofs. Figure presents that, local Nusselt number shows almost sinusoidal shaped variation for gable roofs and its value smaller than that of saltbox roof. Again, gambrel roof shows wavy variation but its value is smaller at the left corner due to long distance between hot and cold walls. Variation shows monotonically increasing for saltbox roof and highest local Nusselt numbers are obtained for this type of roof.

CONCLUSION

Temperature distribution and flow fields of natural convection are presented for three types of the roofs using a numerical technique. It was observed that single cell was formed for gable and gambrel roofs but double circulation cell was formed for saltbox roof. It means that the flow strength is higher for these types of the roof. The lowest heat transfer was performed for gambrel roof at the low Rayleigh numbers. On the contrary, almost same values are formed for both saltbox and gambrel roofs at the highest Rayleigh number.

NOMENCLATURE

g	gravitational acceleration (ms^{-2})
Gr	Grashof number
H	height of roof (m)
L	length of bottom wall (m)
Nu	Nusselt number
Pr	Prandtl number
Ra	Rayleigh number
T	temperature (K)
u, v	axial and radial velocities (ms^{-1})
x,y	cartesian coordinates (m)
X, Y	non-dimensional coordinates
Greek Letters	
υ	kinematic viscosity (m^2s^{-1})
Ω	non-dimensional vorticity
θ	non-dimensional temperature
β	thermal expansion coefficient (K^{-1})
α	thermal diffusivity (m^2s^{-1})
Ψ	non-dimensional streamfunction

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Calculating the Heat Transfer of Wall Structures in Non-stationary Cases

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Summary

In order to get an accurate knowledge of the energy consumption of buildings and of comfort parameters inside of closed rooms depending on the time, we developed a mathematical model. Based on Fourier's differential equation, this mathematical model allows us to calculate the instantaneous heat loss, the amount of the heat flow stored in the wall between the two time values, and the plotting versus time of the temperature generated on the contact surface of a double-layer wall structure. In addition to the outside temperature, in the mathematical model we can take into account the radiation intensity reaching the external wall structures by way of sun radiation, as well as the plotting versus time of these meteorological characteristics. The boundary conditions of the developed equation system, which consists of 5 equations, can be determined on the basis of meteorological databases or our own measurements.

Introduction

For an accurate knowledge of the energy consumption of buildings and of comfort parameters inside of closed rooms depending on the time the development of mathematical models is necessary in the field of building structures and thus in the field of wall structures as well, which models enable the necessary calculations to be performed on a scientific basis but in a simple way and using relatively short CPU time. The results of these calculations are expected to provide the following numerical data both in winter and summer modes:

- The function describing the changing versus time of the heat loss in winter, and of the heat load in summer of the rooms, taking into consideration the thermal energy stored in the wall structures for some time.
- The values of the internal surface temperature of the wall structures, which allow us to evaluate the given room's thermal comfort if we know the other comfort parameters (air temperature, air velocity, humidity, clothes, activity level).

All of these calculated values should be determined with the knowledge of what we can regard the two most important meteorological variables, namely the outside temperature and the radiation intensity during any given day.

Methods

Calculating the heat transfer of wall structures in non-stationary cases is based on Fourier's differential equation. In addition to the necessary material characteristics (c: specific heat, ρ: density and λ: thermal conductance), this equation contains, as its last component, the effect of a heat source (E, in W/m^3), which allows us to also take into account the solar radiation reaching the wall surface.

$$c \cdot \rho \cdot \frac{\partial t}{\partial \tau} = \frac{\partial}{\partial x}\left(\lambda \cdot \frac{\partial t}{\partial x}\right) + \frac{\partial}{\partial y}\left(\lambda \cdot \frac{\partial t}{\partial y}\right) + \frac{\partial}{\partial z}\left(\lambda \cdot \frac{\partial t}{\partial z}\right) + E\,, \quad (1)$$

On the basis of this equation, and taking into consideration other factors as well, we can formulate an equation system consiting of 5 equations, two of which are differential equations that allow us to solve the equation system numerically if we choose the right steps.

The the equation system's unknown values, which have to be determined in relation to any given time period, are as follows:

- the value of the instantaneous heat loss: $\dot{Q}$,
- the heat flow stored in the wall between two time periods: $\Delta\dot{Q}$,
- the surface temperature of the internal wall structure: t_{iw},
- the temperature at the point of contact of the brick wall and of the heat insulation on its external surface: t_c.

In order to solve the equations, we need to know the meteorological characteristics, as well as the plotting versus time of the surface temperature of the external wall structures – out of these characteristics, the changing of the radiation intensity and of the air temperature for the given day was taken from an appropriate meteorological database [1], while the changing of the surface temperature of the external wall structures was determined by measurement.

Results

In order to define the mathematical model, we have used a double-layered wall structure that contains a brick structure capable of storing heat, and an outer heat insulation. Further, in relation to heating, we applied the following presumptions:

- By the wall's heat loss we mean the heat flow arriving at the internal wall structures, whose instatntaneous value, at the time of $\tau = 0$ is $\dot{Q}_0$.
- The heat flow stored in the wall in between the 1st and 2nd time period is: $\Delta\dot{Q}_1$.
- The heat flow on the external wall structures is the difference between the two quantities above, that is $(\dot{Q}_0 - \Delta\dot{Q}_1)$.

The mathematical model consists of the following equations:

$$\begin{aligned}\Delta\dot{Q}_1 = {} & \frac{c_1 \cdot \rho_1 \cdot \Delta x_1}{6 \cdot \Delta\tau} \cdot \left[\left(t_{iw1} + t_{c1}\right) - \left(t_{iw0} + t_{c0}\right)\right] + \frac{c_2 \cdot \rho_2 \cdot \Delta x_2}{6 \cdot \Delta\tau} \cdot \left[\left(t_{c1} + t_{ew1}\right) - \left(t_{c0} + t_{ew0}\right)\right] + \\ & + \frac{\lambda_1}{3} \cdot \frac{\left(t_{iw0} - t_{c0}\right)}{\Delta x_1} - \frac{\lambda_2}{3} \cdot \frac{\left(t_{c0} - t_{ew0}\right)}{\Delta x_2} + \frac{I_r \cdot a}{3} + \frac{\alpha_i}{3} \cdot \left(t_{i0} - t_{iw0}\right) - \frac{\alpha_e}{3} \cdot \left(t_{ew0} - t_{e0}\right)\end{aligned} \quad (2)$$

$$\dot{Q}_0 = \alpha_i \cdot \left(t_{i0} - t_{iw0}\right), \quad (3)$$

$$\dot{Q}_0 = \frac{c_1 \cdot \rho_1 \cdot \Delta x_1}{6 \cdot \Delta \tau} \cdot \left[\left(t_{iw1} + t_{c1} \right) - \left(t_{iw0} + t_{c0} \right) \right] + \frac{c_2 \cdot \rho_2 \cdot \Delta x_2}{6 \cdot \Delta \tau} \cdot \left[\left(t_{c1} + t_{ew1} \right) - \left(t_{c0} + t_{ew0} \right) \right] +$$
$$+ \frac{\lambda_1}{3} \cdot \frac{\left(t_{iw0} - t_{c0} \right)}{\Delta x_1} - \frac{\lambda_2}{3} \cdot \frac{\left(t_{c0} - t_{ew0} \right)}{\Delta x_2} + \frac{I_r \cdot a}{3} + \frac{\alpha_i}{3} \cdot \left(t_{i0} - t_{iw0} \right) + \frac{2 \cdot \alpha_e}{3} \cdot \left(t_{ew0} - t_{e0} \right) \quad (4)$$

$$t_{c0} = \frac{\lambda_1 \cdot \Delta x_2 \cdot t_{iw0} + \lambda_2 \cdot \Delta x_1 \cdot t_{ew0}}{\lambda_2 \cdot \Delta x_1 + \lambda_1 \cdot \Delta x_2}, \quad (5)$$

$$\Delta \dot{Q}_{j+1} = \dot{Q}_{j+1} - \dot{Q}_j, \quad (6)$$

The indexes used in the equations indicate the following:

1 and 2	the values of the brick wall and the heat insulation
iw and ew	internal and external wall structures
c	contact point between the brick wall and the heat insulation
i and e	internal and external
r	radiation
0, 1, j	time periods and running index

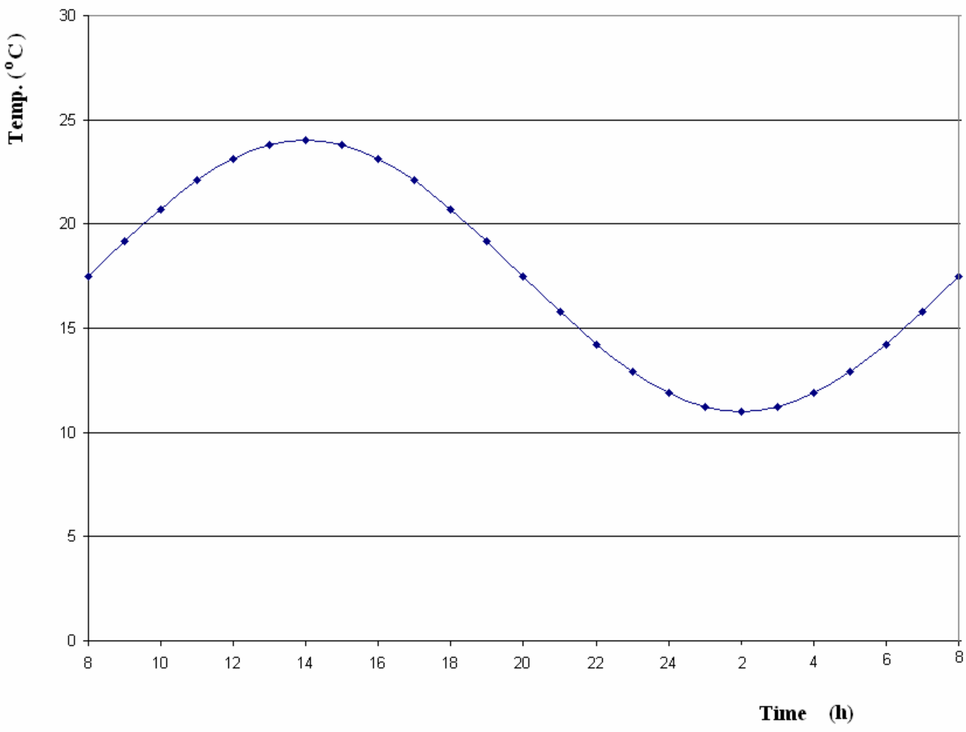

Figure 1. Daily outside temperature fluctuation in Hungary during the month of September

Figure 1 shows the external temperature characteristics of the month of September in case of clear weather. While analyzing the meteorological data, we found that, given Hungary's climate conditions, the average daily temperature measured at 8 am in the morning is the same in the case of clear or cloudy weather. In view of this fact, we determined the following equation for the plotting versus time of the temperature during a given day:

$$t = t_m + A \cdot \sin \frac{2 \cdot \pi}{24} (\tau - 8), \quad (7)$$

Where t_m is the average daily temperature, while A is the amplitude of the daily temperature fluctuation.

Figures 2 and 3 show the daily temperatures measured on the internal and external surfaces of the wall structures facing East. (The wall structure was made of 30 cm wide perforated bricks, plastered on both sides.) The daily fluctuation of the radiation intensity is described on the basis of curves that connect the time periods with the same or similar radiation intensity [1], Figures 4.

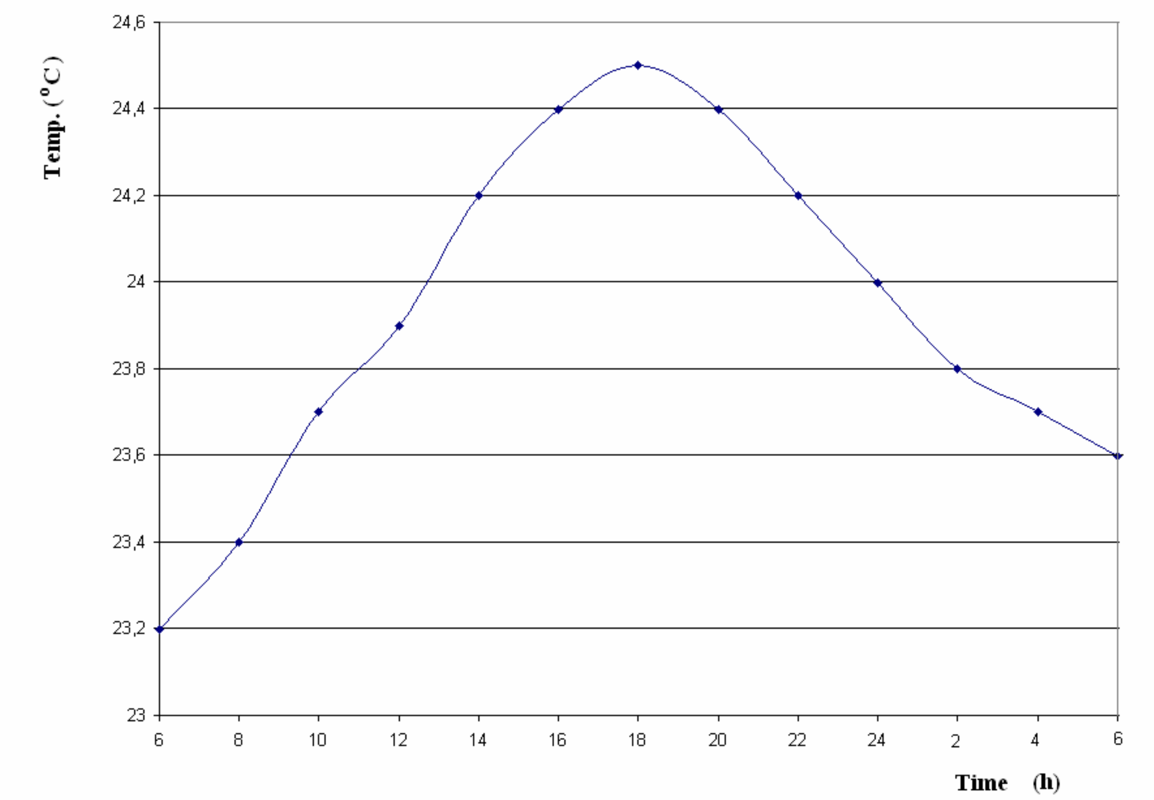

Figure 2. Daily surface temperature fluctuation on the internal wall surface of the wall structure facing East, measured in September 2006

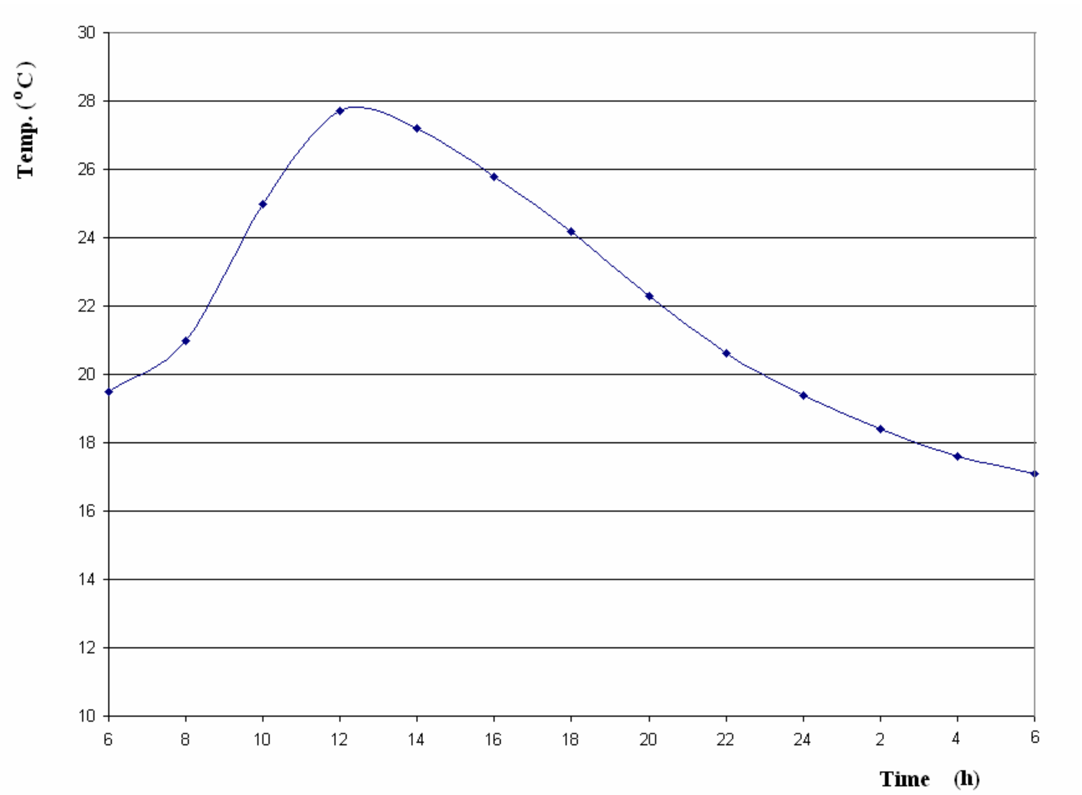

Figure 3. Daily surface temperature fluctuation on the external wall surface of the wall structure facing East, measured in September 2006

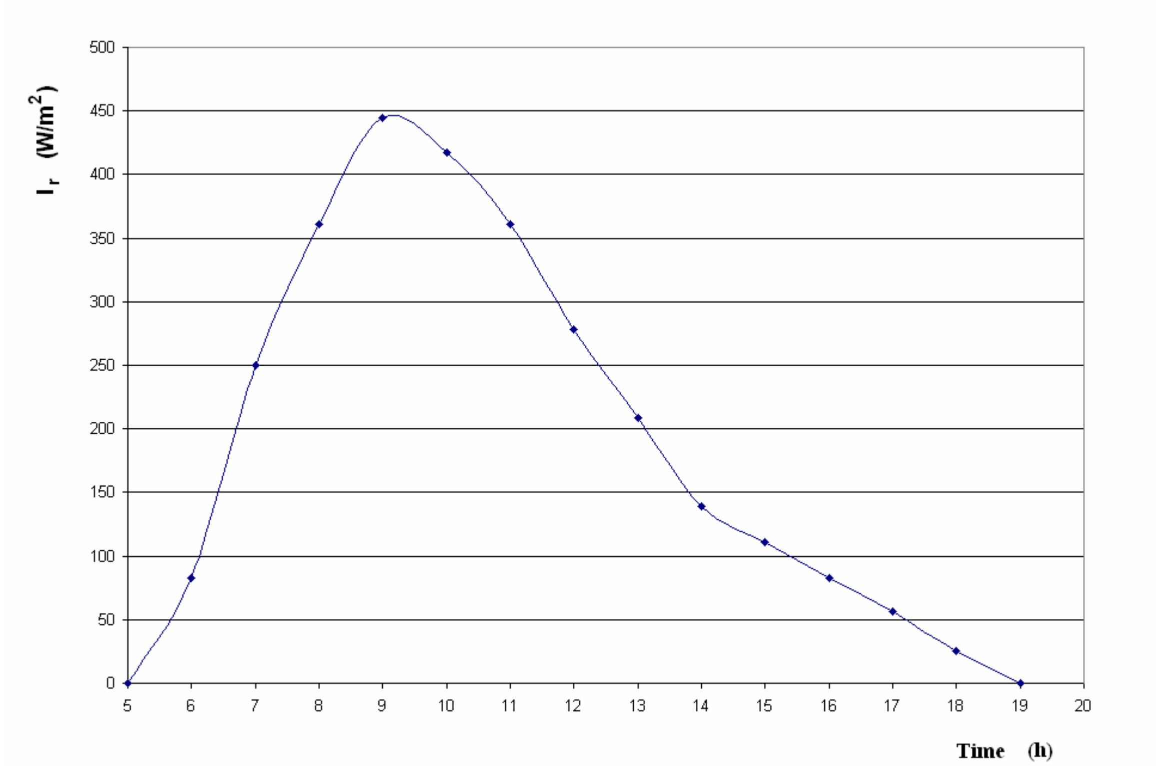

Figure 4. Daily fluctuation of radiation intensity in Hungary, on an average day in September

Discussion

To sum up the research work, we can draw the following conclusions:

- The equation system defined in (2) – (6) can be solved if we know the boundary conditions as defined on the basis of measurements and meteorological databases. The equation is best solved by applying a computer algebra system. When recording the $\Delta\tau = 1$ hour interval, we need to write down a total of 122 equations for one day, which allows us to define the unknown factors.
- From the measuring results of the internal and external wall surface temperatures the damping effect of the brick wall becomes quite obvious: while on the external wall structures we can detect approximately 11 °C daily fluctuation, this value on the internal wall structures is only 1,2 °C.
- As shown in Figures 2 and 3, after a cooler night (in comparison to the preceding nights), the value of t_{iw} will not drop to the value of the previous period's, but will increase to a small extent (approximately 0,3 °C). This temperature difference will be covered by the thermal energy that had been previously stored in the wall structure.

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Sustainable Approach to Healthy Buildings Indoor

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SUMMARY

Concepts of sustainability in buildings and cities are many and there are various programmes for healthy cities and researches on indoor air quality. Wellbeing indoors depend on external factors also. The fourth skin or microbiosphere above the building envelope has not been studied well. Sustainable building or green building is being designed but such building functions well only in sustainable cities and often indoor air quality is creating sick building syndrome. Concepts of solar envelope or shadow umbrella are new thinking with the study of urban geometry and bioclimatic design coming from tradition provides new solution with modern tools. Micro urban climate can be controlled with understanding of indoor and outdoor conditions of building envelope.

INTRODUCTION

There are growing concepts of sustainability in buildings and cities. There are multiple definitions, musrhoom concepts and parameters. These can be grouped into (a) broader aspect of environment, ecology, city and region, and (b) buildings and structures. The first group includes concepts like ecological footprint, factor four, green and brown agenda, life cycle assessment, environmental justice, ecoware etc. and the second group includes bioclimatic design, green building, recycling and regeneration, sick building syndrome, dematerlisation, ecovillage, zero emission etc. [1]. The concept of sustainability has widened from the original definition in the report of Burndtland commission. There are various programmes and projects . environmental management plan, healthy cities programmes, sustainable cities etc. Often these are uncoordinated and inside-out or outside-in i.e. relation between building and city is not well defined. A building is part of physical and nonphysical urban systems. Performance of building depends on sustainability triangle with socio¬economic, physical-technical and environment-ecological parameters. There is considerable amount of research on indoor air quality and there is growing research on outdoor environment but interaction between indoor and outdoor under different conditions in cities has not been researched well. The paper deals with these aspects. Wellbeing indoors depend on external factors as well. In a city often standards and criteria are given for environmental health, preventive health, safety health, reproductive health and health care for special groups etc. The management of healthy buildings and surrounding environment depends on occupants/users also.

THE FOURTH SKIN

The human skin is protected by clothing and apparel which are second skin and they live within a building envelope which is third skin. Now buildings are affected by external factors like pollution of air, water, noise, climate . sun, wind and other factors like natural disaster, sanitation, construction fault etc. On the other hand emissions from air conditioning, kitchen exhaust, disposal of waste etc. from buildings affect surrounding buildings. Mosquitos grown in overhead drinking water tanks have affected communities with malaria is many tropical countries. Heat island in highdensity development has effects on microclimate. There are now legal court cases in USA on the effect of perpetual shadows of tall buildings on surrounding low buildings. There is much knowledge and development on the thrid skin i.e. building envelope . façade, wall, roof, protective devices from sun, wind, ventilation, climate etc. together with energy saving devices but inside is also affected with sick building syndrome for the users.

There are now certain common aspects which are studied and results are applied in sustainable healthy buildings . use of renewable or less energy, recycling of waste or zero waste, recycling of water, use of recycled building materials, use of indoor greenery etc.

The invisible fourth skin of micro biosphere requires studies as this regulates health and wellbeing also. A new science of sensasory ecology provides new knowledge.

A safety pin reinvented from the Roman times is a multifunctional tool, strong and durable and yet it is made of very small amount of recyclable material. A Bedouin tent is a sustainable shelter. It can be moved, it can be closed, its height can be adjusted and it can be fixed according to movement of sun and wind and it can be folded and taken away in the desert on the camel back A building does not have the same advantages. It requires a holistic approach, from fragmented approach to integrated approach. In the cities building rules for construction are often not derived from townplanning zoning regulations. Despite huge investments the cities have become un-sustainable and buildings unhealthy. In old paradigm there was belief that the dynamics of the whole could be understood from the properties of the parts but in the new paradigm the properties can be understood only from dynamics of the whole [2].

The science of ontology comes here for consideration. Ontology is defined as an .explicit specification of conceptualization, ontology is a description of the concepts and relationship. [3]. It is defined as formal and explicit specification of a conceptualization[4]. In other words an ontology is a structured collection of unambiguously defined concepts. It is a knowledge management and sustainability has many ambiguous concepts. In computer simulated model, indoor-outdoor correlation can be established and ambiguities can be removed.

SUSTAINABLE BUILDINGS

A discusion on some aspects of indoor air pollution will be relevant though it is not intention of this paper. The main part is sustainable building and construction. There are many green codes. Sustainable building has minimum adverse effects on the built and natural environment, it reduces resource and energy consumption, prevent pollution, enhances natural environment etc. Out of principles of sustainable buildings and green buildings has come Environmental Architecture and its five principles are [5]

- Healthful Interior Environment
- Energy Efficiency
- Ecological Benign Materials
- Environmental form
- Good design

From this a concept of Ecological Building has developed with major areas in environmental building fabric with building technology and sustainable urban design concerning land form, microclimate, land use, site design, transportation, infrastructure efficiency, on site energy

resource etc. [6]. .Reduce, reuse, recycle, reintegrate, repair, restore, refurbish etc.. are new paradigms. Takashi Kawanaka, of National Institute for Land and Infrastructure, Tsukuba, Japan conducted research and developed formula for resource and energy-conscious urban planning [7] Out of 31, two sets of formula can be quoted.
Formulae for urban vegetation . A network of green spaces in urban areas, promote planting of tall tree in each lot, promote roof planting, promote the covering the building walls with vegetation.

Fromulae for construction in urban areas . utilize the advantages of accumulation in urban areas, larger buildings save more energy, consume less cement and steel, maximize the service life of buildings, Assess life cycle, Co2, Recover and dispose chloofluor carbous, (CFC) at steady rate.

European Charter for solar energy in architecture and urban planning was drafted in March 1996 in Berlin. It emphasised new thinking in the design of buildings and urban spaces. The charter states that the form of the urban and landscape structure need to be governed by the following factors [8].

- Orientation of street and building structures to the sun
- Temperature control and use of daylight in the public realm
- Topography (land form, overall exposure , general situation)
- Direction and intensity of wind (alignment of streets, sheltered public spaces, sytematic ventilation, cold-air corridor).
- Vegetation and distribution of planted areas (oxygen supply, dust consolidation, temperature balance, shading and windbreaks).
- Hydrogeology (relationship to water and waterway system) With the concept of sustainability in construction and public buildings, green accounting or environmental auditing becomes the new criteria. Such accounting is becoming an essential process before and after the construction and after a period when occupants users can react. Sustainability is now from paradigm to measurement. The use of Life Cycle Assessment (LCA) as a dignostic tool for determining the environmental health of production process including input, output, resources used and waste generated, use of energy, water etc. has been recognised in many evaluations including US Green Building Council.s assessment. Various evaluation rating have been adopted such as LEED . leadership in energy and environmental design and BREAM Building research environmental assessment and management. The LEED building rating systems are based on erosion and sedimentation control, alternative transport, the reduced site disturbances, storm water management, landscape, water use reduction, innovative waste water technologies, optimum energy performance and renewable energy system, building storage, building reuse, construction waste management, recycled content, carbon dioxide monitoring, thermal comfort, daylight and views, innovation in design etc. Many projects have been certified according to a rating system. The Royal Institute of British Architects[10] has given guidelines on key indicators of sustainability which are land and ecology, materials, community energy, health and water. The American Institute of Architects. Sustainability Task Force has also identified 10 measures on similar aspects including landuse and site ecology, bioclimatic design etc. [11]. Sustainable buildings function well in sustainable cities.

Sustainability is not new and it is in every religious texts. Accordingly to Buddhism, all outer and inner phenomena, the mind and its surrounding environment are understood to be inseperable and interdependent, human beings, society and nature are interconnected. According to Hinduism there is soul in every plant and animal and in the religious text it is said, .I am pervading the universe. All objects in the universe rest on me as pearls on the thread of a garland.. In the Hinduism, the creation is the result of union of humanity (Purush) with nature (Prakrity). The World Council of Churches said, the integrity of creation has a social aspect which we recognise peace with justice, and an ecological aspect which we recognise in the self-renewing, sustainable character of natural ecosystems. The Orthodox Church says that the humanity ought to perceive the natural order as a sign and sacrament of God. In other religions similar thinking is there [9].

INDOOR AND SICK BUILDING SYNDROME

The whole healthy building system can be subdivided into several components (a) Indoor air quality, building related diseases and human response (b) Indoor climate (c) Design and operation of healthy buildings (d) Materials systems and technologies for healthy buildings (e) Policies and practice issues in creating healthy buildings in all subitems [12].

Indoor environment depends on airquality, ventilation, thermal comfort, noise and visual comfort. Many traditional architecture was solar sensitive and buildings were designed accordingly. Attention is now diverted to occupants/users. reaction in indoor environment. Sick Building Syndrome is an increasing phenomenon.

Sick Building Syndrome (SBS) is a situation in which building occupants experience acute negative physical and mental effect that is linked to time spent in a contaminated and overcrowded area. According to the World Health Organisation SBS may be diagnosed if greater than 20 percent of building occupants complain symptoms of sickness, irritation, infection etc. The ventilation standard has been revised to provide a minimum of 15cfm of outdoor air per person.

In recent times energy saving and various measures in green buildings have been adopted but WHO pointed out energy efficient but sick buildings often cost society far more than it gains by energy saving and peoples. confidence in the effectiveness of health and building authorities may be seriously harmed if sick building becomes a common phenomenon [13]. The causes of Sick Building Syndrome are (a) Inadequate ventilation (b) Chemical contaminants indoor and outdoor and (c) Biological contaminants and solutions may include (1) Pollutant source removal or modification (2) increasing ventilation (3) Air cleaning (4) Education and communication [14].

There are many hazardous indoor pollutants. In the beginning of the 20th Century there were 50 materials to construct a building, now there are 55000 with half of them synthetic [15]. There is increasing asthma in UK. Home dustmite, dampness etc. which are associated with respiratory diseases, surrounding environmental conditions and changing of air temperature, moisture generation and ventilation rates are also responsible [16]. In developing countries, about 50% of people use solid fuels (wood, coal, dung etc) for cooking resuting high incidence of respiratory diseases [17].

THE SOLAR ENVELOPE & SHADOW UMBRELLA

According to Dr. Ralph Knowles, for solar sensitive healthy buildings the shape and size of buildings will change and a concept of solar envelope has been developed by research at MTT and at Univ of Southern California, USA [18]. Orientation of streets, placement of greenery and water bodies, control of radiation and pollution, reduction of heat island effect and many other healthy measures can provide the fourth skin for a healthy building. The Solar Envelope is proposed as a zoning device to gurantee access to sunshine now and in the future. It is an imaginary contanier to regulate development within limits. It does not shadow surrounding buildings but there is assured solar access with a new design strategy.

Urban geometry affects heat island which is a dome of stagnant warm air at the heavily built up areas of cities. Emmanuel writes that the urban geometry of a city is characterized by a repetitive element called urban canyon.a three dimensional space bounded by street and buildings thus restrict the view of skydome and gives multiple refections of solar radiation and restricts the free movement of air. [19]. Emmanuel developed concept of shadow umbrella to reduce radiation in outdoor environment which is based on urban massing, use of water and vegetation.

BIOCLIMATIC DESIGN

Shadow umbrella is part of bioclimatic design Traditionally tropical passive solar architecture adopted bioclimatic design.wind catchers, courtyards, plantations, water, shading devices etc. The symbiotic relation between indoor and outdoor is seen in many historic area. The bioclimatic design eliminates heat islands and enhances windflow and aerodynamics. Offices, shopping centres and other places visited by large number of people have large parking areas and many mechanical devices, a bioclimatic environment creates buffer and reduces sick building syndrome. Bioclimatic design in green building does not mean design with greenary. It is solar sensitive environment and greenery which help to promote healthy environment by converting carbondioxide into oxygen which is to be incorporated. Architectural solution for healthy indoor is a result of implementation of ecological principles. According to Dr. Giovani bioclimatic design can control climate at micro-urban scale with understanding of indoor and outdoor conditinos of building envelope [20].

CONCLUSION

The fourth skin or microbiosphere around building envelope has not been studied properly. A healthy and sustainable building and its indoor air quality has symbiotic relation with outdoor. Sick building syndrome can be avoided. Astudy of urban geometry with concepts like solar envelope and shadow umbrella is needed. Bioclimatic design is recommended to develop healthy buildings.

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Energy efficient building design

The ASHRAE GreenGuide: One Means of Establishing a Link between Sustainable Design Practitioners

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SUMMARY

The paper discusses the newly revised ASHRAE GreenGuide, particularly as it relates to the topic of indoor environmental quality. The updated Guide includes a new chapter on LEED Guidance for Mechanical Engineers and a new chapter on building systems Impact on the Local Environment- both indoor and outdoor. This chapter is intended to describe how HVAC systems interact with the building indoor and outdoor environment, and includes (among others) sections dealing with indoor environmental quality, cooling tower systems and chemical water treatment, acoustics, and the science of designing healthy buildings. One of the more useful concepts in the Guide is the inclusion of “Green Tips”, which are sidebar summaries of specific technologies that can be used to design a high-performance (green) building. Three building specific Green Tips are discussed, with particular emphasis on topical areas related to indoor air quality concerns.

INTRODUCTION

The ASHRAE Green Guide (Guide) [1] is a primary reference for mechanical engineers working on high performance building projects in the United States. The original edition of the Guide was released in January of 2004, and the second edition was introduced at the U.S. Green Building Council’s Greenbuild Conference in Denver, CO, USA in November, 2006. The Guide is intended to be a living document and the process is in place to maintain this with new editions that will contain updated material.

The Guide is not intended as a complete compilation of all interactions that buildings, and heating, ventilation and air-conditioning (HVAC) systems in particular, have on and with the environment. Many of the issues are common knowledge as being important to “green” building design among engineers and lay people alike; such areas as energy consumption, location of buildings and the construction process are prime examples. The Guide is primarily intended to convey ideas on how to improve buildings and their systems. There are, however, some areas that are either not intuitively obvious as being potential impacts of HVAC systems, or are items that are some may not consider to be truly “sustainability” issues. While not being the sole target market, the Guide is written such that practitioners in the early stages of their career will particularly benefit.

Regardless of your definition of sustainability or the various labels and compartmentalization, it is assumed that the reader of the Guide is interested in designing buildings and their systems that provide for the needs of the occupants while minimizing the their adverse impacts. Therefore, the chapter on local environmental impacts provides examples of several areas that the HVAC engineer may not initially think are important when minimizing environmental impacts, but truly are significant.

This paper discusses the importance of continuing to build a link between ASHRAE, REHVA and other global building engineering design associations. Important differences in the integrated design process will be identified from a U.S. perspective. The issues will be evaluated and discussed with the goal of working toward the optimization of the high performance building design process globally. The example presented for this discussion is the Second Edition of the ASHRAE Green Guide.

What is New to the GreenGuide

The updated Guide includes a new chapter on LEED Guidance for Mechanical Engineers and a new chapter on building systems Impact on the Local Environment- both indoor and outdoor. This chapter is intended to describe how HVAC systems interact with the building indoor and outdoor environment, and includes (among others) sections dealing with indoor environmental quality, cooling tower systems and chemical water treatment, acoustics, and the science of designing healthy buildings. One of the more useful concepts in the Guide is the inclusion of "Green Tips", which are sidebar summaries of specific technologies that can be used to design a high-performance (green) building. Some of the chapters from the First Edition have been reorganized in an attempt to more accurately mirror the path that an actual project would take from Pre-Design to Post Occupancy. Content has been added and edited in all of the chapters, with significant updates in the subject areas of Building Automation Systems, Renewable Energy Options, combined heat and power and ground-source heat pump systems. The Second Edition also provides a more global perspective with a specific section titled "International Perspective", along with SI/IP dual units provided for all.

METHODS

Developing a guideline on Indoor Environmental Quality (IEQ) intended for a broad audience while still technically sufficient can be a difficult task. One weakness of the original edition of the Guide was a lack of information on IEQ, and this has been corrected with the new second edition. This section first presents a summary of the IEQ information content in the new Guide and how it relates to other chapters within the Guide. Next, a description is given of three examples on how information contained in the Guide can be applied.

The terms indoor environmental quality (IEQ) and indoor air quality (IAQ) are sometimes confused as being one in the same. In reality, *IEQ* is a broader, more encompassing concept which includes *IAQ* as one of the key factors. Other areas are also considered key to providing good overall IEQ, such as:

- Air quality and ventilation
- Thermal comfort
- Acoustics and noise
- Lighting levels
- Visual perception
- Building materials and envelope

When considering indoor air quality, the outdoor environment can have a negative impact on the building HVAC and the indoor environment, or vice-versa depending on the specific situation. Location of outdoor air intakes near a known contamination source (such as a loading dock with potential idling engines) can seriously degrade the indoor air quality by introducing, rather than removing, contaminants. Similarly, building exhausts can contaminate the local area near the exhaust discharge, making this air unsuitable for human exposure or re-intake into the building. Chapter 44 of the 2003 ASHRAE Applications handbook [2] (Building Air Intake and Exhaust Design) contains more information on this topic.

Assuming no contamination of the local air surrounding the building, then good indoor air quality is possible by providing adequate ventilation and distribution within the space; for example if the design met the requirements as specified in ASHRAE Standard 62.1 [3].

Similar to lighting levels, thermal comfort affect the occupants and overall building indoor environmental quality. Thermal comfort of the occupied space is covered in Chapter 7 of the Guide. The interaction with the local environment has minimal impact on thermal comfort.

The acoustical environment can also be an important factor in determining good indoor environmental quality. Sound and vibration are the often unheralded contributors to occupant comfort and health that should be an integral part of green building design and should not be forgotten. While noise is not always an obvious problem, human productivity and performance can be impacted by the acoustical environment.

Adequate lighting levels are required for the building occupants. Lighting levels required vary according to the design purpose of the room or building zone. The local environment, in the form of trees, landscaping or other buildings, can influence the lighting that may enter the space and hence affect the lighting levels inside. Lighting and its impact on HVAC load determination is discussed in more detail in Chapter 6 of the Guide.

This is another area that influences how a person perceives the indoor environmental quality. Rarely would the HVAC system interact with visual perception of the indoor space, with one possible aspect being exposed ductwork. Any HVAC system interaction with visual perception will likely be dealt with by the project architect. The Guide addresses these issues in Chapter 4.

Recent years have seen a marked increase in recognition of the impact that building materials (envelope, furniture, paints, flooring, etc.) have on indoor air quality. This was the primary reason for changes to the ASHRAE Standard 62.2 outdoor air ventilation requirements to include an allowance for the building area and not just total number of occupants.

The indoor air quality can be negatively affected by off-gassing of chemicals building materials or chemicals used during the construction or fabrication of the components. The LEED program contains a number of credit point items that related to reducing the introduction of potentially harmful materials into a building environment. It also describes methods to help ensure that key HVAC components, such as ductwork, do not become contaminated during construction and act as a source of indoor pollution after occupancy.

Focusing on the Indoor Air Quality concept and how the Guide can assist in advancing the latest design concepts that optimize air quality, several example are given indicating how the Guide would be used.

The first example presented is how “green” practices can be applied in general to laboratory buildings. Building-specific ‘Green Tips’ have been included in the Second Edition of the Guide. The Green Tip for Lab Buildings indicates a number of design concepts for consideration as they relate to IEQ issues.

The second example is a summary of the Green Tip for Healthcare Facilities, and the final example is a similar compilation of techniques as applied to the specific design challenges of a university student residential facility. The strategies outlined can also be applied to hotels and multi-unit residential complexes, including luxury condominium developments.

RESULTS

Example 1: Laboratory Buildings

One of the user friendly features of the Green Guide are the Green Tips. If used, these tips can help the designer make informed decisions on what systems may be properly integrated on a specific project. These tips also allow the engineer to identify benefits and costs in order to work with the constructors and the owner to implement concepts that best suit the needs and ideals of the project.

Building-specific Green Tips have been included in the new Second Edition of the Guide. The Green Tip for Laboratory Buildings focuses on safety, energy and occupant comfort considerations. The following design concepts are given for consideration as they relate to IEQ issues:

Safety

1. Fume hood design and associated air distribution and controls must be designed to protect the users and the validity of the laboratory work.
2. Pressurize rooms consistent with the ASHRAE Laboratory Design Guide and any other code required standards. Utilize building pressurization mapping to develop air distribution, exchange rate and control strategies.
3. Optimize air exchange rates to ensure occupant safety while minimizing energy usage.
4. Chemical, biological and nuclear storage and handling exhaust and ventilation systems must be designed to protect against indoor pollution, outdoor pollution and fire hazards.
5. Intake/ exhaust location strategies should be modeled to ensure that lab exhaust air is not reintroduced back into the building air handling system.

Energy Considerations

1. Heat recovery for spaces served by air handling units with 100% outside air capability or over 50% outdoor air component. The Guide contains several Green Tips associated with air-to-air energy recovery covering heat exchange enthalpy wheels, heat pipe systems and run-around systems.
2. Utilize variable air volume (VAV) systems to minimize air exchange rates during unoccupied hours.
3. Consider low flow fume hoods with constant volume controls where this concept can be properly applied.
4. When appropriate, consider decoupling outdoor air conditioning from the space loads by using chilled beams.

Occupant Comfort

1. Air systems should be designed to allow for a collaborative working environment. Acoustic criteria should be adhered to in order to maintain acceptable levels of noise control.
2. Day light and views should be considered where lab work will not be adversely affected.

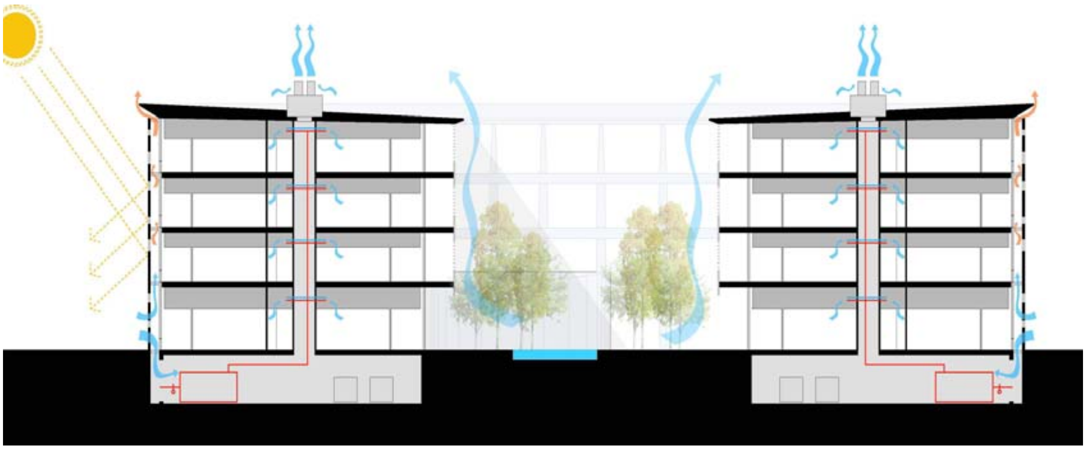

Figure 1. Conceptual diagram indicating optimization of indoor air quality and minimization of cooling loads for a proposed project in Algeria.

Example 2: Healthcare Facilities

Healthcare facilities are infrastructure intensive and include many different types of spaces. The HVAC systems for these different types of spaces must be designed to address the specific needs of the spaces being served. The first considerations should always be safety and infection control. In addition, optimizing energy efficiency and positively affecting the patient experience should also be important design team goals. A summary of the key points contained in the Guide's Green Tip for healthcare facilities is given in the following topical sections.

Safety and Infection Control

1. Consider HEPA (high-efficiency particulate air-filter) filtration for all air handling equipment serving the facility.
2. Consider air distribution and pressurization strategies in operating and trauma rooms that balance air flow zones from most clean to least clean. The order of zone cleanliness starts at the operating and thermal plume location at the patient, then moves to the zone around the doctors, the zone around the room, and then the zone outside of the room.
3. Pressurize rooms consistent with American Institute of Architect (AIA), ASHRAE or other local or national codes and guidelines.
4. Provide air exchange rates in excess of AIA guidelines in operating rooms, intensive care units, isolation rooms, trauma rooms, and patient rooms.
5. Redundancy of equipment should be designed for fail-safe operation and optimal full and part load energy efficient operation.

6. Intake and exhaust location strategies should be modeled (computational fluid dynamics or wind tunnel) to ensure no re-introduction of exhaust into the building.

Energy Considerations

1. Heat recovery for spaces served by air handling units with 100% outside air capability.
2. Utilize VAV systems in non-critical spaces working in conjunction with lighting occupancy sensors.

Occupant Comfort

1. Acoustics of systems and spaces must be designed with patient comfort in mind.
2. Daylighting and views should be provided but design to minimize the HVAC load impact of these benefits.
3. Provide individual temperature control of patient rooms with capability of adjustment by patient.
4. Building pressurization relationships/ odor issues should be carefully mapped and addressed in the design and operation of the building.

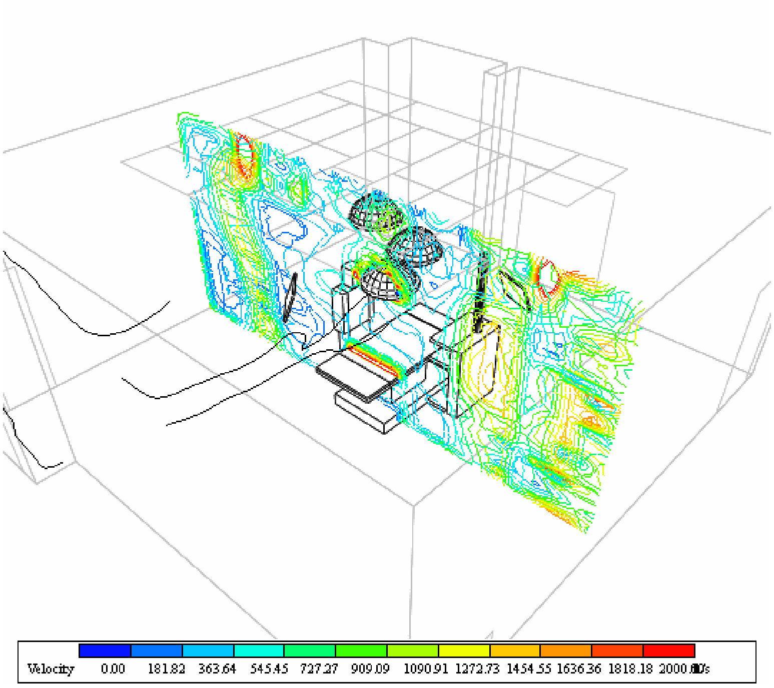

Figure 2. CFD results showing particle path tracing in an operating room. (UMass Memorial Lakeside Addition, Worcester, MA, USA)

Key Elements of Cost

1. HEPA filtration costs are significant in both first cost and operating cost. The engineer should work closely with the infection control specialists at the healthcare facility to determine cost/ benefit assessment of the filtration strategies.
2. Heat recovery strategies should be assessed using life cycle analyses. All components of the strategy must be taken into account, including the negative aspects, such as adding fan static pressure, and therefore, using more fan energy, when heat wheel or heat pipe strategies are considered.

Example 3: University Residential Facilities

Student residence halls are made up primarily of living spaces (bedrooms, living rooms, kitchen areas, common spaces, study spaces, etc.). Most of these buildings also have central laundry facilities, assembly/ main lobby areas, and central meeting/ study rooms. Some of these spaces also include classrooms, central kitchen and dining facilities. The strategies outlined below can also be applied to hotels and multi-unit residential complexes, including luxury condominium developments.

Energy Considerations

1. Heat recovery for spaces served by air handling units with 100% outside air capability serving living units (exhaust taken from toilet rooms and supply air to occupied spaces).
2. Utilize VAV systems or induction systems for public spaces.
3. Investigate methods to provide natural ventilation or hybrid natural ventilation strategies, as appropriate to the local climate. An example airflow path of such is given in Figure 2.
4. Utilize electronically commuted motors (ECM) for fan coil units.
5. Utilize ground source heat pumps where feasible.

Occupant Comfort

1. Systems should be design to appropriately control noise in occupied spaces.
2. Daylight and views should be optimized while minimizing load impact on the building.
3. Consider providing occupant control in all bedrooms, which will be a balance of cost impact with improved indoor environmental quality.

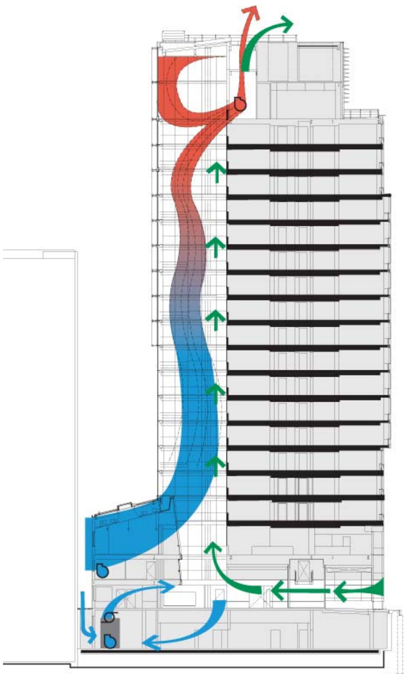

Figure 3. Path of natural ventilation in high-rise atrium which also optimizes views while minimizing solar gain (Suffolk University Residence Hall, Boston, MA, USA).

Key Elements of Cost

1. While there is a premium to be paid in first costs for ECM motors, many utility companies have energy rebate programs that make this concept viable, even on projects with tight budgets.
2. Heat recovery strategies should be assessed using life cycle analyses. All components of the strategy must be taken into account, including the negative aspects, such as adding fan static pressure, and therefore, using more fan energy, when heat wheel or heat pipe strategies are considered.
3. Hybrid natural ventilation strategies could be utilized using operable windows, properly designed vents using venturi effect to optimize natural airflow through the building, and shut down of mechanical ventilation and cooling systems during ambient temperature ranges between 15 and 17 C. This will save significant operating costs. The costs of the operable windows and vents will need to be weighed against the energy savings.

SUMMARY

Indoor air quality is influenced by all of the inter-related aspects of building design. The ASHRAE Green Guide is a valuable tool for mechanical engineers striving to meet the goals of resource use optimization and a comfortable, healthy indoor environment through an integrated design team approach.

Sources of Further Information

The following are sources of further detailed information regarding the three Green Tip examples for different building types listed in this paper.

ASHRAE Laboratory Design Guide, NFPA 45 Standard on Fire Protection for Laboratories using Chemicals, and Labs 21 Environmental Performance Criteria, http://www.labs21century.gov/

ASHRAE - HVAC Design Manual for Hospitals and Clinics, NFPA-99 Healthcare Facilities 2002, ASHE Green Guide for Healthcare, www.gghc.org

BRESCU, BRE, "Natural Ventilation for Offices" Guide and CDRom, ÓBRE on behalf of the NatVent Consortium, Garston, Watford, UK, March 1999, Svensson C. and Aggerholm S.A., "Design tool for natural ventilation" Proceedings of the ASHRAE, IAQ AQ '98 Conference, New Orleans, October 24-27, 1998, ASHRAE Fundamentals 2005, Chapter 27, pages 25.10 - 25.12.

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Workshop Integral Design Methodology

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SUMMARY

Integral design shows high promises to reduce failure costs and to improve design quality. Based on this assumption, the Royal Institute of Dutch Architects (BNA), the Dutch Society for Building Services (TVVL) and Delft University of Technology (TUD) started a research project on Integral Design in year 2000, which resulted in a series of workshops for architects and HVAC consultants. This project was succeeded by new research within the Knowledge Centre Buildings and Systems (KCBS), in which Eindhoven University of Technology (TU/e) and the Netherlands Organization for Applied Scientific Research (TNO) cooperate. The ongoing research utilizes workshops, in which already over 220 professionals from BNA and the Dutch Association of Consulting Engineers (ONRI) participated, for development and evaluation of integral design methodology in the domain of sustainable comfort systems. Both workshops and 'integral design methodology' are used as part of the education programme for continuous BNA-ONRI professionals' personal development.

This paper presents theoretical background for knowledge sharing and knowledge creation within building design teams, implemented in form of 'ID(Integral Design)-methodology'. Additionally, development of workshop experimental setting for measurement of its effects is described.

INTRODUCTION

The present situation in building (design) practice, where "it is hard for building partners to give a collective good answer for variety of questions from the society" [24], is determined by large number of different and often mutually influencing factors that require a broad approach on a variety of levels. For improvement of this situation, changes on three levels are needed:

1. process level – in order to improve design process to fit all involved design disciplines;
2. product level – to improve the end product (building as a whole, as well as its parts);
3. culture level – to bridge the gap between 'Design' and 'Engineering' worlds, in case of building design specifically between architects and (building services) consultants.

To realize all these three aims, an integral approach, as defined by Quanjel and Zeiler [24], is needed: "Integral approach represents a broad view on the world around us that continuously needs to be adapted and developed from sound and documented experiences that emerge out of interaction between practice, research and education. This integral approach can eventually lead to integral process, team and method – all the required conditions for design of the end product." [24]

For such high ambition, a true culture change, it is not enough to just 'prescribe' new ways of doing things. History shows that of design processes through prescriptive methods is not sufficient for large scale design problems/situations [6]. This is partly due to a design problems' peculiarity, being described as 'ill-structured' [30] or even 'wickedness' [25].

The prescriptive methods are often based on experiences of the researchers who develop them. These experiences are either recognized, and as such considered 'open doors', or not recognized and turned down. Jjust reading about design methods is not enough to really get grip on the real philosophy behind them. Designers have to be thought and trained how to work with design methods in order to be able to implement them. Because one has to endure himself that something is indeed worthwhile pursuing, a 'learning by doing' course is used in our pursuit for culture change in building design practice. Insights acquired from observing this implementation have to subsequently be used for further improvement of design methods.

DESIGN PROCESS

Our aim is to improve conceptual design (process level) by defining an 'integral design (ID) methodology' that increases potential for creation of integral building designs (product level). We assume that positive results on these two levels, which we try to demonstrate in our research, eventually will trigger and support culture change in building design practice. For this to happen, continuous implementation of achieved results in the setting of research-education-practice triangle is crucial. Besides research and development goals, this is the major function of workshops.

The main reasons for focusing on design process instead on design product level are because of *subjectivity of* design task *interpretation* and of design (as product) *evaluation*. (A representation of any stage in design development, from initial sketches, models and drawings to prototypes and final spatial objects is considered 'design as product'.)

It is known that designers/architects tend to reinterpret initial program of requirements [15], being it in rational (through analysis) or intuitive way (by framing design situation) [21,29]. However, in both cases this reinterpretation can't be considered objective. Moreover, a designer often makes different interpretations of the same design assignment each time he/she is confronted with it again. In these types of situations it is hard to compare design results (as products) which are based on different interpretations, even though the designer might be the same. We therefore argue that objective comparison concerning integration aspects within building designs, made by *different* designers, is not possible. Even in case of independent experts' deployment, the measurements regarding evaluation of integral designs remain subjective [9].

Design solutions and design problems are evolving together [29]. Because of this duality, initial requirements and subsequently required essential design information often change according to the in time increased insights about design task. Within traditional design process organisation types some disciplines that have to provide parts of this information at the start of design process, simply aren't there. Instead, one must try to find this information using reference books, databases, case studies etc. However, information always has to be *transformed* into design, and this requires certain skills, which we regard as *implicit* design knowledge. Implicit knowledge is coupled to actual person/designer (looking at design from different viewpoints…), while information is discipline based. Considered separately from persons and their skills, information could also be described as a special form of knowledge. In this sense, a certain discipline can indeed be characterized by *explicit* knowledge that it represents. The previous are the reasons why we focus on term knowledge instead on information. Besides, the emphasis on availability of information implies that precise (objective) definition of design task is needed. Moreover, it indicates that objective definition of design task is possible, which in the past proved as a pitfall leading to defining designing as purely rational (scientific in classical meaning) activity [18,22].

DESIGN (TEAM) KNOWLEDGE

In contrast to clients, constructors and managers, design team's disciplines posses *object design knowledge*. 'Object knowledge' is knowledge on the characteristics and properties of artefacts and their materials [33]. Van Aken's distinction between object, realization and process design knowledge proves to be very helpful in explaining what we are trying to do: integrate *explicit* discipline based 'object design knowledge' through implementation of 'process design knowledge' (represented in our case by developing 'ID-methodology').

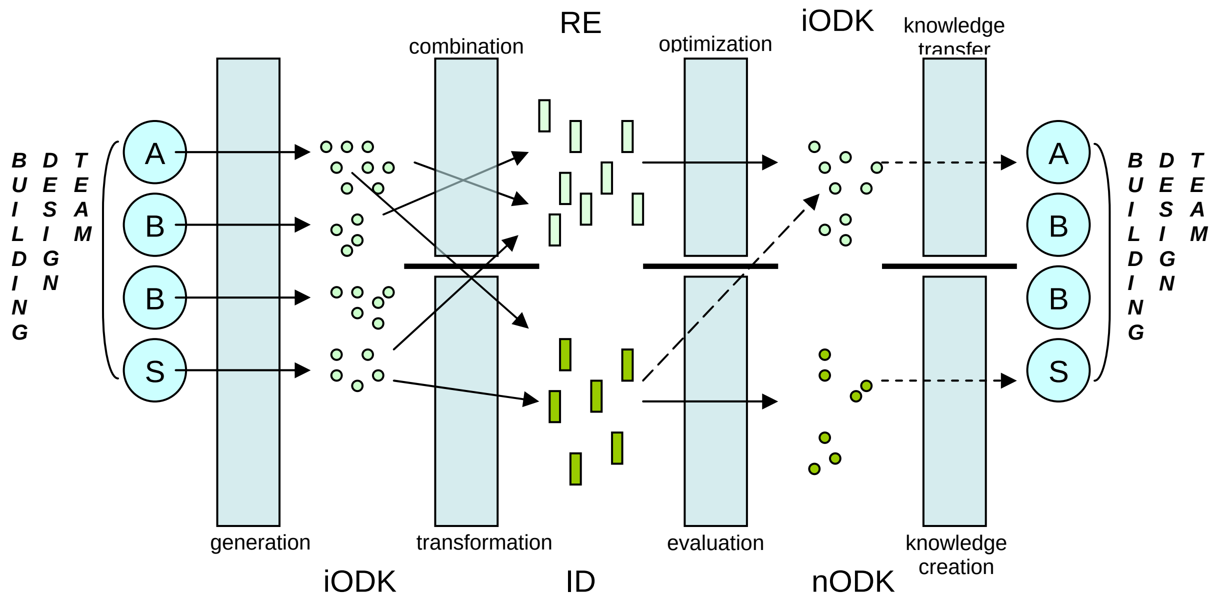
To be able to relate knowledge of different disciplines, we have adopted the view of designing as *the* most central activity in engineering [20]. However, it has been confirmed [28] that at present most of building design team disciplines actually don't act as designers during (traditional) design processes – meaning that there exists an artificial separation between 'design' (as a generalist; architect's) and 'engineering' (as a specialist; consultant's) activity. This duality forms an obstacle that has to be overcome. If we consider relations between different disciplines within design teams as a form of social system, the question is if (in our case design) activities can be imposed on 'design team system' through external (meaning outside intrinsic design activities) management. Certain approaches suggest that each system, if we look at its constituting entities, can only *organize itself* [1] – resulting in what is called emergent behaviour [17]. In order to enhance building design processes, communication between various disciplines has to be improved. Currently, cooperation between design disciplines is unsatisfactory; better organization of building design process is necessary [13]. Communication between different members of a design team is generally a notoriously difficult problem, especially at the early stages of design process [10]. It is important to stress that communication in the first place needs to be transparent; not only internally for design teams themselves, but also for external stakeholders. Communication within groups can generally be discerned in social-emotional and task neutral [2]. Similar distinction is made in literature on design teams, where distinction between task and team work, or content and process activities has been made [11,32]. In our 'ID-methodology' development we are primarily interested in task related communication. Given the fact that in our setting design teams have restricted amount of time to work on design tasks, we assume that this will automatically lead to more task related communication. Some research results support this assumption by showing that time pressure prevents teams from engaging in 'social niceties' [8].

KNOWLEDGE TRANSFORMATION

Besides communicating object design knowledge to each other, design team members/disciplines have to be able to use it for designing. Theoretical background on how design knowledge could be transformed into integral design concepts is found in "C-K theory" [14]. C-K stands for concept-knowledge relation. This theory defines design as a process generating co-expansion of two spaces, space of concepts C and space of knowledge K: "A design concept is a proposition that can not be logically valued in K... Concepts are candidates to be transformed into propositions of K, but are not themselves elements of K (properties of K can however be incorporated into concepts)… If a proposition is true in K, it would mean that it already exists and that we know all that we need about it (including its feasibility). Design would then immediately stop. There is no design if there are no concepts.

Without the distinction between the expansions of C and K, design disappears or is reduced to mere computation or optimization." In our view, optimization through merely (re)combination of already existing object design knowledge leads only to redesign (Figure 1, RE).

We focus on possibility of expanding concept space with integral design concepts (Figure 1, ID) and on producing new object design knowledge (Figure 1, nODK). A concept not being true or false (within K), the design process aims to transform this concept and will necessarily transform K [14]. At the end of process of generation and integration of concepts, transformation of existing object design knowledge within design team into new object design knowledge takes place, allowing design team members to acquire new insights in this 'learning by doing' approach.



A – architect
BP – building physics consultant
BS – building services consultant
SE – structural engineer
iODK – initial 'object design knowledge'
nODK – new 'object design knowledge'
RE – redesign (concepts)
ID – integral design (concepts)

Figure 1. Combination vs. transformation, knowledge transfer vs. knowledge creation; 'ID-methodology' design model

In our case K is defined by initial object design knowledge that participants bring into design team (iODK). Making this knowledge *explicit* enables designers/participants to use it for creation of design concepts. What we are curious about is if these concepts are *integral* (ID), some would even call them innovative, or just plain combinations (RE). The problem regarding innovative concepts / knowledge is that they are mostly related to the present state-of-art. We, on contrary, are focusing only on *knowledge within design team itself*. Now, if we assume that communication aspect can be measured, for example using Bales' Interaction Process Analysis categorization [2], the question is how we can measure (design) knowledge? Our operational definition is the one of only explicitly presented/communicated object design knowledge. Implicit knowledge is considered not directly transferable to other design team members and, as such, isn't treated as a research subject. If we were concerned how designers utilize implicit knowledge, in other words if we were researching *individual* design thinking, it would indeed be one of the essential aspects. Instead, we are interested in how explicit object design knowledge is transformed/integrated within building design *teams*, assuming that different design disciplines are (on average) of the same standard, because of their similar level of education and daily practice.

The essential aspect of design (thinking) is often referred to as creativity [7]. Although we are aware of significance of creativity in design thinking, we are not interested in how this process is unfolding, but if, when and how often it takes place within building design teams setting. People can be credited with creativity in two senses, described as P-creativity and H-creativity: P stands for psychological and H for historical [4]. The P-creativity represents creation of ideas which are new to the person that 'comes up' with them, whether this person immediately realizes their significance or not. These ideas are 'new' no matter how many other people may have had the same ideas already. H-creative ideas are fundamentally novel with respect to the whole of human history, and people usually have them in mind when they're speaking of 'real' creativity and 'real' innovative proposals. In case of design teams we consider transformation of object design knowledge, introduced by different design team disciplines, as design teams' P-creativity process. In other words, with application of 'ID-methodology' we are concerned how to stimulate design teams to produce, for themselves new, integral concepts. Seen in this way, *design within our integral approach represents realisation of potential for creation of new object design knowledge through integration of discipline based explicit object design knowledge into integral design concepts*.

By observing if proposed 'ID-methodology' for building design teams enhances emergence of integral design concepts (ID), we can say that (within specific context of a particular design team) potential for creation of new object design knowledge is realized. This new knowledge increases the possibility of arriving to 'satisficing' final solutions in subsequent design phases of a given situation (within specific design team, regarding given design task and using 'ID-methodology'). *The number of integral design concepts produced by design team is then the measure for this potential.*

INTEGRAL DESIGN METHODOLOGY SETTING

A suitable environment for integration of activities of a building design team is believed to be workshop setting. The workshops are seen as a self-evident way of working for designers, that occurs both in practice as during their education. They are however not predominant way of working in practice, where most of time different disciplines work separately. The actual designing, in the full design team line-up happens only occasionally and mostly at the very beginning of the project. Even then the purpose of (workshop) meetings is often just to get better acquainted with each other. Besides full design team line-up there are a number of other advantages of workshops with regard to standard office situations, while at the same time retaining practice-like situation as much as possible: the possibility to gather a large number of professionals in a relatively short time, repetition of the same assignment and comparison of different design teams and their results. The openness of participants for new methods is also bigger than during daily routine, something that can't be emphasized often enough.

Until now 12 workshops involving design teams were organized, with more then 220 participants. In all except one, workshop participants worked as design teams. A total of 65 teams were observed / worked with. The development of workshop setting was also a 'learning by doing' process. Instead of making a theoretically 'optimal' configuration, the approach we used was rather adaptive. Starting with 'standard' practice-like building team setting for the first sessions in 2001, workshops have evolved to two full-day series. The first workshops were organized during 'Integral Design' project [23,24] that was conducted by the Dutch Society for Building Services (TVVL), the Royal Institute of Dutch Architects (BNA) and Delft University of Technology (TUD), which involved mainly architects and building services consultants. The main focus of that project, which was initiated in 2001 and ended in 2003, was to raise the awareness of different disciplines about each others positions and problems in relation to building design. During this project a total of seven workshops were organized. The first one was an explorative session, which confirmed suitability of workshops for integration of activities of a building design team. Based on this result a workshop concept was developed in which the participants, members of BNA and TVVL, had to change their roles. The architect acted as building service consultant and vice versa. The awareness of position and needs of 'the other' was believed to be most evident if one had to play the role of 'the other'. A series of six identical workshops provided us with an important, and at the same time very surprising insight that either the level of knowledge about not only 'that other', but also about one's own field of expertise, was generally not that high; at best, it couldn't be understandably communicated to the other party in design process. This left us with conclusion that a way for structuring and confronting the respective (object design) knowledge of design disciplines needs to be found.

Hugely oversimplified, these were the most important conclusions regarding direction of further development of 'Integral design' workshops. Much more information is available in Quanjel's report [23], which unfortunately is only available in Dutch.

The basic framework for structuring knowledge of design team members was found in 'Methodical design' [34], a model which is problem oriented and distinguishes, based on functional hierarchy, various abstractions and/or complexity levels during different design stages and design phase activities. This framework proved its potential within (mechanical) engineering domain [3], and makes it possible to explicitly think and act on a specific abstraction level. A distinguishing feature of 'Methodical design' is the use of morphological overviews, both for the overall description of design stages as for separate design activities. Morphological overviews were first used by Zwicky [38], and are listed as one of 'Design methods' in the book by the same name [19]. Jones states that "morphological charts are intended to force divergent thinking and to safeguard against overlooking novel solutions to a design problem", and that "experienced designers in mechanical and structural engineering have quickly learned to use it with enthusiasm and success in areas in which they have some knowledge of problem structure and feasibility." The fact that workshop participants are also experienced designers is the main reason we assume that also within field of building design this method can be applied. Since, according to 'integral approach', the basis for culture change is formed by relation research-education-practice, we also implement findings from workshop based research into master education program within Department of Architecture, Building and Planning at TU/e. As such, we were able to confirm one of other Jones' statements: "graduate design students who have tried the method have found considerable difficulty in defining *functions*" [28].

Emphasis on working with functions is based on experienced designers' preference for function-oriented strategy [12], instead to phase-oriented that is often recommended by (engineering) design methodology [22]. Besides, definition of functions during interpretation of design task makes it possible to assess client's needs on a higher, but better workable, abstraction levels than program of requirements (which is often too detailed) provides. Based on definition of functions, various design complexity levels can be separately discussed and, accordingly, possible solutions generated. This way interaction with the client is aided, and at the same time design process is structured. The process of continuous interpretation and solution feedback transparently narrows field of possible solutions leading to well thought-out integral building concepts, while actively involving the client in design process [26].

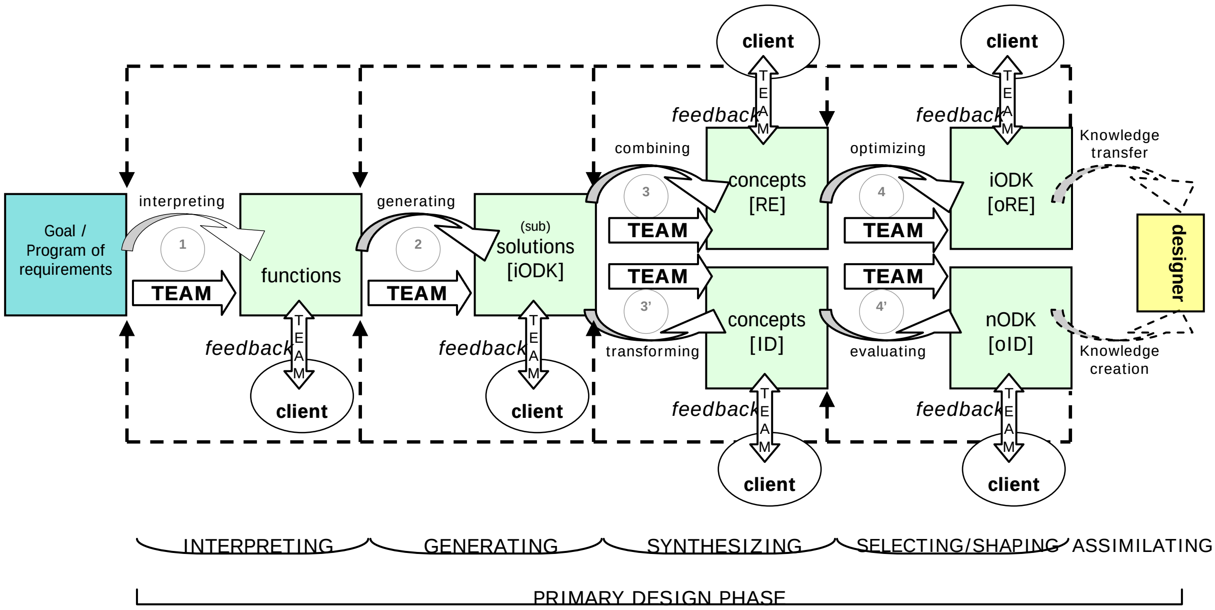


Figure 2. 'ID-methodology' design model; continuous feedback between design team and client additionally structures design process.

Using morphological overviews as a design tool all interpreted functions (step 1, Figure 2) and all generated (sub) solutions (step 2, Figure 3), represented by 'chunks' of object design knowledge, can be structured. During following activity (step 3/3', Figure 2), it is important to understand that integration of initially presented discipline based design object knowledge is something different than plain combination of (sub) results from various abstraction levels. Combination can only lead to redesign (RE), while in literature much referred designer's 'creative leap' is needed for integral concepts (ID). This is the major step in understanding how to work with morphological overviews, and needs 'designerly' [5] attitude. Because concept integration involves *transformation* of design knowledge (C-K theory), it requires design thinking / creativity. Contrary to redesign, the connections design team (members) make between presented (sub) solutions / design aspects in order to produce ID-concepts are subjective, design task and context dependent. Therefore, they cannot be objectified and/or rationalized. This is the reason why 'ID-methodology' can't be automated, even though the structure of morphological overviews makes it very tempting to try (Figure 3).

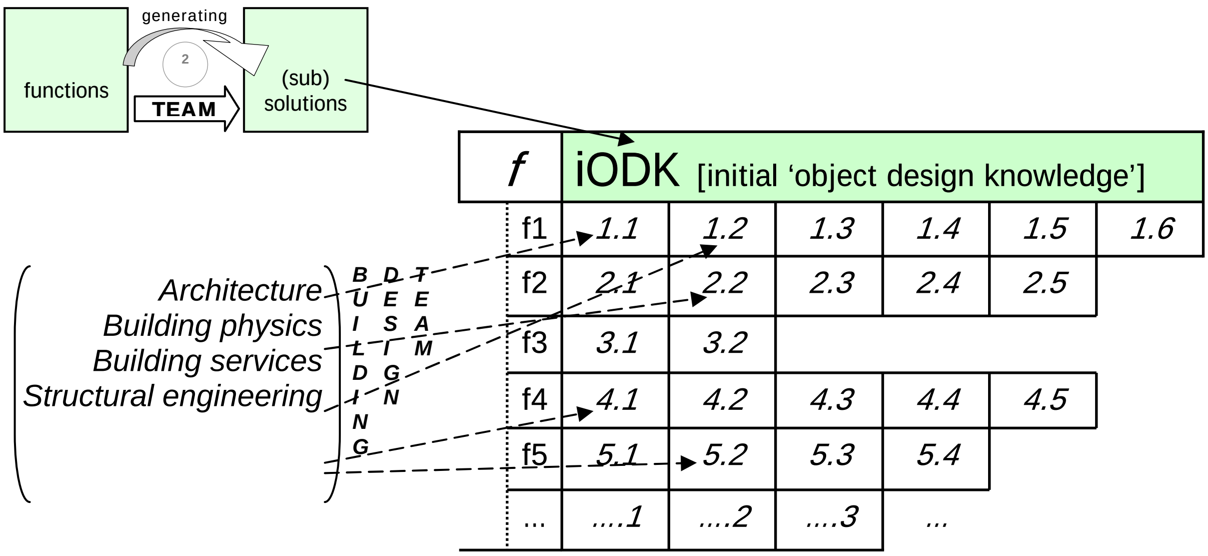


Figure 3. Morphological overviews show the initially available object design knowledge

The essence of 'ID-methodology' is strict separation between synthesising design proposals, being it RE or ID concepts, and selecting suitable ones. This selection step, which once again emphasises importance of interaction between design team and client, represents extension of methodical design model [34] as introduced by Zeiler [37].

DISCUSSION

'Integral design (ID) methodology', based on a reductive (positivism) framework [18,22,31,34] for design teams, in which concepts and knowledge (C-K) [14] constructs (phenomenology) [29] of individual designers are structured, provides suitable ground for creation of conceptual integral building designs. Subsequently, these integral concepts provide potential for creation of new (object design) knowledge. We have tried to build up this chain of reasoning as much as possible on our own observations of design teams during series of workshops with professionals. The development of workshop setting mirrors developments of 'ID-methodology'. Because we don't want this methodology to be yet another set of rigid prescription methods, but a flexible framework that most designers could fit to their own needs and ways of working, we have tried to modify and reduce it to a complexity level that would satisfactorily meet these needs. Taking into account that a fairly large number of new elements were introduced: workshop setting, design team configuration, methodical design, pressurised timeframe, forced client feedback, sustainable comfort systems, new design tasks and above all abstract thinking, a statement from W. Ernst Eder [18], made during one of design conferences, warns: "'If you have to use a new method, under time pressure, on a new problem – you will guaranteed fail!". Bearing this in mind, our framework was largely simplified (instead extended, as was initially expected). The only remaining tool, morphological overviews, however proved to be sufficiently suitable for 'learning-by-doing' explanation of integral approach involving 'ID-methodology'. They do require abstract thinking, for which an adaptation phase and a certain amount of training is needed, but they do provide framework for self-organization of design teams.

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Modelling and optimization of multi-energy source building systems in the design concept phase

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SUMMARY

A growing interest has been addressed to multi-energy systems in buildings that involve the integration of different energy sources to cover the thermal and electrical loads of the building. A modelling approach to multi-energy systems in buildings based on the concept of hybrid energy hub is presented. The model has been customised to be used in the concept phase of the building design, either as a system simulation tool or as a system design optimization tool. Given the prices of technologies and of energy-wares, under a certain set of constraints it is possible to determine the configuration that minimises the capital cost, the primary energy consumption or the life-cycle cost. This approach allows avoiding a burdensome simulation and ranking of a set of different systems. In the end, an application on a case study is provided.

INTRODUCTION

The energy system of a building is the complex of building plants that transform primary (chemical, thermal, solar, wind) and secondary (electricity, hydrogen) energy-wares into energy used to cover the building energy demand.
In recent years a growing interest has been addressed to zero energy homes or buildings that produce more energy than that they consume. The efforts to attain this goal have produced a progressive integration of renewable and conventional energy sources. This means that new technologies, such as solar plants, biomass plants, geothermal heat pumps, co/tri-generators, fuel cells, are spreading rapidly.
The application of these technologies leads to a sort of building that can be referred to as a multi-energy source building, since the thermal, cooling and electrical loads are covered by a mix of energy sources, at least one of them renewable. The aim is to design and manage such systems with the best efficiency [1].
It is necessary to investigate the coupling between a building, characterized by energy demand profiles for heating, cooling and electricity, and a system, characterized by production profiles (solar, wind, electricity, etc…). The study of the integration of different energy sources in buildings has to be the foremost in order to fully exploit the energy savings potential of renewable sources.
Being conscious that the degree of the design effort is greater during the program pre-design and schematic design phases [2], it is of a great importance to concentrate the research activities on the elaboration of a methodology to model and optimize the coupling between energy demand and energy supply in a building at the design concept phase.

MULTI-ENERGY SOURCE BUILDING SYSTEMS

Several examples of multi-energy source building systems can be found in literature, combining cogeneration with solar energy (both thermal and PV) and with wind energy [3], exploiting geothermal and solar energy through solar assisted heat pumps [4], combining CHP with absorption chillers and desiccant cooling [5], exploiting solar energy to produce both heating and cooling [6], exploiting wind energy through a fuel cell stack [7].
One of the main problems of a multi-energy system [8] is the mismatch between energy supply and energy demand. This is especially true for renewable sources. This problem, as for the plants that exploit solar energy to produce cold [6], is usually dealt with the integration of a storage. Typical storage mediums are water (refrigerated or ice), ground, PCM or hydrogen. As an example, in the wind fuel cell hybrid energy system described in [7] electricity surplus provided by a wind turbine is utilized to produce hydrogen that is used, later when necessary, in a fuel cells stack, which also represents a cogeneration system.
The objectives of multi-energy systems are the followings [8]:

- to reduce the primary energy consumption;
- to produce energy on site;

It is out of doubt that in the near future this kind of systems will be used both for new buildings and for renovations.

MULTI-ENERGY SOURCE BUILDING SYSTEM MODELLING

The presented modelling approach to multi-energy systems in buildings is based on the concept of hybrid energy hub, which was developed by Andersson, Fröhlich, Geidl *et al.* [9,10] to model and optimize the multi-carrier energy network of the future [11].
The building is modelled as an energy system that is supplied with different energy carriers and must meet heating, cooling and electrical loads. The coupling between the sources (inputs) and the loads (outputs) is established by a coupling matrix that depends on the technology and on the conversion efficiencies of systems and plants installed. Within the same modelling approach, different levels of complexity are possible depending on the input data and on the coupling matrix entries.
With reference to the optimization of a multi-energy source building system in the design concept phase, the data selected to model flow and conversion of energies within the hub are the design power P and the annual (or seasonal) energy E. Each energy carrier is identified by a superscript (e.g. *e* for electricity, *t* for thermal). The subscript identifies the demand (*out*) or the supply (*in*).
The coupling between energy demand and energy supply can be written as

$$\mathbf{E}_{in} = \mathbf{D}\,\mathbf{E}_{out}$$

where $\mathbf{E}_{in}$ is the vector of energy inputs $[E^e_{in}, E^t_{in}, \ldots]^T$, $\mathbf{E}_{out}$ is the vector of energy outputs $[E^e_{out}, E^t_{out}, \ldots]^T$ and $\mathbf{D}$ is the coupling matrix which depends on systems and plants installed and on their conversion efficiencies.
The same relation can be written for design power, assuming design efficiencies instead of mean seasonal efficiencies.
Each matrix entry d^{ab} accounts for both connection and conversion efficiency between energy carriers. A value of d^{ab} equal to 0 means that no connection between the energy carrier a and energy carrier b is provided by the hub; a value of d^{ab} equal to 1 means that all power of energy carrier a flows into energy carrier b without conversion losses.

Multi-energy source building system modelling application procedure

The application procedure to model a multi-energy source building system is the following:

1) identify the set of available energy sources and the set of building loads;
2) identify the components that can be used to cover the building loads, given the set of energy sources available: this can lead to two different approaches:
   2.1) a *generic hub* which takes into account all conversions (and the relative components) that energy sources can undergo before covering the loads;
   2.2) a *tailored hub* which takes into account only the conversions (and the relative components) that are of practical application and of interest to the building owner;
3) identify all the parameters (such as efficiencies) that can model the performance of the components both at the design condition and at operational conditions, and assign to them a numeric value;
4) define the costs of the energy-wares and of the technologies adopted;
5) perform an optimization to select the best multi-energy system.

MULTI-ENERGY SOURCE BUILDING SYSTEM OPTIMIZATION

Several criteria can be adopted to optimize a multi-energy source building system. Some of them are the followings:
- reduce the running costs;
- reduce the capital costs;
- reduce the amount of energy consumed in a period of time;
- maximize the efficiency of the system (both in terms of energy and of exergy);
- reduce the environmental impact of the building and services;
- maximize the use of natural resources that are available at no cost.

The most appropriate optimization criterion for the design concept of a multi-energy source building system in the design concept phase seems to be the economic approach. It has been adopted in the case study presented. Two different procedures can be outlined.
The first one is based on the net present value calculation. The procedure steps are the followings:

1) set a reference configuration (converters, allowable conversions, power flows, design powers of the converters) of the system;
2) set the period (number of years) for the investment analysis;
3) determine the capital cost difference and the running cost difference between alternative scenarios and the reference case;
4) calculate the net present value for each scenario based on capital and operational cost differences over the fixed period.

As regards the design alternatives, usually after a greater capital cost in the first year, savings on running costs are expected. The optimal scenario is the one that has the greater net present value (this comparison can be done if all net present values are calculated assuming the same period of time. The determination of the internal rate of return (IIR) is also useful.

The second procedure tends to minimise a yearly cost based on the sum of one year running costs and of the investment costs divided by the years of expected life of each component. This procedure can be implemented without defining a reference configuration of the system. In any case, the lifetime of components installed must be defined in order to compare capital and running costs.

APPLICATION TO A CASE STUDY

The methodology presented has been applied to the design of the energy system serving the Valcasotto castle in Piedmont.
A tailored energy hub procedure has been carried out as follows:

1) The available energy sources are: wood (*w*), LPG and electricity (*e*) from the network; the building thermal and the electric load are to be covered. Four demand scenario (the first one considering thermal energy only) have been identified in terms of design power $\mathbf{P}_{out}$ and annual demand $\mathbf{E}_{out}$ and are reported in Table 1.
2) The converters that can be used to cover the building loads are a wood boiler (WB), a boiler (B), an internal combustion engine (ICE) and a steam turbine (ST). The internal combustion engine and the steam turbine provide both electricity and heat. The heat-power ratios are respectively 1 and 2. The internal combustion engine is fed either by wood through a wood gasifier (WG) or by LPG. The steam turbine is considered to be an ensemble of a wood boiler and a steam turbine. The schematic representation of the energy hub considered is designed in Figure 1. The ε coefficients represent the load fractions covered by a certain converter (the superscript identifies the energy carrier, the subscript the converter): for example ε^{t}_{WB} represents the fraction of the thermal load covered by the output of the wood boiler.
3) The efficiencies η of the components (divided into thermal and electrical for cogeneration converters) are reported in table 2 including both design efficiencies and yearly mean efficiencies.

Table 1. Components of $\mathbf{P}_{out}$ and $\mathbf{E}_{out}$ vectors (respectively in kW and MWh/year)

Energy	Scenario 1		Scenario 2		Scenario 3		Scenario 4	
	$\mathbf{P}_{out}$	$\mathbf{E}_{out}$	$\mathbf{P}_{out}$	$\mathbf{E}_{out}$	$\mathbf{P}_{out}$	$\mathbf{E}_{out}$	$\mathbf{P}_{out}$	$\mathbf{E}_{out}$
Thermal	1857	6169	1284	4267	929	3085	573	1902
Cooling	0	0	0	0	0	0	0	0
Electricity	0	0	573	1902	929	3085	1284	4267

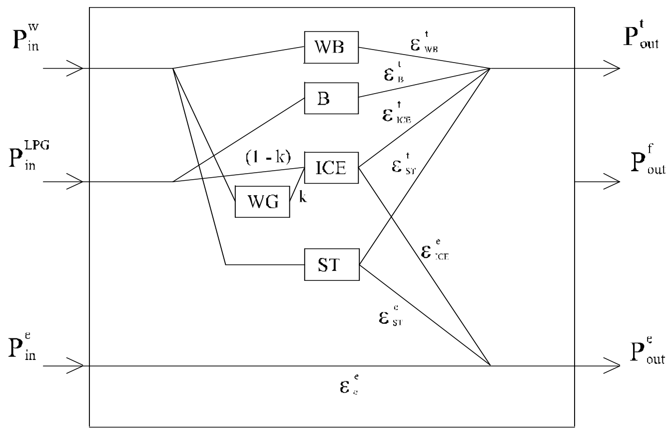

Figure 1. Schematic representation of the Valcasotto tailored energy hub

Table 2. Design efficiencies and mean annual efficiencies of the converters

Design efficiency		Mean efficiency	
η_{WB}	0.75	η_{WB}	0.65
η_{B}	0.95	η_{B}	0.85
η_{WG}	0.75	η_{WG}	0.65
η^{t}_{ICE}	0.40	η^{t}_{ICE}	0.30
η^{e}_{ICE}	0.40	η^{e}_{ICE}	0.30
η^{t}_{ST}	0.60	η^{t}_{ST}	0.45
η^{e}_{ST}	0.30	η^{e}_{ST}	0.23

After having defined the efficiencies of each converter η (for example η^{t}_{ICE} is the thermal efficiency of the internal combustion engine) the input powers at the entrance port of the hub can be determined summing up the contributions of each converter as follows:

$$
\begin{aligned}
P^{w}_{in} &= \frac{1}{\eta_{GCL}}\varepsilon^{t}_{WB}P^{t}_{out} + \left(\frac{1}{\eta^{t}_{ST}}\varepsilon^{t}_{ST}P^{t}_{out} + \frac{1}{\eta^{e}_{ST}}\varepsilon^{e}_{ST}P^{e}_{out}\right)/2 + \left(k\frac{1}{\eta^{t}_{ICE}\eta_{WG}}\varepsilon^{t}_{ICE}P^{t}_{out} + k\frac{1}{\eta^{e}_{ICE}\eta_{WG}}\varepsilon^{e}_{ICE}P^{e}_{out}\right)/2 \\
P^{LPG}_{in} &= \frac{1}{\eta_{B}}\varepsilon^{t}_{B}P^{t}_{out} + \left((1-k)\frac{1}{\eta^{t}_{ICE}}\varepsilon^{t}_{ICE}P^{t}_{out} + (1-k)\frac{1}{\eta^{e}_{ICE}}\varepsilon^{e}_{ICE}P^{e}_{out}\right)/2 \\
P^{e}_{in} &= \varepsilon^{e}_{e}P^{e}_{out}
\end{aligned}
\tag{1}
$$

Assuming a linear correlation between inputs and outputs, this can be rewritten in a matrix form as

$$
\begin{bmatrix} P^{w}_{in} \\ P^{LPG}_{in} \\ P^{e}_{in} \end{bmatrix} = \begin{bmatrix} \frac{\varepsilon^{t}_{WB}}{\eta_{WB}} + \frac{\varepsilon^{t}_{ST}}{2\eta^{t}_{ST}} + k\frac{\varepsilon^{t}_{ICE}}{2\eta^{t}_{ICE}\eta_{WG}} & 0 & \frac{\varepsilon^{e}_{ST}}{2\eta^{e}_{ST}} + k\frac{\varepsilon^{e}_{ICE}}{2\eta^{e}_{ICE}\eta_{WG}} \\ \frac{\varepsilon^{t}_{B}}{\eta_{B}} + (1-k)\frac{\varepsilon^{t}_{ICE}}{2\eta^{t}_{ICE}} & 0 & (1-k)\frac{\varepsilon^{e}_{ICE}}{2\eta^{e}_{ICE}} \\ 0 & 0 & \varepsilon^{e}_{e} \end{bmatrix} \begin{bmatrix} P^{t}_{out} \\ P^{f}_{out} \\ P^{e}_{out} \end{bmatrix}, \tag{2}
$$

where the products $\varepsilon^{e}_{ICE,ST}P^{e}_{out}$ and $\varepsilon^{t}_{ICE,ST}P^{t}_{out}$ are related, in the case of a cogenerator, by the relation

$$
\varepsilon^{t}_{ICE,ST}P^{t}_{out} = \varepsilon^{e}_{ICE,ST}P^{e}_{out}\frac{\eta^{t}_{ICE,ST}}{\eta^{e}_{ICE,ST}}, \tag{3}
$$

that must be included as a further equation to (3) in order to model the hub performance. The optimisation of the hub can be performed by minimizing the objective equation

$$
C_{y} = f\left(\mathbf{E}_{\mathrm{in}}, \mathbf{P}_{k}\right) \tag{4}
$$

which takes into account the energy costs in a period of time and the capital cost of the converter devices installed. It is therefore necessary to determine the design power of each converter $\mathbf{P}^{k}$ and then (5) can be rewritten as:

$$\min C_y = \min\left(c^{\mathrm{w}} E_{\mathrm{in}}^{\mathrm{w}} + c^{\mathrm{LPG}} E_{\mathrm{in}}^{\mathrm{LPG}} + c^{e} E_{\mathrm{in}}^{e} + \frac{c_{WB}\, P_{WB}}{y_{WB}} + \frac{c_{B}\, P_{B}}{y_{B}} + \frac{c_{ICE}\, P_{ICE}^{e}}{y_{ICE}} + \frac{c_{WG}\, P_{WG}}{y_{WG}} + \frac{c_{ST}\, P_{ST}^{e}}{y_{ST}} \right) \quad (5)$$

where cost for energy consumed c, costs of converters (in terms of design power) c_k are resumed in tables 3 and 4. The expected number of years is fixed to 20 years for each converter. The minimisation of this yearly cost function has been performed by means of a commercially available reduced gradient method algorithm.

The problem is then to find the values of ε that minimize the cost function

$$\varepsilon_i^{\alpha} : \min \quad f\left(\mathbf{E}_{\mathrm{in}}, \mathbf{P}_{\mathrm{k}}\right)$$

under the constraint of the hub:

$\mathbf{E}_{\mathrm{in}} = \mathbf{D}\ \mathbf{E}_{\mathrm{out}}$; $\mathbf{P}_{\mathrm{in}} = \mathbf{D'}\ \mathbf{P}_{\mathrm{out}}$

$\mathbf{E}_{\mathrm{in}} \geq \mathbf{0}$; $\mathbf{P}_{\mathrm{in}} \geq \mathbf{0}$

and the constraints relative to the ε coefficients:

$$0 \leq \varepsilon_i^{\alpha} \leq 1 \ \ \forall\, \alpha \qquad , \forall\, i \in \{GCL, GC, ME, \ldots\}$$

$$\sum_i \varepsilon_i^{\alpha} = 1 \ \ \forall\, \alpha$$

$$\varepsilon_{ME,TV}^{t} P_2^{t} = \varepsilon_{ME,TV}^{e} P_2^{e} \frac{\eta_{ME,TV}^{t}}{\eta_{ME,TV}^{e}}$$

Table 3. Cost structure c of the energy wares

Energy-ware	[€/kWh]
Wood	0.014
LPG	0.060
Electricity	0.150

Table 4. Cost structure c_k of the energy converters

Energy converter	[€/kW]
Wood boiler	200
Boiler	100
Wood gasifier	50
ICE	900
Gas turbine	1100
Electricity	100

The results of the optimization of each scenario are shown in figure 2.

The model can be used to simulate, for a given demand scenario, other hub structures. In the case of scenario number 2, it is possible to determine the primary energy that must be supplied using the internal combustion engine feed by gas via the wood gasifier or the internal combustion engine fed by LPG instead of the wood gasifier and the steam turbine. This can be done by fixing the values of coefficient ε instead of letting them to freely change as in the optimisation. Those two alternative scenarios simulated are presented in figure 3 and it can be verified that the resulting values of C_y are higher than that of the optimized configuration.

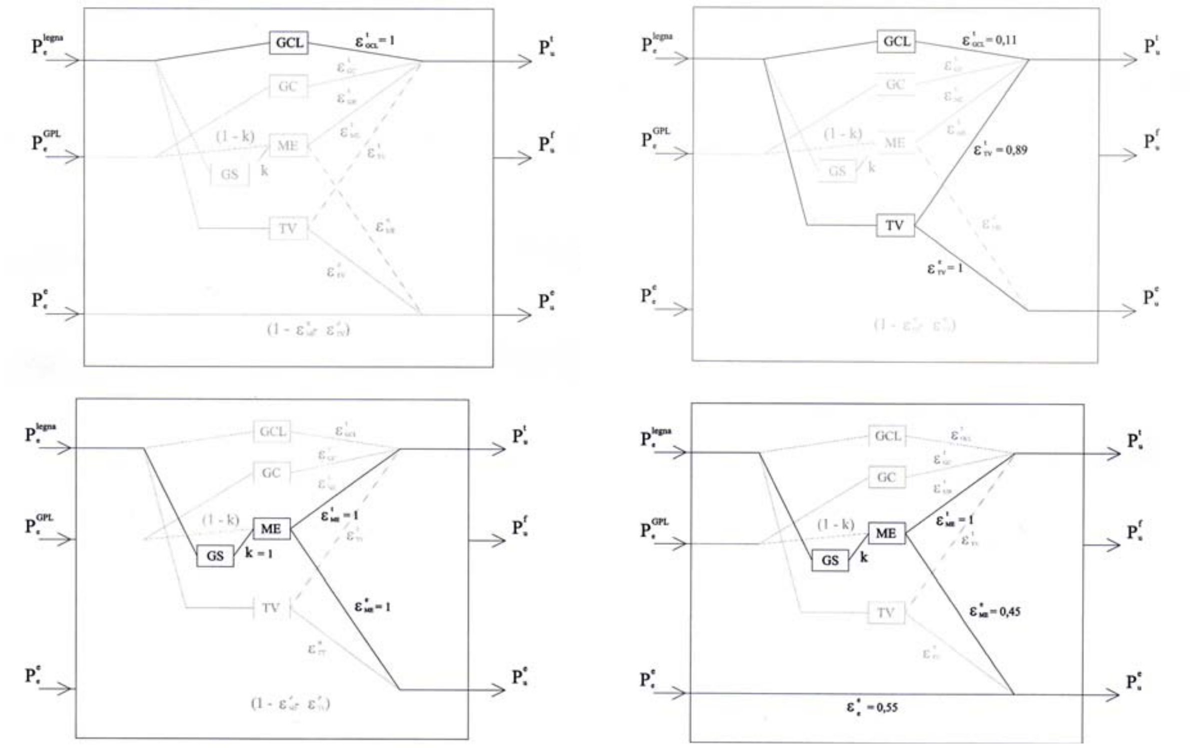

Figure 2. Schematic representation of the Valcasotto tailored energy hub optimized for four different demand scenarios (1 to 4 from right to left)

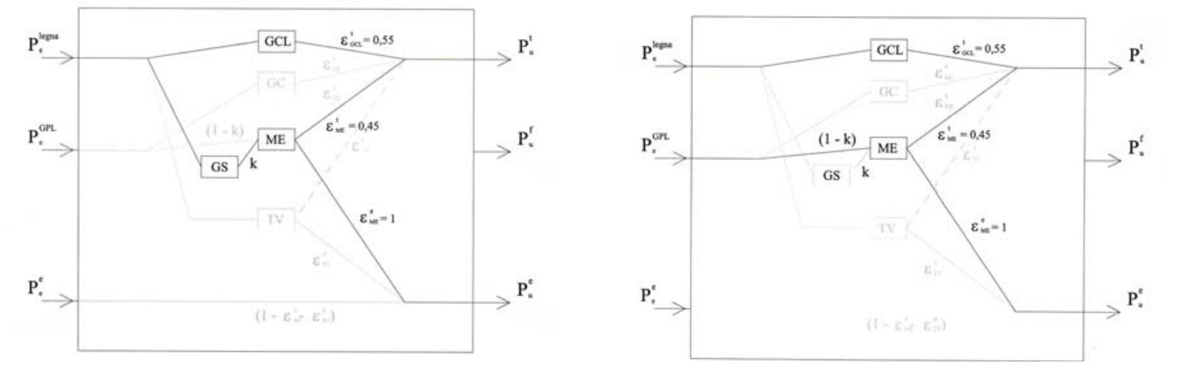

Figure 3. Schematic representation of two further simulation of the Valcasotto tailored energy hub for the second demand scenario

DISCUSSION

A modelling approach to multi-energy systems in buildings has been presented. It is based on the concept of the hybrid energy hub has been customised to be used in the design concept phase. The model is quite simple and allows analysis to be performed in presence of design power and annual or seasonal energy demand data only.

Such a model meets the requirements of simplicity that characterize the design concept phase, but a factor of uncertainty is represented by the choice of the values of the efficiencies. Values of mean seasonal efficiencies greatly affect the results and appropriate values of these properties are difficult to determine *a priori* and must be based on the consultant experience. Further research activity is currently carried out to overcome this drawback; values of efficiencies dependent on design power installed and on the part load ratio may be integrated into the model.

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Integral approach for adaptable indoor comfort; Building and occupants follow the sun

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SUMMARY

The focus on the needs and drives for adaptation of the building automatically leads to changing needs and demands of the occupants of the building. Building should really take care of its occupants and show adaptable behaviour and reaction to the changing outdoor environment during the day. Design for adaptability should start with the occupants needs for comfort and indoor air quality. These are partly influences by the changing environmental forces as wind and sun. Weather predictions and the aggegated voting of users about their thermal comfort, should be the leading parameters to adaptable comfort and the adaptable building.

INTRODUCTION

Building design is a fascinating ,it starts with a blank sheet and ends with a building with spaces and materials. Since the future is unknown, it is only possible to paint scenarios, possible futures, buildings have to meet. A building usually has to comply with more demands than mentioned in the list of specifications, for example because they were overlooked by the client, or necessary changes during the design process. In the continous analysis, there are hard to quantify demands, such as comfort, beauty, social acceptance and safety. The designers has to pay due attention to all aspects, how trivial or unexpected they may seem.

Inappropriate formation of the design team may result in ineffective designprocess and solutions. Many problems emanate from a lack of integration between architectural design and design of indoor climate. After the industrial revolution, the natural relationship between design, construction and the built environment disappeared and has been replaced by a complex system of decision making, complex legislation, a subdivision of a whole integrated building into subsystems and disciplines in the construction process.

METHODOLOGY

The making of the built environment has become complex. In the conceptual design phase, in order to create conditions that assure a built environment that gets better, the ingenuity of the whole design team existing of different disciplines should be used, not only architecture. The quality of the team should be combined with a well considered process of decision making. Techniques are selected and put together by a team in an integral design process. In addition to the application of proven construction methods, the integral approach demands an attitude of openness and appreciation of the other participating disciplines and their positions.

This approach makes it possible to combine and develope different kind of aspects in interaction to each other. During the design proces participants and their decisions are structured at several levels of decision-making; the infill-level, the support-level and tissue-level. On each level there has to be made a balance between the performance of supply and demand for the building during the life-cycle. The basis of this *'level-thinking'* is that of the Open Building [1]. As often such new ideas take a long time before being implemented. The Osaka Gas NEXT 21 Project in Shimizudani, Tennoji-ku, Osaka City, Japan, realized in October 1993 is a very nice example of the realisation of the open building approach, see figure 1.

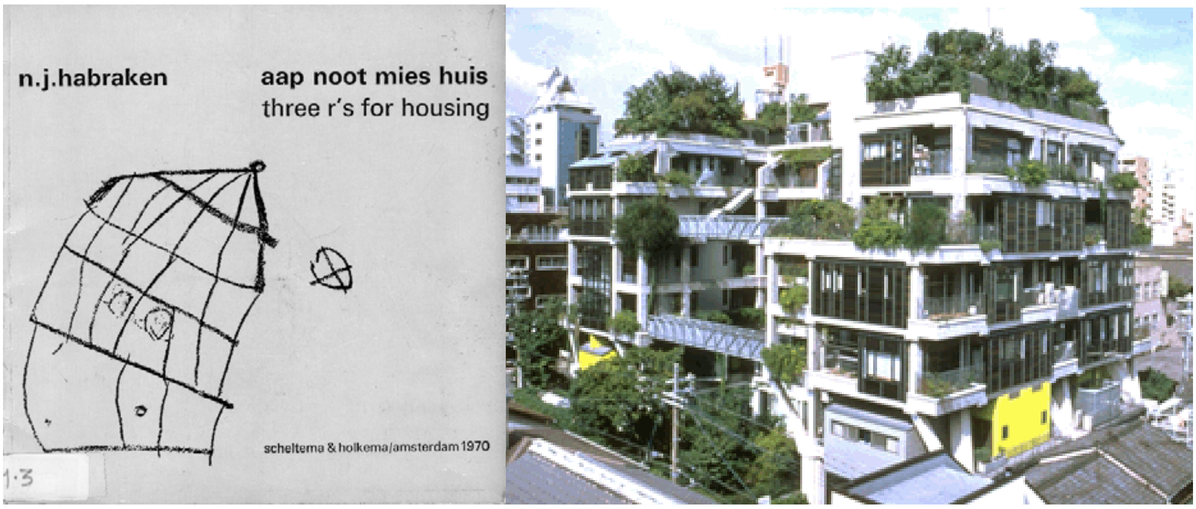


Figure 1: Publication by Habraken, NEXT project interesting realisation of the ideas

Open building is primarily intended as an organised way of responding to the demands of diversity, adaptability and user involvement in the built environment. In open building the built environment is approached as a constantly changing product engendered by human action, with the central features of the environment resulting from decisions made at various levels.

A central idea in Open Building is to respond to the various needs of individual users through the phasing of the design and implementation process. In order to provide prospective occupants with the opportunity to influence their building, the elements decided by the occupants must be easy to change. Thus adaptability is not merely a means for modifying the dwelling during use; it is first and foremost a strategy for enabling the fulfilment of individual wishes without compromising. Thinking in levels is the basic Open Building principle.

Techniques are available to help further organize the complexities of environment as a subject into its component part and wholes. Dissecting a subject into its intrinsic parts and then shifting to how those parts are organized into a whole, allows access to content as well to the way parts connect and, therefore, to a full definition. To characterize connections a hierarchy is needed to out line sequence and progression and to illustrate part-whole relationships.

During design support, it is important to transfer the essentials of the proposed structures and mechanisms, without overloading other member of the design team with unwanted details. This information control can be achieved by use of abstraction. So far, many building teams have been sending their partners detailed drawings, thus relying on the addressees to make the necessary abstraction themselves. With the increasing use of product information models, it is now possible to incorporate multiple abstraction levels in the design representation. Abstraction is the mapping from one representation of a problem to another, which preserves certain desirable properties and reduces complicity [2]. Abstraction is the selective examination of certain aspects of a problem. The goal of abstraction is to isolate those aspects that are important for a particular purpose and suppress those aspects that are unimportant [3]. This enables representations to take an appropriate (abstract) form that matches the needs of the design specialist, thus saving much time and confusion.

Throughout the different levels of abstraction, the description of the building design gradually becomes more and more detailed. The various levels of abstraction should be considered as representations of a particular view on the total information available for a design.

This integrated design model must:

- be able to distinct related information,
- support distinctions related to the different levels of abstractions (views) by being structured into corresponding sub-models,
- ensure the satisfaction of consistency and completeness constraints linking different levels of abstraction in the design process.

Methodical Design [4,5,6] can be described at the conceptual level as a chain of activities which starts with an abstract problem and which results in a solution.The original methodical design process is extended from three to four main phases, in which eight levels of functional hierarchical abstraction can be distinguished, see figure 2.

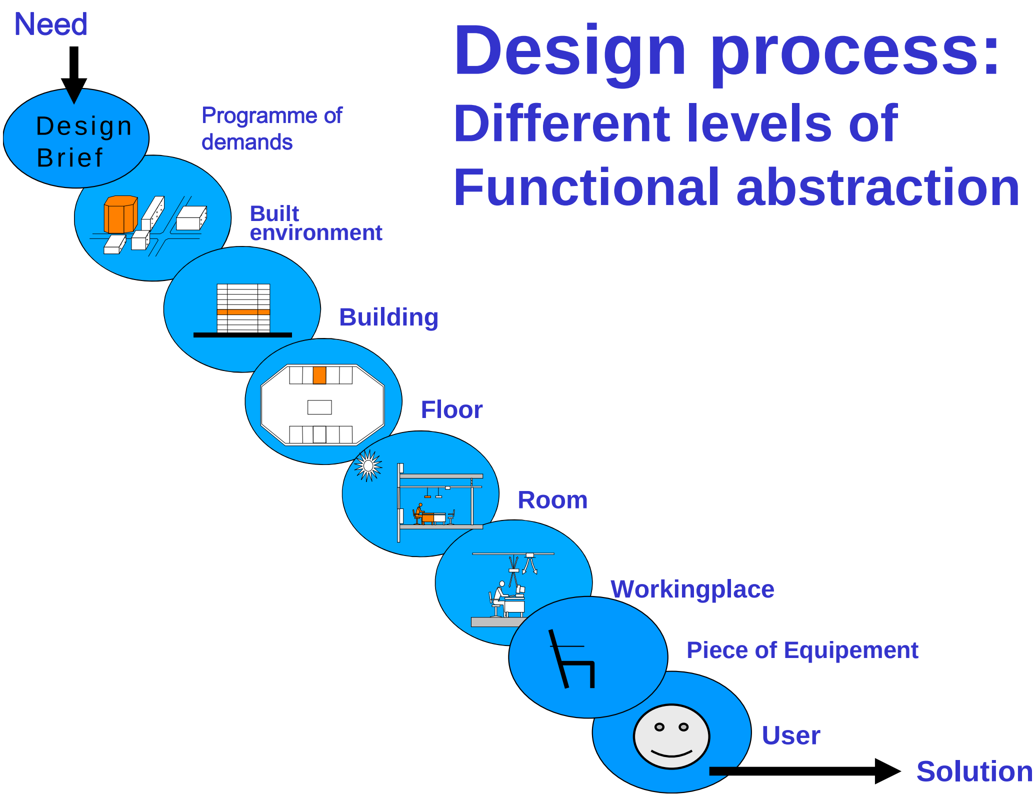


Figure 2: Functional Hiearchy Methodical Design

Hierarchy can be used as a simplifying organizational tool to order and relate the parts of a subject in range-large to small most important to least, etc., used in a neutral sense where all parts are unique and important. Hierarchy theory in this way, frees minds from the analytical mode without requiring them to reject it and provides, as well, glilmpses of a wider, more diverse world. Hierarchies suggest an interplay between parts and wholes and between the two basic modes of thinking:

- Analysis: separation of a whole into its component parts, the examination and study of each part;
- Synthesis: the composition or combination of parts or elements so as to form a whole.

This organizational structure provides a way to study and design complex issues. With regard to the built environment the parts human, built and natural environments can be analysed separately and then synthesizing the parts together to form a composite definition to obtain more meaning like a word integrated into a phrase and then into a sentence.

To work effectively with methodology, practitioners should learn to work with, and understand, the role of methodology in the building design process. Often means and goal are mixed up. More and more the insight is growing that it is not the building to be designed that should be central but the needs of the humans for which the building is intended.

FOLLOW THE OCCUPANTS NEEDS

For about 40 years there is an evolution of thinking and research in this way, strongly related to the integration of disciplines in proces and product. The most important new idea is taking man in his environment as the departure point, variable and criterion. The comfort of the occupants becomes leading.

The first guidelines for thermal comfort in the Netherlands were developed in the late seventies and eighties. They were based on the theory of Fanger and resulted in the PMV-PPD relationship, which predicts the percentages of dissatisfied occupants. Extensive field research however by de Dear showed that people have various adaptation mechanisms with most important people's expectations of the building's climat, based on the actual outdoor temperature. New thermal comfort models based on the human adaptability were developed over the past years. Applying adaptive thermal comfort, a distingtion was made between different types of buildings, usage and climatic circumstances. An important feture to distinguish between the differences here is the possibility of individual control [7].

The representation of situations with multi end-users, multi-individual control, was realized by developing an individual voting sytem. This voting system implied that every user in a thermal zone could enter his or her vote, warmer or colder, within a voting period, e.g. one hour, while seeing the aggregated voting of other users in his zone at the moment of voting.

The new user behaviour control strategy was implemented in a BMS(Building Management System). This BMS was extended with an external real-time information system to improve energy and comfort control.

To further enhance comfort and at the same time reduce energy consumption of buildings, new control technology is needed. Technology in which the end-user behaviour and preferences are integrated into a responsive ambient surrounding. Improvement of the energy consumption is made possible by agent-based systems for energy management in buildings, as well as new possibilities occur to enhance individual comfort of occupants.

On this philosophy, design teams can develop buildings with contrastful attributes for temperature, wind, humidity, daylight and smells. Buildings similarly gain a contrastful microclimate that changes in accordance with the time and the outdoor conditions, as well as a more pleasant interior, a space that will endure longer than we may imagine.

INTEGRAL APPROACH

The search for a building that functions better within its surroundings forces architecture into every closer company with engineering. These buildings take advantage of the prevailing climatic conditions in combination with the familiar physical principals of natural draught, sunlight, wind, precipitation etc. The climatic conditions of the location are highly determinative for the form of the building, thus producing a climate-related and regionally distinctive architecture. Intensive collaboration with technical consultants takes place from the outset in these designs.

In 2000, the Royal Institute of Netherlands Architects, BNA, TVVL and the Delft University of Technology have participated in a research project called Integral Design. Observations in the construction industry show a fractionated process by different parties achieving their own aims, working on the same building. Different cultures and different traditions, many times conflict with the common aim of completing the building. The initiators of the Integral Design project had the vision that breaking the barriers of different trades may be the first step to a better built environment. It was named Integral Design and unfolded ways to implement, investigate, teach and learn the integral approach (2). The Integral Approach in relationship to Integral Design has to deal with different scale-levels and different 'kind' of aspects. To give the range of aspects which are related to the Integral Approach, not in the way to give a definition, in the way of describing, shows the interweaving meachism between the several apsects and the area and approach which belongs to the word 'Integral'[8]

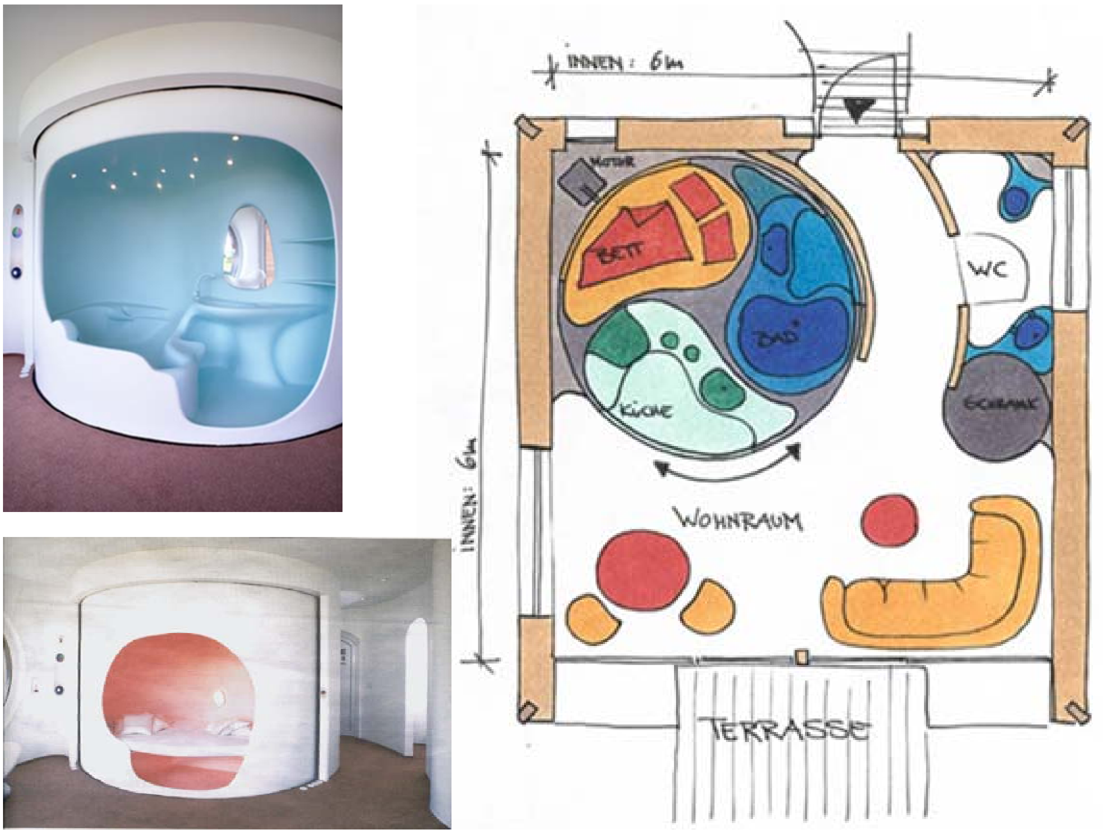
The 'umbrella-aspect' is the flexiblity and bandwidth in process, designmethodology, constructionmethodology and facility-management. The integral approach will guide, interdependent to the value-frameworks, will control and correct and will leave room to the specific method or sollution for the several participants during the total process.

The Integral Approach is conceptual and therefore inhabits the several 'specific methods' such as for instance Strategic Design, Sustainable Design and LCA-Design. The Integral Approach is related to the longer term for different (time, site, process and program) settings; a sustainable approach [9][10], [11].

The integrated building design process is one which considers all aspects of a building, its environment and life cycle, and is undertaken by a team which includes all relevant professionals and stakeholders working together throughout the process rather than sequentially and independently.

INTERNAL ADAPTABILITY

Internally, structures satisfy and symbolize human spatial needs, wants and values; externally they provide and symbolize a protective envelope, an interrelationship with the outside environment. The external enclosure can be considered the “third skin” of the human body and its comfort (light, fresh air, optimum temperatures, etc.) is based upon human senses and either provided for by natural means (windows for light, solar energy for heat, shade for cooling, etc.) or by artificial means (electric lights, heating, air conditioning, etc.). The project ‘rotor haus’of Colani shows how internal adaptability of use of specific functional rooms is possible. Nevertheless because of the major influences of the sun, outside adaptability is more important in relation to energy and esthetics.



Picture 1 Rotorhaus, L.Colani, 2004

OUTSIDE ADAPTABILITY: ADAPTIVE BUILDINGS TO THE SUN

The external character of a building form is shaped not only by internal requirements, but also by fitting a building to its elusive environmental context. Designing with local climates and sun orientation, with similar construction methods and materials, can increase human comfort and reduce energy consumption. Building, indeed, must consider and respond to its built environmental context but also to its natural environment. Technologies should resolve conflicts between what the natural environment provides and human needs, sustenance and comfort.

To let the building react on the thermal comfort demands, it is effective to let the building react to the path of the sun. By letting rotate the building, its sun protection or both, with the turn of the sun the heat of the sun can be left outside the building or when needed let into the building. Differnet solutions were already realised in built projects. But there are more solutions to make buildings adaptive to the sun and the comfort needs of the occupants. Some of the realized projects show completely different ways of solution.

Figure 3. Draai test woning ECN Petten 2002, Waterwoning H.Hertzberg, Middelburg 2002 and Heliotropp, Rolf Dish, Freiburg, 1994

Figure 4: Gemini Haus, Erwin Katenegger, Steiermark, Oostenrijk, 2001

Figure 5: Ellesse International, Perugia, Italië,

If a building is a working whole, a system, we can distinguish subsystems: the load bearing construction, the envelope, inner partitions, HVAC and MEP installations and many more, as can be seen clearly in figure 4 the Gemini Haus designed by Erwin Katenegger and built in Steiermark, Oostenrijk, 2001. Instead of such a drastic and complete approach there are also solutions which are less drastic e.g. the headquater of Ellesse International in Perugia. In this project only the sunprotection is moving around the building and following the sun. Still the effect of such an adaptation to the sun gives a interesting dynamic architectural appearance. It is no longer a dream; A building that can anticipate on an unknown future consisting of building parts with different cycles of change, co-ordinated, yet uncoupled, designed by different disciplines, collaborating in a team, serving the client as well as the end user.

CONCLUSION

For the focus on the needs and drives for adaptability building comfort a process aproach is needed in which the thinking of the architect designer is linked in the design-process itself with the engineer. Support both parties with information on the tasks and decisions of the other party and at the same time supplying an explanation of this information will greatly enhance understanding performance of combined efforts. The structuring of the communication of the process between architecture and consultants is based on abstraction, for instance the exchange of abstract descriptions of a design. Through the different levels of abstraction, the building model is gradually described in an increasingly more detailed manner. Thus decomposing complex design-tasks into manageable size problems.

The integrally working design team not only designs the building, it designs the design process as well. Continuous adaptation of the design and final decision making need to be balanced. Thus the design process is cyclical, spiralling up towards the final plan. The plan with its focus on internal and outside adaptability also determines the possibilities and impossibilities of construction, maintenance and management, It is therefor important to inform and hear the construction partners and the parties .responsible for running the building. It is quite likely that a certain team will never operate again in the same configuration. It is therefor advisable to report on and evaluate the process in order to gain from the accumulated experience and techniques applied.

ACKNOWLEDGEMENTS

The research is embedded in the KCBS (Knowledge Centre Building Systems TNO-TU/e) . The current proposal fits within the research theme “Sustainable energy systems”. One of the objectives of this theme is to provide tools, knowledge and procedures for integrated design processes which lead to innovative applications of comfort systems for building designs with a balanced attention to the value systems of the building occupants, building owner, society and environment.
KCBS, Kropman and the foundation PIT support this research

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Energy Autarky of the Monte Rosa Cabin – A Challenge for the Building Services Engineering Concept

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SUMMARY

Close to Zermatt (Switzerland) the new Monte Rosa mountain cabin is going to be built in 2008 at 2810 metres above sea level. This hut is designed to accommodate 120 mountaineers. Furthermore, it is going to mark a milestone in high alpine building presenting an attractive, unconventional architecture combined with both a surpassing comfort and a high degree of energy autarky of over 90%.

Of course several measures have to be taken to achieve such a high degree of autarky while always keeping a limited budget in mind. Apart from the technical systems, which are based on existing technology (combined heat and power unit, photovoltaics, thermal collectors), there will be an innovative wastewater treatment and, first and foremost, an ingenious energy management. Only an optimised operation taking into account external conditions such as weather forecast and anticipated occupancy schedules is likely to achieve the demanding goal of the high degree of autarky. Detailed thermal simulation is used on the one hand to dimension the building services engineering systems and on the other hand to develop the best energy management strategies.

This is a joint project of the Swiss Alpine Club (SAC) together with the Swiss Federal Institute of Technology in Zurich (ETH), while the energy strategy and building services technology concepts are worked out by the Centre for Integral Building Technology (ZIG) at the University of Applied Sciences of Central Switzerland, Lucerne (HTA).

Figure 1. The new Monte Rosa cabin amidst impressive glaciers showing the Matterhorn in the background (computer visualisation by ETH).

INTRODUCTION

The new Monte Rosa cabin is planned in the context of the 150-year anniversary of the ETH in a joint project with the SAC. The hut is situated 2810 metres above sea level amidst pure nature, surrounded by the Gorner, Grenz, and Monte Rosa glaciers and about 10 kilometres away from the world-known health resort Zermatt. The Monte Rosa cabin is one of the best-known and most visited alpine huts of Switzerland and the starting point for several mountain hikes.

For this new building the engineers aim high. The mountain cabin shall offer an unheard of comfort for such a mountain shelter. Rooms will be heated and mechanically ventilated, the hut will be supplied with enough fresh water such that the visitors can enjoy a hot shower and the wastewater is going to be purified on-site to be reused as greywater. All of the above-average comfort shall be provided without negative effects on the hut's required energy supply. Indeed the hut will feature an energy autarky of 90% meaning that 90% of needed energy is obtained locally from renewable sources.

The concept worked out in detail by HTA incorporates technologies and procedures by which the high requirements concerning comfort, energy demand, and autarky can be achieved – all the time bearing in mind that the budget is limited. Thermal simulations turned out to be of crucial importance to both reach the high planning confidence and permit to establish that the set goals can be achieved.

This paper offers an overview of the aims, the approaches, and the proposed solutions to building services engineering.

METHODS

The simulation package IDA ICE 3.0 [1] is used to set up a very detailed simulation model both of the building and the energy and building services system based on the formulation of the energy strategy and building services engineering concept described below. This model takes into account the climate data, the geometry, the physical properties of the building envelope, all components of energy supply and building services, as well as the thermal loads of persons, equipment, and lighting. The entire system, room temperatures and conditions, as well as the energy flows are being calculated dynamically using hourly steps over a whole year.

These simulations pursue three main aims:

- Represent the actual project status,
- Identify the important factors and system sensitivity on these,
- Optimise the building envelope, the building services, as well as the operation (energy management).

At the same time the investment costs are calculated in a second computing model since the cost is a dominant factor for project realization. This second computing model integrates the investment costs and finds how the energetic aim of 90% autarky are reached at the lowest cost.

Starting from the first design solution, different scenarios with varying exterior conditions are calculated to analyse the stability and robustness of the technical solution examining:

- How does the system react to changed climate conditions (e.g. fewer sunny days than expected)?
- How does the system react if the number of visitors exceeds expectations?
- How does the system react to the failure of single components (system redundancy)?

The following chapter comments on the results of the first phase of the project "Definition, dimensioning, and optimisation of both energy and building services technology". The second phase of the project "Energy management" is currently subject to ongoing work at the HTA.

RESULTS

Water

Water supply and wastewater treatment are top-ranking questions considering the hut's location far away from any infrastructure [4]. On top, the water demand surpasses the average demand of mountain shelters due to the elevated comfort requirements of the cabin (hot water showers, toilets with water flushing).

The calculations show that even using water saving devices, 219 m^3 of fresh water are needed annually. This amount can only be won by catching the melt water around the hut. At the same time it is essential to take into account that melt water accumulates primarily during the summer months (typically during two months only). Therefore the melt water needs to be collected and stored until the next summer. Should the water tank not be built high enough above the hut, a booster pump is necessary. Although the quality of this water is quite good in general, it cannot be directly used as drinking water.

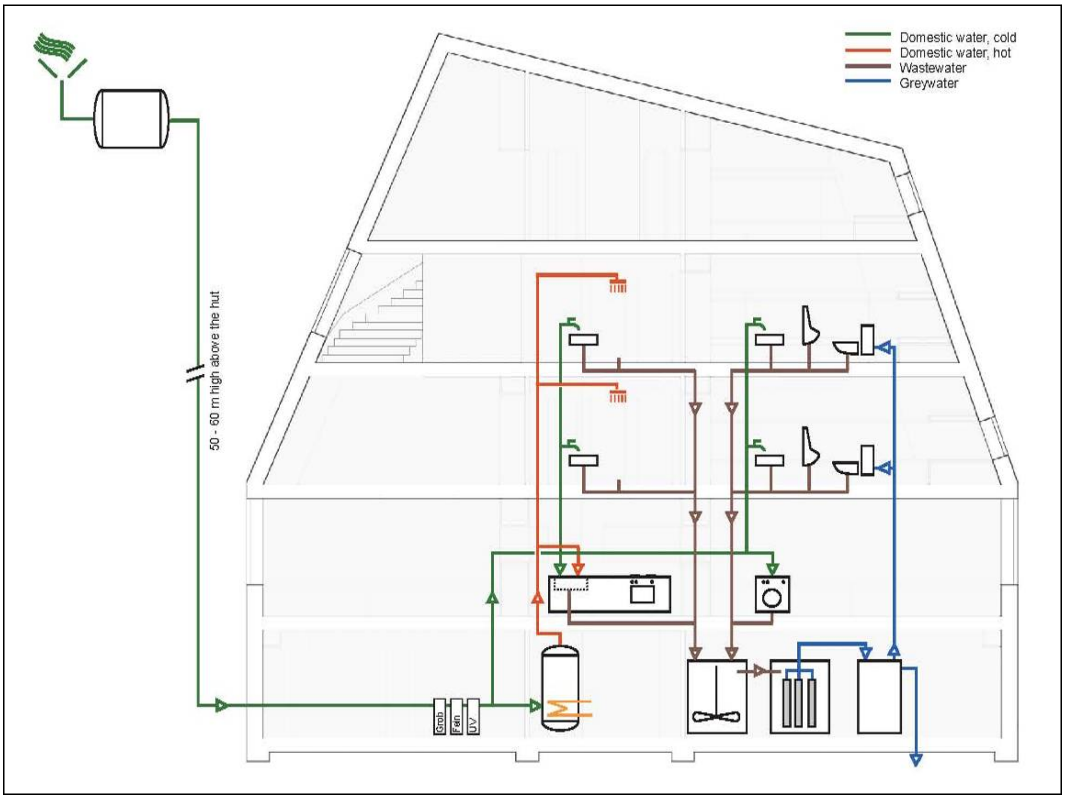


Figure 2. Concept of water supply and wastewater treatment.

After suitable treatment the melt water is used for showers, for cooking, for personal hygiene, and laundry. The toilets are flushed with greywater that originates from the wastewater treatment system. This system is new for such sites and based on the principle of microbiological sewage plants [2]. Tests on the nearby mountain *Hohtälli* show very promising results.

The water balance presents itself as follows:
Supply water: 219 m^3/a, thereof 120 m^3/a hot water
Greywater: 100 m^3/a
Wastewater: 320 m^3/a

Figure 3 shows the water demand prediction over the year, the expected melt water volume, and the resulting water level at the water tank. Clearly recognisable are the weekends with high water demand and the peak weekends with the maximum demand (about 3500 l/d which equals 30 litres per guest). Well recognisable is the fact that the critical phase of storage is reached in early summer just before the melting begins. Depending on the effective water level at the water tank it is generally necessary to save water in spring, which is easily and most effectively done by shutting off the showers.

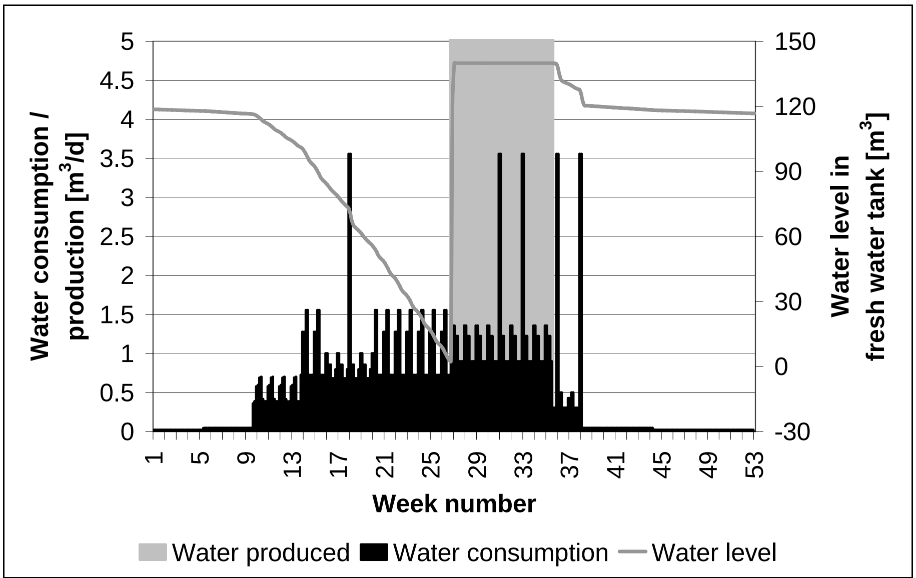


Figure 3. Water supply (20m^3/d during summer) and use, water level at the water tank.

Another point of importance concerns frost protection of the water tank since it will be placed outside the hut. Using good insulation and covering the water tank several metres high with soil solves this problem. In addition, the water tank will be heated during winter by the excess electricity from photovoltaics so that the water temperature will never fall below 5 °C.

Energy

For the needed electricity and heat energy there are photovoltaics (about 84 m^2) and thermal collectors (about 56 m^2) available, respectively. Optionally, a wind generator could supplement those facilities. Although no manufacturer could be found to date who would guarantee its operation at this exposed site. Solar energy can cover 90% of the energy demand not counting cooking, which in a later phase could be covered by solar energy as well. A combined heat and power unit (CHP) powered by gas or rapeseed oil serves as a backup and to cover the demand during peak hours. Both a hot water storage tank of 6000 litres (4500 litres for heating and 1500 litres for sanitary hot water) and batteries having 184 kWh capacity buffer the occurring differences between heat supply and heat demand.

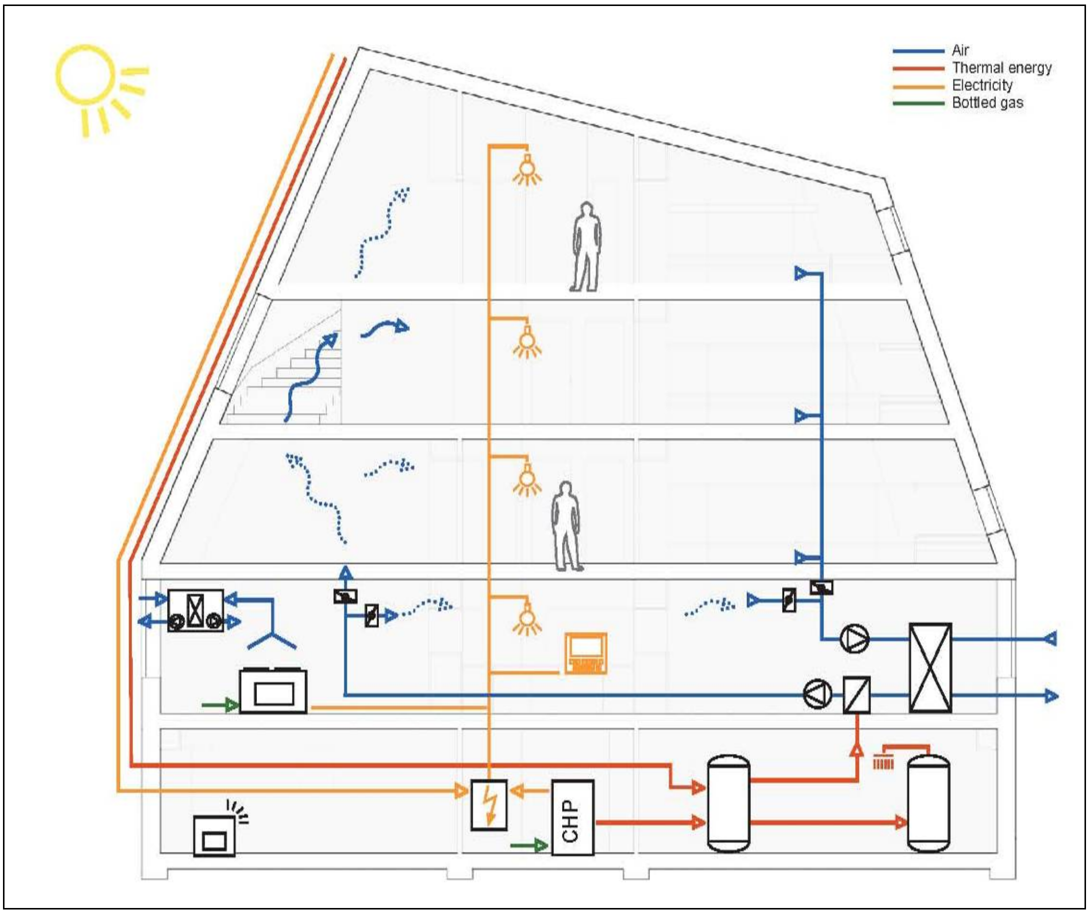


Figure 4. Basic scheme of energy and building services technology.

Building services technology
The cabin will be mechanically ventilated, which is a novelty for mountain shelters. The mechanical ventilation serves two purposes: First it helps to avoid structural damage mainly by mould (the risk is quite high since the building envelope is designed to be extremely airtight) and second, it ensures a basic thermal comfort for the occupants. At the same time the building will be heated by air. The supply air is blown into the open staircase and the exhaust air is drawn off the rooms. After heat recovery, the air is blown outdoors. During the operation of the hut (beginning of March till end of September) the internal heat loads (i.e., persons, equipment, and lighting) together with passive solar gains suffice to ensure the required air temperature of 5 - 20 °C, whereas the target value lies at 15 °C.

The dynamic simulations of the building services technology show that with the chosen concept it is necessary that especially during winter time, in the absence of the warden, the combined heat and power unit, is switched on to ensure the temperature of 5 °C to avoid freezing of piping. The buffers will be charged to 100% while they get uncharged during the few peak weekends when the CHP is used for the production of electricity.

The above described dimensioning of different parts of the facility is based upon elaborate dynamic simulations. Those simulations served to optimise, amongst others, the following parameters:

- Size of batteries and hot water storage tank,
- Optimal division of the available area into photovoltaics and thermal collectors,
- Clarification of whether an improvement of the building envelope or the augmentation of the area of thermal collectors (or the hot water tank) is more cost-effective.

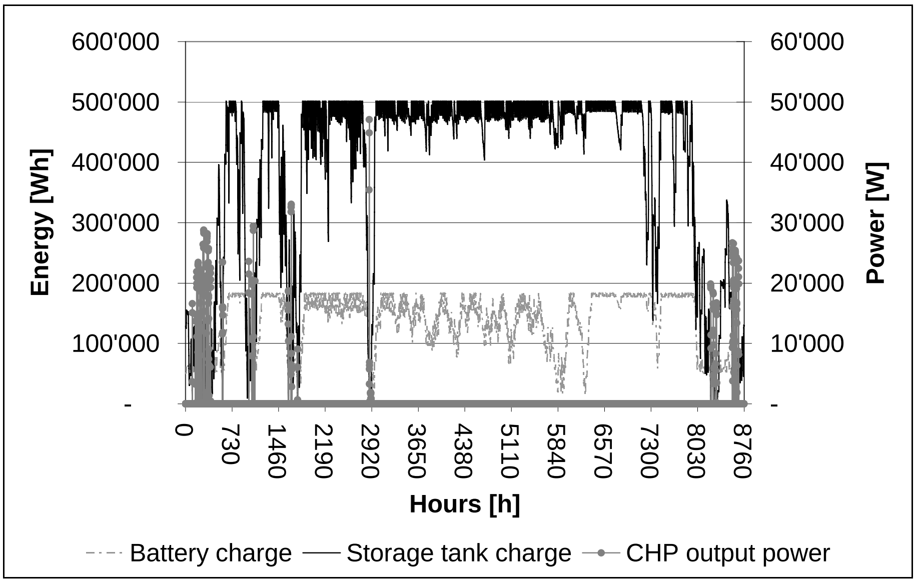


Figure 5. Results of simulations: State of charge of batteries and hot water storage tank, net total power production (heat and electricity) of CHP.

Figure 6 shows an example diagram for this optimising process. For different fractions of a certain total area covered by thermal collectors one can read off the possible degree of autarky on the ordinate. The abscissa hereby displays the total area covered by both photovoltaics and thermal collectors.

The figure shows that to reach a degree of autarky of 90% and using the best ratio of photovoltaics and thermal collectors one would need at least 125 m^2 total covered area. The concept of HTA proposes to cover up 140 m^2, taking into account the local situation.

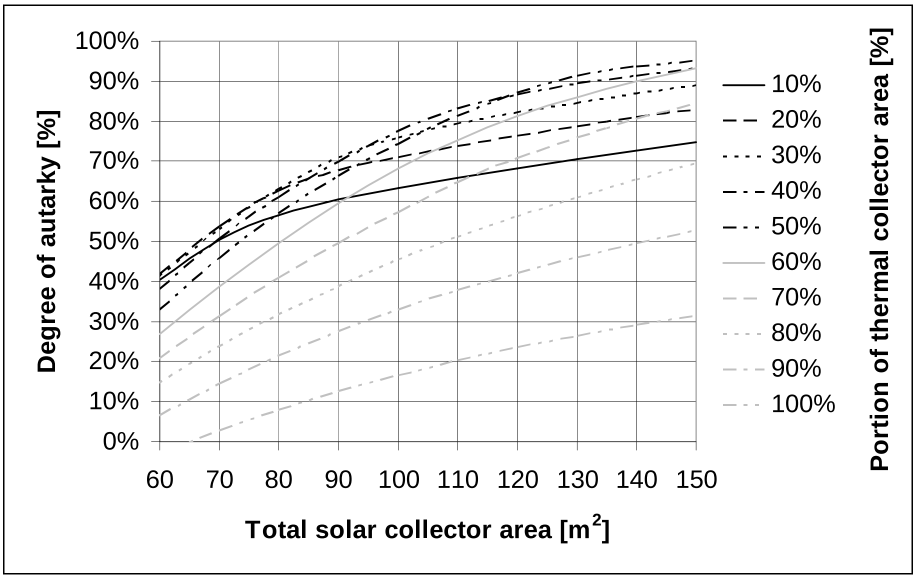
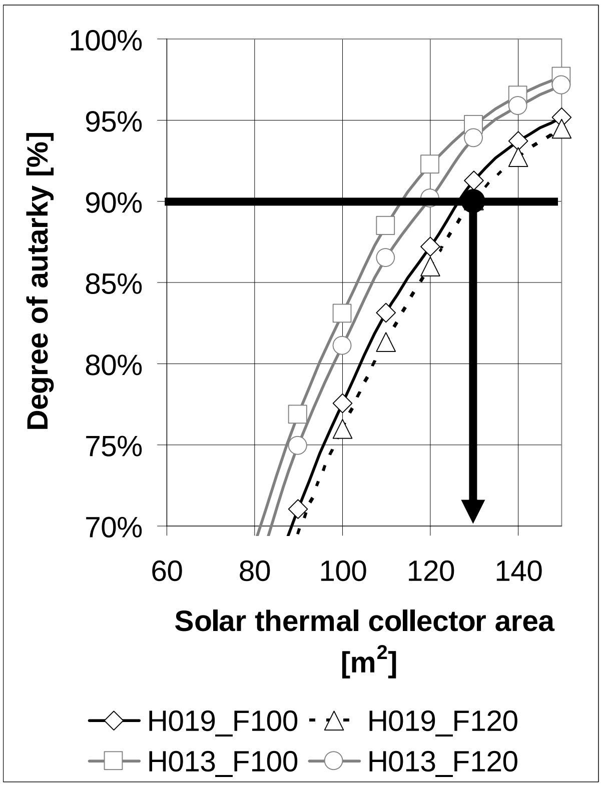


Figure 6. Optimisation of the total area covered with collectors and the resulting degree of autarky. The different curves represent the portion of thermal collectors to the total collector area.

Another optimising process concerns the building envelope (U-value of opaque parts and windows) and the area covered by thermal collectors taking into account the investment cost (see Fig. 7). The figure shows clearly the interrelationship of the area covered by thermal collectors and four different qualities of U-values of the façades. The right figure shows distinctively that even using a comparatively bad building envelope (U-value opaque 0.19 W/m^{2}K, U-value windows 1.2 W/m^{2}K, code 'H019_F120') the investment cost for the building services would not increase much. In contrast, building a good façade (U-value opaque 0.13 W/m^{2}K, U-value windows 1.0 W/m^{2}K, code 'H013_F100') would increase primarily the building envelope cost. Therefore one reaches a better cost-benefit ratio planning a "poor" building envelope and increasing the area covered by thermal collectors.



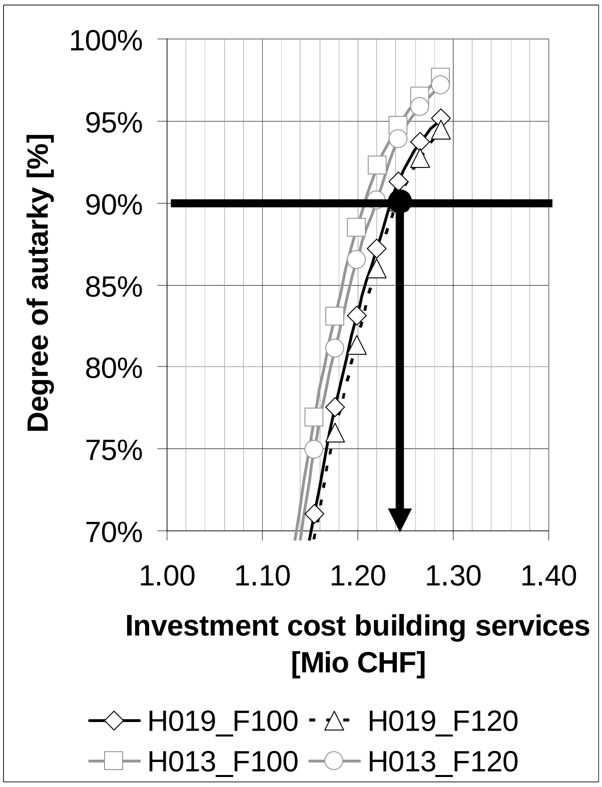


Figure 7. Optimisation among total area covered by collectors, the quality of building envelope ("H019_F100" etc.), investment cost for building services, and degree of autarky.

DISCUSSION

The part "energy and building services technology" for the new Monte Rosa cabin shows that it is certainly possible to reach an extraordinarily high degree of autarky at absolutely affordable investment costs. It is vital that because of the remote location only proven and robust systems are employed. No unproved equipment or sophisticated gadgets will be installed because one cannot take any chances at a hard-to-reach, remote location as an alpine cabin in the middle of the mountains. Despite the reliability of the employed systems (photovoltaics, thermal collectors) it is paramount to provide redundancy and therefore plan a backup system to ensure full security of supply. Furthermore, the paper shows that energy is just the second largest challenge in the conceptual design of such a building. Much more demanding is the supply and recycling of water since, after all, wastewater cannot be disposed of untreated as is quite common nowadays [3, 4].

One important insight will affect the next steps of the project: The presented concept will reach the desired degree of autarky with a clever optimised energy management. This energy management system incorporates the weather forecast and expected occupancy schedules to allow an optimal management of both batteries and hot water storage tank. This is also important to guarantee an optimal operation of the wastewater treatment system. The wastewater tank should be emptied before the peak weekends so that the energy-intensive treatment system can be operated at off-peak times when free electricity is available. Further dynamic simulations will focus on the development of such algorithms for the building automation system and the energy management as well as their validation. Another crucial element is going to be the investigation of sensitivities and the analytical testing of different scenarios to determine the robustness of the planned energy strategy and building technology concept.

ACKNOWLEDGEMENT

We thank Prof. Meinrad K. Eberle and his team at ETH Zurich for the possibility to be working on this interesting and innovative project and the inspiring cooperation. We also thank the Swiss Alpine Club, SAC, for the constructive cooperation. Last but not least we thank all entrepreneurs who supported us so far in the development work providing information and sharing their operating experience.

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Energy Savings in Blocs of Flats Due to Individual Heat Metering

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SUMMARY

The installation of thermostatic valves and cost allocators represents the main modernization measure of the heating installation. In this study we determined theoretically and experimentally their effects upon the energy savings. Their adaptive behavior to the dynamic climatic conditions leads to 15% energy savings. Further on, the people reduce the indoor temperature in order to reduce the price of the energy. This effect represents another 9%, leading to a total of 24% energy savings. The north apartments need 15% more thermal energy than the south oriented apartments. The installation of these devices raise un other problem, the measures to correlate the heat production to the heat need.

INTRODUCTION

The modernization of the heating and hot water installations from blocs of flats in Romania is an operation that has started in 2000. The main operations consists in the installation of heat meters on the branch pipes of the two installations, the installation of individual water meters in bathrooms and kitchens and the installation of thermostatic valves with cost allocators for each radiator. These modernization measures offered the consumers the possibility to deal with the thermal consumptions as goods.

The paper presents a comparison, both theoretically and experimentally, of the heat consumption of a bloc of flats before and after the installation of thermostatic valves. The paper also presents o comparison between the heat consumptions of the north oriented apartments and the south oriented ones. The objective of the paper is to identify the consequences of this modernization measure.

The paper presents the description of the building, the theoretical model, the experimental protocol, the results, discussions of the results and the implications upon the actual district heating.

METHODS

Building description

The analysis was carried out on a bloc of flats (basement, ground floor and 10 more levels) composed of four stairwells with four apartments on each floor of a stairwell, in Bucharest Romania. The front wall of the building is north oriented, meaning that 80 apartments are north oriented and other 80 apartments are south oriented. The central heating installation of the building was gradually modernized:

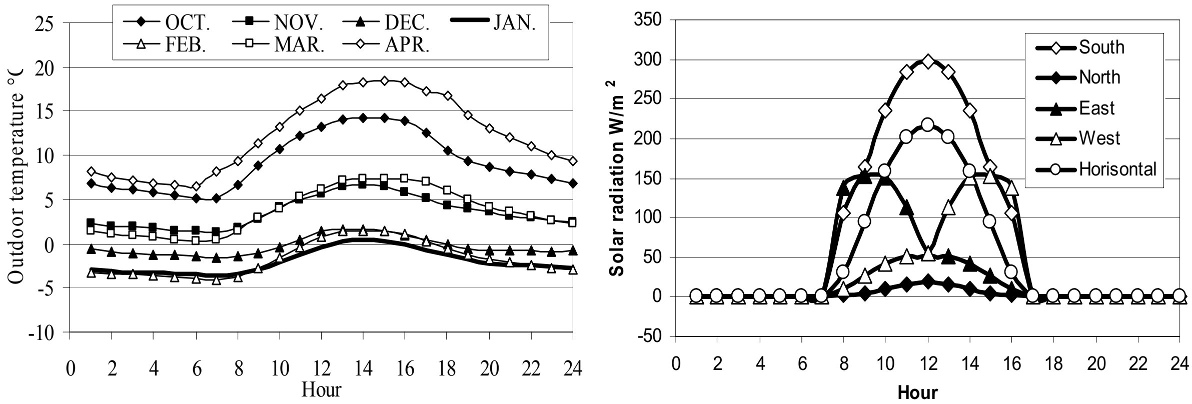
- heat meters were installed on branch pipes of heating and hot water installations, in 1999,

- individual water meters were installed, in 2004,
- thermostatic valves and cost allocators were installed in the 2005 summer.

Both the theoretical and the experimental analysis were carried out during the cold season, considered from October till April.

Theoretical approach

Theoretic evaluation of the heat consumption was carried out in two situations: with and without thermostatic valves installed. In the first case, we considered that the set point of the thermostatic valves was 20°C. The outdoor climatic database for Bucharest (temperature and solar radiation) corresponds to an average intensity of the cold season. The database was composed of 30 similar days for each month, but different from one month to another (Figure 1).



a) b)

Figure 1. Variation of climatic parameters: a) temperature, b) Solar radiation

The evaluation of the heat loss of the building represents the sum between the transmission heat loss through building envelope elements and the heat loss due to outdoor air infiltration.

In the second situation (no thermostatic valves), the unsuited temperature of the hot water in the heating installations generate a raise of the indoor temperature. The dynamic heat transfer calculus was based on the hypothesis of constant temperature inside the walls. The thermal behavior modeling of the radiators was achieved by means of stationary relations. We considered that the global thermal transfer coefficient of the radiator depends on the hot water temperature entering the radiator. Thus the temperature of the hot water at the end of the radiators is the result of the complex thermal transfer phenomenon building – heating installation.

The outdoor temperature used for the calculus cumulates both convective and solar radiation heat transfer phenomena (Equation 1 for heat transfer through walls and Equation 2 for heat transfer through windows).

$$t_{EO} = \frac{\alpha_a}{\alpha_o} \cdot I + t_o \qquad (1)$$

$$t_{EO} = \frac{\alpha_a \cdot \tau}{\alpha_o} \cdot I + t_o \qquad (2)$$

where t_{EO} is the equivalent outdoor temperature, t_o is the outdoor temperature, α_o is the outside convective thermal transfer coefficient, α_a is the absorption coefficient of the outside wall surface, $(\alpha_a \cdot \tau)$ is the absorption-transparency coefficient and I is the solar radiation.

The heat consumption represents the time integration of the heat flux during the cold season. The heat consumption was calculated in both situations of the installation: with and without thermostatic valves.

Experimental approach

The first type of recordings is the heat consumption of the entire building, measured for several years on the branch pipes of the heating and hot water installations.

The second type of recordings is the individual heat consumption due to the installation of the thermostatic valves before winter 2005-2006. These heat consumptions were estimated by means of the monthly recordings of the cost allocators and the indoor temperatures.

RESULTS

Theoretical approach

The theoretical approach is based on the hypothesis that the occupants of the apartments do not modify the position of the thermostatic valves; thus the initial setting of the valves corresponding to a 20^{o}C indoor temperature becomes permanent.

The heat consumption of the apartments with thermostatic valves is:

- similar to that of the apartments without thermostatic valves during evening and night period (20:00-07:00), whatever the orientation of the apartment,
- smaller than that of the apartments without thermostatic valves during morning and midday period (07:00-20:00), whatever the orientation of the apartment (Figure 2).

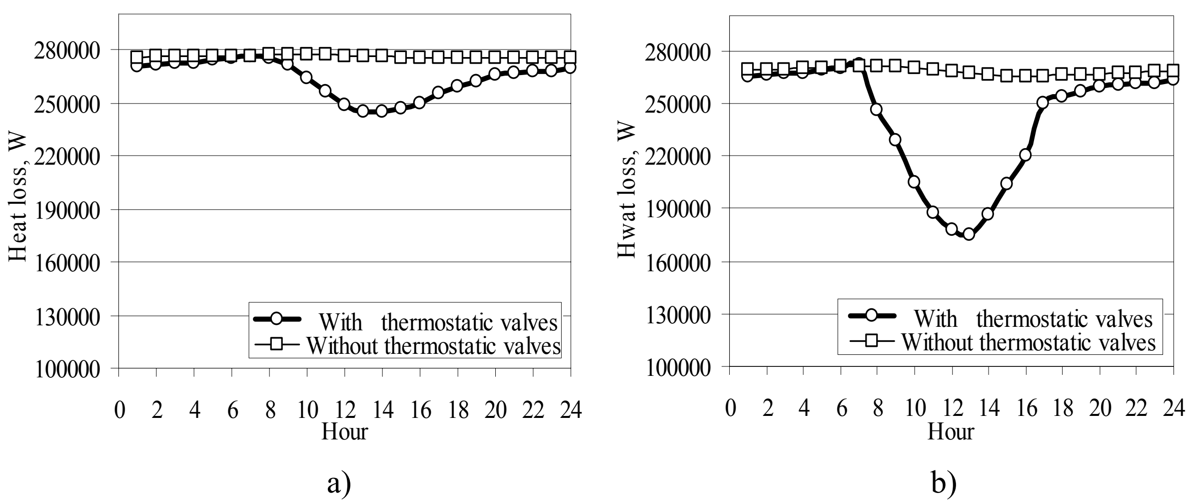


Figure 2. Daily variation of the heat losses of the apartments on a) NORTH b) SOUTH side of the building, for an average day on JANUARY

Comparing the heat loss of the north and south oriented apartments we note that:

- the simple valves do not adapt according to heat gains, and the heat losses is almost similar whatever the orientation,
- the thermostatic valves react to heat gains and lead to important energy savings (Figure 2).

The integration of heat losses during the winter season (October – April) lead to the values of the heat consumptions (Table 1).

Table 1. Specific heat consumption

Orientation	with thermostatic valves (kWh/m^2.an)	without thermostatic valves (kWh/m^2.an)
SOUTH	**105,176**	**127,561**
NORTH	**120,323**	**132,338**

The following conclusions can be drawn from this analysis:

- In the case of simple valves, there is only a very small difference (3.7%) between the heat consumptions of north and south sides of the building. This value is due to different solar radiation between the two sides of the building.
- In the case of thermostatic valves, north oriented apartments present a 14,4% higher heat consumption that the south oriented apartments,
- In the case of south oriented apartments, the thermostatic valves generate a 21.3% energy saving compared to the simple valves,
- In the case of north oriented apartments, the thermostatic valves generate a 10% energy saving compared to the simple valves,

The same specific heat consumption index was calculated for the entire building in the two situations (with and without thermostatic valves). In the case of simple valves the heat consumption is 129.95 kWh/m^2an, while in the case of thermostatic valves it is 112.75 kWh/m^2an.

We conclude this analysis with the remark that this modernization measure (installation of thermostatic valves and cost allocators) lead to 15% energy savings.

Experimental validation

Before the installation of the thermostatic valves and cost allocators, the evaluation of the heat consumptions of the north and south oriented apartments is not possible, thus this paragraph presents only two aspects:

- a comparison of the heat consumptions of the north and south oriented apartments for the winter 2005-2006, and
- a comparison of the recorded heat consumptions of the same building between the winters 2004-2005 (the thermostatic valves were not yet installed) and 2005-2006 (the thermostatic valves were installed), in order to experimentally establish the effect of the thermostatic valves.

The recorded heat consumption of the north oriented apartments are, obviously, higher than for the south oriented apartments (Table 2). The recorded thermal energy savings of the north oriented apartments compared to the south apartments considering the entire cold season (14,74%) is almost equal to the theoretic result (14,4%). This represents a good validation of the theoretic model.

Table 2. Heat consumption comparison between north and south oriented apartments

Month	Heat consumption, %		Reduction, %
	NORTH	SOUTH	
nov. 05	53,74	46,26	14,97
dec. 05	53,25	46,75	12,99
jan. 06	52,71	47,29	10,87
feb. 06	53,23	46,77	12,9
mar. 06	53,15	46,85	12,59
apr. 06	56,04	43,96	24,13
cold season	**53,69**	**46,31**	**14,74**

In the case of the second analysis, we mention that the two cold seasons were characterized by different climatic conditions, leading to an improper comparison of the heat consumptions (Table 3).

Table 3. Temperatures and specific heat consumptions for winters 2004-2005 and 2005-2006

Month	Winter 2004-2005 without thermostatic valves			Winter 2005-2006 with thermostatic valves		
	t_o ºC	t_i-t_o ºC	ϕ kWh/m^2	t_o ºC	t_i-t_o ºC	ϕ kWh/m^2
November	7.4	12.6	22.36	5.8	14.2	19.84
December	3.06	16.94	27.58	2.7	17.3	24.95
January	2.18	17.82	29.91	-2.2	22.2	25.08
February	-0.4	20.37	31.89	0.6	19.4	22.21
March	5.3	14.7	23.21	6.4	13.6	18.41

In order to surpass this inconvenient, we compare the heat consumptions divided by the indoor-outdoor temperature difference (Figure 3).

The heat consumptions for the 2005-2006 winter are smaller then those of 2004-2005 winter with 12% (December) up to 33% (January). We can conclude that the installation of the thermostatic valves generate an average value of 24% thermal energy saving during the cold season. The difference between the experimental result (energy reduction of 24%) and the theoretic result (energy reduction of 15%) is probably due to human behavior; people tend to modify the indoor temperature in order to accord the thermal comfort to the desired cost.

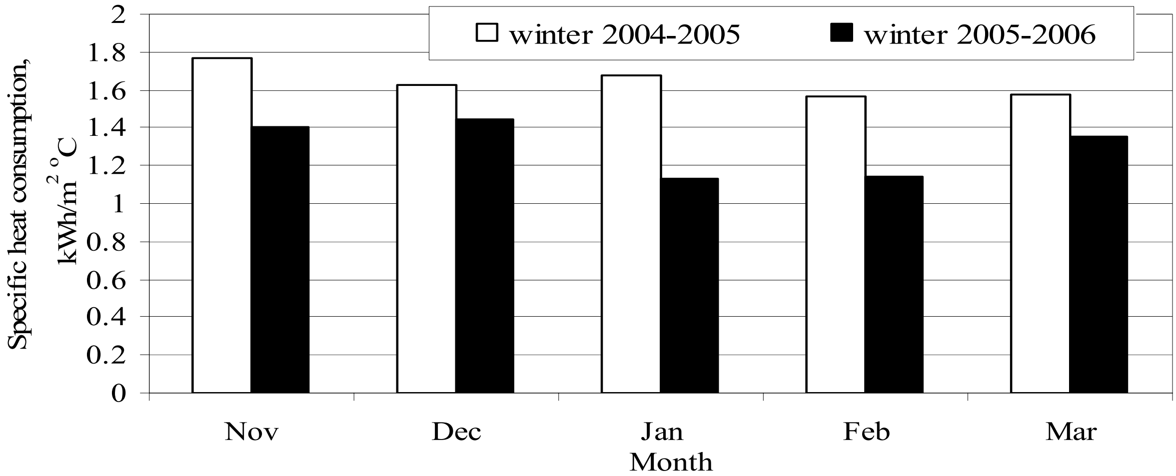


Figure 3. Specific heat consumption comparison between winters 2004-2005 and 2005-2006

DISCUSSION

The theoretical and experimental research followed two objectives. The first is to determine the effect of the thermostatic valves installations upon the thermal energy consumption. The theoretical simulation shows that there is a 15% energy savings while the experimental work showed a 24% energy savings. The 9% difference is probably due to the initial hypothesis for the theoretical model, that indoor temperature is constant 20^{o}C, while in reality people tend to reduce the indoor temperature in order to reduce further more the expenses. The second objective is the comparison of the heat consumption of the apartments oriented north and those oriented south. The advantage of the thermostatic valves is that they can adapt the water flow and by consequently the heat consumption according to the heat gains. We obtained a 14.5% energy savings of the south oriented apartments compared to the north oriented apartments.

Both analyses show the advantages of this modernization of the heating installation, the installation of thermostatic valves and cost allocators. However, if they represent a very important and economic measure for the consumers, the thermostatic valves have a major impact upon the heat producer.

In today’s conditions of rising price of thermal energy, the lack of individual control of heat consumption leads to the impossibility of payment from the consumer. Many district heating company were obliged to stop the thermal energy delivery. Thus the installation of thermostatic valves becomes not only a modernization measure but also a necessity. Another influence of the thermostatic valves consists in the running strategy of the heat production. If before, there was a constant production of heat, and the entire district heating was functioning this way, now the thermostatic valves present a dynamic thermal energy need. The district heating company from Bucharest, RADET, is now confronted with this aspect of correlation between the heat production with the dynamic heat need. The national research project ENERGSYS was developed in order to find the suitable measures to adapt the heat production to the variable heat need.

ACKNOWLEDGEMENT

We acknowledge the contribution of the district heating company of Bucharest, RADET SA, and TECHEM to the present study.

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Sustainable and Energy Efficient Buildings Class Curriculum

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SUMMARY

A workshop was held in the fall of 2006 for the purpose of developing courses related to "Energy Efficient Integrated Sustainable Design for the Built Environment" for university, college, and vocational schools. The objectives of this workshop were to determine the key elements for students to know related to integrated energy efficient sustainable design; to identify specific learning objectives for each of those elements; and to identify resources available to use in these courses. The workshop was attended by a selected interdisciplinary group of well recognized engineering, architecture and building science professors, representatives of related professional organizations and national laboratories, and individuals working with the US DOE Building America teams. This paper presents a listing of the key topics identified and the learning objectives related to each of those topics.

INTRODUCTION

Sustainability and energy efficiency are topics which are covered in wide variety of different college courses in America. They are now being widely taught though community colleges, professional organizations' continuing education programs, and general university and adult education programs. The basics however traditionally have been in the curricula of architecture, engineering and building science programs.

The accreditation board for professional degree programs in architecture is the National Architectural Accrediting Board which "… is committed to the provision of effective professional architectural education through the establishment and application of accrediting procedures …" and "… strives to foster an educational foundation that prepares students who are both broadly and professionally educated for the profession of architecture." [1] For the purpose of accreditation, NAAB indicates that graduating students must demonstrate an understanding or ability in 34 different areas. Most relevant to the purposes of this paper is "#15 Understanding of the principles of sustainability in making architecture and urban design decisions that conserve natural and built resources, including culturally important buildings and sites, and in the creation of healthful buildings and communities." Similar references are also made to the areas of: site conditions, environmental systems, building envelope systems, building systems integration, building materials and assemblies, construction cost control and comprehensive design.

Specific references to sustainability are not as explicitly described in the accrediting body in the field of engineering, ABET. This group indicates in the program outcomes and assessment that a school undergoing accreditation must demonstrate that their students attain an ability, understanding, recognition, or knowledge in eleven different areas. The one related to the area of sustainability indicates that students must demonstrate "(c) an ability to design a system, component, or process to meet desired needs within realistic constraints such as economic, environmental, social, political, ethical, health and safety, manufacturability, and sustainability."[2]

In a meeting of land grant institution teachers and researchers discussing energy efficient residential construction it was recognized that there many different approaches and materials used to discuss energy efficient sustainable design. During the breakout session it was also recognized that there was a pressing need for a systematic discussion to share ideas and basic concepts between the various educational disciplines, professional organizations and research teams. This discussion needed to determine the knowledge base of what was needed by students as to the key elements of integrated sustainable design, to develop curriculum related to these key elements, and to identify the resources used in the delivery of these courses.

METHODOLOGY

In order to address the needs identified, a workshop was held in the fall of 2006 for the purpose of developing courses related to "Energy Efficient Integrated Sustainable Design for the Built Environment" for university, college, and vocational schools. The objectives were to determine the key elements for students to know related to integrated energy efficient sustainable design; to identify specific learning objectives for each of those elements; and to identify resources available to use in these courses. Although the topic was considered to be applicable to all building sectors, it was determined that the first focus should be on the low-rise residential market. The workshop was attended by an interdisciplinary group of 30 selected well recognized engineering, architecture and building science professors, representatives of related professional organizations and national laboratories, and individuals working with the US DOE Building America teams.

RESULTS

The interdisciplinary group met for a 3 ½ day intensive workshop and identified the key elements of knowledge which a student should know in this area. While the extent of the courses might vary between the professional degree programs (such as architecture or engineering) and the two-year associates degree programs; the key elements remained the same. It was thought that these key elements could be used as the basis for the first course on energy efficient sustainable design for the built environment and were therefore laid out as separate modules. Depending upon the level of instruction (i.e. professional design instruction such as engineering versus a technical school) the modules could be modified to suit the needs of the audience. Each one could be the topic of a one day adult education workshop, one-week instructional modules in an introductory course, or serve as the basic outline for an entire semester in an advanced course.

The underlying learning objectives for each of the modules were then identified.

Seventeen modules were identified and learning objectives for each were developed. The modules and the topics contained and learning objectives for each are given below.

Module 1. Setting the Stage for a Sustainable Energy-Efficient Built Environment – The Context of Energy Issues and Building Solutions

- Define terms of building science, ecological systems, economics of consumption
- Relate building science perspective, ecology, social science
- Explain historical energy and environmental issues related to buildings

- Compare site and source energy
- Examine the health, safety and comfort issues in buildings
- Examine the general context for building solutions (The goal of net-zero energy green home with durability.)
- Explain a basic overview of the sufficiency of alternative energy (total solar flux and wind availability)
- Examine cash flow to homeowners
- Demonstrate ability to find, evaluate and synthesize knowledge regarding building performance and sustainability
- Define Business case – career opportunities
- Explain appropriate technology and systems (and how to research them with every lesson)
- Define interconnections / inter-relationships among building systems

Module 2. Introduction to Sustainable Design & Building Performance
Learning Objectives:
- Recognize that a building works as a system
- Learn about the roles of air, heat, water, and vapor flows
- Recognize the importance of climate-specific design details
- Understand the issues on health, IEQ and productivity.
- Building performance and relations to overall sustainability
- Differentiate between the available fuel choices and characteristics
- Recognize roles, responsibilities and respect of the participants in design and construction

Module 3. Site Related Flows: Air, Heat, Water, Vapor
Learning Objectives:
- Comprehend specific issues related to pressure- and temperature-induced flows
- Grasp the significance of water flows and their roles in building details related to the drainage plane and other building elements
- Recognize the need to manage relative humidity and prevention of condensation
- Understand the air change rate and its relationship to above concepts

Module 4. Building Materials and Their Properties
Learning Objectives:
- Understand building material porosity and the impact it has on properties, such as wetting and drying, capillarity action
- Understand and be able to use:
  - Vapor perm ratings
  - Air perm ratings
  - R-values/U-values – all materials, including glazing
- Understand the difference between individual material properties and assembly performance
- Understand the impact of mass and phase change materials
- Recognize the life span of materials and their embodied energy
- Understand the need for waste reduction, regionally appropriate and ecological materials

Module 5. Climate and Designing with Nature

Learning Objectives:

- Understand the importance of climate-appropriate design
- Recognize relationships among temperature, precipitation, and construction techniques
- Learn about hygro-thermal regions and how design and construction details vary across them
- Understand building details related to seismic conditions, hurricane-resistance, wind, corrosion and other climate-specific factors that affect structural durability
- Understand solar geometry, daylighting, natural ventilation
  - Understand building geometry related to the local environment
  - Understand daylighting performance in building design
  - Understand shading devices in building design
  - Understand building designs that utilize natural ventilation

Module 6. The Process of Building Design, Systems Engineering, Commissioning

Learning Objectives:

- Understand air leakage control through design details and blower door testing
- Understand HVAC design and integration issues as well as testing protocols
- Gain competence in performance of building commissioning and management of commissioning records
- Understand the role of design details, specifications, and trade contractor scopes of work with respect to quality and high performance
- Understand diagnostics techniques and reports

Module 7. Site: Drainage, Pest Control, Landscaping

Learning Objectives:

- Understand the role of site grading and water run-off
- Understand management of termites, rodents, and other pests
- Understand proper placement of vegetation, mulch, and other decorative land cover
- Comprehend soil properties and soil conditioning
- Know native vegetation and the role of irrigation and reduction of water usage

Module 8. Foundation: Moisture Control and Energy Performance

Learning Objectives:

- Understand foundation construction techniques essential for the prevention of moisture and soil gas entry
- Understand the contribution of the foundation system to overall building energy performance
- Understand climate-specific use of alternative foundation insulation systems

Module 9. Building Envelope: Moisture Control and Energy Performance

Learning Objectives:

- Learn roof and wall assembly materials and techniques essential to water management (including flashing)
- Learn roof and wall assembly materials and techniques essential to air infiltration
- Learn roof and wall assembly materials and techniques essential for the prevention of vapor intrusion and drying of interstitial spaces
- Learn climate- and design-specific use of alternative glazing systems

- Understand selection criteria and application details of cavity and attic insulation materials
- Understand alternative approaches to vapor retarder and house wrap systems
- Understand the energy and lighting implications of alternative window treatments
- Gain an appreciation and understanding of the dual roles of weather barrier and energy performance

Module 10. Mechanicals/Electrical/Plumbing: Systems Engineering, Energy Performance, Occupant Health, Safety, Comfort, and Envelope/Mechanicals Management, Part I
Learning Objectives:
- Understand equipment and duct issues
- Understand integration of mechanical system design and architectural design
- Understand best practices of selection and installation of mechanical equipment
- Understand efficiency standards and appliance ratings
- Understand the importance of plug loads, appliances and lighting systems
- Understand the principles of the systems for
  - Space temperature conditioning (heating and cooling systems)
  - Hot water distribution
  - Duct layout
  - Controls and monitoring
  - Pollutant source control
  - Evaporative cooling

Module 11. Mechanicals/Electrical/Plumbing: Systems Engineering, Energy Performance, Occupant Health and Safety, Comfort, and Envelope/Mechanicals Management, Part II
Learning Objectives:
- Understand applicable ASHRAE ventilation Standards (ASHRAE 62.2)
- Understand backdrafting issues, combustion gas management, and sealed combustion systems
- Understand proper placement and penetration sealing of electrical wires, plumbing pipes, and HVAC ducts
- Understand the role of indoor relative humidity in building performance and the conditions-based need for dehumidification/humidification
- Understand the basics of alternative heating systems
  - Dual fuel heat pumps
  - Wood burners and fireplaces
- Understand on-site generation systems
  - Photovoltaic and wind
  - Solar thermal
  - Combined heat and power (CHP)
  - Fuel cells

Module 12. Energy codes/standards, Building Simulation and Tools, Measurement and Prediction
Learning Objectives:
- Understand applicable energy codes, standards and best practice recommendations
- Demonstrate the ability to use tools to analyze buildings and make design decisions (energy, environment, etc.)
- Know how to measure performance and make adjustments.

Module 13. Field Issues: Construction Management, Building Codes, and Other Regulatory Matters

Learning Objectives:

- Understand practical matters that affect implementation of design details, specifications or purchasing requirements, and scopes of work including construction labor issues and homebuyer concerns
- Understand code enforcement and zoning ordinance issues that may obstruct the construction of high performance housing and effective counter strategies
- Understand the impact of codes/standards on building performance and sustainable design
- Understand local public policy; Impact of policy, regulation and enforcement
- Be aware of the processes of policy development and change
- Understand contracts and contract law

Module 14. The Business Case, Communications and the Community Scale Perspective

Learning Objectives:

- Understand valuation methodologies for building performance and sustainable design
- Gain communications skills / tools to achieve effective team functioning
- Value the ability to use peers as a resource not a competitor
- Understand the relationship between single building & site land use, infrastructure and ecological impacts
- Learn about utility systems and interconnections
- Comprehend rate structure, base and peak usage
- Understand time-of-day usage and electrical marginal dispatch
- District heating and cooling
- Transportation – people, waste, water and energy

Module 15. Putting it all Together: Experiential Learning in the Field / Office

Learning Objectives:

- Through a partnership with a high performance builder, shadow a construction manager for an assigned time during a one-week period
- Be able to apply the principles to
  - Case studies
  - Performance verification studies
- Analyze examples given in videos

Module 16. Homeowner Education (Communicating with the Consumer)

Be able to communicate the following with the consumer

- Tax incentives
- Financing & insurance
- Occupant lifestyle impact
- Energy improvement mortgages
- Commissioning, punch lists, owner manual
- Operation and maintenance
- TED
- Selling energy efficiency
- Home energy audits
- Cleanliness of the job site

Module 17. Conclusions, Implications, Directions of Future Research
Learning Objectives:
- Highlight and review materials covered and studio and field experiences
- Understand current research issues regarding high performance buildings
- Understand need for life long learning
- Reflection

The resources to be used in the modules were also identified in the workshop. This material will be expanded upon in another workshop to be held in the summer of 2007.

SUMMARY AND CONCLUSIONS

A workshop with faculty and building industry leader participants was held to develop university and community college courses related to the design of energy efficient and sustainable buildings. The objective was to determine the key topics and the elements of these topics which need to be taught and to develop the learning objectives for each of the segments. Courses were developed to be taught at the architectural and engineering schools, at community colleges and vocational schools, and for general education in colleges, universities and general adult extension education. The topics and learning objectives for each of the topics are presented in the paper.

ACKNOWLEDGEMENT

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Comparing Economics of Various Methods of Improving Energy Efficiency of Commercial Buildings

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SUMMARY

Various means of improving the energy efficiency of commercial buildings while preserving and/or improving their internal environmental quality have been extensively studied. In the recent past, the Gas Technology Institute (GTI), produced several of such studies oriented exclusively to identifying the benefits of applying various energy efficiency technologies to different types of commercial installations in the U.S.A. and also in Europe. Independently, these involved technologies for desiccant dehumidification and/or systems of combined heating, cooling, and power generation, (CHP). The applications and their performance were simulated by internally developed programs such as the Building Energy Analyzer, using the DOE2.1E computational engine. Studies identified potentials of targeted application of combined technologies in comparison to using the traditional building heating, ventilation, and air conditioning equipment.[1]

The latest study, which is the subject of this paper, expands on the previous ones. Authors compare the economics of investing either in installing the energy saving mechanical equipment, such as CHPs and solar thermal cooling, or applying the modern means of improving the building envelope and reducing internal loads. As in previous studies the results were found to be “case specific”, yet for the assumed installations the preferred ways of improving a building performance were identified and are reported in this paper. This has been accomplished for the specific case of selected buildings on the campus of the University of Hawaii, yet the implications can be valid for other locations with similar climatic conditions.[2]

INTRODUCTION

In a renewed effort of making the American economy less dependent on foreign oil several approaches to improving the efficiency of power conversion systems are getting higher attention again. One of those is the concept of co-generation, a concept that over a half century of its modern evolution went through a series of names and modifications. Currently, most often labeled as combined heat and power systems, they usually incorporate various kinds of natural gas-fired prime movers and electricity generators in a configuration with equipment providing air conditioning and heating services. A solid wheel desiccant dehumidification of the indoor air, powered by heat recovered from a CHP system found to be attractive in many applications.

One of the original objectives of the Gas Technology Institute previous studies was to determine the economical value of CHP systems and their contribution to energy savings in commercial building applications. In a broader spectrum, that study was aimed at finding the benefits of recovering heat for space heating/domestic hot water, desiccant dehumidification, and absorption cooling for five specific building types, considering climatic conditions and local utility rate structures of several U.S. cities.[1]

Subsequently, these studies were expanded to cover other means of improving the energy efficiency of commercial buildings while improving their economics, such as those shown later in this report. While these additional studies were carried out for one location only, (i.e. the University of Hawaii at Manoa), the results can be considered as providing an important guidance in the process of implementing energy conservation measures for all locations with similar climatic conditions, and as such are presented here.

BACKGROUND AND OBJECTIVES

Energy is a key factor in the success of Hawaii's economy. Efficient use of Hawaii's indigenous and sustainable energy resources can reduce the state's high dependence on imported oil, increase local economic development, and reduce the potential negative economic impacts of oil price fluctuations.

As a major research university, the University of Hawaii presented an ideal setting to research energy savings by way of viable technologies and strategies that can be used to reduce energy consumption and cost while increasing energy self-sufficiency.

The Hawaiian Electric Company, (HECO) owns and operates oil-fired power plants that produce electricity for the island of Oahu. Crude oil used in the plants is imported primarily from foreign sources and then locally refined. Naphtha, a byproduct of refining process is converted by the Gas Company, (TGC) to clean-burning synthetic natural gas (SNG). Utilizing the SNG displaces the need to import additional barrels of oil. Consequently, one of the targets of GTI's study was to evaluate ways to maximize the use of this indigenous energy sources.

The following collection of energy-efficient and sustainable technologies and strategies were evaluated in terms of how much energy could be saved annually and the economic feasibility of implementation:

- SNG-fired reciprocating engine for combined heat and power (CHP) - using waste heat for domestic hot water, and absorption cooling.
- SNG-fired reciprocating engine-driven vapor compression chiller with engine heat recovery for domestic hot water production. No power production.
- Solar cooling based on the application of medium-grade heat (approximately 180°F) thermal collectors driving absorption chillers.
- Energy-efficiency strategies recommended in the local Hawaii Model Energy Code (HMEC).

In this paper we concentrate on the issues of recommended Energy-efficiency strategies as the represented the most economical means of improvements.

BUILDING SELECTION

Based on the results of evaluating the existing campus buildings it was decided to develop prototypical models that closely resemble typical existing structures as well as new buildings projected to be built on Manoa campus as well as West Hawaii and Hilo. The prototype models were then used to determine appropriate methods for targeting and applying energy-efficient strategies and technologies.

Below is a summary of the characteristics for the three models:
1. Dormitory - Twelve-story, 65,300 square-foot facility with three independently characterized zones (dormitory wings, common area, and kitchen and laundry). The zones are cooled by a central plant consisting of an electric water-cooled centrifugal inlet-vane controlled chiller. The dormitory wings make up 60% of the occupied space and the common area 35%. The kitchen and laundry area makes up the remaining 5% of the space. Building glazing is 25%.
2. Lab/Class/Office - Four-story, 93,100 square-foot facility with three independently characterized zones (classroom/office, lab/shop, and mechanical). The zones are cooled by a central plant consisting of two electric water-cooled centrifugal inlet-vane controlled chillers. The classrooms and offices make up 60% of the occupied space and the labs and shops 35%. The mechanical area makes up the remaining 5% of the space. Building glazing is 25%.
3. Library – Four-story, 135,000 square-foot single-zone facility. The facility is cooled by a central plant consisting of two electric water-cooled centrifugal inlet-vane controlled chillers. The HVAC system is configured to maintain an appropriate humidity level for library material. Building glazing is 50%.

BUILDING ENERGY MODELING

Building Energy Analyzer (BEA) computer energy modeling tool was used to generate energy and economic models for each of the building case-studies. BEA consists of hour-by-hour computer simulation models for various building types, heat and power generation equipment, and HVAC equipment. Within the BEA models, equipment (e.g. lighting, HVAC, etc.) and building parameters (e.g. wall material, window designs, roofing, etc.), energy rates, and geographical weather data can be defined for specific applications.

BEA forecasts and reports annual hour-by-hour heat and power loads along with hour-by-hour fuel requirements. GTI enhanced the software for this study with solar thermal analyses module.

BEA uses weather data from the typical meteorological year (TMY2) data sets derived from the 1961-1990 National Solar Radiation Data Base (NSRDB). The building model data streams are typical for weather during the TMY2 time span.

Alternative Energy-Efficient Technologies

As an alternative to CHP, a collection of energy-efficient technologies and strategies, based on the Hawaii Model Energy Code (HMEC), was applied to the prototypical building models to determine energy reductions and associated cost benefits. Technologies and strategies that were considered in the analysis are as follows:

<u>HVAC</u>

- Application of an economizer cycle (although typically not used in Hawaii) that takes advantage of free cooling using outside air. The system selects outside air to condition building if its enthalpy is lower than that of the building comfort set point.

- Reduced cold-deck temperature from 55°F to 50°F (the lower limit on supply air temperature) for greater humidity reduction and lower air flows and fan power. This does not apply to the Library as the building was already designed for 50°F.
- Minimal HVAC system oversizing – Oversizing was reduced from 20% (often a rule-of-thumb) to 5%.
- Addition of a desiccant wheel dehumidifier with enthalpy relief air heat exchanger. The base system is a standard gas-fired reheat system. The alternative implements an enthalpy wheel system that provides both sensible and latent heat exchange between the relief air and outdoor air. A desiccant wheel with gas-fired regeneration is also used to remove excess latent heat from outdoor air entering building.
- Addition of variable frequency drives (VFDs) on fans and pumps to control flows at partial loads – The air handling and cooling tower fans and the chilled water pump are configured with VFDs.
- Substituting standard electric centrifugal or screw chillers with an engine-driven centrifugal or screw chiller and engine heat recovery to hot water supply of the traditional heating system.

<u>Lighting</u>

- Increase lighting efficiency from ASHRAE standards to HMEC standards (roughly 30% to 50% reduction in watts per square foot).

<u>Shading</u>

- Reduced glazing percentage by 25% to account for overhang wall and window shading and radiant barriers that may be implemented on the buildings.

DISCUSSION OF RESULTS

Figure 1 and 2 show energy and cost savings respectively for the different energy-efficient technologies and strategies that were modeled for the dormitory building. Each of the technologies and strategies is represented by a multi-color bar that shows the contributions from gas and electricity. The individual bars are laid over one bar that is the width of the chart and represents the baseline model without any of the energy-efficient technologies applied. The "T" symbols indicate the change in gas levels relative to the baseline model.

In addition, all of the energy-efficiency strategies were collectively applied and the resulting scenario was titled "Collective EE" as shown in the figures.

Tables 1 through 3 show annual utility cost savings for the dormitory and the other two buildings; lab/class/office building and library, respectively.

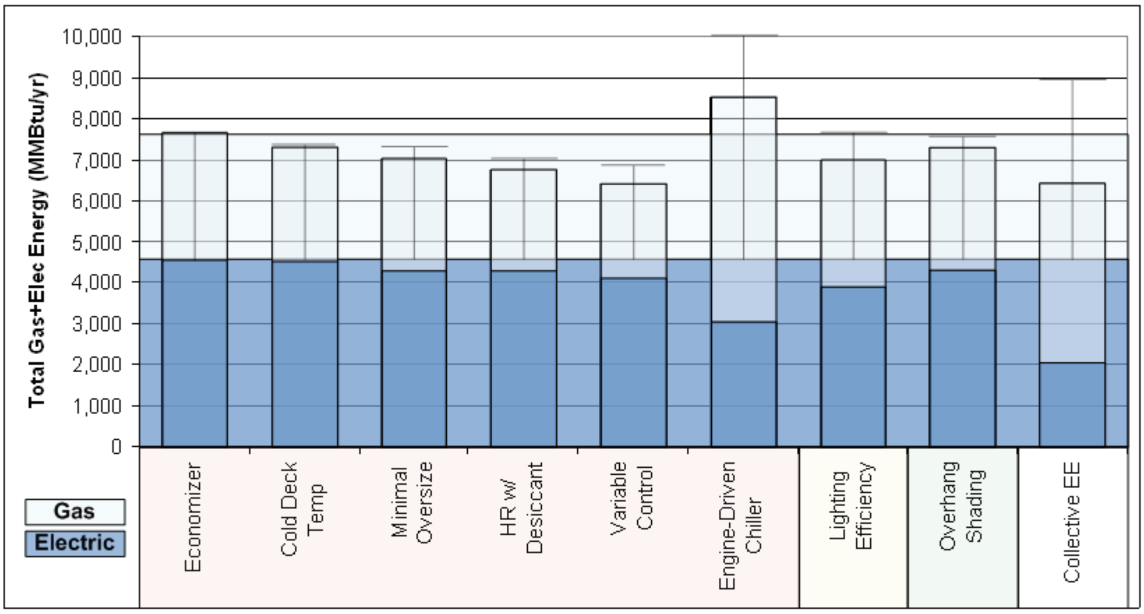

Figure 1 – Annual Energy Savings from Energy Efficient Technologies at the University of Hawaii (Dormitory).

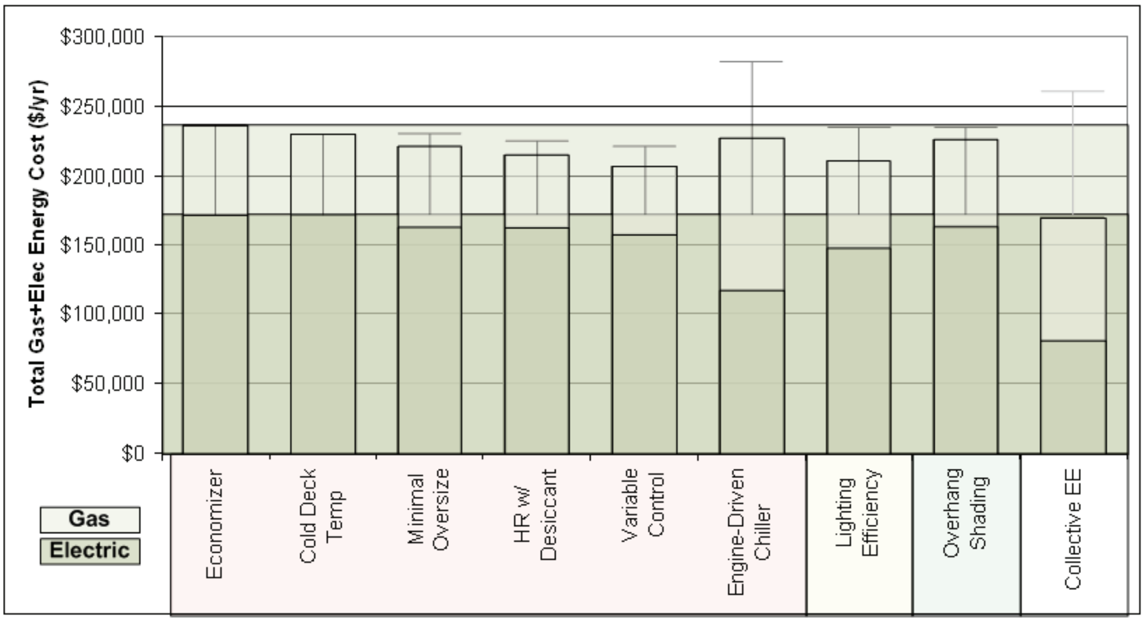

Figure 2 - Annual Cost Savings from Energy Efficient Technologies at the University of Hawaii (Dormitory).

Table 1 – Energy-Efficient Technology Savings (Dormitory).

Technology/Strategy	% Cost Savings	Annual Savings on $236,000/yr*
Economizer	0%	$0
Cold Deck Temp	3%	$6,000
Minimal Oversize	6%	$15,000
HR w/ Desiccant	9%	$21,000
Variable Control	12%	$29,000
Engine-Driven Chiller	4%	$9,000
Lighting Efficiency	10%	$24,000
Overhang Shading	4%	$10,000
Collective EE	28%	$66,000

Table 2 - Energy-Efficient Technology Savings (Lab/Class/Office).

Technology/Strategy	% Cost Savings	Annual Savings on $450,000/yr*
Economizer	0%	$0
Cold Deck Temp	4%	$18,000
Minimal Oversize	6%	$27,000
HR w/ Desiccant	16%	$74,000
Variable Control	18%	$82,000
Engine-Driven Chiller	8%	$36,000
Lighting Efficiency	5%	$22,000
Overhang Shading	2%	$9,000
Collective EE	31%	$138,000

Table 3 - Energy-Efficient Technology Savings (Library).

Technology/Strategy	% Cost Savings	Annual Savings on $460,000/yr*
Economizer	0%	-$1,000
Minimal Oversize	10%	$46,000
HR w/ Desiccant	1%	$2,000
Variable Control	23%	$107,000
Engine-Driven Chiller	3%	$13,000
Lighting Efficiency	12%	$52,000
Overhang Shading	7%	$32,000
Collective EE	29%	$133,000

To put these results into the perspective of the other two evaluated energy saving measures, it's suffice to say that none of the considered systems (i.e. particularly those of CHPs and solar thermal cooling technologies considered as retrofits to existing buildings) could match the savings of the most of the above discussed energy efficiency measures. Consequently, also their return on investment would not warrant their deployment.

Summary of the important results for the alternative energy-saving strategies is as follows:

- An economizer cycle has almost no effect on energy consumption, thus no effect on energy cost for all the three modeled buildings.

*These are the assumed operating costs for prototype buildings with the base equipment as originally designed.

- In some cases, HVAC equipment may be oversized to account for extreme weather conditions that may seldom occur, or just due to the rule-of-thumb engineering. The model indicates that over-sizing the equipment by 20% as opposed to only 5% could cost about $15,000/year. This is caused by inefficiencies of partial loading of the equipment and unnecessary pumping and fan power consumption during non-extreme weather conditions. However, variable speed fans and pumps can more than compensate for the additional cost of oversizing by reducing the excessive flows. In fact, the model indicates that equipping the HVAC system with VFDs could save $29,000/year even if the HVAC system is 20% oversized.
- An engine-driven chiller consumes more energy than an electric centrifugal chiller as shown in Figure 1. However, an engine-driven chiller with heat recovery to domestic hot water can reduce the annual cost as shown in Figure 2.
- Collectively, energy-efficient technologies and strategies could reduce annual energy cost for a prototypical dormitory by almost 30% while increasing the consumption of clean-burning gas by over 40% and reducing the consumption of electricity by 55%. Similar values are shown in Tables 2 and 3 for the two other buildings.
- As with the dormitory, the best cost savings for the Lab/Class/Office ($82,000/yr) comes from installing VFDs on the fans and pumps. The savings from VFDs are still based on a 20% oversized system. However, a 5% oversized system with VFDs also reflects significant savings ($68,000/yr).

CONCLUSIONS

The following are the most important conclusions of this study:

- Collective implementation of energy-efficient technologies, such as engine-driven chillers, fluorescent lighting and HVAC strategies could reduce annual energy cost for prototypical UH facilities by as much as 30% while increasing the consumption of clean-burning gas by over 50% and reducing the consumption of electricity by up to 50% at some UH facilities. As such they could be very effective.
- At least 70% to 90% of recoverable heat from power generation must be recovered in a manner that directly displaces heat from the heating source, for CHP systems serving individual UH prototypical facilities to fall within a 5-year payback range. CHP host facilities must present the proper criteria in terms of persistent heat loads, operating hours, and sufficient size to optimize the return offered by CHP. At the University of Hawaii Manoa campus, such facilities are a challenge to identify.
- Solar thermal cooling is not as yet feasible, given the current electric rates, cost of solar panels, and cost of absorption chillers. A significant subsidizing would be needed for implementation.

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Collaborative Integral Design of Active Roofs

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SUMMARY

In the world of design and engineering, gaps of knowledge between these disciplines are recognized. The learning capacity of the building industry – as well as in other industries – is becoming a main issue, also within Architect-organizations. To link the parts of the knowledge-triangle practice, education and research forms the basis for possible solutions – within the context of the building design-engineering. This context can be represented by the Product-Process-Organization model. This integral approach is the basis for integral solutions, by structuring knowledge of design and engineering within the design team.

A model for structuring knowledge on different abstraction levels is found in Methodical Design, a system theory based on the combination of the German design school (Pahl, Beitz and others) and the Anglo-American school (Archer, Krick, Jones and others). Methodical Design is a problem oriented model based on functional hierarchy, which can be applied on several levels of abstraction and makes it possible to link these levels of abstraction with the phases in the design process itself.

This paper describes the research methodology, based on Methodical Design, as used in a doctoral design related to practice and the 6th European framework research project EURACTIVE ROOF-er. The research methodology – as quasi-experimental design – uses the structuring method of the Methodical Design to investigate how this specific design method and associated design tools can support the collaboration between designers and engineers.

INTRODUCTION AND PROBLEM DEFINITION

Roofs play a special role in buildings. Their value and impact often significantly surpass the cost ratio they represent in the total investment cost of the building. Traditionally, roofs have a protecting function and their basic design has changed little over hundreds of years. Nowadays however, they are increasingly used as preferred location for mounting additional functions such as photovoltaic systems, roof lights, ventilation devices, insulation and safety devices. The roof will contain more and more aspects which are strongly related with the comfort of the building as a whole. Looking in a wider context, the build environment is dominated by the circumstances related to energy use. As results of Global Warming become more and more prominent, it is necessary to look for new ways to save more energy and to generate more sustainable energy for the comfort in the building environment. [24, 25]. In current building design primarily the façade, as the most prominent building component, is used as integral part of these sustainable comfort systems. This integral approach is lacking for the roof, where these systems are mostly treated like add-on components to the already completed conceptual building design.

Until now the roof was not actively developed to meet the new demands. Active Roofs should change this. Active Roofs is the concept related to this change: the possibility or need to change the culture, process and or product related to the roof. Active roofs, with the described possibilities, implicate a more active role in the process for the roofer and roof-advisor. An active attitude of the total roof-culture is needed in order to design and construct innovative and better roofs. The approach needed for the development of these knowledge and skills is integral as defined by Quanjel and Zeiler [8]. Integral design is meant to overcome, during design team cooperation, the difficulties raised with the early involvement of consultants. This is achieved by providing methods to communicate the consequences of design steps between the different disciplines at early design stages. Related to the specific field of the roofer this means the direct connection of construction /user- and design-related knowledge. A domain-related methodology, therefore has a large group of different users with different backgrounds, which will influence the set up for methods and tools.

The actual state is that there is a gap exists between solutions and application in design practice of active roofs (EURACTIVE ROOF-er, 2005). Roof design and roof engineering with all its existing – traditional – and new functions and applications are most of the time handled like separate and add-on aspects. As complexity and scale of design processes in architecture and in building services engineering increase, as well as the demands on these processes with respect to costs, throughput time and quality, traditional approaches to organize and plan these processes may no longer suffice [26]. This implies defining a process methodology that acts as a “bridge” between architectural elements such as shapes and material on the one hand, and the aspects of indoor climate issues such as overheating and ventilation on the other; an integral approach where all design members have shared understanding – with their own background – on the project [27].

Offering design teams and product developers an appropriate methodology will result in decision support for integral roof-design, within the setting of the primal design. Therefore a decision supportive methodology, within the integral design approach of active roofs, is developed. This implies defining a design methodology that acts as a “bridge” between architectural elements such as shapes and material on the one hand, sustainable energy use and the aspects of indoor climate issues such as overheating and ventilation on the other.

The active-roof design and -engineering, as an integrated product development task, involves solving a design problem. Design problems are a special type of problems and have the following characteristics [28]:

- design problems tend to be large and complex, have both logical and creative components and are wicked
- actors search for a solutions of the design problem within a certain solution space, this solution space is undetermined and the available information is incomplete, since project specifications are never complete or without ambiguity; design problems are therefore ill defined and/or ill structured
- during the design process, actors iterate between design problem and its solution(s)to support the decision making process for these generated design/engineering solutions
- design problems are open ended; it is often not clear when actors have solved the design problem; there is also not one best solution for solving the design problem

Given these characteristics, a methodology to support the design team during the development of the building design is of great importance to clarify the problems and to structure possible solutions. To support the design of large-scale, complex design processes, such as one has in the building industry, a method is presented based on the Methodical design methodology; a matrix orientated approach used in the mechanical engineering domain. Thesis of the research is that, through the use of the characteristics of the Methodical Design, an appropriate support tool is available for Collaborative Design Teams.

WORKING PRINCIPLE

The Methodical Design method is based on system theory and a combination of ideas of the German design school of Pahl, Beitz and others[11, 12, 13, 14, 15, 16] and the Anglo-American's,Archer, Greogory, Krick, Jones and others [17, 18]. Methodical Design combines German and Anglo-American process model approaches. Methodical Design is a problem oriented model based on functional hierarchy, which can be applied on several levels of abstraction and makes it possible to link these levels of abstraction with the phases in the design process itself [19, 20, 21, 22].

The essential element in this model is the design process [21]. The characteristics of the design process can be split up into those related to: strategies, stages and activities. Within the setting of Methodical Design several design-support tools are used: the morphological overview and the Kesselring-method [29]. These are practical tools to structure several functionalities, generate and select possible solutions and can be used for different aspects and abstraction levels.

Due to these characteristics, the methodical approach can accommodate the different subjective interpretations of the requirements, inherent to the design team approach. By structuring the requirements, within each complexity level, the development of the shared understanding in the design team is encouraged. More insight by the design-team-members of different possibilities, from the different requirements, can generate more possible solutions. Through iteration cycle of interpretation-generation steps the set of requirements is continuously refined, and with it also the design solution proposals. The research is focused on the added value of the introduced design-tools, as part of the Methodical Design, within the collaborative design team for the preliminary design-phase.

The research methodology, as quasi-experimental design, uses the structuring method of the Methodical Design. This method can be applied on several levels of abstraction and makes it possible to link these levels of abstraction with each other. By using 4 different levels of abstraction as formulated as functionalities of the problem – integral design / collaborative engineering / sustainable comfort systems / active roofs – the research has a clear framework. Within each level of abstraction - though the research is about developing a methodology for design collaboration - the approach for developing edge conditions and possible approach is the same. This is applicable as well for the research methodology as for the design methodology. For each level the approach (analyze / generate / select / modify) is similar, the development will be different.

Within the Methodic Design, the relationship of the several functionalities and steps can be shown in the scheme below (fig. 1.). In order to structure the research for each level of abstraction (integral design / collaborative engineering / sustainable comfort systems / active roofs) there will be a problem definition / working principle to develop solutions / choice of developed solutions / shape of chosen solution (blue arrow) – for all the functionalities related (vertical column / green arrow).

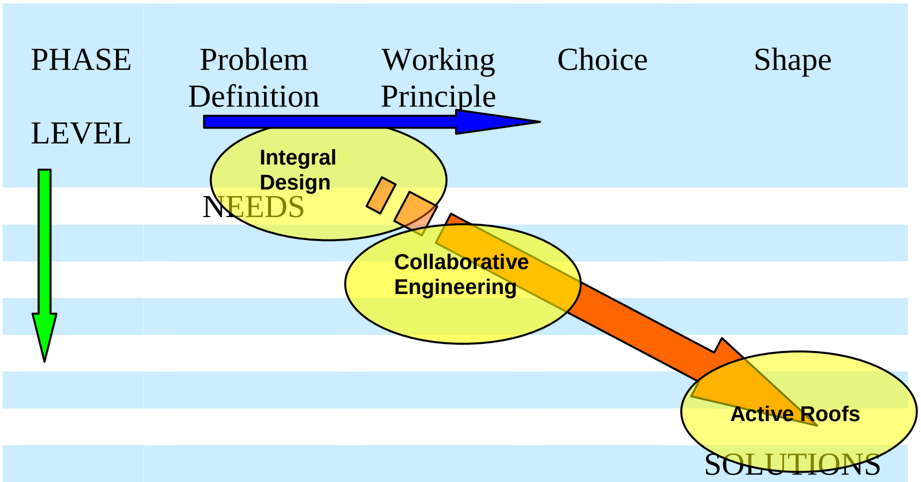

Figure 1. The modular system of hierarchical abstraction as basis of the research methodology

As shown in figure 1, the problem definition generates the working principle for the research methodology. For each level of abstraction within the defined research topic, the several stages and their specific functionalities / needs have to be taken into account. These stages will be described in this paragraph.

The integral approach of the research methodology means different viewpoints on the same topic [8]. For this research the two main viewpoints are that of the designer – the architect – and the engineer – the roofer. By using different typologies of quasi experiments – case studies, scenario's and design task workshops – comparing, verifying and clarifying these experiments, also on research level the different viewpoints are used. By using quasi-experiments, step by step a next – more concrete abstraction level – will be analyzed and verified in order to get a clear view on the influence of the several functionalities / needs related to the specific abstraction level (Integral Design / Collaborative Engineering / Sustainable Comfort Systems / Active Roofs).

Before investigating solutions of 'bridging the gap between architects and roofers, the research will start with investigating the needs and knowledge which causes this gap of design solutions and application engineering.
Therefore the following analysis is made:

- needs in primal design phase for designers / architects
- supply / knowledge from engineers / roofers
- analysis of the differences and similarities between needs and skills of both disciplines

By analyzing all these aspects, an overview of which functionalities are necessary for possible design decision tools to support the participants (architect and roofer) in the setting of collaborative engineering, will be generated.(fig. 2)

Through this first step a further analysis of the roof engineering / installer industry is possible, to develop a model of competence profiles. This model will show which steps are necessary, as a path to success, for creation of appreciation for this specific industry on different levels in the context of collaborative engineering: organization / process / product-level. To have a

reference with practice the competence profile of the façade engineering / installer industry, as a successful path-to-success, is used.
Next step is to set up functionalities related to design- and engineering aspects and generate them in a database useful for both architects and roofers. Related to the different users, two different menus are developed to search and combine the knowledge needed, to design and/or engineer an active roof. The database will be developed in a web-based-setting to facilitate the different users with the needed knowledge, related to design and engineering active roofs.

From the practice, case studies are used to show which kind of problems are there in the traditional design process for roofs and facades, as part of the total building design, in relationship with engineering aspects.

Three different projects will be analyzed (to prove the assumption of the problem);

- comparison of traditional process and collaborative process-approach
- SWOT-analysis with focus on communication and information sharing
- in relationship with architectural concept product/system requirements and technical facilities
- determine criteria of specialist interaction aspects of collaborative design and engineering
- feed-back for aspects of a competence-model and aspects engineering knowledge supply for roofers in the design process

The designed competence profile will be used to set up several, process-based scenarios, used for testing and modifying the data-base-structure as part of Methodical Design methodology. The experiments for testing will redefine the data-base-structure and give a more precise view on the influential and important aspects and functionalities of the knowledge needed in the primal design/engineering phase. Through these steps more precise aspects which are necessary for using more optimal the Methodical Design Tools and Database, can be developed. (fig. 3)

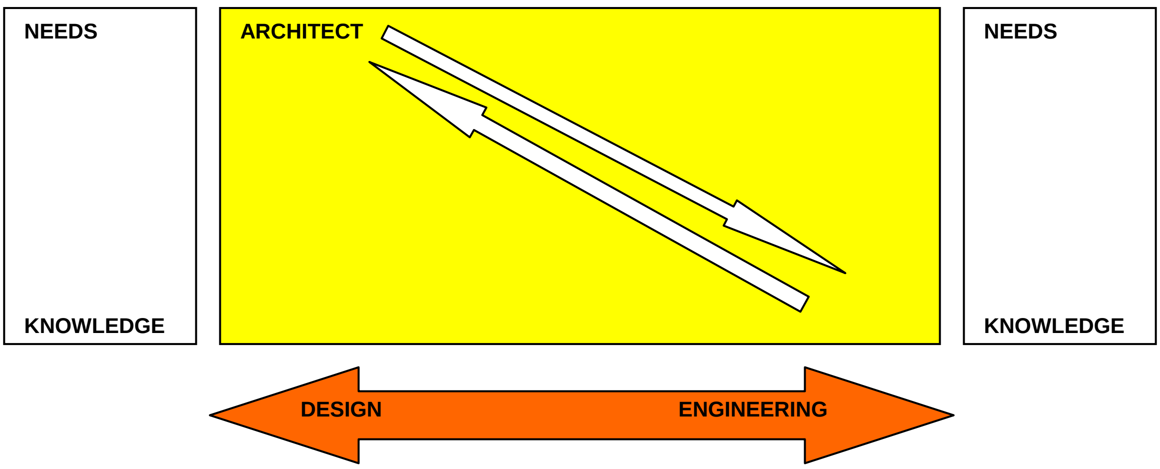

Figure 2. Relationship of needs and knowledge of architect and roofer, from engineering to design

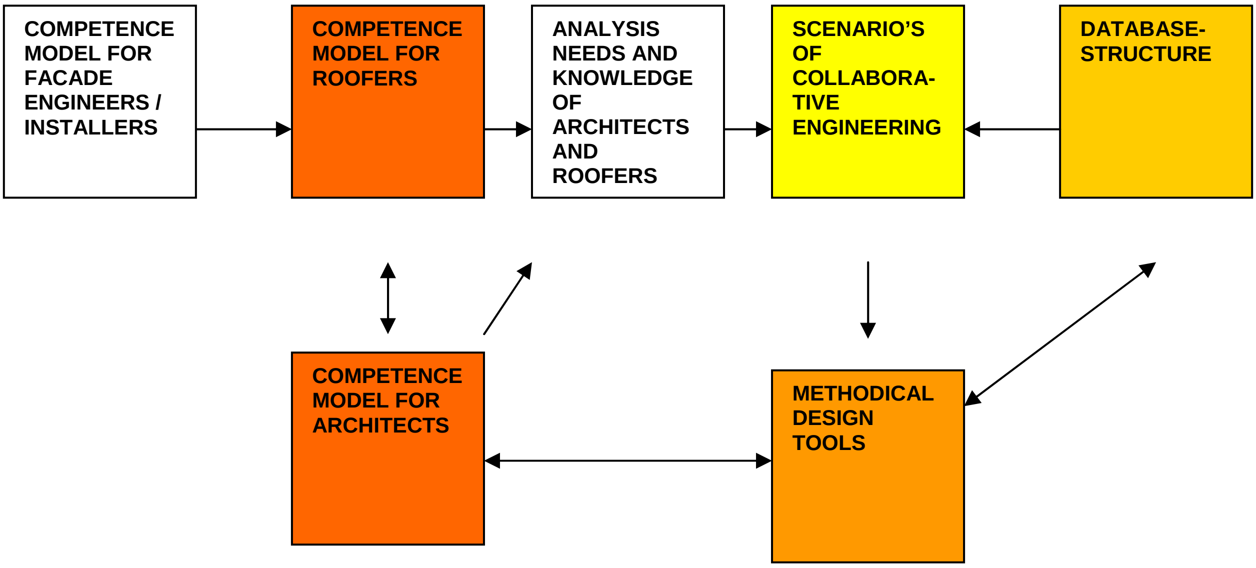

Figure 3. Process scheme for designing / engineering active roofs

Crucial point by using the experiments, in relationship to the 'theoretical model of the Methodical Design, is the connection to a 'realistic model which is part of the design-practice. For constructive implementation / exchange of knowledge within the design process there are 3 main different possibilities of knowledge exchange / generation:

- reflection in action [30]: which connects the design situation in interaction with a framework for the several disciplines
- shared knowledge – heterogeneous engineering [31, 32, 33]: a process of aligning cognitive and social-political elements to create and realize a good design
- 'bricolage' [33, 34, 35]: the use of situational resources tendering with the resources at hand.

The setting which is chosen is that of 'the reflexive practice [32, 36]. This setting has characteristics which makes it, comparing to the other two possibilities, a realistic setting which we can use to verify and validate our theoretical model of Methodical Design. It is a setting which inhabits characteristics to, as in a design-process, have a rational feed-back on the former design-steps and design-decisions and to support the next design-steps and – decisions in a realistic setting with several disciplines.

Mostly the verification of a new methodological concept is done by experiments with student groups (novices) [37] or with design groups within one company [38]. The relevance of the research methodology for practical use in a realistic setting is improved by using experienced designers and participants (professionals), as there is a major difference in approach between novice and experienced designers [39, 40].

CHOICE / SELECTIOIN PHASE

With the found functionalities from the Working Principle – in relationship to the use of the Database-structure and Methodical Design – the scenarios for a serial of quasi experiments will be set up and modified. This is a set up for a prescriptive method to develop appliances for supporting the collaborative design- and engineering-process in combination with engineering knowledge supply.

The quasi experiments will have the focus on the following aspects:

- communication between the design- and engineering solutions for active roofs
- generation of more possible design / engineering solutions for active roofs
- the use of the support decision tool to generate these design / engineering solutions

The same format, related to the Methodical Design, will be used for the set up of the quasi experiments as well for the verification and clarification.

The quasi experiments will be build up in a step by step format with alterations on the group setting (collaborative engineering), group tasks (use of sustainable comfort systems, design /engineering active roofs). The experiments will have the format of design task workshops and/or master classes related to design tasks. The same methodology used on the level of research as a whole is also used on the other abstraction levels within the research; in this case the quasi experiments and the verification of these quasi experiments. (fig. 5.) As example, related to the setting of Collaborative Design and the use of the Database and / or Methodical Design tools we can define the following functionalities / needs related to these quasi experiments:

- situation of the design-team without the use of database or methodical design tools
- situation of design team by using database or methodical design tools, without training
- situation of design team by using database or methodical design tools, with training

The experiments will be done in a serial with feed-back; comparison / selection of the type of experiments as the use of several verification methods as well as the results of the quasi experiments itself. The specific functionalities and results of the experiments will be verified with the use of different verification-methods.

SHAPING PHASE AND DISCUSSION

By using the Kesselring-method a comparison of the different abstraction-levels can be made, finally through the quasi experiments to optimize the experiment and the introduced tools for communication / knowledge exchange. Realization and functioning of the most successful generated tools can be improved. Based on the found and verified functionalities / needs for each abstraction lever an overview of the most successful support tools and scenario's can be generated as part of the developed methodology. Further improvement and research topics for 'how to get there can be evaluated. Through this iteration process more insight in the knowledge exchange of architect and roofers is generated and requirements for the support tools will become clear and can be modified when the circumstances change – the modifying phase. Within a wider range of the EURACTIVE ROOfer project this iteration process can be used for developing the use of the Database structure and the Methodical Design Tools training-set up by the several Collaborative Design team-members for the Active Roof Design [41].

The actual phase of the described research methodology is the selection-phase for defining and applying the case-studies and quasi-experiments. Final results of the research methodology are:

- insight into the needed knowledge / skills of designers (architects) and engineers (roofers) in designing Active Roofs, as part of the total building design

- development of design-decision tool(s), within the setting of Collaborative Design teams, which can structure the needed knowledge from the different design-team members in order to design/engineer these Active Roofs

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A method for evaluating the problem complex of choosing the ventilation system for a new building

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SUMMARY

The application of a ventilation system in a new building is a multidimensional complex problem that involves both quantifiable and non-quantifiable data e.g. energy consumption, indoor environment, building integration and architectural expression. This paper presents a structured method for evaluating the performance of a ventilation system in the design process by treating quantifiable and non-quantifiable datasets together. The method is based on general morphological analysis and applies cross-consistency assessment to reduce the problem complex, thus treating the multi-dimensionality, the uncertainty and the subjectivity that arise in the design process on a sound methodological and scientific basis. Using a distance analysis of the shared values, the solution scenarios may be plotted relative to each other, which provides the designer with an illustrated 'space of solutions'. Herein the designer may view multiple ventilation solutions and navigate between them, evaluate the differences and choose a suitable ventilation system in terms of energy consumption, indoor environment and architectural quality.

INTRODUCTION

Analysing problem complexes like choosing the ventilation system for a new building presents the engineer and the architect with a number of methodological difficulties. The issue involves both quantifiable and non-quantifiable variables that are intercorrelated and due to the early stage in the design phase information is incomplete, missing or undetermined.

In this contribution we use morphological analysis to decompose and structure the problem complex into both objective (technical) and subjective (architecture) variables while treating incomplete information in a structured manner. One of the advantages of morphological analysis is that such combinations are valid. General morphological analysis has mainly been used for socio-technical problems [1] but also in a building related context e.g. to improve the comfort of various ventilation concepts [2].

Here we couple morphological analysis with cross-consistency assessment and a distance analysis to establish the feasible 'space of solutions' for different ventilation concepts within different building envelopes. This process is time-consuming but we have automated it by implementing it in a simplified yet fully dynamic building simulation program BuildingCalc [3] programmed in Matlab [4].

In the Methods section we describe the requirements and general theory for the method while illustrating it with a limited example. The results section contains the results and the xx at last we discuss the consequences that may be derived from the method and results.

METHODS

The methods section contains a chronological description of the methodological approach: establishing the building requirements, generating the morphological box, performing the cross-consistency analysis and the evaluation criteria and tools. Xx example

Requirements

The requirements for the ventilation system are specified in prEN15251 [6] and we use these to establish the initial design criteria for the ventilation system. In prEN15251 the ventilation and thermal comfort criteria are specified for different building types. The criteria are given as intervals in three categories I, II, and III for different building types. Thus we require initially the building owner to specify the use of the building, the desired category of indoor climate, the type of materials to establish the building pollution and the overall placement of the building on the premises to establish external shadows and prevailing wind conditions.

Table 1. The overall requirements specified by the building owner. Data is obtained from prEN15251 [6] and Danish Building Regulations [7].

Building type	Indoor climate category	Building pollution (no smoking)	Total vent. rate [l/s/m^2]	Temp. range [°C]	Energy frame [kWh/m^2]	External shadows	Wind conditions
Single office 18m^2	I	very low (0.5 l/s/m^2)	1.5	21.0-25.5	95.5[1]	None	Good enough for nat. vent.

Morphological analysis

The background for general morphological analysis is to identify and investigate the total set of relationships or configurations contained in a problem complex.

The morphological analysis was developed by Fritz Zwicky and is described in [5]. It is used to structure *messes*, which are complex issues without well-defined form, into problems of unambigiuous form and dimension.

In this contribution the morphological analysis is used to generate a problem matrix that enables us to decompose the problem complex into sets of functional subsystems. The decomposition is carried out hierarchically and continues until arrived at simple building functional components that can be described with a single variable. There are no formal requirements that the variables are of the same unit; technical, aesthetical and architectural elements, known or hypothesized, are treated together to create a space of solution for the designers. Hence the method is flexible xx the example which is used here does not encompass all ventilation variables or dimensions possible.

[1] At a heated floor area of 3000 m^2.

The decomposed problem is put into a 'box' which is depicted as a chart in Table 3. The dimensions and variables are listed in the left columns and the values are listed in rows. The combination of variables with arrays of values makes up the morphological box.

To start the morphological analysis we need to:

1. Identify the *aspects* of the problem complex. Aspects are also referred to as *dimensions* because they represent the sides in the n-dimensional morphological box. In Figure 1 comfort, energy and building characteristics, and aesthetics are chosen as dimensions.
2. Each dimension is governed by a set of variables. E.g. comfort is achieved through fresh air supply and the ability to cool or heat when necessary (Figure 1).
3. Each variable may attain a well-defined range of values or conditions. In practise the ranges are discretized on the basis of a user perception of low, medium and high values.[2] Thus uncertain or hypothesized values are represented on a backwards traceable methodological basis.

Table 3 illustrates the morphological chart. The imaginative values (high, medium, low) are exchanged with real numbers that *we* regard as representing high, medium and low values. The chart has been reduced for the purpose of illustration and to reduce computation time. Thus the parameters in Table 2 are fixed.

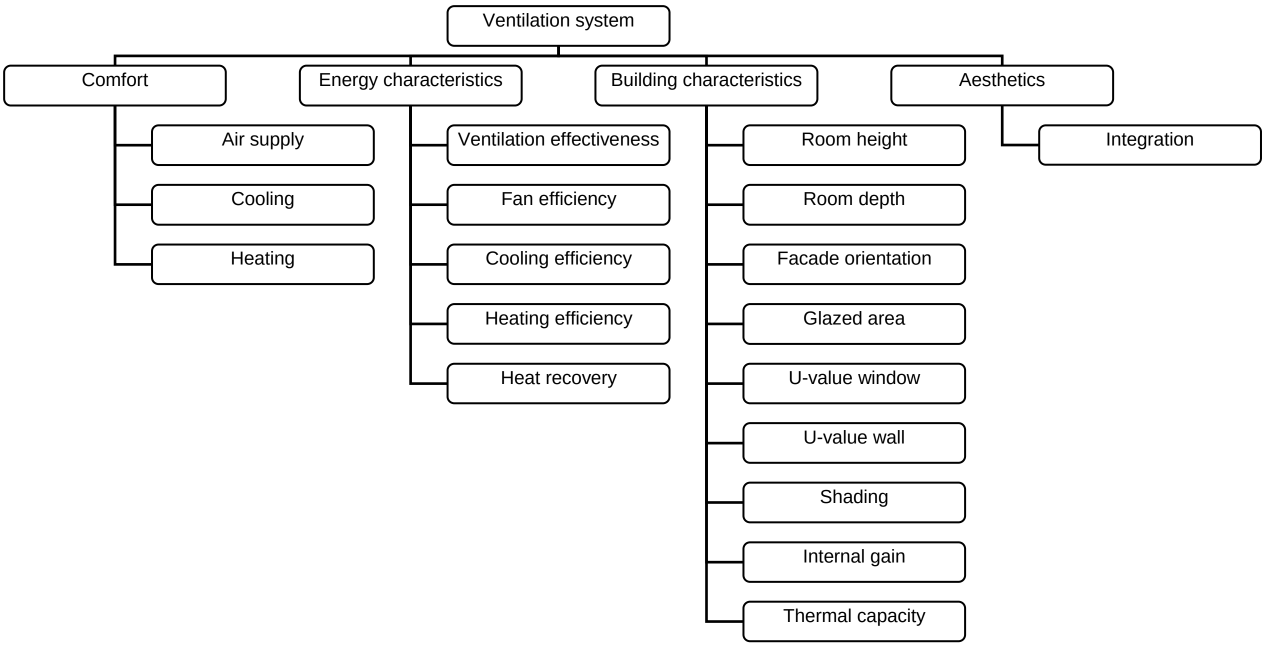

Figure 1. The problem complex decomposed into dimensions and variables.

Table 2. Fixed and excluded values from the morphological chart in Table 3.

Depth of room	Average U-value of window	Average U-value of wall	Façade length	Façade orientation	Min. shading factor
6 m	1.4 $W/(m^2K)$	0.15 $W/(m^2K)$	4 m	South	0.2

Four system/user profiles have been specified: working hours, outside working hours, heating season and outside heating season.

[2] The exact number of discrete values is user defined.

Table 3. Example of a morphological box for a ventilation system in a building. The greyed out areas represent the reduced problem which is used in CCA in Table 4.

Dimensions	**Variables**	**Values**			
Comfort systems	Ventilation 1 Supplied air* [l/s/m²]	Natural **calculated**	Fan-assisted **1.5-3.0**	CAV **3.0**	VAV **1.5-7.5**
	Ventilation 2 Supplied air* [l/s/m²]	Natural **calculated**	Fan-assisted **1.5-3.0**	CAV **3.0**	VAV **1.5-7.5**
	Cooling 1 Max cool. power [W]	Mech. increa-sed vent. rate **calculated**	Night cooling **calculated**	Cooling unit **10**	Chilled beams **30**
	Cooling 2 Max cool. power [W]	Mech. increa-sed vent. rate **calculated**	Night cooling **calculated**	Cooling unit **10**	Chilled beams **30**
	Heating 1 Max heat. power [W]	Radiator ∞	Heating coil **10**	Heat pump **10**	
	Heating 2 Max heat. power [W]	Radiator ∞	Heating coil **10**	Heat pump **10**	
Energy characteristics	Vent.effecness [-]	Low **0.7** (natural)	Medium **0.9** (mixing)	High **1.3** (displacemnt)	
	Fan efficiency SFP** [kJ/m³]	Low **0** (no fan)	Medium **1** (extract only)	High **2**	Very high **2.5**
	Cooling efficiency COP† [-]	Low **1** (cooling w/ outdoor air)	Medium **3**	High **4**	
	Heat recovery [%]	Low **0** (no heat rec.)	Medium **65**	High **85**	
	Heating efficiency [-]	Medium **1** (waterbased)	High **3** (heat pump, COP)		
Building characteristics	Room height [m]	Low **2.5**	Medium **3**	High **3.5**	
	Glazed area [%]	Low **25**	Medium **40**	High **60**	Very high **80**
	Internal gain [W/m²]	Low **10**	Medium **25**	High **40**	
	Thermal capacity [kJ/(m²K)]	Light **144**	Medium **288**	Heavy **432**	Very heavy **576**
Aesthetics	Integration of vent. system	Visible	Somewhat visible	Invisible	

* 'Supplied air' is the required minimum by prEN15251 during occupancy. Some of the systems (CAV, VAV) may be able to supply additional air (for cooling purposes). The user defines the desired ventilation ranges of the systems.

** SFP: annual average specific fan power. A measure of the efficiency of the ventilation fans and the pressure loss in the system.

† COP: annual average coefficient of performance. A measure of the efficiency of the cooling system.

Cross-consistency assessment

After the initial setup and definition of the dimensions of the problem, we apply cross-consistency assessment (CCA). This is to accomplish two things: eliminate the illogical or normatively unwanted configurations and to detect *input garbage*. Input garbage consists of vaguely defined problems, terms or variables with no clear meaning or with more than one meaning etc. If anything is insufficiently defined or described one must go back to the morphological box and redefine or refine input. Table 4 demonstrates the cross-consistency assessment derived from the morphological box in Table 3. All values are pairwise compared by the user for logical incompatibility (denoted by I) or because they are normatively unwanted (denoted by U). The remaining blank intersections are valid combinations.
It is important in the cross-consistency assessment to focus on each value pair to avoid biased assessments that unintentionally exclude value combinations that may be perfectly compatible but are incompatible on a normative basis. In general only strictly illogical combinations should be excluded.

If the intersection between two values is marked all the configurations containing the value couple is excluded from the simulation. For an n-dimensional morphological box this leaves xx

Table 4. The CCA matrix from the morphological chart in Table 3. 'I' denotes illogical value pairs, 'U' unwanted value pairs based on a normative judgement.

		Vent1		Vent2		Cool1		Cool2		Heat1	Heat2	Effectiveness			SFP				COP			Heat rec.		Hieght		Glazed area			Internal gain			Capacity		
		Nat.	VAV	Nat	VAV	Increase	Night	CoolCoil	Beams	Raidtator	Raidtator	0.7 (nat.)	0.9 (mech. mix)	1.3 (displcment)	0 (÷fan)	1	2	2.5	1 (nat)	3	4	0 (none)	0.85	2.5	3	25	40	60	10	25	40	144	288	432
Vent2	Nat.																																	
	VAV	U																																
Cool1	Increase	I		I																														
	Night																																	
Cool2	CoolCoil	I		I																														
	Beams	I																																
Heat1	Radiat.																																	
Heat2	Radiat.																																	
Effectiv.	0.7		U		U	I		I																										
	0.9	I		I																														
	1.3	I		I																														
SFP	0 (÷fan)		I		I	I		I					I	I																				
	1	I		I								I																						
	2	I		I								I																						
	2.5	I		I								I																						
COP	1 (out.air)							I	I					U																				
	3					I	I																											
	4					I	I																											
Rec.	0 (none)		U		U								U	U																				
	85	I		I								I			I																			
Height	2.5	U		U								U																						
	3																																	
Glaz.	25								U																									
	40																																	
	60																																	
Gain	10								U																									
	25																																	
	40																																	
Capac.	144	U		U			U					U							U															
	288																																	
	432																																	
Integrt	Visible		U		U								U	U																				
	Somewht																																	
	Invisible	I	I	I	I				I			I	I	I																				

Examples of questions for assessing the CCA matrix: useraction related xx:

- Is it possible to have a building scenario where a cooling coil in the second cooling system coexists with a night cooling system in the first cooling system? Obviously this is possible. Hence no marking is used.
- Is it possible to have an efficient heat recovery of 85 % and a specific fan power of zero? This is illogical, so the pair is marked with an I.
- Is it possible to have a room height of 2.5 m together with a natural ventilation system? This is a possibility, but the litterature states that natural ventilation performs better if the room height is 3 m xx. Hence an U is used because the user assesses that on a normative basis this particular combination is unwanted.

Evaluation criteria

The morphological approach generates a large number of configurations or scenarios as depicted on Figure 2. We evaluate the scenarios on the basis of total energy consumption, indoor climate (PPD-hours) and annual total cost. For electrical appliances a primary energy factor of 2.5 is used [7].

PPD-hours are calculated from the annual sum of hourly values of predicted percentage dissatisfied [8]. Local discomfort is not considered and relative humidity and air velocity is fixed to 50% and 0.15 m/s respectively. The ventilation rate is not considered as we assume that the minimum ventilation requirements are always meat.

Total costs are based on the sum of the annualized constructing and yearly running costs of the ventilation system.

Distance analysis

One of the most important goals in visualising data is to provide the viewer with a sense of distance between the plotted points. While it is straight-forward to plot scenarios based on two criteria, e.g. energy consumption and PPD-hours, the designer is not provided with the overall picture. With a large number of dimensions (>3), it is very difficult to visualise distances unless the data can be reduced to 2 or 3 dimensions. Hence a dimension reduction is necessary. Metric multidimensional scaling (MDS) is a set of statistical methods that address this type of problem. The methods are available in a toolbox in Matlab and we use them to plot the selected scenarios relative to each other to obtain an instant sense of the distances between them. The interscenario distances are not Euclidean but the *configurational* or technical distances, or more precise the pairwise dissimilarities between the scenarios. I.e. a technical distance of 1 between two configurations indicates that they have been produced from the same set of values except for 1 value that has been replaced by its horizontal neighbour in the morphological box.

It is evident that a certain amount of distortion has to be tolerated when a multidimensional dataset is reduced to two or three dimensions. In typical MDS applications an investigation into the significance of the error is conducted via Kruskals Stress criterion ref xx (stress1) where values are excellent below 0.1 and unacceptable above 0.15. The stress1-value on Figure 4 is 0.12. If the criterion is exceeded the user must reduce the number of selected configurations or raise the number of depicted dimensions from 2 to 3.

RESULTS

In this section we show the results that may be obtained using the method. The initial requirements established by the fictive building owner are shown in Table 1. A total of 248,832 scenarios was simulated with BuildingCalc taking approximately 80 minutes on a laptop with a Pentium M processor running at 1.86 GHz and 1 GB of RAM.[3]

In Figure 2 all the configurations derived from Table 3 and Table 4 have been simulated and depicted with respect to energy consumption and PPD-hours. With the BuildingCalc tool it is possible to pick the most interesting configurations, typically with respect to low energy consumption and low PPD-hours. Figure 3 depicts the predominant value settings of the 30 selected configurations. With the predominant values as starting points it is possible to reformulate the morphological box and resimulate with refined values.

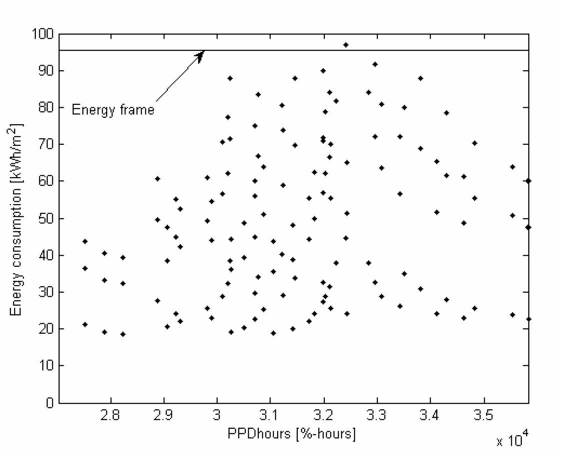

Figure 2. The distibution of ventilation configurations produced via the morphological box.

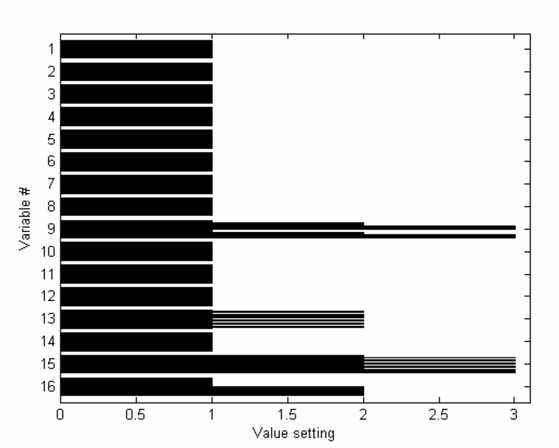

Figure 3. Predominant values for the 30 selected configurations. The variables are numbered consecutively as in Table 3.

Establishing a linear correlation between bubble size and total costs we use the MDS method to plot the selected scenarios relative to each other in
Figure 4. We may navigate between them and explore their similarities simply by clicking on the points with the mouse.

[3] Only a part quantity of the total number of configurations requires lengthy thermal simulation time.

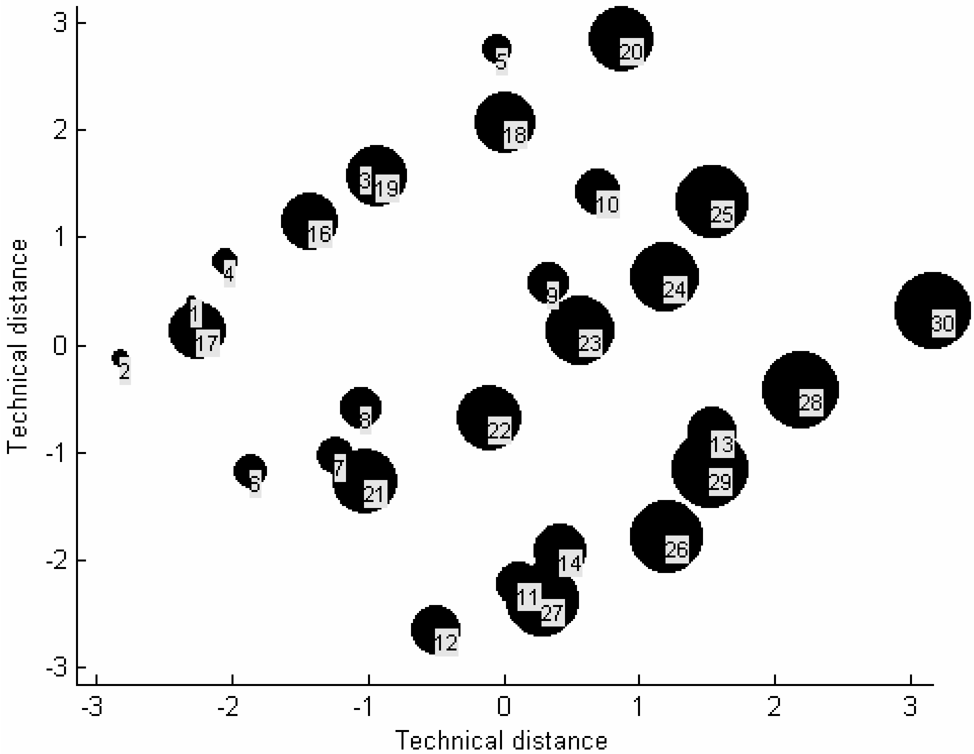

Figure 4. The relative technical distance between the 30 selected scenarios. Bubble Size xx

xxThe distances between the points are the distances expressed in terms of 'technical distance'. Two points closely spaced represents two ventilation scenarios that are very similar to one another. The MDS method visualizes the morphological chart to the engineer and he is able to distinguish between multiple scenarios because their visual distance is a measure of their similarity. Thus
Figure 4 illustrates a space of solutions, where it becomes clear that scenario 3752 xx is better than 2030 if 3758 is not entirely suitable to the engineer or the architect.

DISCUSSION

The method described in this paper offers a deep insight into the problem complex of choosing a suitable ventilation system for a building. The morphological approach exempts the engineer from generating biased scenarios and by coupling the morphological analysis with cross-consistency assessment we ensure that the subsystems are unambiguously defined. The method is fully scaleable to the needs of the user and it allows for treating quantifiable and non-quantifiable datasets parallelly while remaining backwards-traceable. However the method also requires a certain amount of skills and practise to be able to setup the morphological box, cross-assess and iterate, but the procedure helps to keep out garbage input and to decompose the problem complex correctly.

The MDS method is limited to a certain number of configurations and dimensions, but it provides an excellent visualization and overview of the feasible configurations to discuss among engineers and architects in the design phase.

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Energetic Sustainability Assessment about Passive Solar Systems by a Finite Differences Code

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SUMMARY

The use of bioclimatic solutions in architecture has been for a long time in Europe a consolidated procedure, especially developed in northern countries. In the last years, in Italy too, It has been followed through a good policy to increase the construction of buildings equipped with solar energy harnessing system; many architectural competitions expressly ask for passive solar architectonic solutions. In this study, It has been assessed the energetic performance of a solar passive building by a finite differences code (TRNSYS).
Many tests and simulations have been carried out on a residential building, changing the configurations of the building envelope and transparent or opaque surfaces.
It has been calculated the thermal performance of a south-facing building, in which the façade, chancing configuration, is made up of opaque wall and windows, or a completely transparent surface, testing many kind of glasses (direct gain sunspace) or small sunspaces against a thermal storage matt wall.
Thanks to these final results, It has been possible to make some considerations about different kinds of proposed passive solar solutions, picking out the more convenient ones as to their energy performances and appropriate lighting and thermal indoor comfort.
This work would arise a contribution about:

- a correct way of bioclimatic design, by using transparent insulating materials,
- new kinds of direct gain sunspaces, that allow energy saving of 50%,
- sustainable energy use of building envelope.

INTRODUCTION

New Italian law about energetic saving (D.Lgs n. 192 of 19 August 2005), impose to equipping new constructions of energetic certification, exposed as a simple label to the door of the building (figure 1).
The normative system develops new limits more restrictive than previous laws (L. n° 10 of 1991) fixing transmittance acceptable maximum values, relatives to 2006 and to 2009, for the building envelope components, that new constructions will have to respect.

Figure 1 – Energetic label

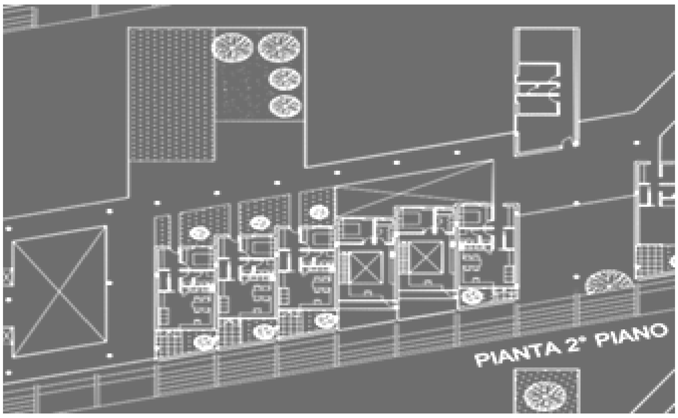


Figura 2 – Detail of the studied plan

In the light of this new laws, object of this work is the energetic behavior of a bioclimatic residential building, characterized from a very particular facade exposed to south (figure 2-3).
Various configurations of this architectonic element have been assumed: the “traditional” one, made of masonry and rectangular windows, the passive solar solution with attached sunspaces to a thermal storage wall, and an other one, made of a completely glazed façade and characterized with “innovative” type glasses (low emissivity triple glass with interstices in krypton) with very low thermal transmittance values (U=0,4 W/m ² K).

METHODS

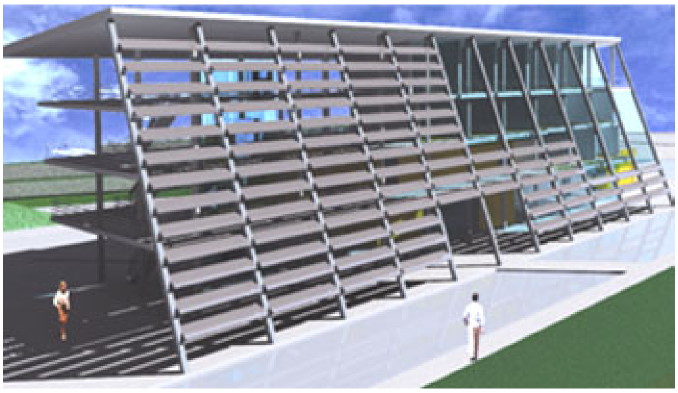
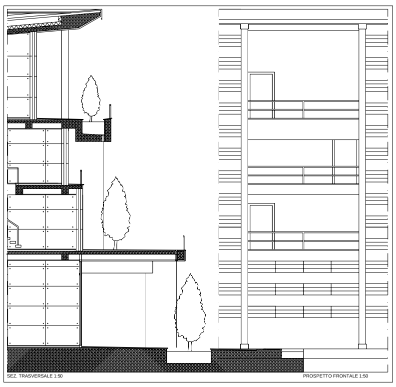


Figure 3 – View of the southern facade

Thanks to a thermal transient simulating software (TRNSYS), it has been possible to calculate the inner temperatures of lodgings and the energetic requirements of the building envelope during the winter period. In order to obtain the temperatures data, the simulations have been carried out supposing a building lacking in heating system, moreover to achieve results about energetic requirements, It has been studied the binomial *building - heating system*, regulated to work every time the dwellings inner temperature falls below 21°C. The final scope has been that to estimate and to confront the behavior of these new transparent materials in relation to passive solar solutions, as classic sunspaces, wide dealt about.
The energetic requirement has been analyzed in the winter time, that is the period in which in Rome is allowed working of heating system, (from 1 November to 15 April, equivalent to 166 days, for a total of 3984 annual hours).

STUDIED CONFIGURATIONS

1. Building with standard windows



Picture 1. Section and front of the building with standard windows

The figures 4, 5, 6 and 7 represent the temperatures and the energetic requirements of the residential construction assuming the southern facade as an opaque wall with standard window. The thermal transmittance of the external walls is U=0,48 W/m² K , the type of window instead is made of double glass with U=2,8 W/m²K and transparency G=75%. Heating system enters in action when the room inside temperature gets below 21°C:

analyzing the temperatures diagrams It is easy to note that with this type of structure the temperature is always under the 21°C, that means that the building has a high thermal dispersion, in winter time about 137 GJ/year.

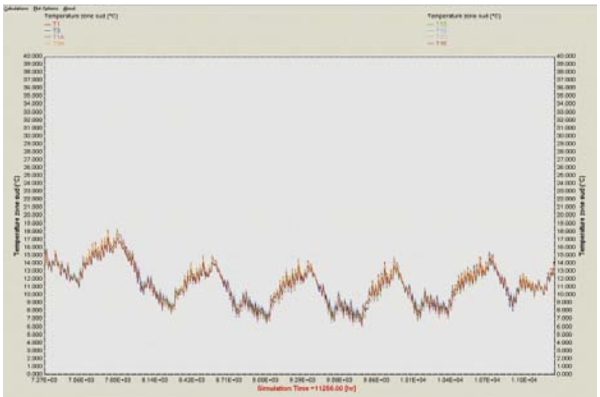


Figure 4 – Diagram of T [°C] of the southern zones of the building.

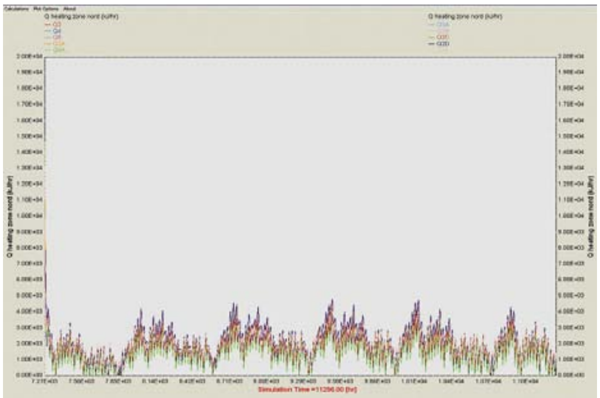


Figure 5 – Diagram of T [°C] of the northern zones of the building

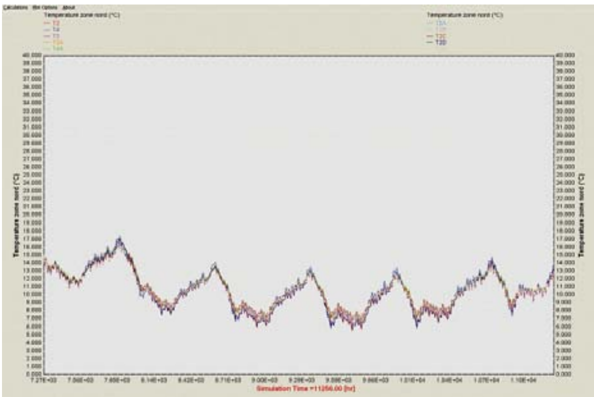


Figure 6 – Diagram of Q [kJ/hr] of the northern zones of the building

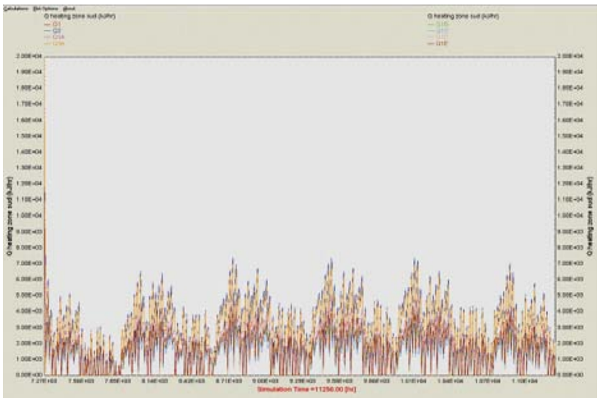
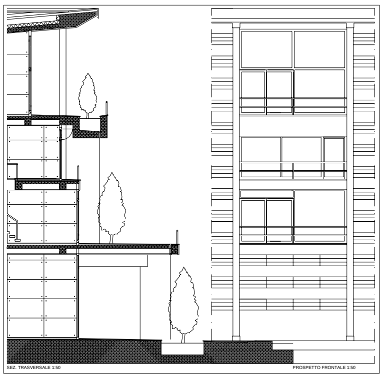


Figure 7 – Diagram of Q [kJ/hr] of the southern zones of the building

2. Building with a completely glazed southern façade

The external walls thermal transmittance is U=0,48 W/m²K, instead for southern facade, many types of glass have been analyzed with different values of thermal transmittance (U) and transparence (G).

The represented diagrams in figures 8, 9,10 and 11 regard the temperatures and the energetic requirements of the residential structure supposing to use triple glazed panels made of an extra-clear glass on which it is applied, by vacuum packed cathode pulverization, a low emissivity noble metals powder. These transparent innovative materials have the skill to achieve very high values of luminous transmission G=60% and thermal insulation, thanks to those noble metals powder characteristics and krypton gas of which It is filled up with (U=0,7 W/m² K).



Picture 2. Section and front of a building with whole glazed facade

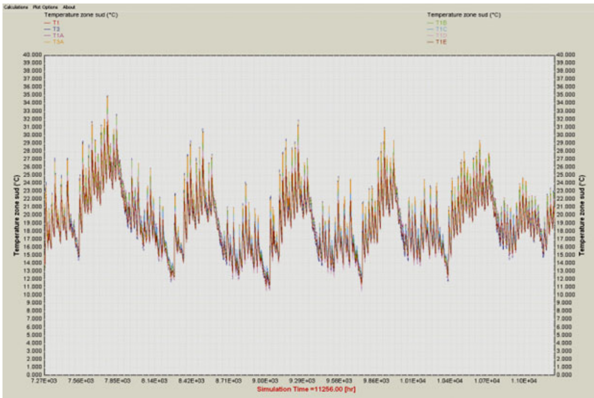


Figure 8 - Diagram of T [°C] of the southern zones of the building

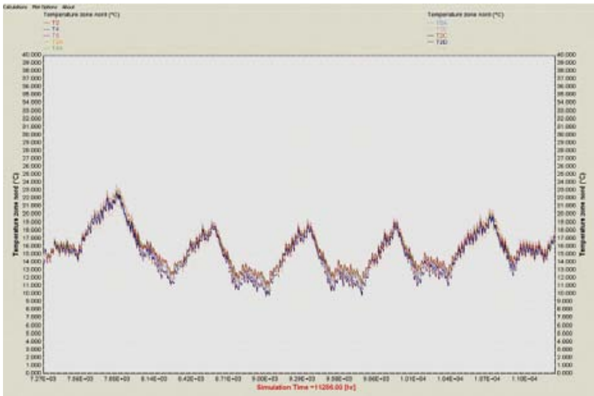


Figure 9 - Diagram of T [°C] of the northern zones of the building

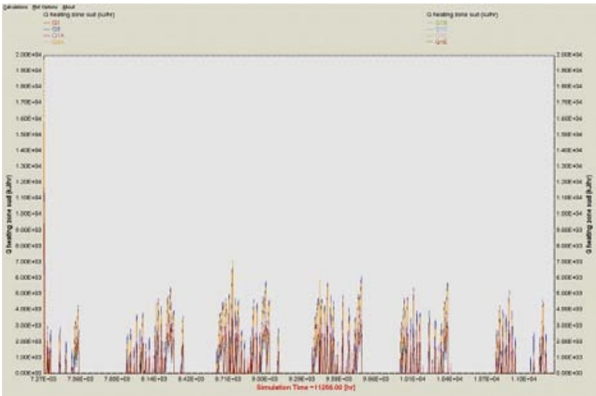


Figure 10 - Diagram of Q [kJ/hr] of the southern zones of the building

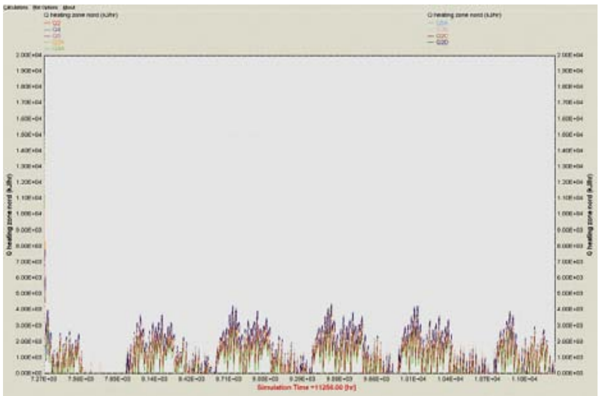


Figure 11 - Diagram of Q [kJ/hr] of the northern zones of the building

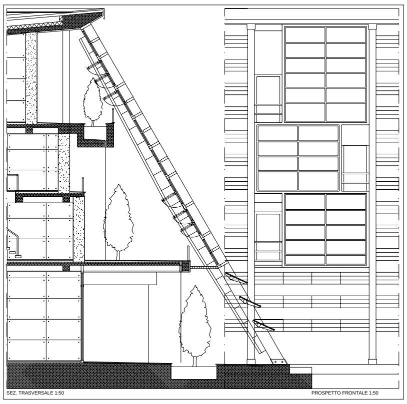
Making use of a single glass, for southern glazed façade, would involve high energetic requirements, because of its transparence value (G=85%), that lets great part of the solar radiation in, with consequent excessive elevation of the inner temperatures on day, while during night, because of its high thermal transmittance value, whole accumulated heat in daytime is leaked out, making go down the temperature one. Instead, in an other studied case, using glasses with lower transmittance and transparence values, energetic requirements diminish, saving energy about 1/3 as regards use of a single glass. Every time the glass transparence value is less than 60%, the energetic requirements increase because the solar radiation cannot go through the almost completely opaque glass and directly irradiate all rooms to rise indoor temperature. The energetic requirement of this solution, made of low emissive triple glass, is lower than the other transparent ones, that is 57 GJ/year (Table 1).

Table1. Building Energetic requirements in proportion to U and G values

Type of glass	**U-values**	**G-values**	**Q Tot. GJ/year**
single glass	U=5,8 W/m²K	G=85%	147
double glass	U=2,8 W/m²K	G=75% (air)	82
double glass	U=1,3 W/m²K	G=64% (argon)	61
double glass	U=1,1 W/m²K	G=63% (krypton	65
triple glass	U=0,8 W/m²K	G=50% (krypton)	70
triple glass low-E	U=0,7 W/m²K	G=60% (krypton)	57
triple glass low-E	U=0.4 W/m²K	G=40% (krypton)	79

3. Building with window and sunspace attached to a thermal storage wall.

The figures 12, 13, 14 and 15 describe temperatures and energetic requirements of the residential building, supposing that southern facade is made of a wall with a standard window and a sunspace attached to a storage wall. The applied heat transfer coefficient of all opaque walls is U=0,48 W/m ² K. For sunspace inclined surface has been analyzed the same transparent configurations already used for studied case 2. The use of a single glass in sunspace southern inclined wall is more convenient than the other types of glass, because of its transparence G=85% allows to have a great thermal gain on the storage wall and all accumulated heat by day can be released by night-time hours. Glasses with smaller transmittance and transparency values increase the energetic requirements, because of reasons already explained before, therefore the storage wall absorbs less heat and the inside room temperatures are lower. The diagrams in figure 12, 13, 14 and 15 regard the configuration with the most convenient type of applied glass (table 2), that is the single one with U=5,8 W/m² K and G=85%: in this case the building energetic requirements are about 102 GJ/year.



Picture 3. Section and front of the building with window and sunspace attached to a thermal storage wall.

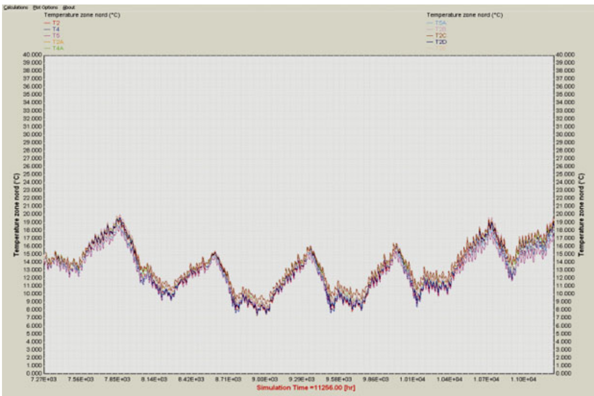


Figure 12 - Diagram of T [°C] of the southern zones of the building

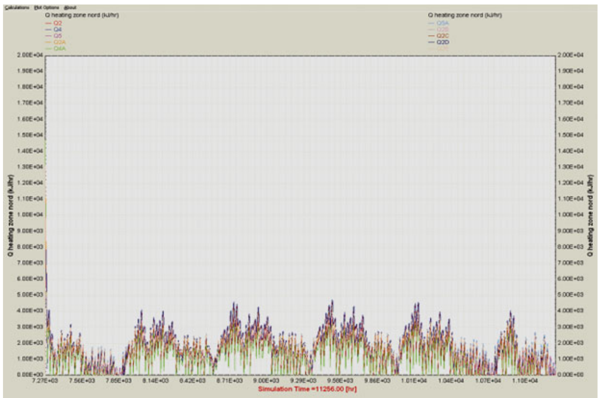


Figure 13 - Diagram of Q [kJ/hr] of the southern zones of the building

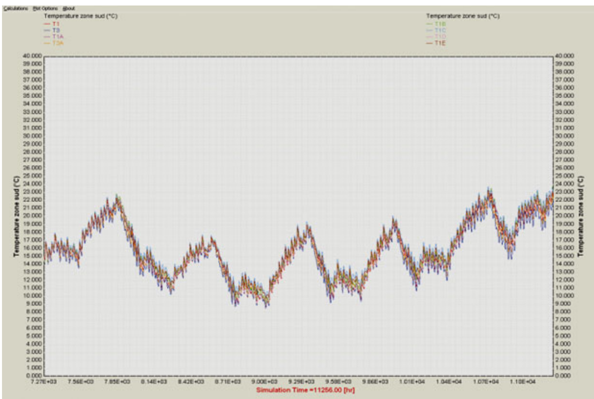


Figure 14 - Diagram of T [°C] of the northern zones of the building

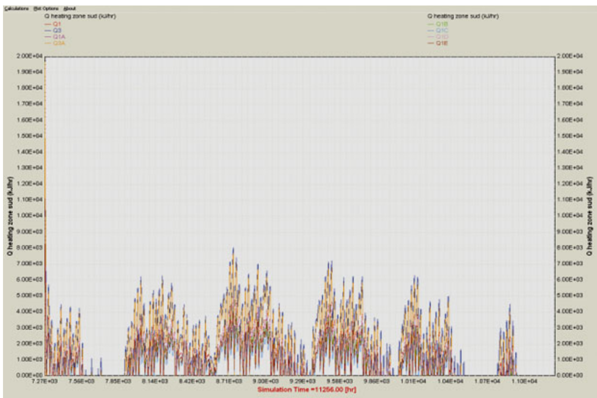


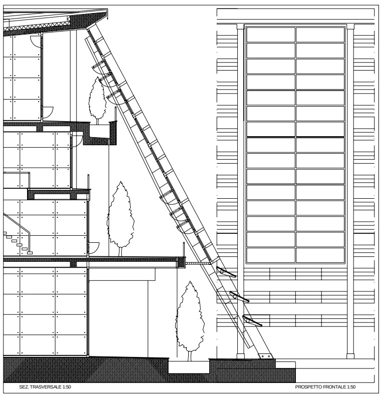
Figure 15 - Diagram of Q [kJ/hr] of the northern zones of the building

Table2. Building energetic requirements in proportion to U and G values

Type of glass	**U-values**	**G-values**	**Q Tot. GJ/year**
single glass	U=5,8 W/m²K	G=85%	102
double glass	U=2,8 W/m²K	G=75% (air)	106
double glass	U=1,3 W/m²K	G=64% (argon)	109
double glass	U=1,1 W/m²K	G=63% (krypton	109
triple glass	U=0,8 W/m²K	G=50% (krypton)	123
triple glass low-E	U=0,7 W/m²K	G=60% (krypton)	114
triple glass low-E	U=0.4 W/m²K	G=40% (krypton)	132

4. Building with sunspaces attached to a glazed wall

In this other studied case, It has been simulated the building configuration, applying to inclined surface the triple low emission glass, instead to glazed wall behind the sunspace, the use of single glass has been very advantageous, because of its high U-value, heating all adjacent rooms. Many configurations have been analyzed (table 3) and the most energetically sustainable solution has been represented in Figures 16, 17, 18 and 19.: applying triple low emissivity glass, characterized by U= 0,7 W/m^2 K and G=60%, the energetic requirements are about 94GJ/year.



Picture 4 - Sunspaces attached to a glazed wall

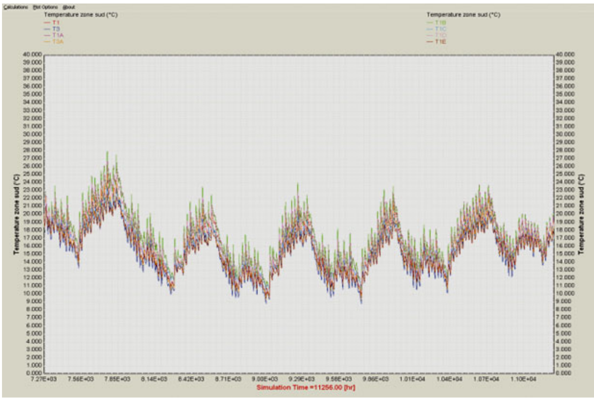


Figure 16 - T [°C] of the southern zones

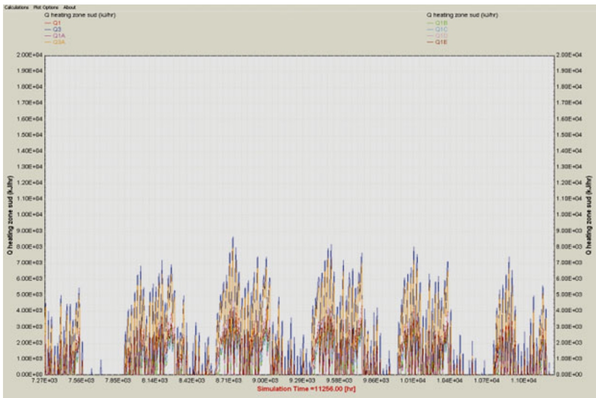


Figure 17 - Q [kJ/hr] of the southern zones

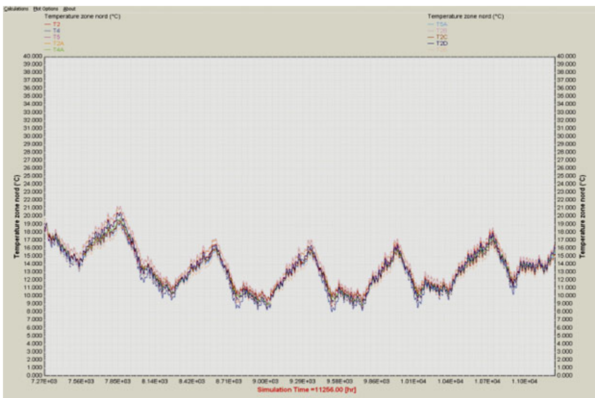


Figure 18 - T [°C] in northern zones

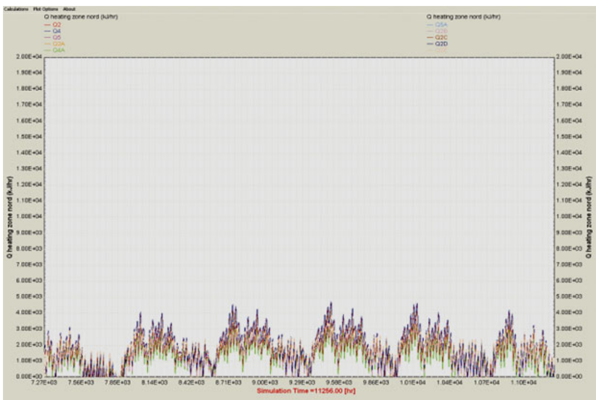


Figure 19 - Q [kJ/hr] in northern zones

Table 3. Building energetic requirements in proportion to U and G values

Type of glass	**U-values**	**G-values**	**Q Tot. GJ/year**
single glass	U=5,8 W/m²K	G=85%	133
double glass	U=2,8 W/m²K	G=75% (air)	110
double glass	U=1,3 W/m²K	G=64% (argon)	96
double glass	U=1,1 W/m²K	G=63% (krypton	97
triple glass	U=0,8 W/m²K	G=50% (krypton)	105
triple glass low-E	U=0,7 W/m²K	G=60% (krypton)	94
triple glass low-E	U=0.4 W/m²K	G=40% (krypton)	112

DISCUSSION

Comparing the several solutions represented in tables 4 and 5, the results are that It is possible to get the best energetic performance, in a southern façade, by employing a triple low-E glass filled of krypton to a whole glazed wall configuration . Therefore, this architectonic solution is the only one that fulfills all new restrictions of D.Lgs 192/2005; in fact its energetic requirement value is less than utmost allowable one in *D climatic zone* of Rome.

Table 4. Energetic requirements of building envelope in winter time

Building characteristic	F.E.P. GJ/year
Traditional windows double glass	137
Southern whole glazed façade(triple low-E glass)	57
Sunspace attached to a storage wall	102
Sunspace attached to glazed wall (triple low-E glass)	94

Table 5. Energetic requirements of building envelope in winter time
ex D. Lgs 192/2005 related to D climatic zone

Building characteristic	F.E.P. kWh/m^{2}year (49.9 max value)
Traditional windows double glass	84.5
Southern whole glazed façade(triple low-E glass)	35.8
Sunspace attached to a storage wall	62.9
Sunspace attached to glazed wall (triple low-E glass)	58.1

The results of the façade made of window and sunspace attached to a thermal storage wall have not been so appreciable. This solution demands less primary energetic requirement than that made of rectangular windows on an opaque wall, but compared with the whole glazed facade, this one presents twice as much annual consumption.
Therefore in any case, this study puts in evidence that all more classic configurations used in the bioclimatic architecture, as sunspace attached to a storage wall, are less convenient than solutions that employ innovative transparent materials, that in fact They do not fulfill italian law parameters.

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Complex Building Automation
Energy Efficient New Construction of Hagen Sparkasse by Klaus auf der Springe

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THE PLANNING TEAM

In mid-2002 the Friedberg architects' bureau Bremmer, Lorenz, Frielinghaus, which had been successful in the architecture competition, was commissioned to plan the new construction of the main branch of the Hagen Sparkasse (savings bank). The Sparkasse had already decided to demolish the 22-storey 1970s office block known for miles around as "Langer Oskar". Redevelopment of the Hagen landmark was not economically viable and instead its spectacular blowing up was to open the door to technical progress. To complete the planning team Skiba Ingeniurgesellschaft für Gebäudetechnik from Herne, which in the meantime has merged with THS Consulting GmbH, Gelsenkirchen, was commissioned with the planning of all the internal technical facilities, including the extensive media installations and the preparation of an integrated facility management and operating plan. The planning team was rounded off with structural planners, sound insulation experts, a fire safety consultancy company and the EGS-plan engineering company for energy, structural and solar technology. The project management for the building was assumed by DBB, Deutsche Bautec Baumanagement GmbH based in Mainz, which originated out of DAL Bautec.
In collaboration with our engineering bureau, EGS-plan drew up the integrated energy plan for the new building, which in addition to the necessary requirements for the shell of the building also had a strong influence all the building's internal technical systems. As the Sparkasse was incurring high energy costs because of the ageing technical systems and façade of the existing building, the client had placed particular value on the economic operation of the new building, without impairing the convenience, amenities or the attractiveness of the building. The use of natural resources formed apart of further considerations, was discussed together with the clients and led to the overall design of the building approved for further planning.

THE STRUCTURE OF THE NEW BUILDING

A five-story building was erected on an area of around 125 m × 50 m. The basement acts as an underground garage with 145 parking spaces and is also used as a storage area, strong-room and for the archives. It also contains the heating and cooling control systems. On the ground floor, in addition to the elliptical entrance and the cash machine areas accessible

outside opening hours, is the expansive customer foyer over 20 m in height with consultation desks, privacy and waiting zones.
An events area, the conference and events room, a Mediterranean-style restaurant and integrated VIP area, a coffee bar, an atrium with a glass roof (22 m high) as well as five shop units make up the remaining space.

In addition to offices, the Sparkasse's conference rooms are located on the first floor. The conference room above the main entrance has been designed in a particularly exclusive manner, taking on the shape of the main entrance. The second to fourth floors are predominantly used as offices, with the Sparkasse's casino also being located on the 4th floor. The Sparkasse offers offices to rent on the floors 1 to 4 of the southwest wing of the building.

INTEGRATED ENERGY CONCEPT

In addition to the user requirements in terms of functionality and convenience, the integrated consideration and planning of a building also includes the subsequent operating costs of the building. In regard to the economic viability of the investment and the subsequent operating and maintenance costs, the drawing up of a sustainable energy concept plays a central role in planning.

In the discussions between all the parties involved in the planning the following objectives were defined:

– Window ventilation rather then mechanical ventilation,
– Air-conditioning or cooling as an exception,
– Reduction in heating and cooling costs,
– As much use of daylight as possible
– Individual room regulation,
– Use of natural resources,
– High level of sound-proofing from outside as the building is located directly adjacent to several bus stops.

The concept also included training of the building users in order to harmonise their expectations and behaviour with the new building.

BUILDING SHELL

In the initial phase of consideration decisions on the building shell were made.

First decision: the office will have box windows, which in the case of rooms facing the street will also have a “deflector pane” for sound-proofing reasons. This deflector pane effectively reduces the noise of arriving and departing buses but still allows the windows to open for natural ventilation as narrow gaps between the deflector pane glass and the facade allow air to be exchanged.

The glass-covered area of the customer foyer, the events area and the inner atrium as well as the entrance area designed as a glass facade are constructed with different qualities of glass in order to create optimum thermal insulation conditions and a feeling of comfort in the interior.

VENTILATION

The building was divided into zones with different ventilation concepts.

The principle of natural ventilation before mechanical ventilation was taken into account in the offices.

Offices adjoining the customer foyer area or the atrium are provided with displacement ventilation via air outlets integrated into the lower sections of cupboards. The air drifts through the room and is directed via sound-proofed facade ventilation elements into the customer foyer/atrium.

The total quantity of air in the offices in conjunction with the adjoining customer foyer and atrium take is balanced out via shutters for natural and/or mechanical ventilation and venting of these areas depending on the weather.

An extensive ventilation shutter control system in the façade and glass roof also ensure pleasant temperatures in the customer foyer and atrium.

A weather station records climatic data, wind strength and direction so that the necessary shutters can be opened for optimum ventilation of the foyer without draughts being felt or rain coming in. Depending on the assigned individual room settings of the offices and customer foyer, the optimum shutter opening is set. However, there are areas in the new Sparkasse building that require mechanical ventilation and venting in the interests of user comfort, in particular the conference rooms in which many people are frequently expected, but also in the gastronomic areas of the restaurant, coffee bar and casino. These areas are mechanically ventilated, whereby of course ventilation devices with heat recovery integrated as circulatory composite systems are used. The ventilation controls are located on the roof behind a technical membrane.

SMOKE EXTRACTION

As well as the ventilation plan, smoke extraction forms an integral part of the installation technology and the associated natural ventilation and venting, particularly in the customer foyer. As part of the integrated planning of the building a concept was developed in collaboration with the fire safety consultancy company Lorenz that makes use of the existing ventilation and venting technology as far as possible.

This means that the ventilation and venting shutters, as well their control systems, assume an important dual function in extracting smoke from the customer foyer. If the event of a (hopefully never to occur) incident the weather station data is also used, but this time for the purpose or extracting smoke from the foyer as quickly as possible. Here the key is not an absence of draughts, but opening the façade and roof shutter so allow the greatest possible movement of air. At the same time the opening of shutters that could cause banked-up pressure in the foyer must be avoided. In accordance with the fire safety plan, in addition to the corridor areas from which smoke must be extracted, there are three smoke extraction zones – the atrium, events area and inner courtyards. Smoke is extracted from the atrium via smoke-heat removal windows (SHR windows) in the roof, which are also used for natural ventilation and venting. Electric doors to the shop area which in the event of a fire alarm in the atrium can be opened by way of a fire alarm system (FAS) contact within 90 seconds serve to provide an additional air flow, and at the same time the FAS controls the closure of the roller grilles in front of the doors. The partition wall to the events area should be closed or remain closed, the ventilation and air-conditioning switch of and the natural ventilation and venting shutter close. The SHR windows in the roof of the atrium are arranged in such a way that groups can be formed in various areas so that depending on the wind direction and strength at least two of the four areas can be opened. The specifications were determined for each flow situation. A similar smoke extraction plan applies in area 2, the events area. Here, the shading systems must be moved into the “open position” in the event of a fire.

Smoke is also extracted from the inner courtyard via SHR windows in the roof that are used for natural ventilation and venting. Here, electrically operated additional flow openings in the outer façade provide additional flow and these can also be opened within 90 s by way of an FAS contact. The partition wall between the atrium and events area as well as the smoke extraction openings in these areas must be closed in the event of a fire alarm in the inner courtyard, the ventilation and air-conditioning systems switch off and the natural ventilation and venting shutter close.

HEATING, COOLING, BUILDING ELEMENTS ACTIVATION SYSTEM

Via the comprehensive building elements activation system provided in the office floors a basic temperature equalisation of the building is achieved. As a three-pipe system the building elements activation system can be used for both heating and cooling. However, in order to keep to the principle of optimising energy use cooling has been largely dispensed with. Therefore it is permissible for office temperatures not to be kept constant throughout the day in the summer months as is the case with cooling ceilings, but to fluctuate in accordance with changes in the outside temperature. If this basic temperature equalisation is inadequate the building is heated by heaters. The offices have individual room regulators which in addition to radiator valve regulation provide important room data to the building process control systems. Evaluation of this data optimises the deployment of building element activation. In the entrance hall, customer foyer, consultation desk area, events area and conference area underfloor heating/cooling is built in in combination with a double floor as a three-pipe system. The consultation, training and board rooms are also supplied via a three-pipe system, though in connection with single room regulation, cooling ceilings and cooling sails. Depending on the recorded room and weather station data the building component activation system is charged during the night. The cooling machines are designed for acyclic use by the consumers, i.e. not all consumers can be cooled at the same time. The system was optimised to use the building element activation system at night, whereby a basic load through constant consumers of cold, e.g. the dataprocessing and server rooms was added.

The cooling system allows the following four operating modes:
1. Normal operation
2. Heat coupling to the three-pipe system
3. Free cooling operation 1,
4. Free cooling operation 2.

As the various operating modes can change smoothly from one to another, the operating management for automatically selecting the relevant operating mode has been optimised to this use.

Normal operation is activated at external temperatures of over 20 °C . It is defined in that free cooling operation is not possible due to too high external temperatures and no more heat is being requested by the building element activation three-pipe system.

At external temperatures below 20 °C heat energy is required for the three-pipe system (operating mode heat coupling). Free cooling operation 1 (external temperatures below 14°C) is a combination of free cooling operation with simultaneous operation of the cooling machines. It is primarily intended for concrete core cooling, but also for the "server room" cooling cycle. Due to the hydraulic design of the installation as a dual-circuit system and the by-pass regulation on the cooling machines, simultaneous operation of the cooling machines and free cooling can be achieved.

Free cooling operation 2 is classic free cooling operation, without simultaneous cooling machine operation. Free cooling is triggered at external temperatures of below 7°C or if the specified value of the flow temperature of the "server room" cooling circuit exceeds the external temperature by more than 5 K.

ROOM AUTOMATION

For room temperature regulation and control of solar protection, daylight and artificial lighting use the building the building will have a bus-capable room automation system as a result of the system architecture based on the LON Mark standard.

The full functionality of the room automation stations is shown on the central building control system. The individual room management system allows individual necessity-related regulation of thermal by each room user with simultaneous minimum energy requirements. The integration of all room functions, in particular the combination of heating, ventilation, climate control with daylight utilisation, lighting, solar protection, access control and safety systems is also possible.

The solar protection system has variously shaped and coated panels which in the upper section direct light into the room. Depending on the position of the sun the panels are automatically moved into the optimum position so as to provide protection from the sun at the same time as continuing to guarantee optimum utilisation of daylight. By way of a daylight measuring sensor, which not only transmits the light intensity, but also the elevation and azimuth to the room automation devices, depending on room occupation the lighting, solar protection and, where available, also antiglare systems are moved into the optimum position. The solar protection system also minimises external heat gain or, in winter, serves to enable solar energy to be used for room heating.

The following take place via the bus interface
– individual, group and zone control of the individual rooms,
– energy requirement signalling to energy production and distribution systems,
– individual use with the start-up and service software,
– individual adaptation to changing room conditions and
– access to every information point of the connected regulation and control devices.

Local operation is via room controls. Programmable are functions such as, for example:
– remote control and display via LonMark on LON bus,
– manual recording of room use via room controls, or automatically via presence recorders
– recording of variables, e.g. by way of window contacts (energy block),
– regulation of the room temperature in heating and/or cooling operation and air exchange (comfort operation),
– reducing or increasing the room temperature when not in use (stand-by operation)

– individual allocation of timer programs,
– if necessary integration of all room functions including the combination of heating, ventilation, and climate control regulation with lighting and solar protection
– rapid warming or cooling,
– free night cooling,
– morning purging,
– summer/winter compensation.

RECORDING CONSUMPTION AND ENERGY MONITORING

In order to be able to monitor the ambitious targets of optimised building operation, the Sparkasse Hagen has decided to use remote-readable metering devices and to commission EGS-plan with monitoring the energy for two years.

In addition to the advantage of allowing easy-to-understand running cost accounting with the tenants through extensive measurements and meter data, the measurements recorded by the central building control system are to be evaluated in collaboration with all those involved and used for further optimisation of installation operation and energy use. In this way both the operator and user can see the sustainability of the planning and optimisations.

Of course, the optimisation phase assumes the collaboration of users and operators. But caution is required as measures not accepted by the users will rapidly lead to dissatisfaction with the building. Eagerly awaited therefore is the first building data which after the introductory phase of the extensive building control systems should provide information about the achieved level of economic viability and user comfort.

Facts and technical data

Heating:
2 gas condensing boilers each with approx. 790 kW modulating, distributed to 20 heating circuits

Cooling:
3 cooling machines as screw-type compressor each with 438 kW cooling output at 139 kW compressor output distributed to 26 cooling circuits including 3 pipe system

Sprinkler system:
CEA/VdS-tested sprinkler system with side wall sprinklers as wide-angle nozzles in the offices, umbrella sprinklers and mist sprinklers in the other areas, pump output 90 kW.

Argon gas extinguishing systems:
For the archives, server rooms, back-up rooms, central distribution rooms data-processing with automatic fire alarm system and a fire early detection system.

Sanitary installations:
Fountains for the customer foyer, greenery in the atrium and entrance, sanitary fittings in various qualities.
2 fat separators

Ventilation and air-conditioning:
12 ventilation control systems with fresh air supply and outgoing air of 6000 m^3/h to 45,000 m^3/h with heat recovery and smoke extraction function
2 fat separators for the gastronomy areas and canteen
Garage CO- and smoke extraction system
Jet ventilators for garage smoke extraction and CO control

High voltage electricity
2 transformers 10 kV/400V each with 1000 kVA for bank operation
1 transformer 10 kV/400V with 1000 kVA for tenants
1 mains back-up system 700 kVA
1 no-break power supply system 240 kVA
2 no-break power supply systems 11 kVA
2 no-break power supply systems 7 kVA
2 no-break power supply systems 5 kVA
4 low-voltage main distribution frames (2 secure supply, 1 normal mains supply, 1 tenants' supply)
Security lighting system 35 kVA wit 15 sub-stations in function preservation
50 differently equipped sub-distribution frames

Telecommunications and IT installations:
DP network FTTO with mini-switches in the offices, designed completely as an optical fibre network
ISDN telephone system with 400 users
Camera system in accordance with accident prevention guidelines
Break-in alarm system partly class “C” with certificate
Fire alarm system with push button alarms, optical smoke alarms, smoke extraction systems
Venting installation
Access monitoring system with time recording
Emergency exit controls
Security cell DP certified in accordance with ECBS (European Certification Board Security Systems), quality class R60D; type-tested in accordance with EN 1047-2

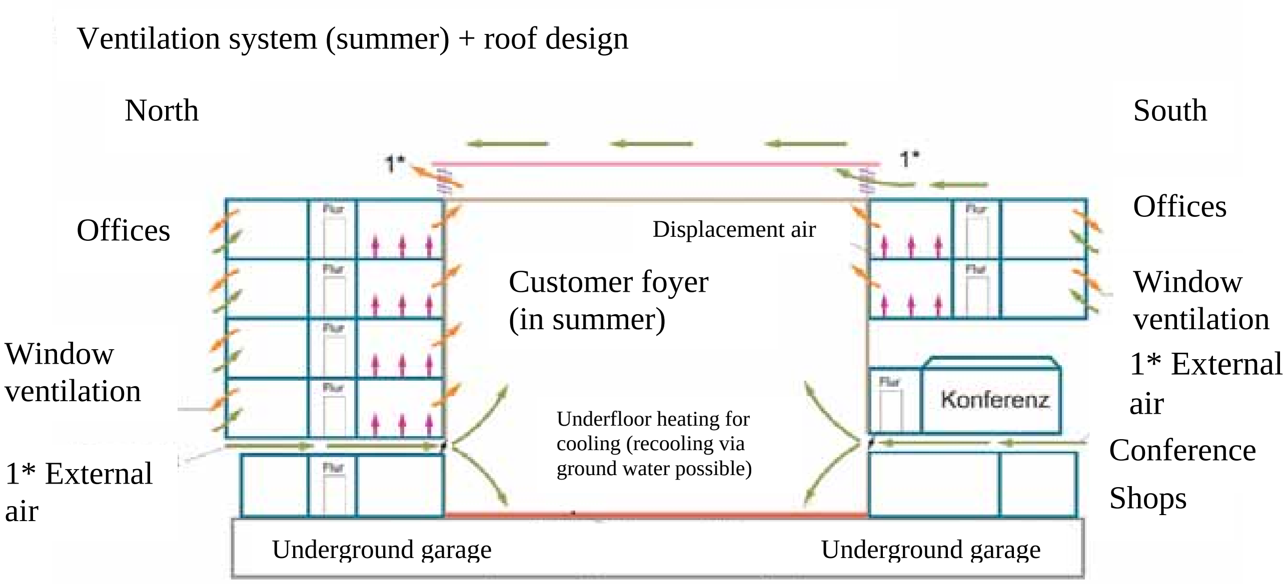
Media technology:
PA systems, including amplifiers, mixing panels, loudspeakers
Stage lighting systems
Plasma screens
Interactive information points (Infopoints)
Screens
Projection systems with uplighting and rear projection systems
Movable ceiling stands
Light rails
Light scanners
Control systems
AV cupboard
Transportable camera system
Training system
Conference system
Touch screen
Monitors

Software

Measuring, control and regulation systems, central building control:
Control system with 2 servers, ODBC interface to other systems
2 operating stations
1 touch screen at atrium reception
Individual room control systems LON
9 information centres
approx 12,000 data points hardware
Weather station
Consumption data recording via LON

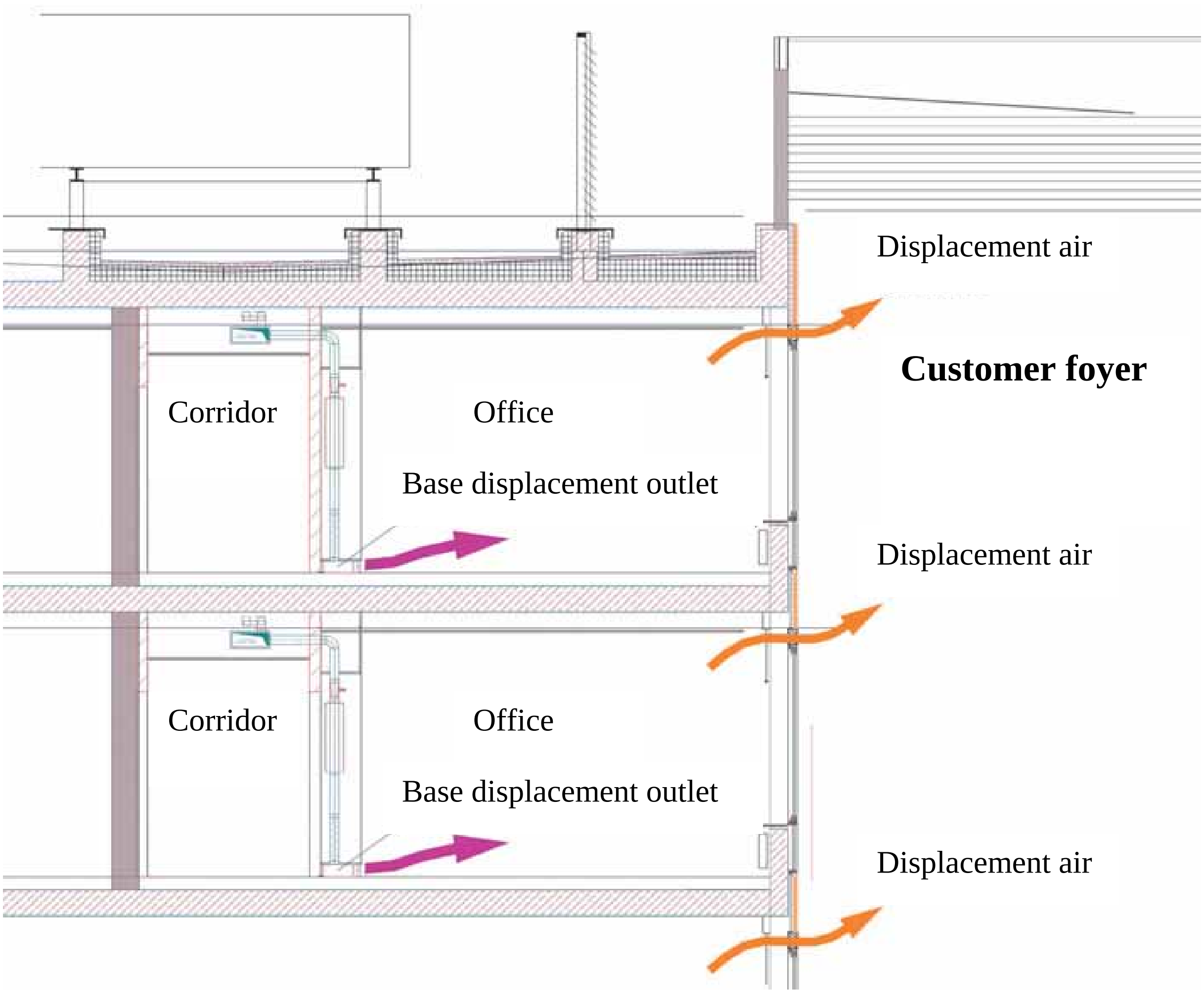
1 New building Sparkasse Hagen

2 Large customer foyer with “functional roof”



1* Shutter control of the external air and outgoing air via the roof façade depending on the temperature, CO2 volume in the customer foyer, wind speed and wind direction

3 Ventilation



4 Ventilation principle for the offices adjoining the foyer

5 Cooling centre

6 Events area with professional stage lighting and PA technology

7 View into the covered atrium to the restaurant “Hagen1“

8 Planted atrium with rest areas

9 Clever architecture creates consultation areas

10 Lift shaft as light source

11 Reception area

12 Roof structure with ventilation panels

13 Fassade of the customer foyer

Microclimate and air quality in capital town of Vojvodina, Serbia through 2002-2005.

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SUMMARY

Novi Sad is capital administrative center of the Province Vojvodina, one of two towns with active oil refinery crossroad of transit traffic artery and until recently – great industrial town! In this paper are presented data of imission for basic pollutant matters in the air of the town, microclimate and health condition of children and adult population according to permanent reports of health services for preschool, schoolchildren and physician in primary health care (general population) for the period 2002-2005. Climate changes (increased temperature, relative humidity change in the air etc.), then endangered urban air quality, have influenced the increase of specific morbidity from respiratory diseases in vulnerable population categories. Results of the paper indicate to monitoring set up towards specific pollutants followed by bio-indicator analysis. In further period, multidisciplinary studies can enable to become a part of some European project of environmental effect analysis to population health.

INTRODUCTION

Climate change is a new and rapidly developing topic of scientific inquiry and health risk assessment [1]. Anthropogenic **climate** change signifies that for the first time the aggregate **global** impact of humankind exceeds the physical and ecological limits of the biosphere [2]. The potential consequences of this and other **global** changes (including stratospheric ozone depletion, loss of biodiversity, worldwide land degradation, and depletion of aquifers) are wide ranging [1,3,4].

The aggregate environmental impact of humanity has begun to change some of the earth's great biophysical systems. Such human-induced systemic environmental change is unprecedented. Some health effect are related to climatic and weather variability, while others are connected to climatic changes [5,6,7]. Changes in frequency of temperature extremes and stagnation periods increase episodes of thermal stress and air pollution.

There has been a clear worldwide warming trend. Global temperatures have increased by amount 0.6 ^{o}C in the last 100 years. In Europe the warming trend is slightly greater, 0.8 ^{o}C [3,7]. Increases in minimum temperatures have been far than changes in maximum temperatures, and the observed temperature rise has been the most marked during the winter period [3,7].

The effects of climate change go beyond the gradual spread of disease [8,9]. Extreme events such as floods, droughts and heat waves are likely to increase under global warming and will challenge our ability to manage health risks and test the resilience of our infrastructures in many areas, including health service delivery. In 2003, Europe experienced summer temperatures that were unprecedented in the instrumental record. In France, over 14 000 more deaths were reported during the August heat wave than were typical for that time of year, and the total for Europe was in the region of 20 000. In Paris, the number of deaths increased by 140% over usual figures. The modest climate change that occurred between the mid-1970s and the year 2000 is estimated to have caused the annual loss of over 150 000 lives and 5 500 000 disability-adjusted life-years" [9,10].

Exposure to ambient air pollution has been linked to a number of different health outcomes, starting from modest transient changes in the respiratory tract and impaired pulmonary function, continuing to restricted activity/reducing performance, emergency room visits and hospital admissions and to mortality [4].

For the first time in history, the economic activity of the human population has become so vast that it is beginning to change the gaseous composition of the lower and middle atmospheres. This is now called human induced global climate change (HIGCC), which in turn will have a significant impact on a future generation of children [11]. Why the children have at high risk? Because: Children breathe more per unit body weight than adults; have smaller airways and lings; has a different rate of toxification and detoxification; time spent outdoors; increased ventilation with play and exercise; high prevalence of asthma and other diseases; high ruts of acute respiratory infections [4].

Which population groups are at high risk, also? Undoubtedly the elderly people, with cardio respiratory disease. When compared with healthy people, those with respiratory disorders (such as asthma or chronic bronchitis) may react more strongly to a given exposure, either as a results of increased responsiveness to a specific dose and/or as a results of a larger internal dose of some pollutants than in normal individuals exposed to the same concentration. Increased particle deposition and retention have been demonstrated in the airways of people suffering from obstructive lung disease [4,11-13].

Ussually, the air pollution problems are associated with oil and heating oil furnaces, industrial facilities with dirty technologies and motor vehiclecs [3,4,14-17].

METHODS

Since 1967 Institute of Public Health, Novi Sad - Department of Hygiene and Environmental protection follows-up everyday presence of basic and some specific pollutant matters in the outdoor air of the city. Air sediments, are continually followed during 30 days/12 months at 20 localities in the city, accordingly to our legal regulations and recommendations [17,18].

Data on the number of diseased from respiratory infections at adults, registered in the General practice of Health Center, preschool and school children at regarding Dispensaries, are acquired from the Department of Social Medicine, Institute of Public Health, Novi Sad. Climate data (2002 -2005) are acquired from the Meteorological Center - Novi Sad. [17,18].

Statistical data processing is performed with SPSS for Windows, graph presentation with Harvard & Graphics program and the whole text - in text processor Microsoft Word for

Windows. In the paper are applied descriptive statistic parameters: mean value, standard deviation (SD), minimum and maximal values, variation coefficient (CV).

RESULTS

Air-sediment is analyzed in the period 1989-2005 at 20 localities in Novi Sad (Figure 1a), and in details for the period 2002-2005. The highest average monthly values are registered in May and June (within recommended values) for each investigation year. Maximal value of total aero-sediment quantity was: in 2002 – 1332.9 mg/m^2/day in May (mean annual value: 158.8; C_{98} 500; SD 135.1; CV 85.09); in 2003 – 874.41 mg/m^2/day in June (mean annual value:174.8; C_{98} 710; SD 147.4; CV 84.34); in 2004 - 1567 mg/m^2/day (mean annual value: 169.5; C_{98} 541.1; SD 154.52; CV 91.15) in June and in 2005 – 778.7 mg/m^2/day in May (mean annual value: 778.7; C_{98} 452.9; SD 110.29; CV 69.61). According to localities, values above recommended values are registered in the parts of the town with the greatest building activity.

The lowest pH is determined in February 2003 (2.6), and then in 2005 (3.0), also determined in the winter period. Survey of the average annual values pH is presented in the Figure 1b. Although our regulations do not have limited concentrations for calcium, sulfate, chloride and nitrate in air-sediment, we still present them in order to have general view of the air quality (Figure 1c). These results are presented to enable the view of “acid rains” in this territory.

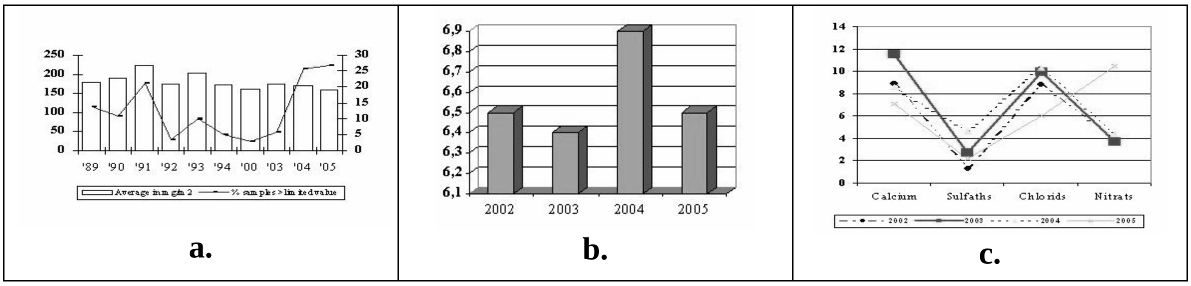
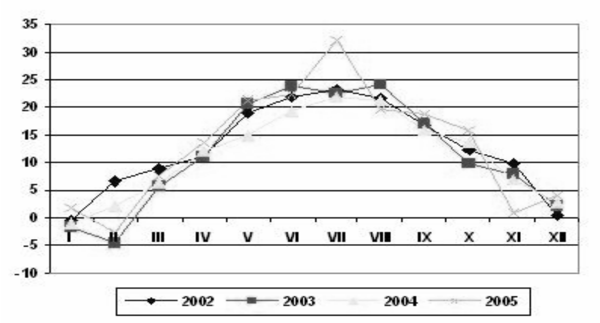


Figure 1. a) Average Concentration of air-sedimentation in the capital town of Vojvodina, Novi Sad, in the period 1989 -2005, b) Outdoor pH in the city of Novi Sad in the period 2002 - 2005, c) Content of calcium, sulphates, chlorides and nitrates in the air-sediments, in the period 2002-2005 (mg/m^2/day).

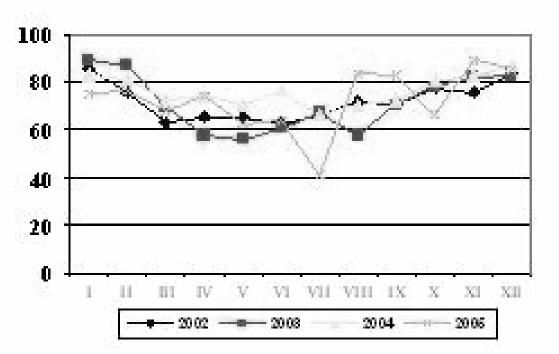
In Figures 2a, 2b, 2c and 2d are presented climate parameters: temperature, relative humidity, atmospheric pressure and air circulation velocity by localities during the period 2002-2005 [17,18]. Volume of temperature peaks ranged in this sequence: in 2002 from – 11.6 to 30.4^{0}C; in 2003 from –15.7 to 31^{0}C: in 2004 from – 8.8 to 28.1^{0}C; in 2005 from –2.0 to 36.2^{0}C. Increase of daily maximal temperatures is evident (from 28.1 to 36.2^{0}C) as well as change in minimal temperatures (from –15.7 to –2.8 ^{0}C). Average air circulation velocity ranged from 2.6 to 2.2 m/s, while minimal air humidity ranged from 5.9 to 7.1 %. Atmospheric pressure was standard for city area in investigated period (from 1006 to 1008 mbar).

Specific morbidity from respiratory diseases at pre-school and schoolchildren and adult population in Novi Sad is presented in Figures 3a, 3b and 3c. The following registered diseases are analyzed (2001-2005): Pneumonia (N^o 169; numbers* from: Int. diseases clasif.-10, Federal Institute of Public Health, Savremena administracija, Belgrade; 1996), *Bronchitis acuta et bronchiolitis acuta* (N^0 170*), other diseases in upper part of breathing system (N^{0}174*), *Tracheitis ac., Emphysema* and other obstructive lung diseases (N^{0}175*) as well as

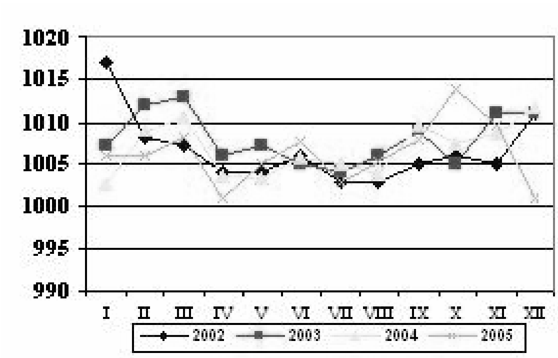
Asthma bronchiale ($N^0$176*), which is base on medical documentation (routine evidence and reports of health services for health care of preschool and school children as well as General medicine service). There is evident increasing number of the following diseases: *Bronchitis ac. et bronchiolitis ac.* (N^0 170*); *Tracheitis ac., Emphysema* and other obstructive lung diseases ($N^0$175*) during analyzed period at all three population categories, which can be in correlation to urban air quality and climate changes.



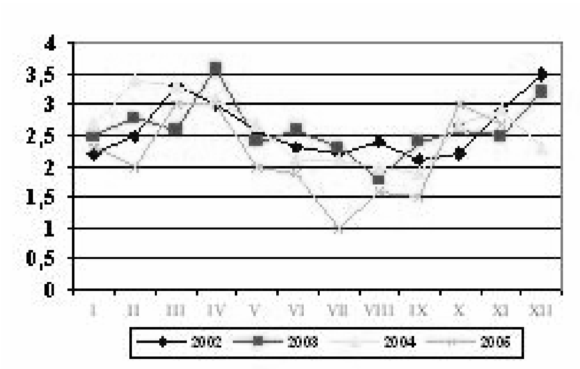
a.



b.

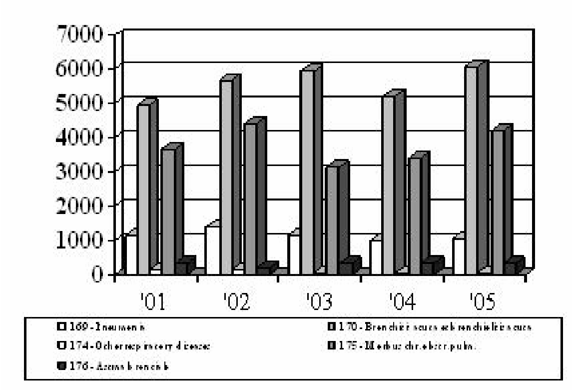


c.

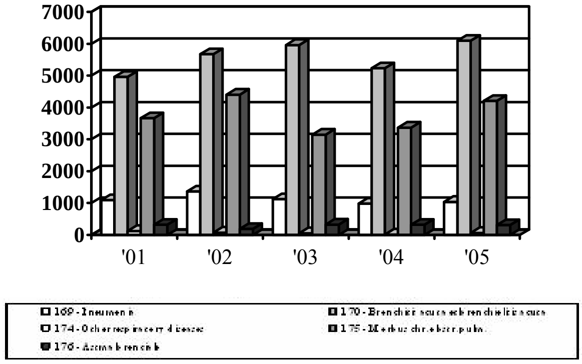


d.

Figure 2.a) Air temperature (^{0}C), b) Relative air humidity (%), c) Atmospheric pressure (mbar), d) Air circulation velocity (m/s).



a.



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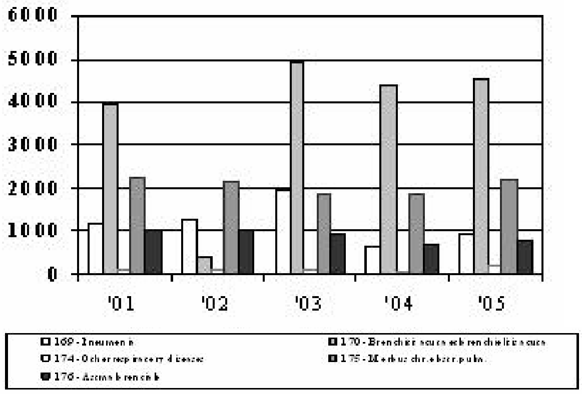

c.

Figure 3. a) Specific morbidity from respiratory diseases at preschool children in Novi Sad (aged 0-7), b) Specific morbidity from respiratory diseases at school children in Novi Sad (aged 7-18), c) Specific morbidity from respiratory diseases at adults in Novi Sad (general practice)

DISCUSSION

Air pollution in urban areas is a major concern for environmental health in Europe, especially the effects of particulates [19-22]. The dissemination and concentration of air pollutants (both particles and gases) depend on the prevailing weather conditions air current, temperature variation, humidity and precipitation. Large, slowly moving anticyclones may cover an area for several days, are a week or more, and give rise to conditions that readily allow pollutants to accumulate. Predicting the impact of climate change an average local air pollution concentration is therefore very difficult.

Exposure to thermal extremes cause direct: altered rates of heat and cold - related illness and death; as An altered frequency and/or intensity of other extreme weather events cause direct: deaths, injuries, mental disorders, damage to public health infrastructure [12].

Several recent studies have observed that elderly people are especially vulnerable to heat-related illness [21]. By the year, 2025 the population of adults aged ≥ 60 years will increase by 37 million (50%) [22]. The population vulnerable to thermal stress will therefore increase, and climate change will represent an additional burden on this population. Meta-analyses by Martens [23,24] estimated that an increase in global temperature of 1^{o}C could reduce winter cardiovascular mortality in Europe.

Climate change effects on population health will reflect the conditions of the ecological and social environments in which humans live. Projections on future climate change in Europe are derivate from global climate model experiments [24,25]. An overall increase in average annual temperatures is projected, and thus increase is likely to be greater in high latitudes than in mid-latitude Europe. Summer temperatures will increase more than winter temperatures. In general, continental regions may become dryer while maritime regions may become wetter.

Air pollution (AP) is associated with increased upper and lower respiratory symptoms in children; may increase bronchitis and cough and aggravate asthma symptoms; it is uncertain whether current levels of ambient AP (traffic - generated air pollution) contribute to cancer development in children. Does air pollution influence the development of the lung? Studies of lung in children suggest that: living in areas of high air pollution is associated with lower lung function; long-term air pollution is associated with lower rates of lung function development.

Particulate matter and acid aerosols seem to be the main agents of acute effects all-cause mortality in European cities [26]. In most epidemiological studies on air pollution effects, temperature is treated as a confounder. There is some evidence of a physiological synergistic effect between high temperatures and pollutants [2,4,7, 9-13].

The regional "negative" effect of sulfate aerosols in central Europe could offset some of the time-dependent 'positive' effect of carbon dioxide on climate warming [26,27]. The local effects of aerosols on precipitation patterns are also highly uncertain [27-29].

Our previous investigations indicated to correlation of respiratory and malignant diseases not only in Novi Sad but also in the widen area of AP Vojvodina and the country as well. Influence of basic pollutant matters originating from the communal air ambient (sooth, SO_2, NOx) and correlation with some microclimate parameters (temperature, velocity and relative air humidity) gave positive association [14,16,17,27-29].

ACKNOWLEDGEMENT

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Statistic Selection of Coincident Solar Irradiance, Dry-bulb and Wet-bulb Temperatures for Determining Design Cooling Loads

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SUMMARY

Near-extreme solar irradiance, ambient dry-bulb and wet-bulb temperatures are fundamental data for determining the peak building cooling load. Design solar irradiance has been separately and independently selected by both ASHRAE and CIBSE. This may result in over-estimated cooling loads, and in turn over-sized air-conditioning systems. Hence, a statistic method based on probability theory and heat transfer principles was developed for rational selection of the three coincident design weather data used for calculating peak cooling loads in a building with thermal lag less than one hour. The new method was applied to historic weather records of 25 years in Hong Kong to rationally generate design weather data. These data were compared with those produced by the traditional method. Results show that traditional design solar irradiance, dry-bulb and wet-bulb temperatures may be significantly overestimated when a room or building faces east, south and west. Generating sequences of three coincident weather variables for heavy buildings is also discussed.

INTRODUCTION

Near-extreme weather conditions, including dry-bulb temperature, moisture content and solar irradiation, are essential and fundamental data for the design of building air-conditioning systems [1,2]. They simultaneously act on a building, and are driving potentials of heat transfer though building envelope and direct mass exchange by infiltration and ventilation. Near-extreme coincident design weather conditions are required in determining the peak cooling load for sizing air-conditioning systems [1,2]. Improper design weather conditions may lead to oversized or undersized HVAC system, which will result in unnecessary extra capital cost and low part-load efficiency or frequent failures in providing sufficient cooling. Even a small difference in the design temperature and solar irradiance may have significant economic implications. Hence, coincident solar irradiation, dry-bulb and wet-bulb temperatures should be properly selected.

Engineering design involves a trade-off between maximizing system reliability and minimizing cost [3]. The probability of failure in meeting the required load is expressed as the ratio of the number of hours that the capacity would fail to meet the load to the total operating hours. This probability is call as risk factor here, which is the complement to the system reliability, and equals the difference of one subtracted by the system reliability. It is highly desirable that the design method and the associated design data can help optimize this trade-off. Several professional institutes have revised their design guidelines to incorporate probabilistic design concepts or reliability into design. For instance, the American institute of

Steel Construction Load and Resistance Factor Design specifications and the European and Canadian structural design specifications have been revised [3].

Many researchers have made valuable contributions to the development of weather design conditions in the past decades. ASHRAE has launched a number of research projects, such as ASHRAE RP-754 [4], RP-828 [5], RP-890 [6], and RP-1171 [7]. These ASHRAE research project were aimed at providing, improving and rationalizing the fundamental weather data for HVAC system design.

Thom [8] developed a statistical method for proper selection of outdoor design dry bulb temperatures in winter for heating system sizing. The method took into account the probability that the system could fail to meet the actual load. He found five design dry bulb temperatures with different cumulative probabilities. HVAC engineers may select reasonable design temperatures, based on their experience, for various types of buildings to properly size the system to match the capacity reliability required. Holladay [9] analyzed design weather data from three sources: Summer Weather Data 2.5 percent, the Guide 2.5 percent and Common Use. He found that large differences existed among these data for some locations. He developed a simple weighted average equation for determining the design dry bulb temperature.

Colliver et al [4] statistically determined near-extreme dew-point temperatures and mean coincident dry-bulb temperatures using a 30-year set of hourly weather data from 239 US and 143 Canadian locations. They found that humidity ratios computed from the design extreme dew-point temperature and mean coincident dry bulb temperature is greater, on average, than those from the design extreme dry bulb temperature and mean coincident wet bulb temperature. Colliver et al. [5] further developed criteria and procedures for statistical analysis of weather data and generated sequences of near-extreme weather records for four design parameters. The four parameters are high and low dry-bulb temperature, high dew-point temperature, high enthalpy level, and low wet-bulb depression. They determined four different extreme sequences for each parameter for the 0.4%, 1.0% and 2.0% annual frequencies of occurrence. Colliver et al [6] also analyzed historical weather data to produce annual frequency-of-occurrence design dry-bulb, wet-bulb, and dew-point temperatures with mean coincident values at the design conditions. Their analysis showed that the design dew-point directly derived from coincident dry-bulb and dew-point temperatures had a much higher humidity than that generated from coincident dry-bulb and wet-bulb design conditions.

Mason and Kingston [10] indicated that the use of separately selected coincident dry-bulb and wet-bulb design temperatures would lead to computed cooling loads much higher than actual loads. ASHRAE [1] and CIBSE [2] presented two forms of coincident design dry-bulb and wet-bulb temperatures; one is the dry-bulb temperature at a certain cumulative frequency of occurrence associated with the mean coincident wet-bulb temperature; and the other is the wet-bulb temperature at a certain cumulative frequency of occurrence associated with the mean coincident dry bulb temperature. These coincident design weather data are well applicable to systems the thermal time constant of which is less than one hour [5].

Chen et al [11] pointed out that the dry-bulb temperature at a certain percentile of occurrence together with the mean coincident wet-bulb temperature may not indicate the frequency of occurrence of the cooling load. This is because the defined percentile may not equal the real probability of occurrence of coincident dry-bulb and wet-bulb temperatures. Hence, they developed a statistical method for rational selection of coincident design dry-bulb and wet-bulb temperatures. These design weather conditions allow engineers to do risk-based air-conditioning design when the effect of solar irradiation on the peak cooling load is negligible. Their study shows that the rationally derived coincident dry-bulb and wet-bulb temperatures could be largely different from the traditional ones.

Generally, solar irradiation should have a significant contribution to building cooling loads, especially for passive or active solar cooling buildings [12]. From the above literature review, however, it can be seen that less attentions have been paid to the design solar irradiance. The current design solar irradiance data do not simultaneously occur with the coincident dry-bulb and wet-bulb temperatures for air-conditioning system design. Therefore, this study is first aimed at developing a method for rational selection of coincident solar irradiation, dry-bulb and wet-bulb temperatures. The second objective is to apply this novel method to historical hourly weather data observed in Hong Kong to generate the coincident design weather conditions. These selected design weather data can be used to design an air conditioning system that can match system reliability desired for a building with thermal time constant less than one hour.

THEORETICAL METHOD

For a building or room whose thermal lag is less than 1 hour, hourly cooling load $Q(k)$ may be expressed by [1]

$$Q(k) = \sum_{j=1}^{n} (UA)_j (t_e(k) - t_{rc}) + m_o c_{pa} (t_{db}(k) - t_{rc}) + m_o h_l (W_o(k) - W_{rc})$$
$$+ A_{wd} (IAC) [SHGC(\theta(k)) E_D(k) + < SHGC >_D \; E_{dt}(k)] \qquad (1)$$

where n is the number of external envelopes; k is discrete time, hr; A_{wd} is window area, m^2; A is the area of external envelope components of a building or room, m^2; h_l is air latent heat, 2430×10^3 J/kg; c_{pa} is the sensible specific heat capacity of air, 1010 J/(kg °C); E_D is direct irradiance, W/m^2. E_d is diffuse sky irradiance, W/m^2; E_{dt} is diffuse irradiance, $E_{dt} = E_d + E_r$, where E_r is ground-reflected irradiance， W/m^2; E_t is total solar irradiation incident, W/m^2; *IAC* is inside shading attenuation coefficient; *SHGC* is direct solar heat gain coefficient as a function of incident angle θ; $<SHGC>_D$ is diffuse solar heat gain coefficient; t_{rc} is presumed constant room air temperature, °C; U is the overall heat transfer coefficient of walls, roofs or windows W/(m^2·K); m_o is outdoor air mass flow rate introduced into room, kg/s; W_o is outdoor air humidity ratio, kg/kg; W_{rc} is presumed constant room air humidity ratio, kg/kg; θ is incident angle, °; n is the total number of external envelope components of a building or room; t_e is sol-air temperature, °C and its calculation may be referred elsewhere [1]. In the small range of near extreme weather conditions, outside air humidity ratio may be approximately expressed by $W_o(k) = w_1\, t_{wb}(k) + w_0$, where w_o and w_1 are constants; t_{wb} is outdoor air wet-bulb temperature, °C.

The number of building characteristic parameters in Equation (1) may be reduced by the following methods. Figure 1 shows how the diffuse solar gain coefficient $<SHGC>_D$ varies with *SHGC*(0) in the normal direction for all glazing and window systems listed in ASHRAE Handbook [1]. It can be seen from Figure 1 that $<SHGC>_D$ is approximately equal to *SHGC*(0) since their relation can be expressed by

$$< SHGC >_D = SHGC(0) - 0.063 \approx SHGC(0) \qquad (2)$$

SHGC ratio is defined by

$$r_{SHGC}(\theta) = \frac{SHGC(\theta)}{SHGC(0)} \qquad (3)$$

which is the ratio of *SHGC* at any incident angle to *SHGC*(0) at the normal angle. Analysis of all the glazing and window systems listed in ASHRAE Handbook [1] shows that *SHGC* ratios vary similarly at most of the incident angles with the upper and lower limits. Therefore, all the glazing or window systems listed in ASHRAE Handbook [1] are classified into three classes: I, II and III based on the ratio of $SHGC(\theta)$ to *SHGC*(0). Table 1 shows the average SHGC ratio values at different solar incident angles.

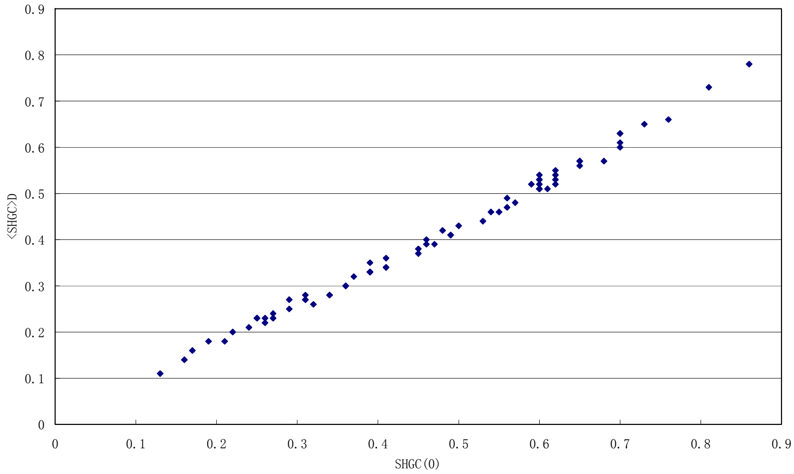

Figure 1 Relation between $< SHGC >_D$ and $SHGC(0)$ of glazing

Table 1 SHGC ratio values at different solar incident angles

Glazing classes	$r_{SHGC}(\theta)$					
	Normal	40°	50°	60°	70°	80°
I	1.0	0.980	0.940	0.920	0.80	0.50
II	1.0	0.955	0.900	0.855	0.70	0.40
III	1.0	0.920	0.860	0.785	0.60	0.30

Application of the above simplifications to Equation (1) yields

$$Q(k) = [\sum_{j=1}^{n}(UA)_j + m_o c_{pa}]t_{db}(k) + m_o h_l w_1 t_{wb}(k) + \frac{\alpha}{h_o}\sum_{j=1}^{n_w}(UA)_j E_t(k)$$
$$+ A_{wd}\,(IAC)\,SHGC(0)\,[r_{SHGC}(\theta(k))E_D(k) + E_{dt}(k)] + C_o \qquad (4)$$

where

$$C_o = m_o h_l w_o - \left[\sum_{j=1}^{n}(UA)_j + m_o c_{pa}\right] t_{rc} - m_o h_l W_{rc} - \sum_{j=1}^{n_{rf}}(UA)_j \frac{\varepsilon \Delta R}{h_o} \qquad (5)$$

where n_w is the number of building opaque envelopes and n_{rf} is the number of roof types; t_{db} is outdoor air dry-bulb temperature, °C; h_o is heat transfer coefficient by long-wave irradiation and convection at outer surface, which is equal to 17 W/(m^2 °C); ΔR is difference between long-wave irradiation incident on surface from sky and surroundings and irradiation emitted by blackbody at outdoor air temperature, W/m^2; $\varepsilon\Delta R/h_o$ = 4 for horizontal surface, =0 for vertical surface; α and ε are the absorptivity and emissivity of the exterior surface of building opaque envelope, respectively.

C_o in Equation (5) is constant when building design and required indoor air conditions are known. It does not impact the selection of design weather data, and hence will not be considered in the statistical analysis of weather data. Only those terms associated with solar irradiance, dry-bulb or wet-bulb temperature will be analyzed. They can be combined into a single index, which may be called as equivalent weather temperature. Neglecting the constant term and rearranging Equation 6 yields

$$T_e(k) = t_{db}(k) + at_{wb}(k) + bE_t(k) + c[r_{SHGC}(\theta(k))E_D(k) + E_{dt}(k)] \qquad (6)$$

with

$$a = \frac{m_o h_l w_1}{\sum_{j=1}^{n}(UA)_j + m_o c_{pa}} \quad , \quad b = \frac{\alpha}{h_o}\frac{\sum_{j=1}^{n_w}(UA)_j}{\sum_{j=1}^{n}(UA)_j + m_o c_{pa}} \quad , \quad c = \frac{A_{wd}(IAC)\,SHGC(0)}{\sum_{j=1}^{n}(UA)_j + m_o c_{pa}} \qquad (7)$$

Coefficients *a*, *b* and *c* in Equation (6) are constant for a given building or room. It can be easily observed from Equations (7) that these coefficients only depend on building design parameters and some constants. Therefore, they reflect the thermal characteristics of buildings, and describe the effect of wet-bulb temperature and global solar irradiance incident on opaque and transparent envelope on cooling loads, respectively. In addition to the three weather variables, the *SHGC* ratio and $r_{SHGC}(\theta(k))$ in Equation (6) also varies with time. Nevertheless, the annual distributions of the *SHGC* ratio are fixed for a building or room with one external envelope in a given orientation. Thus, the selection of coincident weather conditions should only depend on the building characteristic parameters, *a*, *b* and *c* as well as glazing types, and the required capacity reliability for a building or room with one external envelope in a given orientation.

NEW DESIGN WEATHER CONDITIONS

Air-conditioning system design should aim to optimize system reliability and cost. Coincident design weather data to be selected should allow engineers to design an air-conditioning system that can satisfy the target system reliability or risk. Examination of Equations (4) to (6) indicates that the variation of cooling loads is proportional to the equivalent weather temperature. This means that the percentile of coincident solar irradiance, and dry-bulb and wet-bulb temperatures should be equal to the capacity risk of air-conditioning systems designed with the corresponding weather data.

The Hong Kong Observatory observes dry-bulb and web-bulb temperatures, horizontal global solar irradiance as well as other weather parameters hourly at latitude 22º18 ʹN and longitude 114º10 ʹN [13]. The hourly weather records of 25 years from January 1979 to December 2003 were used for statistic analysis of the peak cooling loads of different buildings. The hourly horizontal global irradiance was first decomposed into direct and diffuse irradiance using the formula given by Lam and Li [14]. The horizontal irradiance can then be converted to the vertical surfaces in eight orientations: north (N), northeast (NE), east (E), southeast (SE), south (S), southwest (SW), west (W) and northwest (NW), using the method given in ASHARE Handbook [1].

Building characteristics largely impacts the selection of design weather data. It can be fully described by the three building characteristic parameters *a*, *b* and *c*, as well as glazing type. In addition to these three parameters, three classes of glazing are also taken into account. However, no windows were considered on the roof at this stage.

Table 2 shows a sample of coincident design weather data for buildings with the same building characteristic parameters *a* (0.60), *b* (0.03) and *c* (1.00), and glazing class III for the percentile or risk factor of 1.0. The combination of different coincident weather conditions may result in the same cooling loads [11]. In the other words, there are many equal cooling loads occurring at different weather conditions, month, day and time. The design conditions given in Table 2 represent the statistic center of them, which means most of the equal cooling loads occur in or around the design conditions shown in Table 3. Note that the 21st of each month can be used as the design day on which the peak cooling load is calculated, following the ASHRAE method [1].

Table 2 Coincident design weather data derived by the new method for Hong Kong

	Glazing class = III, *a* = 0.60, *b* = 0.03, *c* = 1.0, Risk factor = 1.0%								
Orientations	N	NE	E	SE	S	SW	W	NW	H
Equivalent temperature (°C)	215	324	444	531	589	675	667	501	72
Dry bulb temperature (°C)	29.9	29.4	25.3	20.5	20.6	22.5	28.9	29.8	31.3
Wet bulb temperature (°C)	25.9	25.8	21.3	15.5	15.1	17.4	24.3	25.4	26.4
Beam irradiance (W/m^2)	28	158	244	322	388	466	457	327	828*
Diffusion irradiance (W/m^2)	155	166	185	203	205	194	181	160	
Month	6	7	10	12	12	11	7	5	7
Hour	15	11	10	11	12	15	16	16	13

Table 3 shows coincident horizontal design solar irradiance, and dry-bulb and wet-bulb temperatures generated by the traditional and new methods for the percentile or risk factor of 0.4 and 2.0. The building characteristic parameters *a*, *b* and *c* that were used for generating Table 3 are respectively equal to 1.0, 0.03 and 1.0, and window glazing was class III. It can be seen that the rational design solar irradiance decreases with the increase of risk factor on the horizontal surface. The design solar irradiance is kept constant for all the risk factors because the traditional calculation of design solar irradiance does not depend on both the risk factor and the other weather parameters. Difference between the traditional and rational design solar irradiances is negligible at the percentile of 0.4, but about 19.5% at the percentile of 2. Difference between the traditional and rational design dry-bulb temperature is about 1.5 °C and the wet-bulb temperature 0.6 to 0.8 ºC.

Table 3 Comparison of traditional and new design weather data on horizontal surface

Risk factor	0.4%			2.0%		
Glazing class III *a*=1.0, *b*=0.03, *c*=1.0	Dry temp	Wet temp	Solar irradiance	Dry temp.	Wet temp.	Solar irradiance
Traditional	33.0	27.1	907	32.2	26.8	907
Rational	31.5	26.5	895	30.6	26.0	759

Table 4 shows the coincident design dry-bulb and wet-bulb temperatures generated by the two methods for a building with a = 0.60, b = 0.03, c = 1.00, glazing class III and risk factor = 1.0%. These design weather data are used for calculating the peak cooling load through vertical building envelopes in different orientations. The 'Month' shown in the table means that in the indicated month, the peak cooling load occurs for that vertical envelope in the given orientation. Careful examination of Table 4 shows that all the traditional design dry-bulb and wet-bulb temperatures are overestimated as compared to the rationally derived design temperatures. Difference between the dry-bulb and wet-bulb design temperatures generated by the two methods is generally very large for the vertical envelope in most of the orientations, especially in the east, southeast, south, southwest and west. The traditional design dry-bulb and wet-bulb temperatures could be more than 6 ºC higher than the two rationally derived design weather temperatures.

Table 4 Comparisons of traditional and new design dry-bulb and wet-bulb temperatures

Orient.	Month for Peak Load	Traditional		Rational	
		Dry temp.	Wet temp.	Dry temp.	Wet temp.
N	6	32.1	26.7	29.9	25.9
NE	7	32.6	26.9	29.4	25.8
E	10	29.9	24.8	25.3	21.3
SE	12	24.0	20.4	20.5	15.5
S	12	24.0	20.4	20.6	15.1
SW	11	27.0	22.1	22.5	17.4
W	7	32.6	26.9	28.9	24.3
NW	5	30.8	26.0	29.8	25.4

Note: a = 0.60, b = 0.03, c = 1.00, glazing class III and risk factor = 1.0%

DISCUSSION

New near-extreme coincident design solar irradiance, dry-bulb and wet-bulb temperatures have been generated using the novel method given in the second section. It can be seen from the model described by Equations (1) to (7) that the annual percentile of coincident design weather data is now statistically equal to the capacity risk factor of air-conditioning systems designed with the corresponding design weather conditions. This allows engineers to design air-conditioning systems that match the target system reliability probably required by the client. In the other words, the use of these outdoor design weather data will result in consistent sizing air-conditioning systems in different conditions.

Comparison between the traditional and rational design weather data shows that all the traditional data are higher than the rational ones. These differences are significant in many cases, especially when a room or building faces east, southeast, south, southwest and west. The current design solar irradiance could be more than 19% higher than the rational value. The current design coincident dry-bulb and wet-bulb temperatures could be more than 6 ºC higher as compared to the newly generated design dry-bulb and wet-bulb temperatures. This will significantly impact the peak latent as well as sensible cooling load.

The new method can also tell us when the peak cooling load occurs, and hence HVAC engineers can avoid calculating 24 hour cooling loads on one design day in each month of the year. This significantly simplifies the design cooling load calculation.

Although the new design weather data can be used only for direct determining the peak cooling load in a building with a thermal lag less than one hour, the new method provides a basis for the further generating coincident design weather data for any buildings.

Based on the principle of the new rational method, all the independent heat sources may be described with the discrete Fourier series [15]. Cooling load, Q_{cl}, due to these sources can be computed by

$$Q_{cl} = \sum_{i=1}^{4} \sum_{n=-N}^{N} H_i(j\omega_n) u_{n,i}(j\omega_n) \exp(j\omega_n t) \qquad (8)$$

where H_i is Fourier transfer function with respect to any heat source, $u(j\omega_n)$ is the coefficient for each harmonic (ω_n), t is time, n is the harmonic number, and $j = \sqrt{(-1)}$. The amplitude and phase angle of the complex transfer function represent the heat transfer level and thermal lag of building envelopes. Like building characteristic parameters a, b and c in Equation (7), the amplitude and phase angle of transfer functions fully represent the physical characteristics of medium and heavy buildings. Therefore, the selection of sequences of three coincident design weather variables fully depends on these parameters. Similar to Table 2, sequences of the design weather data can be uniquely determined by glazing type, building characteristic parameters, and desired risk factor.

ACKNOWLEDGEMENT

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Study on Optimizing and Designing of Passive Solar Building in Sitsang of china

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SUMMARY

Sitsang belongs to cold Region in china and heating energy consumption in winter is very large. There is fairly scarce petrochemical energy, and that the oxygen content in plateau section is lower, which makes combustion become insufficient, so the cost of conventional energy is considerably high. However, compared with other regions in china, the best advantage of Sitsang is that it possesses abundant solar energy resources. According to climate conditions and energy resources in Sitsang, the paper studied passive solar houses heating system using EnergyPlus software, which is capable of rapid and detailed analysis of energy consumption in buildings, and analyzed carefully such influencing factors as architecture plan layout, the thermal resistance of envelope, the window shading coefficient, area ratio of windows and walls and indoor heat mass. Sufficiently considering the heat gain, heat loss and heat storage, the paper presented the optimum design proposal. The analysis on annual energy consumption and indoor thermal environments is performed. The results showed that when the structure was composed of well thermal insulation materials the solar energy can completely satisfy the heating demand in winter in Sitsang.
Key words: solar building; passive solar heating; sustainable energy

INTRODUCTION

Current situation of energy sources in Sitang

In Sitang, petrifaction energy used in building heating are mostly transported from other areas, so the cost of energy sources is very high. The local resident of rural heat mainly though sapless grass and dried cow-dung and the local resident of urban heat though different ways on the basis of the local circumstance. For example, the north of Sitang(Nangqen) is close to Qinghai and the cost of transporting coal is lower, so the boiler of burning coal for heating is adopted in Nangqen. In other areas, since environmental protect and the higher cost of transporting coal, the boiler of burning coal is restricted. The Sitang area processes pipe laying, petroleum is used for heating. However, the lower content of oxygen in plateau area leads to combustion insufficient, which reduces the efficiency of boiler. Decreasing consumption of conventional petrifaction energy sources not only saves hard-earned resource but also helps to protect environment. The environment resource of Sitsang is rare and weak, so the traditional mode of treatment after pollution is absolutely unallowed.

Main Characteristics of Climate in Sitsang

The cities of Lhasa、shiagatse、Nangqen belong to cold or freezing regions in china according to the standard of climate regionalization in architecture (GB50178-93),. The outdoor temperature of the winter in Sitsang is not tremendous different with other cold or freezing regions regions. The characteristic of climate is that the relative humidity is greatly low. The outdoor design relative humidity is respectively 28% and 27% in Lhasa and shiagatse. Because Sitsang is in low latitude and high altitude region, and atmosphere is rarefied, the content of dust and water is low, so the transparence of atmosphere is fairly high. In most areas of Sitsang, the total yearly solar radiation is 6000～8400MJ/m2. The total yearly solar radiation of Lhasa is 8170MJ/m2, and that of Chengdu and Shanghai, which is same latitudinal, are respectively 3687 MJ/m2 and 4375 MJ/m2. The solar radiation of Lhasa is greatly intense. According to criterion of SDM and SMM, Sitsang belongs to the optimal climate region of solar energy heating.

Traditional buildings in Tibet plateau

It can be seen from the traditional building layouts in Tibet plateau that all the designers take the use of solar energy into consideration. The main used rooms of these buildings such as offices, guest rooms are disposed in the south, while the secondary rooms such as storerooms, public toilets are placed in the north. Some buildings just possess rooms towards the south whereas gallery exists in the north, which plays the role of buffer action in heat transfer. The area ratio of window to wall of the external wall of room towards the south is usually greater than 0.4. Solar radiation coming into rooms through window makes these rooms be worthy of the name of greenhouses (direct heat gain passive solar building). Taking good advantage of solar energy is further considered in some buildings. For instance, all the south chambers of a certain hospital are hybrid passive solar house, i.e., the upper glazing of the south external wall directly receives solar radiation heat and the outside surface of the lower solid wall is napped, blacked and closed by a single clear glass, which is treated as solar heat gathering board. The air gap, between the wall and glass, is remained. In addition, the upper and the lower of the wall has ventilating orifices which can be switched off, such that the indoor air flow is natural convection in the air gap and thus absorb solar radiation heat. According to the introduction of the owners, the indoor temperature can be improved in the range of three to five degree Celsius.

Although the designers take the use of solar energy into consideration in the region, the indoor thermal environment doesn't meet the requirements. This is attributed to the poor thermal performance of the building envelope in the region. At present, wall materials of buildings in the region are mostly concrete blocks and rock materials, generally the concrete brick building blocks with a thickness of 200 millimeters produced at local are adopted, whose heat preservation is just better than clay bricks with a thickness of 120 millimeters. Its corresponding heat transfer coefficients greatly exceed the limit thermal conduction value required by civilian building energy economy design standard (the portion of heating supply dwelling buildings). Windows of the buildings mostly use metallic frame single glazing, such as new aluminum alloy frame in recent years and earlier steel frame for buildings.

From the above mentioned situation, we can conclude that the heating supply design in Tibet plateau cannot be completely processed according to conventional heating system in cold region. The design idea also cannot be focused only on heating system design. Three following problems must be solved one by one:With begin to, improving the heat preservation performance of the buildings and reducing heat dissipation of the outside enclosure structures; Then, make good use of the local resource advantages and solar energy (direct and indirect);Finally, optimizing heating design schemes and choosing the most optimal heating system.Only so completely considering heating problem in given environment at plateau can meet the demands of indoor heat comfort environment for occupants and at the same time reduce the conventional petrifaction energy consumption.

This paper mainly considers the using potential of passive solar buildings in the region. The research on direct heat gain passive solar building is conducted through using building energy consumption simulation software EnergyPlus, analyzing the difference of indoor thermal environment of direct heat gain passive building in the region at various external wall constructures.

SELECTION OF THE DYNAMIC ENERHY SIMULATION PROGRAM

The building dynamic energy simulation can compute buildings energy consumption established on mathematical model. These programs can compute hour-by-hour based on 8760 hourly weather data and simulate building indoor thermal environment and energy consumption. In last twenty years, the software of the building dynamic energy simulation has developed father fast. There are about 100 kinds these softwares, in which DoE-2 and EnergyPlus are applied usually. Because doe-2 mainly is used to calculate the heat extraction (heat-addition) rate and air temperature of a room in given conditions, Doe-2 solves these problems by a two-step process. In the first step, the air temperature is assumed to be fixed at some reference value. Instantaneous heat gains (or losses) for the room are calculated based of this constant temperature. Various types of heat gains, such as solar radiation entering through windows, energy form lights, people or equipment, and conduction of energy through the walls, are considered. The cooling (heating) load for a room is calculated for each type of instantaneous heat gain. At the end of the first step, the cooling loads from the various heat gains are summed to provide a total cooling load for the room. In the second step, the total cooling load for a room, along with data about the HVAC system attached to the room and a set of air-temperature weighting factors, is used to calculate the actual heat-extraction rate and air temperature.

In DOE-2, there are two general assumptions made with all weighting factors. The first is that the process modeled can be represented using linear differential equations. Therefore, non-linear processes such as natural convection and radiation must be approximated linearly. The second general assumption is that the weighting factors are constant (that is, they are not functions of time or temperature). This requires system properties, such as film coefficients and the distribution of incident radiation on surfaces, be represented by average values over the time of interest. Both assumptions limit the application of the Doe-2 program. In most case, the DoE-2 is still a valid tool and will continue to be applied in various environments. However, the user should be caution, when used for passive solar buildings or other buildings with heavy construction and for buildings where solar energy provides a large part of the load.

In order to overcome the obvious shortcomings of DOE-2, the EnergyPlus was developed, which had its roots in both the DOE-2 and BLAST. Many of the simulation characteristics have been inherited from the legacy programs. There are also some new features of the EnergyPlus, such as integrated simultaneous solution, Sub-hourly time steps, heat balance based solution, etc. The new features of integrated simultaneous solution and heat balance based solution make the Energy plus can be used to analyze the passive solar building, so the EnergyPlus is selected to use.

Model and Parameter

Simulation Model

Passive solar building is one of the easiest and the most effective modes of heating supply in winter, especially in Tibet plateau where solar energy is abundant. Currently there are mainly three using modes of passive solar building, such as direct heat gain, Trombe Wall, additional solar chamber. Among these three class passive solar buildings, both additional solar chamber and Trombe Wall need to add components at the south of the buildings, which result in difficulty at construction phase and high cost. While direct heat gain mode only considers window area of the south of the building and sunshade in summer, which has less matter at construction phase and a lower cost. Consequently, this paper is mainly on the research of direct heat gain passive building. The Building studied in this paper is showed in the Figure 1, which is the typical layout of the building in the Sitang. The middle standard room is chose in the research.

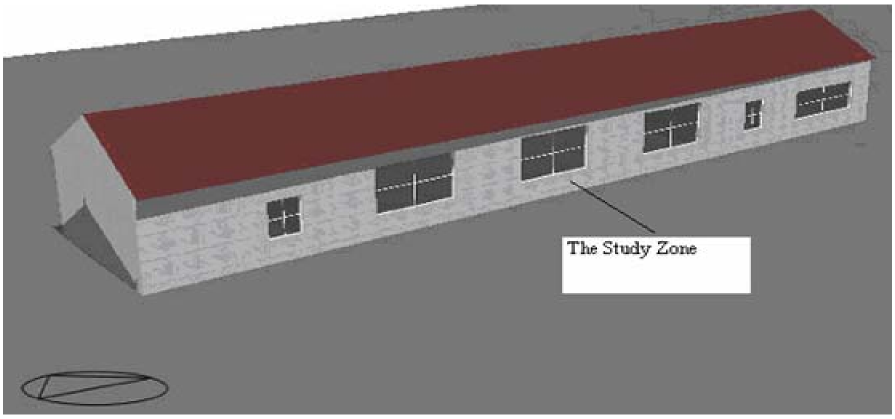

Fig. 1 Simulation model of building

Thermal Design Parameter

In order to compare the indoor thermal environment of the traditional resident building with the new passive building, two different work conditions are simulated. The one is the traditional resident building; the other is new direct gain passive solar building.

The different constructions of traditional and new building envelope are showed in table 1-4. There isn't thermal insulation layer in traditional building construction.

Table 1 Thermal parameter of external window

External window		specification	SHGC	U-Value W/(m2K)
Traditional	Single Clear	4 m	0.858	6.2
New	LowE Double	CLEAR +Air+ LowE (4+12+4)	0.631	2.0

Table 2 The construction and thermal parameters of external wall

External wall	Thickness (mm)		Conductivity W/m.K	R-Value (m2.K)/W		Specific heat kJ/(kg.k)	Density kg/m^3
	Traditional	New		Traditional	New		
Slush mortar	20	20	0.93	0.0215	0.0215	1.05	1800
EPS	0	60	0.04	0	1.25	1.4	15
Concrete Block	40	40	1.350	0.296	0.296	1.0	1800
lime mortar	20	20	0.87	0.0229	0.0229	1.05	1700
RO=∑R (Traditional/New)	0.34 / 1.82(m^2.K/W)						
U-Value (Traditional/New)	2.934 / 0.55{w/m^2-K}						

Table 3 The construction and thermal parameters of Plat Roof

Roof	Thickness (mm)		Conductivity W/m.K	R-Value (m2.K)/W		Specific heat kj/(kg.k)	Density kg/m3
	traditional	new		traditional	new		
Slush mortar	20.	20	0.93	0.0215	0.0215	1.05	1800

EPS	0	50	0.04	0	1.5	1.4	15
Reinforced Concrete	200	200	1.74	0.1149	0.1149	0.94	2500
Slush mortar	20.	0	0	0.0215	0	0	0
Cement mortar plastering	0	20.	0.93	0	0.0215	1.05	1800
RO=∑R (traditional/new)	0.16 / 1.40 (m^2.K/W)						
U-Value (traditional/new)	6.25 / 0.61 (W/m^2.K) (exclude inside and outside convection coefficient)						

Simulation result and discussion

In order to study the potential of passive solar building in Lasha regional, the object room is calculated and analyzed. Compare to the indoor thermal environment of the traditional building and that of new reconstruction passive solar building in the heating season. The results are showed in the Fig.2-3 (the radio of window and wall is 0.5). Fig.1 shows the compare results when the outside condition is bad (such as the outside temperature and solar radiation is low).

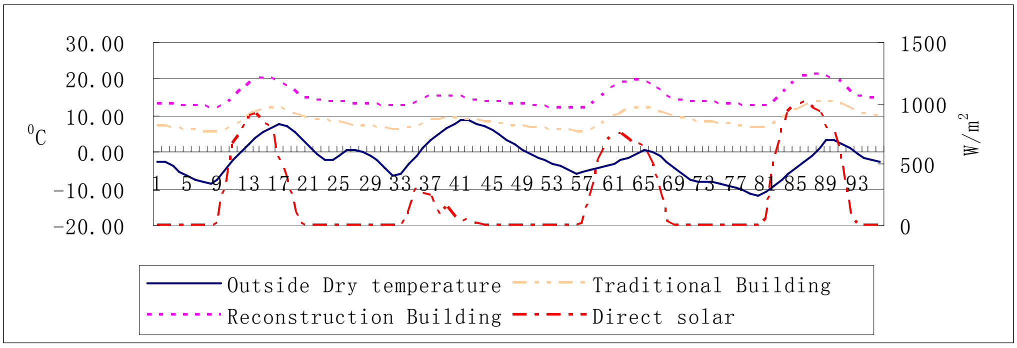

Fig.2 The mean operative temperature of two buildings when the outside condition is bad

Fig.3 shows the compare results when the outside condition is better (such as the higher outside temperature and solar radiation).

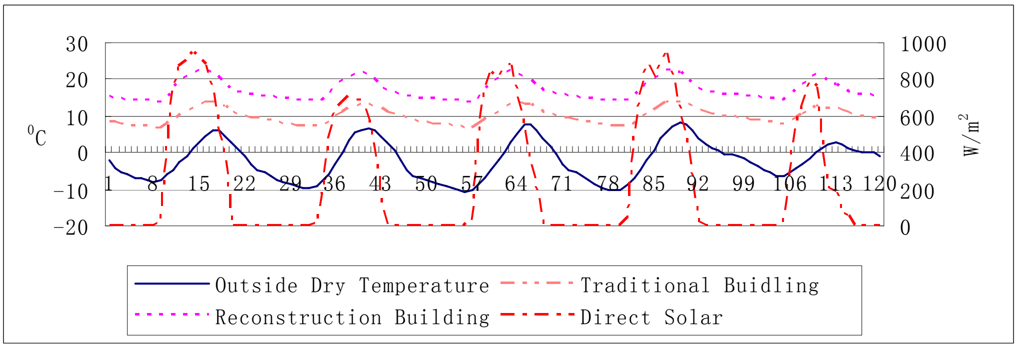

Fig.3 The mean operative temperature of two buildings when the outside condition is better

From the Fig.2-3, we know that no matter what the outside climate is, the traditional building indoor thermal environment is poor, the maximum value of the mean operative temperature is 10 ^{0}C, the minimum is 5 ^{0}C. This can't satisfy the thermal comfort requirement. When the building envelope is insulated, the indoor thermal environment is greatly improved. The value of the mean operative temperature is between 13~20 ^{0}C. In order to exploit potential of using passive solar heating in this regional, the key design parameters (such as the area radio of window to wall, the U-value of external wall) of direct gain passive solar are studied. The analyzing results are showed below.

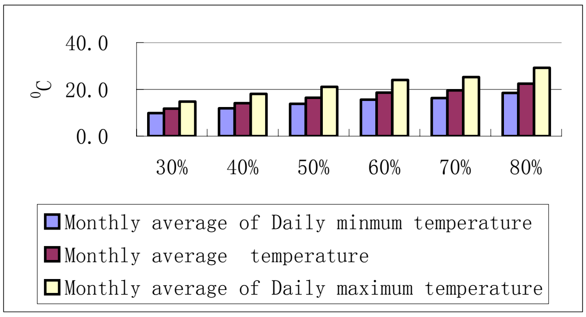

Fig.4 The mean operative temperature of building at different area radio of window to wall

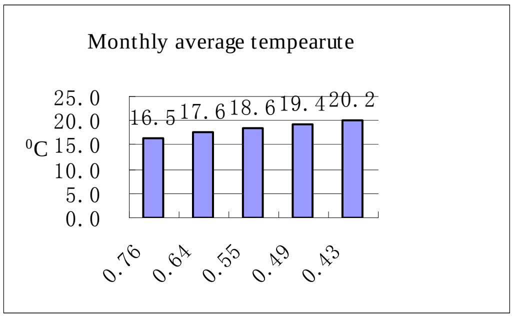

Fig. 5 indoor monthly average temperature at different U-value of external wall

From the Fig.3, we know when the monthly average temperature, monthly average temperature of daily maximum temperature and monthly average temperature of daily minimum temperature are considered together, the value of southern area radio of window to wall should between 0.5%~60%. When the radio is above 60%, the monthly average maximum temperature of each day exceed 24 ^{0}C and the people in this room will feel overheat. When the radio is less than 50%, monthly average temperature is below 16 ^{0}C, which can't satisfy the requirement of thermal comfort.

The fig. 5 gives indoor monthly average temperature at different U-value of the external wall, when window-wall radio is 60%. The U-value changes from 0.76 to 0.43.

From fig.5, we know that when the U-value of external wall is 0.55 and the monthly average operative temperature is about 18 ^{0}C, the U-value is reasonable.

Based on the above analyze, the area radio of window to wall should be 60% and the U-value of external wall should be 0.55. At this case, the yearly operative temperature is calculated and the coldest and hottest month results are showed in Fig6-7.

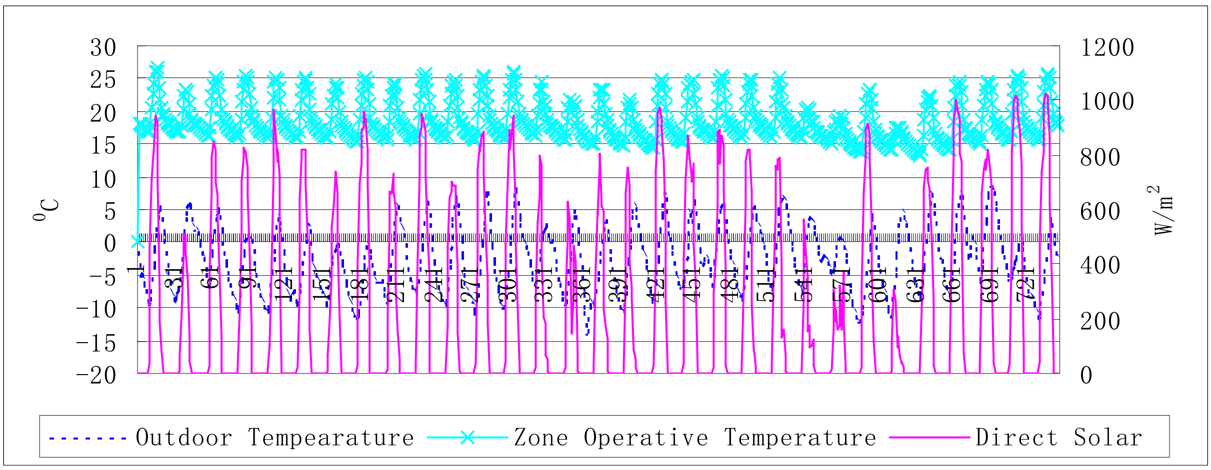

Fig.6 the operative temperature of the coldest month

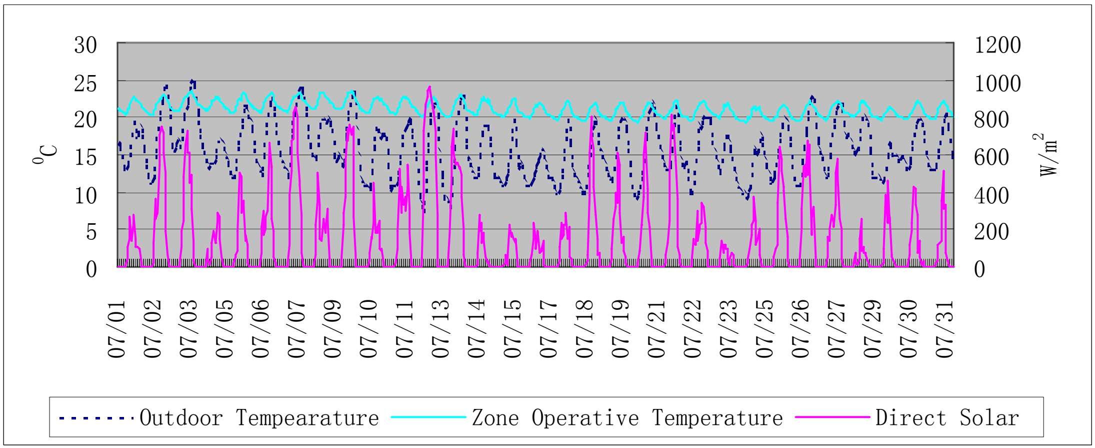

Fig.7the operative temperature of the hottest month

From the fig.6, we know that the operative temperature of the room is 15~25^{0}C at most time and the time of temperature below 15^{0}C or above 25^{0}C is very short. This means that the indoor thermal environment is satisfied and the overheat conditions or clod is not serious.

Because outside temperature of Lasha isn't very high in summer, the maximum outside temperature is about 25^{0}C, so the overheat condition isn't serious in summer. The figure 7 shows the indoor operative temperature variety in summer, the operative temperature is between 20~25 ^{0}C. This shows that the overheat condition isn't serious.We can know that the direct gain passive solar building with reasonable thermal design can provide a satisfied indoor thermal environment without consuming any petrochemical energy in LaSha.

Conclusion

Based on the above analyze, we can draw the conclusion that when the direct passive solar building is optimally designed, it can provide a satisfied indoor thermal environment in winter without consuming any petrochemical energy and the overheat condition isn't serious in summer. This show that the direct gain passive solar building should be applied on a large scale. The utilization of solar energy for building heating can reduce the petrochemical energy consuming and protect the zoology environment.

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Assessing thermal comfort of dwellings in summer using EnergyPlus

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SUMMARY

The simulation of a building as a whole offers an image of its behaviour, reflecting the complex relation established between the outdoor environment factors, the building characteristics, the use by its occupants, and the parameters which intervene in satisfying the comfort criteria. This paper gives an overview of the use of building simulation in Romania. It focuses on summer comfort.

INTRODUCTION

In specific climatic conditions (cold winters and hot summers), the housing fund of Romania, characterised, to a great extent, by a low level of insulation and high occupancy, is a great consumer of energy. Mention should be made that in the social mass housing units the energy is consumed exclusively to heat the space during winter time, the use of air conditioning plant not being the preferred choice. As the thermal climate rehabilitation measures are focused on saving the energy consumed for heating, the preoccupation for achieving comfort during summer time by passive measures appears as markedly topical.

Knowledge concerning the thermal climate parameters, their influence on the occupants and the influence of buildings and systems is today relatively known and established in international standards. Energy consumption of buildings depends significantly on the demands for the indoor environment, which also affects health, productivity and comfort of the occupants. The indoor environment is mentioned in prEN 15251 :2005 Criteria for the indoor environment including thermal, indoor air quality, light and noise. This standard presents IEQ (Indoor Environmental Quality) performance demands that can be used as input or default value for energy calculations, energy evaluations and building quality labeling.

One of the issues that is addressed in prEN 15251:2005 is the maximum allowable upper temperature in summer. For naturally ventilated buildings with a high degree of occupant control (e.g. access to operable windows and no strict clothing policy) the standard allows for the use of an adaptive criterion.

THERMAL COMFORT : MODELS AND CRITERIA

The environmental parameters that constitute the thermal environment are: temperature (air, radiant, surface), humidity, air velocity and personal parameters (clothing together with activity level). Criteria for an acceptable thermal comfort (PMV-PPD index) are evaluated by Fanger model. In the Fanger model the optimum internal condition for a building (i.e. one in which occupants will report comfort) is correlated exclusively to parameters referring to conditions which are internal to the building (mentioned above). Standard ISO 7730 is based on the steady state Fanger model of human physiology. The Predicted Mean Vote is strongly dependent on metabolic rate and clothing insulation. In the technical report 1752 (CEN 1998), are proposed 3 categories of thermal environment.

Table 1. Three categories of thermal environment. Percentage of dissatisfaction due to general comfort and local discomfort (ISO EN 7730, 2005, CR 1752, 1998.),[1]

Cate-gory	Thermal state of the body as a whole		Operative temperature °C		Max. mean air velocity m/s	
	PPD %	PMV	Summer (0,5 clo) Cooling	Winter(1 clo) Heating	Summer(0,5 clo) Cooling	Winter(1 clo) Heating
A	< 6	-0.2 < PMV < + 0.2	23,5 – 25,5	21,0 – 23,0	0,18	0,15
B	< 10	-0.5 < PMV < + 0.5	23,0 – 26,0	20,0 – 24,0	0,22	0,18
C	< 15	0.7 < PMV < + 0.7	22,0 – 27,0	19,0 – 25,0	0,25	0,21

For the last years several new CEN standards have been developed (e.g. USA – ASHRAE 55 2004 and the European draft pr EN 15251 which proposed Adaptive Comfort model based on comfort survey in the field.

The Adaptive Comfort model was proposed by Nicol and Humphreys. It states that people in real, naturally ventilated buildings, tend to adapt their comfort requirements to the prevailing outside temperatures. This model takes into account that people in real life situations are not functioning at constant conditions; instead they vary their activities, posture, metabolic rate and clothing according to the climate and its variations. Thus, the optimum indoor temperature (i.e. one at which occupants will report comfort) varies with the history of outside temperatures; in particular, it has a strong correlation with the average external temperature in the last few days. The adaptive principle is: if a change occurs such as to produce discomfort, people react in ways which tend to restore their comfort. They do this by making adjustments to their clothing, activity and posture, as well as to their thermal environment. The set of conceivable adaptive actions in response to warmth or coolness may be classified into five categories:

1. regulating the rate of internal heat generation
2. regulating the rate of body heat loss
3. regulating the thermal environment
4. selecting a different thermal environment
5. modifying the body's physiological comfort conditions

Design values for the indoor operative temperature as a function of mean monthly outdoor air temperature for buildings without mechanical cooling systems are shown in figure 1:

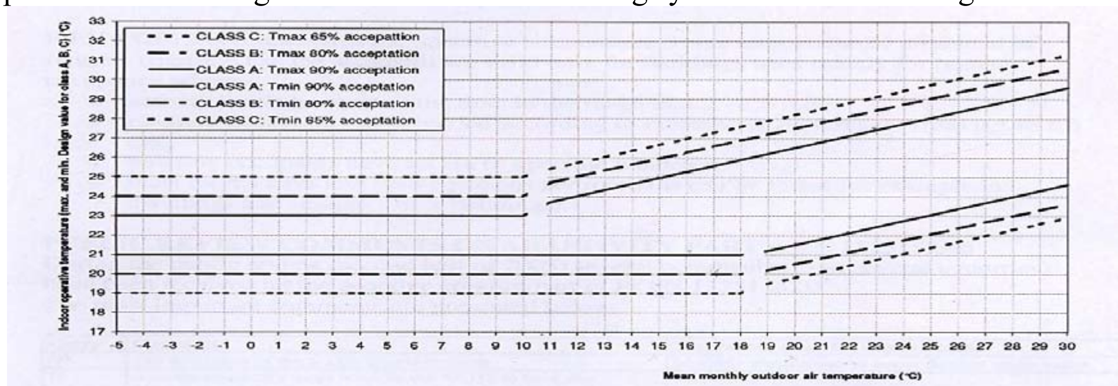

Figure 1. Design values for the indoor operative temperature, [2]

Four categories of adaptive thermal comfort are identified in prEN 15251:2005:

- Categories A: Thermal environment of the building meets the criteria of category A when the room temperature in the rooms representing 95% of the occupied space is not more than 3% of occupied hours a year outside the category A temperature limits.
- Category B: Thermal environment of the building meets the criteria of the category B when the room temperature in the rooms representing 95% of the occupied space is not more than 3% of occupied hours a year outside the category B temperature limits.
- Category C: Thermal environment of the building meets the criteria of category C when room temperature in the rooms representing 95% of the occupied space is not more than 3% of occupied hours a year outside the category C temperature limits.
- Category D: A building belongs to the category D if it does not meet the criteria of category C.

CASE STUDY

The building was modeled in EnergyPlus simulation environment. The main purpose is describing the thermal performance of the building from the viewpoint of external gains and losses (climate) and internal casual gains and losses. The building was modeled with 3 geometry thermal zones (fig. 2). All zones are virtually mutually ventilated by circulating air to keep uniform indoor environment in the entire building. Natural ventilation takes places through operable exterior windows and interior doors in all zones. In addition to natural ventilation, there are cracks in all exterior and interior walls. Cracks in the exterior walls allow infiltration of outside air into the zones. The location of the building is the city of Galati, in July. There was used a Weather File, with Rain Indicators. The occupation set up is the numeric representation of the way, the house is occupied and used by their residents. The important data are: number of occupants (3 in zone A, 2 in zone B), schedule of occupancy, the internal maximum ventilation ratio, and the schedule of weather, the openings (windows and doors, the ventilation set point 21^{o}C). The working hours are from 9.00 to 17.00. The wind pressure coefficient data are taken from literature,[3] for a simple rectangular building shape, surrounded by buildings of equal height. Hourly schedules with 24 schedule events are used for building parameters like window opening. These schedules are reused for all days of the simulation period.

EnergyPlus is an energy analysis and thermal load simulation program,[4]. It calculates the energy required to maintain each zone at a specific temperature for each hour of the day. A zone is an air volume at a uniform temperature plus all the heat transfer and heat storage surfaces bounding or inside of that air volume.

The building model must:

1. Determine heat transfer and heat storage surfaces ;
2. Define equivalent surfaces ;
3. Specify surfaces and subsurface (windows, doors, etc) construction and materials ;
4. Compile surface and subsurface information.

People, lights, equipment, outside air infiltration and ventilation all constitute internal gains for the thermal zone that are described as a peak level with a schedule for each hour.

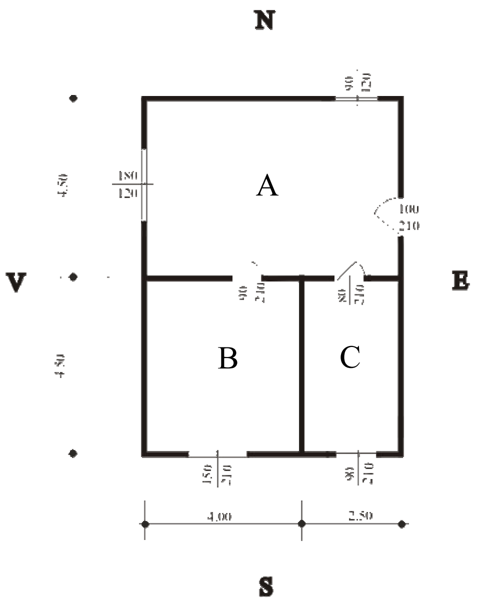

Figure 2. Test house

RESULTS, COMMENTS

By running the Energy Plus program there have been obtained data referring to the parameters which influence the comfort level. There have been considered such factors as indoor air temperature (fig.3), relative humidity (fig.4). The variation in time of the PMV index is shown in fig.5. and infiltrations (fig.6). On the bases of this information, it was possible to analyse the extent to which the comfort exigencies are met as against the Fanger and Adaptive model.

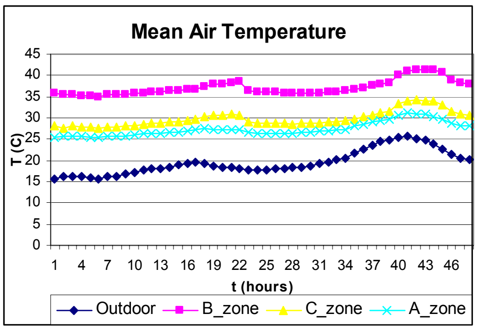

Figure 3. Mean Air Temperature

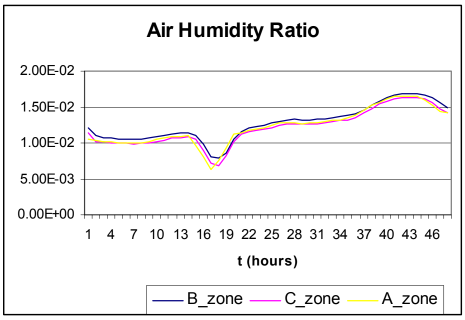

Figure 4. Air Humidity Ratio

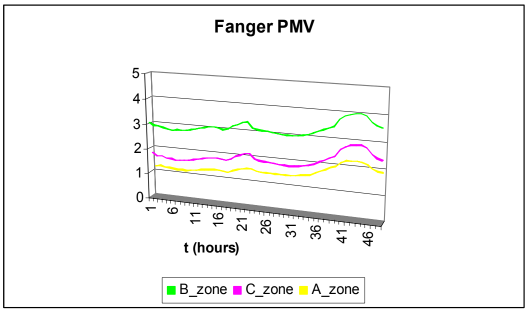

Figure 5. Fanger PMV , Simulation I

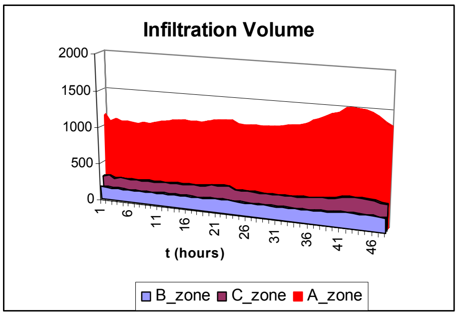

Figure 6. Infiltration Volume

It can be seen from fig.5 that the PMV index for the 3 zones varies between 1, 2, and 4, which means that not even the minimum conditions, for class C of comfort, are met. The least favourable situation is found in zone B, with double exposure.

The Adaptive Model: The operative temperature values for the 3 zones have been calculated and compared to the optimum ones recommended by the Adaptive Comfort Model (table 2, Simulation I). In the given situation the comfort criteria for zone A are satisfied, while in zone C the operative temperature values are relatively near the design ones. The research was resumed for a situation where passive measures were used in view of improving the comfort, such as: replacing glazing (table 2, Simulation II) in zone B and using blinds. The results obtained by running EnergyPlus are shown below:

Table 2. The operative and recommended temperature's values

Outdoor Temp	Design Value	Simulation I Zone operative values			Simulation II Zone operative values		
		A	B	C	A	B	C
17.8	24.7	24.4	33.1	26.2	23.1	23.8	25.0
18.0	24.7	24.4	33.0	26.2	23.1	23.8	25.0
18.1	24.8	24.4	33.0	26.1	23.1	23.8	25.0
18.3	24.8	24.5	32.9	26.1	23.2	23.8	25.0
18.4	24.9	24.5	32.9	26.1	23.2	23.9	25.0
18.8	25.0	24.7	32.9	26.2	23.4	24.0	25.0
19.2	25.1	24.8	33.0	26.3	23.6	24.2	25.2
19.7	25.3	25.0	33.1	26.5	23.8	24.4	25.4
20.1	25.4	25.2	33.2	26.7	24.0	24.7	25.6
20.5	25.6	25.4	33.4	26.9	24.2	24.9	25.8
21.8	26.0	26.0	33.7	27.2	24.9	25.5	26.2
22.7	26.3	26.5	34.2	27.8	25.4	26.1	26.8
23.5	26.6	26.9	34.5	28.2	25.8	26.5	27.2
24.4	26.9	27.4	34.9	28.6	26.4	27.0	27.7
24.8	27.0	27.8	35.2	29.0	26.7	27.3	28.0
25.3	27.1	28.8	37.1	30.9	27.7	28.5	30.1
25.7	27.3	29.1	37.9	31.5	28.1	29.0	30.6

The values presented in table 2, Simulation II, show that the comfort criteria are satisfied, over the simulation time, according to the adaptive model in the three zones, as follows:

Zone A – 100%
Zone B – 83.3 %
Zone C – 66 %

If a difference of 0.2 ^{0}C against the designed value is accepted.

The Fanger Model,[5]: The values of PMV index (fig.7) which vary between 1 and 2.5 for zone A orientated in the most favourable position, show that not even in this situation the minimum comfort criteria (class C) are met.

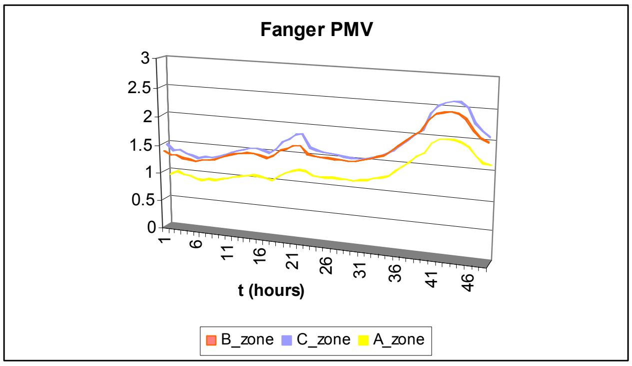

Figure 7. Fanger PMV, Simulation II

CONCLUSIONS

This paper gives an overview of the use of building simulation in Romania. It focuses on summer comfort. Summer comfort,[6] depends on the characteristics of the building envelope; in particular, the transparent components and the materials play a relevant role. In summer time, most of the cooling loads depend on the solar gains, coming from the transparent surfaces of the building. The analysis presented in this paper gives some useful information, concerning energy simulation of buildings. Transparent components deeply affect the energy balance of the building. The simulation of a building as a whole offers an image of its behaviour, reflecting the complex relation established between the outdoor environment factors, the building characteristics, the use by its occupants, and the parameters which intervene in satisfying the comfort criteria.

The Adaptive Comfort Model constitutes an instrument of realistic assessment of the quality of the indoor environment from the point of view of comfort in the naturally ventilated buildings, thus contributing to saving energy in air conditioning. Applying simple passive measures with minimum costs may led to achieving a summer time comfort which is acceptable even in dwellings of low surfaces and high occupancy degree.

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12 June 2007 at 13:30 - 15:00

B04
Energy performance of buildings and building stock

EULEB - European high quality Low Energy Buildings

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SUMMARY

The European research project "EULEB – European high quality Low Energy Buildings" intends to provide information about good examples of energy efficient buildings in use, in order to reduce prejudices and lack of knowledge of many key actors of the building market. Therefore, a multilingual CD and website was produced, containing detailed information about 25 buildings from all over Europe including measured data about energy consumptions, construction costs, comfort and user acceptance. 150.000 copies of the CD resulting from this work have been disseminated in the beginning of 2007 through European magazines reaching the target groups of architects and engineers as well as investors and property developers all over Europe. Furthermore, all information is available on the website www.EULEB.info
The project was performed by a European consortium of five Universities and a European umbrella organisation and it was partly funded by the "Intelligent Energy Europe" programme.

INTRODUCTION

The energy consumption in Europe is rising from year to year. The replacement of limited fossil fuels by renewable energies is proceeding slowly. As a result, the reduction of CO2-Emissions is proceeding slowly, too.

The building sector plays an important role in the total energy consumption. In order to tap this potential for energy savings and reduction of CO2-emissions the energy-efficiency of buildings has to be improved as soon as possible.

One important measure to achieve these goals is the legislation for new and existing buildings. With the European "Energy Performance of Buildings Directive" (EPBD), which has to be turned into national laws by the European member states, an important step into this direction has been done.

But unfortunately energy efficient buildings are sometimes facing prejudices and image problems resulting from bad examples from the past. Many people mistrust the real energy efficiency in use, the quality of architecture, the user comfort and the cost effectiveness. Other people just have concerns about energy efficient buildings as a result of lack of information.

These prejudices and the lack of information can be reduced and eliminated by providing information and detailed data of existing good examples to key actors of the building market. With a collection of 25 European high quality low energy buildings and detailed information about their architecture, their building concept, their measured energy performance and their building costs, EULEB helps to improve the image of energy efficient buildings and supports the implementation of the EPBD.

PROJECT TEAM

The EULEB project team consists of five European Universities of which the University of Dortmund coordinated the project and one international association. The Universities are represented by different institutes, which are working in the field of energy efficiency of buildings. They have formed a successful research and teaching network in which several projects already have been performed. The excellent knowledge of the team's national building stock allowed a detailed identification of buildings from the five largest European countries.

The sixth partner was REHVA, the Federation of European heating and air-conditioning associations. REHVA is a 43 year old umbrella organisation, representing 30 member associations of European experts for building services. Thus, REHVA has direct contact to about 110.000 key actors of the European building market. They supported the project with the expert knowledge and their network which was used for disseminating the results.

Table 1. List of Project partners involved

Universität Dortmund, Lehrstuhl für Klimagerechte Architektur
London Metropolitan University, LEARN
Università degli Studi di Firenze, ABITA
Université de La Rochelle, LEPTAP
Universitat Politècnica de Catalunya, AiE
Federation of European heating and air-conditioning associations (REHVA)

SELECTION OF BUILDINGS

A total of 50 public buildings have been identified, predominantly from the countries where the University Partners are situated. To cover the large variety of climatic conditions in Europe, buildings from the very far north (Scandinavia) as well as from the very south of Europe, (Mediterranean countries), were included. Out of the identified buildings, there had to be a selection of five buildings per University Partner for further examination.

For the selection of buildings a simple evaluation system was designed. In this first step, each building was evaluated concerning its qualification to the project. Seven categories (such as quality of architecture, energy consumption, availability of monitored data etc.) with different weightings were used in this subjective evaluation methodology. Low ratings in some of the criteria had to lead to a direct exclusion of a project (for example lack of monitored data).

After having evaluated the 50 identified buildings, a ranking of the buildings identified by each partner could be established, showing which buildings fulfilled the overall criteria best. This process led to the selection of 25 buildings from all over Europe.

SELECTED INFORMATION

The EULEB project should address to different target groups, namely architects, engineers, investors and property developers. All these groups have a very different knowledge background as well as different interests in terms of building information. As EULEB could never satisfy 100% of the different interests in all details, the challenge was to provide a basic set of general building information. A number of details should address the different target groups and raise the user's interest to search for more information.

Besides the different building use, the locations of the buildings from all over Europe result in a large variety of differences of the buildings´ boundary conditions, especially in terms of climate. Therefore a visualisation of the buildings locations with reference to the climatic zones of KOEPPEN was used to introduce the user to the influence of locations on the building design and building energy consumption.

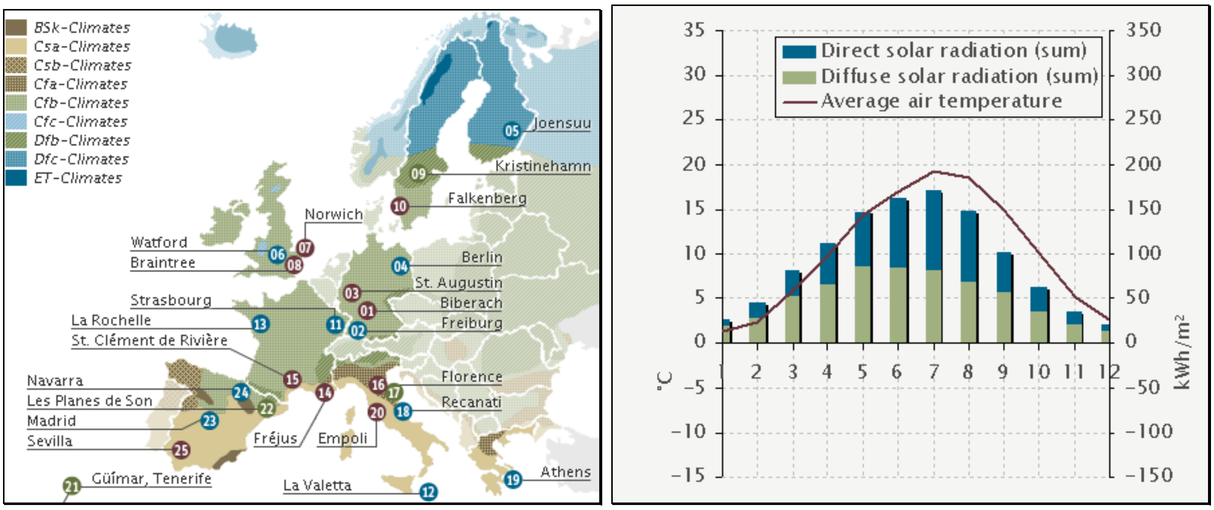

Figure 1. Locations and climatic zones of selected buildings and example of climatic data in EULEB

Each building presentation starts with some general information about the exact building name, location address, relevant persons involved in the design and construction, information about the buildings´ areas and volumes etc. These information are accompanied by images (external and internal views) and plans (site maps, floor plans, elevations and sections) visualising the architecture of the building. To visualise the appearance and the atmosphere of the buildings, short professional video clips provide extra impressions beyond the static images and plans.

The climatic conditions are described in more detail by statistical diagrams and some figures, like the ASHRAE classification [1] which relates the climate to the energy consumption of buildings. The next sections give general descriptions of the building construction including the thermal quality of the building envelope and the general energy concept and technical systems.

Special features of the building concept, which are significant for the energy efficiency of the building, are described and visualised in more detail. These features have been grouped to the categories Insulation, Solar control, Lighting, Heating, Cooling, Ventilation, Materials, Renewable energies, Co-Generation and Rainwater use.

The energy performance of the buildings is evaluated with measured energy consumption. Where available, the consumption is separated by fuel type (electricity, gas, oil etc.) and load type (heating, cooling, ventilation, lighting etc.). This allows giving a detailed overview on the building's energy performance and calculating the respective partly and total primary energy consumptions. All values are related to the building's usable floor area and compared with national average and standard values where available.

The real building costs have been documented in relation to the usable floor area of the building. Where available, the total costs are divided by cost categories and national average costs for a similar building have been used for comparison.

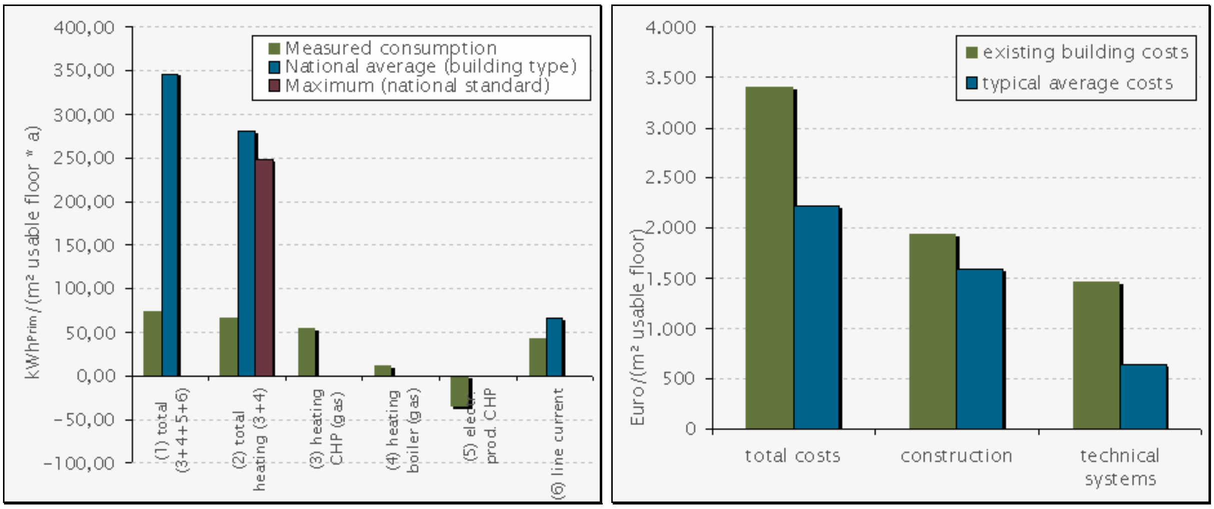

Figure 2. Example of Energy performance and construction costs in EULEB

The visual comfort has been measured by luminance pictures, which have been taken and processed for all buildings in the curse of the EULEB-project. They give an impression of the luminance and possible glare effects in a typical room of the building.

In order to evaluate the quality of the buildings, user acceptance studies have been performed in all 25 buildings. Using a simple questionnaire with 12 questions concerning temperatures, air quality, lighting, comfort, user influence, architecture etc. allowed creating a detailed evaluation and an overall user acceptance.

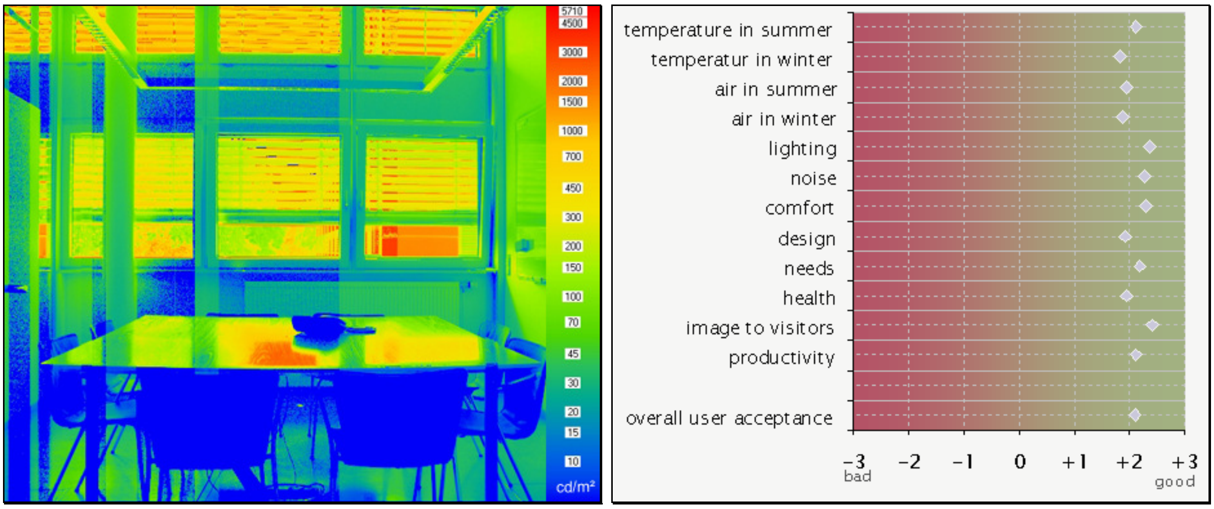

Figure 3. Example Visual comfort expressed by luminance pictures and results of post occupancy studies in EULEB

Based on the building data described above, the “Benchmarking” section provides a comparison of the building costs, the energy performance and the user acceptance of all 25 buildings. These characteristics depend on many variables such as building use, climatic conditions etc. which have to be taken into account. Thus, in EULEB the buildings have been grouped according to building type and climatic zone. Although this of course does not take into account all relevant parameters (user influences, etc.) a rough comparison could be established.

With this compilation of building data, the general and specific interests in terms of design, technical details, energy performance and costs of the target groups Architects, Engineers and Property Developers have been addressed as much as possible.

COMPARISON OF BUILDINGS

The comparison of different buildings generally is a very sensitive process as many different boundary conditions like the use of a building, the climatic background etc. Still then, other factors like the occupants' behaviour will have influences on the results and are very difficult to separate.

In the EULEB project, the 25 buildings have been compared in the section "Benchmarking" in the three main categories "Economy", "Energy" and "Quality". The buildings have been grouped by building type and by climatic zone. The climatic zones have been defined according to the ASHRAE-classification system [1]. This method takes into account the heating and cooling degree days of a certain location and thus takes into account a building's energy demand resulting from climatic conditions.

The Economy-Benchmarking shows a significant variance of the total building costs, within one building type and on climatic zone. But discounting some extreme values, general statements can be found: The total building costs for office buildings constitute generally about 1400 and 2100 €/(m² usable floor area). For the educational buildings this value generally reaches between 1000 and 1500 €/(m² usable floor area).

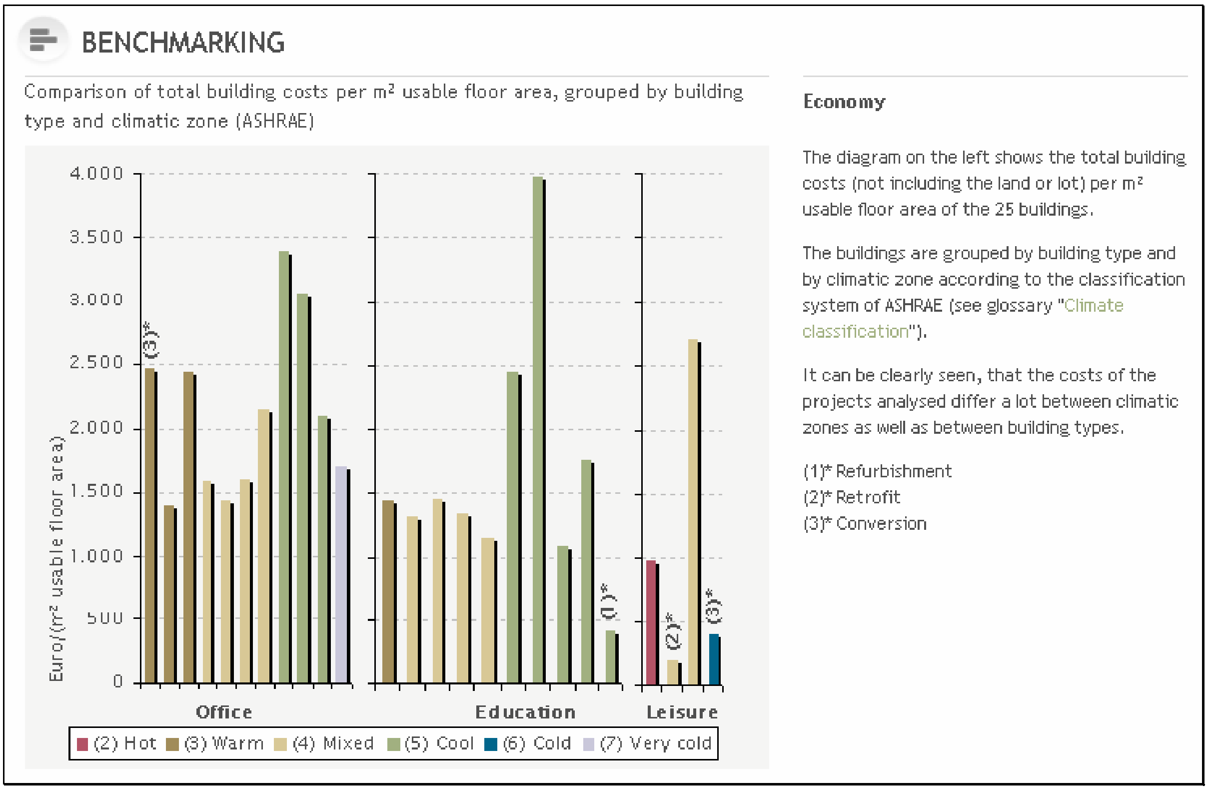

Figure 4. Comparison of building costs in EULEB

Comparing the energy consumptions of the 25 buildings again brought a significant variance even within one building type and climatic zone. But again, a general statement could be, that for example for all three building types, a total primary energy consumption between 100 and 220 kWhPrim/(m² usable floor area) can be achieved.

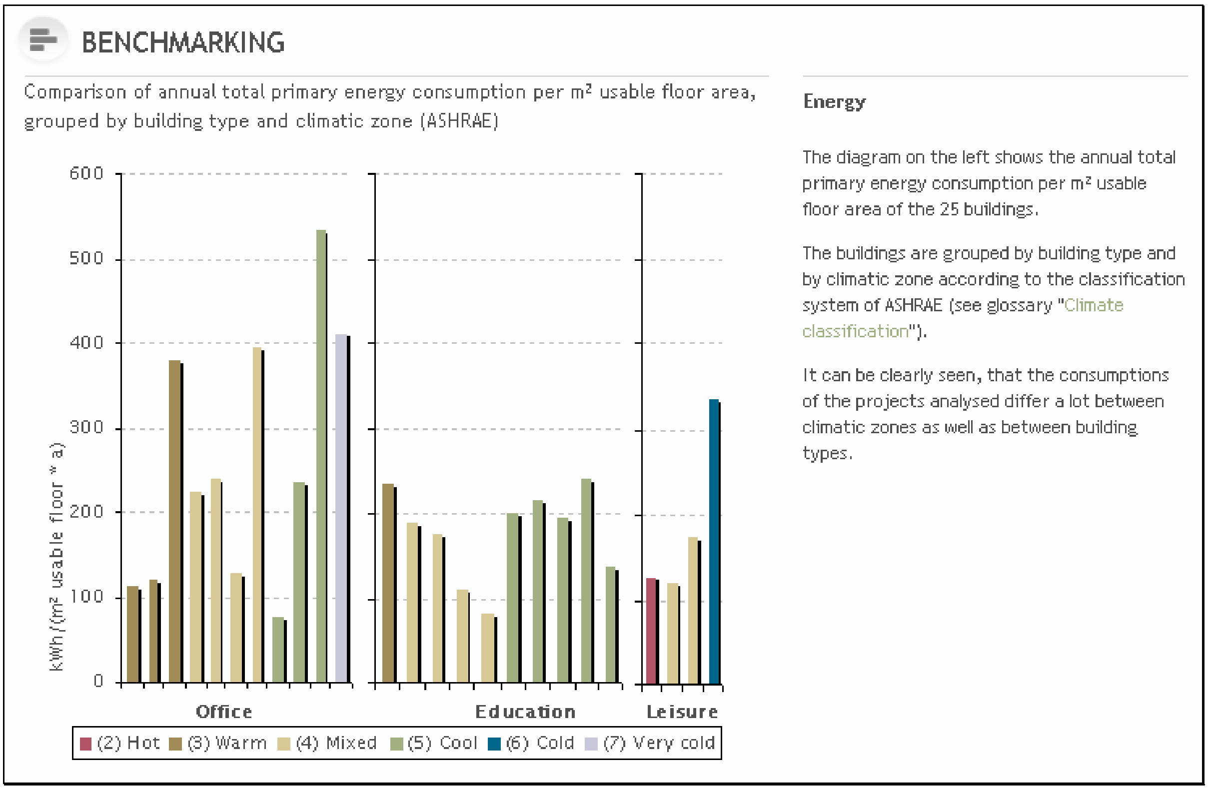

Figure 5. Comparison of energy consumptions in EULEB

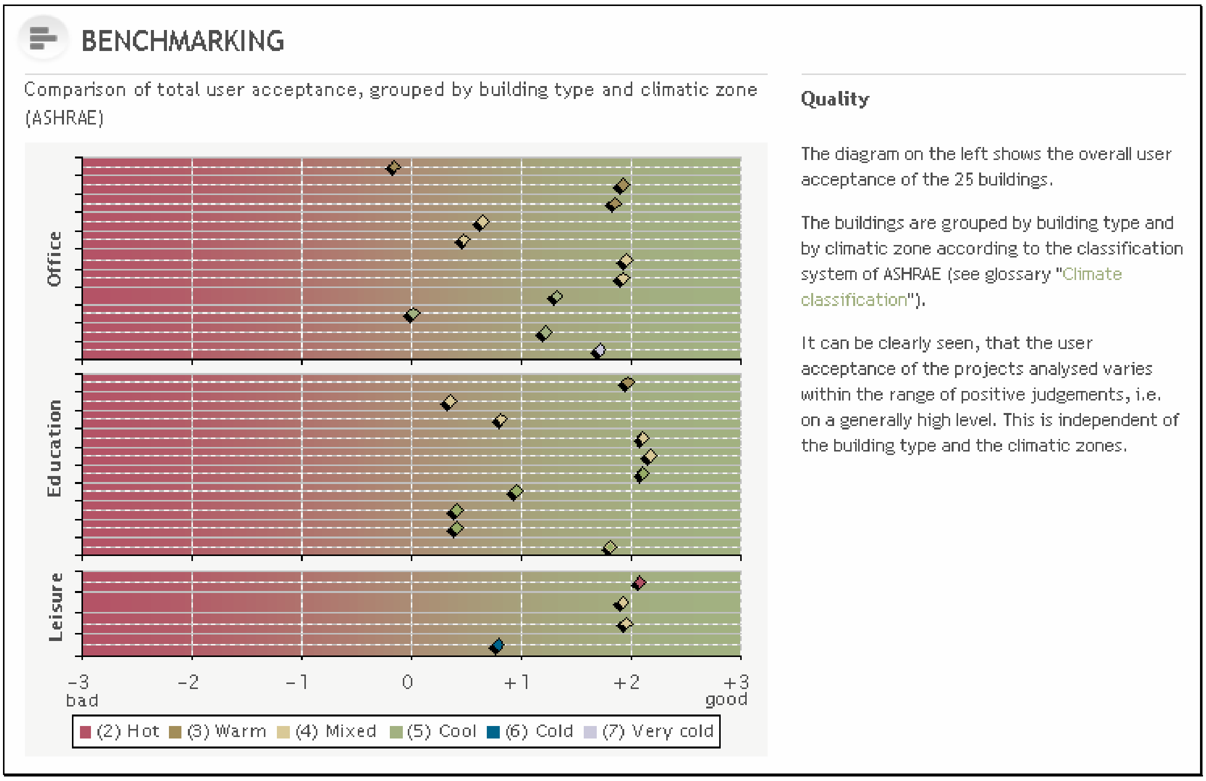

Figure 6. Comparison of overall user acceptance in EULEB

The overall user acceptance has been compared in the Quality benchmarking diagram. Generally positive results are obvious, independent of the building type and the climatic zone.

ACKNOWLEDGEMENT

This project is partially supported by The European Commission.

Intelligent Energy Europe

Figure 7. Support by the European Commission

The sole responsibility for the content of this paper lies with the authors. It does not represent the opinion of the European Communities. The European Commission is not responsible for any use that may be made of the information contained therein.

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Energy efficient service buildings with ecofacility: support for planners and building developers

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SUMMARY

The construction of a building with an optimized thermal and energy performance and an accordingly low energy demand does not necessarily require higher investment costs. The decisive factor is an interdisciplinary and foresighted – an Integrated Energy Design. Although this integrated approach prolongs the design process and increases the planning budget, in return the construction time and the subsequent costs such as energy and operating costs are significantly reduced. A qualitative (using checklists) and quantitative assessment (calculation of heating and cooling demand, building energy simulation) is essential for the identification of flaws in the design. In order to ensure the longevity of the quality of design beyond the realisation of the project, guarantee models can be applied. This is a model of a “slim” integrated planning process, but applying this concept step by step should help to overcome the introduction barriers. The ecofacility consulting services should be the first step.

INTRODUCTION

Newly built service buildings - such as office buildings, shops, hotels or schools - with a final energy consumption exceeding 400 kWh/m²/yr are still common. However, several examples exist which prove that service buildings with a final energy consumption of less than 40 kWh/m²/yr are equally feasible [1].

What is the reason for this considerable disparity in energy consumption levels? Although the use of innovative technologies certainly plays an important role, an even more decisive factor is the choice of an integrated design approach. An approach that focuses not only on the appearance and functionality of the building, but also accounts for its energy performance during the utilisation period. Decisions made at an early design stage, such as those concerning the shape and orientation of a building, have a crucial impact on the building’s future energy consumption [2]. Only an integrated design process at this early stage allows for optimisation in as many areas as possible (architecture, function, building code, comfort, investment costs and operating costs). For a timely identification of areas for possible improvements, a regular assessment of the draft according to certain criteria becomes necessary at the end of every planning stage. At the first design stage – the predesign stage – the main assessment criterion is the net energy demand; later – when the installations have been chosen – the final energy demand becomes relevant. Additionally, an analysis of total investment and running costs is possible with the help of a life-cycle cost estimation.

Concerning life-cycle costs, it is widely recognised that 80% of operating, maintenance and repair costs of a building are already determined during the first 20% of the entire planning process – i.e. during the predesign and design stage [3] (see Figure 1). In the subsequent stages, the chance to lower these costs decreases.

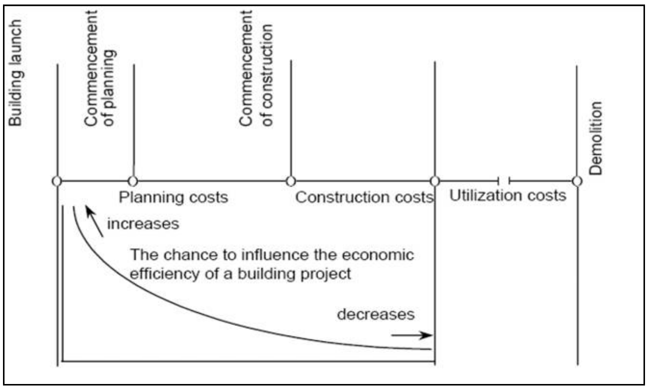

Figure 1: Chance to influence the economic efficiency of a building [3]

The same concept applies to the energy demand of a building: the earlier the energy demand of the building is taken into account and optimised during the design stage, the bigger the chances for success. Due to the fact that at this stage of the design process the amount of time and money invested is still relatively low, redesign is still feasible and financially insignificant. An integrated energy design process and regular assessment provide the basis for a quick and efficient completion of the project, less construction defects and lower repair costs – and therefore help maintaining low investment and repair costs [4].

In the course of implementing the Energy Performance of Buildings Directive (EPBD) on a national level, every member state incorporates minimum standards for the energy performance of buildings in their national building codes. As soon as the implementation process is completed, new buildings as well as existing buildings that are subject to major renovation will have to comply with certain thermal and energy performance criteria. Thus, building developers have to take low energy demand for buildings into account in future.

How can the construction of an energy efficient building be realised? How can real estate developers be supported to fulfil the requirements of national implementation of EPBD? How can the theoretical concepts of integrated energy design, life-cycle cost analysis and optimisation of thermal and energy performance of new buildings be put into practice and become part of the standardised construction process of building developers? How can clients make sure – especially at the project initiation stage and during predesign – that an energy efficient service building is built in the end?

ecofacility, the programme of the Austrian Climate Protection Initiative “klima:aktiv” aims at increasing energy efficiency in private service buildings, has developed a standardised consulting process to achieve these goals. The ecofacility consulting services help building developers to incorporate energy efficiency as a key factor into the planning process right from the beginning. The aim is to design buildings that can operate with minimal energy consumption, i.e. with minimal heating and cooling demand. The better the building envelope is adapted to climatic conditions and the function of the building, the lower the energy demand [5]. The choice of installations should be based on the estimated energy demand. This way, it is guaranteed that the installations fit the building and that investment costs are not increased by the installation of unnecessary and oversized equipment. In addition, indoor environmental quality must not be neglected; simple measures such as allowing for individual indoor temperature regulation by opening the windows raise the comfort level of the building occupants.

METHOD: CONSULTING SERVICES FOR BUILDING DEVELOPERS

The ecofacility consultants make sure that the future energy demand of a building is taken into account at every planning stage – from the first draft to detailed design – and especially at the early design stage. The best results can be achieved when the consulting activities take place within the framework of an integrated energy design process under independent guidance. However, the consultation is based on a modular concept so that the right service can be provided at every stage of the process.

Definition of goals

At the beginning of the whole design process, it is necessary to determine the energy criteria the building has to meet during the utilisation period.
At the beginning of the design stage of a building project, the client usually has only a rather vague and general idea of his objectives. Therefore, it is essential to translate the client's vaguely stated wishes and goals (strategic goals) into clear and measurable requirements[6,7]. These requirements form the basis of the entire design process and can be used in the design competition or the design bid (see Figure 2).

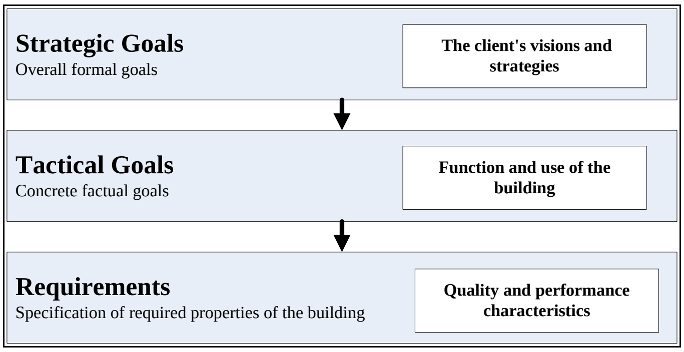

Figure 2: Process of defining requirements [6]

The requirements have to contain criteria, target values and indicators. They can be very general in nature and specify only maximum values of final or primary energy demand, or they can specify in detail maximum values of energy consumed by heating, cooling, lighting, etc. Some requirements even fix details such as minimal U-values for building components, obligatory installation of an exterior shading system, avoiding the installation of a chiller, etc. In addition to the requirements, an assessment method has to be defined which makes it possible to check at every design stage if the decisions made meet the requirements [8]. This specification of requirements combined with an assessment procedure makes it possible already at the initial planning stage to check the compliance with energy efficiency criteria. At the beginning of the design stage, it is necessary to find out which innovative or energy efficient options are available for the building in question. The number of options is determined by some basic decisions concerning e.g. the location or shape of the building which have to be made or are already made when design activities start. If the location of the building has already been chosen, it can be determined whether e.g. the prevailing winds allow for a movable exterior shading system or the noise level allows for natural ventilation.

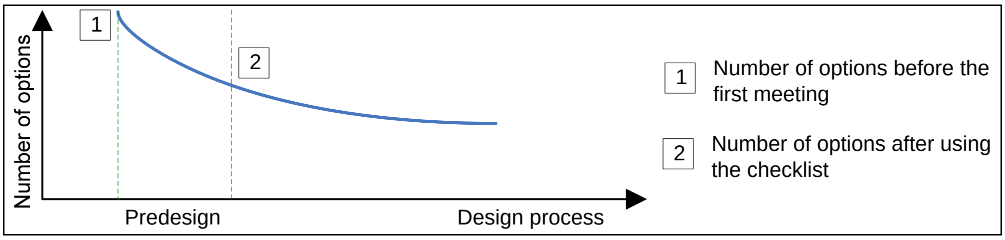

Figure 3: Reducing the number of options with the help of a checklist (Source: Austrian Energy Agency)

The development of comprehensive concepts and a closer look at different options often reveal conflicting goals. Finding an optimal solution which resolves these conflicts allows for significant energy savings, but only if this is done at a very early stage of the design process [9].
In addition, it is necessary to define which of the requirements are mandatory and which of them may still be changed in the course of the optimisation process [9,10]. A list of questions which is discussed with the client in the first meeting assists this process of defining requirements (see Figure 3).

Assessment of building energy performance during predesign

During the predesign phase, the architectural concept is assessed with respect to the estimated energy consumption. Based on the first sketches and drafts, several observations are already possible: e.g. whether a shading system is necessary to lower the cooling demand or the installation of a chiller is required. Afterwards, suggestions for improvements of the preliminary draft as regards thermal and energy properties are submitted that aim at a reduction of the total energy demand. Possible ways to lower the cooling demand or to even avoid the installation of a chiller are the reduction of the amount of glazed surfaces (e.g. by adding a parapet) or the installation of solar control glazing.
This assessment of building energy performance is carried out with the help of two tools:

- Calculation of the heat and cooling demand of the building: the net energy demand is calculated in order to determine whether an improvement of energy efficiency is advisable or not an absolute necessity.
- Qualitative analysis of the preliminary draft based on a standardised checklist: the checklist covers various aspects of energy efficient building practice. The draft is evaluated according to every aspect in the checklist, and – if necessary – suggestions for improvements are noted. The checklist covers the following aspects: shape of the building / architectural concept, orientation of the building / the sections of the façade, compactness of the building, number of internal rooms, effective thermal mass, thermal protection, thermal bridges, summer thermal protection (window-wall ratio, shading systems, orientation of the glazed façades), orientation of the office spaces, passive solar energy use, daylighting and ventilation. The online information site provided by ecofacility (http://tool.ecofacility.at) offers property developers and contractors an overview of the main aspects that determine the thermal and energy properties of the building.

The above two procedures are included in the ecofacility report on the preliminary draft which is presented to the client and serves as a basis for further detailed design of the building.

Assessment of thermal and energy properties during the schematic and detailed design phase

During design development and detailed design, the ecofacility consulting services offer an in-depth analysis: An hourly simulation of the building's thermal and energy performance simulates possible realisations of the building and its installations, including internal and external influences and their synergies. These simulations help in spotting flaws in the design and make sure that the energy performance of the building and its installations are optimised. Based on the results of the different simulations, the estimated energy costs of the building can be calculated. For the calculation of the life-cycle costs, additional operating costs such as maintenance, repair and cleaning costs are estimated and, together with the energy costs, added to the investment costs.

Guarantee models for the operational phase

To ensure that the quality of the design process is entirely reflected in the construction and operation of the building, various guarantee models can be applied, depending on the project delivery method (single product contractor, general contractor, total contractor or integrated system provider). Similarly to Energy Performance Contracting projects, the guarantee models apply to energy costs / energy consumption and quality of realisation. The ecofacility consultations support clients with the integration of these guarantee elements into the contract.

COLLABORATIONS

In order to increase the know-how and thus continuously improve the services, ecofacility collaborates with several EU projects. The project "Network for the Promotion of RTD results in the field of Eco-building technologies, small Polygeneration and renewable heating and cooling technologies for buildings" (PEP-NET) within the Sixth Framework Programme of the European Community for Research, Technological Development and Demonstration is developing technology profiles for energy efficient office building, creates dialogue platforms between planners, technology developers and manufacture and makes pilot studies to support building developers when applying innovative products. In the framework of "Integrated Energy Design in Public Buildings" (INTEND), an "Intelligent Energy – Europe" project, several countries compare their processes for thermal and energy performance optimisation of service buildings, assess other concepts and develop common approaches. In the course of the EIE project KeepCool, the promotion-project of 'sustainable cooling' in the service building sector", extensive documents and checklists concerning summer comfort have been developed in the last few years which are now used in the consulting process.

CONCLUSION

The Integrated Design Process is a minimum requirement for developing energy efficient buildings in the tertiary sector. Taking many aspects into account, starting from the first planning phase is the key for optimising the building. But on the other hand, professional office building developers often don't have enough time and resources to apply such a comprehensive approach in the planning phase. These developers need support in every planning stage, edited on their specific requirement, in order to develop energy efficient buildings. Ecofacility developed a consulting service process for building developer. This service is modular in order to be applied in every planning stage. This approach should support to overcome the introduction barriers in applying more comprehensive planning strategies like integrated planning. However, this is not an broad integrated planning process, but applying this concept step by step is for the target group of real estate developer in the tertiary building sector more realistic.

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Life Cycle Optimization of Extremely Low Energy Buildings

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SUMMARY

A global methodology is developed to optimize concepts for extremely low energy dwellings, taking into account energy savings, environmental impact and financial costs over the life cycle of the buildings. Energy simulations are executed with TRNSYS. The ecological impact is evaluated through a life cycle inventory of the whole building, whereas costs are evaluated through a cost-benefit analysis. The multi-objective optimization problem is solved with a combination of genetic algorithms and the Pareto-concept. Firstly the optimization methodology is presented. Subsequently the main results are presented and a hierarchy of cost-effective energy saving measures is derived. Finally the impact of economic parameters, such as price evolutions and discount rate is discussed together with the strengths and weaknesses of extremely low energy buildings from an economic point of view.

INTRODUCTION

Since the oil crisis of the early seventies energy consumption in buildings is a hot item. During the first years most research focused on the reduction of transmission losses, later also on the optimization of passive solar gains. The next step was the search for better performing heating systems and the reduction of ventilation losses through heat recovery systems. From the early nineties on, the interest shifted towards the overall sustainability of buildings, taking into account the environmental impact over the whole life cycle of the building [1].

Currently extremely low energy dwellings are in the picture, ranging from passive houses [2,3] over zero energy houses [4-6] to energy autarkic buildings. The latter often is considered as the ultimate objective for sustainable buildings [5]. Though, these concepts assume the application of a good many technologies resulting not only in much more embodied energy and pollution, but also in more resources and higher costs. Although these buildings have much smaller energy consumption during the usage phase, the projects hardly ever show clearly if the global balance of energy, ecology and cost is finally positive.

An important tool for evaluation and quantification of environmental impact is life cycle assessment (LCA) [7]. Traditionally LCA is mostly concerned with materials, components and product design and rarely buildings as a whole are subject of a detailed LCA. Furthermore, LCA is mainly used as an instrument to quantify in- and outflows and environmental impact, but lacks to be an instrument to optimize decision making [8-11].

Optimization of a building as a whole is also a complex problem due to the amount of parameters and variables, the non-linear relations and second order effects. During the two last decades evolutionary computational optimization techniques, which are adaptive methods inspired by the genetic processes of biological organisms, have been receiving increasing attention for their potential for such complex problems [12]. One of their main advantages is that they do not have much mathematical requirements about the optimization problem [12]. However, application in building related engineering is rare until now [13-15].

So, the current trends for sustainable buildings lack an underlying scientifically based methodology to develop and evaluate dwellings that are optimized for energy, ecology and costs. The development and implementation of such a methodology is presented here. Basic principle is a well-founded evaluation of the environmental impact and financial cost during the whole life cycle of the building and its installations, by coupling LCA and cost assessment with advanced optimization techniques. This results in concepts and guidelines for globally optimized extremely low energy building concepts.

METHODOLOGY

The developed methodology consists of three main pillars. The first pillar concerns the multi-objective evolutionary optimization strategy. The second pillar consists of databases for LCA and investment costs whereas the third pillar concerns the building energy simulations.

Reference dwellings and energy saving measures

The objects for optimization are residential buildings. In order to obtain representative results for Belgium, a number of reference dwellings are designed following the statistical average of the Belgian residential sector [16]. To include the compactness, being the proportion of heated air volume and heat loss area, as a parameter, five typologies are defined, resulting in five references dwellings: one terraced dwelling, one semi-detached house, two individual dwellings (one with a simple square plan and one with a fragmented plan) and an apartment flat. To assess the impact of the energy saving measures, the non insulated version of the dwellings is defined as the reference situation. The geometry of each reference dwelling is fixed. Only the glass area per room can vary in order to include its impact on net heat demand and summer comfort. The thermal capacity is included by incorporating both light weight (wood frame) and massive variants (cavity wall and massive wall with outer insulation).

The parameters for optimization are related to energy saving measures applied to both the building envelope and the heating system. The optimization itself is performed in two steps. In the first step only envelope-related energy saving measures are considered. Insulation measures are taken for the roofs, attic floor, façade and ground floor. The insulation thickness in each envelope component can vary from zero to a maximum and different types of insulation material can be applied. Table 1 gives the maximum applicable insulation thickness per envelope component, as well as the applicable materials.

A large number of different glazing types is considered with a U-value varying from 2.8 W/m²K to 0.4 W/m²K and a g-value varying from 0.76 to 0.21. Different types of window frames are considered, such as frames of wood, PUR, PVC and/or aluminium with a U-value varying from 6 W/m²K to 0.65 W/m²K (passive house frames). Summer comfort can be affected by adding movable internal or external shading with an opaque fraction of 0 to 100% and/or by choosing one of four ventilation scenarios for extra summer or night ventilation.

These ventilation scenarios are combined with four levels of air tightness, ranging from the average of new built dwellings [17] to the passive house standard for air tightness.

Table 1. Maximum insulation thickness and applicable materials per envelope component

Envelope component	Maximum thickness [cm]	Applicable materials
Flat roof	30	MW, PUR, CG
Sloped roof	40	MW, PUR, EPS, XPS, CF
Attic floor	40	MW, PUR, EPS, XPS
Façade massive variant	30	MW, PUR, EPS, XPS
Façade light weight variant	30	MW, CF
Floor	15	MW, PUR, EPS, XPS

Legend: MW = mineral wool; PUR = polyurethane; CG = cellular glass; EPS = expanded polystyrene; XPS = extruded polystyrene, CF = cellulose fibre

In the second step, the measures on the building envelope are combined with system-related measures. This includes systems for distribution, emission, production and storage of heat, natural and mechanical ventilation systems with or without heat recovery, systems for local electricity production and control systems. Traditional systems, such as high efficiency or condensing boilers, are considered, but aim of the research was also to focus on more innovative technologies, such as heat pumps, cogeneration of heat and power, different levels of heat recovery of ventilation losses, combination of mechanical ventilation with heat recovery and integrated electrical heater and renewable solar energy systems.

Multi-objective optimization

To deal with the large number of parameters and variables that are involved, the evolutionary optimization method of genetic algorithms (GA) is selected. The algorithm starts from a set of individuals called a population, in which each individual, called a chromosome, represents a possible solution to the problem. Solutions from one population are selected to create a new population. Solutions that are selected (=parents) to form new solutions (=offspring) are selected according to their fitness, which means the more suitable they are for the problem, the higher their probability to be selected for reproduction [18]. This normally leads to a new and improved population. Applied to the reference buildings, each potential variant of these buildings is represented unequivocally as a set of parameters, joined together in a string of values. This string forms the chromosome, in which each value represents one parameter, being the insulation thickness, glazing type, type of sun shading, glass area, etc. for a particular building design. The fitness of a solution depends on its primary energy consumption, net present value and global warming potential.

To deal with the multiple objectives that need to be optimized simultaneously, the Pareto concept is selected [19-21]. This concept treats all objectives equally during optimization and deduces the trade off between them by determining the non-dominated solutions. A solution is non-dominated if no other feasible solution exists that decreases one objective without causing simultaneously an increase in at least one other objective. For two objectives this results in a curve of non-dominated solutions (Pareto-curve), for three objectives in a surface of non-dominated solutions (Pareto-surface). For each solution in a population the Pareto score equals the number of solutions in the population by which the solution is dominated. Non-dominated solutions have score 0. Constraints, such as summer comfort and insulation standard, are treated through a penalty function. If one or more constraints are not met, a penalty is added to the Pareto score, proportional to the violation of the constraint. With the

Pareto concept all information on the objectives is maintained during the optimization process and weighting of the objectives can be postponed to the decision making process.

Life cycle inventory for buildings

A life cycle inventory (LCI) database for building-related materials and components is established to provide the ecological input in the optimization process. It also allows an analysis of the relation between the energy savings realized with extremely low energy buildings and the embodied energy needed to create these buildings.

As the optimization process aims at developing building concepts that are not only globally optimized, but also satisfy boundary conditions for indoor comfort and IAQ, the LCI does not focus on materials or building components, but considers the building as a whole. However the very long lifetime of buildings makes hypotheses on processes of the end phase highly uncertain and even unrealistic. Moreover, most buildings undergo several refurbishments during their lifetime, often resulting in thorough modifications. Therefore not the whole life span of the building is considered, but only the impact of one generation. This results in the scenario that the building is designed and constructed and then used and maintained by one generation during 30 years. Thus the phases of extraction, production, transport, usage and replacement are taken into account, whereas the end phase of the building is not considered.

The LCI is mainly limited to an inventory of energy flows and emissions. No impact indicators are calculated, except for the global warming potential. For each phase, LCI models are developed for materials, components and systems and aggregated data are calculated based on LCI data from the ecoinvent2000 database [22]. The energy consumption and CO_2 emissions during the usage phase are determined with dynamic building simulations with TRNSYS. The input data for ventilation and infiltration are calculated with COMIS. The weather data are hourly average data for the Test Reference Year of Brussels, Belgium. Summer comfort is evaluated with the method of weighted temperature exceeding hours according to [23]. Finally the sum of energy flows from all phases and the corresponding emissions form the optimization criteria for energy (total primary energy consumption) and ecology (total global warming potential). A sensitivity and uncertainty analysis is performed on the LCI building model. It showed that the sensitivity for errors and the propagation of errors is limited and that the errors of the different input data neutralize each other somehow. A more detailed description and discussion of the LCI building model can be found in [24].

Cost assessment

The economic impact of the building concepts is integrated in the optimization model through a cost-benefit analysis for a private building owner. Several economic criteria are calculated, such as investment cost, yearly energy cost and cost savings, total and net present value, static pay back time, etc. However, only one cost criterion is incorporated in the optimization process. As in the first step no assumptions are made on the heating system, no energy cost can be calculated yet. Therefore the initial investment cost serves as cost criterion in this phase. In the second phase all costs can be calculated. For this phase the net present value (NPV) serves as optimization criterion, to be able to extract the economically viable solutions.

All cost data for materials, components and systems are based on up to date (from 2005) price information and include working hour cost and VAT. Mostly a discount rate of 4% is adopted, but as the real discount rate is uncertain, different scenarios have been analyzed: a discount rate of 2%, 4% and 8%. Energy prices are based on private consumer prices for Belgium for May 2006. For the energy price evolution three scenarios are adopted based on [25]: low (+0%), medium (+2%) and high (+3 to 4% depending on the energy carrier).

RESULTS

Figure 1 presents the results for the individual dwelling with simple plan, with all variants analyzed during the optimization process in grey and the final optimal solutions in black. For each variant the total primary energy consumption over 30 years is given together with the corresponding NPV over 30 years. The results are valid for a low energy scenario.

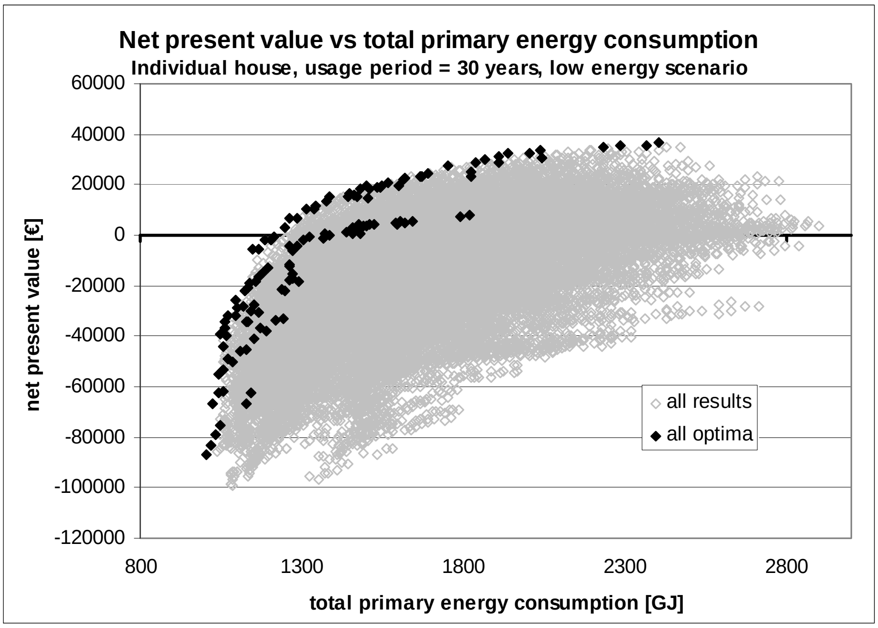

Figure 1. Globally optimized solutions for the individual dwelling

Some optimal solutions seem to be suboptimal as solutions exist with higher NPV for the same total primary energy consumption. This misleading visual effect is caused by the fact that the optimisation process considers three objectives, whereas the figure only shows two of them. Thus the optima in figure 1 are a 2D projection of that 3D trade off front, which does not coincide completely with the 2D trade-off curve.

Obviously the optima with a positive NPV are of most interest, as they are economically viable, even in the improbable case that the energy prices remain constant. Similar results are found for the other reference dwellings. Table 2 presents a summary of the results for all reference dwellings. Cost results are valid for the low energy scenario and a discount rate of 4%. Most values are expressed per m³ heated air volume (energy related values) or per m² heated floor area (cost related values). The insulation level in Belgium is expressed as a K-value, that depends on the mean U-value and the compactness. Therefore table 2 also gives the legal maximum for the mean U-value per reference dwelling.

Figure 2 presents the initial extra investment cost for the energy saving measures on the building envelope and installations as a function of the yearly energy cost. The graph is valid for the terraced house, but similar results are found for the other reference dwellings.

Table 2: Summary of the results for the global optimization of all reference dwellings

	Terraced house	**Semi-detached house**	**Individual house**	**Architectural house**	**Flat, centre**
Number of optima economically viable	79 16	138 48	125 62	36 15	35 17
Insulation level $U_{mean, optima}$ [W/m²K] $U_{mean, legal}$ [W/m²K]	K15-27 0.20-0.37 0.60	K13-34 0.15-0.40 0.52	K13-40 0.14-0.43 0.48	K13-20 0.13-0.20 0.45	K9-38 0.15-0.69 0.90
Net energy demand [MJ/m³,a]	15-90	15-90	25-100	30-55	< 40
Total primary energy consumption over 30 years [MJ/m³]	2000-4500	2240-4650	2230-5330	2500-3700	1300-2500
Total GWP over 30 years [kg/m³]	90-330	100-290	95-330	120-230	100-190
Extra investment cost [€/m² floor area]	40-500	80-800	3-840	100-720	50-340
Yearly energy cost [€]	300-880	300-900	300-1000	300-650	130-510
TPV over 30 years, low energy scenario [€/m²]	1100-1300	1000-1600	1000-1700	1000-1500	800-1050
NPV over 30 years, low energy scenario [€/m²]	(-260) - 60	(-360) - 300	(-500) -200	(-200) - 400	(-160) - 70
Static payback time [yrs]	9 - 50	3 - 57	(-4) - 62	2 - 23	(-3) - 35

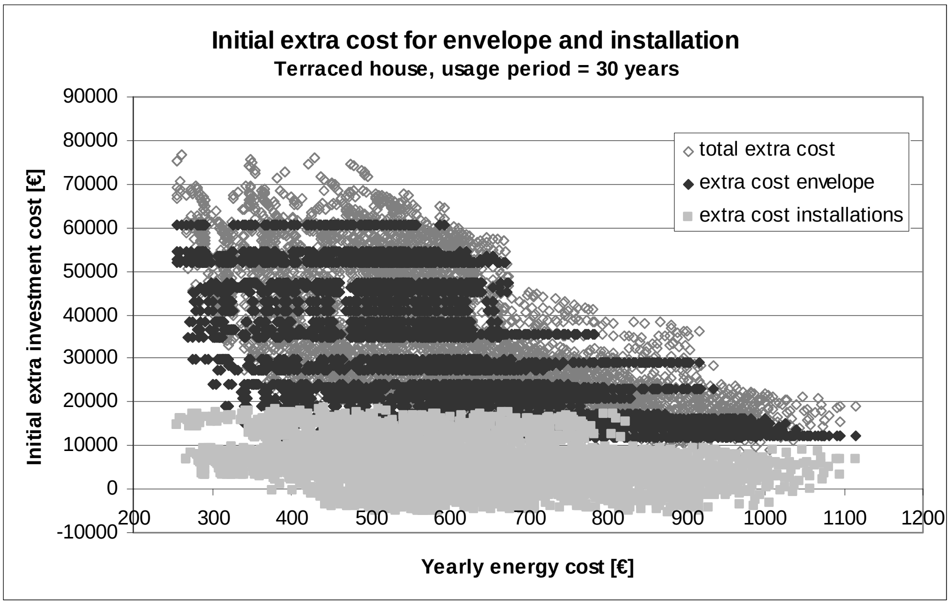

Figure 2. Terraced house: extra investment cost for building envelope and installations

DISCUSSION

Without any insulation the terraced house with a heated volume of 446m³ and a heat loss area of 220m², has a yearly energy cost of ca. 1600€. To halve the yearly energy cost, a total extra initial cost of at least 8.000 to 9.000 € is needed, consisting of 12.000€ extra cost for the building envelope and a negative extra cost of -3.000 to -4.000€ for the installation due to smaller dimensions of the installation. Further decrease of the yearly energy cost leads to an exponential increase of the initial extra cost. To realize an extremely low energy variant of this house with a yearly energy cost less than 300€, an extra cost of 60.000€ is needed for the highly insulated building envelope (U_{mean} = 0.14W/m²K) and 20.000€ extra for the installation (heat pump and mechanical ventilation system with heat recovery), compared to the non-insulated version. Similar results were found for the other reference dwellings. As table 2 shows, the NPV over 30 years for these variants is negative and thus, these variants are not economically viable.

Of importance however is that large cost and energy savings can be realized, also with economic viable concepts, if the hierarchy of energy saving measures is respected. This hierarchy is derived from the results and consists of the subsequent steps of most cost-effective energy saving measures:

1. Firstly invest in a good insulation level with good air tightness and a well designed natural ventilation system. The economic optimal insulation level lies far beneath the legal Flemish insulation requirement ($U_{mean, optimal} \approx$ 0.3-0.4 W/m²K ↔ $U_{mean,legal}$ = 0.5-0.6 W/m²K for compactness of 1.5 to 2m)
2. Secondly select a well performing heating system, at least a high efficiency boiler or condensing boiler with variable water temperature. Combined with the economic optimal insulation level, this represents the overall economic optimum. If the budget is available, a heat pump can be a good alternative, as it has a better energy performance. However due to its higher investment, it is beyond the economic optimum.
3. Finally, if for any reason, further decrease of the energy consumption is needed, heat recovery of the ventilation losses, a solar driven system and/or CHP is an option. However, although these measures improve the energy performance of the building, none of them is not economically viable over 30 years.

Analysis of energy price evolutions and discount rates showed that the economic optimum not only remains economic viable for all analyzed cases, the optimal combination of measures is also independent of these cost scenarios. Also the hierarchy of energy saving measures appeared to be independent of these scenarios. Obviously the economic viability of some measures, such as the application of heat pumps or mechanical ventilation with heat recovery, will depend on the cost scenarios, but in contrast to what sometimes is assumed, none of the scenarios causes a shift in the hierarchy of energy saving measures.

Concepts for extremely low energy dwellings have been determined for all reference dwellings, but none of them appeared to be economically viable for the current energy prices or discount rates. Only in case of much higher energy prices, some of these concepts will become cost-effective over 30 years. However, the largest barrier for all these concepts is the extremely high investment cost. Without financial support or incentives, these concepts will be limited to a minor part of consumers with a high environmental consciousness that is willing to invest such a large budget in an extremely energy saving house.

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Bringing an energy neutral built environment in the Netherlands under control

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SUMMARY

In this paper insight is given in the potentials for energy efficiency and renewable energy sources, specifically when applied to the built environment in the Netherlands. To this end, an analysis is presented of the building stock development from now to 2050 and building concepts and scenarios for a mid-century energy-neutral built environment in the Netherlands. Special attention is given to the potential of energy management systems in buildings.

1. INTRODUCTION

Our world is strongly on the move. The climate changes. More and more uncertainty exists concerning the energy supply. The discussion is no longer about the need to contemplate the future of energy supplies but about the way we can meet the future energy demand in a sustainable way. Both national and international policies are developed in the climate field and people devote themselves to energy transition.
In the Netherlands more than one third of the energy is used in the built environment. Insulation of buildings, more efficient comfort installations and local production of sustainable energy have strongly improved the energy performance of buildings in the previous decades. The potential for even better energy performance however has still not been exhausted. The urgency to bring all measures for improvement of the energy performance into action, and thereby connecting to nationally and internationally pursued policy, increases.
The Dutch research institutes TNO and ECN have started the strategic cooperation Building Future (BF) in the field of energy in the built environment in order to jointly give an impulse to this transition. Both institutes believe that by the middle of this century energy neutrality in the Dutch built environment can be reached, provided that the developments to this end are tackled energetically.

2. SCENARIO ANALYSIS

2.1. The scenario analyses tool

To assess the potentials of technological developments use is made of a scenario-analysis tool. Specific details about this tool can be found in [1].
The energy consumption profile of the Dutch built environment in 2000 is illustrated in Figure 1. The figure illustrates the distinct difference between residential and non-residential buildings: in residential buildings 65% of the total primary energy is used for heat (for space heating and domestic hot water), whereas in the non-residential sector electricity is dominant (52%). It is interesting to note that, although the total primary usage associated with the non-residential sector is lower than that of the

residential sector, the average energy per m^2 floor area is much higher in the non-residential sector. This is illustrated in Figure 2.

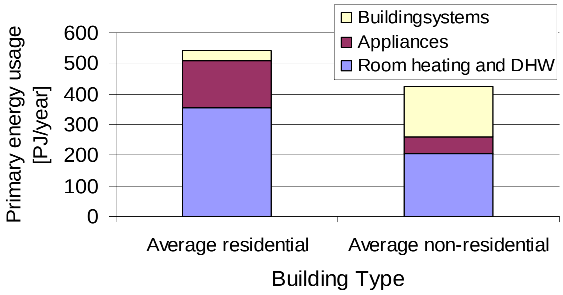

Figure 1. Building energy consumption profile for the built environment in 2000

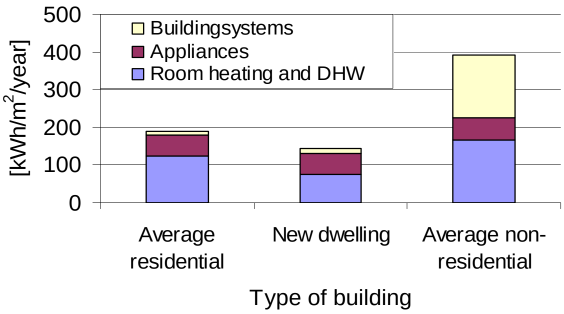

Figure 2. Building energy usage in 2000

Figure 2 shows that in the non-residential sector the higher primary energy usage per m^2 floor area is, for the main part, due to the larger amount of energy used for building systems (mainly HVAC and lighting).

3. ENERGY IMPACT MEASURES

3.1. Breakthrough technologies

In order to bring about the sought for transition to an energy neutral built environment, implementation of measures are considered which comply with the following philosophy:

1. Maximum demand reduction. With respect to buildings this implies measures such as:
   a. optimal insulation of the building envelope, minimal infiltration
   b. use of heat recovery on ventilation air and domestic hot water
   c. use of energy efficient appliances (A-label, energy conservation lamps)
   d. avoid standby losses of electrical appliances
2. Optimised use of renewable energy (RE), implying:
   a. optimal use of passive renewable energy such as solar heat, daylight and natural ventilation
   b. high-efficient renewable energy production, for instance from (high temperature, vacuum) solar collectors (possibly in combination with a μ-Organic Rankine Cycle (μ-ORC)), PV(T) panels, (small scale) windturbines
   c. balance demand and supply of renewable energy, both in energy sort (heat or electricity, temperature or power) and in time
   d. make use of (compact) energy storage, either local or using grid-connected / community systems
3. Technologies which make efficient use of fossil energy sources such as:
   a. heat pump (compression, sorption, magneto-caloric or thermo-acoustic)
   b. μ-Combined Heat (Cold) and Power Generator (μ-CH(C)P), stirling driven or using fuel cells (SOFC/PEMFC)
   c. community systems for instance using waste heat from waste processing plants

4. ENERGY IN THE BUILT ENVIRONMENT

4.1. Business as usual

Future energy consumption of the built environment mainly depends on the development of the building stock, but also on global and national developments, such as climate change, national turnover / economy and policy-effectiveness. The building related modifications influencing energy consumption involve construction of new residential buildings, demolition, renovation, extensive maintenance and installation replacement.

Taking all these developments into account, future energy consumption according to the 'Business As Usual' scenario can be outlined [2], [3].
Figure 3 illustrates that the expected total primary energy consumption in the building sector remains approximately 1000 PJp per year. Two significant trends contribute to this stabilization. Due to improved building insulation and more efficient comfort installations, gas consumption is expected to decrease. On the other hand, residential electricity consumption increases, mainly due to increasing cooling needs and increased use from household appliances (HHA). A growing number of ICT related applications contribute to increasing electricity consumption in the non-residential sector.

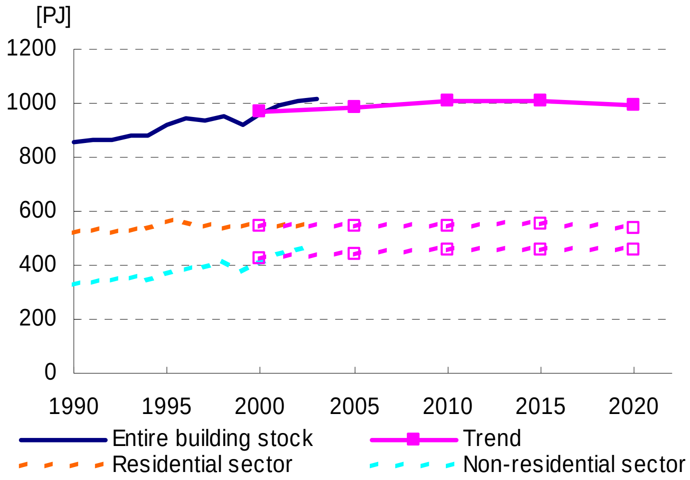

Figure 3. Total annual primary energy consumption in the building sector according to the 'Business As Usual' scenario.

4.2. Building Future scenario

The research program of Building Future (BF) aims at reducing the total net energy usage of the built environment to energy-neutrality around the middle of the century. To this end several technological and non-technological developments have to be undertaken linked to the building stock improvement moments.
To assess the potential of measures with the scenario tool first the effect of the measures separately is examined:

- Replacement of heating installations (every 15 years) including compact heat storage and thus reducing the energy for heating and domestic hot water (DHW), by 50% overall through system replacement, will result in a reduction of 130 PJ/year by 2050.

- If from 2015 onwards all new to build residential buildings would be energy-neutral a reduction of 65 PJ would result by 2050.
- Even if from 2015 onwards the introduction of net energy producing residential buildings would be taken on (starting at app. 4.5 GJ/year/building), a maximum reduction of 80 PJ/year could result by 2050.
- On the other hand, when all effort is directed towards renovation, for instance by implementing renovation packages that will reduce the demand for heat and DHW by a factor 4 and additionally integrate Renewable Energy (RE) from the sun in the building, the total reduction will not exceed 145 PJ/year by 2050. With a more modest package, but compared to current best practice still ambitious, consisting of only 50% reduction in demand for heat and DHW, a reduction of approximately 70 PJ/year by 2050 results, which is in the same order as the variant with energy neutral new buildings.
- Reduction of energy used by HHA by 2% per year from 2015 onwards might result in a total reduction of 145 PJ/year by 2050.

The above findings illustrate that, if an energy neutral built environment is to be reached, focus on only one of the building stock developments will not suffice; all the building stock improvement moments have to be used and a reduction of the user-related energy (HHA) should be accomplished as well. The required combination of measures (which are interrelated) for the residential sector is listed in table I.

Table 1. Impact of developments in residential sector on energy consumption [%] compared to the original situation according to BF scenario.

Mutation	Heat (%)	Electricity (%)	RE (kWh/a)
- new buildings	-40/year	-	4000
- renovation	-75	-20	2000
- maintenance	-15	-20	-
- installation replacement	-50	-50	-
- household appliances	-	-2	-

On top of these measures, the total energy consumption of the non-residential sector should decrease from 0% in 2010 up to 5% per year in 2025. This decrease in fact implies the same trend in energy consumption as for the residential sector. Most of the targets listed, albeit ambitious, can be achieved with introduction of the required building concepts around 2015, using existing technologies and technologies currently under development.

If all of the abovementioned measures are set in place the resulting development of the net energy usage of the built environment is depicted in Figure 4. Note, that the effect of the separate measures can not simply be added to obtain the result for the combination of measures, since some are interrelated. Figure 4 illustrates that building related measures (for new to build residential buildings and the existing stock) are not sufficient to reach energy neutrality. Also reduction of user related energy (HHA) and measures on district level (yellow line) should be introduced.

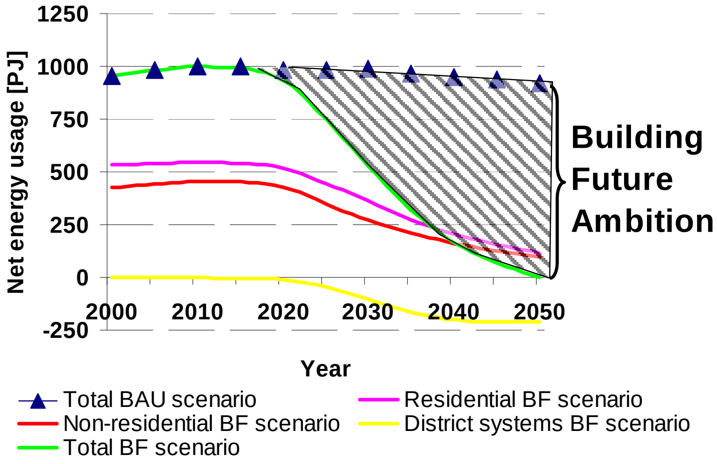

Figure 4. Development of net energy usage in Dutch built environment.

The effect of the Building Future scenario in terms of CO_2 emission directly related to residential buildings and user related electricity, is an emission reduction from 31.4 Mton in 2000 to 5.5 Mton in 2050. This implies an 83% reduction of building related energy for the residential sector with respect to the reference year 2000. With respect to Business as Usual in 2050 it implies a 2.5 times higher reduction of Mton CO_2.

5. BUILDING CONCEPTS AND KEY TECHNOLOGIES

The above conclusions on required ambitions for building concepts set the R&D agenda for Building Future. Energy management systems play an important role.

5.1. Building concepts

In the previous sections it is shown that in order to reach an energy-neutral built environment in the Netherlands around mid-century, there is a demand for building concepts with a higher (energy) ambition level than currently available. These ambition levels are the set point for a number of building concepts for residential buildings and non-residential buildings and are based on expected availability of technology in the year 2010, with estimated payback times of the complete concepts of approximately 15 years. In all concepts a strategy is used in which as a first step the energy demand is reduced as much as possible, in line with PassivHaus targets. The second step is to fulfil the need for energy wherever possible with renewable energy sources and, thirdly, using fossil fuels if still needed in an energy efficient way. As an example a short description is given of a concept for residential retrofit:

- Increasing the insulation levels of walls, floor and roof to R_C=7.5 m^2.K/W. Windows are replaced by triple glazing U=0.6 W/m^2.K.
- Improving the air tightness to 0.6 air changes/hour at 50 Pa.
- Heat recovery from ventilation air by using either a central balanced system or decentralised solutions like a facade integrated HVAC or a window-frame integrated system (new development).
- Application of external shading to prevent overheating.

As a reference case a typical Dutch terraced house of the late sixties is selected, since the major part of the housing stock currently up for renovation resembles the building characteristics of this type. The aim is to reduce the total energy demand for gas and electricity 'on the meter' to one quarter of the existing energy end-use before renovation. This means that besides building related energy (heating, domestic hot water, ventilation, lighting, etc.) also building related measures to reduce energy consumption of household appliances have to be taken into account. In addition to the reduction of the energy demand for space heating, described above, the following measures are incorporated:

- Heat recovery for DHW from the (renewed) shower
- Installation of intelligent sockets designed to minimise standby losses of HHA
- Heat recovery and storage connected to water distribution system of dish water and washing machine
- Replacement of standard boiler by a micro-Combined-Heating-Cooling-Power-generator micro-CHCP operating in combination with a solar collector.

The energy usage before and after renovation are illustrated in Figure 5.

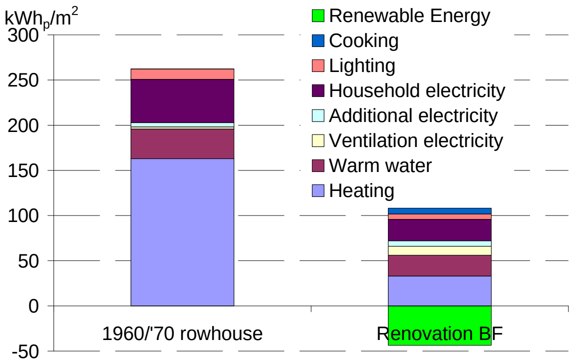

Figure 5. Energy usage before and after renovation.

5.2. Energy Management systems

Recent research has shown that the technical savings potential of advanced control systems in the whole Dutch built environment is estimated to be in the order of 190 PJ of primary energy per year (19% of the total energy usage of the Dutch built environment) and 12 Mton CO_2 emission reduction per year.

In table 2 and 3 below the estimated savings potential of some of the most promising control measures for the residential buildings and office buildings are shown. In addition to environment-adaptive control also user-adaptive and user-educational control is taken into account. User-adaptive control is characterized by adaptation of control to the behaviour and the preferences of the user. User-educational control goes a step further and aims to influence the behaviour and the preferences of the user, for example by providing feedback on the consequences of the current behaviour.

The estimates for the savings potential in the residential sector are based on model calculations for a typical Dutch terraced house built in the 70s. The estimates for office buildings are based on literature and practical experiences. They are calculated for office buildings built in the 80s. Both types of buildings are to be renovated shortly. To arrive at an estimate for the total savings potential the applicability of these measures in the current building stock has been taken into account. [4]

Table 2. Technical savings potential of advanced control measures in the residential sector.

control measure	savings potential per dwelling (GJ_p/a)
environment-adaptive	
irradiation control with solar shading	2.3
user-adaptive	
presence-based control of ventilation	3.0
window-closure-based control of space heating	0.3
disconnection of appliances in standby-mode	7.2
presence-based control of space heating	4.6
user-educational	
feedback on ventilation behaviour	1.9
general feedback on measured energy use	2.6
influencing hot water use to match available solar heat	0.4

Table 3. Technical savings potential of advanced control measures in office buildings.

control measure	savings potential per m^2 (MJp/m^2)
environment-adaptive	
daylight-dependent control of lighting	23
summer night ventilation, (incl. heat load reduction)	20
irradiation control with solar shading	14
user-adaptive	
presence-based control of ventilation	210
presence-based control of lighting	23
disconnection of appliances in standby-mode	41
user-educational	
improving operation and maintenance through monitoring	359
close windows during heating	20

From the tables it is clear that user-adaptive and user-educational control measures represent a large potential in addition to current state-of-the-art control measures (including environment-adaptive control). In the residential sector user-adaptive control measures represent two-thirds of the estimated savings potential, whereas user-educational control measures account for one-thirds of this potential. In office buildings the majority of the potential is covered by improved operation and maintenance through continuous monitoring. Presence-based ventilation also has a large per m^2 contribution, but in most of the situations where it is applicable this is already applied.

In addition to the use of advanced control systems for the reduction of primary energy use, it is also possible to introduce demand response in the built environment. This does not directly lead to the reduction of the primary energy use, but it will be necessary in future electricity grids with a high penetration of intermittent, renewable energy sources. In the residential sector there is a significant potential for demand response by shifting the electricity demand of various appliances: 34 PJ electricity demand response per year.

These figures indicate the large energy saving potential of energy management systems. At TNO and ECN, research is focussed on developing energy management systems, ranging from environment-adaptive, user-adaptive up to user-educational. First experiences in the latter category have been acquired in the EU-FP5 project EBOB (Energy Efficient Behaviour in Office Buildings [5]), see Figure 6. The effect of the system in an office building situated in Sweden, showed that more than 50% energy saving could be accomplished on indoor climate systems, with the major part on the heating system.

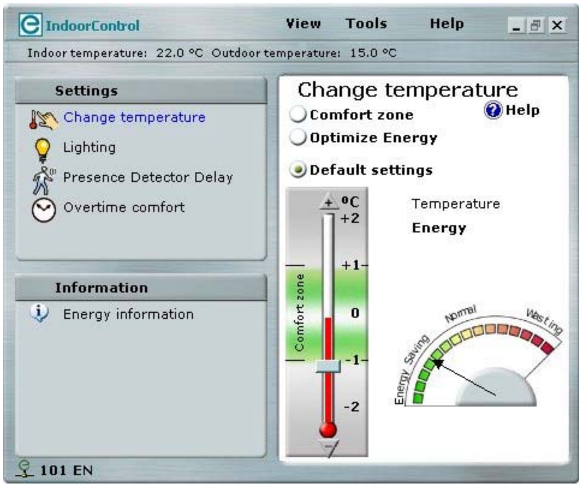

Figure 6. User Interface, providing feedback on comfort and energy.

6. CONCLUSIONS

The developments concerning the built environment indicate that in a 'business as usual' scenario, the energy usage will not decrease although a great potential for reduction exists. This potential can only be achieved by bringing about a transition to a sustainable energy system in the built environment, borne by all responsible parties.

From the Building Future scenario, associated with this transition, it can be deduced that all available measures for improvement should be used. This implies making best practice (such as Passivhaus measures), common practice and developing necessary technologies for essential concepts such as energy producing new residential buildings and factor 4 renovation concepts.

Potentially successful concepts incorporate technologies for building related and user related energy reduction as well as integration of renewable energy technology and intelligent energy management systems.

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Household Energy Consumption under Different Lifestyles

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SUMMARY

This study investigated the impacts of lifestyle in terms of family patterns, life schedules and climate factors upon household energy consumption, as well as to unfold the main causes of household energy consumption under various climatic lifestyles. Based on the findings of the previous studies, it was assumed that space heating, cooling, lighting and entertainment/media equipments usages are the major lifestyle and climate related sources of household energy consumption. Based on these aspects, together with the typical life schedules of different occupation groups, climatic data and typical family patterns, lifestyles related household energy consumptions were calculated for Hokkaido (cool temperate climate), Saitama Prefecture (temperate climate) and Kogoshima Prefecture (varies from temperate to sub-tropical climates) of Japan, for years 2005, 2010, 2015, 2020 and 2025. The results confirmed that regional basis lifestyles in terms of family patterns, life schedules and climate factors are significant in dictating the household energy consumption trends.

INTRODUCTION

The Intergovernmental Panel on Climate Change (IPCC) has predicted a rise in the average global surface temperature of about 2°C between 1990 and 2100, and the average sea level is also projected to rise by about 50cm over the same period [1]. The projected climate changes are expected to result in significant adverse impacts on ecological systems and socio-economic sectors. In view of the continued rise of greenhouse gas (GHG) concentrations in the atmosphere, which may lead to dangerous interference with the climate system, the United Nations Framework Convention on Climate Change (UNFCCC) sets an ultimate objective of stabilizing GHG concentrations in the atmosphere. Under the Kyoto Protocol, the participating developed countries are committed to reduce their GHG emissions on an average of about 5% by the target years of 2008 to 2012 [2].

GHG emission is closely related to energy consumption. Based on the projection by the International Energy Outlook 2006, world carbon dioxide emission from the consumption of fossil fuels is expected to grow at an average rate of 2.1 percent per year from 2003 to 2030 [3]. The world CO_2 emission from the consumption of fossil fuels is predicted to increase from less than 20,000 million metric tons in 2003, to more than 40,000 million metric tons by 2030, as shown in Figure 1. Hence, in handling GHG emission issue, energy consumption should firstly be looked into.

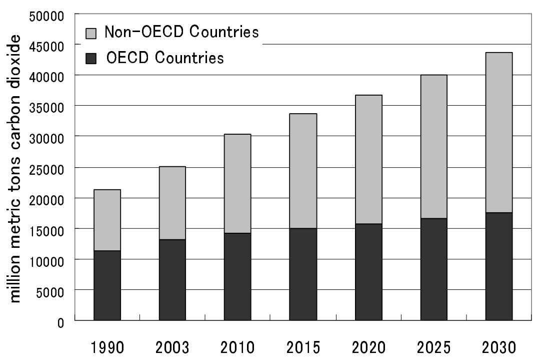


Figure 1. World CO_2 emissions from energy use, 1990-2030 [3]

In response to the above issues, the present authors had previously undertaken a study to investigate the urban energy consumption trends [4]. The said study was based on an integrated approach to investigate the embodied urban energy consumption over a period of 50 years using the System Dynamic model (SD model). The study found that household energy consumption and lifestyle aspects are among the significant factors in dictating the overall urban energy consumptions trend. The current study is a continuous work from the previous investigation and it was focusing on understanding of the interrelations between household energy consumption, lifestyle and climate factors.

In Japan, there are a number of existing researches on household energy consumption and lifestyle. However, little has been done in cogitating about the effects of lifestyle in terms of family patterns and life schedules of family members on household energy consumption. In view of these shortfalls, the purpose of this study is to shed new light on the impacts of lifestyle (in terms of family patterns and life schedules) and climate factors (under different climate zones of Japan) upon household energy consumption.

In most the previous studies on lifestyle and household energy consumption, questionnaire survey and long-term on-site monitoring/measurement methods were commonly used. Despite the better accuracy of data, the disadvantages of these methods are that they are very costly and time consuming. This study thus stressed to estimate the lifestyle related household energy consumption using on the existing life schedule data, which is low cost and time saving, and yet able to provide reasonably accurate estimate of energy consumption. The objectives of the study are to understand the effects of family pattern, life schedule and climate aspects on household energy consumption.

METHODS

The climate of Japan is predominantly temperate, but due to the large north-south extension of the country, the climate varies strongly in different regions. Generally, northern Japan is experiencing cool-temperate climate, central Japan is temperate, while southern Japan is under sub-tropical climate. In order to understand the differences in domestic energy consumption patterns under different climatic characteristics, the case study was divided into three parts according to the climate zones i.e. Hokkaido for cool-temperate zone, Saitama Prefecture for temperate zone, while Kagoshima Prefecture is experiencing climates that varies from temperate to sub-tropical. (cf. Figure 2).

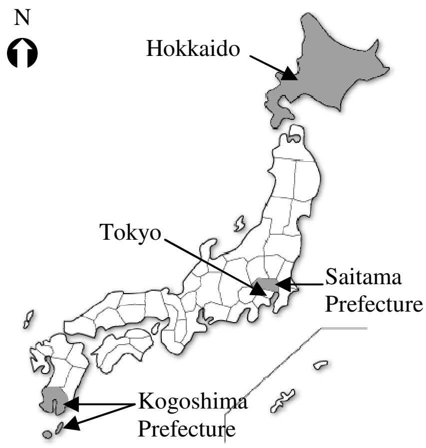
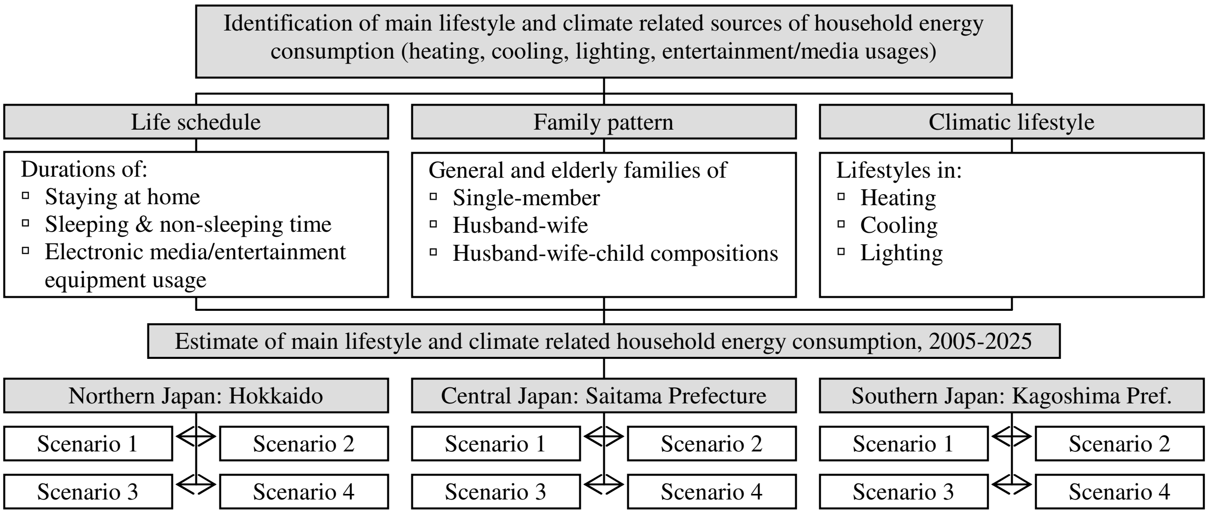


Figure 2. Locations of case study areas

Figure 3 outlines the study methodology of the current investigation. From the literature review, it was pointed out in the previous studies [5,6,7] that space heating, cooling, lighting, electronic entertainment/media equipments and hot water supply are the major sources of household energy consumption. The current study adopted these factors as the major lifestyle and climate related sources of household energy consumption, however, hot water supply was excluded in this study due to absence of life schedule data on this aspect (hereinafter, household energy consumption by space heating, cooling, lighting and entertainment/media equipments shall be referred as ‘lifestyle related household energy consumption’).



Note: Details of Scenarios 1, 2, 3 and 4 are described in Table 1.

Figure 3. Study methodology

As mentioned above, the purpose of this study was to investigate the impacts of life schedule, family pattern and climate factors on household energy consumption. For this reason, detailed analysis had been carried out on these three aspects.

For the life schedules, typical life schedules of people of different occupation groups (working groups, housewives, retirees, students, children below schooling age) were identified based on the life schedule survey by the Japan Broadcasting Corporation in 2005, on 12,600 respondents throughout Japan [8]. In this study, the extracted data include the average durations of time staying at home, sleeping & non-sleeping times and for media/entertainment equipment usages (television, radio, CD/MD/cassette player and video).

Based on the above life schedule data and climatic information (sourced from the Japan Meteorological Agency and National Astronomical Observatory of Japan), as well as other relevant studies [5,7] together with necessary assumptions, typical trends of heating, cooling, lighting and media/entertainment equipment usages under the three climate zones were determined.

For family pattern, based on the projection by the Japan National Institute of Population and Social Security Research, cf. Figure 4, it was predicted that by the year 2025, about 79.5% of the families in Japan would be in the categories of single-member family (i.e. 1 person), husband-wife family (i.e. 2 people) and husband-wife-child family (i.e. 3 people) [9]. Also, it was projected that the proportion of population age 65 and above will increase from 19.9% in 2005 to 24.4% in 2025, and 35.7% by 2050 [10]. Based on this information, this study focused on these three family groups and the elderly family groups for the estimate and analysis of household energy consumptions.

Figure 4. Japanese family patterns [9]

Based on the above data and assumptions, typical lifestyles related household energy consumptions were estimated for Hokkaido, Saitama Prefecture and Kagoshima Prefecture, for years 2005, 2010, 2015, 2020 and 2025. The projections were carried out according to the official household projection data of each region [10]. The assumptions for energy consumption rates were based on the typical energy consumption rates of the relevant household electrical appliances in Japan, as well as reference to the household energy consumption survey carried out by the Architectural Institute of Japan (AIJ) [5].

The projections for lifestyle related household energy consumption were carried out for four scenarios as described in Table 1. Under these projections, it was assumed that there is no change of energy efficiency and no extra energy saving effort as compared to the present trend.

Table 1. Projection scenarios

Scenarios	Descriptions
Scenario 1	Projection based on the existing typical lifestyles and the official population projection statistical data [10].
Scenario 2	Instead of the official household projection data, it was assuming that the proportion of family groups in year 2005 to be remained unchanged over the projection period, while population growth was based on the official projection data [10] as in Scenario 1, and other factors as per Scenario 1.
Scenario 3	Instead of being housewives in the traditional Japanese families, this scenario assumed that the proportion of working women would be increased, while other factors as per Scenario 1.
Scenario 4	It is assuming that the average daily working hours of the working groups would be reduced by 1 hour over the projection period, while other factors as per Scenario 1.

RESULTS

Before presenting the analysis results, it is necessary to describe the trends of population and household growths in the three chosen case study areas. The 2005 populations of Hokkaido, Saitama Prefecture and Kagoshima Prefecture were about 5.7, 6.9 and 1.8 millions respectively. Projection by the Japan National Institute of Population and Social Security Research shows that the population of Hokkaido and Kagoshima Prefecture will gradually drop over the next 20 years, while in Saitama Prefecture, the population will slightly grow until 2015 then decrease gradually [10], cf. Figure 5.

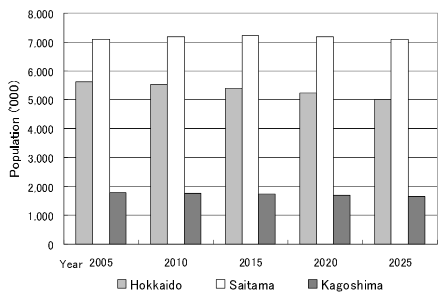
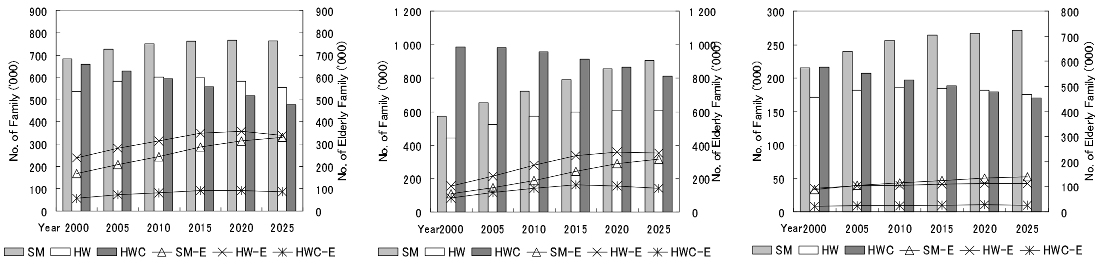


Figure 5. Population projections for Hokkaido, Saitama Prefecture and Kagoshima Prefecture [10]

In the next 20 years time, as shown in Figure 6, the number of single-member families (SM) and husband-wife families (HW) will increase steadily, while the number of husband-wife-child families (HWC) will continue to drop in all these three regions. It must also be noted that the proportion of elderly families are projected to increase significantly, particularly in the single-member family and husband-wife family sectors. Projection shows that the family pattern in Japan is moving towards small family's direction, whereby the single-member families and husband-wife families will constitute more than 50% of the overall Japanese families by the year 2025.

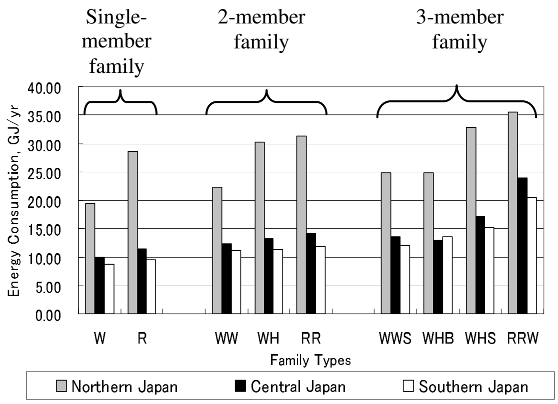


Hokkaido   Saitama Prefecture   Kagoshima Prefecture

SM: single-member family, HW: husband-wife family, HWC: husband-wife-child family, SM-E: elderly single-member family, HW-E: elderly husband-wife family, HWC-E: elderly parent husband-wife-child family

Figure 6. Household growth trends of Hokkaido, Saitama Prefecture and Kagoshima Prefecture, 2000-2025 [10]

Figure 7 illustrates the calculation results of the typical lifestyle related household energy consumption (per household per year) of different family patterns in northern, central and southern Japan. Validation of results has been done by comparison with the residential energy consumption survey conducted by AIJ [5]. This data were used as the basis for the analysis in this study. This figure indicated three energy consumption trends that worth mentioning, (1) households with retirees or housewives generally consume more energy due to longer hours staying at home, (2) northern Japanese households generally consume more energy due to space heating and lighting needs, and (3) the bigger the family, the higher the energy consumption is expected but the per person energy consumption rate is obviously less.



W: working, R: retiree, WW: both working, WH: working husband & housewife, RR: both retirees, WWS: working parent & schooling child, WHB: working husband, housewife & child below schooling age, WHS: working husband, housewife & schooling child, RRW: retiree parents & working child

Figure 7. Typical lifestyle related household energy consumption for each family

Figure 8 presents the calculated results of total lifestyle related household energy consumption for Hokkaido, Saitama Prefecture and Kagoshima Prefecture (for all households of each region) under **Scenario 1**. This scenario serves as the reference case for Scenarios 2, 3 and 4. Despite the expected continuous decrease of population in Hokkaido and Kagoshima Prefecture, the total lifestyle related household energy consumption increases until 2015 and 2020 respectively. For Saitama Prefecture, the population is expected to drop after 2015, cf. Figure 5, but the total lifestyle related household energy consumption will increase until 2020. The increase of energy consumption despite population drop is mainly attributed to the reduction of family sizes and growth of elderly households. The calculated results also show that energy consumption in Hokkaido is higher compared to the southern regions. It is basically due to energy consumption for heating. Similar findings were also showed in the residential energy consumption survey carried out by AIJ [5]. Another reason of higher energy consumption in the northern regions was the energy consumption for lighting. In Hokkaido, generally more than half of a year the sunset time is earlier than 6.00pm, while in Kagoshima it is only about 3 months.

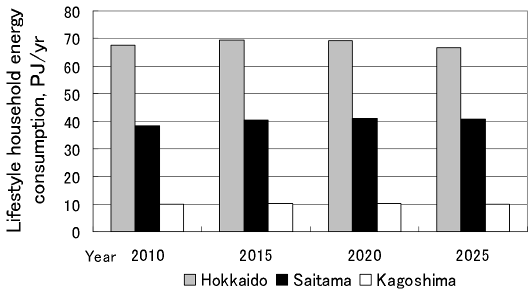
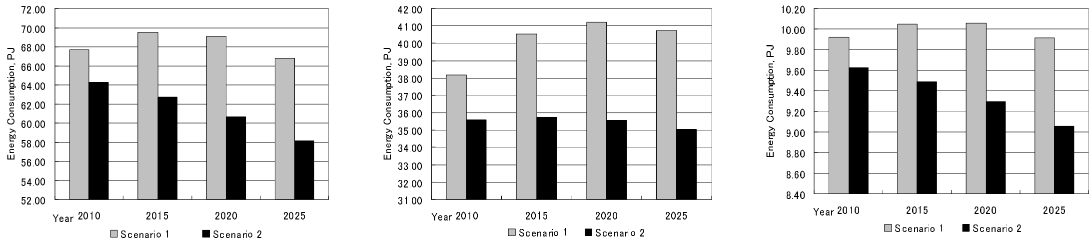


Figure 8. Lifestyle related household energy consumption under Scenario 1 (*based on official household projection data*)

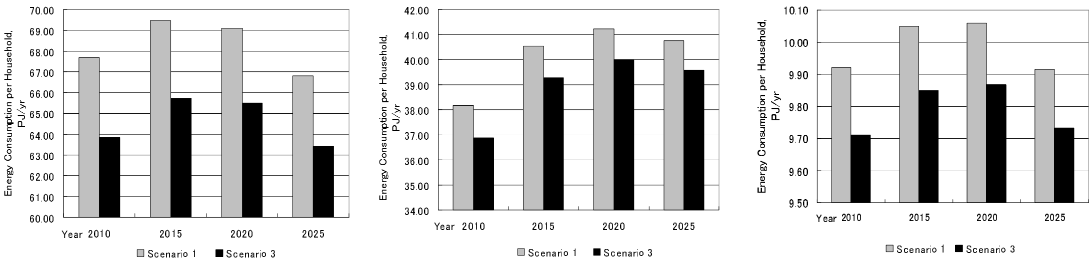
Figure 9 shows the calculated results of **Scenario 2** in comparison to Scenario 1. Under Scenario 2, the energy consumption was calculated based on the assumption that the 2005's family pattern (the proportions of each family group) will remain unchanged over the projection period. It is obvious that the energy consumption would be significantly reduced if there is no change of family pattern. Under the official household projection, the shares of single-member families and husband-wife families will significantly increase in future (cf. Figure 6). The reduction of family size results in higher energy consumption because of higher per-person energy consumption rate and increased number of households/residential units. The result concludes that family size has inverse relation with household energy consumption. In the case of Hokkaido, under Scenario 1, the energy consumption increases until 2015 but under Scenario 2, the energy consumption consistently drops over the projection period. The huge different of energy consumption under Scenarios 1 and 2 in Hokkaido was because of drastic drop of husband-wife-child families and significant increase of the share of single-member families under Scenario 1.



Hokkaido Saitama Prefecture Kagoshima Prefecture

Figure 9. Lifestyle related household energy consumption under Scenario 2
(*assuming that the family pattern of year 2005 remains unchanged*)

It was indicated previously in Figure 7 that households with housewives generally consumed more energy because of longer hours staying at home. **Scenario 3** attempts to validate this statement. Under this scenario, energy consumption was calculated based on increased proportion of working women (reduced proportion of housewives), which results in shorter duration of energy usage for heating, cooling and lighting. As shown in Figure 10, regardless of energy consumption at the workplace, increased proportion of working women will reduce the household energy consumption.



Hokkaido Saitama Prefecture Kagoshima Prefecture

Figure 10. Lifestyle related household energy consumption under Scenario 3
(*increased proportion of working women*)

Scenario 4 calculates the lifestyle related household energy consumption under assumption of reduced working hours. The Japanese Government is presently encouraging people to reduce working hours and spend more time with family at home. This scenario investigates its impact on household energy consumption if the average working hours of the working groups were reduced by one hour per day. The calculated results obtained (cf. Figure 11) shows that reduced working hour will result in increase of household energy consumption. It is because longer hours at home mean spending more household energy, particularly on space heating, cooling and lighting.

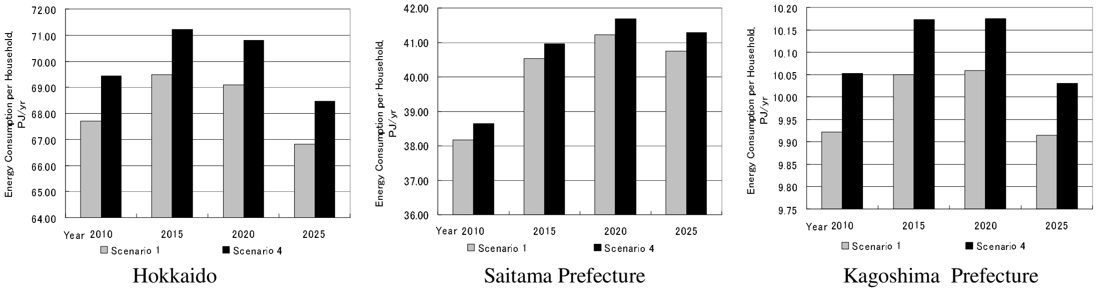


Figure 11. Lifestyle related household energy consumption under Scenario 4
(*reduced working hours*)

DISCUSSION

From the above results, it can be inferred that lifestyles in terms of life schedule and family pattern, and climate factors have significant effect in predominating over the household energy consumption trends.

In most of the previous studies on lifestyle and household energy consumption found in literature, the focuses were put on the energy saving potential by practicing energy saving smart lifestyle in term of the usage of energy consuming household appliances. However, it is difficult to get long-term cooperation from all households. For example, in the survey conducted by AIJ [5], the respondents were not practicing energy saving lifestyles in most of the aspects in household daily life.

Besides the aforementioned lifestyle aspect, there are also some other aspects of lifestyle that have indirect effect on household energy consumption. This study pointed out that lifestyles in terms of life schedule and family pattern do have significant impacts on household energy consumption. The longer time staying at home results in higher energy consumption, this is even more notable in the northern regions where long hours heating and lighting are necessary. Hence, the higher proportions of housewife and retiree population, who tend to spend more time at home, implying higher household energy consumption.

In terms of family pattern, smaller family size results in higher per-person energy consumption. In Japan, the family sizes are getting smaller and smaller (cf. Figure 4), as mentioned earlier, smaller family size is expected to result in higher household energy demand. The situation is worse when the smaller family size coupled with the rapid increase of retired elderly who consumes more household energy due to longer duration at home. Hence, in order to reduce energy consumption for space heating/cooling and lighting, it is deemed necessary to encourage the usage of smaller power space heating/cooling and lighting equipments. Besides improvement of energy efficiency, it can also be achieved by having smaller and enclosed partitions/room in the residential buildings (particularly the living room and dining room), so that same space heating/cooling and lighting effect can be achieved with smaller power equipments. Also, in term of lifestyle, the family members should be encouraged for shared usage of space heating/cooling and lighting by spending more times in the common spaces such as living room, particularly during evening time when most of the family members are at home.

Another aspect of lifestyle that has significant impact on household energy consumption is the climatic lifestyle. This study indicated that households in the northern regions of Japan (cold region) generally consume more energy in comparison with the southern regions (warmer region). As mentioned above, the main reasons are energy requirements for heating and lighting. Thus, as part of the efforts to reduce the energy consumption in the northern regions, it is necessary to reduce the heating and lighting requirement by passive design measures such as introducing solar radiation in winter through windows facing south, double glazing windows, airtight building, direct solar radiation for lighting, etc. for the residential buildings in the cold regions. Besides, it is also important to reduce the energy consumption for heating and lighting by introducing higher energy efficient heating and lighting equipments.

Although this study has effectively pointed out the importance of some aspects of lifestyle in household energy planning, it must also be noted that there are some limitations with this study. Firstly, fixed energy consumption rates were applied for the projections of future energy consumptions. So it may not reflect the actual amount of future energy consumption. Secondly, the household energy consumptions were calculated based on the typical life schedules and energy consumptions of household appliances and climatic data. Thus, it may not represent the actual energy consumption of each household. However, in this study, the main purpose is to understand the energy consumption trend rather than predicting the future energy demand. Hence, the results were deemed sufficient to illustrate the possible future household energy consumption trends. Nevertheless, these limitations will be taken into consideration in the future studies.

This study based on the case of Japan is deemed important as Japan is often used as a model for development in other Asian countries, hence the findings will have implications beyond Japan. Japan is now experiencing social change such as aging society and shrinking family sizes. This study found that these social changes will indirectly result in increased household energy consumptions. It is believed that the existing developing countries will also experience similar social change in future, which will eventually affect the household energy consumption trends. Hence, it is vital to take these aspects into consideration in their long-term energy planning.

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Investigating the Thermodynamic Parameters of the Residential-Commercial Sector: An Application of Turkey

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SUMMARY

The energy utilization of a country can be evaluated using exergy analysis, which is a way to a sustainable development, to gain insights into its efficiency. The authors have conducted various studies on analyzing the energy utilization efficiencies of Turkey and extended here these studies by dealing with the investigation of the thermodynamic parameters in the Turkish residential-commercial sector (TRCS). These parameters are determined for the components of the TRCS in an attempt to assess their individual performances and are also compared to each other, while the analysis is done based on the actual data. The present study has clearly indicated the necessity of the planned studies towards increasing exergy efficiencies in the sector studied and especially the critical role of policymakers in establishing effective energy-efficiency delivery mechanisms throughout the country.

INTRODUCTION

Thermodynamic analysis including exergy analysis method is a powerful tool, which has been successfully and effectively used for estimating energy utilization efficiencies of countries. Based on the earlier studies conducted on the sectoral energy and exergy analysis of countries by many authors, the approaches used to perform the exergy analyses of countries may be grouped into three types, first two approaches; namely Reistad's approach and Wall's approach, as denoted by Ertesvag [1] and the last one Scuiba's approach [2]. The authors reviewed in detail analyzing and evaluating the energy and exergy utilization of countries [3] .

The concepts of exergy, available energy, and availability are essentially similar. The concepts of exergy destruction, exergy consumption, irreversibility, and lost work are also essentially similar. Exergy is also a measure of the maximum useful work that can be done by a system interacting with an environment which is at a constant pressure P_0 and a temperature T_0 [4,5].

Dincer et al. [6] reported that, to provide an efficient and effective use of fuels, it is essential to consider the quality and quantity of the energy used to achieve a given objective. In this regard, the first law of thermodynamics deals with the quantity of energy and asserts that energy cannot be created or destroyed, whereas the second law of thermodynamics deals with the quality of energy, i.e., it is concerned with the quality of energy to cause change, degradation of energy during a process, entropy generation and the lost opportunities to do work. More specifically, the first law of thermodynamics is concerned only with the magnitude of energy with no regard to its quality; on the other hand, the second law of thermodynamics asserts that energy has quality as well as quantity. By quality, it means the ability or work potential of a certain energy source having certain amount of energy to cause change, i.e., the amount of energy which can be extracted as useful work which is termed as exergy. First and second law efficiencies are often called energy and exergy efficiencies, respectively. It is expected that exergy efficiencies are usually lower than the energy efficiencies, because the irreversibilities of the process destroy some of the input exergy.

Exergy is the expression for loss of available energy due to the creation of entropy in irreversible systems or processes [6]. The exergy loss in a system or component is determined by multiplying the absolute temperature of the surroundings by the entropy increase. Entropy is the ratio of the heat absorbed by a substance to the absolute temperature at which it was added. While energy is conserved, exergy is accumulated

Exergy analysis provides a method to evaluate the maximum work extractable from a substance relative to a reference state (i.e., dead state). This reference state is arbitrary, but for terrestrial energy conversion the concept of exergy is most effective if it is chosen to reflect the environment on the surface of the Earth [8].

The main objective of the present study is to investigate the effect of thermodynamic parameters on the energy utilization efficiencies of the RCS by giving an application of Turkey. In this regard, thermodynamic relations used to perform energy and exergy analyses under varying reference (dead) state temperatures are given first. The relations for the RCS and its subsectors, such as water heating (WH), spaces heating (SH), cooking and electrical appliances (EA), are then presented. Finally, these relations are applied to the TRCS, while the results obtained are discussed.

METHODS

Thermodynamic parameters

A key discussion of the relevant theory of thermodynamic parameters for energy and exergy analyses is stated in this section. At the restricted dead state, the fixed quantity of matter under consideration is imagined to be sealed in an envelope impervious to mass flow, at zero velocity and elevation relative to coordinates in the environment, and at the temperature T_0 and pressure P_0 taken often as 25°C and 1 atm, respectively [5]. Some of the key aspects of thermodynamics in terms of energy and exergy used in the modeling are taken in Table 1 [9,10].

Energy and exergy efficiency calculations in residential-commercial sector

Residential-commercial sector includes space heating (SH), water heating (WH), cooking and electrical appliances (EA). In all activities in this sector, based on electricity, heat is produced. Quality of heat is Carnot factor that is strongly depend on temperature. Quality factors for some energy carriers and forms are 1.00, 0.60, 0.2-0.3,0-0.2, 0 for mechanical energy, steam (600°C),district heating (90°C), space heating (20°C), and earth respectively [11,12]. Energy efficiency is simple comparison of energy content of input and output energy carrier or flow. However, energy efficiency is defined as a function of requirement temperatures in system. Energy and exergy efficiencies values of these categories are determined separately as follows:

Space Heating: Heating options specified for heating systems are district heating, central heating, individual heating and stove. Firstly energy and exergy efficiencies of each other

energy carriers considered for all systems efficiencies are determined than heating system and fuel preference of dwelling units are determined according to their utilization ratios.
43-52% of all direct fuel use was for space heating, which is done entirely by home heaters having the same first law efficiencies of home stoves. Energy and exergy efficiency values of these categories are determined separately as presented above. Energy and exergy efficiencies of each other energy carriers considered for all systems efficiencies are determined and heating system and fuel preference of dwelling units are determined according to their utilization ratios. Quality factors of energy carriers are obtained from ref [13]. Reference state temperatures are from 0°C (273K) to 25°C (298K), SH here needs a temperature of 50°C (323K) Achieved energy and exergy efficiency values are illustrated in Table 2.

Table 1. Some of the key aspects of thermodynamics in terms of energy and exergy used in the modeling [9,10].

Process	Energy efficiency	Exergy efficiency
System Balance	$\dot{Q} + \sum \dot{m}_{in} h_{in} = \dot{W} + \sum \dot{m}_{out} h_{out}$	$\sum_{in} m_{in} ex_{in} - \sum_{out} m_{out} ex_{out} + \sum_{r} Ex^{Q} - Ex^{W} - I = 0$
Specific exergy		$ex^{PH} = (h - h_0) - T_0 (s - s_0)$
General Efficiency	*ε_1=(Energy in products / Total energy input) 100*	*ε_2=(Exergy in products/ Total exergy input) 100*
Electrical heating	$\varepsilon_{1e,h} = Q_p / W_e$	$\varepsilon_{2e,h} = Ex^{Qp} / Ex^{We}$ $\varepsilon_{2e,h} = [\,1 - (T_0 / T_p)]\, Q_p / W_e$ $\varepsilon_{2e,h} = [(\,1 - (T_0 / T_p)]\, \varepsilon_{1e,h}$
Fuel heating	$\varepsilon_{1f,h} = Q_p / m_f H_f$	$\varepsilon_{2f,h} = Ex^{Qp} / m_f \varepsilon_f$ $\varepsilon_{2f,h} = [(\,1 - (T_0/T_p)]\, Q_p / (m_f \gamma_f H_f)$ $\varepsilon_{2f,h} = [\,1 - (T_0/T_p)]\, \varepsilon_{1f,h}$
Electrical cooling	$\varepsilon_{1c,e} = Q_p / W_e$	$\varepsilon_{2c,eh} = Ex^{Qp} / Ex^{We}$ $\varepsilon_{2c,eh} = [(\,1 - (T_0 / T_p)] Q_p / W_e$ $\varepsilon_{2c,eh} = [\,1 - (T_0 / T_p)]\, \varepsilon_{1c,eh}$
Shaft work production via electricity	$\varepsilon_{1e,w} = W / W_e$	$\varepsilon_{2e,w} = Ex^{W} / Ex^{We} = W / W_e = \varepsilon_{1e,w}$
Shaft work production via fuel	$\varepsilon_{1f,w} = W / m_f H_f$	$\varepsilon_{2f,w} = Ex^{W} / m_f H_f = W / (m_f \gamma_f H_f) = \varepsilon_{1f,w} / \gamma_f$
Fuel driven kinetic energy production	$\varepsilon_{1f,ke} = m_s \Delta ke_s / m_f H_f$	$\varepsilon_{2f,ke} = m_s \Delta ke_s / m_f \varepsilon_f = m_s \Delta ke_s / (m_f \gamma_f H_f) = \varepsilon_{1f,ke} / \gamma_{f,ke}$
Improvement potential	$IP = (1 - \varepsilon_2)(Ex_{in} - Ex_{out})$	
Carnot factor	$Q_{carnot} = 1 - (\,T_0 / \,Tp\,)$	
Space/Water heating	$\varepsilon_2 = \left(\frac{\varepsilon_1}{q_{fuel}}\right)\left\{1 - \left[\frac{T_o}{T_2 - T_0}\right] \ln\left(\frac{T_2}{T_0}\right)\right\}$	
Overall SH,WH,EA,	$\varepsilon_{1osh} = [(a_1 * \varepsilon_{1,1c}) + (a_2 * \varepsilon_{1,2c}) + (a_3 * \varepsilon_{1,3c}) + \ldots\ldots.. + (\,a_9 * \varepsilon_{1,9c})]/100$	
Overall SH,WH,EA,	$\varepsilon_{2osh} = [(a_1 * \varepsilon_{2,1c}) + (a_2 * \varepsilon_{2,2c}) + a_3 * \varepsilon_{2,3c}) + \ldots.. + (\,a_9 * \varepsilon_{2,9c})]/100$	
Overall fuel	$\varepsilon_{1of} = [(f_{sh} * \varepsilon_{1osh}) + (f_{wh} * \varepsilon_{1owh}) + (f_c * \varepsilon_{1oc}]/100$	$\varepsilon_{2of} = [(f_{sh} * \varepsilon_{2osh}) + (f_{wh} * \varepsilon_{2owh}) + (f_c * \varepsilon_{2oc})]/\,100$
Cooking/ Cooling	$\varepsilon_2 = \varepsilon_1 \left[1 - \left(\frac{T_0}{T_2}\right)\right]$	$\varepsilon_2 = \varepsilon_1 \left[\left(\frac{T_o}{T_3}\right) - 1\right]$
All sector	$\varepsilon_{1,orc} = \frac{(\varepsilon_{1e} * e_{rc} + \varepsilon_{1of} * f_{erc})}{(e_{rc} + f_{erc})}$	$\varepsilon_{2,orc} = \frac{(\varepsilon_{2e} * e_{rc} + \varepsilon_{2of} * f_{exrc})}{(e_{rc} + f_{exrc})}$

Water Heating: In WH activities, various energy carriers are used that are natural gas, LPG, wood, dried dung, solar and electricity. First law efficiencies of energy carriers in water heating are determined by producer using temperatures 25^{o}C (298) and 60^{o}C (333K) for instance electric water resistant energy efficiencies is 98%, home gas water heater 60% Energy efficiencies of direct fuel use for water heating are assumed to be 27-80% [13,14]**.** We obtained the figures ranging from 3.10% to 10.80% for all energy carriers, as indicated in Table 2.

Cooking: In cooking activities, various energy carriers are used that are natural gas, city gas, LPG, electricity, wood and dried dung. Energy efficiencies used in the previous study, dead state (ambient) and process temperatures used in calculations are for cooking applications 20^{0}C and 120^{0}C respectively cooking efficiencies are assumed to be 50% for gas cooking stove, LPG, natural gas, 80% for electric cooking stove, 22% for wood, 20% for dried dung. These assumptions yield second law efficiencies ranging from 4.1% to17.2% for fuel use over the years studied, as indicated in Table 2.

Table 2. Energy and exergy efficiencies of residential-commercial sector at varying reference state temperatures [13]

		Temperatures (^{o}C)						Temperatures (^{o}C)					
	Energy carriers	0	5	10	15	20	25	0	5	10	15	20	25
		Energy efficiencies (%)						Exergy efficiencies (%)					
Space heating	Coal stove	90	81	72	63	54	45	7.14	5.75	4.52	3.44	2.51	1.74
	Coal district	84.21	77.19	70.18	63.16	56.1	49.12	7.85	6.56	5.39	4.34	3.41	2.60
	Fuel-oil	100	90	80	70	60	50	8.26	6.65	5.23	3.98	2.9	2.01
	Natural gas	99	96.95	92.9	86.54	80	70	10.33	9.22	7.99	6.66	5.44	4.15
	Wood	80	72	64	56	48	40	6.35	5.11	4.02	3.06	2.23	1.55
	Electricity	100	95	90	85	80	75	13.55	12.09	10.72	9.44	8.24	7.14
	Geothermal	100	88.89	77.78	66.67	55.6	44.44	29.03	22.45	16.74	11.89	7.88	4.71
	LPG	99	96.95	94.74	92.1	86.7	75.83	9.6	8.57	7.57	6.59	5.48	4.18
Water heating	Natural gas	100	91.67	83.33	75	66.7	58.33	10.44	8.72	7.17	5.78	4.54	3.46
	Wood	100	86.12	72.22	58.33	44.4	30.56	9.33	7.32	5.55	4.01	2.7	1.62
	LPG	100	93.24	85.47	77.7	69.9	62.16	9.67	8.16	6.77	5.51	4.38	3.39
	Electricity	100	96.25	94.17	92.25	89.3	79.33	14.77	13.43	12.38	11.39	10.3	8.56
	Fuel-oil	100	91.67	83.34	75	66.7	79.33	9.6	8.11	6.66	5.37	4.21	3.21
	Geothermal	100	87.5	75	62.5	50	37.5	23.03	17.53	12.8	8.85	5.63	3.15
	Solar	100	91.67	83.34	75	66.7	58.33	10.3	8.61	7.07	5.70	4.47	3.41
Cooking	LPG	51.43	48.91	46.51	44.23	42.1	40	15.86	14.45	13.14	11.93	10.8	9.77
	Natural gas	51.43	48.91	46.52	44.23	42.1	40	17.62	15.55	14.14	12.84	11.6	10.51
	Wood	29.64	28.40	27.17	25.93	24.7	23.95	7.99	7.23	6.56	5.91	5.29	4.72
	Electricity	94.95	90.29	85.87	81.66	77.7	73.85	28.98	26.41	24.02	21.81	19.8	17.84
Electrical appliances	Refrigeration	1.03	1.05	1.07	1.09	1.11	1.13	3.02	4.91	6.79	8.68	10.6	12.45
	Air condition	1.98	2	2.02	2.024	2.05	2.07	-3.46	0	3.53	7.111	10.8	14.47
	Clothes Dry	100	93.34	86.67	80	73.3	66.67	21.54	18.77	16.18	13.79	11.6	9.58
	Dishes Machine	100	90	80	70	60	50	14.71	13.24	11.76	10.29	8.82	7.36
	Iron	100	96.67	93.34	90	86.7	83.33	34.75	33.59	32.43	31.28	30.1	28.96

Electrical Appliances: Electrical appliances includes, lighting, refrigerator, air condition, dish washer, washing machine, iron, television, computer and others appliances, as hair dryer, mechanical drive. In RCS, electricity is primarily consumed for refrigeration, lighting, air conditioning and other activities. For electrical appliances, input exergy is equal to input energy. However, in refrigeration and air conditioning applications, purpose is extracting heat instead of producing heat. It is assumed that the temperatures inside freezers and refrigerators are approximately –8^{o}C, the room temperature near the refrigerator coil is 20^{o} C and the coefficient of performance (COP) is calculated. It is assumed that the temperatures for air conditioning 33^{o}C and13^{o}C the COP is 2.0. Approximately 35-38% of all electrical use was for lighting. Lighting is assumed to be 80% incandescent and 20% fluorescent with first and second law efficiencies of about 5% and 4.5%, and 20% and 18.5%, respectively [13]. Combining the relevant first and second law efficiencies for lighting, we calculated ε_1 =9.5-15.5% and ε_2 = 8.70-14.3% for the years considered. 6-7% of all electrical use was for television and computer. First and second law efficiencies are assumed to be 80%. Dish washer and iron are heat-producing appliances that are evaluated using the relations given in Table 1. Mechanical drive requirements in the RCS are covered by small electric motors, while both energy and exergy efficiencies are about 70%. Electricity consumption values of other electrical appliances, for instance, computer and hair drying are estimated as given in Table 3, while the first and second law efficiencies of these appliances are listed in Table 2. In this sector, overall first and second law efficiencies for the entire RCS are calculated by aggregating both purchased electrical energy and direct fuel use. Using the numerical values obtained using in these models, the weighted mean overall energy and exergy efficiencies for the entire RCS were found.

RESULTS AND DISCUSSION

An application of the Turkish residential and commercial sector

The present application analyzes of the energy and exergy uses of the TRCS are based on thermodynamic parameters of process. This analysis is also based on the actual data for 2004, and it is used in the calculations. Total energy and exergy inputs for the same period according to energy carriers are obtained from Ref [14]. By 2004, Turkey's population and dwelling unit are determined to be 71332000 and 16487640, respectively. Total energy and exergy inputs to the Turkish sector were 3758.23 and 3710.24 PJ in 2004, respectively. In addition per/capita for total energy consumption and for RCS were determined to be 48.08 GJ/capita and 10.78 GJ/capita in studied year. In 2004, of Turkey's total end-use energy, 42% was used by the industrial sector, followed by the residential-commercial sector at 30%, the transportation sector at 20%, the agricultural sector at 5%, and the non energy (out of energy) use at 3% [15].

Energy and exergy utilization in the Turkish residential-commercial sector

The highest contributions came from fuel with 38.45 %, renewable resources (includes wood) with 38.20% and electric with 23.31% in 2004. In 2004, the highest contributions came from wood with 192.20 PJ [16]. However, natural gas usage has continuously increased in the TRCS for space heating, water heating and cooking purposes in several cities. Share of the energy utilization in the residential-commercial modes is as follows: space heating with 41%, water heating with 25%, cooking with 11% and electrical appliances with 23% in the year studied. These values are determined for 2004 obtained from Refs. [16,17].

Estimation of component and overall efficiencies for SH ,WH and cooking

Various fuels given in Table 3 are used for the purposes of WH, SH and cooking activities in this sector. Energy utilization values for the TRCS are also indicated in Table 3. SH requires the largest fraction of fuel with about 44%, while WH and cooking are responsible for 35%, and 21% of the total fuel inputs in this year, respectively. Based on the values obtained from Turkey's population census, the fuel preferences of dwelling units in the year of 1998 in Turkey [14]. The first and second law efficiencies of space heating were calculated as given below: 44% of all direct fuel use was for space heating, The needed temperature for the SH equipment is 50^{o}C and the ambient temperature is from 25^{o}C to 0^{o}C. Heating system and fuel preferences must be considered for calculating. Those preferences are given in Table 3 for the year of 2004. Overall energy and exergy efficiencies of SH were calculated as follows: Substituting the relevant numerical values into equations at state temperatures from 25^{o}C to 0^{o}C, we obtained ε_{1sh}= 48.74% to 90.02%, $\varepsilon_2$sh= 2.25 % to 7.18% in 2004 for SH.

Table 3. Distribution of residences according to fuel types and components of fuel uses and values for energy and exergy utilization of electric and saturation values of EA in 2004 (%).

Space heating					Water heating	Cookin g	Electrical appliances		
	Ratio of residence				Ratio of residenc	Ratio of residenc		Ratio of saturatio	Utilization of electric
Energy carriers	District	Central	Individua	Stove			Component	(%)	(%)
Coal (stove)				72.5			Lighting	100	36
Coal	58	39					*(Incandescent)*	*75*	*75*
Fuel-oil	21	24	6	0.8	2.04		*(Fluorescent)*	*25*	*25*
Natural gas	18	37	94	3.9	2.10	9.5	Refrigeration	99	42
LPG				1	42.5	90	Washing machine	87.1	3
Electricity				3.7	7.8	0.3	Dishes machine	32.5	1
Wood				18	32	0.1	Vacuum cleaner	89.4	2
Geothermal	3				0.3		Air conditioning	2.5	3
Solar					13.16		Television+Computer	99	8
Dried dung				0.1	0.1	0.1	Iron	96	1
Total	**100**	**100**	**100**	**100**	**100**	**100**	Clothes drying	82	1
Ratio of	**4**	**6**	**6**	**84**	**100**	**100**	Other	3.2	3
							Overall electrical utilization		100

[a]The values for 2004 are obtained from Ref.[13].

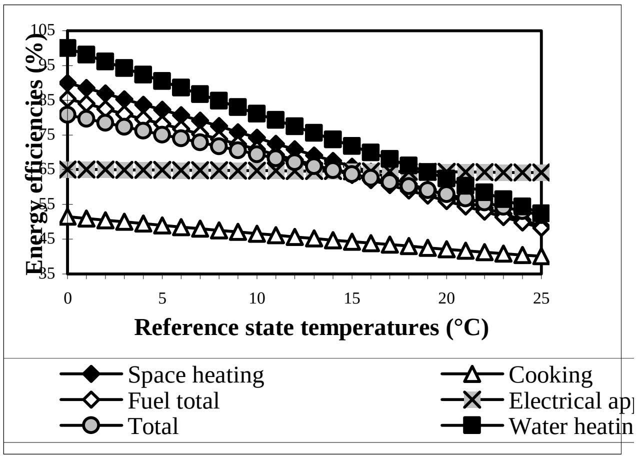
Based on the values achieved from Turkey's population census, the fuel preferences of dwelling units for wafer heating were determined for each province in 2004, 35% of all energy carrier use was for WH. The efficiencies of WH is calculated below. Fuel preferences must be considered for calculating that are given in Table 3 for the year of 2004. Overall first and second law efficiencies of WH are calculated; Using Eqs. and the numerical values assumed, we found $\varepsilon_{1wh,f}$= 52.43% and 100%,and $\varepsilon_{2wh,f}$=3.22% and10.10% in 2004 for WH. Overall first and second law efficiencies are found from 9.85% to16.05% and from 40.06% to 51.51% at reference state temperature changes for cooking in 2004, respectively.

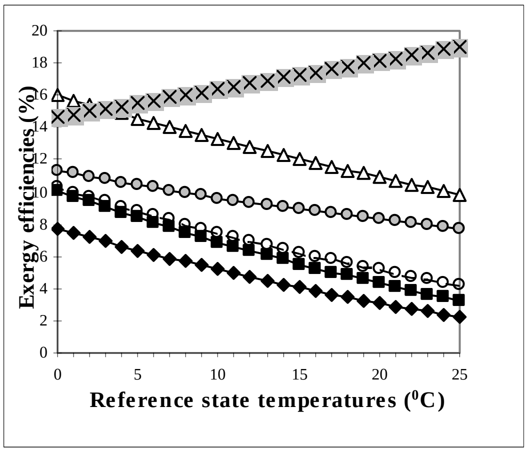
Estimation of overall efficiency and effectiveness values for electric utilization

Energy utilization values and the saturation values of EA for the TRCS are indicated in Table 3. Refrigeration requires the largest fraction of electricity with 42% in the year studied, followed by lighting with 36%. The overall efficiency and effectiveness values for electric utilization are estimated as follows: Lighting is assumed to be 75% incandescent and 25% fluorescent with first and second law efficiencies of about 5% and 4.5%, and 20% and 18.5%, in 2004, respectively [12,13]. Combining the relevant first and second law efficiencies for lighting, we calculated $\varepsilon_{1,l}$ =8.75-15.5% and $\varepsilon_{2,l}$ = 8.0-14.3% for the year considered. Refrigerators are consumed huge share of electricity. 40% of all electrical use was for refrigeration [13]. It is assumed that the temperatures inside freezers and refrigerators are approximately –8^o C, the coefficient of performance (COP) is 1.0 and the room temperature near the refrigerator coil is from 25 to 0^{o}C. Using equations, these assumptions yield second law efficiencies from 3.44 to 12.44% in 2004. Assuming that the COP value of the electric air conditioning unit is 2, this unit extracts heat from air at 14^{o}C and the outside temperature is from 33^{o}C to 8^{o}C, we found $\varepsilon_{2,a}$=14.46 and -3.46% in the year studied. 8% of all electrical use was for television and computer. Annual electricity consumption of television and computer has increased compared to previous years due to an increase in the number of TV channels, and usage of computer spread in daily life as at school and work in Turkey. Energy and exergy efficiencies are assumed to be 75%. Dish washer, iron, are heat-producing appliances that Using Eqs. and the numerical values assumed, we found first and second law efficiencies ε_{1dw}= 50% and 83%, and ε_{2dw}= 7.35% and 28.95% for 25^0C,and ε_{1i}= 100% and 100%, and ε_{2i}= 14.70% and 34.75% for 0^0C in the year studied for dishes washing and iron, respectively. Mechanical drive requirements in RCS meet to by small electric motors and both energy and exergy efficiencies of about 70%. We found energy efficiencies from 64.19 to 65.09% and exergy efficiencies from 14.65% and 19.02% for electrical use in 2004. This changes of efficiencies are illustrated in Figures1 a and b.

Energy and exergy utilization efficiencies in the entire Turkish RCS

Overall, first and second law efficiencies for the entire Turkish RCS were calculated by aggregating both purchased electrical energy and direct fuel use using equations. The weighted mean overall energy utilization efficiencies for the entire residential-commercial sector were found to vary from 51.89% to 80.75% in 2004 for reference state temperatures changes. There is major differently energy efficiencies for reference state temperatures changes in TRCS as shown in Figures 1a and 1b. In terms of energy loses this sector rank rather differently for about 19-38%. Overall exergy efficiencies in the year studied for the entire RCS were found to range from 7.74% to 11.16% in 2004 for reference state temperatures changes. This sector shows considerably important and comparable losses of energy and exergy. In the regard of exergy loses, this sector grades very differently, accounting for about 88-92% of all exergy loses.





a) Energy b) Exergy

Figure 1. Energy (a) and exergy (b) efficiencies of residential-commercial sector in Turkey at varying reference state temperatures for the year of 2004

CONCLUSIONS

In this study, we investigated the thermodynamic parameters of energy and exergy utilization in the RCS. The main results derived from the present study may be summarized as follows:

a) The determination of energy utilization efficiencies is needed for identifying energy efficiency and/or energy conservation opportunities, as well as for dictating the right energy and exergy management strategies of a country.
b) Changes in the reference state temperatures do not have an important impact on exergy efficiencies. Besides, these changes are displayed considerably effect on energy

efficiencies because of energy efficiencies are defined as a function of requirement temperatures in system.

c) A case study included the analysis of the energy and exergy utilization of the TRCS in the year of 2004. This analysis is done based on the actual data for the year of 2004. The energy efficiency value for the TRCS is found to be from 51.89% to 80.75% and the exergy efficiency value for that is obtained to be 7.74% to 11.16% for reference state temperature changes in 2004.

d) This study indicated that exergy utilization in Turkey was even worse than energy utilization. Turkey represents a big potential for increasing the exergy efficiency. It is clear that a conscious and planned effort is needed to improve exergy utilization in Turkey.

As a conclusion, the authors expect that the analyses reported here will provide the investigators with knowledge about how effective and efficient a country uses its energy resources.

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Energy consumption vs. Energy performance?

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SUMMARY

The discussions around the implementation of the Energy Performance of Buildings Directive (EPBD) [1] – not only in national legislation but also in real practice – have revealed open questions and caused different interpretations. This paper deals with one question, and is based mainly on discussions and experiences in Finland.

Studies to prepare background material for energy calculation guidelines have been done, and several proposals for national legislation on energy performance certificates have been prepared. Proposals have been on review, with different levels of ambition, and with split views expressed in replies. The approach described here is completely independent of this legal process. It could, however, contribute as a complementary tool beyond the regulatory level.

The core of our approach is: make a clear difference between "energy performance" and "energy consumption". If this difference is vague, then a common misunderstanding of "lower energy consumption automatically means better energy performance" will stay and make an obstacle in all efforts for improved indoor environment and for more effective use of the building. The latter is valid for some building types, especially in public buildings like schools and sports facilities. Complementary tools are still to be developed to describe all elements of energy performance in parallel – suggestions for these will be discussed in the paper.

INTRODUCTION

The aim of this paper is to contribute in the "EPBD discussion" from the "Well Being Indoors" point of view: how to achieve good indoor environment with low energy consumption? This point of view is necessary, because otherwise "energy performance" is understood as "minimum energy consumption", ignoring the demand for good indoor environment. In the near future, more and more efforts in Europe and probably also globally will be put on reducing emissions and energy use, and this may become a real threat to our buildings and their indoor environment.

However, there are means already available to put together the "contradictory" demands for good indoor environment and low energy consumption. In other words, we should really aim at ENERGY PERFORMANCE, which takes into account the indoor environment and the actual usage profile of the building.

In 2003 the European Commission issued a new directive on Energy Performance of Buildings, the **EPBD**, 2002/91/EC [1]. This directive required that all member countries by January 2006 should have implement the directive in national building legislation. By today, delays have been occurred in a number of member countries because of several reasons.

If we go back a little bit, concern has been expressed in several countries about the costs of the obligatory measures (certificates and inspections) and about lack of competent persons to perform these measures. The progress in Finland has brought up also a tendency towards simplified certificates especially for existing buildings, based on the measured consumption only. This actually does not give an "energy performance certificate". For new buildings, the certificate will be based on calculations and is much up to the reliability of the calculation method and the assumptions made in the input data.

There are two major disadvantages in the "energy consumption certificate":

-It does not properly take into account the indoor environment. Closing off the ventilation system is rewarded; aims for high-level indoor environment are punished

-it does not properly take into account the real usage profile of the building. Really efficient use of the building (e.g. using a school building for various activities during evenings and weekends) is punished, keeping the building unoccupied for exceptionally long periods are easily rewarded

Energy performance is a very complicated issue, and therefore it would take probably many years before the methods and practices are established in Member States, and even more time before they become harmonised throughout Europe. On the other hand, there is a need to start the implementation process step by step, to keep the first steps rather simple in order that the process towards a more comprehensive implementation (also in practice) goes towards the right direction from the very beginning. In this way, the "simple" obligatory measures can also provide a good basis – and also encouragement - for complementary voluntary measures.

One of the practically open questions is: how to estimate the cooling load and energy? This may sound a minor issue regarding the overall energy performance. If the cooling loads and summertime temperatures are calculated by a simple monthly basis, this may result in serious underestimation of the loads. As a consequence, the client makes a decision to skip air-conditioning, or to have big windows, and when the building is completed and during the first warm summer days the indoor temperature goes up far over 30 ºC. Then it is too late to keep the temperatures in control by any reasonable combination of architectural means and air-conditioning. This has happened in reality, and this problem is increasing also in (well-insulated) residential buildings! And either the occupants will suffer and lose their productivity further, or the new building will be subject to early and expensive renovation.

To overcome contradictions in efforts for good energy performance and for good indoor environment (IEQ), a methodology is being developed on voluntary basis for guidance on technologies and design for simultaneous optimisation of energy performance and IEQ. If, like is in the case of many household appliances, everyone wants the highest energy performance, this is of course a good objective but what will happen if other key elements of a good building – architecture, and especially **indoor environment** - have to be sacrificed in the name of energy performance. **Figure 1** gives one possible approach [2] to have these aspects in parallel in design, and this methodology can be extended further to include also the regular inspections – using the same system and product data as the basic reference.

The EPBD gives a clear justification to consider IEQ in parallel with energy performance. However, no further guidance is given how to do this in real practice.

THE METHODOLOGY

Main principles and contents

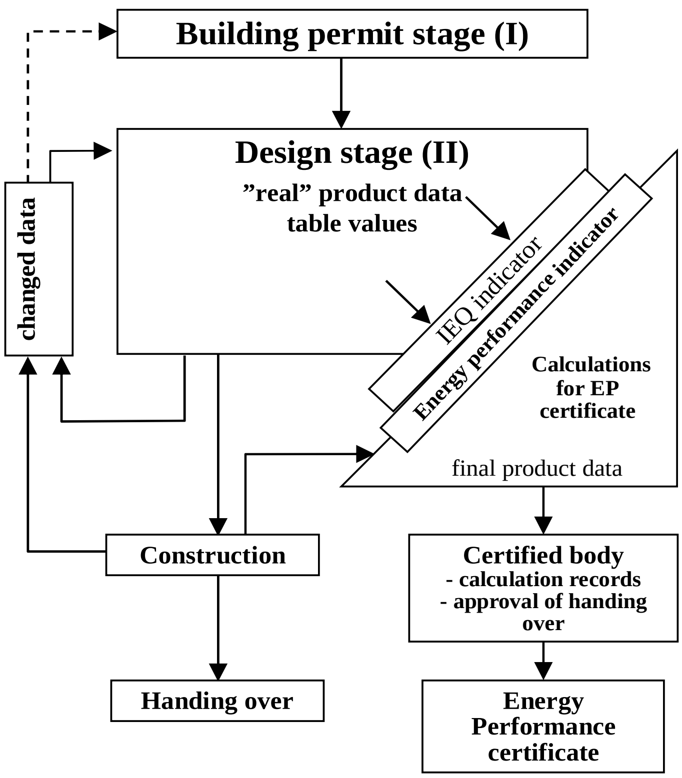
The main principles of the methodology are:
-the energy performance shall be calculated using the data that is available for the building and its building services systems and equipment.
-if, at the early design stage, there is not sufficient data available, default values shall be used for calculations.
-the calculations are revised as soon as exact data is available

Because this methodology was originally developed for other applications, the first stage to develop the procedures was to adjust the principles to fit in the EPBD implementation purposes. Testing of the methodology will take place in real practice.

The key issue is that the TARGET LEVELS for the indoor environment, as well as for energy performance, is defined in the very beginning of the design process. These target levels have to be defined in measurable quantities (indoor temperature summer – winter, energy consumption per surface area, etc.). Because in early design stages (building permit stage) the construction details and HVAC systems are not usually decided, the calculations have to be made first using default values and later on checked with real system and product data. This data should be also used as the basic data for energy performance certificate and for inspection of heating and air conditioning system. One main objective for further development of the methodology is to link together these three main elements of the EPBD.

Figure 1 presents the main flowchart of the methodology. The details will of course be different in different countries depending on how the design process proceeds, and also on the type of building. For example in residential buildings the design flow can be simplified a lot, and data from other similar building projects can be more used as a basis. In complex buildings, the process may become more complicated and contain more backcouplings.

The target level for energy performance could be specified as one figure like "xyz kWh/m^{2}a" or as an energy performance class e.g. according to prEN 15217 [3].



STAGE I

-"Basic" architectural and structural design

-Define TARGET LEVELS for

--Energy Performance (e.g. "A")

--Indoor Environment (e.g. "S1")

-verify by calculation that the legal requirements are fulfilled at standardised loads and operation

STAGE II

Design/Energy performance certificate

- the energy performance, based on standard usage – shall be calculated for the real building and its selected building services systems, and for defined Indoor Environment class

-the Energy Performance class shall be at least as good as targeted and calculated at stage I. If not, then backcoupling to STAGE I

-**Energy performance certificate - calculate** using the final product data (use of certified products should be encouraged and rewarded)

-the real energy consumption, based on the actual use of the building, can be calculated separately, and it may deviate from the consumption estimated for the certificate ("standard consumption")

Figure 1. Principles of design methodology for good energy performance and good indoor environment.

The same system and product data can provide the basis of inspections (obligatory based on Articles 8 and 9 of the EPBD, or voluntary), too. It shall be also pointed out that Figure 1 really concentrates on the elements of design process, just giving a general connection to energy performance certificate. It should be completed with blocks for inspections.

The methodology applies also for existing buildings, but in a different way. In existing building, subject to major renovation, sale or rent, an energy performance certificate is needed. In most existing buildings the data needed for the certificate is far from sufficient. Maybe even the original documentation is incomplete, or changes in the building envelope or building services systems are not completely available. For existing buildings it is very important to encourage the owner/ end user to improve the energy performance. For example it could be stated that in order to be certified to one of the highest categories, certain elements for the indoor environment have to be validated in one way or another (e.g. measured total ventilation air flow rate, recent balancing of ventilation, plus proper documentation). If the owner/ end user has already taken care of these issues by himself, then no more validation is needed. Of course, simple measures are often not sufficient (especially in complex buildings), but in many cases these are the necessary start – and a good wake-up and helping the user to pay attention to both IEQ and energy issues in a proper and balanced way.

Key elements

The stages of the methodology consist of the following calculations. This list does not describe the construction and handing over (commissioning) stages, or how to use the final data as basic information for energy performance certificate or inspections. This list is applicable mainly for newbuildings. In buildings subject to renovation a careful study of the systems and their documentation is necessary in the very beginning. In other existing buildings a case-by-case approach is needed and the list is only partly applicable.

Stage I *(building permit stage)*
-basic data for calculations
--targets for energy performance and indoor environment
--weather data
--indoor environment parameters
--building envelope
--hours of operation and occupancy
--lighting and other internal loads
--ventilation and infiltration

-energy demand calculations to verify that the legal requirements are fulfilled
--heating energy demand
--indoor temperature summertime -> need for cooling
--cooling energy demand
--system energy
--building net energy

Stage II *(design stage – detailed calculation)*
-detailed system calculations using real system and product data as soon as available
--ventilation system
--domestic hot water
--heating system
--lighting system
--cooling system

-delivered energy: heat, electricity etc.

As an example (still just a proposal), we may apply a "star classification" – the example in Table 1 is prepared for multi-storey buildings, both new and existing ones.

Table 1. Example of data sheets for indoor environment and energy performance design

Category ➔		*****	****	***	**	*
Quality criteria:						
Air temperature						
Winter	°C	21 +0,5	21±1	21±2		
Summer	°C	24 ± 0,5	25±1	26±2		
allowed degree hours beyond the given range	°Ch	30/30	70/70			
vertical temperature difference	°C/m	1,5	2	3		
Air velocity						
Winter	m/s	0,14	0,17	0,20		
Summer	m/s	0,2	0,25	0,30		
Noise level	dB(A)					
living rooms, bedrooms		22	25	28		
Air quality and ventilation						
concentration/CO_2	ppm	700	800	900	1200	1500
air flow per person	l/s,person	12	10	8	6	4

	Categories for new buildings	
	Categories for existing buildings	

ENERGY CONSUMPTION

Category		**A**	**B**	**C**	**D**	**E**	**F**	**G**
Overall energy consumption	kWh/ m^2	105	125	150	180	210	250	300
- building envelope								
- electricity for appliances, lighting…								
- DHW energy								
- heationg of ventilation air								
- electricity for fans								
-electricity for pumps								

Energy performance of air handling system													
	qv (m^3/s)	**Specific fan power SFP (kW / m3/s)**						**Annual heat recovery efficiency (%)**					
		1	**<1,5**	**<2**	**<2,5**	**<3**	**<4**	>80	>70	>60	>50	>40	>30
Building (overall)	3,5				**x** (2,3)					**x**			
AHU 1	2			**x**					**x**				
AHU 2	1					**x**					**x**		
FAN 3	0,5	**x**											

OPEN QUESTIONS

The energy performance certificate has been discussed much more than the methods and inspections. But whatever the principles, final schedules and limitations are on national level in different Member States, it is of extreme importance that the certificate in new buildings is linked to the design process in one way or another, and that it based on what the building and systems really are – not only a "calculation just to satisfy the authorities". Figure 1 gives one approach to link the calculations and certificate together. In existing buildings, perhaps the most important issue is to find out the real performance of the building, in order not to sacrifice the indoor environment.

One big question is that lots of independent experts have to be trained in a relatively short time. Another question is: if, like is in the case of many household appliances, everyone wants the highest energy performance, this is of course a good objective but what will happen if other key elements of a good building – architecture, and especially indoor environment - have to be sacrificed in the name of energy performance. Figure 1 gives one possible approach to have these aspects in parallel in design, and this methodology can be extended further to include also the regular inspections – using the same system and product data as the basic reference.

DISCUSSION

In the implementation of the EPBD we should make a clear difference between **implementation in legislation** (the official implementation) and **adopting of the principles in real practice.** The latter is based on the official implementation, which should be completed by year 2009, and will probably take much more time, and require a high number of voluntary efforts in addition to the legal measures to ensure the fulfilment of the minimum requirements of the Directive.

The methodology presented in this paper is still under development, but based on elements already included in the good design and building practice. More detailed guidance and testing the methodology in real practice is needed, as well as tools to encourage the designers and clients to fix the data needed as early as possible. Testing the methodology in different types of buildings is also important to make the tools really practicable for different purposes, to fit also for use in existing buildings, and to avoid unnecessary complexity and unnecessary repetition of routine calculations in the design and construction process.

Tools for manufacturers – partly extending or revision some existing standards and product certification schemes are also needed so that certification of products will become really rewarding. The classification already applied in mandatory labelling schemes can be useful also in voluntary schemes.

It is too early to give estimates of the schedule of the "**adopting of the principles in real practice".** If we take the experiences from energy-labelling of household products, the future looks very promising: everybody wants to buy "Class A". For washing machines, people want the highest class also for other important characteristics: the washing and drying performance. So, why not to aim at the best energy performance and indoor environment simultaneously?

However, the complexity of decision-making in construction may slow down the development. Where indoor environment is clearly recognised as a key element for productivity, it is surely considered properly (or at least should be). In housing, the main target is still too often "to fulfil the minimum legal requirements with minimum investment".

The EPBD gives many opportunities. The main challenge is, however, to increase the common knowledge among our profession and among our customers, about the benefits of good energy performance and good indoor environment. The presented methodology provides one platform in this mission.

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Improving building design and indoor conditions in the mid-highlands of the French tropical island of La Réunion. Application to a green building high school "Le Tampon Trois Mares".

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SUMMARY

This paper deals with the use of passive solutions to reach thermal comfort conditions in an educational building under specific tropical climatic conditions. A case study of a green high school building located in the French tropical island of "La Reunion" at an altitude of 600m was used. The building designers initially planned for mechanical air conditioning in the classrooms and in the administrative section however dynamic simulations pointed out that with an optimisation of passive solutions such as solar protection, insulation, natural and mechanical ventilation; active air conditioning systems could be avoided. Condensation risks during the winter season were analysed also. The high school has been in service since September 2003. This research was the first scientific study conducted in the highlands of a tropical island. This paper presents the methodology used and describes the results and passive solutions recommended which are now widely used by building owners to design new building projects located in the same range of altitude.

INTRODUCTION

The methodology and results presented in this paper were developed within the framework of a research project funded by the Regional council of La Reunion and SEMADER, which is the company responsible for the construction of the high school. The island of La Reunion is a small French tropical island located in the Indian Ocean (21° of latitude south and 55,5° of longitude east). The island is only 70km long and 50 km wide (see Fig.1) and has a growing population. It is estimated that in the next 15 years, the population will have reached the billion marks with 25% of them being less than 20 years old. Ten more high schools are needed to account for this growing population, highlighting the importance of influencing green construction for new educational buildings in every sector –elementary schools, primary schools, colleges, high schools and university buildings. The urban areas mainly affected have so far been located along the coastline (see figure 1), but the recent regional urban planning act focuses now on developing the mid-highlands of the island rather than extending new urban areas along the coastline. This is mainly because most of the land is dedicated for the sugar cane industry. Electricity output in these highland areas is restricted and the area is mainly oil-powered (60%) and it is predicted that the energy supply will encounter difficulties with the 7% per year growing energy demand. The Regional Council is aware of the energy problem that is facing the island of La Reunion and has decided to launch a wide program of demand side management and development of the use of renewable energy [1]. The final aim of this programme is to have the island energy autonomous by 2025. A part of this program is dedicated to the construction of green buildings, which is the subject of this paper.

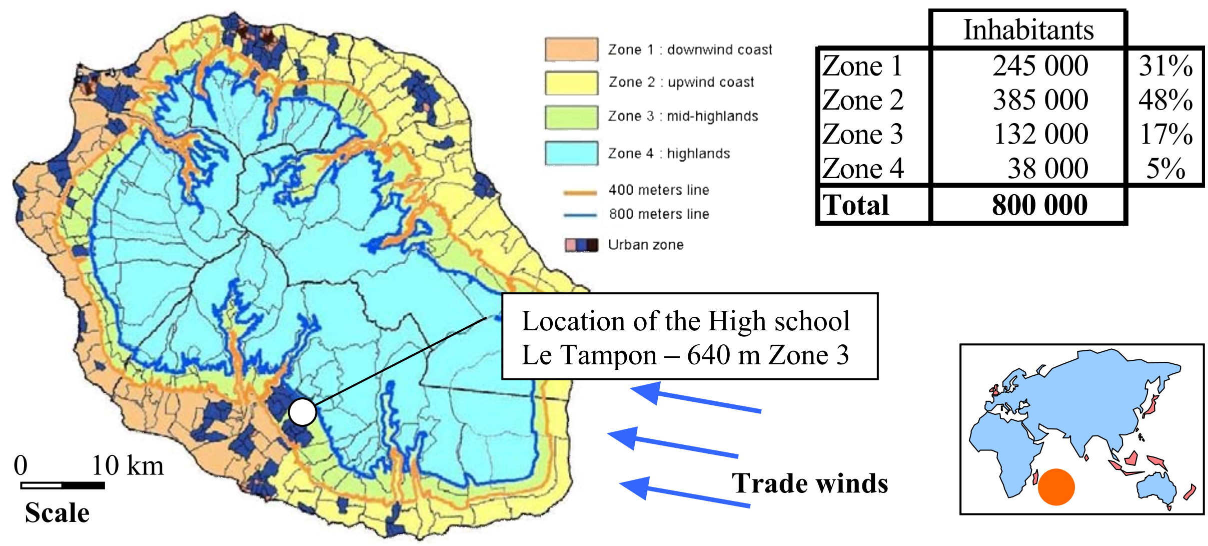
METHODOLOGY

The project consisted of the following steps:

1. Elaboration of typical hourly climatic data representative of the summer season and of the winter season;
2. Choice and Modeling of the typical buildings (that is, classroom, office and computing room);
3. Determination of the outputs and the simulations tools;
4. Simulation and analysis of the results;
5. Synthesis of the results and conclusions.

Climate in La Reunion and typical hourly data.

As shown in Figure 1, the climate in the island of La Reunion can be divided into four main zones [2]. The first two zones correspond to standard coastal tropical climates. They are both located below 400 metres and differ mainly by the average wind speed. The downwind coast, named zone 1, is the driest, the sunniest and the warmest. This part of the island is protected from trade winds by the mountains. The upwind coast (that is, zone 2) is wet because of its wind exposure. The mean global solar radiation is the same as in zone 1 (5000 kWh/m²/year). The highlands, named zone 4, are situated above 800 metres. The temperature is cool in summer and cold in winter with a minimum that can go down to minus 1°C. The period of sunshine remains good and the humidity is very high.



	Inhabitants	
Zone 1	245 000	31%
Zone 2	385 000	48%
Zone 3	132 000	17%
Zone 4	38 000	5%
Total	**800 000**	

Figure 1 : Urban areas and climatic zoning of the island of La Reunion

The high school is located in zone 3 in the suburbs of a town called Le Tampon. For this reason, we will focus on this specific zone. The main climatic features are given in Table 1. Zone 3 is a mid-highlands zone located between 400 and 800 metres above sea level. The average yearly temperature is 18.2°C. The mean solar radiation and the humidity are high all year round. The climate in this zone does not correspond to an existing classification mainly because of the altitude. Comparison with a “standard tropical climate” highlights the main differences to be summer conditions which can be hot and humid with temperature and humidity values that can go up to 30°C/78% and winters that can be cold for a tropical climate with temperature and humidity values that can go below 12°C/70%. Due to this, the existing references on recommendations for building in tropical climates can not be applied to this zone [3]. The recent publications concerning the thermal comfort in classrooms deal with the thermal conception of buildings in tropical climates in Singapore or Hawaii [4] [5] and mainly focus on the comparison between the thermal sensation of the occupants and the ASHRAE Standard 55-92 [6]. Even the recent work of Kwok in Japanese schools deals with a field survey and a comparison of the occupants preferred thermal state to the ASHRAE comfort standard [7]. While the climate there is close to zone 3, with hot and humid summers and mild winters, wider discrepancies exist between the mean temperatures in summer and winter. These are 27°C and 4°C respectively instead of 22.8°C and 17°C for zone 3. Other research concerning the energy performance of school buildings has been conducted in Israel by Becker [8] for hot and dry conditions in summer and moderate conditions in winter. The climate which seems to be the closest to that of zone 3 can be found in the work of Lam where passive design principles have been defined for different climatic zones in China by using a bioclimatic approach. In the southern part of China, the average annual temperature is around 21°C with hot and humid summers and mild winters. Lam recommends using natural ventilation for passive cooling in summer and assesses that passive solar design is sufficient enough to ensure the comfort is maintained during winter.

Table 1: Main features for Zone 3 (mid-highlands) –annual, summer season, winter season

	Air temperature (°C) and associated Relative humidity (%)						Diurnal temperature range ΔT T_{max} - T_{min}	Mean global solar radiation (kWh/m²/day)	Mean wind speed (m/s)
Typical day	T_{mean}	RH	T_{max}	RH	T_{min}	RH			
Annual	**18,2**	**76%**	**25,1**	**64%**	**16**	**73%**	**9,1°C**	**4916**	**1,5**
Mean summer	22,8	79%	28,3	68%	18,9	80%	9,4°C	5812	1,4
Extreme summer	23,5	75%	30,1	62%	19	78%	11,1°C	7008	2,5
Mean Winter	17	73%	21,9	62%	13,2	74%	8,7°C	4855	1,7
Extreme winter	14,8	75%	18	67%	12	70%	6°C	2846	3,5

With reference to school building design, it can be seen that the existing material for school design recommendations is not suitable for the climate in question. That is, in zone 3, the building design has to provide for both tropical climates in summer (use of cross ventilation and solar shadings) and temperate climates in winter (passive solar heating and condensation risk). A main charactistic of the mid-highlands climate of La Reunion is that neither air-conditioning nor heating are necessary if some basic passive design principles are applied. This was the purpose of this study to demonstrate this assessment through the use typical meteorological data file and dynamic thermal simulations. The recommandations through this research could then be applied to new building projects in the area and on a wider scale through setting local thermal standards [2].

The generation of hourly typical meteorological days for zone 3 has been performed by calculation methods developed by David [10], [11]. These methods are based on algorithms of classifications applied to several years of data. An eleven year period from 1993 to 2004 has been considered to generate the typical sequences which can be read as a TMY format file by our simulation code. The climatic data include outdoor air temperature, relative humidity, global and diffuse radiation, wind speed and wind direction.

Main features of the high school

The high school consists of a three storey building with a capacity of 720 college students (which can be expanded to 900). The total floor space is 8352 m². Typical to most schools, it is composed of different sections comprising the administration part and different use classrooms (for tutorials, practical work and computer activities). A specific building on the east side is dedicated to sport activities. All the walls are 16 cm thick concrete walls with no insulation. The roofs are double skin consisting of a terrace concrete roof with 6 cm of external insulation and a 10% slope corrugated iron roof above this. At the design stage of the project, all the buildings were already cross ventilated with sun shading devices, louvres and overhangs. Most windows are louvres but some windows have slide-windows with an additional louvre on the side. Heating and cooling air conditioning systems were supposed to be installed in the administrative building, staff room, and in the on-site row houses of the administrative staff. Computer rooms and classroom are naturally cross-ventilated. No mechanical ventilation or air conditioning was planned for these rooms.
The high school has a rain water storage tank that collects water for the toilet and watering use only. 391m² of PV panels are installed on the northern roof, which represents an installed power of 51 kWp. The annual production is estimated to be 66,300 kWh.

Figure 2 : The three storeys main building with sun shading devices, external corridors facing north and louver windows (right). All the classrooms are cross ventilated with sliding windows, louvers and air fans (left).

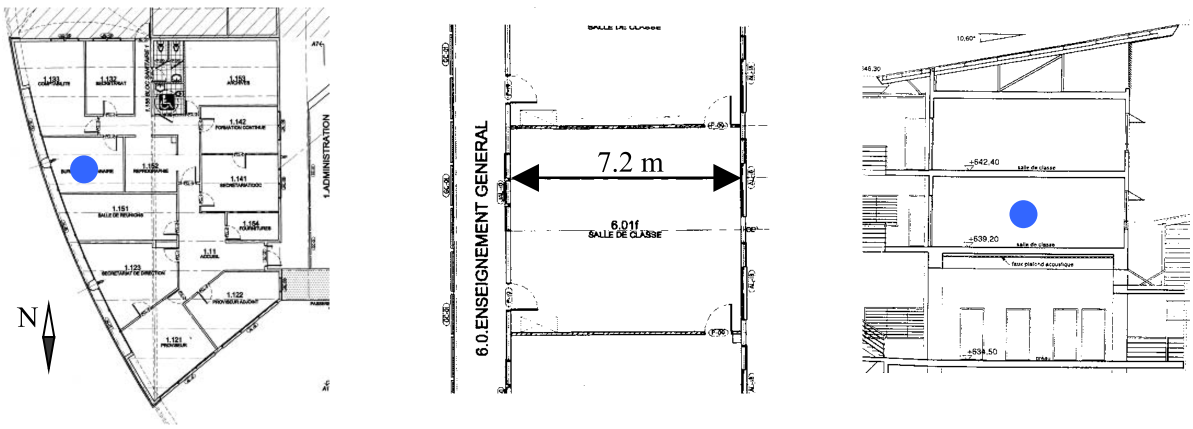


Figure 3 : Plan view of the administrative offices (left), of a typical classroom (middle) and cross section of a typical building with the double skin roof (right). The typical rooms used for the simulations are spot marked.

This study mainly focused on three typical rooms as they represented 80% of the floor space. These were one administrative office, one classroom, and one computing room. The size of the two classrooms is the same. Only the internal loads due to the computers differ.
For each typical room, different scenarios of building components (insulation) and air renewal have been tested. As previously stated, the main problem of thermal design in this area comes from the humidity during winter time [11]. The lack of insulation coupled with no mechanical ventilation is the main cause of such problems. At the same time however, the summers can be hot and uncomfortable. Due to low temperature at night during summer, night cooling is a possible solution to consider. This passive cooling technique (used mostly in European countries) has not been tested as yet in La Reunion [12].

Outputs and simulation tools

A thermal and air flow simulation program was used, which in conjunction with a model of large openings under pressure, showed the impact of natural ventilation on thermal comfort. (See reference [13] for a detailed description of Roldan's mass transfer model. The model takes into account one air volume per room neglecting stratification. The comfort indices chosen for summer time were Givoni's comfort diagram [14] and the resultant temperature T_{res}. The comfort zones allow us to find the percentage of points within each comfort zone (0 $m.s^{-1}$, 0.5 $m.s^{-1}$ and 1 $m.s^{-1}$) as well as the number of hours of discomfort. T_{res} is a weight equation between the air temperature T_a and the average wall temperature T_{rm} for a naturally ventilated room In summer $T_{res} = 2/3.T_a + 1/3\ T_{rm}$. During winter time, the Predicted Mean Vote (PMV) [15], the resultant temperature and the condensation risk were used as outputs. In winter $T_{res} = 0{,}45.Ta + 0{,}55.T_{rm}$ which is the usual equation used for low-level indoor air velocities.

RESULTS AND DISCUSSION

As a large number of simulations and data have been carried out and analysed, the results presented in this section will focus on a typical office only. The most important conclusions for the other rooms studied will be presented however. The whole research report is available by request to the corresponding author or by downloading it at http://lpbs.univ-reunion.fr.
The office room has 13m² floor space with a window facing west which has a solar factor of 0.3 (one meter deep solar shading device with clear glass). The internal loads consist of one person (occupancy from 0800 to 1700hrs), artificial lighting from 1400h to 1700h (13 W/m²) and one computer from 0800h to 1700h. The main objectives are to check the possibility of removing air-conditioning devices while ensuring a good level of comfort is maintained all year round and avoiding problems due to condensation.

Comfort conditions during summer – typical office room

Different scenarios of mechanical air renewal listed in Table 2 have been considered.

- Case 1 : one ACH (ie18 m^3/h) 24 hours a day;
- Case 2 : one ACH from 8h to 19h then 20 ACH from 20h to 7h, window remains closed ;
- Case 3 : one ACH from 8h to 19h then 20 ACH from 20h to 7h, windows is opened from 14h to 17h.

One can notice that the mechanical night cooling lower the morning resultant temperature by 4°C. It remains below 26°C until 13h. With opened window, the mean maximum temperature never goes above 27°C during the whole summer. The effect of opening the window during the afternoon (case 3) lowers the temperature by 1°C compared to case 2.

Figure 4 shows the impact of mechanical night cooling in terms of thermal comfort. Case 3, which is definitely the best scenario, (grey tint in Table 2) allows to have 100% of the time inside the 0,5 m/s comfort zone (see Table3). That means that air-conditioning is not necessary. The speed of 0,5 m/s can be obtained by using cross ventilation and/or coupled with the use of one air fan. Even if for acoustic reasons the window remain closed (case 2), the use of air-fan is sufficient enough to ensure the comfort conditions 100% of the time.

Table 2 : Impact of the different scenarii of mechanical air renewal on the resultant temperature

	T_{res} 8h-12h	**T_{res} noon**	**T_{res} 13h-17h**	**Mean $T_{res,max}$**	**Abs $T_{res, max}$**
Case 1 : 1 ACH 24h/day	27.5	28.5	30.4	30.7	32.2
Case 2 : 1 ACH /20 ACH (20h-7h)	23.6	25.2	27.4	27.8	29.1
Case 3 :1 ACH 20 ACH (20h-7h) windows opened in the afternoon	23.5	25.1	26.1	26.8	28.4
Outdoor temp	25.6	26.2	25.1	27.3	30.1

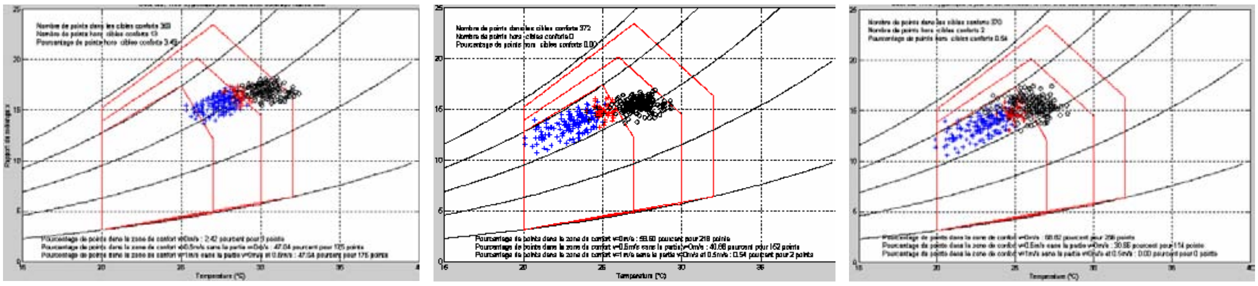


Figure 4 : Impact of the air change rate and mechanical night cooling on thermal comfort : case 1 (left), case 2 (middle), case 3 (right).

Table 3 : Percentage of dots within the comfort zones and number of hours of discomfort

	Office Case 3	Classroom Cross ventilated	Computing room Cross ventilated	Computing room Single side windows
Comfort zone - 0 $m.s^{-1}$	69%	32%	22%	14%
Comfort zone - 0.5 $m.s^{-1}$	31%	60%	60%	30%
Comfort zone - 1 $m.s^{-1}$	0%	4%	13%	49%
% outside the comfort zones	**0%**	**4%**	**5%**	**7%**
Hours of discomfort	***none***	***36 h***	***54 h***	***70 h***

Table 3 synthesises the results in terms of thermal comfort for the all the typical rooms. It is shown that thermal comfort conditions are reached most of the time. The worst configuration is for single sided computing rooms where 70 hours of discomfort were calculated.

As for the impact of the different cooling solutions on the installed power and on energy consumption, several simulations have been carried out with different scenarios that are listed in Table 4. A Coefficient of Performance (COP) of 2 has been considered to determine the electric power of HVAC systems. Case 1 and case 2 represent the initial project with and without artificial lighting. Case 3 and 4 are an intermediate solution where air conditioning is only used in the afternoon thanks to the effect of night cooling. Case 5 and 6 only use mechanical night cooling and ceiling fans. It is noticeable when comparing case 1 with case 2, that the use of artificial lighting increases both the power and the energy use by 30%. The use of night cooling allows a reduction in electric power by 10% and the energy consumption by 40% (see case 3 and case 4). The best solution was found to be the use of air fans that reduce both the power and the energy consumption by a factor of 12 and achieving good levels of thermal comfort.

Table 4 : Impact of the different cooling solutions on the installed power and on the daily electric comsumption

	$P_{max\ el}$ (kW)	*Installed power* *(W_e/m^2)*	Cooling load kWh_{cool}	Electric consum. *kWh_e/day*	Energy Index *($kWh_e/day/m^2$)*
1. Air cond. all day /natural lighting/1 ACH	0.67	*42*	77.5	25.83	*1.61*
2. Air cond. all day/artificial lighting/1 ACH	0.85	*53*	100.73	33.58	*2.1*
3. Afternoon air cond. / natural lighting 20 ACH at night/1 ACH during daytime	0.58	*36*	36.41	12.14	*0.76*
4. Afternoon air cond. / artificial lighting 20 ACH at night/1 ACH during daytime	0.76	*48*	56.03	18.68	*1.17*
5. 1 hour use of ceiling fan /natural lighting 20 ACH at night/1 ACH during daytime	0.05	*3*		2.2	*0.14*
6. 3 hours use of ceiling fan /artificial lighting 20 ACH at night/1 ACH during daytime	0.05	*3*		3.3	*0.21*

Comfort conditions during winter – typical office room

As for the study of comfort during winter, two solutions have been evaluated: case 1, one ACH 24 hours a day, and case 2, one ACH only during the day (8h-19h). Preliminary simulations conducted without mechanical equipment have pointed out that condensation occurred most of the time on the internal surface of windows and even on the walls if they are not insulated. Table 5 shows that when the mechanical air renewal is off at night, the temperature is slightly better (0.7°C) but condensation occurs almost every time on the window. On the contrary, if it is used for 24 hours, the condensation risk is totally avoided. The comfort conditions remains good with only a few hours outside the comfort zone for 0 m/s (see Figure 5). The low-level risk of condensation is confirmed. Most of the humidity/temperature pairs are far away from the saturation curve. The simulated PMV is only 12h below -0,5 throughout the entire winter period.

Table 5 : Impact of the mechanical air renewal on the resultant temperature in winter

	T_{res} 8h-12h	**T_{res} noon**	**T_{res} 13h-17h**	**Mean $T_{res,min}$**	**Abs. $T_{res,min}$**	**Condensation risk**
1 ACH 24h/day	21.5	22.7	24.8	20.3	19.1	none
1 ACH 12h/day	22.2	23.3	25.3	21.1	19.8	Yes. 90% of the time on the indoor window surface.
Outdoor temp.	18.2	19	18.2	13	11.6	

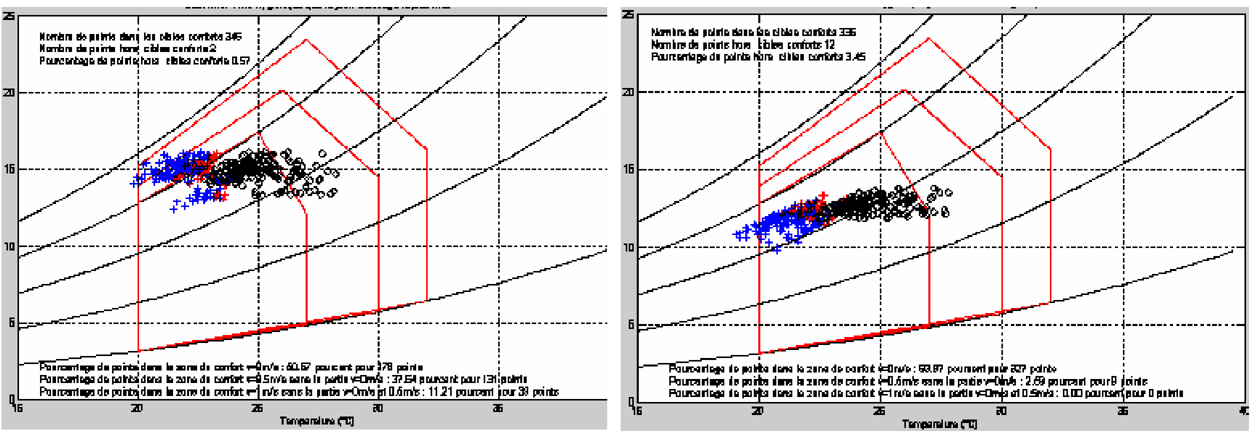


Figure 5 : Impact of the mechanical air renewal on thermal comfort and on condensation risk 1 ACH 12h during daytime (left), 1 ACH 24 hours a day (right).

SYNTHESIS AND CONCLUSION

The keys findings of this study are as follows. The mid-highlands climate in tropical islands must be considered apart because one must cope with the duality of the design principles for tropical climates in summer (use of cross ventilation and solar shadings) and temperate climates in winter (passive solar heating and risk of condensation). Nevertheless, this study has demonstrated that neither air-conditioning nor heating are necessary if some basic passive design principles are applied. With respect to the building envelope, a minimum of 10% of openings for cross ventilation and a minimum solar factor for windows is required. The values of solar factor function of the window orientation are given in [2]. Air treatment in the administrative offices is needed with a 20 ACH of mechanical night cooling coupled with ceiling fans proving to be an interesting alternative to air conditioning during summer time. During winter time, one ACH of mechanical air renewal is mandatory to avoid condensation problems. The comfort conditions are almost reached all year round with only 12 hours of PMV <-0,5. This proposed solution is the first of its kind in La Reunion. It has since been applied to other building projects, either office or educational buildings. In the others typical rooms studied (that is, classroom and computing rooms), natural ventilation and use of air-fans is sufficient enough in summer to ensure good comfort conditions. During winter, louvers must remain slighlty open to avoid condensation. The PMV for this scenario is below -1 only one hour per day.

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An adaptable urban house designed for the southern Brazilian climate – emphasis on summer and winter thermal comfort

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SUMMARY

The climate in southern Brazil is characterised by mild winters and hot-humid summers which requires the design to be adaptable to the often conflicting summer and winter requirements. In the residential sector, air conditioning consumption is still low, but it has been growing significantly along with an increase in people's purchasing power which emphasizes the importance of encouraging a change in construction practices [1]. The aim of the research is to design a residence that combines traditional and contemporary techniques and technologies, along with a smart design, providing acceptable summer and winter thermal comfort. This was pursued by considering the advantages found in some precedents, such as, passive heating and cooling strategies that respond to the climate; and by controlling the relationships between buildings and outdoor spaces to ensure a response to the different seasons. Environmental design approach is found to go far beyond the quantification of energy consumption through the use of different materials or strategies.

INTRODUCTION

To reduce energy consumption, the adequacy of the architectural standard is the item that requires the lowest investment and provides one of the highest energy savings.

Finding successful means of energy reduction and a solution to the environmental effects of air conditioning is a strong requirement for the future. Possible solutions include the adaptation of buildings to the specific environmental condition of cities in order to incorporate renewable energy technologies efficiently to address the radical changes and transformations of the radiative, thermal, moisture and aerodynamic characteristics of the urban environment. This involves the use of passive and hybrid cooling and heating techniques to decrease cooling and heating energy consumption and improve thermal comfort.

OBJECTIVES OF THE BUILDING

2.1 Project Brief

It is a house designed for a family of 4 to 5 people that enjoy being outdoors and having freedom in the house. The average size required is around 200 m². The programme is composed by Living Room, Barbecue Room, Kitchen and Study Room - mostly occupied during the whole day; 3 Bedrooms - occupied during the night; Laundry, Bathrooms and Garage - occasionally; and Swimming Pool - summer.

2.2 Conceptualization

According to the study of the built examples some conclusions were drawn, such as the importance of the engagement of the architecture with the environment, about rethinking the way that people live and how the design should respond to the climate [2].

The house should be adaptable, able to be changed according to the seasons, responding at the same time to the climate and to the occupants' attitudes and expectations of comfort. In winter it should be more compact and well insulated to keep the internal and solar gains inside the house and save energy when mechanical heating is needed, but at the same time it should be able to be more permeable in the summer and be fully integrated with the outside.

DESIGN GUIDELINES

3.1 Chapecó's Climate

The context of this work is the city of Chapecó, which is located in Southern Brazil, in the west of Santa Catarina State at Latitude 27°14' S, Longitude 52°37' W and Altitude 668 m.

The following description of the climate of Chapecó is based on 10 years of hourly meteorological data recorded in Chapecó's airport processed through the Software Meteonorm [3].

3.1.1 Dry bulb temperature

The annual mean temperature is 19.2°C. The hottest month is January, presenting a mean temperature of 22.8°C and the coldest is June, with a mean temperature of 13.5°C. The annual mean daily range of temperature is 9.3°C. In January, the mean maximum temperature is about 28°C and the mean minimum is 16.9°C, producing a daily range of temperature of about 11.1°C. In June, the daily swing of temperature is also about 11.1°C, ranging from a minimum mean of 7.9°C to a maximum mean temperature of 19°C.

It was observed that during the winter, or even in the summer, there are periods of heat and cold, respectively.

3.1.2 Relative air humidity

The annual mean relative humidity is 73%. The months with the highest average relative humidity are May and June (around 78%) and November is the month with lowest average (around 68.6%).

3.1.3 Wind speed and direction

The annual mean wind speed is 2.5 m/s. March is the month with the weakest winds and September has the highest speeds, both average and maximum.

Wind frequency, relative to its direction, shows the north-eastwards and northwards directions as predominant throughout the year. The north-eastwards direction is predominant during summer and the northwards direction in August. The second highest frequency occurs south-eastwards in most part of the year, mainly in winter and in the months of June and September.

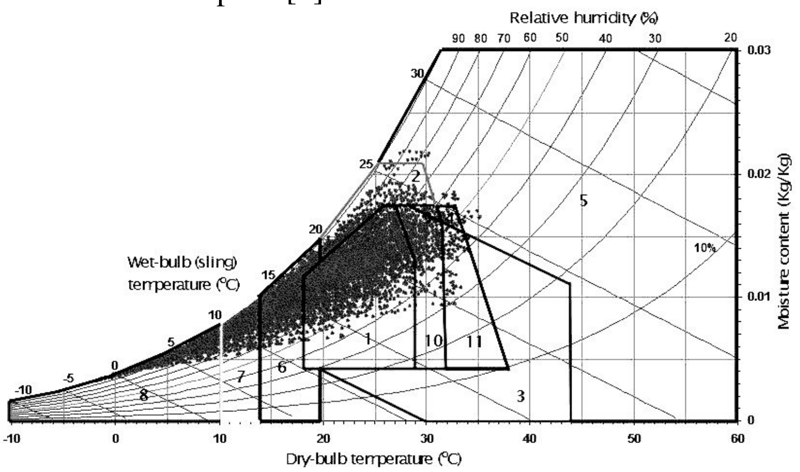
3.1.4 Cloudiness and Insolation

In Chapecó's climate, winter months feature lower cloudiness rates than during summer months.

Annual average cloudiness is 5.4 (part of the sky covered by clouds: from a 1 to 10 scale, that is, greater than 50%); also, cloudiness tends to increase from September to February, reaching 59% in January and February. May is the month with the lowest cloudiness rate and is the only one to fare below 50%.

3.2 Passive Strategies Investigation

To obtain some building design guidelines the result obtained was analyzed by applying the program developed at LABEEE (Laboratory of Energetic Efficiency in Buildings – Federal University of Santa Catarina – Brazil), based on Watson's and LAB's method (1983), thus allowing the crosschecking of all hourly climate data on the Bioclimatic Building Plan tailored to Chapecó [4].



1. Comfort zone
2. Ventilation
3. Evaporative cooling
4. Thermal mass for cooling
5. Air-conditioning
6. Thermal mass/Solar heating
7. Solar heating
8. Mechanical heating
9. Ventilation/Thermal mass
10. Vent./Mass/Evap. cooling
11. Mass/Evap. cooling

Figure 1: Chapecó's climatic year

DESIGN DEVELOPMENT

According to the local climate analysis the primary need is to increase winter heat gains and reduce summer heat gains during the day. This was attempted through a comprehensive set of actions that start on a large scale by controlling the relations between buildings and outdoor spaces.

In the design stage the advantages which were found in the three different houses' approach of designing and passive strategies that would respond to the climate were taken into consideration [2], as a result an adaptable house responding to the local climate is proposed.

4.1 Site layout and Building Form

As a starting point for the design, it was decided to separate the living area (daytime) from the sleeping area (night-time), thus more rooms could get the winter sun and have cross ventilation also allowing the buildings to be connected in winter and disconnected in summer which could lead to a more interesting site layout.

The sleeping area was located at the back of the site, consequently further away from the street and noise. It would comprehend the whole site's width (17 m) to ensure that all the bedrooms would face north. The living area would be more compact allowing 4 m setbacks at the sides to decrease the overshadow created by the neighbouring houses and walls and also for the creation of side gardens and courtyards that would allow the extension of the inside space when the outdoor temperature is comfortable.

The average distance found to ensure that the sleeping area would not be overshadowed by the living area was around 8 m (Fig. 2 and 3). It was considered that the first volume had an average height of 4.5 m, which is going to be used at the final project.

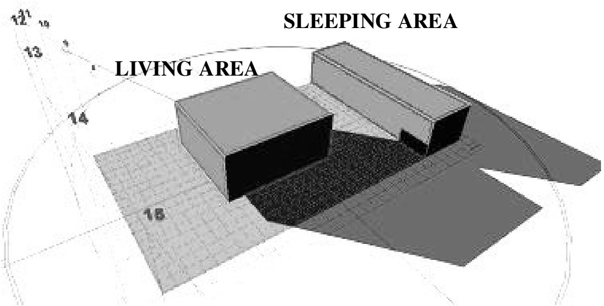


Figure 2: Shading study on June 21st at 9am

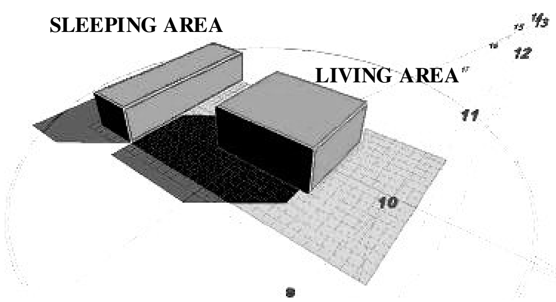


Figure 3: Shading study on June 21st at 3pm

4.2. Internal Layout

The house was designed with an internal layout where the main rooms such as barbecue room, living room and bedrooms are facing north as they are the rooms that will be most occupied at daytime and evening in winter. The study room, the second most used space, is facing east, and kitchen and laundry facing west as they also can be shaded in summer. Openings are not placed on the south façade as it does not have any solar gain in winter and in the summer it gets some sun, and the south façade might require extra insulation to avoid loss of heat gains.

The leisure area of the house would be on the roof of the ground floor as the swimming pool requires maximum solar radiation to heat the water (Fig. 5). The swimming pool also works as heat storage through its thermal capacity. The ceiling, which is cooled by heat conduction through the water, acts as a radiant/convective cooling panel for the space under it. Thus, the indoor air and radiant temperatures can be lowered without elevating the indoor humidity level.

4.3 Design Project - *Summer and Winter House*

The house proposed is adaptable through moveable devices. It can change its configuration through the occupant's interaction responding to the local climate and to the seasons change.

It was focused to design the versions of the house in summer and in winter as they are the most extreme seasons. Thus, in winter the house is more compact and well insulated to keep the internal and solar gains inside the rooms and save energy when mechanical heating is needed (Fig. 4). In summer it is more permeable and the relationship with the outside is stronger (Fig. 5).

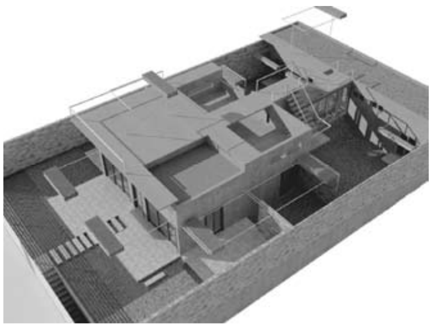


Figure 4: Winter House

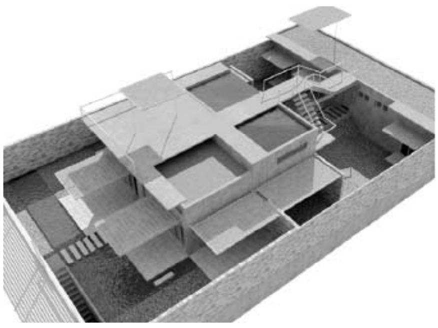


Figure 5: Summer House

The floor plans of the house are presented below with the strategies and the design elements pointed at the floor plans are explained (Fig. 6, 7, 8 and 9).

1. Thermal storage - water
2. Reflection - gardens
3. Thermal mass – stone (walls and floor)
4. Thermal mass – brick (walls)
5. Cross ventilation - sliding doors fully opened

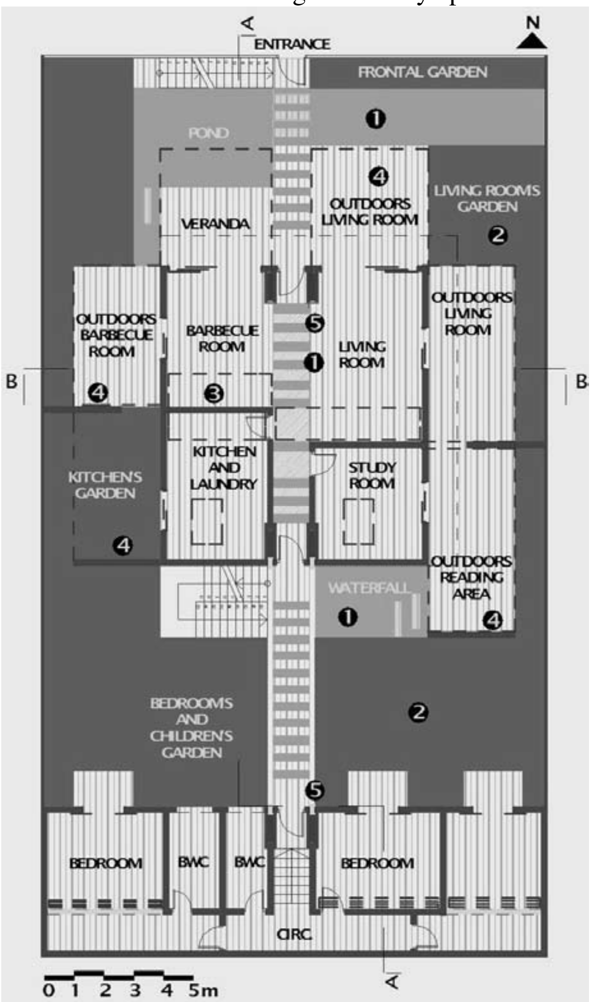


Figure 6: Ground Floor Plan – *Summer House*

1. Thermal mass and wind protection - concrete
2. Compactness - sliding doors fully closed

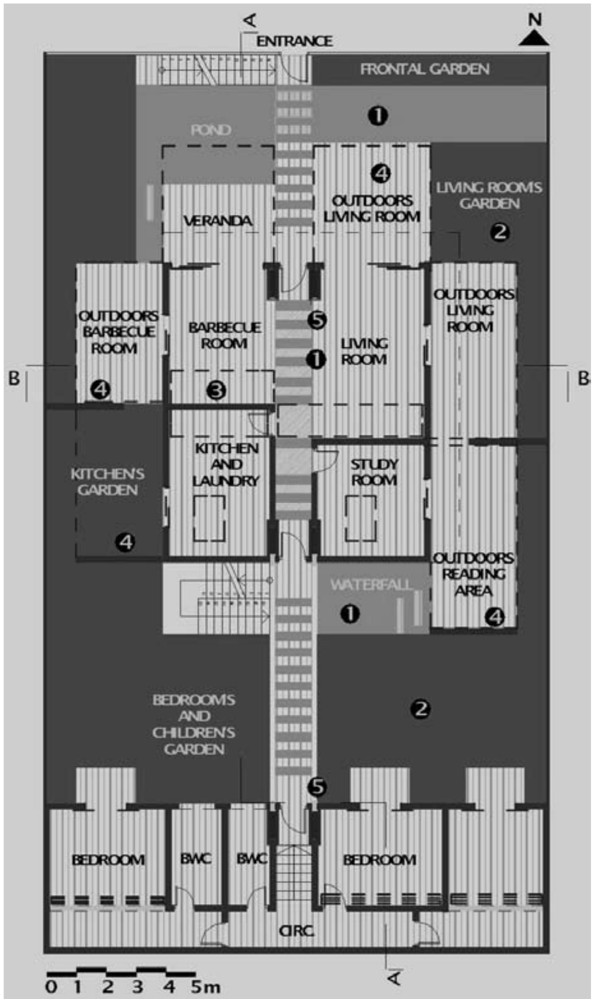


Figure 7: Ground Floor Plan – *Winter House*

1. Moveable shading device/night shutters - position: horizontal fully opened
2. Moveable shading device/night shutters - position: horizontal fully closed
3. Moveable shading device/night shutters - position: horizontal almost fully closed
4. Moveable shading device/night shutters - position: horizontal semi-opened
5. Reflection – green roof
6. Thermal storage - swimming pool
7. Fixed shading device

1. Solar heating - roof light
2. Thermal insulation – green roof

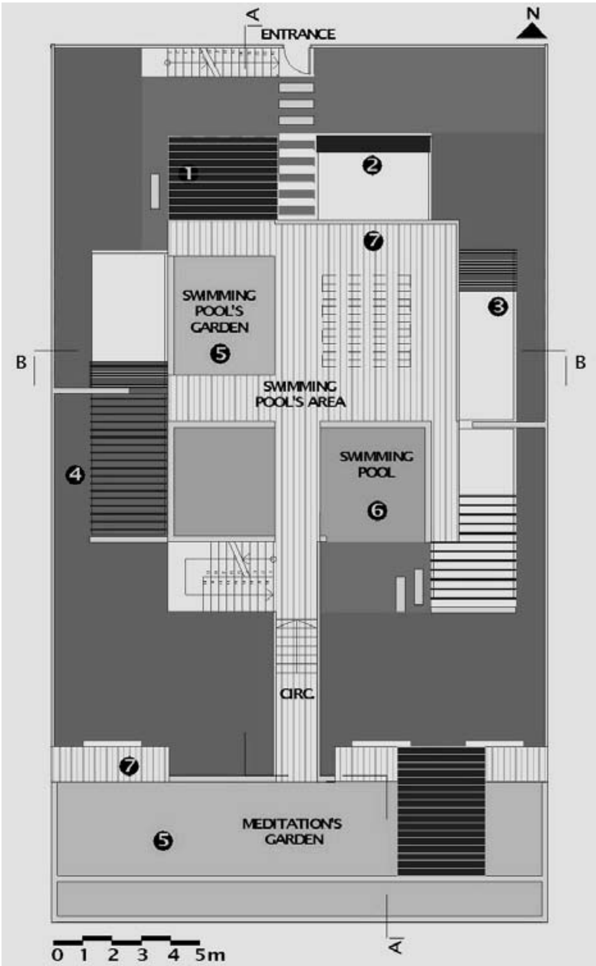


Figure 8: Roof Plan – *Summer House*

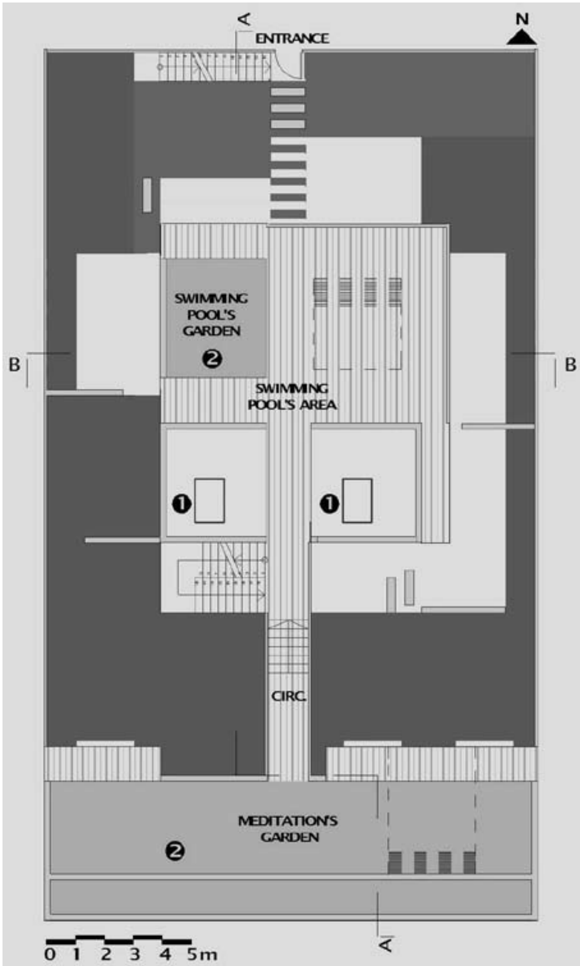


Figure 9: Roof Plan – Winter House

4.3.1. Thermal Mass

Thermal mass was applied through the brick and stone walls, concrete ceilings and floors (Fig. 6). It is a strategy suitable for summer, decreasing the peak of cooling loads and releasing, with a delay, the heat to indoors. In winter, the solar diurnal gain can be stored in the thermal mass and transferred into the building at night, when heating is required.

According to Balcomb [5], thermal mass is best increased by maximizing surface area rather than increasing thickness. Thus, thermal mass was also applied on the walls surrounding the building creating courtyards (Fig. 10 and 11). These walls can store some heat during the day and release it at night creating warmer and more comfortable outdoor spaces.

Figure 10: Barbecue Room's Courtyard – *Summer House*

Figure 11: Living Room's Courtyard –*Summer House*

4.3.2. Ventilation

Ventilation happens through the openings located at the occupant's level, providing comfort ventilation and located at a higher level in the bedrooms, barbecue room and living room (Fig. 12). However, as in these spaces there is a moveable insulated ceiling which can be closed decreasing the height of the room in winter, thus in winter the higher windows stay above the insulated wood ceiling and closed (Fig. 19). All the openings, with the exception of the kitchen and study rooms, are facing northwards which is the main wind direction.

Figure 12: Barbecue Room's openings and shading devices – *Summer House*

4.3.3. Shading Devices/Night Shutters

The shading devices applied in the design are mostly moveable allowing the occupant to control the penetration of direct solar radiation and also daylighting levels, to control direct gain passive heating, maximising useful gains while avoiding overheating. They are more used in summer, however, it was observed that during the winter, or even in the summer, there are periods of heat and cold, respectively. Therefore, within the same season, days may occur with characteristics of the opposite season consequently if the devices are moveable they can also provide shade for the lower winter sun.

As the configuration of operable shading devices can be changed, their performance can be much better than that of fixed devices. They are composed of fins that can be rotated in both directions and if they are not in use they can be folded to one side (Fig. 13). The horizontal and vertical shading devices also work as night shutters and as insulators for the walls covering its whole extension (Fig. 14).

Figure 13: Different configuration of shading devices

Figure 14: Different configuration of shading devices

The higher windows of the living room, barbecue room and bedrooms are also shaded to avoid unwanted heat gains in summer. The shading devices are fixed, and are the floor of the Swimming Pool's Area and the sits of the Meditation's Garden.

According to insolation studies on the 21st December, the sun just heats the northern surfaces between 10.30 am and 2.30 pm, before and after that it is towards the South, so the shading devices needed are quite small (Fig. 15).

According to Shaviv and Capeluto studies [6], the winter performance of the thermal mass is very sensitive to the roof's insulation and the better it is the less energy is required for heating. On the other hand, this is not the case in summer, when the better the roof's insulation, the more energy is required for cooling. This is because the poorly insulated roof allows night cooling by heat conduction and by long wave radiation. Consequently, higher transmittance in the roof is recommended.

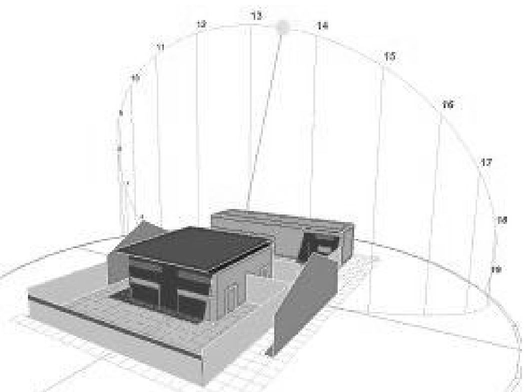


Figure 15: Fixed shading devices for higher windows

Thus, besides the function of shading the higher windows the floor of the Swimming Pool's Area is partially suspended shading the roof below from the Living and Barbecue Rooms. It also allows some cross ventilation as it is open on both sides (Fig. 16). In winter it is closed to avoid the contact of the roof with the outside temperatures (Fig. 17).

Figure 16: Suspended roof – *Summer House*

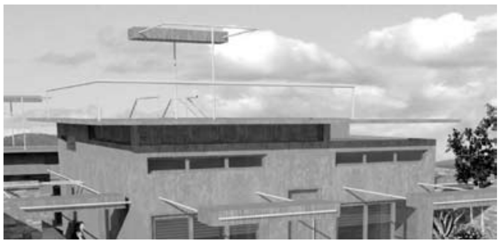


Figure 17: Suspended roof closed–*Winter House*

4.3.4. Swimming Pool and Ponds

Water is used on surfaces outside and inside the building as it has a very high thermal capacity, about twice that of common masonry materials and also has the advantage that convection currents distribute heat more evenly throughout the medium also providing some evaporative cooling. The water way inside of the house in summer is partially open and with moveable glass sheets, so it is possible to see the water through it (Fig. 20). In winter this floor is replaced by insulated wood panels, becoming fully closed. The same happens at the water way that connects the living area to the sleeping area.

The water that circulates around the house and from the waterfall is not the same as that from the swimming pool as one of the requirements for the swimming pool is to have its water heated, so it should be fully under the sun and not in movement.

The swimming pool was located above the kitchen and the study room, thus their roof can be cooled by evaporation and the ceiling then acts as a passive cooling element for the space below. The ceiling, which is cooled by heat conduction through the water, acts as a radiant/convective cooling panel for the space under it. Thus, the indoor air and radiant temperatures can be lowered without elevating the indoor humidity level.

In the winter the swimming pool is empty allowing the entrance of light through the roof lights to heat the study room and the kitchen. The dimension and the location of it were done by shading studies (Fig. 18). According to the shading mask the roof light would get full sun between 10 am and 2 pm, before and after that it could be closed and very well insulated avoiding loss of heat to the outside.

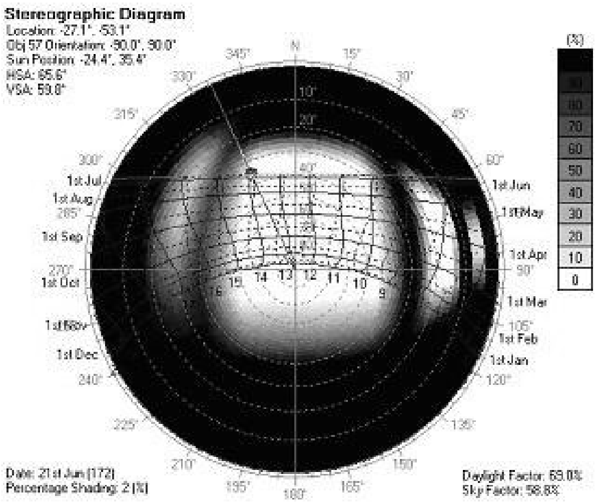


Figure 18: Shading mask of roof light in the kitchen – *Winter House*

4.3.5. Gardens
The roof is the building element which receives the most amount of sun, especially in summer, and that is more exposed to nocturnal cooling by radiation. Vegetation protects the roof from direct solar radiation and enhances its thermal behaviour providing a cooling effect in summer as green surfaces reflect sun radiation and storage heat causing a cooling effect. In winter it can be used for its thermal insulation effect though the air cushion within the vegetation and the fact that the cold wind does not hit the earth surface consequently protecting the surface below. Thus, grass was used on the Bedrooms' and on the Barbecue Room's roof (Fig. 8).

4.3.6. Solar Heating and Architectural Features
Solar Heating was applied at the design stage as storage element exposed to the indoor and outdoor spaces to store excess solar energy during the sunny hours and release it back during the night. This strategy was applied through the mass of the building fabric itself, floor, in the internal layers of external walls, internal partitions, on the roof and in the furniture surface area.

To improve the solar heat and internal gains of the building, a moveable ceiling and sliding doors were applied in the Bedrooms, Barbecue and Living Rooms. The moveable ceiling is composed of wood filled with insulation. There are 4 or 5 sheets of 1 m each that can be fully closed in winter leaving the room with a height of 2.6 m (Fig. 19) and in the summer they can be folded (Fig. 20), leaving the room with a height of 3.5 m. In the summer the height improves ventilation through the high windows (just visible in summer); in winter, the floor-to-ceiling lower height helps to keep the internal gains and solar gains in the room. They also have wood sliding doors that make it possible to isolate the rooms in winter (Fig. 19) and leave them open in summer increasing ventilation (Fig. 20).

The circulation from the living area to the sleeping area is also created by sliding doors that can be fully opened in summer and fully closed in winter, keeping the access between the two zones less exposed to the weather (Fig. 4 and 5).

Figure 19: Moveable insulated ceiling and sliding doors – *Winter House*

Figure 20: Moveable insulated ceiling and sling doors – *Summer House*

CONCLUSION

The design of more enjoyable spaces and their interrelation in a more interesting way was pursued at the same level as the outside penetration/interference at the inside spaces, and, on its occupants interaction with the building's envelope and features.
In response to this, the design started by challenging attitudes in house designing, pointing out that the occupant should be in control of the house in such a way that he feels that he is interacting with the design, having the possibilities of achieving what he wants from the space, mostly by moving architectural features instead of switching on equipments. Its occupants can deliberately see very visible differences between "summer" and "winter" houses through a clear reading of the different devices.

It was found that environmental design combining passive strategies and the intention of achieving a quality design might be more efficient in the overall environmental design picture. The occupant's thermal and psychological comforts were placed as the main issues to be addressed and not energy savings, as the latter is a consequence of the two former considerations.

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Comparative study regarding the energy supply of a high-tech building

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SUMMARY

The paper presents a comparative study among six different solutions regarding the energy supply of a high-tech building. All the solutions take into account the coupling of the sources of energy and the energy storage, considering equipments like: boilers, compression and absorption refrigeration machines, thermal motors, fuel cells, storage tanks and the use of the underground water and an artificial lake. The functioning of the different solutions was simulated using Simulink - Matlab. The second part of the paper presents technical and economical considerations of the proposed solutions, permitting the choice of an optimal one.

INTRODUCTION

This work presents some results of a technical and economical study concerning different energy supply systems (including heating, cooling, and electrical energy consumption) for a complex (industrial and office) building. The complex building represents a research and development (R&D) centre, having a 35000 m^3 volume, situated in London, England.
The energy supply solutions were designed taking into consideration the modern solutions (cogeneration using internal combustion engines, fuel cells), and analysing the size and costs for each type energy supplied (thermal energy for heating/cooling, electrical energy).
The essential objective of the study is the comparison among different solutions, in order to provide a recommendation by considering their reliability, technical feasibility and their environmental impact.
The specific demands of the client are illustrated by the phrase «the whole building must rise at standards which may not yet have been reached in the UK». This conception leads to the following characteristics: all the services are placed on a common architectural support; the whole building, including installations and the energy units is part of a public space, so a special attention is giving for the integration within the environment; all the services must be flexible, in order to provide a future modification.
The general design data for this building are presented in Table 1.
Because of the great interdependency and the numerous functioning conditions, the design and the estimation of gas and electricity loads are very difficult. In order to facilitate the dynamic analysis of the installations, we simulated the functioning of each solution by using specific modules in Simulink.
The dimensioning of each equipment for energy supply was done for each solution. After dimensioning, the economic calculation was done, including investments costs, maintenance costs and annuities.
The dimensioning of the energy supply equipment and of the storage vessels was based on the variations in heating, cooling and electricity loads of the complex building for two different design days, considered as the most disadvantaged cases:

- the summer normal working day;
- the winter week-end day (because in this particular day we can not use the important heat released by the technological processes inside the building, in the production zone).

All the data regarding the variations in heating, cooling and electricity loads are presented in figure 1 (for a summer normal working day) and in figure 2 (for a winter week-end day).
The heating/cooling are generally (90%) made by using warm/cool air. Air conditioning with humidification for VIP zone, computer rooms, industrial zones and dehumidification for some industrial zones was also provided.
The main technical-economical results for six solutions for energy supply, obtained by using a numerical modelling in Simulink are presented in Table 2.

Table 1. General design data

Industrial and administrative area	40000 m^2
Museum area	4000 m^2
Number of employees	1000-2000 persons
Number of visitors	300-350 persons
Kitchen / Restaurant	600 persons/day
Exterior temperature	summer: 30°C dry bulb, 20°C wet bulb winter: – 4°C (saturated air)
Consumption hot water temperature	minimum: 55°C, maximum: 60°C
Heat transfer coefficients	walls: 0.45 W/(m^2·K), frontal face glazing: 1.80 W/(m^2·K)
Interior temperature (offices)	winter: minimum 20°C
Interior temperature (comfort zone)	22°…26°C
Cooling system for offices	radiant ceiling

Table 2. Energy supply solutions.

	Description	T.I.C. [Euro]	T.I.A. [Euro]	A.F.C. [Euro]
1	Boilers (1 x 1000 kW, 1 x 800 kW, 1x 400 kW) Vapour compression refrigeration machines (4 x 750 kW) Cool storage vessel (1 x 250 m^3) + A.L. (Artificial lake)	1,871,500	148,549	1,190,359
2	Boilers (1 x 1000 kW, 1 x 600 kW) Fuel cells (2 x 350 kWe) Absorption refrigeration machine (1 x 500 kW) Vapour compression refrigeration machines (3 x 860 kW) Cool storage vessel (1 x 250 m^3) + A.L.	2,620,750	230,242	1,203,256 (without E.) 1,132,849 (with E.)
3	Boilers (1 x 600 kW, 1 x 400 kW) Internal combustion engines (2 x 300 kWe) Absorption refrigeration machine (1 x 900 kW) Vapour compression refrigeration machines (3 x 750 kW) Cool storage vessel (1 x 250 m^3) + A.L.	2,160,750	182,342	1,150,346 (without E.) 1,125,177 (with E.)
4	Boilers (1 x 1000 kW, 1 x 800 kW, 1x 400 kW) Cool storage vessel (1 x 250 m^3) + Phreatic water	1,514,000	127,227	1,045,670
5	Boilers (1 x 1000 kW, 1 x 600 kW) Fuel cells (2 x 350 kWe) Absorption refrigeration machine (1 x 500 kW) Cool storage vessel (1 x 250 m^3) + Phreatic water + A.L.	2,495,750	236,083	1,121,937 (without E.) 1,033,985 (with E.)
6	Boilers (1 x 600 kW, 1 x 400 kW) Internal combustion engines (2 x 300 kWe) Absorption refrigeration machine (1 x 900 kW) Cool storage vessel (1 x 250 m^3) + Phreatic water + A.L.	2,076,250	187,073	1,064,701 (without E.) 1,029,356 (with E.)

T.I.C. - total investment cost, T.I.A. - total investment annuity, A.F.C. - annual functioning cost
E. – electricity selling

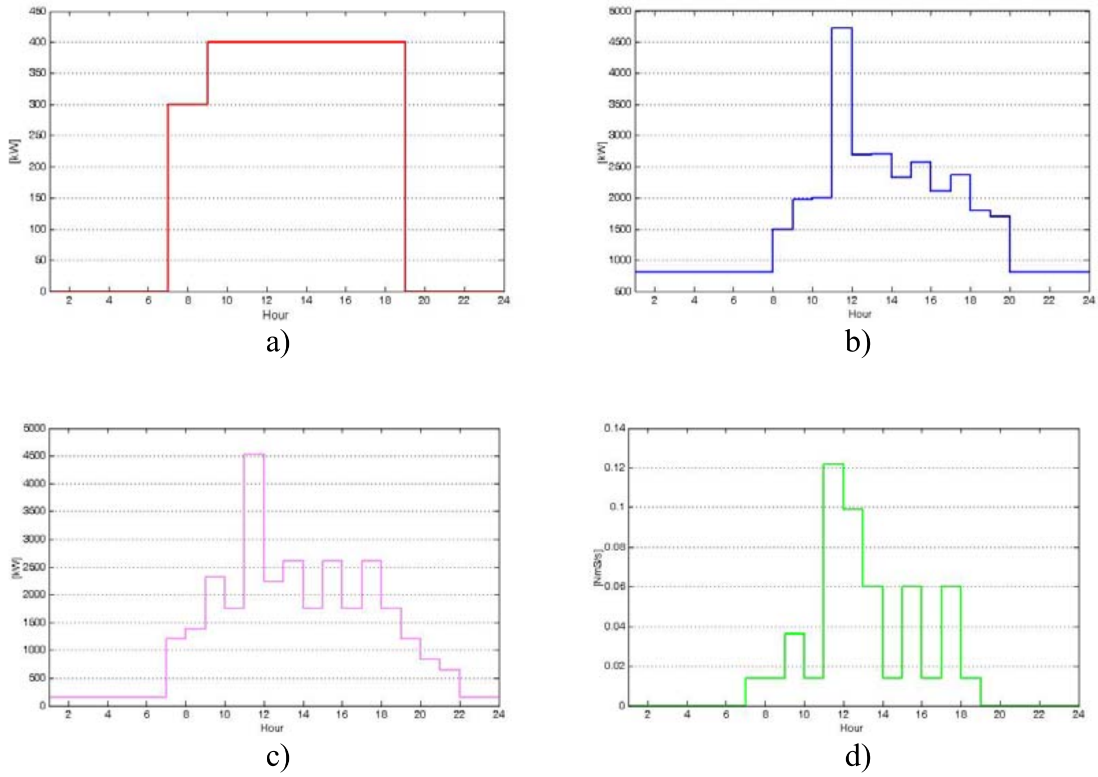

Figure 1. Loads for a summer day – normal working day.
a) Heating, b) Refrigeration, c) Electricity, d) Natural gas.

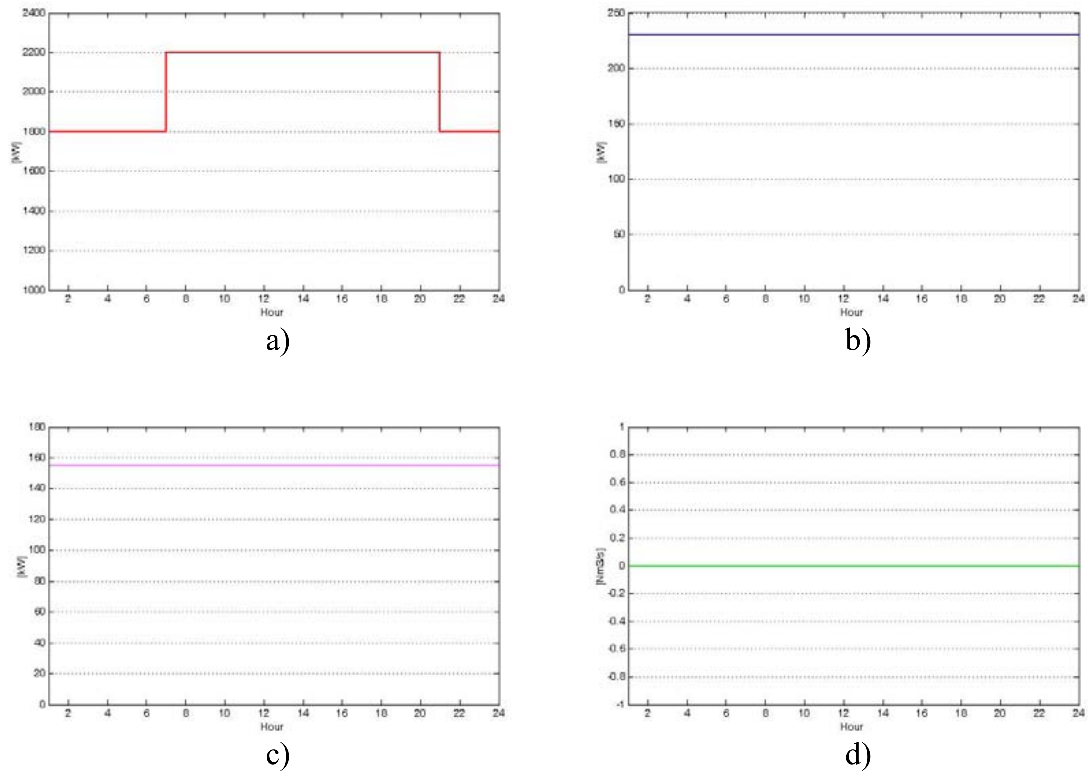

Figure 2. Loads for a winter day – week-end day.
a) Heating, b) Refrigeration, c) Electricity, d) Natural gas.

ANALYSIS OF THE ENERGY SUPPLY SOLUTIONS FOR THE COMPLEX BUIDING

An energy supply system must produce each moment all the energy needed by the consumer, without overproduction (that is lost when the thermal storage is not used). The system must have the capacity of self regulation, in order to follow the variations in the energy loads.
The primary energy in use is the electrical energy – 2 high-voltage lines of 8 MWe each, and the natural gas, at a flow rate capacity of 2100 Nm^3/h.
The boilers are modern ones, for the production of heating water at 65/45°C temperature. The boilers were equipped with modular combustors.
The cogeneration was considered by using internal combustion engines (as a “classical” solution) and fuel cells (as a future alternative, approved by the European Union).
The phreatic water is available at a level of 6°C, and can be used for cooling.
The cooling is designed for producing cold water at 8/14°C temperature, by using vapour compression refrigeration machines and an absorption refrigeration machine.
The low level heat recovery is taken into account for the air conditioning design cooling load.
The medium level heat recovery is considered from the technological processes inside the building: the compressed air, the vacuum units, the autoclaves, the motor benchmarks units.
An artificial lake was located inside the building complex, as an innovative design solution, leading to energy sparing, a beneficial impact on the environment by means of a non-polluting effect and an ambient harmonisation, with remarkable aesthetic results. The thermal modelling using Simulink shows the feasibility of using the lake as a cooling source for the refrigeration system.
The solutions 1, 2, 4, 5 are presented below, in the figures 3 to 6. The solutions 3 and 6 are schematically the same with solutions 2 and 5, the difference consisting in replacing the fuel cells with internal combustion engines.

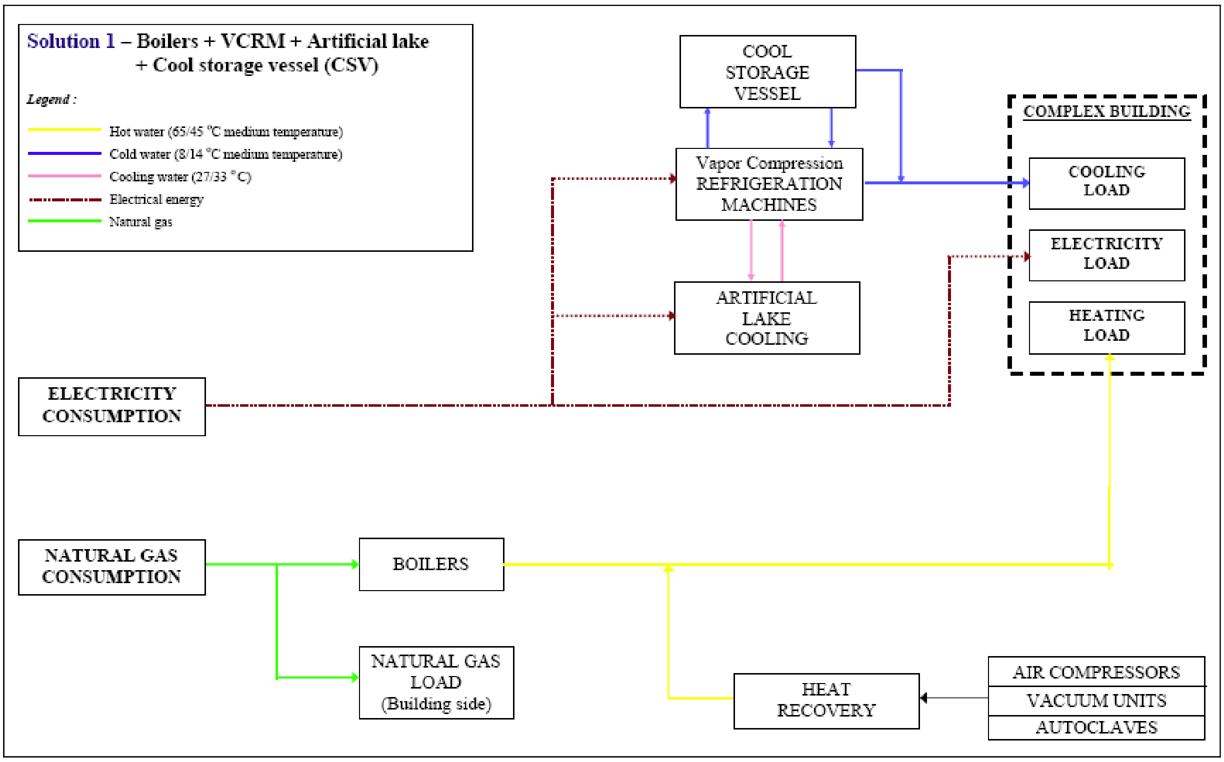

Figure 3. Solution 1 (Boilers + Vapour compression refrigeration machines + Artificial lake + Cool storage vessel).

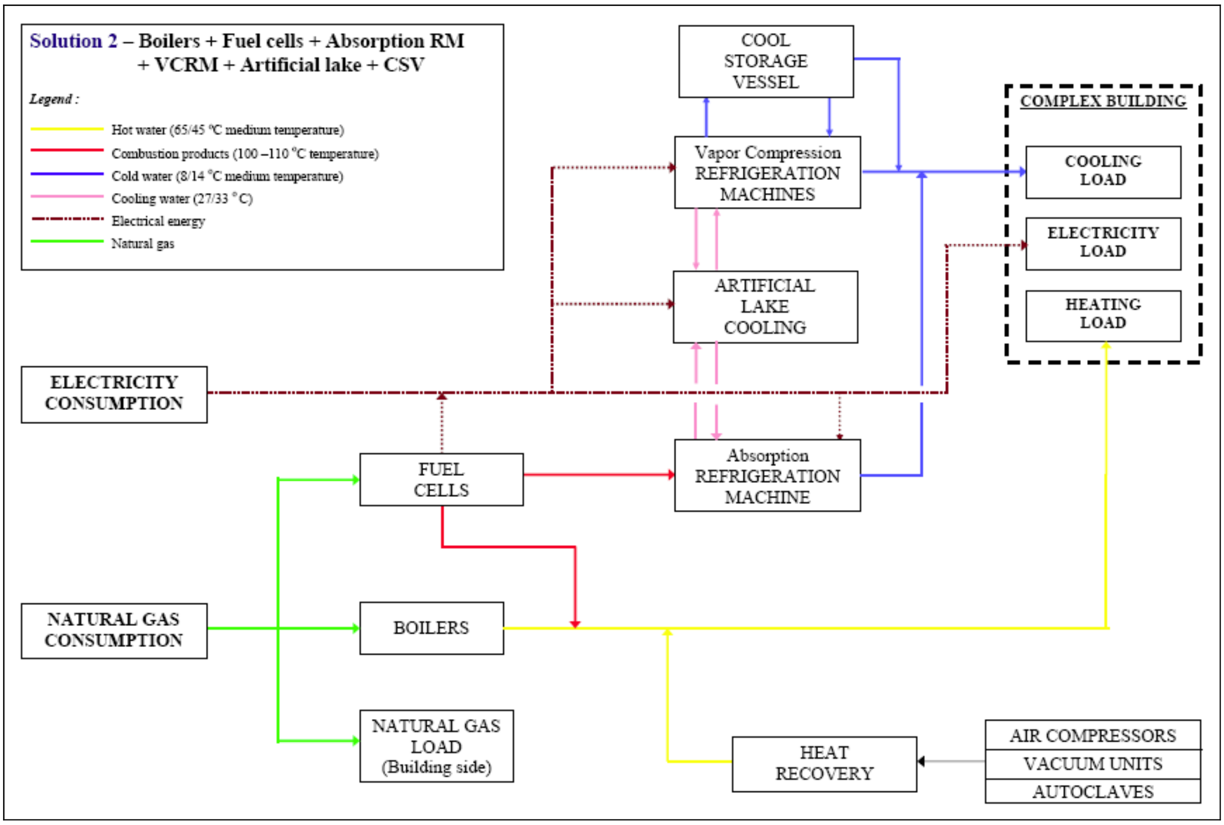

Figure 4. Solution 2 (Boilers + Fuel cells + Absorption refrigeration machine + Vapour compression refrigeration machines + Artificial lake + Cool storage vessel).

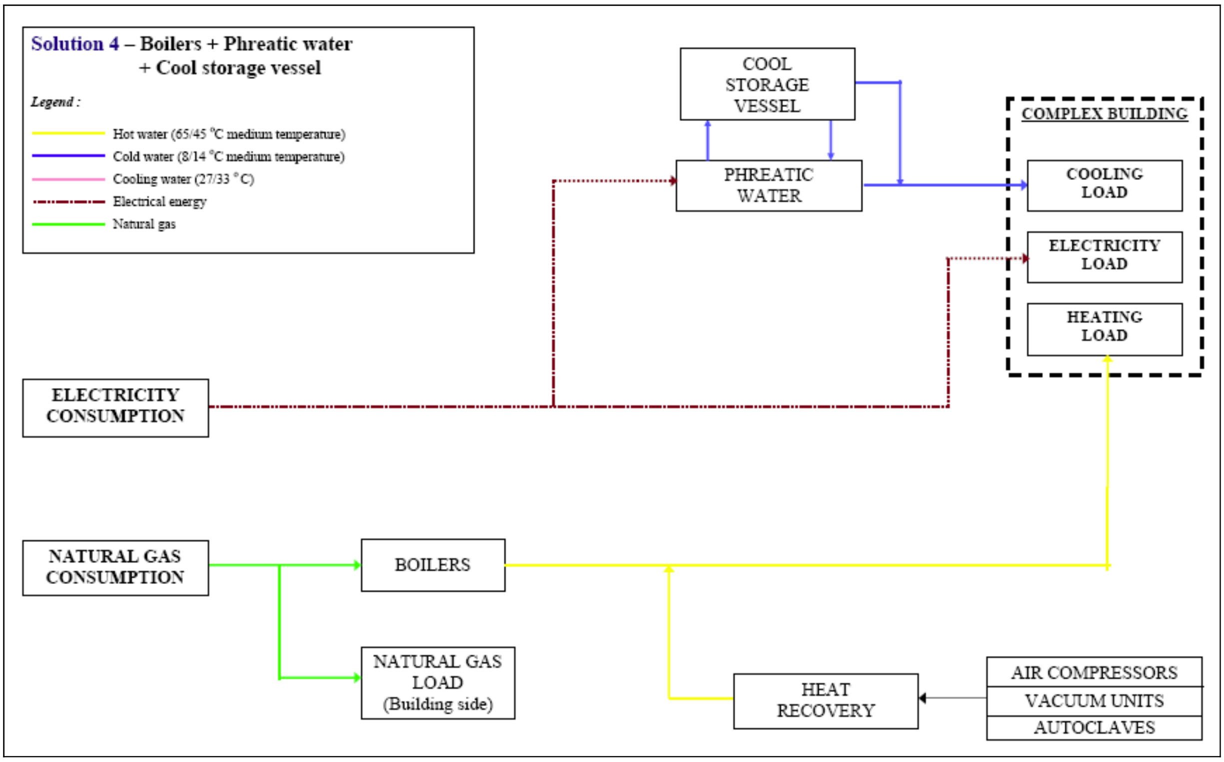

Figure 5. Solution 4 (Boilers + Phreatic (underground) water + Cool storage Vessel).

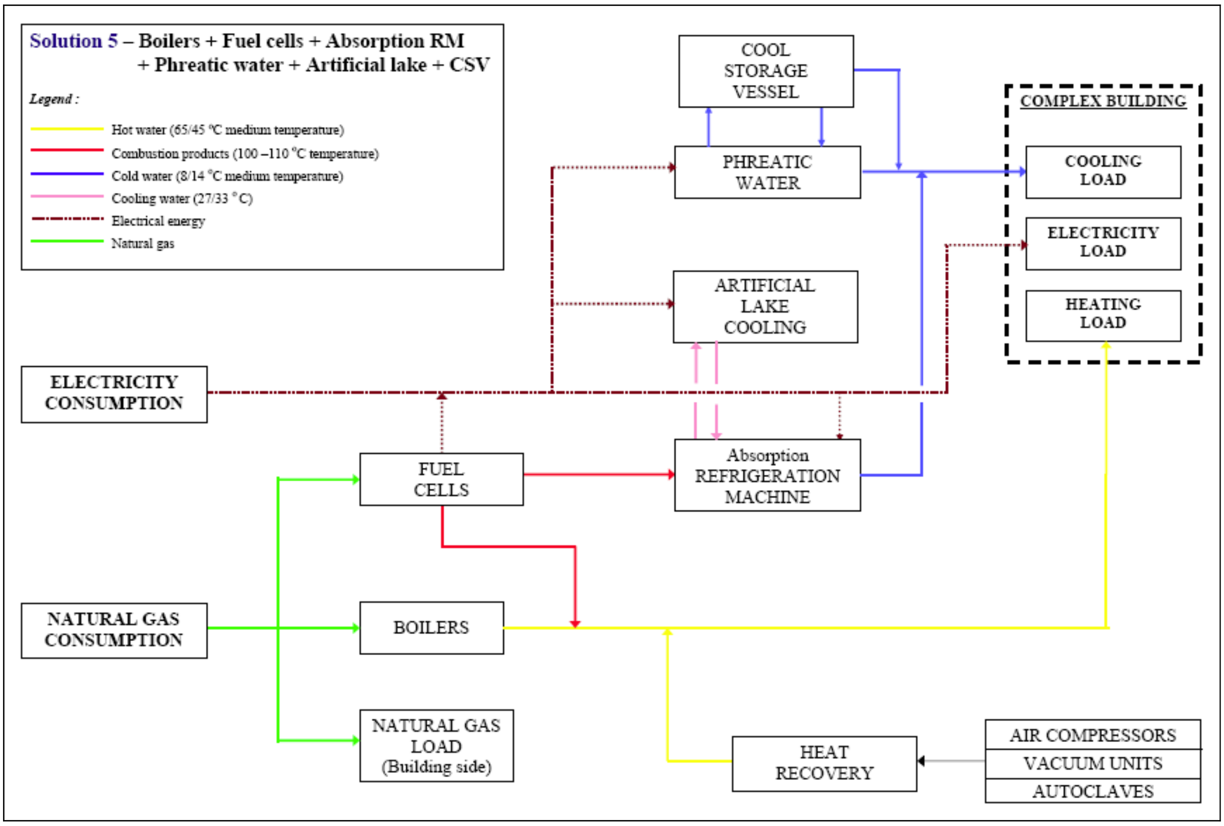

Figure 6. Solution 5 (Boilers + Fuel cells + Absorption refrigeration machine + Phreatic water + Artificial lake + Cool storage vessel).

ECONOMIC EVALUATION OF THE PROPOSED SOLUTIONS

The economic evaluation plays an important role in choosing the right solution, especially if balancing the present and the future needs and costs. Therefore, the economic evaluation was made taking into account the investments costs and the maintenance and operation costs for a long period.
In this study the economic evaluation was done concerning the building investment, the total investment for equipments, the total investment, the total annuity, the fixed costs, the maintenance costs, the total costs of primary energy.

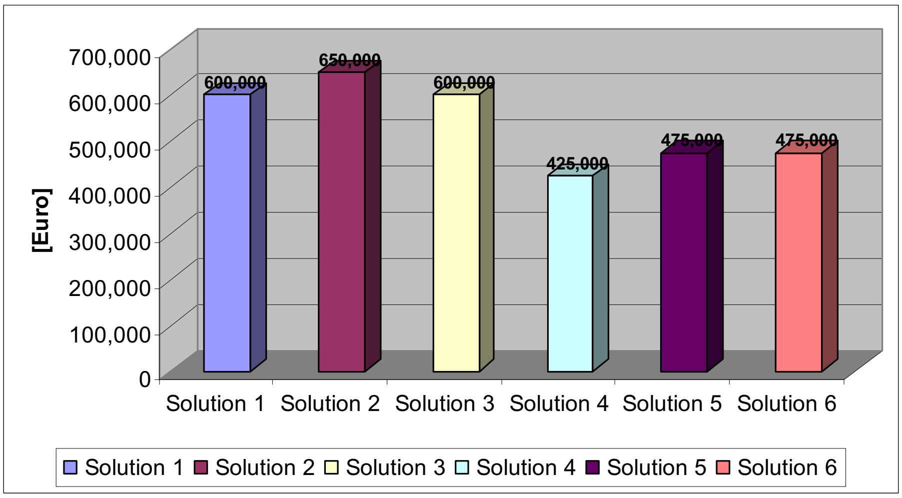

Figure 7. Building investment.

Because the equipments used for different solutions have different life-times, the comparative economic analysis regarding the investments costs is done by considering a mean life-time. In this respect, we calculated a total annuity for each solution. This value is the sum of the mean annuity of the installations (calculated with a 15 years life-time) and the building annuity (calculated for a 50 years life-time).
We can observe in figure 7 that the solutions utilising phreatic water (4, 5 and 6) implies the lower building investment. The solutions 1, 2 and 3 need bigger constructions because of the dimensions of supplementary equipment.
Concerning the cost of the equipments (figure 8), the solution 2 and 5 are clearly more expensive than the others, because of the high price of the fuel cells used. The most inexpensive solutions are the classical ones (1 and 4, with boilers).

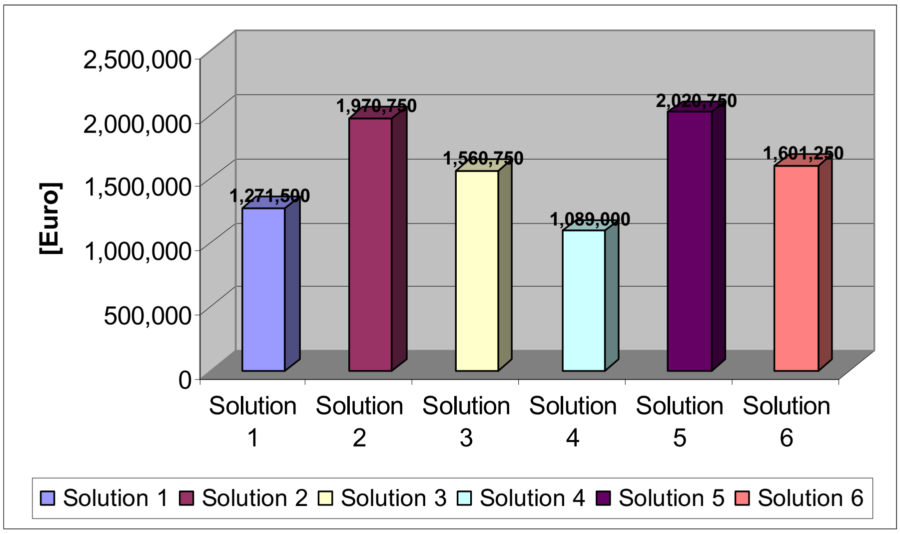

Figure 8. Total investment for equipments.

Finally, the total investments costs (figure 9) reveals three different categories: the expensive solutions 2 and 5 (with fuel cells), the medium-price solutions 3 and 6 (with internal combustion engines) and the cheap solutions 1 and 4 (with boilers).

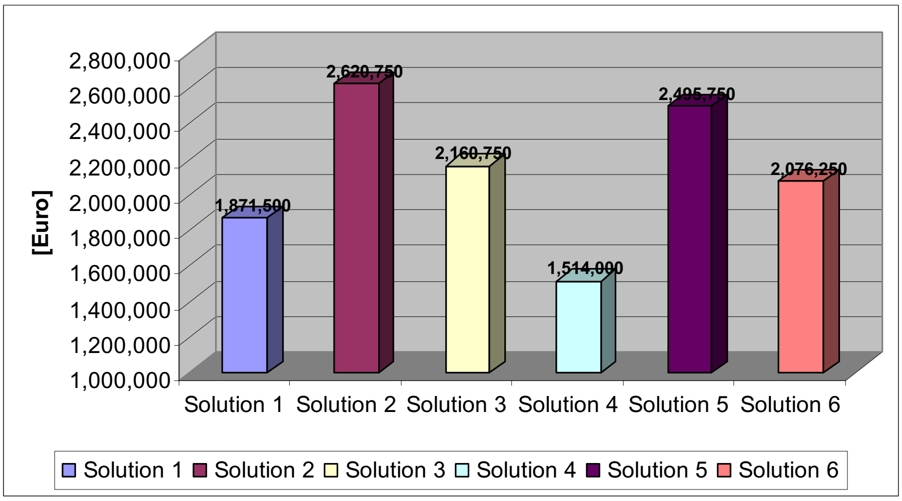

Figure 9. Total investment.

It must be noticed that the calculation of the total costs of primary energy was done for two different hypotheses: without and with selling of electrical energy.

A first observation is that the solutions using phreatic water need less primary energy than the other solutions. Another observation is that the solutions using cogeneration (with fuel cells / internal combustion engines) are more economical than the classical solutions (with boilers). The cogeneration with fuel cells needs less primary energy. As an example, the solution 2 (with fuel cells) has a total annual primary energy cost of 754500 Euro, and the solution 1 (classical solution with boilers) has a corresponding cost of 913200 Euro; it results an economy of primary energy costs of 158700 Euro, that is 18% !

CONCLUSIONS

In order to choose an optimal solution, one must consider different weighting concerning the essential criteria. In our case, we considered the following criteria:

a) economic criterion (investment costs and total costs);
b) legal criterion (financial support for fuel cells);
c) environmental criterion (environmental impact);
d) marketing criterion (commercial impact).

We consider the marketing and the environmental criteria as the most important ones (for the case of a client with a high financial power).
The solutions 1 and 4 raise no interest for a client interested in a commercial impact; these solutions are also expensive in time.
The solutions 2, 3, 5, 6 remain as the basis for choosing the optimal decision.
The solutions with fuel cells (2 and 5) have higher total annuities, and the lower costs of primary energy. The solution 5 has a negative commercial impact, so the final recommended solution for this application and for this particular client was the solution 2.

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A Two Year Measurement of Energy Consumption And Indoor Temperature of 9 Houses in a Cold Climatic Region of Japan

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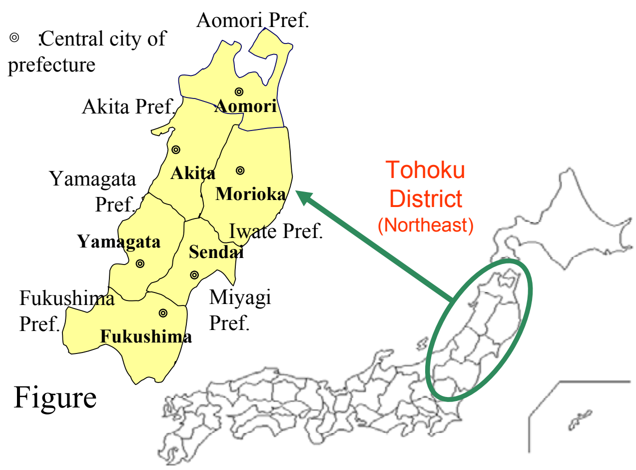
SUMMARY

The energy consumption and indoor temperature of 9 housing units sited in the northern region of Honshu Island, Japan were investigated for a full two years from Dec. 2002 to Nov. 2004. Three of the houses were installed with all-electric equipment. The annual and daily energy consumption profiles of two typical houses were analyzed, and the annual energy consumption per house was found to range from 40GJ/year to 120GJ/year. In some houses, energy consumption during the second year decreased due to a rise in energy saving awareness. The results indicated that the characteristics of energy consumption were not only greatly influenced by regional climate but also by the use of household equipment and lifestyle.

INTRODUCTION

In line with the recognition that a sustainable global environment is the most important environmental issue, global warming must also be handled urgently. As a result of improved living standards throughout the world, energy consumption by households is also growing. In Japan, the average annual growth rate of residential energy consumption was 4.2% from 1965 to 2004 (Handbook 2006). For this reason, it is important to take immediate action to minimize energy consumption by the housing sector.
The objective of this study is to understand the current condition of energy consumption and thermal environment in the daily life of households located in the northern region of Honshu Island, Japan. Detailed measurements of energy consumption for each end use as well as indoor temperature and humidity were carried out over a period of two years.



Figure

MEASUREMENTS

Outline of the houses

In this study, 9 houses located in the northern region of Honshu Island, Japan were investigated (Fig. 1). The outline of these houses is shown in Table 1. Seven detached houses and two apartments in Miyagi, Iwate, Akita and Fukushima prefectures were involved in this investigation. All the detached-type houses are made of wood and the two apartments are built by SRC or RC. The floor areas of the detached houses and apartments are 110~ 180m^2 and 70~ 80m^2, respectively. When these houses were newly constructed, they were well insulated and airtight. The value of heating transmission rate was calculated based on the design plans and components. The value of equipment leakage areas per floor area was measured by the fan pressurization method. In addition, the main energy sources were kerosene, electricity and gas. The three houses investigates used only electricity. In these houses, electric thermal storage heating equipment was installed.

Items of measurement

The major items investigated in the measurement are given in Table 2. These include energy consumption, thermal environment, capacity of electrical appliances and the occupants' lifestyle, etc. The measurement intervals for electricity consumption, gas consumption, kerosene consumption, and temperature were 1 minute, 5 minutes, and 15 minutes.

Categories of energy consumption

End use of energy consumption was classified into six categories, i.e. space heating/cooling, hot water supply, kitchen, audio visual & information, healthcare, lighting & others. The category of "lighting & others" means the energy consumed for lighting and unidentified usage. Receipts for electricity, gas, and kerosene consumption were used to compensate for missing data in situations where the measuring instruments were not functioning.

Table 1. Description of houses investigated (Survey period: from Nov. 2002 to Nov. 2004)

No.		Location	Number of occupants	Floor area [m^2]	Heat loss coefficient [W/m^{2}K]	Equivalent leakeage area [cm^2/m^2]	Construction	Energy sources by usage			
								Space heating	Space cooling	Hotwater supply	Kitchen
Detached house	D01	Miyagi	5	159.0	1.88	0.85	Wood	Kero.2*	Elec.	Kero.	Elec.
	D02	Miyagi	3	115.9	1.72	0.76	Wood	Kero.4*	Elec.	Kero.	Gas
	D03	Akita	3	109.3	1.77	0.87	Wood	Elec.5*	Elec.	Elec.	Elec.
	D04	Akita	3	141.6	1.79	0.77	Wood	Elec.3*	Elec.	Elec.	Elec.
	D05	Akita	4	160.6	1.84	2.20	Wood	Kero.2*	Elec.	Kero.	Gas
	D06	Iwate	4	140.0	1.01	0.70	Wood	Elec.3*	Elec.	Elec.	Elec.
	D07	Iwate	4	178.0 1*	1.16	0.40	Wood	Kero.2*	Elec.	Kero.	Elec.
Apartment	A01	Fukushima	3	72.3	2.47	1.74	SRC	Elec.5*	Elec.	Gas	Gas
	A02	Fukushima	3	78.0	-	0.47	RC	Elec.5*	Elec.	Gas	Gas

Notes Kero.: Kerosene, Elec.: Electricity, 1*:It includes 58m^2 of undreground spaces

Space heating equipment: 2*: Hot water heating panel, 3*: Electric thermal storage heating equipment, 4*: Portable kerosene heater, 5*: Air conditioner

Table 2. Measurement items

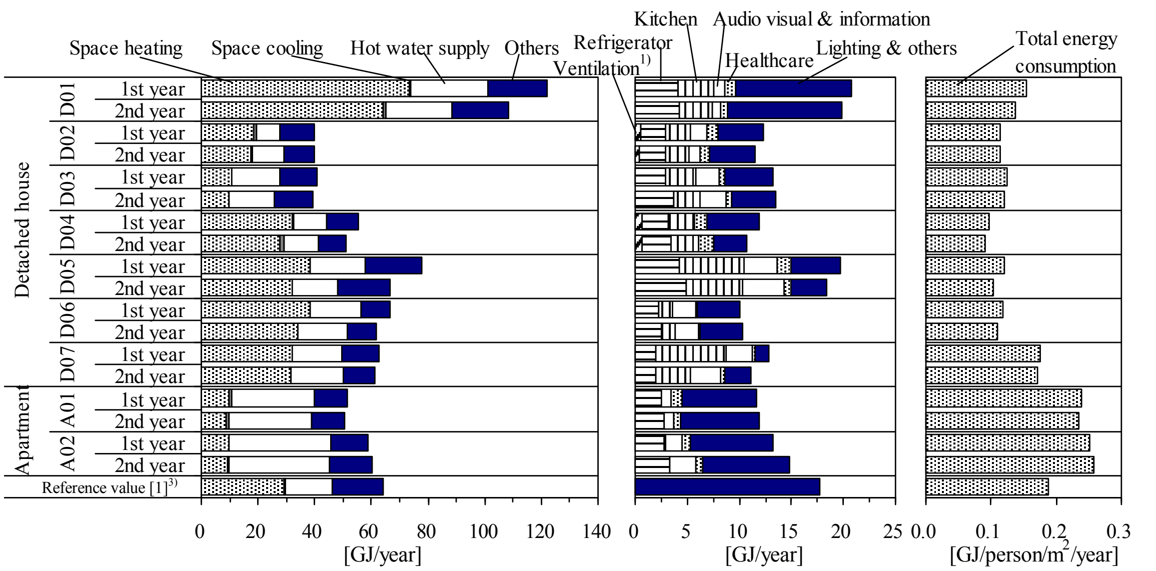
Investigated		Measurement items
Measurement	Electricity	Home energy consumption recording system with interval 1 minute
	Gas	Digital camera signal data logger with interval of 5 minutes
	Kerosene	Fine flow rate fuel oil meter with interval of 5 minutes
	Temperature & relative	Small sensor and data loggers with interval of 15 minutes
Questionnarire		Life-style, Energy saving consciousness, Annual income, Usage frequency and Capacity of equipment
Interview		Family structure

ENERGY CONSUMPTION AND TEMPRATURE OF ALL INVESTIGATED HOUSES

The annual energy consumptions of nine houses are shown in Fig. 2, with the statistical reference values from literature (Handbook 2003) for the Tohoku District in Japan. The periods of the first and second years were from Dec. 2002 to Nov. 2003 and from Dec. 2003 to Nov. 2004, respectively. The statistical reference value is the average of 2002. Energy consumption by end use is classified into four categories; space heating, space cooling, hot water supply and others (Fig. 2.-a). It is noted that the hot water supply includes gas consumption for cooking in the case of apartments. The category of "others" divided as ventilation, healthcare, lighting & others is shown in Fig. 2.-b. Energy consumption for audio visual & information includes consumption for lighting & others in house D04. The statistical reference value is not divided. The total energy consumption per person square meter is shown in Fig. 2.-c.

Difference between houses As shown in Fig. 2.-a, the annual total energy consumption per house varied between 40−120GJ/year. The major energy consuming activities for detached houses and apartments were space heating and hot water supply respectively. The energy consumption in D01 was the largest, which was nearly twice that of the reference value. This house consumed energy largely for space heating because they used it all day. In addition, the floor area was large and required more energy to keep the rooms warm. In contrast, the energy consumption of D02 and D03 was about two thirds of the reference value. In these two houses, occupants practiced energy saving awareness as revealed by a questionnaire. Fig. 2.-b shows the energy consumption of "others", which was used by refrigerator, kitchen & others. The characteristics of end use were different between houses.

Difference between detached houses and apartments Fig. 2.-a shows the trends of energy consumption according to end use for detached houses and apartments. The percentage of space heating and hot water supply in the case of detached houses were approximately 50% and 20% respectively, and 20% and 50% in the case of apartments. In both cases the energy



(a) Total energy consumption (b) Energy consumption of "others" (c) Total energy consumption per person per square meter

1) Ventilation: Electricity consumption for ventilation was measured only in D02 and D04
2) D04: Audio visual & information is included in lighting & others
3) Reference value [1]: Average value of the Tohoku district in Japan

Figure 2. Energy consumption by end use consumed for space heating and hot water supply was above 70% of the total energy consumption.

Comparison between the first and second year The results show that energy consumption in the second year decreased compared to the first year for most houses, especially in house D01, D04, D05 and D06. The rate of decrease in these four houses was 11%, 8%, 14% and 8% respectively. In these houses, energy consumption for heating decreased largely and it affected the decrease of total energy consumption. Based on the questionnaire survey, it was found that the energy saving awareness of the occupants in these houses became higher in the second year. One example of concrete action for saving energy was in house D01 where the warm water heating system was turned off when the house was unoccupied. In the case of D04, they turned the heating preset temperature down in the second year. In house D05, number of occupants changed from 4 to 3 because of husband-alone transfer since May 2004. It is considered as a key factor in decrease of energy consumption. In addition, space heating was used for the entrance, lavatory and toilet in the first year, but was not used for these spaces in the second year. In the case of house D06, electric thermal storage heating equipment was installed for the living room, dining room, bed room and Japanese-style room, and energized shorter hours in bed room and Japanese-style room in the second year.

Energy consumption per person per square meter Fig. 2.-c shows the energy consumption per person per square meter. The values for detached houses were less than those of apartments; approximately 0.12 and 0.23 GJ, respectively.

COMPARISON OF THE ENERGY CONSUMPTION OF TWO TYPICAL HOUSES

Outline of the two houses under investigation

The energy consumption of house D01 (Fig. 3) that used electricity and kerosene as sources of energy, is compared with house D06 (Fig. 3) that used only electricity. The energy consumption of houses D01 and D06 were the largest among the all-electrified houses and non-electrified houses. These two houses were located in Sendai and Morioka cities. The number of occupants in D01 and D06 were 5 and 4, respectively.

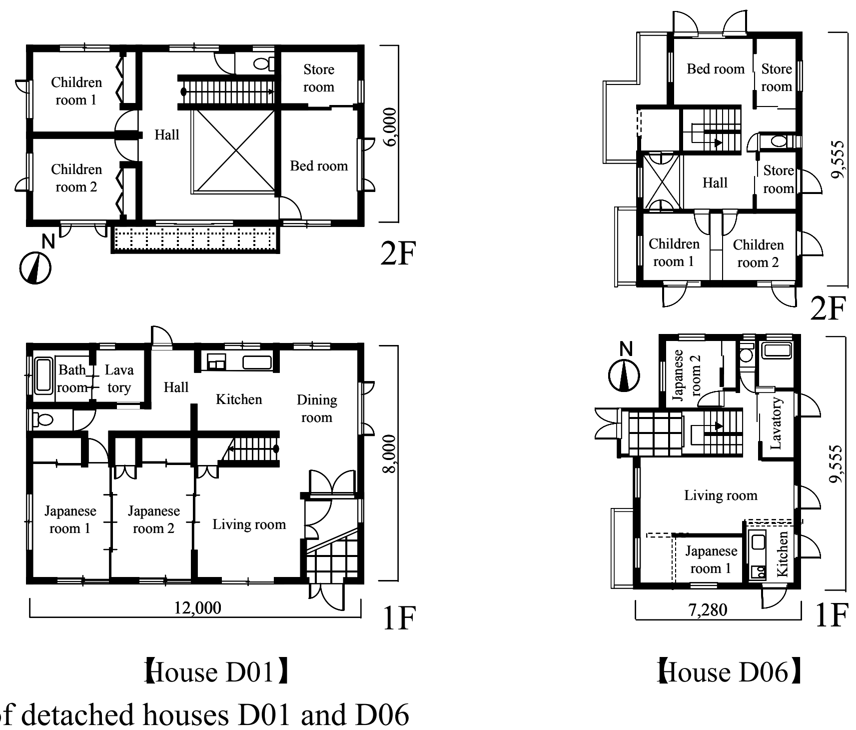


Figure 3. Plan of detached houses D01 and D06

Two-year profile of daily energy consumption

Profiles of daily amount of energy consumption and daily mean temperature of these two houses over a two-year period, from Dec. 2002 to Nov. 2004, are shown in Fig. 4. In house D01, the living room temperature changes in a range of 21˚C to 29˚C during the measured period, while that of house D06 is between 18˚C to 29˚C. Energy consumption in both houses increased in the winter due to space heating, but it is noted that the energy consumption of house D01 is about double that of house D06. The difference is because space hating was used entirely for house D01 but only partially for house D06. Another reason is that the value of heat loss coefficient of house D01 was 1.88 [W/m^{2}K] while house D06 was only 1.01 [W/m^{2}K]. The energy consumption of kitchen, audio visual & information, and healthcare, lighting & others are almost constant. However, the energy consumption of lighting & others changes with different season during the period of the investigation. The total energy consumption is very different due to the different lifestyles in both houses. In winter, the maximum difference of energy consumption between D01 and D06 is about 400MJ/day. On the other hand, in summer, the energy consumption of each house is about 100MJ/day.

Energy consumption profile on typical days

The profiles of energy consumption and temperature during three days in winter/summer are shown in Fig. 5. The middle day among three days was the coldest/hottest day with the lowest/highest outdoor air temperature during the investigation period. Each diagram of Fig. 5 is divided into three sections: the higher section shows the outdoor air and living room temperature, the middle section gives the energy consumption oh three uses (space heating/cooling, hot water supply, and others), and the lower section illustrates the energy consumption of the other items (kitchen, audio visual & information, healthcare, and lighting & others). The energy consumption is shown at intervals of 15 minutes.

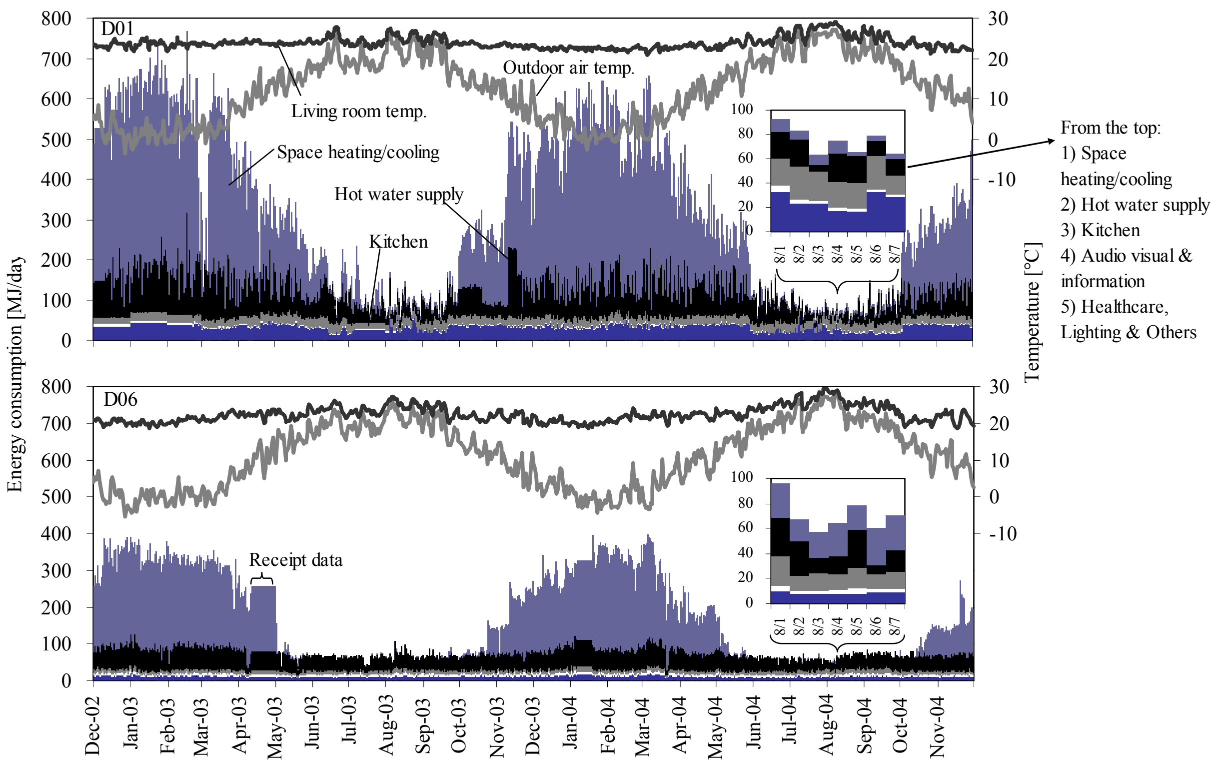


Figure 4. Two-year profile of daily energy consumption and temperature

The coldest day The temperature of the living room in D01 and D06 is almost constant during the coldest period, at 23˚C and 20˚C, respectively. This is due to the fact that the space heating is used all day in both houses. However, the energy consumption profiles are significantly different between the two houses. As the hot water heating panel was operated for the whole day in house D01, there was no peak energy consumption (except for hot water supply). In house D06, the energy was mainly consumed from 11 pm to 7 am by the thermal storage heating equipment using nighttime electric power. As shown in the lower section of diagram in Fig. 5, there are several peaks of energy consumption due to cooking activities.

The hottest day The temperature change in the living room ranged from 27˚C to 30˚C for house D01 and 25˚C to 30˚C for house D06. There was no obvious difference in the energy consumption trends of house D01 and D06, except energy consumption for hot water.

Ratio of energy consumption end use The ratio of energy consumption by end use during three day is shown in Fig. 5. In the coldest day, the percentage of energy consumption for space heating and hot water supply are 70% and 15% approximately in both houses. On the other hand, in the hottest day, the energy consumption for space cooling is small. In house D01, the energy consumption for space cooling and hot water supply share the same percentage of 15%. In house D06, those values are 10% and 50%, respectively.

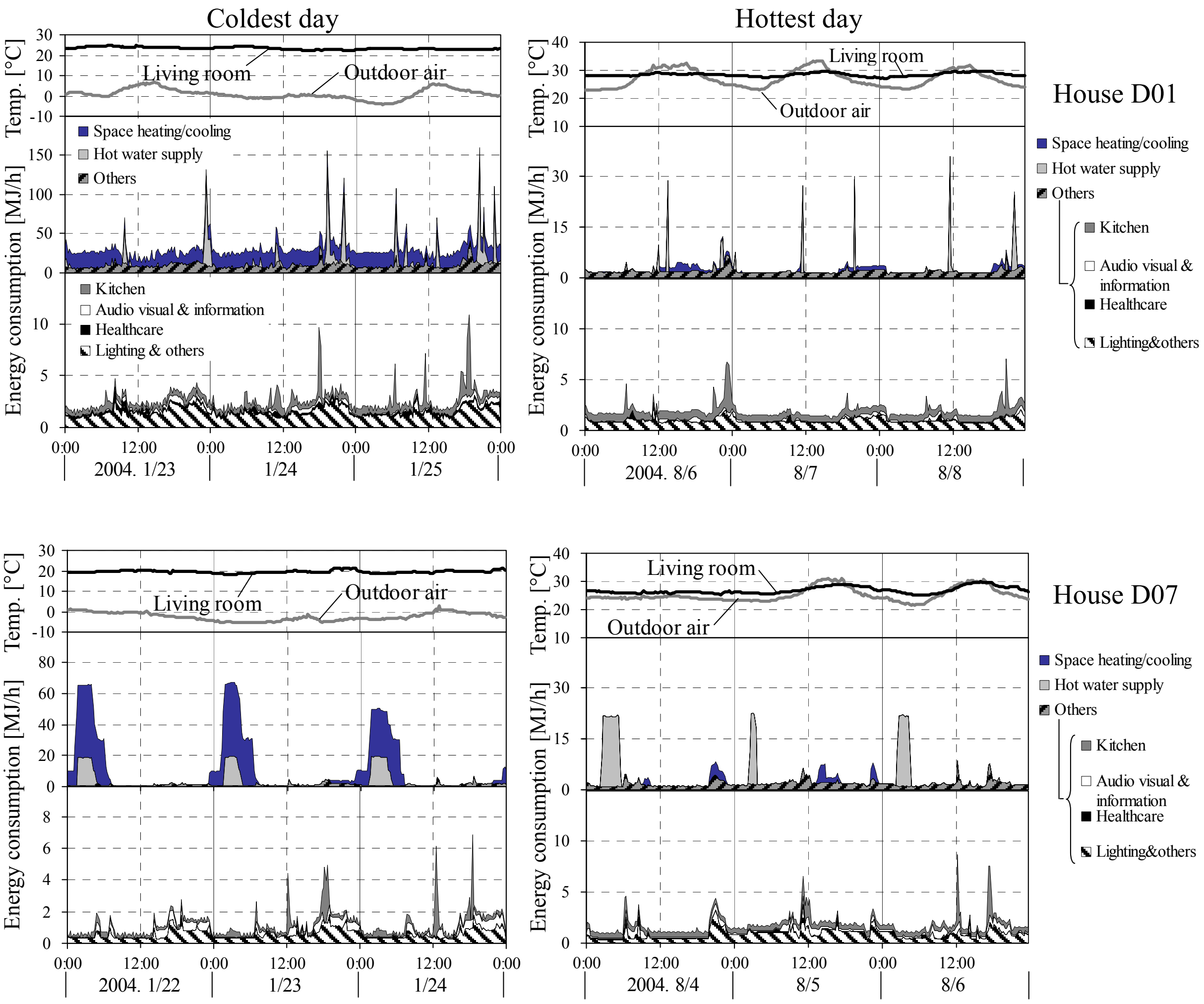


Figure 5. The change of temperature and energy consumption for three days, when the middle of three days is the coldest or hottest

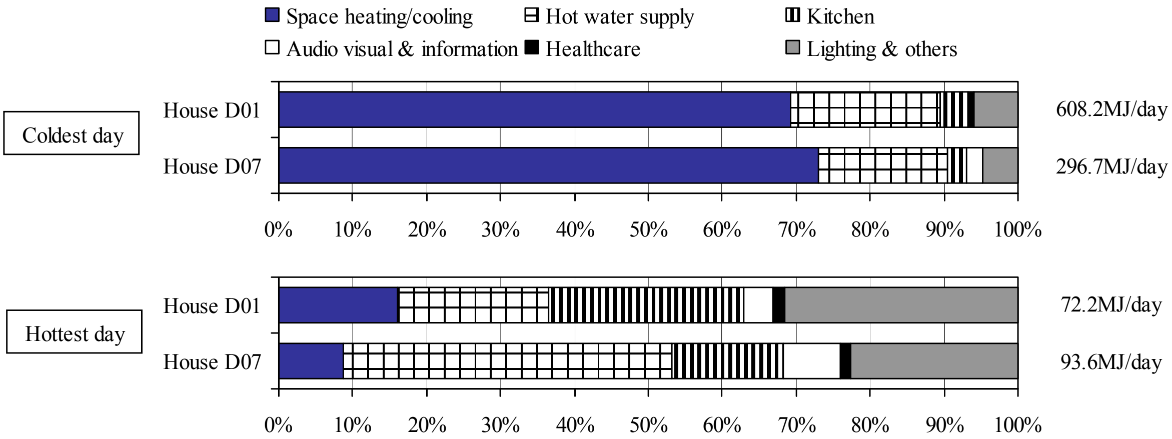


Figure 6. Ratio of end use energy consumption during three days including the coldest/hottest day

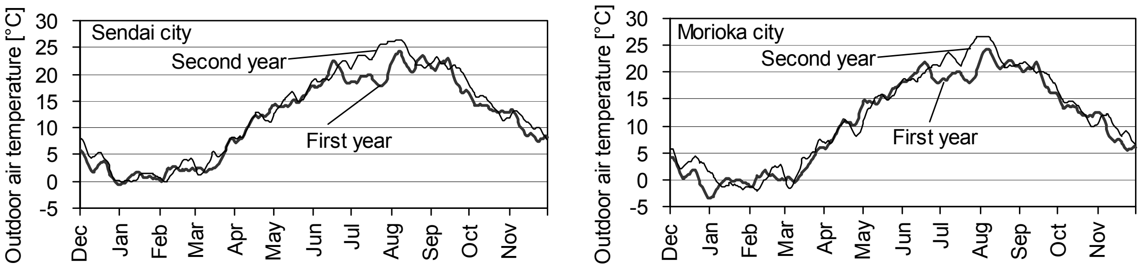


Figure 7. Comparison of outdoor air temperature between two years in two cities

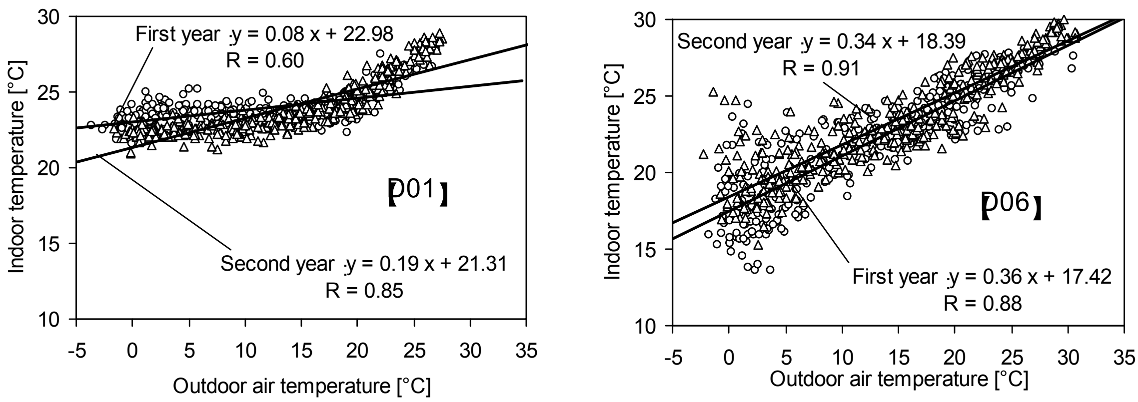


Figure 8. Relationship between daily average temperatures of indoor and outdoor air

Indoor and outdoor air temperatures

(1) Comparison of outdoor air temperatures between two years

Fig. 7 shows comparison of outdoor air temperatures between two years in two cities where the houses located. Temperatures in winter stayed about same in both cities. On the other hand, in summer, there were big differences. It was cold in first year and extremely hot in second year.

(2) Relationship between indoor and outdoor air temperature

The relationships between daily living room and outdoor air temperature in two houses are shown in Fig. 8. As to house D01, the living room temperature in the first year is commonly stable although the outdoor air temperature is remarkably varied. But in the second year, indoor temperature changed under the influence of outdoor air temperature. Based on the questionnaire survey, preset temperature of heating/cooling turned down/up in the second year. In the case of house D06, the relationship in the first year was almost the same as the second year.

CONCLUSIONS

Energy consumption and temperature for 9 houses

(1) Annual energy consumption per house varied from 40 to 120 GJ/year. The major energy consuming activities for detached houses and apartments were space heating and hot water supply respectively.
(2) Energy consumption in the second year decreased compared to the first year for most houses. The energy-saving consciousness of the occupants became stronger in the second year than that in the first year.
(3) Energy consumption per person per square meter for detached houses was less than those of apartments; approximately 0.12 and 0.23 GJ, respectively.

Energy consumption of two typical houses

(1) Energy consumption increases significantly in winter in both houses due mainly to space heating. The energy consumption of kitchen, audio visual & information, healthcare, lighting & others are almost constant. The annual profile was similar between the two years.
(2) In house D01, the living room temperature in the first year is commonly stable although the outdoor air temperature is remarkably varied. But in the second year, indoor temperature changed under the influence of outdoor air temperature. Based on the questionnaire survey, preset temperature of heating/cooling turned down/up in the second year.

This study found that energy consumption patterns of different houses varied significantly. It indicated that there is high energy saving potential by the change of lifestyle. The second year's energy consumptions of some of the investigated houses were lower than that of the first year. It was mainly due to the increased awareness of the inhabitant on the importance of energy saving and improved knowledge on energy saving lifestyles. The findings of this study serve as the basic data for further investigations in energy consumption saving strategies in Japan.

ACKNOWLEDGEMENT

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The Field Measurement of The Sustainable Office Building with The Environmental Adjustable Systems

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SUMMARY

HVAC load weighs a large portion of overall building energy consumption. A low storey building of 20,000m2 floor area adopted a task ambient conditioning system (TAC) to reduce HVAC load while maintaining work environment. Other design techniques such as double-skin facade and structural thermal storage system, were also applied and the TAC system achieved comfortable personal work-environment with reduced demand.

INTRODUCTION

Generally, HVAC system load is one of the major energy consumption sources of a building. Reduction of such load shall play a significant role on the environment effect of the building but often in return of degraded work environment. This paper introduces a case of a low storey building with floor area of 20,000 m2 for its environmentally conscious design.

Project Outline

The building is a headquarter office in the suburbs of Tokyo.
The building is a four storey building with typical floor plan consisting of two wing zone with an atrium in between. Each wing is pillar-less and 33.6 m x 38.4 m / approximately 1500 m2. South Façade, typical floor plan, and outline of the building are shown in Figures 1, 2, and Table 1.
This building is built on a quake-absorbing structure. The main air conditioning method is under floor air distibution.

Figure 1. Facade of building

Table 1. Outline of building

Building data	Building type	Office
	Site area	4,782 m^2
	Gross area	19,169 m^2
	Floors grade	B1F-4F
	Max height	25.81m
	structure	SRC
M/E data	Heat source system	Ice storage system with Gas fired absorption chilled and hot water unit
	Air conditioner system	Under floor air-conditioning system only in interior zone
	Power supply	High voltage electric supply (main and sub)
	Supply of water system	Public water supply and the utilization of rain water and the well water

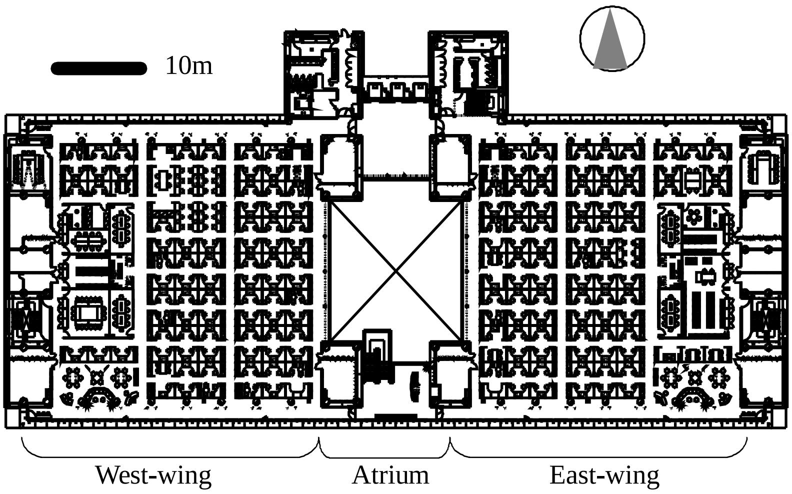

Figure 2 Typical floor plan

System Outline of Double-skin Facade

Double-skin façade (DS) consists of whole surface glasses of full storey height. Its advantages are to make the occupant unaware of the confinement and maximum utilization of natural lighting. In Japanese climate, the double-skin glass is required to satisfy various functions such as the discharging the heated air between the glasses in the summer, the thermal insulation in the winter and the natural ventilation in spring and autumn. These functions are realized by the automatic control ventilation dampers on Double-skin facade. DS has various ventilation dampers such as the top ventilation dampers, the bottom ventilation dampers, the top ventilation dampers of atrium, and each floor ventilation dampers.

During the summer, the top and bottom dampers are opened, to exhaust heat in DS by the natural ventilation and the thermal load to the building is decreased. The dampers are closed during the winter for better insulation.

In intermediate seasons such as spring and autumn, the building can be naturally ventilated by opening the bottom ventilation dampers, floor ventilation dampers, and top ventilation dampers on the atrium.

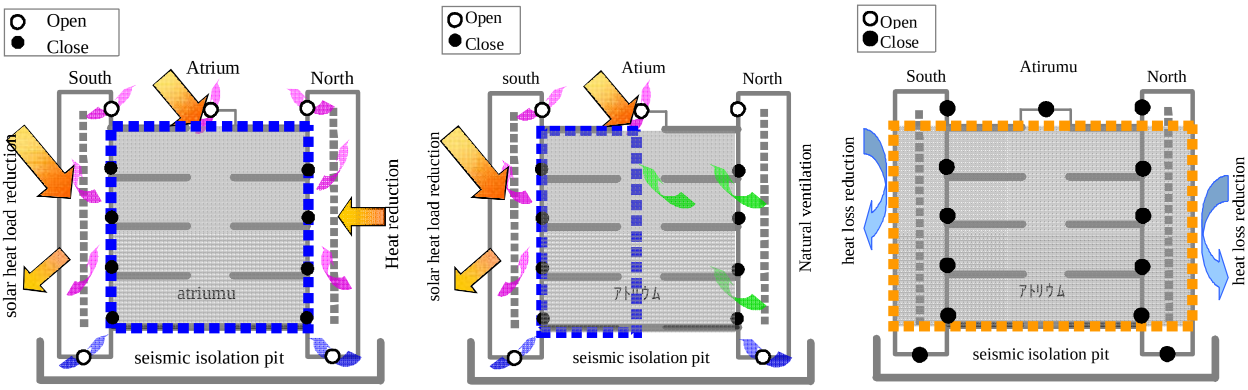

Figure 3. System outline of DS

Design Concept of Air-conditioning System

The air flow from floor distribution system reaches the occupied zone (FL+1,700) to form the temperature stratification between the occupied and unoccupied zone. (Fig.4)
Task air conditioning was introduced to the cubicles parted by partitions equipped with air outlets.
Moreover, by setting the main aisle in the DS, the main aisle serves as a perimeter buffer-zone. As a result, the building does not require dedicated air-conditioning system for perimeter.
In this building, the air-conditioning that suits with the special characteristic of a high ceiling and a wide space was designed by combining a vertical zoning and horizontal zoning.

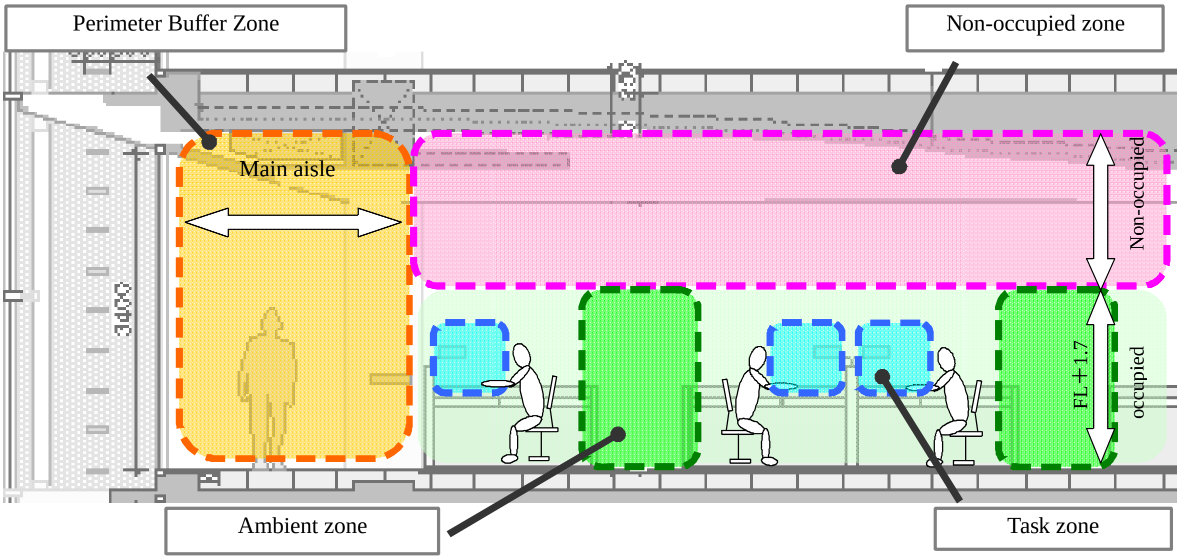

Figure 4. Outline of air-conditioning system

Environmental Assessment (CASBEE)

CASBEE (Comprehensive Assessment System for Building Environmental Efficiency) in Japan is a tool intended to implement the environmental assessments based on new concepts including BEE (Building Environmental Efficiency).[1]
The concept of this project is also “Improvement of environmental quality and performance” and “Reduction of environmental load”.
At the design stage, it was evaluated as S-rank (BEE=3.7) (=”Excellent”) with CASBEE.

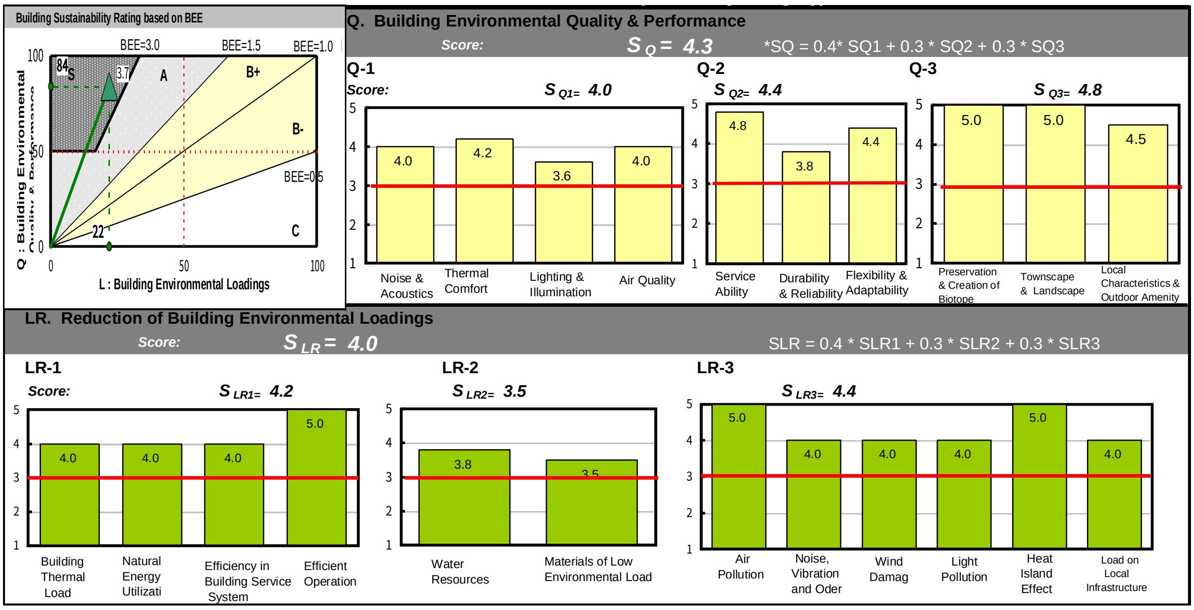

Figure 5. Environmental assessment in Japan

METHODS

Field Measurement Outline of DS

We predicted the performance of DS by the simulation that used the heat ventilation network [2] at the design stage. In 2005, we measured temperatures, the glass surface temperatures a velocity in DS, and a solar radiation in the room etc to confirm the DS performance in operation at each season of summer, autumn, and winter. Table 2 shows the outline of the measurement

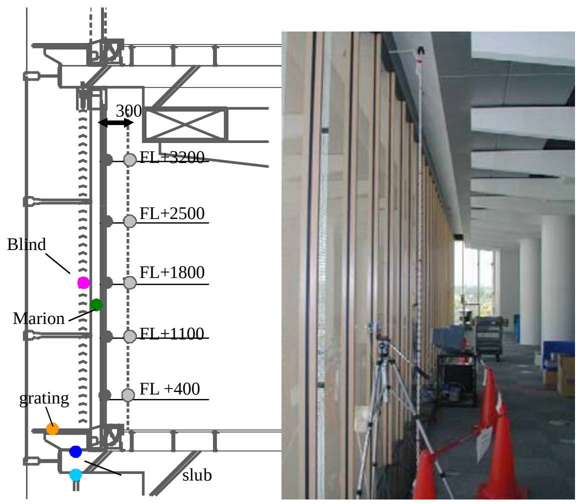

Figure 6. Outline of fielded measurement

Table 2. Outline of fielded measurement

Measurement item	Measurement point
Indoor glass surface temp.	5 p／F×4 place
Indoor air temp. by DS	5 p／F×4 place
Outdoor solar radiation	1 p（Outdoor）
Indoor solar radiation	1 p（3F/south）
Outdoor air temp.	1 p（Roof）
Outdoor velocity	1 p（Roof）
Thermal Graphic	1 place（3F/south）
Velocity in DS	Each Floor x 2 zone/F x 2p
Air Temp. in DS	Each Floor x 2 zone/F x 2p
Object temp. in DS	Grating, slab, blind

RESULTS

Thermal Performance of DS

In summer, the inside temperature of DS on the fourth floor was 35.4 degrees C on the south side, and 36.5 degrees C on the north side. The north and the south sides recorded similar temperatures despite the radiation of sun light.

The surface temperature of inner glass on the 3rd floor was approximately 32 degrees C for both north and south sides and this was almost equal to the outside air temperature. This shows that DS effectively blocks the radiation load by sunlight screening and heat exhaust.

The room temperature around the inner glass (300mm from glass surface) was equal to the AC room temperature setting (27.5 degrees), and the thermal environment in the perimeter section was maintained.

As for the DS in autumn, difference of temperature of the fourth and the first floor was larger with ⊿ T = 9.6 deg than it was in summer. On the north side, natural ventilation leveled the temperature in DS from the fourth floor from the first floor. The indoor glass surface temperature was 30 degrees C and north side 24 degrees C on the north side. The air temperature around DS was 24-25 degrees C, and good thermal environment was maintained.

All ventilation dampers are closed in winter(a day with a little sunlight). During the night, the temperature in DS was around 7 degrees C when the outdoor air temperature was 0 degrees C. The indoor glass surface temperature then was 15 degrees C. In addition, air temperature around DS was 22 degrees C in the daytime, and it was 20 degrees C at night. From these measurements, the heat loss by DS, was shown to be fewer.

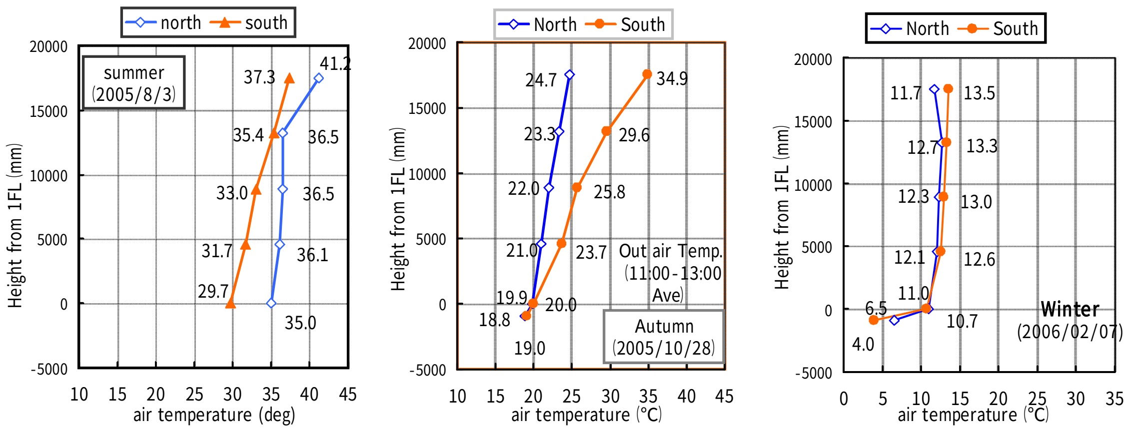

Figure 7. Profile of temperature (DS)

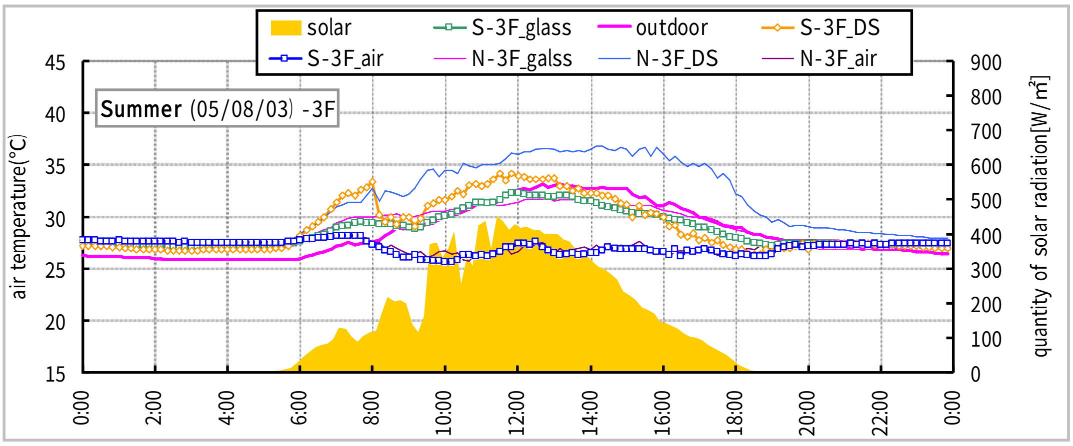

Figure 8. Result of Field measurement on 3F in summer

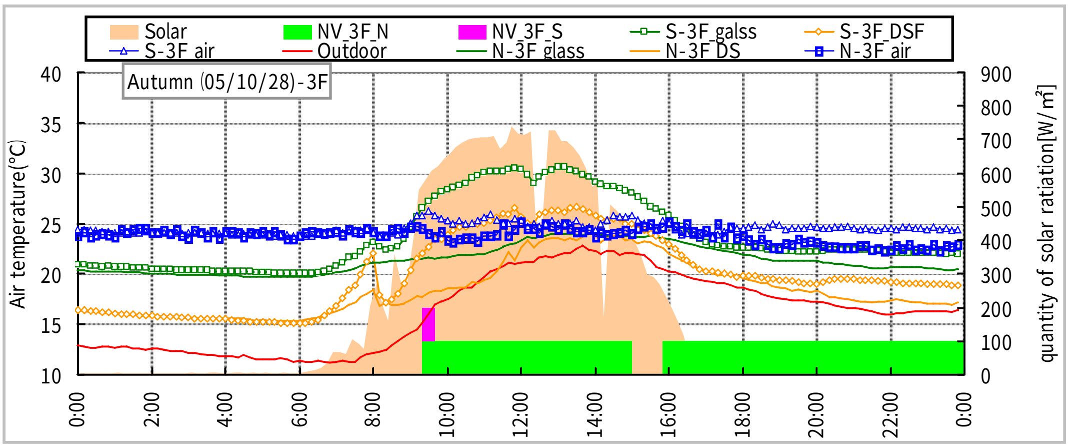

Figure 9. Result of Field measurement on 3F in autumn

Figure 10 shows the measurement result concerning the solar heat gain ratio of DS. The solar heat gain ratio was approximately 23%, showed high solar screening.

The measurement result concerning the thermal insulation efficiency in winter is shown in Figure 11. The over-all heat transfer coefficient of DS was about 2.7W/m2･K.

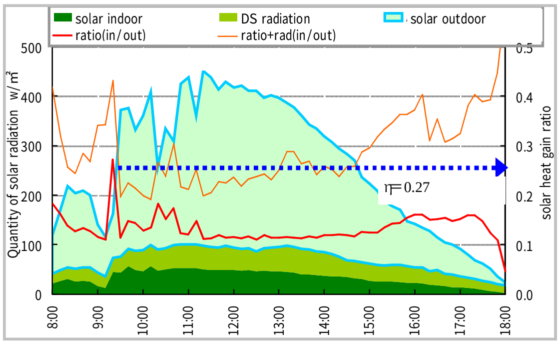

Figure 10. Solar heat gain ratio of DS

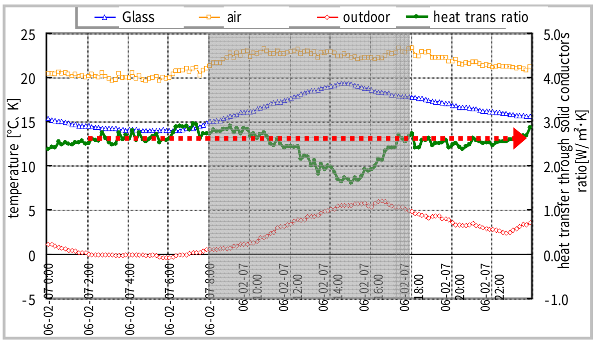

Figure 11. Over-all heat transfer coefficient of DS

Survey of "Cool-Biz"

In this building, a trial operation of "Cool Biz" was carried out together in 2005, 2006. TAC is assumed to maintain a thermal comfort even when the room temperature is setting high (28 degree C in summer).

We executed the survey on "Cool-Biz" to workers.

90 % agreed with the execution of "Cool Biz". A number of people reasoned as "Cooperation to energy conservation" .

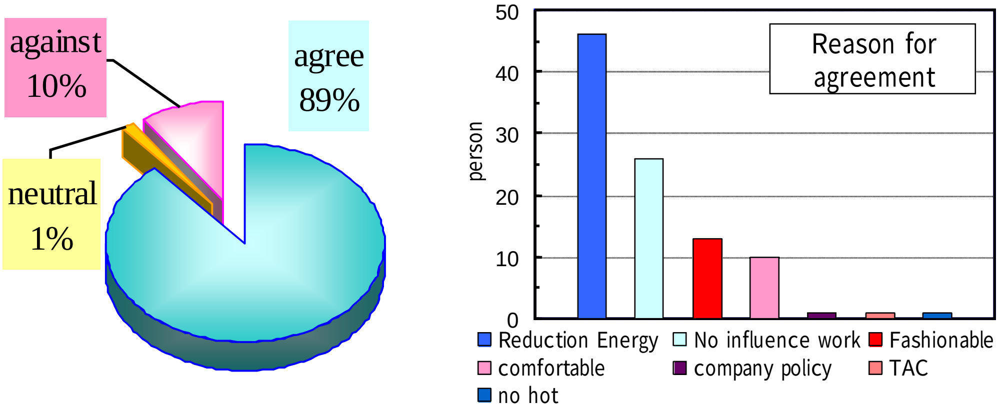

Figure 12. Survey about "Cool-Biz"

Filed Measurement of Energy Demand Profile

Figure 13 shows the annual Primary equipment load. Because of the reduction of thermal load by environmental technologies (DS, natural ventilation, etc), hot water demand in winter, and cold and hot water demand in intermediate season.

Fig.14 show the annual primary energy consumption of this building from July, 2005 to June, 2006. The primary energy consumed value was 1,657 MJ/m2/ year,
It was estimated that 21.9% (comparison object value : 2,121MJ/m2/year) reduction in the primary consumed value was attained by the DS, TAC, and various technologies (e.g. : natural ventilation, cool and heat tubes, using sunlight, etc). And, it was 16.3% less in the primary consumption compared to the standard office building in Japan (1,980MJ/m2/year).Table X. shows primary energy equivalent value when calculated.

Table 3. Primary energy equivalent value

	Electirc power	Natural gas
Value	9,760 MJ/kWh	46,100 kJ/Nm3

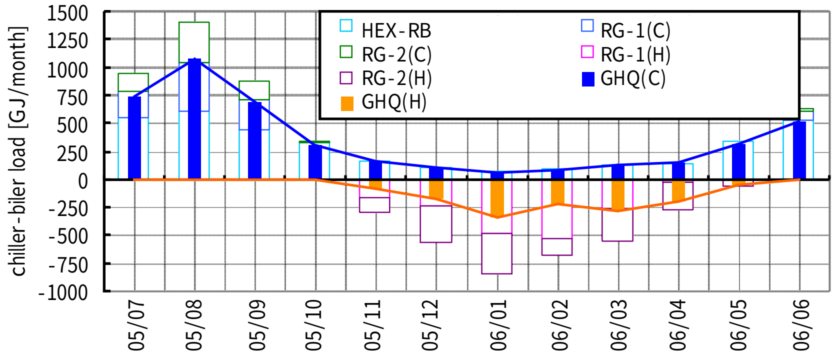

Figure 13. Annual primary equipment load

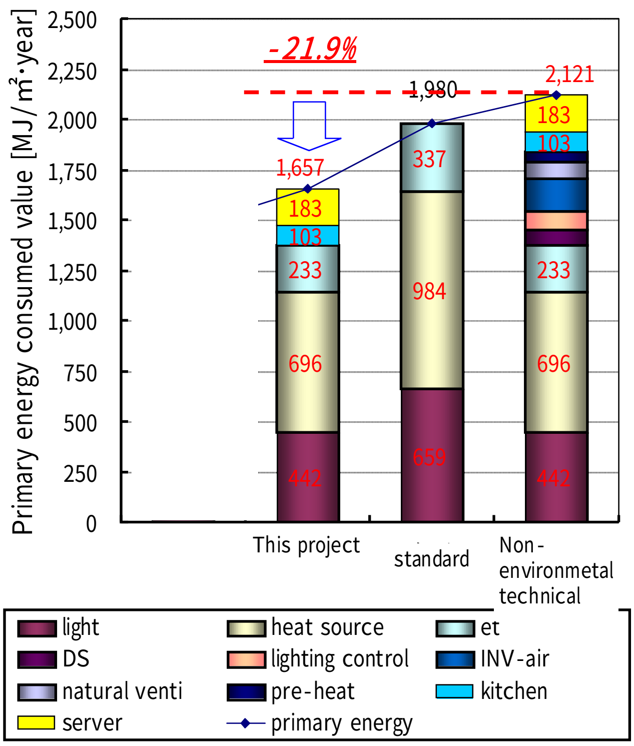

Figure 14. Primary energy consumed value

DISCUSSION

The concept of this project is "Improvement of environmental quality and performance" and "Reduction of environmental load" . So, this building integrated various environmental technologies. Field measurement verified that the building, estimated that 21.9% reduction in the primary consumed value could be attained by the DS, TAC, and various technologies, was sustainable.

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Energy consumptions in Hospitals: preliminary results of the ICEOs Project

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SUMMARY

The ICEO Project quantifies the thermal and electrical energy consumption in hospitals of an Italian region (Lazio Region). 26 out of the 57 (45,6%) hospitals of the region were selected for a questionnaire in order to acquire information about activities, structural characteristics, technologies and the energetic consumption during the years 2001-2003. Only 61,5% of the selected hospitals supplied the required information. The consumption data show a great variability between hospitals, not always explained by their technological level or by their complexity. In Italy the energy consumption for housing and other civilian uses is increasing and today it represents the third part of the total energetic consumption. To reduce consumption, economic solutions should be adopted such as effective maintenance programs. It could be also useful to increase the adoption of renewable energetic sources and of a strong training program for the hospital staff.

INTRODUCTION

A hospital represents a classic example of a complex system where every structure is different each other and its characteristics are determined by the variety of supplied facilities, the size and the geographic location, sometimes with sophisticated solutions, but where every plant and technical equipment requires attention, in order to supply the service 365 days a year.

Besides, if the leading mission of a hospital is to supply high-grade medical treatments, in a short lapse of time, in optimal general conditions and at the lowest cost, it's also true that the various equipments employed to achieve that goal, and the whole operation, imply a relevant use of both thermal and electric energy.

The major amount of thermal energy consumption in hospitals is generally absorbed by heating and ventilation systems, covering the 50-60% of total consumption [1], whereas the remaining percentage is employed by the so-called "technical service equipments", such as: hot water production, cookery, laundry, sterilization.

The electric energy, instead, covers an average of the 20% of the entire energy consumption, but it could consist up to an average of 30-35% of the total cost [1].

The increase of electric consumption in the last few years is certainly due not only to the operation of a variety of new diagnostic and therapeutic equipments, but, mainly, to the increasing use of television sets in every rooms and in-patient areas, and to a more generalized use of HVAC systems [1].

The study performed by the ENEA (*Ente Nazionale Energia e Ambiente*) in the '90s (1), estimating a thermal energy consumption equal to the 80%, and the remaining 20% of electric energy consumption, already recorded a valuable waste of energy certainly due to a general obsolescence of plants and buildings, but also to an inadequate management. On the whole, the pure cost of energy constituted just a portion of the total consumption.

Referring to the same study in the '90s, it turned out that in Italy the cost of the National Health Service (NHS) amounted to approximately 70,000 billions of Lire in 1991, about the 50% of which was attributable to hospital service. The incidence of energy was estimated, even in other European countries, around the 4-5% of the entire cost of the hospital services and this entails that the energy bill of the Italian hospitals, at that time, was up to 1,500 billions of Lire [1], around 775 millions of Euro, namely 17,445,000 MWh (1.5 Mtep) yearly.

The average energetic cost, for a hospital bed, is 1,700 Euro/year, whereof 1,300 Euro are due to heating cost.

A remarkable amount of energy could be saved reducing the service cost by 10%, improving the energy efficiency, thanks to a better management and rationalization [1]. Therefore, it is important to make aware the hospitals and their staff about the need of a rational use of energy.

OBJECTIVES

Aim of the ICEO Project (translation: investigation on hospital energy consumption) was to quantify the thermal and electrical energy consumption by the hospitals of an Italian Region (Lazio Region), in order to identify the factors associated with the consumption and to define possible strategies of control.

METHODS

The study covered the 45.6% (26 out of 57) of the regional hospitals. By mean of a questionnaire, each hospital has been asked to supply information about activities, structural characteristics, technologies and energetic consumption (thermal and electric) during the years 2001-2003.

Data sheets on energy accounting by ENEL (*Ente Nazionale Energia Elettrica*), data of ENEA researches [1, 2] on the rational use of energy in hospitals, and data used in a previous pilot study [3] were taken into consideration to draw up the questionnaire.

The questionnaire was sent by fax to 26 hospitals selected from the list published in the Ministry of Health website. The acquired data were treated to find out useful energy indicators to compare energy situations in hospitals. Only 2003 energy data, being the others non exhaustive, have been analysed.

RESULTS

16 (61.5%) out of the 26 selected hospitals submitted the questionnaire and just 14 of them provided all the necessary information. It turned out that the majority of them had given the operation of plants in outsourcing.

Table 1 shows hospitals that answered the questionnaire by typology, structure and service.

Considering the relationship with the National and Regional Health services, the complexity level and the financing, the hospitals were divided in four groups: University hospitals, Major General Hospitals, Research Hospitals and Local General Hospitals.

Table. 1 Characteristics of the hospitals involved in the study

	Name	n° of buildings	Volum m^3	n° beds	n° of patients	Average stay (days)
University Hospitals	UH 1	44	1,287,675	154,0	43,985	11.1
	UH 2	1	180,000	450	639	5
	UH 3	1	--	179,0	63,141	8.8
Major General Hospitals (*)	MGH 1	5	415,889.1	105,5	26,972	9
	MGH 2	3	201,387	783	22,428	9
Research hospitals (**)	RH 1	6	66,577	455	24,545	6.8
Local General Hospitals (***)	LGH 1	1	--	99	4,170	6.4
	LGH 2	1	164,500	420	17,102	7.11
	LGH 3	2	31,500	75	3,129	6.73
	LGH 4	1	12,500	41	1,481	4.08
	LGH 5	3	141,703	434	28,000	6.2
	LGH 6	1	85,000	387	11,842	6.8
	LGH 7	1	117,000	369	9,069	9.35
	LGH 8	1	28,000	71	1,839	7.2
	LGH 9	1	57,000	122	3,602	9.5
	LGH 10	1	104,803	324	14,620	6.3

(*) Self administered within the Regional Health Service
(**) Self administered within the National Health Service and financed by the State for research purposes
(***) Administered by the Local Health Units within the Regional Health Service

Figure 1 shows the percentage distribution of Regional and selected hospitals depending on their typology. In the investigated sample the percentage of Research Hospital was lower if compared with the Region, but this defect was acceptable because of the higher percentage of University Hospital.

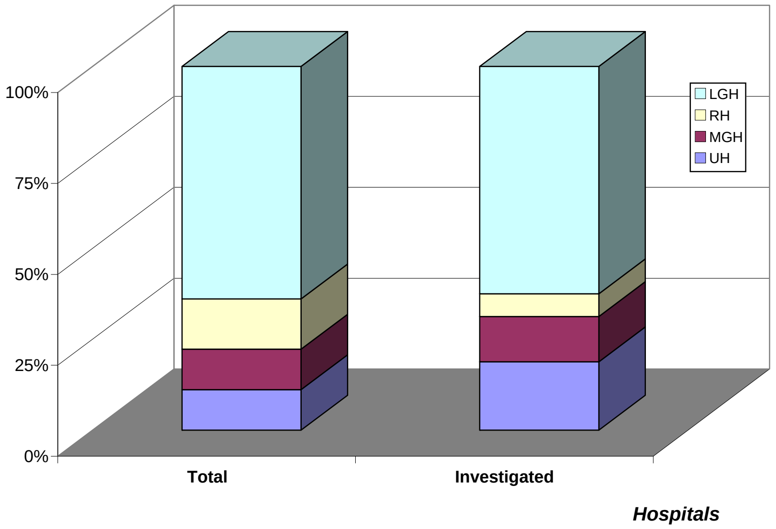

Fig . 1 – Distribution of hospitals by typology: total and investigated hospitals

The largest amount of the hospitals of the Region (73%) use methane as heating fuel (fig. 2), followed by oil (13%), eco-diesel (7%) and Low Sulphur Content (7%).

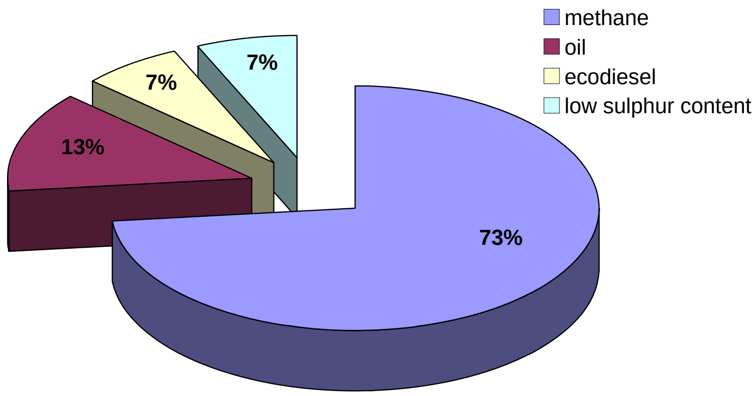

Fig . 2 - Combustible used by hospitals

In regard of electric energy consumption, the yearly average is around 13,500 kWh/bed (min 2,100 kWh/bed - max 23,700 kWh/bed) and the thermal consumption average is around 22,000 kWh (min 9,600 kWh/bed - max 36800 kWh/bed).

Figure 3 illustrates a wide variability among the hospitals, not always explained by their technological level and the level of service supplied. For two hospitals (LGH 6 and LGH 8) data about electric consumptions were not available.

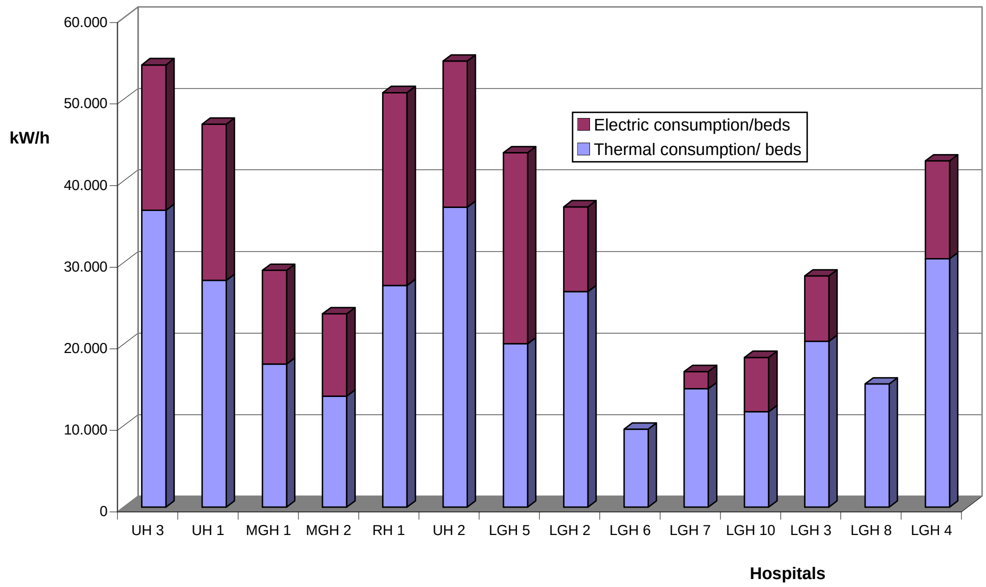

Fig . 3 - Energy consumption (thermal and electric) by hospital

Comparing the average results observed in the study with those obtained by other national and international studies (table 2), we found out that similar proportions of thermal/electric energy requirement are supplied in Italy and in foreign countries.

Table. 2 - Average results observed in the ICEOs study with those obtained in other studies

Study	electric consumption (kWh/bed)	thermal consumption (kWh/bed)
Italy, 1997 (*)	6.000	23.000
Belgium, 1997 (*)	11.000	30.000
Netherlands, 1994 (*)	10.000	36.000
Austria, 1997 (*)	27.000	36.000
Canada, 1997 (*)	23.000	42.000
Milan, 2003 (**)	19.084	69.295
PROST, 2003 (***)	14.600	-
ICEOs, 2003	13.535	21.960

(*) CADDET - Saving Energy with Energy Efficiency in Hospitals, about the average of international consumption [4]
(**) data collected by the “Cà Granda di Riguarda” hospital, Milan, in 2003;
(***) Prost project (Public Procurement of Energy Efficient Tecnologies), financed by Italian Ministry of Environment and the European program SAVE, that shows the average of thermal consumption in the Italian hospitals [5]

DISCUSSION

The energy consumption for housing and other civilian uses represents the third part of the total energetic consumption in Italy. Besides, the introduction of new technologies implies a strong and continuous increase of consumption. In various hospitals the Energy Manager, although required by Italian low in all structures consuming a quantity of energy >1,000 OET/year, isn’t in charge or hasn’t a specific competence or is absorbed by other tasks, thus lacking in his reference role for the energetic consumption control program.

The reduction of consumption could be achieved by means of economic solutions such as a greater care by the staff, the use of high-efficiency lamps and heat-saving heating plants, the adoption of renewable energetic sources and the implementation of strong programs of maintenance.

CONCLUSIONS

Hospitals are supposed to be significantly interested in the adoption of new technologies employing “renewable” natural energetic and, above all, non-pollutant sources.

In accordance with the Italian CIPE directive 57/2002 [6], a solution could be the promotion of the efficiency in the production of energy by means of high-productive technologies to generate electric energy, the spread of plants of co-generation electricity-heat, the use of the energy produced by the plants of thermo-destruction of sewage and the saving of dispersed heat, but also sustaining the use of renewable energetic sources and the promotion of studies and development in the area of less impacting energies.

In this field it’s of value the possible planning of a hospital according to the principles of a sustainable development, mainly using sun power by means of systems of active and passive utilization, dampening environmental and economic expenses [7].

The energy consumption in hospital, nowadays, is 3 times greater than the energy consumption for housing, because the whole plant is supposed to assure an extended seasonal heating supply, higher temperatures, more air exchange and particular microclimatic characteristics during the year [7].

As suggested in the literature [1, 7, 8], useful actions to reduce energetic consumption in hospitals could be:
improving the thermal isolation of both the vertical and horizontal envelope;
realizing a computerized control of the illumination, HVCA, doors control systems;
using renewable energetic sources (active and passive sun power systems);
adopting co generative plants and heat pumps;
reducing thermal dispersion.

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B05
Sustainable energy systems

Advanced sustainable energy technologies for cooling and heating applications

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SUMMARY

Multi-energy sources, for heating and cooling purposes, are considered to have a large potential in contributing to the penetration of renewable energy sources for domestic, building and industrial applications. But it requires that conventional heating and cooling systems have to be adapted or changed for incorporating renewable energy sources. Nowadays, the heating, ventilating and air-conditioning industries are the actors which have access to the market, and they provide services using mainly fossil fuels or electricity as energy sources. The objective is to have them using renewable energy, combining conventional and renewable sources and proposing environmental cost-effective products.

The technical goal of this project is to set-up technology resource centres in Europe, with the aim of building bridges between the technology providers (research centres, universities and industry) and the technology users (manufacturers, engineering companies…). The project outputs are:

- Creation of the clubs/grouping
- Collection of RTD results
- Development of design tools for evaluating and sizing the new technologies
- Best practice and training programme
- Creation of a knowledge resource centre relying on multi language search engines based on craft ontologies and terminologies
- International collaboration with China

For this 13 partners are collaborating to this project. The consortium is composed of centres of excellence in the area of renewable energies, energy agencies and professional associations representing industry.

INTRODUCTION

The growing demand for energy despite limited fossil fuel reserves and growing environmental concerns is probably the major challenge of the 21st century. To achieve a sustainable development, the origin and the usage of energy have to be addressed, and advanced renewable energy technologies and energy carriers have to be developed, requiring significant progress in research and technology.

The objective of this project is to support industrial manufacturers and engineering companies of heating and cooling systems (including energy storage) in their development, by introducing more renewable energy sources in their technology. For European countries, according to the IEA, thermal use of energy represents more than 25% of the gross energy consumption. Multi-energy sources, for heating and cooling purposes, are considered to have a large potential in contributing to the penetration of renewable energy sources for domestic, building and industrial applications. But it requires that conventional heating and cooling systems have to be adapted or changed for incorporating renewable energy sources. Nowadays, the heating, ventilating and air-conditioning industries (HVAC) are the actors which have access to the market, and they provide services using mainly fossil fuels or electricity as energy sources. The objective is to have them using renewable energy, combining conventional and renewable sources and proposing environmental cost-effective products.

The technical goal of this project is to set-up technology resource centres in Europe, with the aim of building bridges (figure 1) between the technology providers (research centres, universities or industry) and the technology users (manufacturers, engineering companies, architects, local actors…). The technology user is a company or a professional organisation which is going to incorporate in its system/service renewable energy technologies.

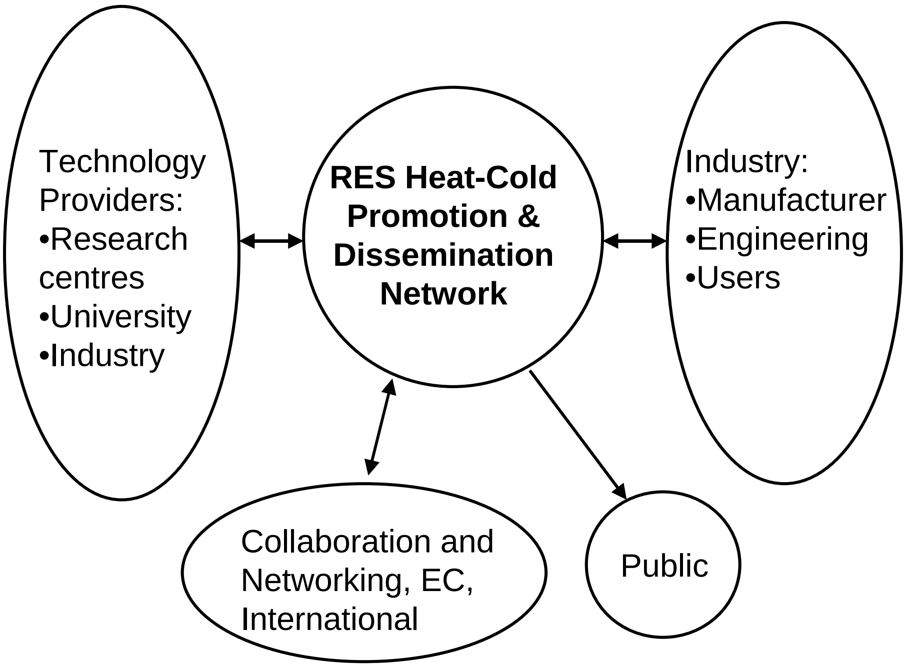

Figure 1 : Overview of the project methodology

A technology resource centre is where industry can found resources for developing and improving their technology. A resource centre might have its own research capacity or collaborates with technology providers such a research centre (private or public) or universities. A resource centre is the focus point of industrial actors in a same area, it associates manufacturers (components and systems), engineering companies, end-users, and energy providers.

Technology resource centres have five main goals:

- To bring new technologies from research to the market
- To promote trans-national collaboration
- To support companies in problem solving
- To be identified as a partner to manage industrial RTD projects
- To access and to share data and expert knowledge base

RENEWABLE ENERGY SOURCES FOR HEATING AND COOLING

To be effective and to take into account the specificity of the market, this project focuses on renewable heating and cooling technologies: **RES-HC**. It concerns **solar heating and cooling, biomass (heat), heat pumps, geothermal heat and energy storage**. The objective is to facilitate the introduction of RTD results into the market by establishing technology resource centres. These centres will operate on a common basis, sharing tools, date-base and exchanging information.

Within this project, 7 technology resource centres will be established and linked by the end of the project. The effective technology transfer will be insured by associating industrial partners to the project. The objective is to get at least 500 industrial partners connected to the network and beneficiating from its output.

A virtual knowledge base resource centre relying on a dedicated multi-language search engine will be created. 6 centres of excellence in the area of RES-HC will contribute by linking their data and project results references. This platform will be open to other centre of excellence wishing to share informations.

This project is technology oriented, and aims in structuring the scientific and technical background for developing and optimising RES-HC systems. It contributes to the technology and industrial development phase of the product life cycle development, as the marketing phase is essentially controlled by non technical issues such as policy measures and financing options.

For RES-HC technologies, it is believed that the market penetration will be insured by associating the traditional actors of the sectors to manufacturers focusing and single RES-HC technology. The growing interest of manufacturers of gas–fired boilers for space heating and hot water to renewable energy sources confirms the strategy proposed in this project.

The traditional actors in the heating and ventilation industry are mainly composed of small and medium enterprises, which have low research and technology development activities. On the contrary, the companies involved in RES-HC are more innovative. Therefore, there is a clear need in supporting the traditional industry in incorporating renewable energy sources in their systems and to associate technology providers of RES-HC for developing and marketing new products.

Individual renewable energy sources have advantages but also drawbacks, such as intermittency of the source or difficulty of supply. To cope with these drawbacks, and to provide an efficient service to the final user, it is necessary to associate various energy sources and energy storage. Solutions have been proposed developed and tested within several research projects and need now to be appropriated by the actual industrial actors. For this they need to get the information, to identify the right partners and to set-up a collaborative agreement for optimising and marketing new products, and these are the role of technology resource centres.

THE POTENTIAL OF RES-HC TECHNOLOGIES

Market Potential

This project focus on renewable heating and cooling technologies for domestic, building and industrial applications. The renewable heating and cooling energy sources are biomass, solar thermal and geothermal (including heat pumps). In 2000, the energy consumption for thermal power generation (including district heating) is 410 Mtoe/year, which represents 25% of the gross EU-25 energy consumption [1]. At this value must be added 16 MWh/year of electricity consumption for building and residential air-conditioning (recent surveys have estimated an electricity consumption of 29 MWh/year in 2010[2]).

For RES-HC [3], the EU 1997 white paper on energy gives a target for 2010, of a market share of 80 Mtoe, 20% of the thermal power generation [1]. In 2001, the market share is 48.7 Mtoe, 12% of the thermal power generation [4]. This requests an average growing rate of 5 to 6% until 2010. Even if the situation has progressed since 1997 (see figure 2), the 20% average market share will probably not be reached in 2010, and an extra 2% must be achieved. In order to reach this objective both market pull and technology push measures have to be adopted. This project will concentrate on developing and promoting cost-effective technologies and systems. The Bielefeld Declaration (14 May 2004), has emphasised that specific measures have to be taken in favour of RES-HC if a 20% market share of the thermal power generation must be reached in 2020.

Even if the overall RTD expenditure on renewable energy sources is slowly increasing in Europe [5], the effort related to RES-HC is decreasing each year since 1980, representing nowadays less than 10% of the total effort on RES. This lower support is in contradiction with the high potential of RES-HC, and a more vigorous support policy must be implemented at the European level [6].

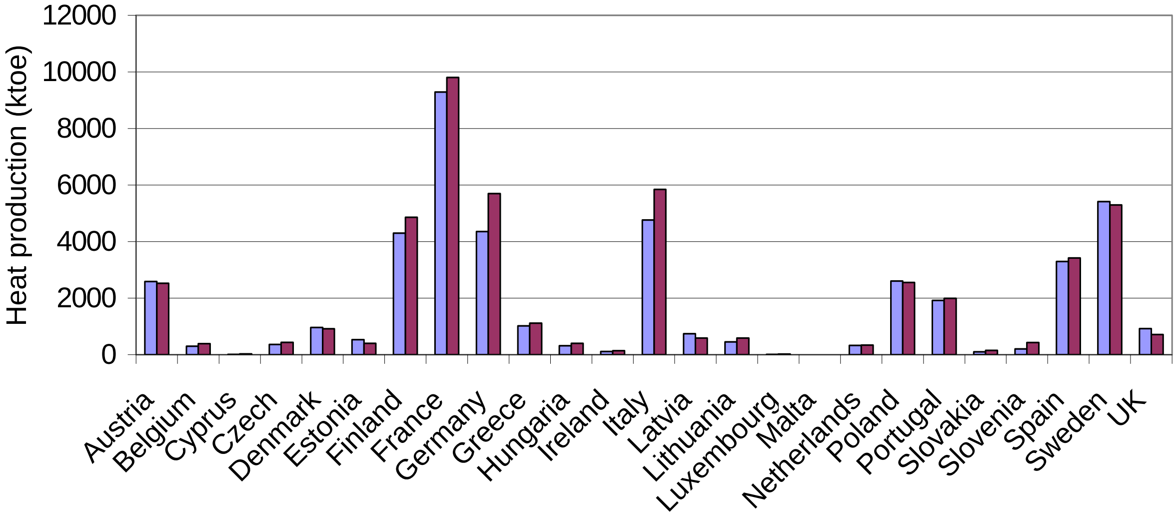

Figure 2: RES-Heat production for the EU-25 for 1997 and 2001

The potential of RES-Heat has been recently evaluated for the European Union [7] and is summarised in the table 1. As it can be seen, depending on the source and the country, there are very significant market penetrations of the renewable energy sources for heating and cooling (RES-HC). Biomass has from far the largest market penetration, having reached almost 50% of its long term potential. For EU-25; biomass and geothermal energies are mature technologies that contribute about 3% to primary energy supply. Solar, heat pumps other new renewables have experienced rapid technology development, but as yet they represent only a small share.

The main figure is that biomass is nowadays the major provider of renewable thermal energy, but it will reach 75% of its potential in 2010. The contribution of the other sources (heat pumps, solar and geothermal) could potentially contribute to the same amount but is nowadays much less important. Heat pumps and solar renewable sources needs another source of energy to provide the total thermal needs, therefore it is only by associating various energy source that these technologies could reach their full potential. This project will concentrate on multi renewable energy sources technologies.

Table 1: Market penetration of RES-HC systems in EU-25 [7-8-9-10]

	Penetration (year 2002)			**Energy (Mtoe)**		
Source	**Minimal**	**Maximal**	**Average**	**2001**	**2010 target**	**Potential**
Biomass	10%	70%	47%	46.90	75.0	100.0
Solar thermal	0%	30%	<2%	0.57	4.0	36.0
Heat pumps	0%	30%	<2%	0.40		32.0
Geothermal (grid)	0%	40%	<2%		1.0	3.0
Geothermal (total)*				0.83		
Total				48.7	80.0	171.0

* The total geothermal heat production is complex to analyse due to various applications and the calculations methods varies from one country to another (contribution of hot springs).

It has been recognised [5] that hybriding fossil fuel and renewable energies will accelerate market penetration of RES. The technical and financial risks can be minimised, therefore being more attractive for investors. Hybrid and multi-sources systems are particularly applicable to renewable heating and cooling.

Social and environmental impacts

Renewable energies and in particular heating and cooling technologies are decentralised markets and create employment essentially at a local stage. The manufacturers involved in such activity are often small and medium enterprises ranging from few employees to several hundreds. Furthermore, the installation and maintenance of these equipments will require specialised workers [5].

The replacement of fossil fuel heating systems by renewables will induce a modification of the individual behaviour and will have impact on employment [11]. Fossil fuel heating systems implies generally relatively low investment and after 10 years of operation, the cost for the fuel is largely the first expense. On the contrary, renewables have high capital cost and low operating cost. Therefore, the financial amount which is spent for buying fossil fuels (mainly imported) is mainly replaced by an equivalent amount in hardware which are mainly produced in Europe. An energy consumption of 80 Mtoe of fossil fuels (gas and oil), correspond roughly to an expense of 40,000 M€ (30 €/MW.h and average energy efficiency of 70%). If this expense is partly replaced by capital costs for investment in renewable technologies, it is expected to induce the employment of 400,000 persons.

This project will contribute in creating employment in small and medium enterprise of the heating, ventilation and air-conditioning industries, by introducing more renewable energy technologies in the systems they are developing and installing. The activity of manufacturers of RES-HC systems (solar panels, biomass heaters, heat-pumps…) will also be strengthened.

From an economic point of view as RES-HC technologies are capital intensive energies, the present period is particularly interesting for the development of RES technologies. This is due to a relatively low cost of money (yearly interest below 4%) and high cost of energy. If the different incitation measures in favour of RES are promoted and increased it will guarantee the development of RES-HC in Europe.

The adoption of the Kyoto protocol implies that Europe has to reduce its CO2 by 5% in respect to its level in 1990, and heating and cooling are particularly concerned by CO2 emissions. Renewable energy sources are by definition environmental clean energies and contribute to preserving natural resources and prevent global warming. The CO2 emissions of such system are significantly reduced compared to fossil fuel system, and this even taking into account the energy required for manufacturing the system (life cycle impact assessment). The use of renewable energies for heating and cooling purpose has indirect environmental impacts. If 80 Mtoe of fossil fuels are replaced by renewable energy it will affect the distribution schemes of energy: (1) less transport of liquid fuels by sea, river, train and road which reduces the harmful effects (pollution risks, number of trucks on the roads…); (2) reduced peak effects for the electric demand in winter time for heating and summer time for cooling and induces a better management of the electric network.

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Towards sustainable energy systems – role and achievements of heat integration

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SUMMARY

This paper presents an established sustainable and integrated design methodology for the efficient heating and cooling of individual buildings and complexes. The methodology includes the design basis for combined heat and power systems, refrigeration, air conditioning and heating with pump systems. It is equally applicable for single family houses as well as large building complexes and meets a major challenge in the design of heating and cooling systems, namely, the complexity of energy and power integration. The efficient use of available heating and cooling resources for serving buildings of various sizes and designations can significantly reduce energy consumption and emissions. Although heat integration (or Pinch Technology) has been around for several decades and has been very successfully applied in large industrial applications, it has only recently been applied for improving the energy efficiency of buildings and building complexes. There is significant scope for application, as energy prices rise and energy related emissions are required to be reduced. This methodology can also be used to integrate renewable energy sources such as biomass, solar PV and solar heating into the combined heating and cooling cycles. Case studies are used to illustrate the methodology. The paper makes suggestions how the methodology can be applied for energy and emissions reduction in single buildings and complexes.

INTRODUCTION

Since 1995 the energy consumption of the EC member countries rose by 11 %, to the value of 1637Mt of oil equivalent [1]. The population of the EC member states however is growing at a much slower rate, approximately at 0.4 %/y [1]. In the UK domestic energy consumption has risen from 35.6 Mt of oil equivalent to 48.5Mt in the period 1971 to 2001, an increase of 36%, despite energy efficiency increases [2]. The increase in energy consumption per household was less, at 5 %, over the same period, better reflecting the increase in energy efficiency in individual dwellings. However in the UK it is proposed that the number of houses built over the next 10 years should increase by 200,000/y, to account for the smaller number of people per household, and the demand for housing [3]. This will have serious implications for energy consumption and greenhouse gas emissions. To date the increase in energy consumption from individuals [2], dwellings, and buildings generally has been met by the burning of carbon based fuels, or in some cases, by the increase in nuclear power. Although renewable energy has seen a large increase, the contribution towards total domestic energy use is still small (in the UK 1.0 Mt of oil equivalent compared to a total use of 47.0 Mt of oil equivalent).

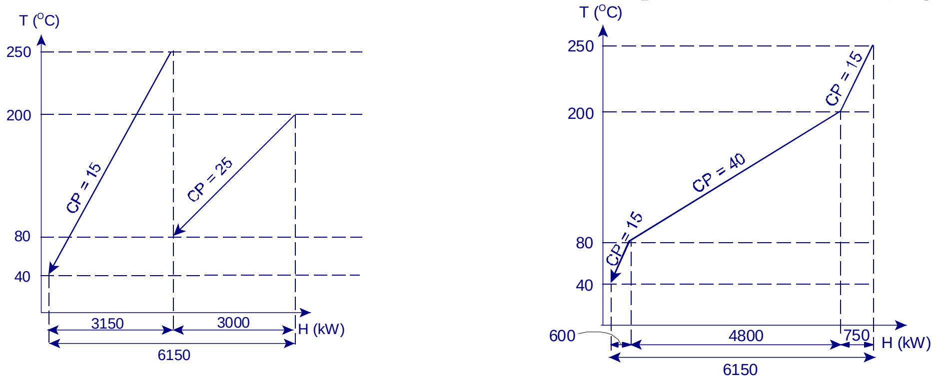
SUSTAINABLE ENERGY AND HEAT INTEGRATION

Heat Integration (or Pinch Technology/Process Integration) is an energy saving methodology that has been extensively used in the processing and power generating industry over the last 30 years. This method examines the potential of exchanging heat between heat sources and heat sinks via the use of heat exchangers and reducing the amount of external heating and cooling required. A systematic design procedure has been developed to provide the final energy reduction design of the system. The method has further been developed to specify the source of heating and cooling required (e.g. steam, hot water, cooling water) and also the potential of power production in the form of shaftwork or electrical production. More details are available elsewhere [4-9]. This methodology is based on the analysis and understanding of heat exchange between process streams through the use of a temperature-enthalpy diagram. The specific steps for drawing the curves in this diagram are presented in Figs. 1 - 3. The methodology first identifies sources of heat (termed hot streams) and sinks of heat (termed cold streams) in the process flowsheet. Tab. 1 presents a simple example.

Table 1 Sources (Hot) and Sink (Cold) streams

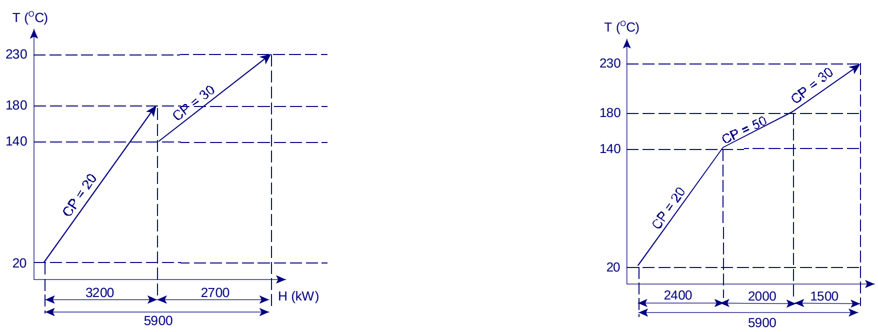
Stream	**Type**	**Supply Temp.** T_S (°C)	**Target Temp.** T_T (°C)	**ΔH** (kW)	**Heat Capacity Flowrate** **CP** (kW/°C)
Fresh Water	Cold	20	180	3200	20
Hot Product 1	Hot	250	40	-3150	15
Juice Circulation	Cold	140	230	2700	30
Hot Product 2	Hot	200	80	-3000	25

Sources of heat can be combined to construct the composite hot stream (Fig.1) and sinks of heat can likewise be combined to construct the composite cold stream (Fig. 2).



(a) The hot streams plotted (b) The composite hot stream

Figure 1. Composing hot streams to create a hot composite curve



(a) The cold streams plotted (b) The composite cold stream

Figure 2. Composing cold streams to create a cold composite curve

The relative location of these curves on the temperature-enthalpy diagram is dependent on the allowable temperature difference for heat exchange. The next step is to select a minimum permissible temperature approach between the hot and cold streams, ΔT_{min}. The selection of the optimum ΔT_{min} is a result of an economical assessment and trade-off between the capital and operating costs (mainly for energy usage) of the process. A large ΔT_{min} implies higher energy use and costs and lower capital costs. For increasing energy cost the optimum ΔT_{min} is reduced, meaning the heat exchanger system is allowed to recover more energy, but at the expense of more capital to pay for the greater heat transfer area. This issue has been discussed in greater detail elsewhere – [10], [8], [11]. In this example a ΔT_{min} of 10 °C was selected for simplicity. Plotting the composite curves (CC) in the same graphical space (Fig.3) allows values to be derived for maximum heat recovery and minimum hot and cold utilities. These are known as targets. In this particular case of ΔT_{min} = 10 °C, the minimum hot utility requirement is 750 kW and minimum cold utility requirement is 1,000 kW.

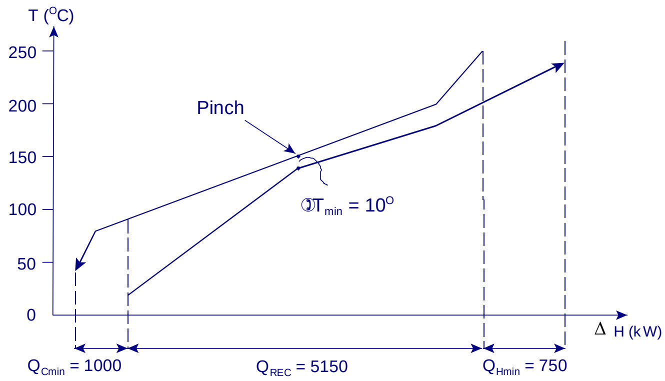


Figure 3. Plotting the hot and cold Composite Curves

In Fig.3 we can also determine the position of the Pinch. The Pinch represents the position where the hot composite and cold composite curves are at their closest (for a ΔT_{min} >0). The Pinch has provided the name for the Heat integration / Pinch Technology and has important features which make a substantial contribution to the design of maximum energy recovery systems and the design of economically efficient heat exchanger networks.

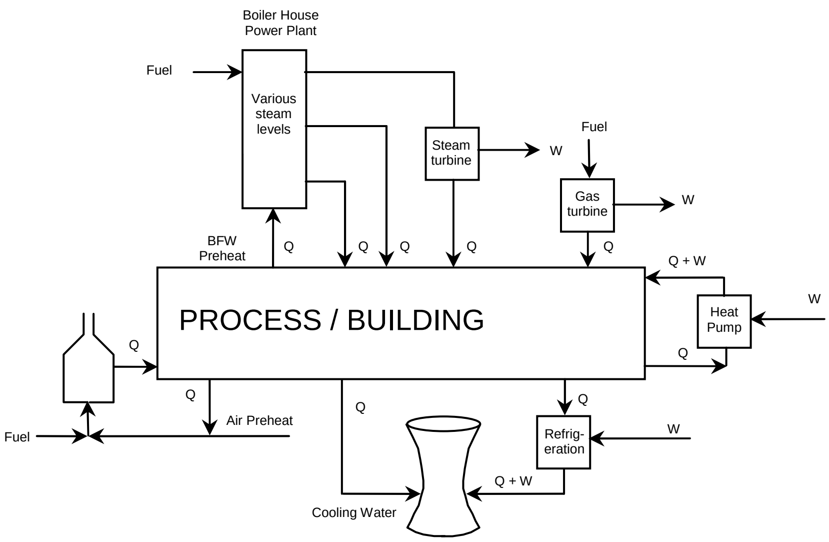


Figure 4. Potential for the choice of hot and cold utilities

Various design methods have been developed which allow these targets to be achieved in practice for both grass roots designs [4], and more importantly, for the retrofit of existing plants [12-14]. These methodologies are supported by process integration software which provides both design and retrofit support and automated design [15, 16]. In most cases there are more than one hot and one cold utility potentially providing heating and cooling requirements after energy recovery. In these situations we have to find and evaluate the cheapest and most desirable combination of the potential utilities that are available (Fig.4). To assist with this choice and to further enhance the information derived from the hot and cold CC, an additional graphical construction has been developed. This is known as the Grand Composite Curve (CC) and provides clear guidelines for the optimum placement and scaling of hot and cold utilities. The GCC together with the Balanced Composite Curves (the CC with the utilities selected added) provides a convenient tool for the optimum placement and selection of hot and cold utilities. An example of selection of utilities and its placement is shown in Fig. 5.

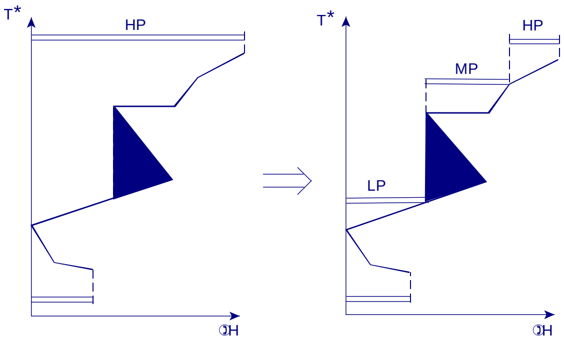


Figure 5. Placement of utilities with the help of GCC

The GCC is also a useful tool for targeting the cooling requirements in sub-ambient processes (such as air conditioning) which require some form of chilling or compression refrigeration. The GCC provides a target for the heat that has to be removed by a refrigeration process, and the temperature at which the refrigeration is needed. However, the overall process/utility system can be improved by using the heat rejected by the refrigeration system to provide low level heating to the process above ambient, thereby saving heat supplied by another utility source (such as hot water). The more detailed description has been provided elsewhere [17].

Traditional Heat Integration assesses the minimum practical energy needs for a process through a systematic design procedure involving five steps: (i) collection of plant data; (ii) setting targets for minimum practical energy requirements; (iii) examination of process changes that contribute to meeting the target; (iv) obtaining the minimum energy design that achieves the target and (v) optimisation which allows a trade-off between energy costs and capital costs.

EXAMPLES – CASE STUDIES

Hospitals are an example of building complexes which are important energy users, especially that of thermal energy. Although considerable research effort has been done to minimise the energy consumption in this sector, there are relatively few studies which apply Heat Integration (Pinch Technology) as a tool to evaluate potential energy savings and thereby reduce emissions.

Herrera et al [18] presented a study of a hospital complex which included an institute, a general hospital, a regional laundry centre, a sports centre, and some other public buildings. The use of diesel as a fuel represented 75% of its total energy consumption and 68% of its total energy cost which was 396,131 US$ in 1999. The hospital complex process diagram is shown in the Fig. 6. In the hospital complex, the heat demand is met by producing steam in boilers fuelled by a high price diesel fuel. There is no heat recovery between the existing heat sources and heat sinks. The hot streams were identified as the soiled soapy water from the laundry and the flow of condensed steam not recovered in the condensation network. The stream data are presented in Tab. 2.

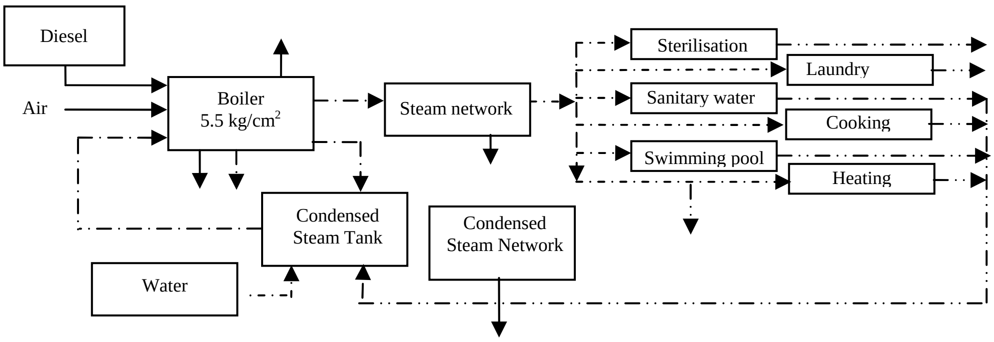


Figure 6. Hospital heat flow block diagram [15]

Table 2 presents the input data extracted from Fig 6.

Streams	**ΔH [kW]**	**CP [kw/°C]**	**Tin [°C]**	**Tout [°C]**
Hot Streams				
Soapy water (1)	23.7	0.53	85	40
Condensed steam (2)	96.32	2.41	80	40
Cold streams				
Laundry sanitary water (LS)	17.60	0.59	25	55
Laundry (L)	77.12	2.20	25	60
Boiler feed water (BF)	7.13	0.24	30	60
Sanitary water (SW)	77.12	2.20	25	60
Sterilisation (S)	12.50	0.14	30	121
Swimming pool water (SP)	151.67	50.56	25	28
Cooking (CO)	59.63	0.85	30	100
Heating (H)	100.82	14.40	18	25
Bedpan washers (B)	4.94	0.05	21	121

It needs to be noted that the CP represents mass flow m multiplied by the specific heat capacity Cp and in this case are considered constants. As can be seen from the CC (Fig. 7) the amount of external heating required, or the hot utility target, of this hospital complex is 388.64 kW. The GCC has been employed to determine the temperature levels of the utilities necessary to satisfy this requirement. This graphical method allows a more precise analysis to be undertaken in order to integrate the thermal utilities considered for the process heating and cooling requirements (Fig.8). This GCC indicates that the required heating of complex should be 388.64 kW. This means a yearly energy requirement of 12.26 TJ. The actual amount of heating provided is in fact 625.28 kW which represents the heat services that are currently transferred to the complex. This results in a potential energy savings of 38 % equivalent to saving 246,000 L of diesel/y (100,000 US$). To reduce energy demands in the form of heating to the value targeted by the analysis, the Heat Integration based analysis suggests that four extra heat exchangers should be added to the network. Two in the laundry to cover part of the heat demand, a third in the machinery rooms which help to heat boiler feed water, and the final one in the condensation tank area that heats the sanitary water. Several other issues could also have been considered to further refine the analysis, such as fouling, pressure drop and non constant heat demand.

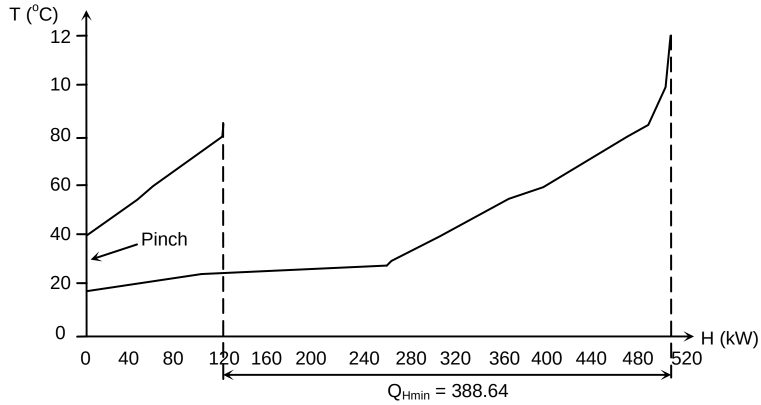


Figure7. CC for the hospital complex process streams

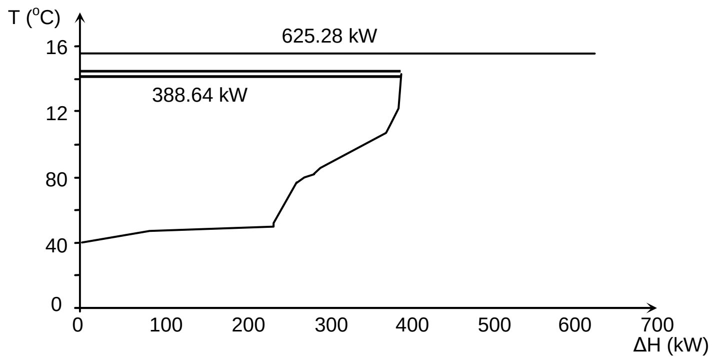


Figure 8. GCC of the hospital [15]

Kemp [9] presented a further more detailed case study of a hospital complex. The main energy demands were found to be space heating, air systems, general hot water supplies and other uses in the laundry and the incinerator. The main part of the complex was fully air conditioned. Central heating water was supplied at 80 ^{o}C, and returned at 71 ^{o}C and then reheated in a gas-fired calorifier. Tap hot water was also provided.

Table 3 Heat requirements and time distribution required during winter

Stream details and locations	No of streams	Hot streams total		Cold streams total		Times of operation
		Temp (^{o}C)	Heat load (kW)	Temp (^{o}C)	Heat load (kW)	
Main air supply	35	20-15	43	5-37	780	24 h: Day
	33	20-15	27	2-37	681	24 h: Night
Space heating	2			71-80	709	24 h: Day
	2			71-80	665	24 h: Night
Domestic hot water	1			5-60	244	24 h: Day
	1			5-60	48.5	24 h: Night
Autocalves	1			162	13	24 h
Kitchen	1			5-30	101	0530-0200
Laundry	3	37-30	67	18-26, 162	797	0700-1630
Incinerator	1	600-217	650			0900-1630
Boilers	1	235-60				24 h
Summary						
Maximum load			760		2,644	0900-1630
Minimum load			27		1407	0200-0530

In some areas (e.g. kitchens) the heat use varied considerably between day and night time. The identified streams are shown in Table 3. Energy use in winter for day time conditions was found to be 2534 kW (all hot utility) whereas in summer this value dropped to 1030 kW (hot utility) and 207 kW (cold utility) as space heating was replaced by air conditioning and the use of some refrigeration. Kemp [9] presented the GCC for the peak demand during winter (7− 9am period). Heat integration analysis of these different time periods provided various schemes for improvement. The first involved rescheduling. This took into consideration the possibility of changing the periods of high utility demand. Suggestions were (i) Moving a cold stream active in 7-9am period to a different time; (ii) Moving a hot stream not operating in 7-9am period to operate during this period. As it was not possible to shut down the central heating or air conditioning systems for this high demand 2 h period, an alternative was that laundry operations could be rescheduled to start 2 h later but finish at the usual time. This would result in the installation of new machines at a high capital cost. For the second suggestion is would be possible to start the incinerator at 7am and run it for a further 2 hours. Retaining the current waste disposal value would result in a reduced exhaust flow rate and heat recovery rate. The daytime total heat recovery would be the same, but one less boiler would be required. It was also found that additional municipal waste could be delivered and that the incinerator could be run at full power over 24 hours. The Heat Integration methodology also investigated the benefits of introducing CHP. The GCC shows that the hospital is represented by four different temperature ranges which were the laundry and autoclaves (160-170 ^oC), space heating (80-85 ^oC), domestic hot water (up to 70 ^oC), and air heating (10-40 ^oC). The heat from the jacket cooling water and exhaust gases could be recovered and be suitable for the space heating system. Standby diesel generators already installed to supply emergency power would be a cheaper option to convert than to provide heat and power from a new diesel generator. These would then provide about 50 % of the peak heat demand. Further options would be to install a micro gas turbine of around 1.5 MW. This would provide all power requirements and the majority of the peak demands. Maintenance for a single source heat and power supplier would leave the hospital complex in a vulnerable state.

Several other case studies have been reported by Linnhoff March Ltd. [19]:
a) A study with the code name BRECSU/BDP was an attempt to define the energy-efficient hospital of the future. LM Ltd looked at all aspects of hospital heating including space heating, ventilation air heating, hot water heating and sterilisation. For hot streams available for heat recovery LM looked at ventilation extraction air, drainage and incinerator flue gas. The major opportunities on an existing building were be in the utility area. There was much detail in the study, including heat generated by the patients in the day (active) and night (sleeping). The work demonstrated that, for a complex new building like a hospital, potential existed for heat recovery between extract and inlet air systems and for heat pumping from the drainage system. The most economic form of heat recovery was from the incinerator exhaust and they also concluded that the optimum utility system would be CHP.
b) Eastbourne Hospital. The largest saving found was in the laundry where dryer exhaust could heat water and fresh air for the dryers and hot dirty wash water could heat fresh water (with storage). A heat engine-based CHP scheme was recommended but this would affect the laundry recommendations because it would provide the necessary hot water.

CONCLUSIONS

Heat Integration (or Pinch Technology) has been used for 20 years in industry throughout the world to increase energy efficiency of any processing plants that have heating or cooling requirements, and also have needs for power to provide electricity or directly drive machinery. Energy savings of over 30% have been recorded, and the methodologies developed have been incorporated into the design offices of all major producing companies. The same methodologies and design rules can also be applied in buildings or their complexes. Examples of two complexes have been given. It is clear that as energy efficiency in buildings and the reduction of emissions becomes a standard issue in existing and new complexes, further examples of heat integration application will be carried out.

ACKNOWLEDGEMENT

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Residential fuel cell systems

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SUMMARY

Fuel cells offer an environmentally sound, distributed energy source for individual buildings. Many governments believe that security and sustainability of energy supply can be improved with this technology. Therefore large public support programmes are under way to facilitate the commercialization of residential fuel cell systems. VTT is actively participating in this development work in a number of projects.

Modelling and simulation tools have been developed to provide the means for studying the interaction between fuel cell system and a building.
VTT initiated in 2004 development of a PEM fuel cell based power stack and developed its control system and operating strategies.
VTT supports local industry in developing technologies for SOFC systems, e.g. fuel processing for natural gas and diesel. The most important accomplishment has been the development of a 5 kW SOFC CHP laboratory demonstration plant fuelled by natural gas. The plant was grid connected and successfully operated for 6500 hours.

INTRODUCTION

In most buildings energy in different forms is used: lighting, cooking, heating, cooling, washing and entertainment all need their share of energy. The most convenient way of using energy is trough electricity. The share of electricity in energy consumption is continuously increasing. However, natural gas and oil are used for heating in many areas. In some countries, like Finland, district heating is widely used. In some areas gas is also used for cooking. The situation can be simplified in dividing the use of energy in a building into electricity and heating/cooling.

Until now electricity has practically always been fed to the building from a centralised electricity grid. Recently a growing interest in distributed energy systems has been seen. It can be realised in many ways. The electricity and heat can be produced within the building using fuel cells, Stirling engines, IC- engines or micro turbines.

In this paper we are dealing with different aspects introducing fuel cells as power and heat sources in individual buildings. First the basic idea and some of the international programmes in the field is described. Secondly some related development work performed at VTT is described, starting from the modelling of distributed energy systems in buildings and continuing with the research and development of polymer electrolyte fuel cells (PEFC) and solid oxide fuel cells (SOFC).

The present market situation

Residential and industrial fuel cell systems for power or combined heat and power production (CHP) in single buildings are developed by a number of companies in several European countries as well as Japan and the USA. The residential units are 1-10 kW electric, usually 1-5 kW. The heat produced is everything from 1 to 4 kW for a 1 kW unit. The industrial systems can be everything between 10 kW and 2 MW. The most common sizes are 250 kW, but 1 MW units are built. The residential systems are installed in one, or a few families buildings, while the larger ones are installed in hotels, hospitals, computer centres and other industrial sites. The cumulative number of small systems built until now is some 5000 units [1]. Of the large stationary some 800 units have been built [2]. Naturally only a fraction of these are in use anymore, because the life time of a fuel cell system is still 30000 h, or less.

The technology can be divided into PEFC and SOFC for residential systems and SOFC, molten carbonate fuel cells (MCFC) and phosphoric acid fuel cells (PAFC) for industrial systems. At present most of the installed residential systems are based on PEFC technology, while most of the installed industrial systems being MCFC with PAFC as a good second. Most of these use natural gas as fuel, while some of the industrial systems use different biogases. The low temperature PEFC requires pure hydrogen, which is produced by reforming natural gas. The electrical efficiency is only 30 % or less. The warm water produced has a temperature of 60 ºC, which means that it has to be used locally. The high temperature fuel cells, MCFC and SOFC have electrical efficiencies of 50 % with 60 % to be expected in the future. They produce water at temperatures of several hundred degrees Celsius. This can be used for producing industrial heat as well as cold, using chillers. Hybrid plants, producing only power, might reach 70% electrical efficiency in the future.

The power sources are used for different purposes. Both in Germany and Japan the residential units are usually operating according to heat need, the electricity produced being an added value. When heat is not needed, the electricity is taken from the grid. Thus they are mainly replacing natural gas boilers, and the additional value compensating for higher investment cost comes from the produced electricity. Taking into account the low electrical efficiency, this is an approach which can be doubted in countries were electricity is produced using high efficiency technologies.

The industrial units are often producing electricity and heat to the grid. In many early applications they are used as back up power for instance in hospitals or computer centers. Because high temperature fuel cells are slow to start, they are often used as main sources, the grid acting as back up.

In Europe, especially in Germany more than one hundred FC units have been installed for demonstration purposes. However, Japan has at the moment the most ambitious program in which 1257 units were installed during 2005 and 2006. These are based on PEM technology with an electrical efficiency of 1 kW. The program continues with an escalating number of units to be installed.

In most areas large government programmes are supporting research and development. In Japan NEDO, in USA USDOE and in Europe EU, in addition to several governments have massive support programmes. It can be estimated that more than 1000 M€ public funding is annually used for research and development. The European program has a goal to reach 1 GW installed capacity in 2015 and annual sales of more than a 100,000 units per annum for a system cost of 1000-2000 €/kW in 2020.

The Finnish funding agency for research and innovation (Tekes) has now started a technology program to support fuel cell commercialisation, including hydrogen technologies.

In Finland one major company is developing 20-50 and later 250 kW SOFC units intended for industrial buildings and APU use in ships.

VTT is doing research and development work both for the development of PEM based power units and SOFC based power units. VTT also looks at the use of residential fuel cells as power sources for power systems of buildings. Several universities are also doing research on fuel cell related technologies.

SIMULATION OF RESIDENTIAL SYSTEMS BASED ON FUEL CELLS

Due to constantly increasing electricity consumption, networks are becoming overloaded and unstable. Decentralization of power generation using small-scale local cogeneration plants becomes an interesting option to improve economy and energy reliability of buildings, in terms of both electricity and heat. It is expected that stationary applications in buildings will be one of the most important fields for fuel cell systems. In northern countries, like Finland, efficient utilization of heat from fuel cells is feasible.

Even though development of some fuel cell systems has already progressed to a field trial stage, relatively little is known yet about the interaction of fuel cells with building energy systems during dynamic operation. This issue could be addressed using simulation techniques but there has been a lack of adequate simulation models. International cooperation under IEA/ECBCS/Annex 42 (http://cogen-sim.net/), aims at filling this gap. Our objective was to provide the means for studying the interaction between a building and fuel cell system by incorporating a realistic fuel cell model into a building energy simulation. A new model for a solid-oxide fuel cell system has been developed using a purpose-justified simplified approach. In this approach a system level model was created, in which a control volume boundary is assumed around a fuel cell power module and the interior of it is regarded as a 'black box', Figure 1.

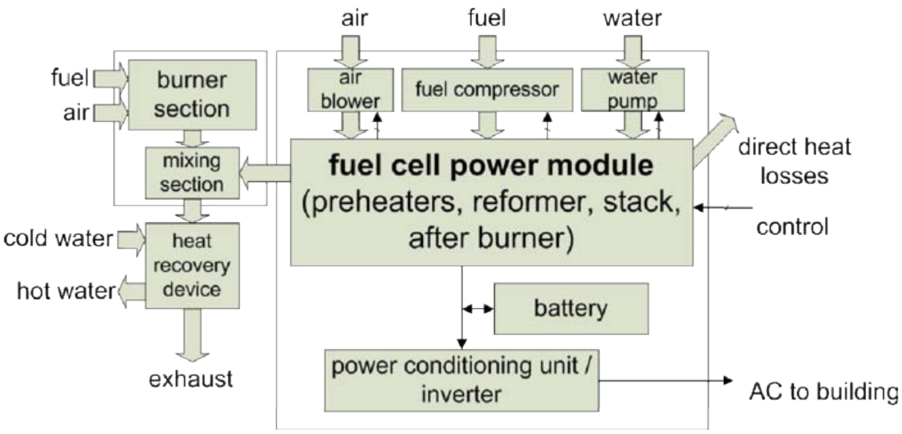

Figure 1. System model diagram according to Annex 42 specification [3].

Inputs to the system are given by flow rates of fuel, water and air. Outputs are electricity, exhaust gas and heat losses from the surfaces.
Subsequently, the system level model has been incorporated into the building simulation tool IDA-ICE (Indoor Climate and Energy) using Neutral Model Format (NMF) language [4]. The first phase of model implementation has been completed and inter-program validation performed within Annex 42 co-operation [5]. In the next phase, model validation and calibration will continue using empirical data. The final goal is to create a comprehensive but flexible model, which could serve as a reliable tool to simulate an annual operation of different fuel cell systems in different buildings, e.g. as in Figure 2.

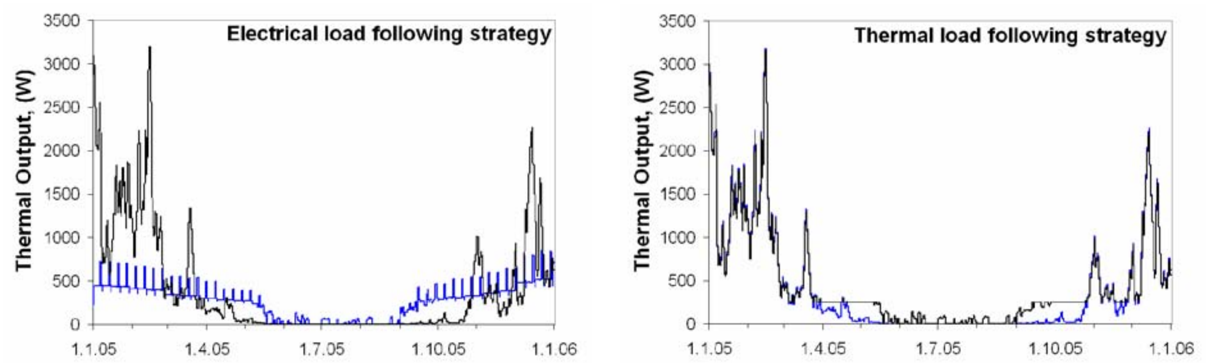

Figure 2. Tentative annual simulation results of heat demand (black curve) and SOFC fuel cell thermal output (blue curve) operated with two different strategies in a single family house. [4]

Another extended application is described in a separate CLIMA2007 paper, in which a community level simulation has been attempted for the first time, covering a group of houses connected by a district heating network. [6]

VTT DEVELOPMENT OF PEFC POWER SOURCE FOR RESIDENTIAL USE

The development of a PEM fuel cell based power pack was started in 2004. The work is co-operation with VTT and Helsinki University of Technology (TKK). Initially the project was restricting to the power level 1 – 2 kW, but at the later phase upgrading to the level of 10 kW has been taken in the program. The construction of the power pack is designed with series production in mind. This project supports the application phase of fuel cell power by deepening the expertise in the essential areas of fuel cell operation. The participating companies include those, which are interested in different phases in the value chain.

The systems are mainly intended to be assembled using commercially available 1-10 kW stacks. Fuel cell stacks are developed in the project mainly in order to understand the technology but also keeping the possibility of own production in mind. The system components are gathered from different manufactures based on performance and cost. System control and control strategies are developed in the project.

Cells and stacks
The initial design of the cell was done in co-operation between TKK Laboratory of Advanced Energy Systems and VTT. Different MEA/GDL constructions were going to be tested in the project in order to know which combination is most suitable for the aimed stack. Because of

limited availability, the tests were performed with two combinations. MEA was PRIMEA Series 56 from W.L. Gore and GDLs were SGK 10-BB from SGL Carbon Group and Carbel from W.L. Gore. Based on the results, the recommended GDL is SGL 10-BB if the cell temperature is relatively low (60 °C), because it was not affected by possible too low or high humidification at that temperature. This also requires that the stack cooling system is efficient enough in order not to increase temperature of some parts of the cell too high. If the cell temperature is higher, the recommended GDL is Carbel.

The effect of impurities [7]

Air contaminants may affect fuel cell performance by many different mechanisms. The effect of NaCl on PEM fuel cell performance was studied. The amount of NaCl used in the experiments presented here was very large. While the exposure time was relatively short, the total salt exposure would be more than most fuel cells could expect to see even under the harshest conditions (marine applications or stationary devices in coastal regions).

Sodium is known to replace protons in Nafion® polymer, thus increasing cell resistance, and chlorine ions adsorb to the catalyst surface, decreasing the number of available reaction sites. In this study NaCl was injected in an operating single cell and degradation of performance observed.

The results showed that the performance loss could be attributed to changes in proton conductivity due to the displacement of protons from the membrane by sodium ions. The effects of chloride ion on catalysis were not evident, although they could be important under longer and higher potential operation. The change in proton conductivity was not accurately reflected by HFR as sodium ion conductivity and ionic conductivity in the electrodes could not be separated.

Optimization based on modeling

Flow field distribution and end plate structure have been modeled and a 2-D fuel cell model that takes into account the inhomogeneous compression of GDL in order to optimize the rib/channel –structure of the flow field plate have been made.

The flow field properties of the cell developed in the project were studied with a finite element method based model that was implemented with commercial finite element software Femlab 3.2. The results showed that for a parallel flow field design, which typically has an uneven flow distribution in the channels, the channel configuration was relatively good. In order to make the flow velocity distribution more even, the distributor channel geometry was modified. The flow velocity distributions in the original and modified distributor channel geometries were also studied experimentally and the results were in agreement with the modelling data.

The main function of the end plates is to offer mechanical support for the stack. The pressure distribution in the stack resulting from the end plates was studied with finite element modelling. The original stack end plates of the POWERPEFC-project cell were two centimetres thick steel plates with a combined weight of approximately 14.2 kg. The results show clearly that despite its bulkiness, the current end plate structure could not provide sufficiently even pressure distributions on the gas diffusion layers of the cell. Different rib structures were tested to improve the gas diffusion layer compression and simultaneously decrease the end plate mass. When aluminium was used as the end plate material, significant improvements on both accounts were reached. [8]

In order to be able to optimize the rib/channel –structure of the flow field plate, the effects caused by the inhomogeneous compression of GDL must be known. The GDL intrusion into the channel was measured with several channel widths and compression values. The evaluation of parameters was done with SGL 10-BA. The gas permeability as a function of thickness was measured with a purpose-build measurement apparatus. The actual model that will be used in the rib/channel –structure optimization has been constructed.

<u>Segmented cell</u>
Anode flow field was divided into 60 segments to study current distribution in the flow field area. The measurement technique used in this work to sense the current from each segment is based on the Hall Effect. The most important result of the measurements has been validation of the calculated result of the shape of the distributor channel.

<u>Design and characterisation of stacks</u>
The stack development has been directed towards cost effective manufacturing and assembling. Due to requirement of low pressure drop on the cathode straight channels were chosen. On the anode and cooling side conventional serpentine channels will be used. Moreover stacks from commercial suppliers have been characterized and also assembled from components.

<u>Bipolar plate development and characterization</u>
VTT Advanced materials has produced composite material with competitive properties for PEM fuel cell bipolar plates. Currently, also steel materials are under study in realistic fuel cell environment.

<u>Development of Instrumentation and BoP for the Power Pack</u>
Currently a prototype version of the balance of plant and control system of a pre-commercial power pack is being tested during operation. The control strategy is towards long life time of the stack, which is achieved by profound understanding of the operation of the fuel cell.

VTT DEVELOPMENT OF SOFC SYSTEMS FOR BUILDINGS

The SOFC systems work aims mainly for larger systems. Here the whole system is dealt with taking into account that mostly hydrocarbon fuels are used. These are natural gas, biodiesel, and gaseous bio fuels. Therefore pre-reformers are developed for all these fuels. The system development includes both the construction and operation of demonstration units and corresponding dynamic system modelling. A 5 kW SOFC demonstration unit fuelled by natural gas and producing electricity to the grid was operated during 6500 hours during 2006 and 2007. The 5 kW SOFC stack was obtained from Forschungszentrum -Jülich in Germany.

<u>SOFC Power station demonstration</u>
The purpose of work is to demonstrate a small-scale combined heat and power plant (CHP) that includes all main BOP components needed in real power plant operation (Figure 3). The system operates at atmospheric pressure using autothermally reformed natural gas as fuel. Alternatively, hydrogen can be use as fuel as well during operation or in system stand-by situations. The system is designed for a planar anode-supported stack. The first 5 kWe stack was delivered by Research Centre Jülich.

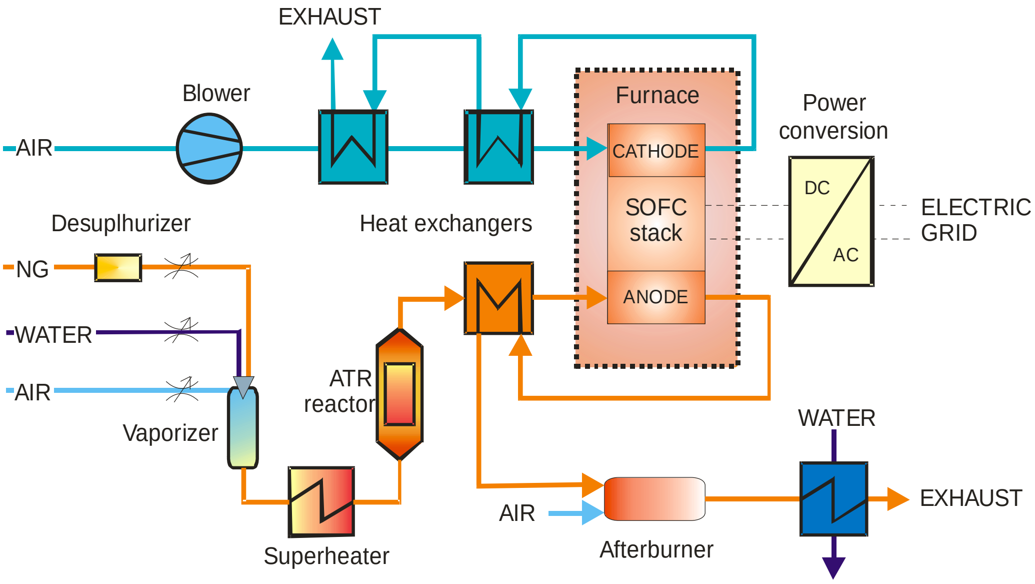

Figure 3. System lay-out of the grid connected natural gas fuelled SOFC based 5 kWe power plant demonstration. The system consist of planar 5 kWe stack, ATR, Hex for fuel, air and exhaust gas, catalytic afterburner, power conversion, grid interconnection, automation, measurement, control and safety system..

The power plant demonstration has been started in May 2006 and currently long period experiments have been stopped after 6500 hours (February 2007). The power plant demonstration produces electricity to the laboratory grid in the power range of 3-5 kWe depending on the operating parameters (Figure 4).

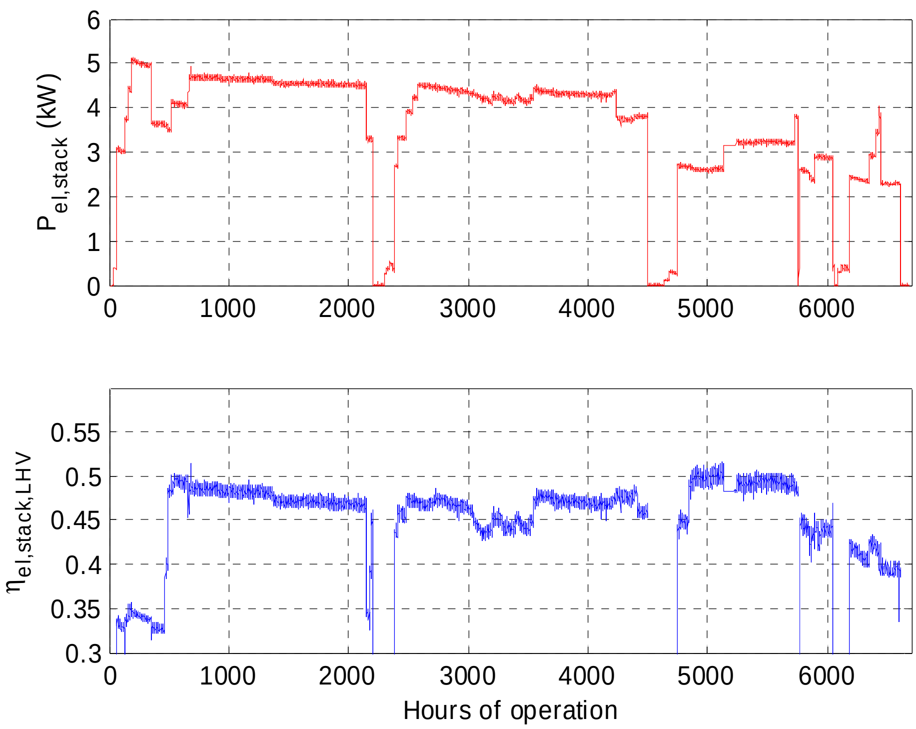

Figure 4. Long period experiments of the power plant demonstration. Degradation is about 1%/1000 hours at 0.35 A/cm2 when fuel utilisation is 70%.

DISCUSSION

Energy production and use is of main concern today in many countries. The main problems dealt with are environmental impacts and reliability of supply. Therefore new solutions are sought for. Fuel cells enable new distributed conversion technology offering high electrical and total efficiency and multi fuel capability. The later makes them especially suitable for high efficiency power production from new types of bio based fuels. Those are renewable hydrogen, biogas, biodiesel, ethanol, gasification gases and other synthetic liquid fuels.

For this reason there is a growing effort to develop fuel cell technology as distributed power sources for buildings. Large international programmes involving private public partnerships have generated and are generating funding for the research and development work.

In Finland VTT is participating in this international effort by supporting the local industry in its work. This work is done in close cooperation with European partners trough EU- funded projects and with international partners trough the International Energy Agency Implementation Agreements. In this work The Finnish Funding Agency for Research and Innovation (Tekes) and its new Fuel Cell Technology Program plays a central role.

ACKNOWLEDGEMENT

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Prospective and adaptive management of small Combined Heat and Power systems in buildings

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SUMMARY

In the context of current promotion of Distributed Generation (DG), micro Combined Heat and Power (CHP) systems for single or small ensembles of (residential) buildings are seen advantageous to combine both decentralized generation and rather high overall efficiency. The latter presupposes intelligent management strategies which have to mediate between energy cost minimization and user comfort aspects. A flexible and auto-adaptive approach of such kind of energy and load management was developed and first results are shown with several operational examples gained from simulation based verification.

INTRODUCTION

Micro Combined Heat and Power (CHP) systems powering up to approximately 10 kW_{el} are considered as a future key technology for energy supply of buildings and settlements from the viewpoints of both heating systems manufacturers and energy suppliers; such CHP plants can be based on conventional Diesel, gas or biomass motors, gas or steam turbines, as well as Stirling engines or fuel cells [1]. In combination with existing public gas and electricity supply these technologies are well suited for cardinal provision of electric and thermal energy in single or multi-family residences and buildings with mixed occupancy of habitation and business establishments. It is evident that reasonable economic operation of CHP systems can only be achieved if power peaks are being moderated; for electrical peaks the public grid connection provides a sound basis, whereas thermal peaks can be smoothed out by both thermal storage and an auxiliary boiler. Efficient operation of such plants pivotally depends on their sound layout for the particular building under regard as well as powerful strategies for energy and load management. Energy management in this context means cost-efficient supply of all (electrical and thermal) loads by intelligent operation of all interacting system components, in particular the CHP unit. Load management means the controlled arresting and releasing of the operation of certain devices, especially larger electro-thermal loads with a significant power demand – for instance a washing machine.

So far, commercial CHP plants do not include sophisticated control structures for flexible and automatic adaptation of their operation to the individual customer behavior, given tariffs and local infrastructure. The existing conventional energy management systems usually control the power of the CHP unit with respect to the actual electricity and heat consumption, but do not apply past or forecasted quantities; user specific consumption behavior as well as dynamic weather influences (such as outside temperature, wind, insolation) are not adequately considered. The potential of installed CHP plants therefore cannot completely be exploited.

In the frame of a current research project – accompanied by manufacturers of CHP plants, network operators and natural gas service providers – efficient strategies for a powerful energy and load management of a micro CHP based energy supply for buildings are being developed and verified on a sound simulation of the complete CHP system. Besides regarding the operational demands and boundaries of the plant components involved, the energy management is designed to fulfill comfort demands by flexible adaptation to user habits (evaluation of past and consequential prognosis of future consumption) as well as local tariffs and infrastructure, and therefore provides economic generation of electricity and heat under inclusion of environmental compatibility. The load management controls the operation time of certain (mainly larger electro-thermal) devices based on evaluation of past user behavior – considering both comfort demands and economic aspects. The management functionalities are elaborated and verified on a detailed operational simulation of the complete CHP system under regard; first results are presented in the following.

METHODS

Simulation model

A versatile Matlab/Simulink® based simulation tool containing a library of Distributed Generation (DG) component models was developed, allowing to arbitrarily compose typical residential micro CHP systems and to investigate the characteristics of the components as well as their operational interdependence within the complete supply system under high temporal resolution and simulation fidelity [2]. The minimum simulation step size is 1 s, while 60 s has proven as a good compromise between accuracy and computation time. This simulation tool was used for test and verification of the CHP energy management focused on here.

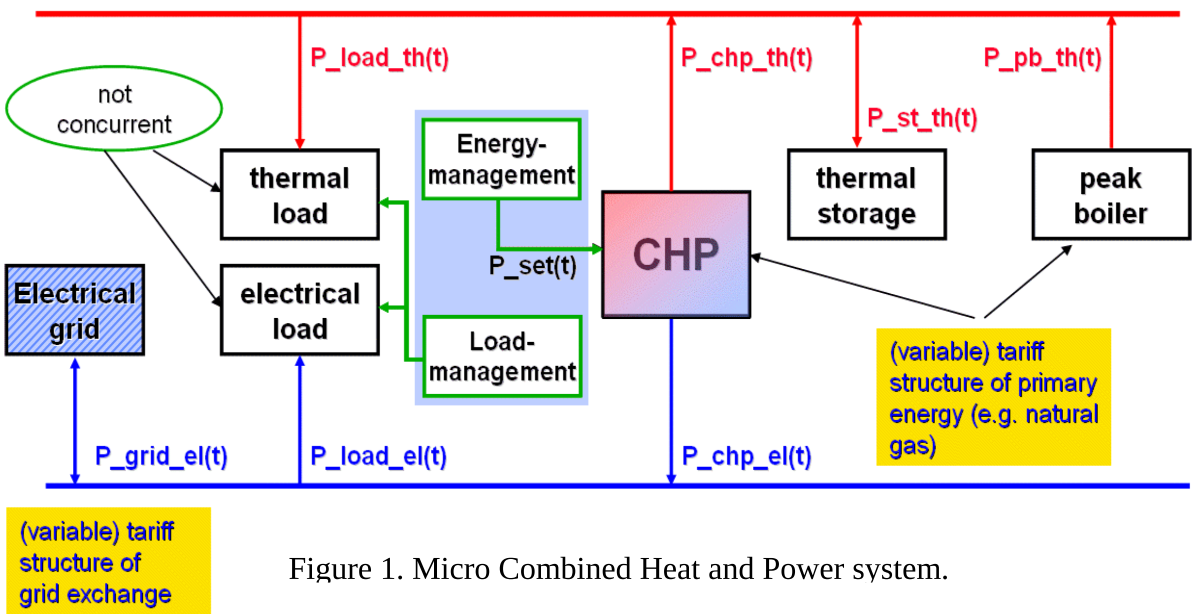


Figure 1. Micro Combined Heat and Power system.

A typical residential CHP system is shown in Figure 1. The system consists of the following main components which were modeled by use of the simulation tool for validating the management functionalities focused on here under largely realistic operational conditions:

Electrical and *thermal loads* are considered as a time series of power samples; these can either be taken from measurements or they can be synthesized by superposition of individual load curves – if required. Innovative energy supply systems are usually installed in connection with modernization of old or construction of new buildings. Therefore, modern building standards and corresponding demand curves were used to develop and verify the CHP plant management. In particular, for the investigations made here, load curves of 140 m^2 low-energy single occupancy houses (4 person's households) over one year in 15 minutes temporal resolution had been available from a German measurement program [3]. As an example, a detailed electrical and thermal (both heating and warm water) demand curve for a cold week in January is shown in Fig.2.

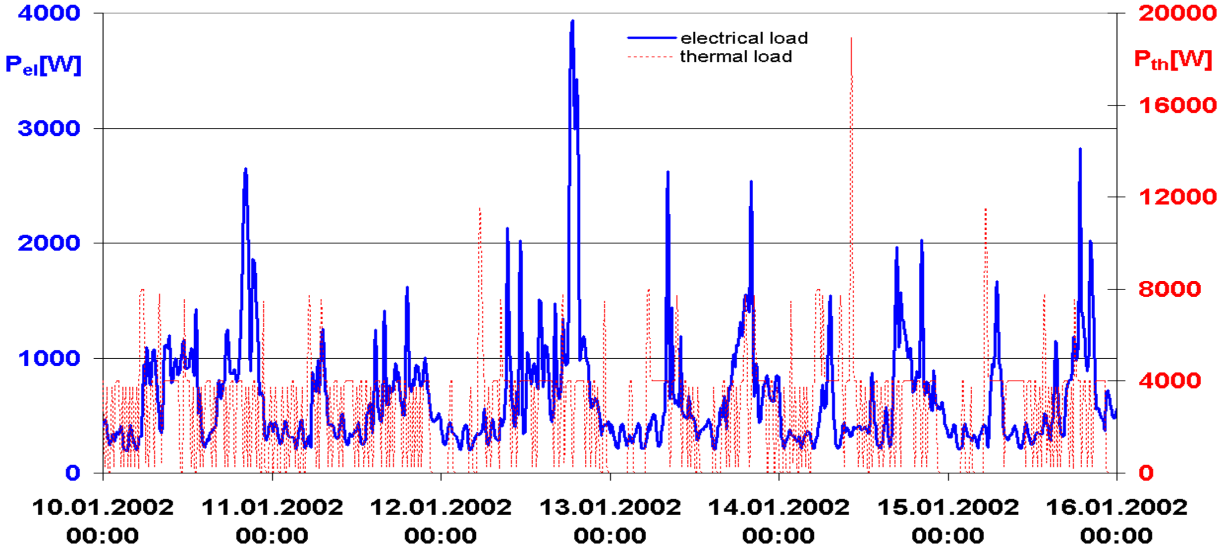
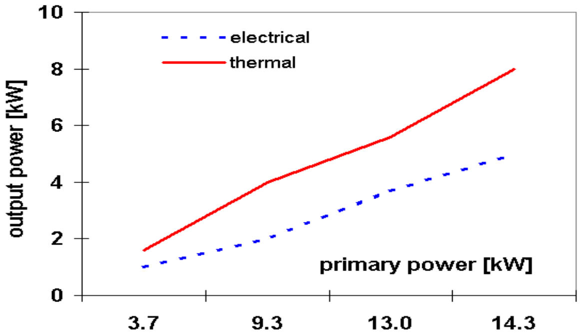


Figure 2. Electrical and thermal load curves in 15 minute resolution (excerpt).

The *CHP unit* is characterized by its stationary power curve including upper and lower operating limits, maximal thermal and electrical power gradients $\delta P/\delta t$, minimal operating and down times, maximal numbers of start-ups and set-point changes per day, performance during the start-up procedure – including auxiliary power demand – and the associated start-up costs. In principle, the simulation system can be set up for various kinds of CHP units as enumerated in the introduction. The examples given in the following are specifically related to a *gas engine based generator set* because such units are presently the most common (and economic) ones used for micro CHP generation. The reference unit has 5 kW_{el}/8 kW_{therm} nominal power and 1 kW_{el}/1.6 kW_{therm} minimal power respectively; the electro-thermal generation interdependency, steady state operating points as well as the start up performance are shown in Figure 3a/b.



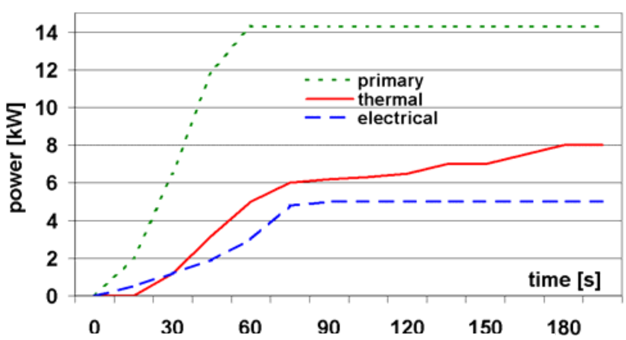


Figure 3. a) Stationary CHP operation points. b) Start-up behaviour of CHP

The *thermal storage* as a crucial component to moderate the thermal power peaks is modeled considering the capacity, maximum charging and discharging powers, maximal power gradients, efficiency of charging and discharging processes, and a fill level dependent loss factor. The storage actually considered here has a capacity of 600 l (≡ 35 kWh at operating temperature); the efficiency of a charge/discharge cycle was set to 95% and a discharging loss factor of 0.1 kWh/h was assumed. The maximal charge and discharge power is 22.9 kW. A dead band control keeps the thermal storage filling level within adjustable threshold values. In case the maximal admissible storage filling level is exceeded the CHP plant is shut down immediately.

The *peak boiler* is operated at nominal power if the thermal storage filling level falls below 20%. Boiler operation is stopped if the storage fill level exceeds 25 %. The simulation model accounts for a characteristic stationary power curve including upper and lower operating limits, and has no gradient limitations.

Coupling to the *electrical grid* – which is considered as unlimited but expensive electricity storage – is constricted by the connection power contracted with the provider.

The *tariff structures* of grid exchange power (feed-in and delivery) as well as primary power (natural gas) consist of both monthly connection rates and time variable kilowatt-hour rates. Even if nowadays electricity and gas rates for private customers are mostly constant (sometimes electricity rates may include special night tariffs for fixed time intervals) new electricity meter information technologies already do exist [4], allowing highly flexible and dynamic tariffs for future Distributed Generation applications; for instance, selective regional price signals can give the system operator an indirect control of a huge number of DG units. In order to prove the flexibility of the system, a tariff structure as shown in Fig. 4 was applied for the examples described in the following. Furthermore, a monthly connection rate of 7.50 € for electricity (each delivery and feed in) and 4.20 € for gas supply were assumed, reflecting actual tariffs in Germany.

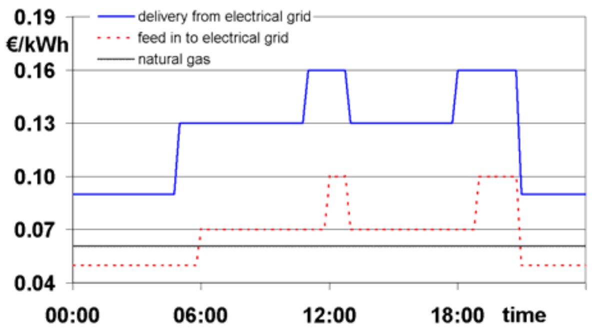


Figure 4. Electricity and gas tariffs applied.

CHP system management

Under regard of the current system states, the operating point dependent component characteristics, their dependencies, varying tariffs, forecasted electrical and thermal loads and potential commercial contracts, ad hoc online cost optimal CHP set-point setting is not a trivial task. The management structure was therefore designed to consist of the three modules *load management*, *load forecast*, and *energy management* (Figure 5), which provide decisions during online operation. The adaptation of the modules to user habits, local tariffs and infrastructure is regularly improved by off-line pre-processing of the archived measurement data, see

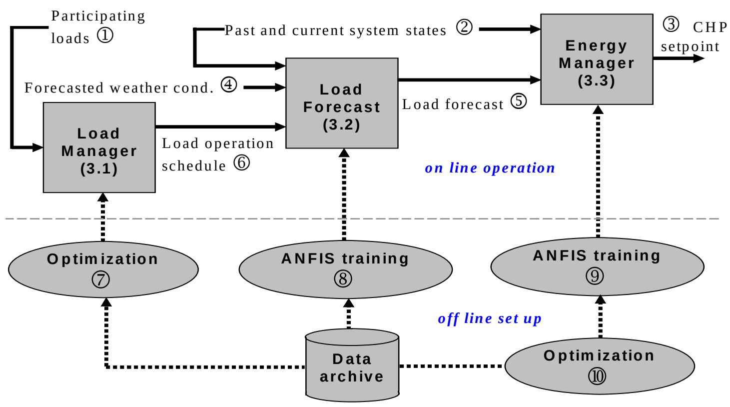


Figure 5. Structure of CHP set point decision making.

Figure 5, lower part. For the very first operation initial default values are implemented. In the following considerations, a weekly pre-processing cycle is applied.

The *load management* module provides a schedule (⑥ in Figure 5) of privileged operation times for each participating load ① one week in advance. These schedules are the result of a cost optimization process ⑦ and are based on the load specific maximal shifting intervals as well as on the measurement data recorded during the previous week. By overruling the proposed schedule, the user still can give emphasis on his comfort demands in which case higher associated costs have to be accepted. Even if controlling particular devices is not yet state of the art, there is current work in progress to establish such techniques, using for instance powerline or wireless communication [5,6]. In the scenarios presented in the forthcoming results section the total electrical load is composed from few controllable devices (e.g., washing machine) and the (uncontrollable) given rest.

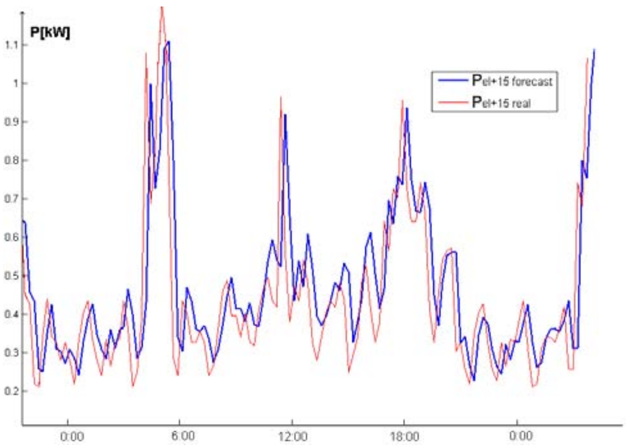


Figure 6. Example of 15 min. el. load forecast.

The *load forecast* module pre-estimates the electrical load at specific, discrete times in the future (15, 30, 60, 120 minutes ahead). Instead of the usually considered influence factors such as temporal and meteorological information the metaheuristic approach applied here (ANFIS, [7]) is based on recognition of characteristic sequences from the data archive. As an example of the forecast quality Figure 6 shows both the 15 minutes forecast and the actual load measured. Owing to the chaotic and hardly predictable instantaneous values of the warm water demand, the expected thermal *energy demand* (warm water and heating) within the next 2 hours is forecasted. The instantaneous difference is balanced by the thermal storage.

The *energy management* finally generates the next power setpoint ③ ensuring cost optimal operation of the micro Combined Heat and Power system. This is achieved based on past, current and forecasted loads ⑤ and tariff information as well as on current and past operating states ②. The appropriate CHP-setpoints are determined by the objective function described below: archived electrical and thermal load curves of the previous week as well as tariff information are applied to the CHP model, and optimal CHP-setpoints are identified with hindsight minimizing the associated costs. The interdependencies between the corresponding inputs and the determined CHP-setpoint values (see above) are periodically identified by means of the metaheuristic approach (ANFIS ⑨) – thereby extending the internal fuzzy knowledge base.

The cost optimal CHP-setpoints and load schedules are regularly determined based on the last week's measurement data, and used to continuously improve the quality of the management system. Owing to the high number of system variables (168 hourly setpoints for one week operation plus additional schedule parameters for all participating loads), the optimization has to be applied *offline* by means of an efficient technique. Taking into account all relevant operating conditions and restrictions, the energy and load management can be formulated as a mixed integer optimization problem:

$$cost_{tot} = cost_{prim\text{-}en} + cost_{grid} - pay_{grid} + cost_{util} \qquad (1)$$

$$\underset{P_{CHP,set},\ load_{lm}}{\mathbf{min}}\ cost_{tot} = f(P_{CHP,set}, tariff_{el,prim}, load_{lm}) \qquad (2)$$

subject to:

<u>energy management:</u>

- 1.6 kW $\leq P_{CHP,th} \leq$ 8 kW
- maximal gradients $\Delta P_{CHP,th}$: +50 / -60 W/s
- start-up cost of CHP : 0.10 €
- maximal number of CHP starts per day: 10
- average time between CHP set point changes: 60 minutes
- minimal CHP run time: 4 minutes,
- minimal CHP stop time: 1 minute
- start-up cost of peak boiler: 0.00 €
- maximal number of peak boiler startups: no restrictions
- maximal no-supply time of loads: 0 s

<u>load management:</u>

- $load_{shift\text{-}min,i} \leq load_{lm,i} \leq load_{shift\text{-}max,i}$

with:

$cost_{tot}$: total costs for supplying all electrical and thermal loads
$cost_{prim\text{-}en}$: costs for primary energy, e.g. natural gas
$cost_{grid}$: costs for connection to and delivery from el. grid
pay_{grid}: refund for feed-in to el. grid
$cost_{util}$: costs for utilities (e.g. start-up, maintenance)
$P_{CHP,set}$: thermal CHP setpoint
$load_{lm,i}$: loads taking part in load management
$load_{shift\text{-}min\text{-}max,i}$: maximal shifting interval in load management

The optimization mode (pure energy- or combined energy/load management) is selected by incorporating one or both of the variables P_{CHPset}, $load_{lm}$ into the optimization process, see Eq. (2). Additional contracts such as minimal CHP overall efficiency for subsidies, or stipulated feed in power can be added if necessary.

In conclusion, the three modules load management, load forecast, and energy management are regularly adapted to user habits, local tariffs and infrastructure by incorporating means of *Computational Intelligence* into the developed control system. The identification of existing correlations between the last week's recorded measurement data (input) and corresponding output data – being results of the optimization or the prognosis – is extended weekly. This process of accumulation and generalisation results in a continuously improving online operation of the CHP system.

RESULTS

In the following first results of the application of the three management modules on a simulated CHP system are presented. The results are based on the load shapes of the January week from Figure 2.

Figure 7 shows the cost optimal plant operation (CHP set points) as determined by the energy management module (load management still disabled). From a broad view it is visible that the energy management in compliance with all active constraints results in

- a shut down of the CHP unit during night hours
- and an individual operation during daytimes.

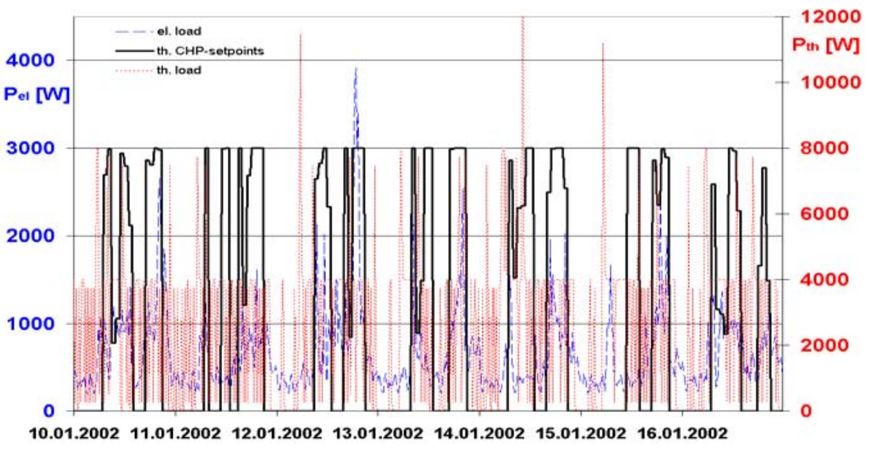


Figure 7. Results of energy management for a cold week in January.

From a more detailed view of the CHP set points, as exemplarily shown in Figure 8 for the Saturday, it can be realized that

- the mutual dependency of the system components,
- the varying electricity and gas tariffs during the course of the day (Figure 4),
- and the constraints as outlined above

do not allow for direct and easy derivation of the optimal system operation from the given electrical and thermal load shapes, thus demanding for intelligent techniques like the approach presented here.

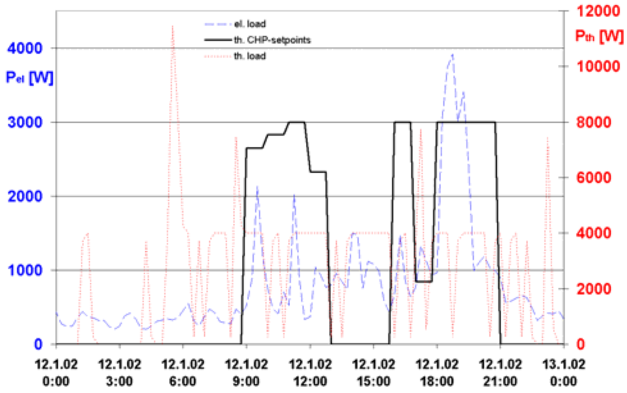


Figure 8. Setpoints determined for Saturday.

As an example of the *modeling detail grade*, the electrical grid exchange power resulting from the CHP operation and the filling level of the thermal storage are shown in Figure 9. It is obvious that the storage is generally charged at daytimes and discharged during the nights, but this is gradually influenced by the electrical grid exchange and the load profiles (Figure 2), thus leading to a rather individual daily operation of the plant.

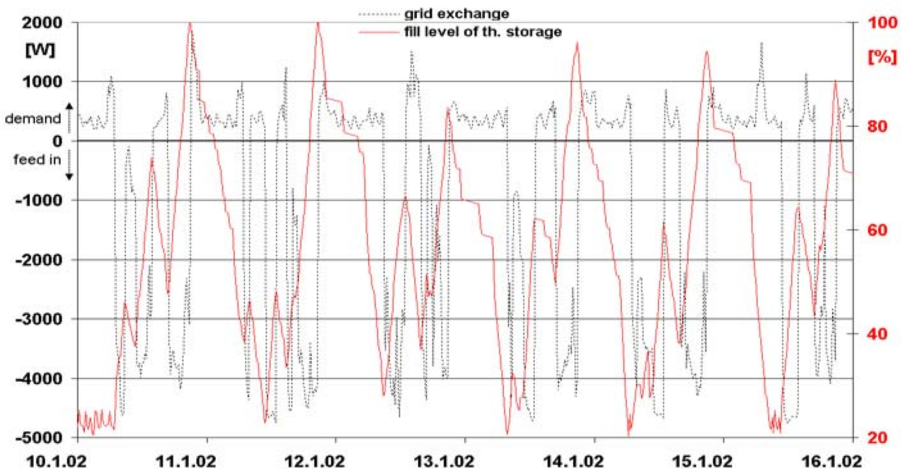


Figure 9. Filling level of thermal storage and el. grid exchange power.

A result of the additional application of the *load management* is shown in Figure 10: for one particular day (January 12th) shifting of certain electrical loads within a 24 hour period has been released. Comparison of the original load curve with the one resulting from load management generally proves that power peaks are defused and high-load periods are shifted to low-tariff regions (according to tariffs from Figure 4).

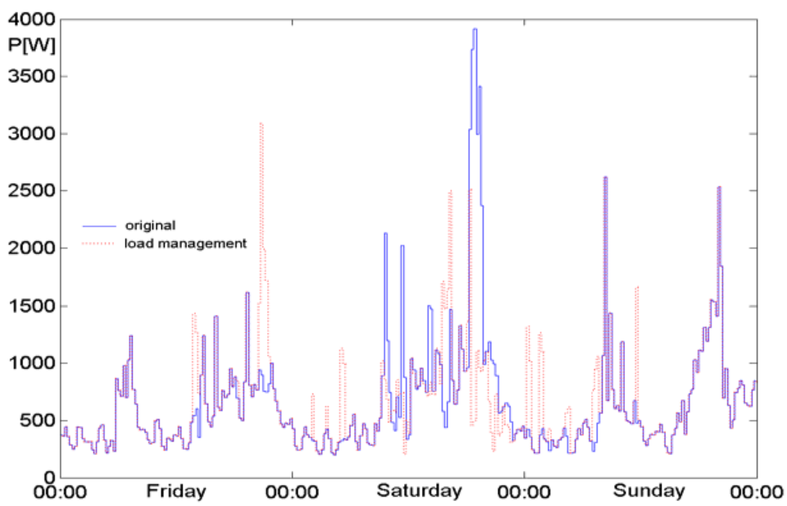


Figure 10. Effect of load management for certain el. devices.

A monetary comparison of different CHP operation modes is finally shown in Table 1 for the winter week under regard.

It is apparent that CHP operation with the energy management enabled (4) is saving approximately 17% of cost compared to no CHP operation - that is, complete electricity demand covered by external grid and thermal demand covered by peak boiler, (3). Having additionally the load management released for one day (see above) saves another 6%, (5). In contrast, blindfold CHP operation at either nominal (1) or minimal (2) constant power is even more expensive than no CHP operation at all.

Table 1: Comparison of energy costs.

CHP operation mode	Weekly cost [€]
1 Constant nominal power	139,16
2 Constant minimal power	54,14
3 No CHP operation	53,22
4 CHP with energy management only 1)	44,00
5 CHP with energy and load management 2)	41,07

1) see Fig. 7-9 2) see Fig. 10

DISCUSSION

Combined Heat and Power (CHP) systems for individual domestic appliances are a promising approach in the context of the recent developments in decentralized energy supply: they are based on an existing infrastructure (gas and electricity systems as well as house installation) and provide good efficiency. Owing to the extreme variation and non-simultaneity of electrical and thermal loads in single households, a flexible and adaptive system management is required. The proposed approach proved in first investigations performed on a detailed plant simulation that

- reasonable set points for the CHP unit are provided by having the energy management rely on historic (archive), actual and forecast data;
- the results are qualified by application of an adaptive load forecast module;
- power peaks are smoothed and heavy-load periods are shifted to opportune times by enabling the load management of certain (electro-thermal) devices;
- the overall energy cost can significantly be reduced.

Thus, by use of intelligent management functionality the benefits of decentralized Combined Heat and Power systems for buildings can fully be exploited.

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Optimal Control of Cogeneration Building Energy Systems

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SUMMARY

We investigate optimal supervisory control of a building energy system with cogeneration of heat and power (CHP). The system consists of a Stirling engine and a supplementary burner, space heating and a domestic hot water (DHW) storage tank. Cost and primary energy (PE)-optimal operation are considered.
The best theoretically possible operating strategy is found using the following assumptions:

- An ideal dynamic model of the system and an ideal prediction of all future disturbance variables (weather, hot-water draws, etc.) are available to the controller ("ideal" here means that model and predictions used by the controller perfectly match "reality", which is used for simulation after applying the control signals)
- The room temperature is allowed to vary within a time-dependent tolerance band (e.g. 21...24°C during the day and 19...24°C at night). Progression of the room temperature is then an output of optimization. A dynamic building model is used, rather than heat demand profiles. A similar tolerance band is used for the DHW storage tank

This strategy defines the so-called *performance bound*, since no real controller can yield a better performance. It is found using model-predictive control (MPC) with moving horizon.
In this general setting, the following results are discussed:

- *Control strategy:* How does the system have to be operated to cover thermal and electrical energy demand with minimal costs, or with minimal PE?
- *Performance assessment:* What annual amount of primary energy and money can be saved by a CHP unit compared to conventional heating?
- *Influence of specific parameters (sizing of Stirling engine)*

The results, although obtained with a Stirling engine, can be used for other CHP units as well.

KEYWORDS: Model-based predictive control, CHP, Stirling engine, performance assessment

INTRODUCTION

Residential cogeneration is an emerging technology with a high potential to deliver energy efficiency and environmental benefits [1]. The concurrent production of electrical and thermal energy can reduce PE consumption and associated greenhouse gas emissions. The distributed generation nature of the technology also has the potential to reduce electrical transmission and distribution inefficiencies and alleviate utility peak demand problems. Leading contenders for residential building cogeneration include fuel cells, Stirling cycles and internal combustion engines.
However, the effective exploitation of the thermal output for space and DHW heating is critical to realizing high levels of overall energy efficiency and the associated environmental (CO_2 and other) benefits. It is believed that building-integrated cogeneration will not deliver the potential benefits outlined above without appropriate control strategies.
This research work has partially been carried out in the framework of the IEA ECBCS Annex42 project called "The Simulation of Building-Integrated Fuel Cell and other

Cogeneration Systems", where also CHP units based on Stirling engines are studied. This paper is based on [2]. Other research projects from Siemens in the field of building automation using the same optimal control approach involve heating applications [3] and integrated room automation [4]. A similar approach has been used in [5].

DESCRIPTION OF THE SYSTEM

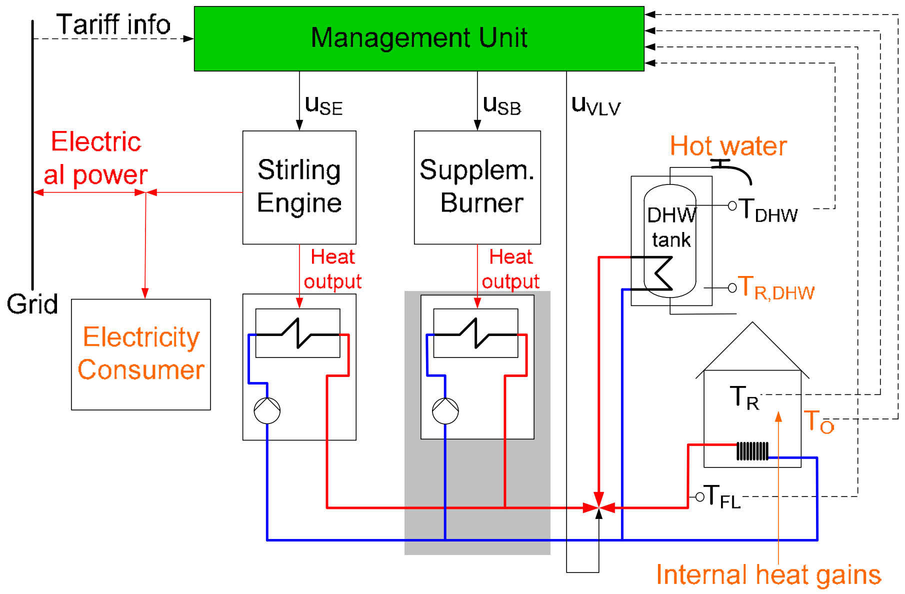

Figure 1: Overview of the system.

An overview on the system is given in Figure 1: A CHP unit heats the building by means of an underfloor heating system, supplies the domestic hot water and may partly or wholly satisfy the electrical demand. The CHP unit is made up of a Stirling engine and a supplementary burner that covers the thermal peak loads. Control signals u_{SE} and u_{SB} represent the gas input of Stirling engine and supplementary burner, and u_{VLV} is the fraction of heat that flows into the heating circuit.

ASSESSED OPTIMAL CONTROL STRATEGIES

Both optimal control strategies must satisfy the following constraints:

Room temperature	$T_R(t) \in [T_{R,min}(t),\ T_{R,max}]$
DHW storage tank temperature requirements	$T_{DHW}(t) \in [T_{DHW,min},\ T_{DHW,max}]$
Maximal flow temperature limitation	$T_{FL}(t) \le T_{FL,max}$
No discharging of DHW tank for heating	$\dot{Q}_{th,DHW}(t) \ge 0$
Minimal and maximal limitation of the control signals (normalized)	$u_{SE}(t), u_{SE}(t), u_{VLV}(t) \in [0,1]$

(1)

Cost optimization

Cost-optimal control minimizes the sum of gas and electricity costs. The cost function is:

$$J_C = \min \sum_{\text{time } t} \left[C_{\text{PE,SE}}(t) + C_{\text{PE,SB}}(t) + C_{\text{el}}(t) \right] \tag{2}$$

where $C_{PE,SE}(t)$ and $C_{PE,SB}(t)$ is the cost of PE, i.e. the gas for the Stirling engine and supplementary burner, and $C_{el}(t)$ is the cost of electricity bought (from the grid) minus the gain from electricity sold (to the grid).

Primary energy (PE) optimization: Electricity credit method

For both PE optimization and performance assessment we compare the CHP system with a system with conventional generation (heat generated in a gas boiler; electricity generated in a gas-driven electrical power plant without exploitation of waste heat). In Figure 2 we first consider the situation *without supplementary burner* (same efficiencies used as later on).

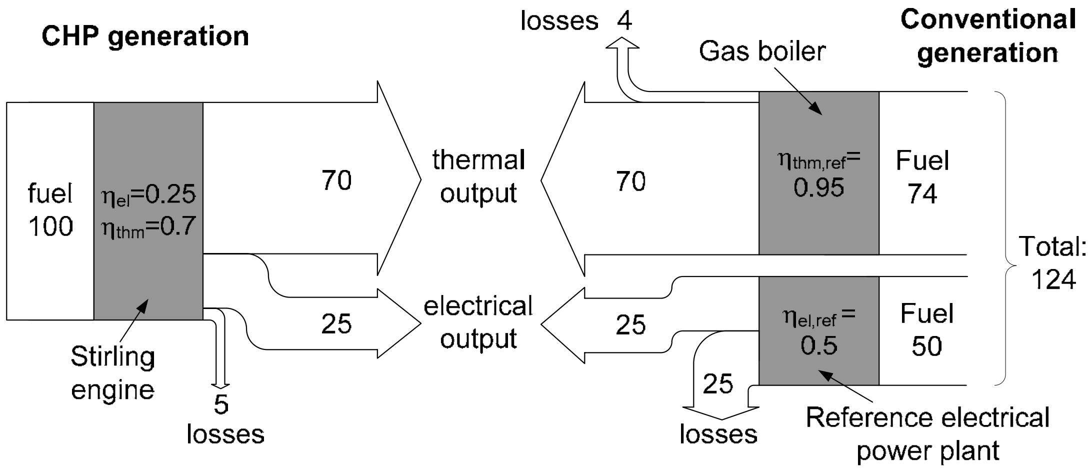

Figure 2: Comparison of energy flows of Stirling engine and conventional generation (situation without supplementary burner).

With 100 kWh of PE (gas), the example CHP unit produces 70kWh of thermal output and 25kWh of electrical output. To produce the same thermal and electrical output with conventional generation (i.e., a conventional gas boiler for heating and a conventional power plant for electricity production), 124kWh of PE would be needed, 50kWh of which for electricity production. For computing the relative savings, we can relate the difference of the PE used to the PE used for heating only with a conventional system, i.e.:

$$relative\,PE\,savings = (124-100)/74 \;=\; 32.4\%. \tag{3}$$

This definition of relative savings is the same as for measures to reduce heating energy consumption, for example improving the building isolation.

There are different possible criteria for PE optimization [9]. Our approach is to minimize overall PE consumption (for heat and electricity, including the grid). Basic idea: The electricity produced by the CHP need not to be produced somewhere else in a conventional power plant (the reference electrical power plant), thus reducing the PE consumption of this reference power plant. These savings (50kWh in Figure 2) are accounted for with a corresponding "credit" in the optimization criterion for PE-optimal control, J_{PE}. We call this method "electricity credit method" [6]. For the system with Stirling engine *and supplementary burner*, J_{PE} is thus:

$$J_{\mathrm{PE}} = \min \sum_{\mathrm{time}\; t} \left[E_{\mathrm{PE,SE}}(t) + E_{\mathrm{PE,SB}}(t) - E_{\mathrm{el,ref}}(t) \right] \tag{4}$$

where $E_{PE,SE}(t)$ and $E_{PE,SB}(t)$ is the PE (gas) consumption of Stirling engine and supplementary burner, and $E_{el,ref}(t)$ is the credit for produced electricity.

Note that without the credit for electricity the Stirling engine is not used at all. PE optimization only favors the Stirling engine if it receives a payback from electricity production. Furthermore, the criterion for optimization should comply with the criterion for performance assessment (assessment of PE used, and of PE savings compared to a conventional system).

Optimization method

We use linear programming (LP) as optimization method. The optimization horizon is chosen to be 24h, with 15-min time steps. A moving horizon technique is then applied to run through the whole period under consideration (e.g. one year).
The optimization problem is formulated in terms of a cost function; see for example eq. (2), that has to be minimized under a set of inequality constraints given by eq. (1). The linearized plant model (in our studies, the plant model was of order 24) is used to express the temperatures $T_R(t)$, $T_{DHW}(t)$ and $T_{FL}(t)$ as a function of the control sequences $u_{SE}(t)$, $u_{SB}(t)$ and $u_{VLV}(t)$ for each time step within the time horizon. The cost function is then minimized by manipulating the optimization variables (i.e., the control sequences).

NUMERICAL VALUES AND PARAMETERS

Building types

We studied three types of buildings with four apartments inhabited by three occupants each: An old one with thin walls and poorly insulated windows, an averagely insulated one corresponding to the German WSV95 building regulation and a well insulated one corresponding to the German EnEV2000 building regulation. More details on these building types can be found in [7]. The apartments all have a floor space of 150 m^2.
The nominal gas input of the Stirling engine has been chosen in such a way as to ensure the basic load, i.e. the engine should supply the average thermal energy needed during a whole year. On the other hand, the supplementary burner is sized so that the peak loads during winter (design temperature of -13.5°C) can be covered by both the engine and the supplementary burner.

Nominal values		Old	WSV95	EnEV2000
Stirling Engine heat output	[kW]	19.9	9.4	5
Stirling Engine electrical power	[kW]	7.1	3.35	1.8
Supp burner heat output	[kW]	41.2	18	8.2
Building heat losses	[W/K]	445.7	194.3	88.4
Building cooldown time constant	[h]	94	162	396

Table 1: Numerical values of the parameters for three building types

Loads and climate

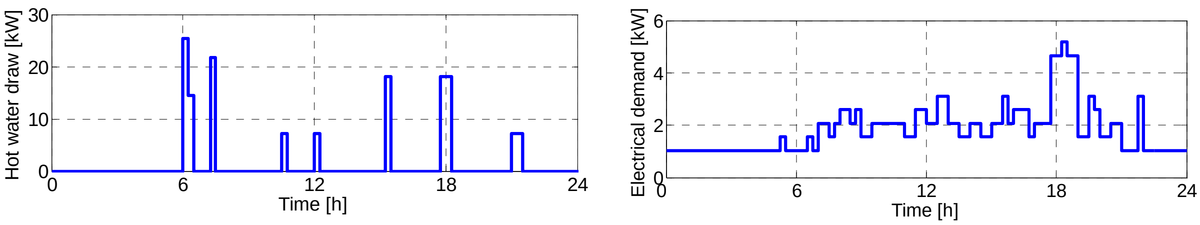

Figure 3: Profiles of hot water draws (left) and electricity demand (right), both in kW. Demand peaks are mapped to 15min-intervals (= sampling time of the control algorithm).

Five disturbance variables are included in our model and are shown in Figure 1 in orange.

- **Hot water**: An overview on typical electric and DHW load profiles is given in [8]. We assume an average daily hot-water consumption of 52 liter/person at a rise of 50K, which corresponds to 36.3kWh for a building with 12 inhabitants. We assume strong peaks in the morning and the evening (see Figure 3).
- **Electricity**: The daily electricity consumption is set to be 43.8kWh with a minimum

consumption of 1 kW and with peaks depending on the time of day (see Figure 3).
- **Outdoor temperature (T_O)**: For whole-year simulations, we use the temperatures measured at Zurich Airport during the year 2000
- **Internal heat gains (persons, electrical equipment etc.)**: Set constant to 0.6kW.
- **External heat gains**: Neglected. Reason: Incorporating solar gains correctly would involve the modeling of sunblinds, including an appropriate sunblind operation.
- **DHW tank**: The air temperature around the DHW tank ($T_{R,DHW}$) is set to be 10°C. The cooldown time constant of the DHW tank is set to be 2 days.

Prices

We use gas and electricity prices of Zurich, Switzerland (all in CHF-cents).
- **Natural gas**:
  - CHP devices: 5.1 cents/kWh
  - Conventional burners: 6.1 .. 7.1 cents / kWh (*lower* for large consumers)
- **Electricity**: Table 2 shows the electricity rates of Zurich.

period	Purchase from grid	Feed-in
Winter day-time	17 .. 19.5 cents / kWh (*higher* for large consumers)	15 cents / kWh
Winter night-time		11 cents / kWh
Summer day-time		7 cents / kWh
Summer night-time	5.0 cents / kWh	4 cents / kWh

Table 2: Electricity pricing, Zurich. Winter: 01.10 – 31.03. Day-time: 6am – 10pm.

Reference electrical efficiency

We adopt a marginal approach with a state-of-the-art gas reference power plant: From the viewpoint of achieving additional electricity capacity with a Stirling engine, the efficiency of an equivalent new central power generation installation should be considered, i.e. 56% for a combined cycle system at the power station, i.e. around 50% with losses resulting from electricity transport from the power plant to the apartment building. (Note that this is an ambitious benchmark. In many publications, the current electricity mix with η_{el} around 35% is used instead.) For more information see [9].
We assume that exported electric energy is consumed in-house or in the vicinity of the Stirling engine. Then the electricity produced by the Stirling engine is not subject to grid losses, in contrast to the electricity from the power plant. The grid losses are therefore included in $\eta_{el,ref}$ for both import and export.

Efficiencies and setpoints

- Efficiencies of the Stirling engine: $\eta_{\mathrm{SE}} = 0.95, \quad \eta_{\mathrm{SE,\,th}} = 0.7, \quad \eta_{\mathrm{SE,\,el}} = 0.25$
- Efficiency of the supplementary burner: $\eta_{\mathrm{SB}} = 0.95$
- Reference electrical efficiency: $\eta_{\mathrm{el,\,ref}} = 0.5$ (after subtraction of grid losses)
- Reference thermal efficiency: $\eta_{\mathrm{th,\,ref}} = 0.95$
- Room temperature: $T_{R,\min} = 19°\mathrm{C}$ between 22:00 and 06:00, $T_{R,\min} = 21°\mathrm{C}$ between 06:00 and 22:00, $T_{R,\max} = 24°\mathrm{C}$
- Flow temperature: $T_{FL} \le 50°\mathrm{C}$
- DHW storage water temperature: $T_{\mathrm{DHW,\,min}} = 20°\mathrm{C}$ and $T_{\mathrm{DHW,\,max}} = 80°\mathrm{C}$.
  ⇨ Note that T_{DHW} is the *average* water temperature of the DHW storage tank, not the measured (sensor) temperature. For the minimal state of charge we assume that 20% of the water in the tank has a temperature of 60°C and 80% has a temperature of 10°C

so that the average temperature is 20°C.

SIMULATION RESULTS

Diurnal progression of the temperatures and control signals (base case)

Figure 4 shows progressions with PE-optimal control over 2 days for a sinusoidal outdoor temperature with a mean value of 7°C, an amplitude of 6°C, and a minimum temperature reached at 3am. The building type is chosen to be an older one for better contrast. Note that the Stirling engine works with full heat output throughout the night. The peaks of the supplementary burner are caused by hot water consumption. The progressions are quite the same for cost optimization with winter tariff as well, but not for cost optimization with summer tariff.

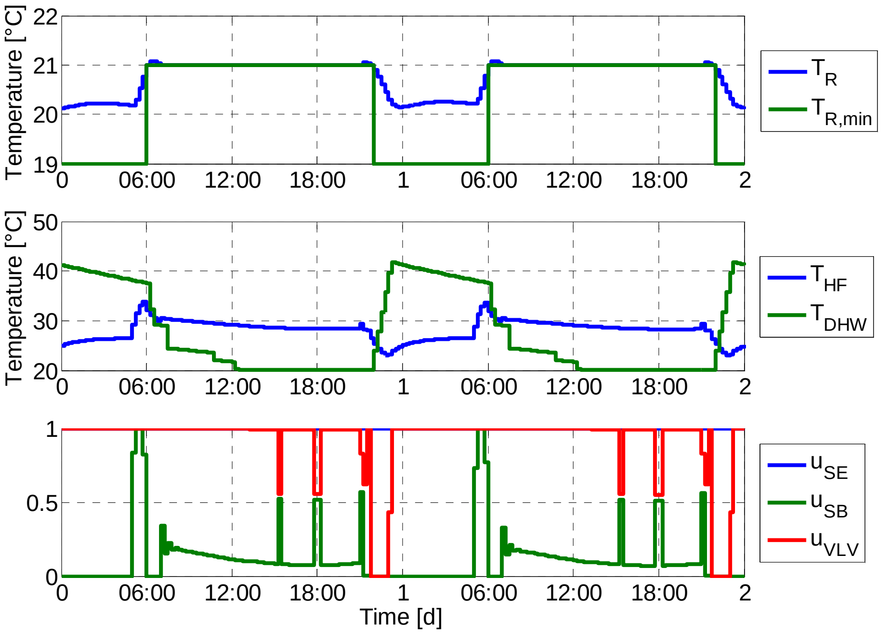

Figure 4: Results of PE-optimal control for an older building with a mean outdoor temperature of 7°C. Middle panel: Flow temperature TFL (blue) and hot water temperature (average over the whole tank, green).

Differences between cost-optimal and PE-optimal control

The characteristics of the control behavior depend in a complex way on many factors. We restrict ourselves to pointing out a few major differences between cost- and PE-optimal control:

- Generally, PE optimization (as well as cost optimization with winter tariff) tries to use the Stirling engine as much as possible and to use the supplementary burner as little as possible.
- As a consequence, the Stirling engine runs throughout the night unless the outdoor temperature is too high. This results in a moderate night setback. The DHW storage may preferably be charged at the beginning of the night.
- Cost optimization with summer tariff tends to shut down the engine, thus producing a more pronounced night setback and postponing the DHW charge to the early morning. As a consequence, more support from the supplementary burner is needed in the morning.

Performance assessment (PA) for whole-year simulations (base case)

The operation cost and PE consumption of the investigated system has been compared to the figures from a conventional generation (reference) system. In the reference system, there is no Stirling engine; however the supplementary burner (conventional gas boiler) is more powerful, such that the total rated thermal output is the same. All electricity is produced by the electrical reference plant ($\eta_{el} = 56\%$). For the computation of the relative PE savings, the PE consumption for heating only with the conventional system is taken as a reference (example without supplementary burner see equation (3)). Table 3 shows the main results:

			Building type		
			Old	WSV95	EnEV2000
Conventional heating (=gas burner alone)		Cost PE consumption for heating	CHF 15,169.-- 196.5 MWh	CHF 8,512.-- 87.4 MWh	CHF 5,650.-- 40.5 MWh
CHP system	Cost-optimal control	Cost saving PE saving	28.5% 20.6%	28.1% 21.9%	23.3% 23.7%
	PE-optimal control	Cost saving PE saving	27.5% 21.4%	25.5% 22.9%	20.6% 24.9%

Table 3: Main PA results for the base case. Reading example for old building: With conventional heating (a conventional gas burner alone), the yearly energy cost is CHF 15,169.- and the PE consumption for heating is 196.5MWh. With cost-optimal control, the one-year energy cost for the reference system with Stirling engine is 28.5% lower than with a gas burner alone, and saves 20.6% of PE. PE-optimal control saves 27.5% energy cost and 21.4% PE.

The differences in cost and PE consumption between both types of optimization (cost/PE) are only a few percent. This means the tariffs in Zurich reward a PE-optimal operation quite well. The equivalent full-load operating time of the Stirling engine is between 260 and 280 days, or about three quarters of the year for all cases. This indicates correct sizing of the SE.
The whole-system PE savings of 21 .. 25% can be related to the PE savings of the Stirling engine alone, which are 32.4% (equation (3)). The latter figure is the limitation for achievable PE-savings with a much more powerful SE. In other words, the base case design (SE for peak load only) achieves 65 .. 75% of the possible savings. This result is backed in the following section.

Variation of the sizing of the Stirling engine

Starting from the base case, the variation of many parameters has been investigated, including time shift of room temperature setpoint profile and electricity tariff profiles; electricity and gas prices; reference electrical efficiency; Stirling and supplementary burner efficiencies; and DHW tank size. In this section, we vary the nominal gas input of the Stirling engine and assess the effect on operation cost and PE consumption. The size of the supplementary burner is varied accordingly, such that the total nominal heat output remains constant. (Note that SE power multiplier = 0 results in the conventional reference system used for the performance assessment, see above).

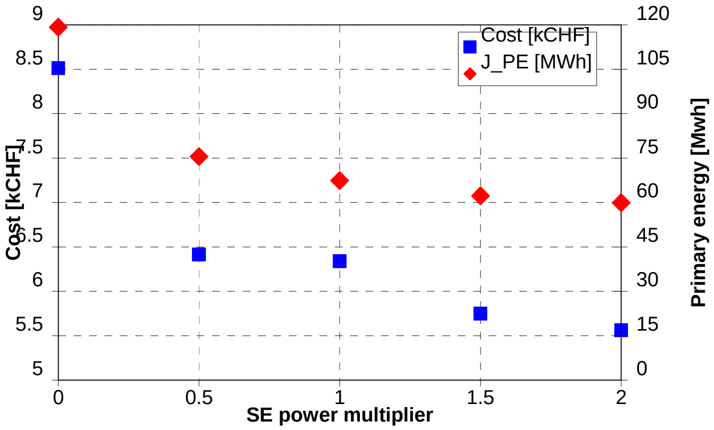

Figure 5: Cost and PE consumption as a function of the SE sizing (power multiplier)

Figure 5 shows the result of PE-optimal simulation over one year with temperatures measured in 2000 for the WSV95 building. We can see that the curves become rather flat when increasing the size of the Stirling engine above the reference case (i.e. SE power multiplier > 1): Doubling the size of the Stirling engine reduces operating cost of only by about 12%. Since on the other hand, investment and maintenance costs increase with the size of the Stirling engine, an SE seems to be correctly sized in the base case.

DISCUSSION

The method described in this paper allows determining the cost- or PE-optimal CHP operation for a given system, with known loads and climate. This optimal solution is the performance bound and can be used for a comparison with real or simpler (suboptimal) control strategies. It also gives us useful hints for developing such simpler strategies.
Numerical values for building model, load profiles etc. used in this paper may be debatable. Nevertheless we are convinced the general quantitative results are usable:

- We investigated a combination of a Stirling engine for the base load and a supplementary burner for the peak load. Using the assumptions on parameters, loads, and climate outlined in this paper, such a set-up can yield a whole-year PE and CO_2 savings are in the range of 20..25% comparing to conventional generation (modern reference electrical power plant with an ambitious $\eta_{\mathrm{el}} = 56\%$).
- Increasing the SE size at the expense of the supplementary burner increases the PE savings only moderately may increase investment costs significantly.

The use of CHP is encouraged by many governments in order to meet the goals of the Kyoto protocol. However in practice, CHP owners and operator will only operate their CHP device in a PE-optimal way if this is financially rewarded. Governments should therefore convince energy suppliers to offer attractive prices for feed-in electricity. (Technically this means that the two cost functions (2) and (4) should be similar, see [2].)

ACKNOWLEDGEMENT

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Economic premises for SOFC cogeneration in Finnish households

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SUMMARY

In this paper, we present the economic analysis of a solid-oxide fuel-cell (SOFC) micro-cogeneration plant in a single-family low-energy house in Finland. Here, we implement a new solid-oxide fuel-cell (SOFC) model in the dynamic building simulation software "IDA – Indoor Climate and Energy" to obtain a match between energy demand and supply. In our computational study, we first estimate the break-even values for both the buyback price of electricity and plant investment that make an SOFC plant financially viable in a comparison with a water-based gas boiler heating system without cogeneration. Second, we determine the sensitivity of break-even prices in terms of the electrical power, overall efficiency and two operational strategies. Our results suggest that the optimal operation would encompass a constant run with electrical power no more than 1 kW_e. In the above case, the break-even buyback price would remain less than 0.04 EUR kWh^{-1}, provided that the overall efficiency exceeds 80 %.

INTRODUCTION

Buildings account for 40% of final energy use, providing a major challenge to the whole energy system. Several studies suggest that the introduction of distributed energy generation would promote the transition to a sustainable energy system (e.g. [1-2]). A solid-oxide fuel cell (SOFC) can be considered a promising alternative to the traditional energy supply for residential buildings due to its general advantages, such as noiselessness and low emissions [3].

The success of new technology in the market presumes, however, that the technology is financially viable with respect to conventional technologies. The key economic drivers are capital costs, energy import and export prices, and plant lifetime [4]. Earlier studies imply that savings in a comparison with traditional residential energy supply solutions may be difficult to obtain by employing solid-oxide fuel cells, partly because of their high capital costs and also due to the high amount of non-utilizable thermal energy (e.g. [5]). The economic key factors that apply to fuel cells, however, strongly depend on the local operational environment and several issues between climatic conditions and taxation policy. A good example is the buyback price of electricity (i.e. the monetary compensation a producer receives for the electricity fed into the grid), which is largely not yet established.

The objective of this study is to evaluate the financial viability of SOFC cogeneration in a low-energy single-family house in Finland. First, we estimate the break-even values for both the buyback price of electricity and plant investment that make an SOFC plant financially viable in a comparison with a water-based gas boiler heating system without cogeneration. Second, we determine the sensitivity of break-even prices in terms of electrical power, overall efficiency and two operational strategies.

METHODS

The applied methodological framework consists of i) creating the hourly electrical and thermal demand profiles of a case building using the advanced building simulation software IDA – Indoor Climate and Energy (IDA-ICE), ii) estimating the hourly electrical and thermal supply profiles of an SOFC plant using the new mathematical model of SOFC implemented in the IDA–ICE and iii) post-processing of the simulation results and conducting an economic analysis in MicroSoft Excel.

IDA-ICE provides an advanced tool for the dynamic simulation of heat transfer and airflows in buildings, allowing the modelling of multi-zone buildings, internal and solar gains and outdoor climate. The software is widely used in Scandinavia both in commercial and research use. It has been developed by the Royal Institute of Technology and the Swedish Institute of Applied Mathematics, reported in detail by Björsell et al. [6] and Shalin [7] and well tested in several studies, e.g. Achermann et al. [8] and Travesi et al. [9].

The SOFC model is based on the principles in Beausoleil-Morrison et al. [10] and is implemented in the IDA–ICE using the modelling language "Neutral Model Format (NMF)", reported by Sahlin et al. [11,12]. Here, the "dynamics" of a 3 kW_e SOFC system is modelled as a modified PI-controller, which drives the fuel-cell system at a limited power change rate. The user provides parameter values manually. Inputs to the system are given by flow rates of fuel, water and air. Outputs are electricity, exhaust gas and heat losses from the surfaces. The model structure with relevant parameters is presented in Figure 1.

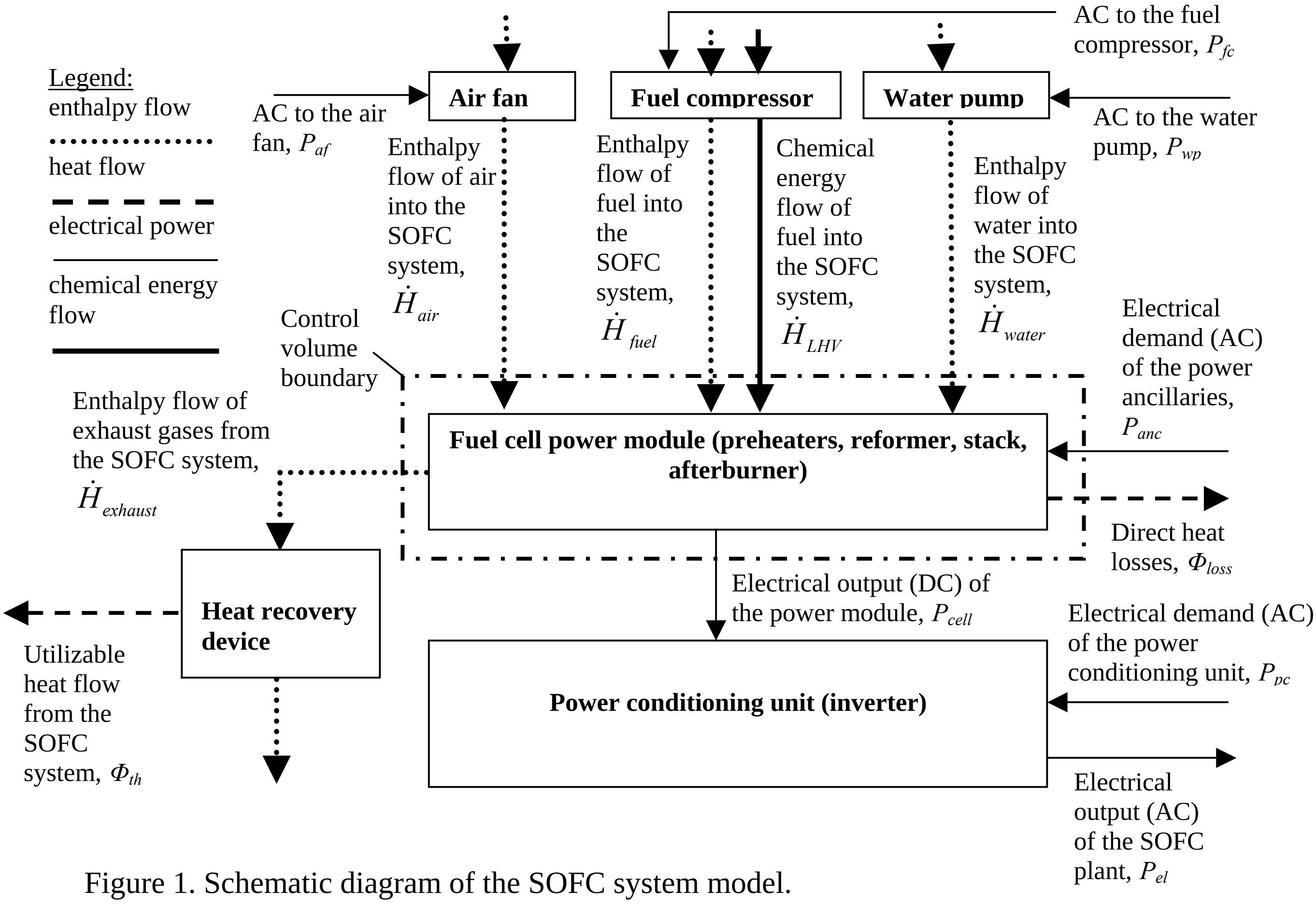

Figure 1. Schematic diagram of the SOFC system model.

The electrical output (DC) of the power module P_{cell} is given by Eq. (1)

$$P_{cell} = \eta_{e,stack} \dot{n}_{fuel} LHV = \eta_{e,stack} \dot{H}_{LHV} \qquad (1)$$

where $\eta_{e,stack}$ is the electrical efficiency, $\dot{n}_{fuel}$ is the molar flow of fuel [mol/s] and LHV is the lower heating value of the fuel. In the present model, the electrical efficiency of the power module $\eta_{e,stack}$ as the function of expected electrical output power is expressed as a polynomial

$$\eta_{e,stack} = (e_{2,\eta} P_{cell}^2 + e_{1,\eta} P_{cell} + e_{0,\eta})(1 - d_{stops} n_{stops}) \cdot \left(1 - MAX\left[\int_0 dt - t_{threshold}, 0\right] \cdot L\right) \qquad (2)$$

where $e_{2,\eta}$, $e_{1,\eta}$ and $e_{0,\eta}$ are polynomial coefficients obtained by experiments. The term "1-$d_{stops}n_{stops}$" is an expression for the degradation of the fuel cell power module's electrical efficiency as a result of stop-start cycling, n_{stops} is the number of stops and parameter d_{stops} is a user-input that represents the fractional performance degradation. The "L-term" represents the degradation of electrical efficiency as a consequence of operation. The time integral indicates the current time of operation, i.e. the accumulated time from the initial system start. L is a fixed, user-defined value that represents the fractional degradation related to operation time and $t_{threshold}$ is a given time when the operational degradation is expected to start.

The energy balance of the power module is

$$\dot{H}_{fuel} + \dot{H}_{air} + \dot{H}_{water} + \dot{H}_{LHV} + P_{anc} = P_{cell} + \dot{H}_{exhaust} + \Phi_{loss} \qquad (3)$$

where H_{fuel}, H_{air}, and H_{water} are the enthalpy flows into the power module by the fuel, air and water, respectively, H_{LHV} is the chemical energy flow into the system by the fuel, $H_{exhaust}$ is the heat flow out of the system by the exhaust gases, P_{anc} and P_{cell} are the electrical draw of ancillaries and the electrical output of the power module, respectively, and Φ_{loss} is the loss heat flow out of the power module.

The enthalpy flows in Eq.(3) are defined as

$$\dot{H}_i = \dot{n}_i h_i \qquad (4)$$

where n_i is the i-th molar flow [mol s^{-1}] and h_i is the corresponding specific enthalpy [kJ/mol]. The relation between temperature T and specific enthalpies is given by the Shomate equation

$$h_i = \frac{AT}{1000} + \frac{B}{2}\left(\frac{T}{1000}\right)^2 + \frac{C}{3}\left(\frac{T}{1000}\right)^3 + \frac{D}{4}\left(\frac{T}{1000}\right)^4 - \frac{1000E}{T} + F - H \qquad (5)$$

where $A,B,\ldots,H$ are polynomial coefficients. Since air, fuel and exhaust gas are mixtures of various gases, the above coefficients are calculated for the gas mixture using molar fraction weighted averages ε_i. The composition of the exhaust is considered stoichiometric. Assuming that the fuel fully reacts, i.e. there is no hydrogen or carbohydrates left in the exhaust gas, the expression for molar fraction weighted averages can be given as

$$\varepsilon_{avg} = \sum_i (\dot{n}_i \varepsilon_i) / \sum_i \dot{n}_i \qquad (6)$$

where ε = {A, B, C, D, E, F, H}.

The electrical demand of ancillaries is defined with separate polynomial equations. The net electrical output power of the SOFC plant can be defined as

$$P_{el} = P_{cell} - P_{af} - P_{fc} - P_{wp} - P_{anc} - P_{pc} \qquad (7)$$

where P_{cell}, P_{af}, P_{fc}, P_{wp}, P_{anc}, and P_{pc} are the electrical powers of the SOFC power module, air blower, fuel compressor, water pump, ancillaries, and power conditioner, respectively. The utilizable heat flow Φ_{th} now depends on the capability of the heat recovery device to transfer heat from the exhaust gases to water circulation. In the present approach, the overall efficiency of the SOFC plant is

$$\eta_{tot} = \frac{P_{el} + \Phi_{th}}{\dot{H}_{LHV}} \qquad (8)$$

In the economic evaluation, savings are assumed to occur during operation, compared to gas heating without a fuel cell (reference). The condition of the financial viability presumes that the discounted savings are equal to the capital costs during a given period of time. This condition is satisfied when

$$C_{I,SOFC} - \frac{(1+r_e)^n - 1}{r_e(1+r_e)^n} \cdot \left[c_{e,p} W_{e,ref} + c_{pr} Q_{pr,ref} - \left(c_{e,p} W_{e,p} - c_{e,s} W_{e,s} + c_{pr} Q_{pr,SOFC} \right) \right] = 0 \qquad (9)$$

where $C_{I,SOFC}$ is the capital (investment) cost of an SOFC plant, r_e is the discount rate for energy costs, n is the number of years on the time period, $W_{e,ref}$ and $Q_{pr,ref}$ are the annual electricity and input energy consumptions in the reference case, respectively, $c_{e,p}$ and c_{pr} are the purchasing prices for electricity and input energy, respectively, $c_{e,s}$ is the buyback price of electricity, $W_{e,p}$ is the annual amount of electricity to be purchased in the case of SOFC operation, $W_{e,s}$ is the annual amount of electricity fed into the grid, and $Q_{pr,SOFC}$ is the input energy consumption of the SOFC plant (directly proportional to H_{LHV}).

RESULTS

The computational study concerns a low-energy single-family house located in the Helsinki area. The habitable area is 131 m^2 and the U-values for envelope, roof, floor, windows and doors are 0.14 W m^{-2}K^{-1}, 0.1 W m^{-2}K^{-1}, 0.15 W m^{-2}K^{-1}, 1.0 W m^{-2}K^{-1} and 0.5 W m^{-2}K^{-1}, respectively. The house is inhabited by four persons; the daily profiles for heat gains and domestic hot water (DHW) demand is illustrated in Figure 2.

The calculations were conducted for the reference case (water-based gas boiler heating) and the SOFC plant with two operational strategies: i) the SOFC plant was operated at a constant power throughout the year and ii) the operation was controlled by the electrical demand of the building. The energy prices were based on the rate schedules of local utilities in December 2006 for both natural gas and electricity. The efficiency of the gas boiler was assumed to be 0.93 [13]. The annual consumption of electricity and natural gas in the reference case were 6089 and 11186 kWh a^{-1}, and the corresponding annual costs were 583 and 771 EUR a^{-1}, respectively. The discount rate of 5 % was used. The annual energy profile of an SOFC system with an overall efficiency of 80 % is presented in Table 1.

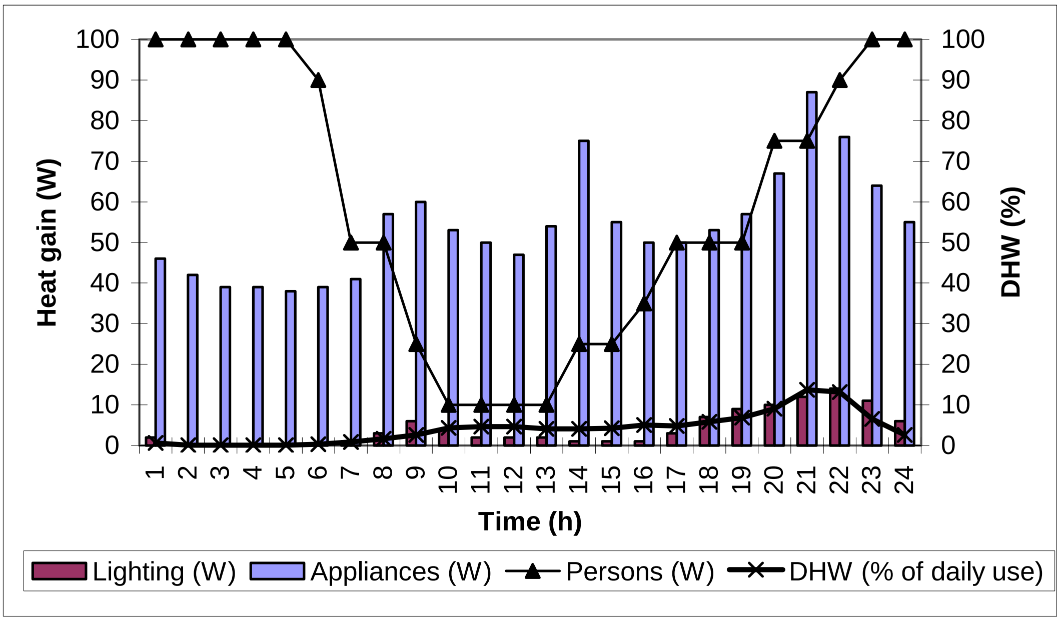

Figure 2. Daily profiles for heat gains and hot water usage [14].

Table 1. Annual energy profile for the SOFC heating system.

Constant operation					
eta = 80 %	Purchased electricity ($kWh\ a^{-1}$)	Delivered electricity ($kWh\ a^{-1}$)	Fuel to SOFC ($kWh\ a^{-1}$)	Fuel to boiler ($kWh\ a^{-1}$)	Fuel total ($kWh\ a^{-1}$)
1 kWe	47	2719	22464	4394	26858
2 kWe	0	11382	39723	2274	41997
3 kWe	0	20108	58204	861	59064
Adapted operation (electricity)					
eta = 80 %	Purchased electricity ($kWh\ a^{-1}$)	Delivered electricity ($kWh\ a^{-1}$)	Fuel to SOFC ($kWh\ a^{-1}$)	Fuel to boiler ($kWh\ a^{-1}$)	Fuel total ($kWh\ a^{-1}$)
1 kWe	87	41	16418	5398	21816
2 kWe	34	55	16565	5351	21916
3 kWe	34	55	16565	5351	21916

As can be seen in Table 1, there is only a minor need to purchase electricity from the grid either when the SOFC plant is operated at constant power or following the electrical demand (adapted operation). The fuel consumption remains quite large in both cases. Instead, in constant operation, the amount of electricity fed to the grid is large. The match between the supply and demand of power in the case of adapted operation (SOFC plant follows the electricity demand) is illustrated in Figure 3.

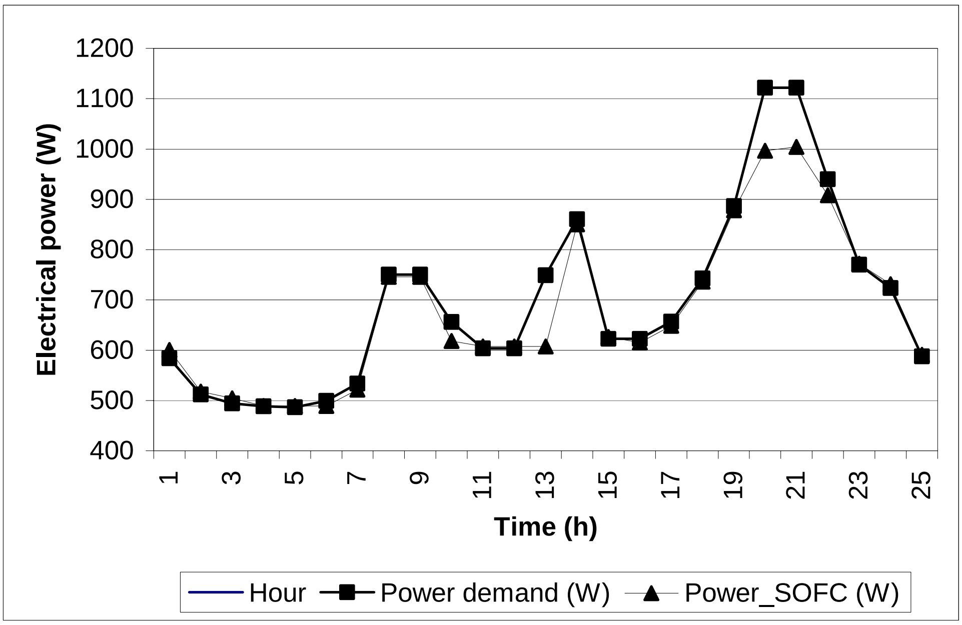

Figure 3. The match between power supply and demand.

The first condition enabling financial viability is that annual savings occur, which presumes here that a break-even buyback price must be first found for electricity (zero savings). Figure 4 illustrates the break-even buyback price of electricity when the SOFC plant is operated at constant power from 1 to 3 kW_e and the overall efficiency, i.e. the ratio of heat that can be transferred from the exhaust gases to the water circulation, varies from 0.5 to 1.0. The result suggests that a significantly low buyback price of electricity is not achievable when the operating power is more than 1 kW_e. Based on the overall efficiency, an obvious reason for this is large, but unavoidable, heat loss in the constant operation, which cannot compensate the incomes from buyback electricity. Figure 4 shows also that annual savings are not possible if there is no compensation for the electricity fed into the grid. The lower the buyback price, however, the more eagerly it will probably be offered by the energy utilities to a private producer. For the above reasons, the further considerations focus on 1 kW_e operation.

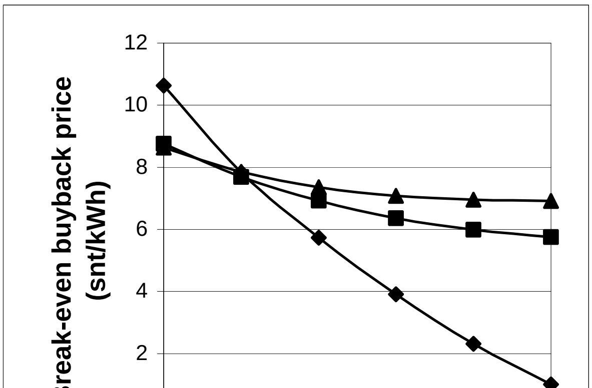

Figure 4. Break-even buyback price of electricity.

In Figure 4, the curve of break-even buyback prices represents zero savings. Considering a 1 kW_e SOFC plant with an overall efficiency of 80 %, the buyback price should be more than 4 snt kWh^{-1} to create annual savings. The relation between the overall efficiency, buyback price, payback period and the maximum allowable capital costs invested in an SOFC plant are presented in Figure 5.

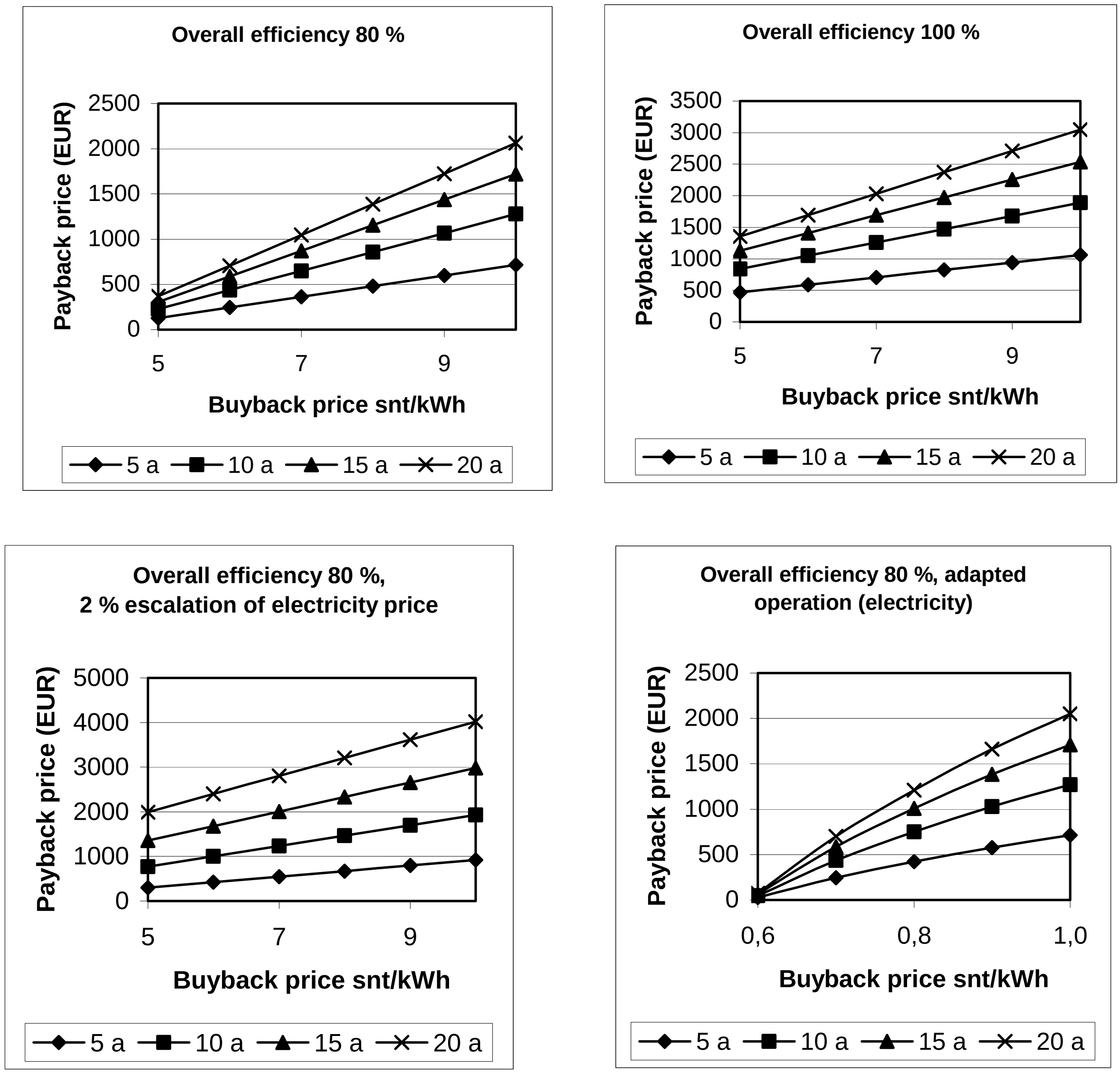

Figure 5. Payback price of SOFC technology.

Figure 5 implies that, in the ideal case (overall efficiency 100 % and payback period 20 years), the maximum allowable technology price would be 3000 EUR. In the sense of earlier studies (e.g. [3]), the life span of an SOFC stack may be less than 10 years, which suggests that an allowable payback period might be rather 5 than 10 years. In this case, the allowable capital costs would remain less than 1000 EUR. The situation from the viewpoint of short payback periods will not improve, even when one assumes that the escalation of electricity price with respect to that of natural gas is 2 % (see Figure 5). Without significant investment support, the investment in SOFC technology does not seem feasible in the case considered here.

DISCUSSION

In this paper, we present the economic analysis of a solid-oxide fuel cell (SOFC) micro-cogeneration plant in a characteristic single-family low-energy house in Finland. Our results suggest that the optimal operation of an SOFC plant would encompass a constant run with electrical power no more than 1 kW_e. Here, the break-even buyback price would remain less than 0.04 EUR kWh^{-1}, provided that the overall efficiency exceeds 80 %. However, taking into account the requirements concerning the payback period and the price of technology, it seems that, without significant investment support, the investment would not be feasible in the case considered here. One should note, however, that this result is only valid for the present case building. Furthermore, our approach did not employ an actual demand profile of a building and the study was unable to analyze the real impact of peak demands. Therefore, more studies are needed to draw further general conclusions about the viability of SOFC cogeneration in Finnish households.

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Experiences on sustainable heating and cooling with an aquifer thermal energy storage system at a Belgian hospital

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SUMMARY

This paper presents the basic parameters and energy flows of an aquifer thermal energy storage (ATES) system combined with reversible water/water heat pumps used for heating and cooling the new hospital Klina (Antwerp – Belgium). The installation system is one of the first ATES projects in Belgium, and its operation is monitored with the aid of a DAQ system. The energy flows, primary energy consumption, CO_2 emission reduction, … were calculated based on DAQ loggings of the first 3 years of system's operation. It is proved that the ATES system is a highly energy efficient system delivering 74% of the total cooling demand. The $COP_{cooling}$ of the ATES system was 45. A primary energy saving of 69% on cooling energy was achieved. These figures prove that in heating and cooling mode, the ATES system is significantly less demanding in terms of primary energy.

INTRODUCTION

The energy use in the Belgian health sector amounts to 8% of service sector electricity consumption and 10% of service sector fuel consumption. To reduce greenhouse gases emitted by heating and cooling plants the health sector needs to shift away from the use of conventional techniques towards renewable energy sources such as the use of underground thermal energy storage [1, 2].

Aquifer Thermal Energy Storage (ATES) was introduced on the Belgian market since 1995 [3]. Since 1998, many companies showed interest in the technology, but this isn't translated into a steady increase of realized projects. This is mainly caused by the hydro geological circumstances. In Belgium, the Northeastern part of the country has very good technical and economical potential for ATES applications. However, the most interesting economical and industrial areas are located outside this region. A thick clay layer covers the Western part of Belgium; the southern part mainly exists of Silurian schist and Devonian rock. At this time, 15 large ATES-systems (> 300 kW) are operational and most installations are monitored.

A number of ATES projects are monitored within the framework of a subsidy program for the stimulation of innovative energy technologies, called "Energy Demonstration Program" (initiative of VEA – Flemish Energy agency - of the Flemish government). For ATES system the monitoring is 3 years in order to have representative results for steady-state performance. Enterprises can get financial support (35% of the extra investment of the innovative investment in comparison to a traditional installation).

In this paper, the monitoring results of an ATES system coupled to reversible heat pumps for heating and cooling a new hospital in northern part of Antwerp is presented with energy and temperature data recordings over 3 years of operation [4].

METHODS

Description of the building

The ATES system and reversible heat pumps (195 kW heating power) was installed at the new hospital "Klina" of Brasschaat, in northern Antwerp (Belgium). It is located at a distance of 12 km from Antwerp center.

The building is located at 4 floors, covering 440 beds, surgery, technical and consultation rooms and a large atrium is implemented between these zones. This imposes simultaneous heating and cooling of different thermal zones, especially during intermediate seasons (autumn and spring). The KLINA hospital was one of the first in Belgium to incorporate comfort cooling in the patient rooms. In order to avoid expensive cooling with traditional refrigeration a system with long term storage of cold and heat in the groundwater was installed. On average, the hospital is in operation 365 days per year and 24 hours per day. The building was designed in the early 90's and the construction works were completed in 1999. The ATES system went in operation in august 2000.

The heat in the patient rooms, kitchen, etc. is distributed by radiators which are designed to operate at temperatures between 80/50°C. The heat pump cannot be used for such a high temperature regime. Gas-fired boilers will cover this demand. The low temperature heat is used in the ventilation installations (air handling units, AHU) and is typically suitable for use with the heat pump and the ATES system.

The predicted cooling power of the building was calculated at 1.6 MW. The top cooling is used in patient rooms, kitchen, consultation rooms, etc at a temperature regime of 11/21°C. The deep cooling demand is especially needed in surgery rooms at a temperature regime of 6/12°C. The central air handling units with water coils (in total 40) provide make up air to the building and the fan-coils are selected for low temperature heating (34/45°C) and high temperature cooling purposes (11/21°C).

Geohydrology

For the application of an ATES system aquifers (or water-saturated sand layers) are necessary. At the site of the hospital a test drilling was made to explore different sediments or formations in the ground. The drilling reports indicates that the ground, up to a depth of 100 m, consists mainly of sand layers separated by a clay layer at the depth of around 80 m. This geology makes the application of ATES preferable. Groundwater can be extracted out of (and injected in) the formations of Diest, Antwerp and Kiel (all water saturated sand layers). The maximum possible groundwater extraction per well is estimated at 100 m^3/h, given 1.2 MW of cooling power.

Description of HVAC and ATES system

The ATES system provides cooling by means of extracting groundwater from the cold well (see also figure 1). This groundwater cools down the building and ventilation air and incorporates the building heat and is afterwards injected in the warm well. At inadequate cooling power or if an insufficiently low temperature is reached by the system, the heat pumps are operated in summer mode (heat pump is working as cooling machine). The condenser heat is also removed by means of the ATES system to the warm well. In the winter groundwater is extracted from the warm well with as aim preheating the ventilation air. This way the winter-cold is taken from the building and injected in the cold well. The heat pump in this mode is used for heating. One of the main advantages of this installation is that the condenser heat is stored in the warm well for use in winter period when the heat pump is operated in summer mode. An overview of the system (operation in summer period) is shown in Figure 1. Table 1 gives the specifications of the ATES system.

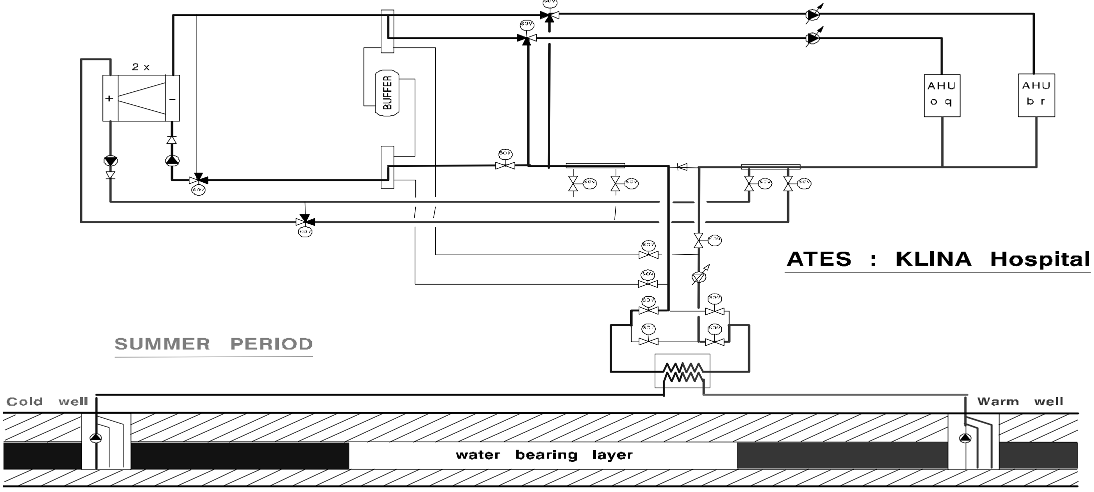

Figure 1. KLINA - ATES operation during summer period.

Table 1. Specifications of the ATES system

Parameter	**Value**
Maximum flow per well	100 m³/h
Maximum cooling power	1.2 MW
Diameter drilling	0.8 m
Depth wells	65 m
Length filters	36 – 40 m
Thickness aquifer	30 – 40 m
Number of cold wells	1
Number of warm wells	1
Distance cold – warm well	100 m
Undisturbed groundwater temperature	11.7°C
Injection temperature warm well	18°C
Injection temperature cold well	8°C

The data logging system

A central BMS is installed, to fully control the operation of all devices of the system and also to provide system monitoring. A full analysis of the ATES system is made by retrieving data on all energy flows (and temperatures), electricity consumption for all auxiliary equipment (boilers, refrigerators, pumps,…), etc. All measurements are stored on a half hourly basis on a computer. For temperature measurements immediate values were stored. Due to some technical problems and data logging incomparability the logging is started at 01/12/2002 (2 years after commissioning). The results presented in this paper cover the period 01/12/2002 - 30/11/2005. The installation and integration of such a groundwater system in combination with a traditional HVAC system isn't self-evident. This was proved with the installation of the first ATES systems where some problems occurred on the integration aspects. A good cooperation between drillers, building/HVAC designers and contractors is absolutely necessary with a clear demarcation of the responsibilities.

RESULTS

Ground water flow and temperatures

During the monitoring period more than 534,000 m^3 groundwater was moved from cold to the warm well and 515,000 m^3 groundwater in the opposite direction. Figure 2 shows the outside air temperature compared to the groundwater flow of the ATES system in the year 2005.

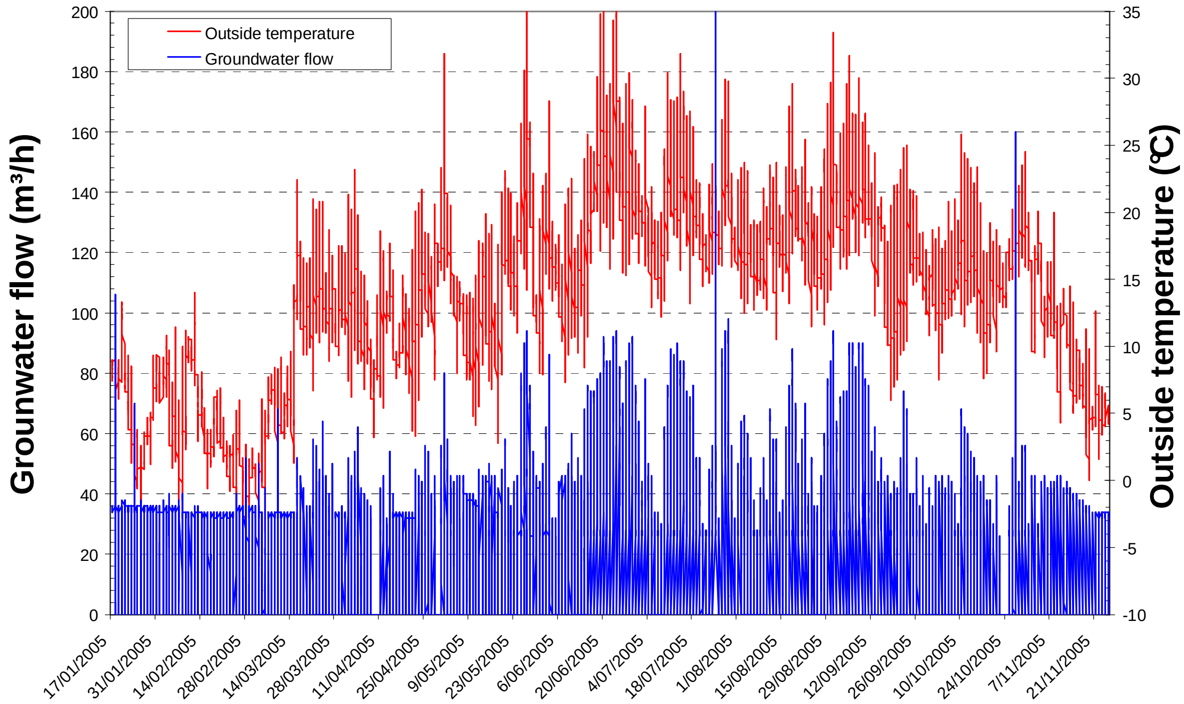

Figure 2. Groundwater flow versus outside temperature (year 2005)

It is observed that the groundwater flow from the ATES system follows the outside air temperature quite well. Due to the large amount of working hours during winter mode the flow is relatively low (35 m³/h) and constant ensuring that enough cold can be stored in the cold well for use in summer period. In the summer period the flow is increased towards the projected value of 100 m³/h. Fluctuations in flow are more likely in the summer period.

Figure 3 shows the groundwater temperature of the warm and cold well. As it can be seen, there is an increase in cold well temperature during summer period (7°C in June 2005 up to 10°C in September 2005). The ground water temperature at the end of the cooling season is still high enough and lower then the natural groundwater temperature (~12°C). This means that there is still some potential for cooling the building and that the ATES system is working properly. Somewhat same conclusions can be drawn for the winter period. Groundwater is extracted from the warm well at temperatures of 16°C and injected in the cold well at temperatures of 7°C. Figure 4 shows the heating and cooling power of the ATES system.

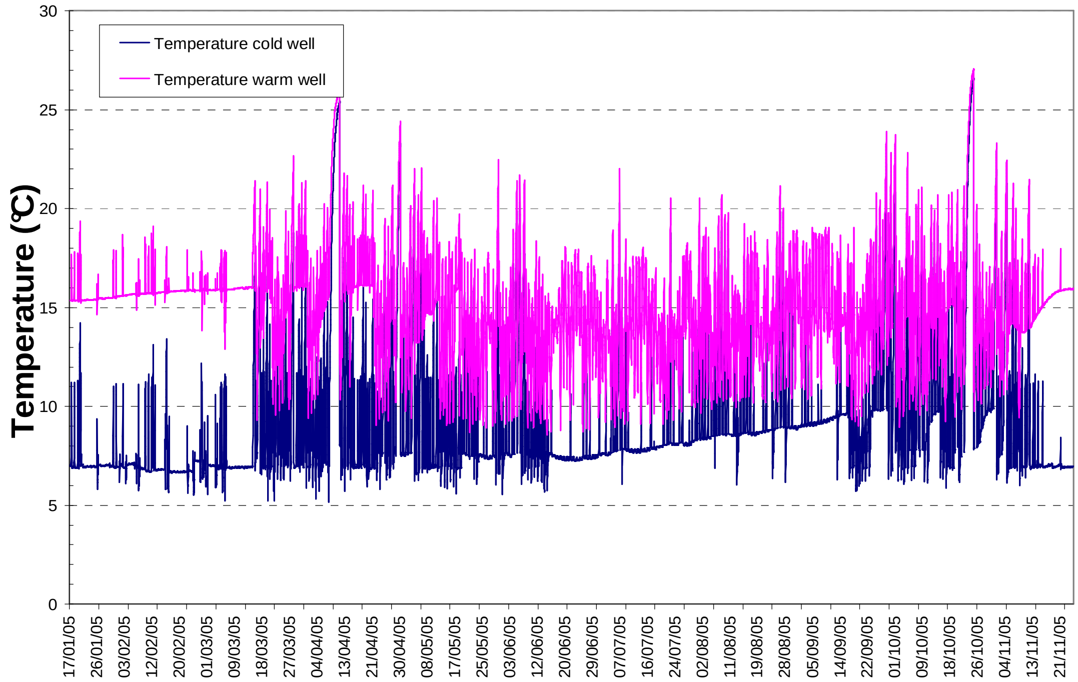

Figure 3. Temperatures groundwater in warm and cold well (year 2005)

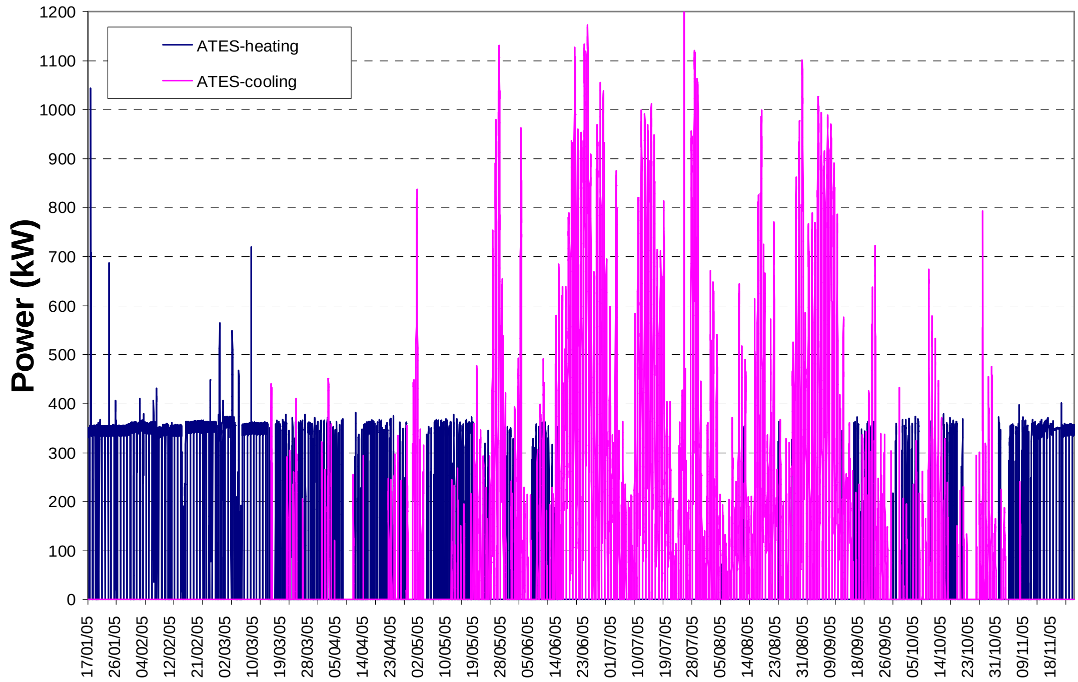

Figure 4. Heating and cooling supply by the ATES system (year 2005)

The ATES system provides a capacity between 100 kW and 1.2 MW. In the winter period there is a more constant but lower capacity supply with an average of 350 kW (provide heat to building or cold charge of cold well). The presence of a heat pump significantly helps in the cold charging process. The number of charging hours increases and is independent of the outside air temperature. Furthermore, the injection temperature is more constant and at an interesting low value. In the summer period the provided capacity varies much more and fluctuates in function of the intensity of overheating of the building.

Energy balance over 3 years of operation

Figure 5 gives a picture of the energy supply of the ATES system and the heat pump covering the entire monitoring period (three years). The largest part (74%) of the ATES cooling is applied directly for comfort cooling of the building (incl. surgery, pharmacy,…). The remaining 26% is used for refrigeration of the condenser of the heat pump during summer situation (operating as a cooling machine). This operation is necessary for supplying cold on low temperature (for dehumidification purposes).
Analogously the groundwater system provides heat, the proportion of direct heat supply to the building is rather limited (17%). Generally, the warm well will provide heat to the evaporator of the heat pump at heat demand. In this way, efficient heat supply is provided to the building and the cold well is charged.

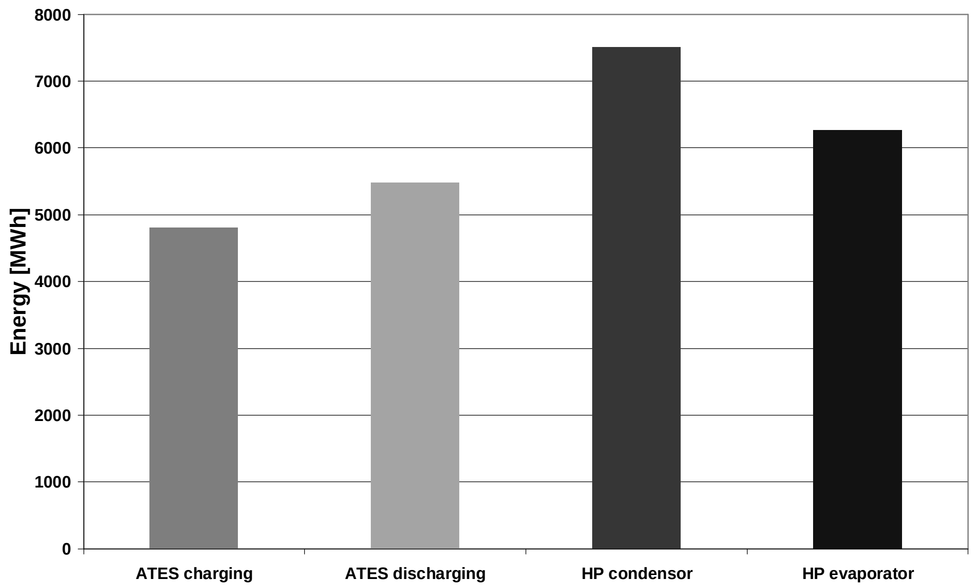

Figure 5. ATES energy amounts (2002 - 2005).

During winter period the heat pump is delivering heat at a $COP_{heating}$ of 6.1 for the first and 5.5 for the second heat pump. During summer period the heat pumps are operating as cooling machine at $COP_{cooling}$ of 5.4 and 4.9. These high figures indicate that the working conditions (temperature levels of condenser and evaporator) are most favorable mainly because the ATES system is designed at relatively high temperatures and the secondary system (the AHUs) at low temperature. The electricity consumption of the ATES system over the monitored period was 229 MWhe given a $COP_{cooling}$ of 45.

It's important to point out that the total heating demand of the building also involves a static heating system with classic radiators and a boiler as heat supplier. It's typical that the ratio static / ventilation heating is about one.

Primary energy savings and CO_2 reduction

In order to determine the primary energy savings, the ATES system is compared with a reference installation based on a water-cooled cooling machine and a gas-fired boiler for heating. Furthermore, the impact of the innovative installation on the CO_2 emissions over the monitoring period of 3 years is calculated and presented in Table 2.

For the primary energy savings and CO_2 emission reduction the following numerical assumptions were made:

- compression chiller efficiency $COP_{cooling}$ = 3.5;
- gas-fired seasonal thermal efficiency η_B = 85%;
- average electrical efficiency Belgian power plant = 44%;
- CO_2-emission factor electricity = 624 g/kWhe,
- CO_2-emission factor natural gas = 181 g/kWht.

Table 2. Primary energy savings and CO_2 emission reduction (2002 – 2005)

	ATES system + heat pumps (installation Klina)	**Gas-fired boilers + cooling machines (reference-installation)**
Electricity consumption	1,474 MWh_e	1,603 MWh_e
Gas consumption	0 MWh_t	9,235 MWh_t
Primary energy consumption	12,060 GJ	39,017 GJ
Reduction primary energy	**- 26,957 GJ**	**-**
CO_2-emissions	**920 ton**	**2,829 ton**
CO_2-reduction	**- 1,909 ton**	**-**
	- 67 %	**-**

The application of ATES reduced the CO_2-production with more than 1,909 ton over the past 3 years. The gas consumption for conditioning of the ventilation air is omitted entirely and is replaced by electricity consumption of the source and heat pumps. Because the heat pumps operate at a very high efficiency, the total electricity consumption remains under this of the reference installation. There is a total primary energy saving of 69% on the climatisation of the ventilation air with a CO_2-reduction of 67 % compared to the reference installation.

Economic analysis

Finally, the feasibility of each option has been assessed and presented in Table 3. Both the investment, maintenance and energy costs (electricity and natural gas) were calculated according to information obtained from the manufacturers and the owner of the hospital. The annual costs are the sum of total energy and maintenance costs. The maintenance costs are assumed to be a percentage of the investments costs.

Table 3. Economic analysis of the ATES system compared to the reference installation (2002 – 2005)

	ATES system + heat pumps (installation Klina)	**Gas-fired boilers + cooling machines (reference-installation)**
Annual costs per year	65 kEuro	100 kEuro
Investments costs	750 kEuro	300 kEuro
SPB ATES system (excluding subsidies)	8.5	-
SPB ATES system (including subsidies)	2	-

DISCUSSION

The HVAC-installation in the new Klina hospital in Brasschaat (Belgium) consists of 2 heat pumps combined with an ATES system (two wells). The system has accumulated 3 years of trouble free operation except the start up period after the commissioning of the installation.

Different parameters such as temperatures, energy flows, … are constantly monitored over this period and the results show that the ATES and heat pump system reaches equilibrium in terms of energy demands. This is also shown in the high COPs of the different components. The primary energy saving reaches 69% (1,900 ton CO_2 is saved) as compared with a reference installation composed of gas-fired boilers and cooling machines.

It's clear that well designed and installed ATES system provides heating and cooling with spectacular efficiency. As a general conclusion it can be stated that of all renewable energy systems (and ATES can be considered as one of them) this technology can present, under good applications, one of the best economical figures and can contributed to the well being indoor quality of hospitals.

ACKNOWLEDGEMENT

The authors are grateful to VEA for supporting this renewable technology and having the opportunity to monitor this ATES system and to gain experience from it. The introduction of ATES in Belgium has been made possible through the exchange of knowledge and experience with other countries through Belgium's participation in the IEA Implementing agreement ECES.

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More sustainable buildings through exergy analysis – Solar thermal and/or ventilation systems?-

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SUMMARY

A more efficient use of energy in the built environment is absolutely necessary. Using the exergy concept delivers more complete information on the use of the energy flows within buildings and opens up room for further improvements within the field. The aim of this study is to gain deeper insight into how the energy is being used in two already efficient building systems, namely a balanced ventilation system with heat recovery and a solar thermal system for space heating supply. Furthermore, desirable optimization chances for them will be pinpointed. The results of energy and exergy analyses of a first study on a typical building with both systems are compared. Annual dynamic simulations were carried out for the space heating case. Another major aim of the present study is to show the key differences between energy and exergy analyses of energy systems in buildings.

INTRODUCTION

Analyses of the exergy flows have been common for the optimization of many thermodynamic systems since the first half of the last century [1]. However, current analyses of the energy consumption in buildings are only based on the study of the energy balance, i.e. on the first law of thermodynamics. To date, the whole energy chain for the supply process is considered by the primary energy factors. Yet, neither the primary energy factors nor the study of the energy balances provide complete information on the use of the potential of the energy, i.e. on the energy quality (exergy) losses during the process [2].

From a thermodynamic point of view, exergy is the maximum theoretical work obtainable from the interaction of a system with its environment, until a state of equilibrium is reached between them [3]. The merit of exergy calculation is that we can explicitly and thoroughly show how different types of energy are used by calculating the flow of exergy and its consumption at different parts of a system [4].

Most of the energy consumed by the building sector is used to maintain constant room temperatures of around 20°C. Because of the low temperature level, the actual demand for exergy in space heating and cooling applications is low. In most cases, however, this demand is filled by high grade energy sources, such as fossil fuels or electricity [5]. Following big mismatching arises between the exergy levels of demand and supply and exergy that shall be used mainly for the production of high quality products is being used instead to meet low exergy demanding applications. Both solar thermal and ventilation systems with heat recovery are able to cover a part of the space heating demands of a building by making use of heat in the environment, i.e. with low exergy content. Thus, they both contribute to matching the exergy content in the demand and supply sides, allowing for a more efficient energy supply for heating in the building sector.

The aim of this study is to get a clearer picture on how energy is used in both systems, and to demonstrate which system leads to a more efficient use of energy within the built environment. Therefore, a first study for a typical building under heating conditions with each system has been carried out and the resulting energy and exergy flows are shown and compared. It is important to underline that the focus of this study is on the whole building facilities system and not on just the individual system components.

SYSTEM DESCRIPTION

The building model consists of a unique square zone (reference zone) whose thermal behaviour will be considered as representative of the whole energy performance of the building. The reference zone has been defined as a corner room with a south/west orientation of the external walls, located within the building at such a height that no influence from the ground and roof exists [6]. In each of the exterior walls, one window of $4.5m^2$ is taken into consideration. The internal heat loads have been set to a constant value year round of $5W/m^2$ [7]. Figure 1 shows a schematic drawing of the particular zone and in Table 1, the main constructive details for the base case are described. Table 2 provides a description of the cases under study.

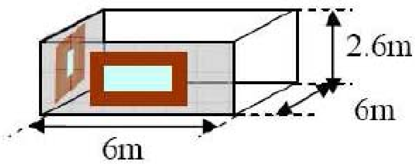


Figure 1. Reference zone for the building under study.

Table 1. Main characteristics of the base case.

Walls	External walls: U_{ew}=0.55 W/m^2K; Internal walls (adiabatic): light gypsum-air construction; Floor and ceiling: U_f= 0.41 W/m^2K
Windows	U_g= 2 W/m^2K; U_f= 1.5 W/m^2K (20% frame surface)
Infiltration rate	$n=0.6h^{-1}$
Heating system	LNG condensing boiler, η=0.95; Floor heating system (30/35)

Table 2. Description of the different cases under study.

Cases studied	Main features
(1)	Base case
(2)	Base case + balanced ventilation system with heat recovery, η=0.8; Mechanical ventilation airflow: 0.4 h^{-1} ; Infiltration airflow: 0,2 h^{-1}
(3)	Base case + 7.5 m^2 flat plate collector field and 890 l storage tank

METHODS

Energy and exergy flows were calculated for the space heating process in the cases stated. The energy/exergy analysis of the processes within the building and its installations for space heating was carried out according to the model developed by Schmidt [8]. This model divides all processes, from the primary energy conversion to the final heat transfer through the building envelope, into several blocks or subsystems. This modular approach aids in developing a better understanding of the processes involved in each subsystem and makes it easier to compare the results obtained for the building installations under analysis. For clearness, the energy required to meet the energy demand of the building is divided into thermal energy/exergy and auxiliary energy/exergy required to operate the different subsystems.

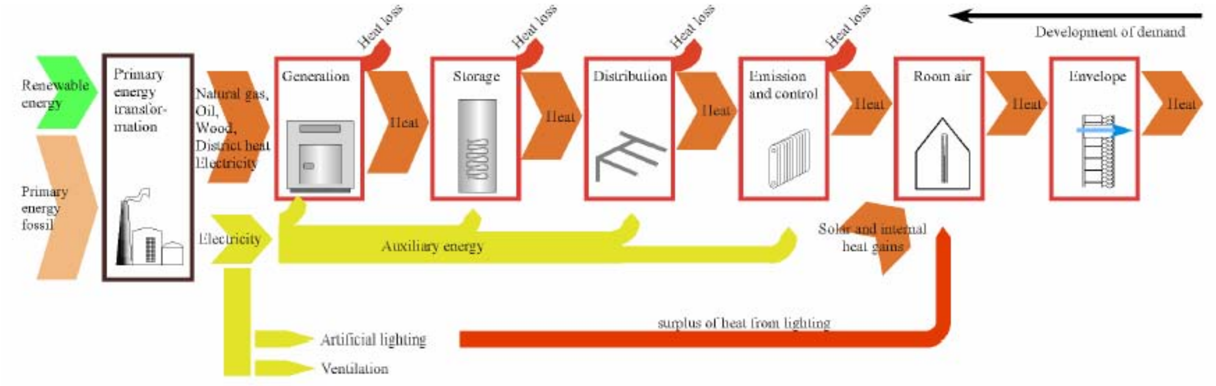


Figure 2. Subsystems and energy flows considered in the model for the energy and exergy analysis in the building. Source: [8]

Energy and Exergy Analysis

The annual energy flows for the heating cases investigated were simulated dynamically with variable time-steps. Primary energy factors for Liquefied Natural Gas (LNG) and electricity are *1.1* and *3.0* respectively, whereas their quality factors are *0.9* and *1.0* [8].

Exergy depends on both the states of the system and its environment, and thus changes on the state of the environment should also be accounted for in the calculation of the exergy flows. Investigations in non-steady state thermodynamics have shown that exergy values calculated with different reference temperatures can be added and compared [9]. The outside air can be regarded as a constant pressure system. Thus, only changes in the temperature of the environment must be taken into consideration. Outside ambient air temperature for the dynamic simulation of the energy flows was subsequently considered as the reference temperature. It is assumed that during each of the variable time steps defined for the energy analysis, no major changes occur in the main variables associated with the process (temperature, energy and entropy). In this way, the exergy associated to the processes can be calculated by applying the quasi-steady-state equations developed by Schmidt [8] to the results of each finite time step of the dynamic simulation of the energy flows and temperatures. In the equations developed in [8], the energy demand and losses of each subsystem were calculated through constant efficiency factors. Here, the energy demands of each subsystem are directly obtained from their dynamic analysis.

The building and the ventilation system were simulated in the same dynamic simulation environment. For the solar thermal system, a different simulation tool was used and the results delivered were introduced into the building model. No real coupling with actual feedback from one simulation tool into the other was done. The output of the solar thermal system is 15% lower when integrated in the building model and simulated in the building dynamic simulation tool. The difference in the solar heat output amounts to 3% of the overall space heating load of the building.

Efficiency Values

To enhance the understanding of the systems analyzed, efficiency values describing their performance were calculated in addition to the absolute energy and exergy flows throughout the subsystems. In many of the involved processes, not all the energy flowing out of the subsystem corresponds to a desired output of it (for instance the return from the floor heating system). Therefore, the rational efficiency as suggested by Cornelissen [10] is the parameter that best describes the actual efficiency of the systems studied here [10]. Rational efficiency is defined as the ratio of the desired output of a process to the useful input required. Equations

(1) and (2) represent the overall energy efficiencies for the generation and primary subsystems.

$$\eta_{overall,prim} = \frac{En_{envelope}}{En_{prim}} = \frac{\Phi_h \cdot}{\left(\Phi_{gen} + p_{aux}\right)}, \qquad (1)$$

$$\eta_{overall,gen} = \frac{En_{envelope}}{En_{gen}} = \frac{\Phi_h}{\left(\Phi_{gen} \cdot F_{p,f} + p_{aux} \cdot F_{p,f}\right)}, \qquad (2)$$

The definition of the overall exergy efficiencies for the generation and primary subsystems is:

$$\psi_{overall,prim} = \frac{Ex_{envelope}}{Ex_{prim}} = \frac{\Phi_h \cdot \left(1 - \frac{T_0}{T_i}\right)}{\left(\Phi_{gen} \cdot F_{q,f} \cdot F_{p,f} + p_{aux} \cdot F_{q,f} \cdot F_{p,f}\right)}, \qquad (3)$$

$$\psi_{overall,gen} = \frac{Ex_{envelope}}{Ex_{gen}} = \frac{\Phi_h \cdot \left(1 - \frac{T_0}{T_i}\right)}{\left(\Phi_{gen} \cdot F_{q,f} + p_{aux} \cdot F_{q,f}\right)}, \qquad (4)$$

The same notation as in [8] has been followed: Φ_{gen} is the energy demand of the generation subsystem, Φ_h stands for the energy demanded by the envelope to the active heating systems, T_i represents the indoor temperature, p_{aux} is the overall auxiliary energy required for the operation of the systems and $F_{q,f}$, $F_{p,f}$, $F_{q,e}$ and $F_{p,e}$ are the primary and quality factors for the fuel (LNG) and electricity respectively.

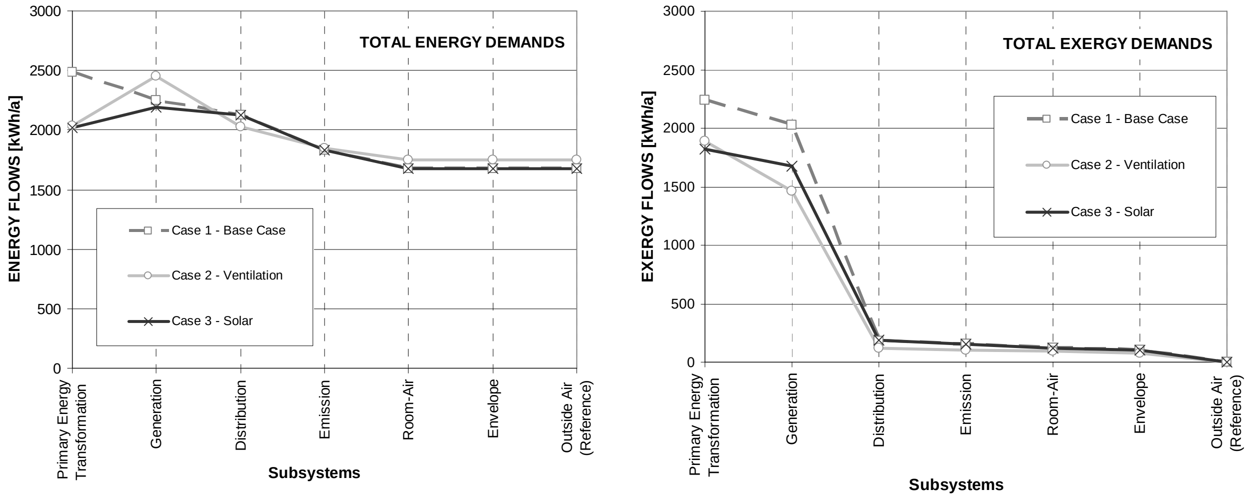
These overall efficiencies represent the degree of matching between the energy and exergy levels of the supply and demand sides.

RESULTS

Figure 3 shows the annual energy (left) and exergy flows (right) through the subsystems for the three cases described under heating conditions. From left to right, energy/exergy is fed into each subsystem, exits it and enters the following one. Because of losses and system immanent irreversibilities and inefficiencies in the heat and mass transfer processes in the components, energy, as well as exergy, dissipates into the environment in each of the subsystems [9].

The energy demand considered in the envelope subsystem does not represent the total losses through the building envelope, but just the part of those losses that has to be provided by the active heating system. The rest of the total losses are covered by solar and internal heat gains in the building. In the primary energy transformation subsystem only fossil fuels are accounted for, i.e. the electricity required for the operation of the ventilation and solar thermal systems as well as fuel and auxiliary energy input required by the boiler. In turn, in the generation system, the energy from the exhaust air which is used by the ventilation system to pre-heat the supply air, and the energy being effectively transferred by the solar buffer tank to the boiler are respectively taken into consideration. This is the reason for the greater energy outputs than inputs that can be seen in the primary energy subsystem in cases 2 and 3.

Despite the relatively high energy demands of the building envelope (around *1700 kWh/a* for all cases), its exergy demands are very low (around *90kWh/a* for all cases) due to the low temperature level required for space heating (*20°C*). When the energy provided by the active heating systems to the indoor air reaches the outdoor ambient its exergy content equals zero since its temperature equals that of the reference state, and the energy flow, no matter how big, has no further potential of performing work.



a) b)

Figure 3. Energy (a) and exergy (b) flows through the subsystems in all cases studied (see Table 2).

The rather horizontal shape of the energy flows in Figure 3 indicates that from an energy point of view, i.e. first law analysis, the systems already seem to be quite efficient. Both the balanced ventilation system with heat recovery and the solar thermal system allow a quite similar reduction of the primary energy that has to be fed into the building for the regarded cases. Overall primary energy efficiency of the building, according to equation (1), subsequently increases: from *67.40%* in case 1 to *85.82%* and *83.05%* in cases 2 and 3 respectively.

The shape of the exergy flows shows quite a different picture: high exergy losses occur in the generation and primary energy subsystems. The overall primary exergy efficiencies amount to only *4.48%, 4.23%* and *5.53%* compared to energy efficiencies of about *80%,* clearly showing that great room for further improvements in the systems still exists. This optimization requirement can be best recognized on the basis of exergy analysis.

The slightly lower efficiency for case 2 as compared to case 1 is due to the high input of electricity (and thus of exergy) required for the operation of the ventilation system and the high primary energy factor associated to electricity. Yet, the efficiency should not be the only parameter for estimating the desirability of one system against another but the absolute values of exergy demand and losses are also to be taken into account.

Both cases 2 and 3 show a reduction in the energy and exergy that has to be fed into the systems and thus represent desirable options compared to case 1. Since both cases 2 and 3 show a similar energy and exergy behaviour, it is worthwhile to take a closer look at them to show differences and optimization possibilities for both systems.

To get a deeper insight into the energy and exergy performance of both systems, their energy and exergy flows are first analyzed without regarding their influence on the overall performance of the building. Table 3 shows the exergy and energy inputs for the ventilation and solar thermal systems. It is to be noted that for the solar thermal system only the energy flowing out of the buffer tank, i.e. being finally transmitted to the boiler, is considered. No energy input or losses within the solar thermal system itself are regarded here. In turn, all existing energy inputs and outputs of the ventilation system have been considered.

The ventilation unit recovers a great amount of energy from the exhaust airflow. However, exhaust air has a temperature level of around *20°C*, and thus the exergy content of this energy flow is very low (see Table 3). To provide this low-quality energy to the building, a high input of electricity is required. In turn, the solar thermal system produces roughly half the thermal energy that the ventilation system makes available, but its exergy content is higher since the temperature of the water coming out of the buffer tank is between *76°C* and *20°C*.

Table 3. Energy and exergy flows through the solar thermal and balanced ventilation systems. "*Th*" stands for thermal energy from the gas boiler and solar system; "*El*" stands for electricity.

Subsystems	Solar thermal system				Ventilation system			
	Energy [kWh/a]		Exergy [kWh/a]		Energy [kWh/a]		Exergy [kWh/a]	
	Elec.	Th.	El.	Th	El	Th	El	Th
Prim. En. Transform.	11.53	-	11.53	-	489.13	-	489.13	-
Generation	3.84	375.43	3.84	39.04	163.04	885.11	163.04	30.84
Distribution	-	375.43	-	39.04	-	708.08	-	16.18
Envelope	-	-	-	-	-	708.08	-	16.18

Efficiencies calculated from the values in the grey cells on the table for each system show how efficient the conversion of fossil fuel input into thermal output coming from the environment is (be it exhaust air or solar radiation). For the ventilation system, these values are *4.34* and *0.099* for the energy and exergy efficiencies respectively, whereas for the solar thermal system they amount *97.76* and *10.16* respectively. These values are not influenced by the primary energy factors but just by the conversion processes taking place in each system. From these values, it can be concluded that for the case under investigation, a solar thermal system is advantageous compared to a ventilation system. The ventilation system could only be competitive if its electrical consumption would first be dramatically decreased.

In the following, the energy and exergy performance of the systems within the building, i.e. of cases 2 and 3, are analyzed and compared in detail. Figure 4 shows energy and exergy flows and losses through the subsystems in the building for cases 2 and 3.

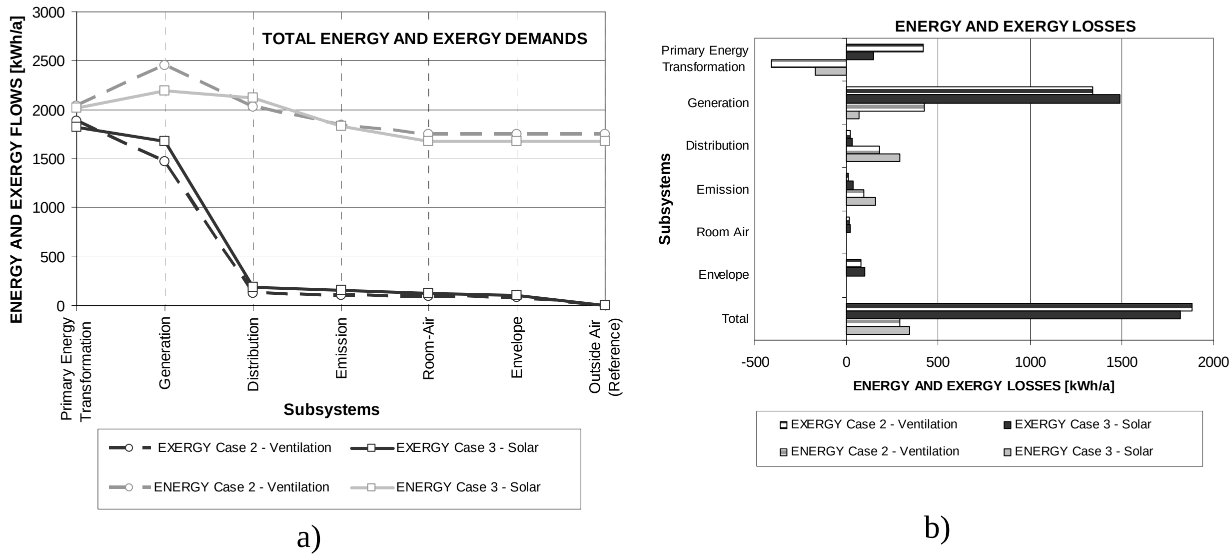


Figure 4. Comparison of the results for cases 2 and 3: a) Energy and exergy flows, b) Energy and exergy losses through each individual subsystem as well as total values.

As it can be seen from the values in Table 3 and in Figure 4, the ventilation system is responsible for reducing the exergy demand of the building envelope. This is due to the low exergy content of the energy it provides to the room-air subsystem, thus saving high quality exergy from LNG to meet that part of the demand. The solar thermal system feeds its output into the generation system, and from there on, its exergy level equals that provided to the distribution system by the boiler, thus making no further reduction of the exergy demand in the envelope possible.

The exergy demand of the generation subsystem is *12.9%* higher in case 3 than in case 2. This is due to the lower amount of energy that the solar thermal system provides (*375.43kWh/a*), as compared to that supplied by the ventilation system (*708.08kWh/a*). This energy is the output from the buffer tank of a solar thermal system whose collector field provides *1062.401kWh/a*. The exergy performance of the solar thermal system could thus be greatly improved if the final heat output for the solar thermal system was higher, i.e. if the energy losses were reduced in the different components within the solar thermal system.

Both systems take more energy from the environment than the primary energy they require, therefore energy losses in the primary system are negative (gains). Despite having smaller gains, exergy losses in that system are lower for case 3 than for case 2 due to the much lower input of electricity it requires (*2.3%* of that of case 2, considering primary energy input of electricity). Energy losses in the generation system are higher for case 2 due to the lower efficiency of the heat exchanger as compared to the boiler. However, exergy losses for case 2 are *l1.5%* lower since energy with very low exergy content gets lost. On the overall balance, exergy losses are *4.2%* lower for case 3, despite the energy balance being favourable for case 2. In Table 4, the overall efficiencies for the primary and generation subsystems, according to equations (1), (2), (3) and (4) are shown. The ventilation system with heat recovery improves both the energy and exergy performance of the building. However, the solar thermal system allows better matching of the exergy levels between the energy supplied and demanded both from a primary and end-exergy perspective.

Table 4. Overall energy and exergy efficiencies for the primary and generation subsystems.

	Case 1- Base Case		Case 2 - Ventilation		Case 3 - Solar	
	$\eta_{overall}$ [-]	$\Psi_{overall}$ [-]	$\eta_{overall}$ [-]	$\Psi_{overall}$ [-]	$\eta_{overall}$ [-]	$\Psi_{overall}$ [-]
Prim. En. Transform.	0.674	0.045	0.830	0.042	0.858	0.055
Generation	0.746	0.049	0.765	0.055	0.714	0.060

DISCUSSION

The ventilation system is very efficient from an energy point of view (*67%*), considering the output it provides to the distribution subsystem compared to the input required for its generation. In addition, it enables to make high amounts of energy with very low exergy content for space heating available, thus contributing to matching the exergy level of supply and demand. However, its high electricity requirements make the overall exergy performance of the system look unfavourable.

On the contrary, the solar thermal system shows a worse energy performance as compared to the ventilation system (*35%*), considering the energy from the collector field as input and the energy delivered by the buffer tank to the boiler as output. In spite of this, the electrical input required by the solar thermal system is 42 times lower, making the overall exergy balance favourable for the building with the solar thermal system.

The electrical demand required for the operation of the fans in the ventilation system has to be lowered. Further research in the development of low consumption fans, which could be called

"LowEx" fans, is absolutely necessary. On the other hand, further reductions in the currently high energy losses through the components of the solar thermal system are also a must to increase its exergy performance. Detailed exergy analysis of the processes in the solar thermal system would be greatly advantageous.

Due to the different nature of the energy inputs and outputs of the systems analyzed, the main differences among energy and exergy analysis have also been shown. Furthermore, exergy analysis has proven to be advantageous in recognizing sources of losses and optimization chances for increasing the performance and reducing the environmental impact of the systems.

FUTURE WORK

The results of a first study for comparing the two systems has been shown here. Further cases where the energy provided by the ventilation and solar thermal systems are equal or equivalent shall be studied, i.e. with bigger solar thermal systems. Furthermore, economic analysis and cost-effectiveness of both systems is to be carried out.

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inhouse4000 – experience with a PEM fuel cell system for stationary applications in buildings

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SUMMARY

Object of the R&D activities of the last 2 years was the development of the new generation of a stationary low temperature PEM fuel cell system (operating temperature max. 80°C) with the following features

- 5 kW electrical power
- 10 T€ / kW electrical power
- robust continuous operation
- lifetime greater than 10.000 h
- system volume less than 1.000 dm³.

This new system is a pre-stage for a market suitable system. With some modifications it also allows an upgrade to a middle temperature system (operating temperature max. 120°C). The accomplished R&D activities are essentially based upon simulations, the development of several versions, experimental analysis in several test systems and a two years lasting field test. The system was integrated into the supply system (heat and electrical power) of a botanical garden. In summary of these activities we could build up a new prototype system which accomplishes all requirements mentioned above. We achieved research results, which allow us the development of a market-driven system suitable for low and middle temperature applications within the next 2 -3 years.

INTRODUCTION

Important international companies, which are developing and producing PEM fuel cell systems, changed their development focus from low temperature PEM fuel cells to high temperature fuel cells (operating temperature up to 200°C, e.g. Plug Power/ Vaillant).This new membrane technology is very CO tolerant and needs no water during operation. Due to these advantages it seems that a high temperature system will be much smaller and cheaper than a low or middle temperature system. But in the present project we demonstrate that proceedings in the R&D of low and middle temperature fuel cell systems have enough innovation potential to be competitive in certain applications. Main reasons therefore are small reformers with carbon monoxide content less than 10 ppm and the high power density of low temperature PEM fuel cells. An increasing of the lifetime and a significant cost reduction for low temperature membranes, the ongoing material and procedural change-over from low temperature membranes to middle temperature membranes are other reasons for it. At least new developments in the nano-technology create new chances for low and middle temperature membranes.

METHODS

We tried to combine theoretical methods and practical tests to get significant results within an acceptable time. One of the central issues is the degradation of the cell. It can be divided into a reversible and an irreversible degradation. Reversible degradation means a degradation of the cell voltage resulting of bad operating conditions (temperature, humidity, mass flow of reactants, pressure), inefficient or inhomogeneous fluid allocation and others. Irreversible degradation means a degradation of the cell voltage resulting of an alteration of the fuel cell components. This includes an alteration and loss of catalysts, carbon corrosion and a disintegration of the membrane caused by radicals. Together with a partner from the german university Hochschule Anhalt we developed a simulation system for low temperature PEM fuel cells. This is an universal mathematical 3 D model and allows the evaluation of physical conditions in a single cell.

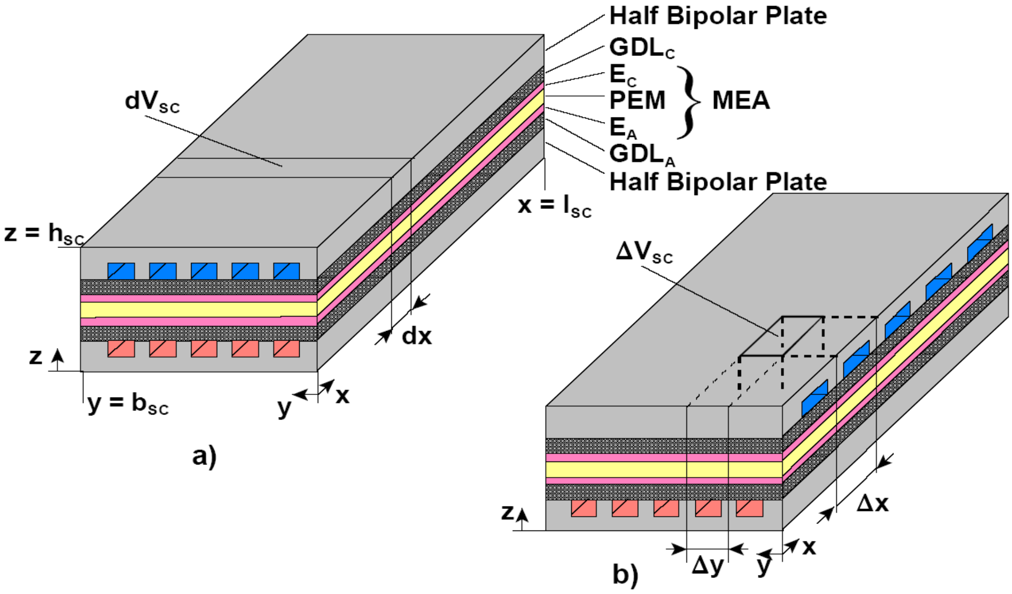


fig.1 illustration of the simulation elements in a cell [1]

The design of the cell and the bipolar plates will be deposited in the simulation. The deposited bipolar plate design will be divided into several simulation elements. One element includes one channel and a half land on each side of the channel. One simulation element includes all layers beginning with the cathode bipolar plate through the gas diffusion layer and the MEA to the anode bipolar plate. The number of modelling elements is selectable. The maximal solution for a single cell is 93 x 93 modelling elements. Each modelling element delivers 50 parameters, which can be analyzed to evaluate the cell.
Most important aim of the simulation was to create a very homogeneous allocation of the fluids, the temperature and the current density. To reach this we diversified the operating conditions (temperature, humidity, pressure, mass flow of reactants) and the structure of the bipolar plates. Another aim was to reach parameters which allow a reduction of costs.

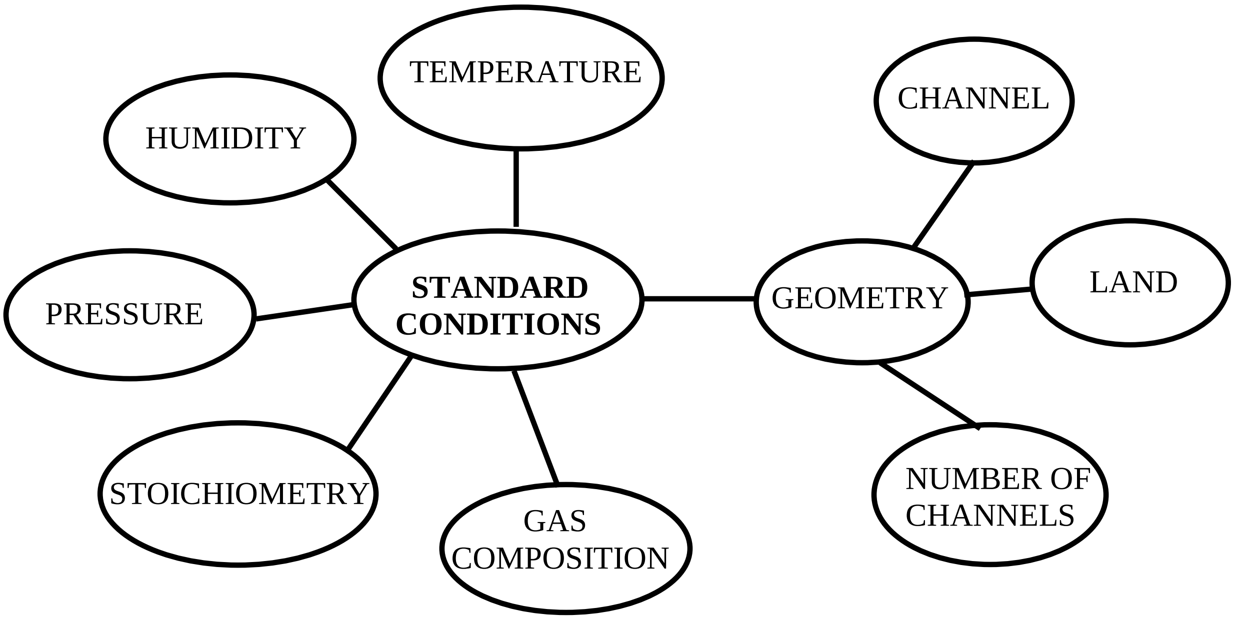


fig.2 variation of parameters for simulation

Another important parameter was the flow direction. All simulations were made with co flow of the fluids and counter flow of the fluids. After the simulation we obtained statements concerning the optimal operating conditions in connexion with the best bipolar plate structure. We could see as shown in fig. 3 the homogeneity of the current density and its changes by differing the operation conditions or flow direction of the fluids.

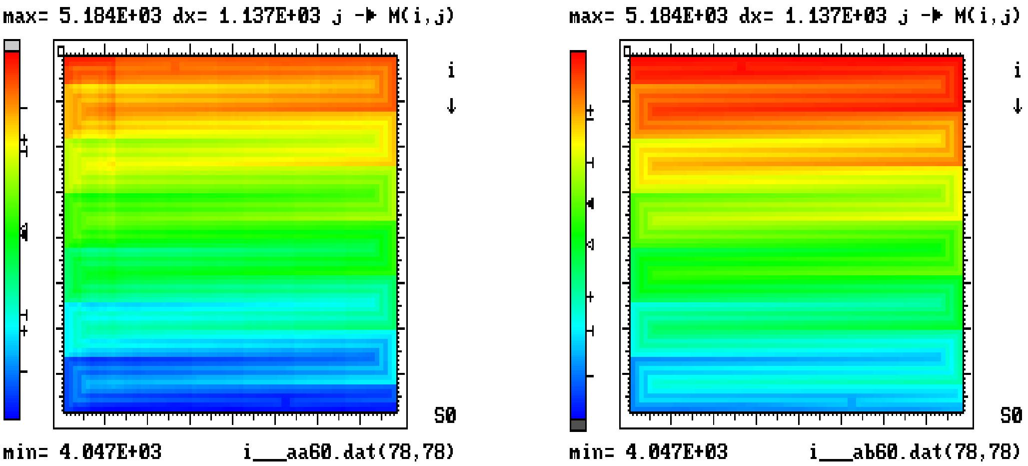


fig. 3 simulation result of one bipolar plate design with different operating conditions [2]

Due to the simulation conclusions we developed several designs for bipolar plates. At first we made some pre-tests. That means we evaluated physical parameters like contact pressures, materials and tolerances. But we also evaluated economical parameters like production costs now and later (mass production). Passing these tests the bipolar plates were integrated into short stacks (300 – 500 W). These stacks were operated several hundred hours. These test results where compared to the simulation results. We obtained a very good consistence between test and simulation. The best short stack/ bipolar plate design was chosen and operated for several thousand hours. The tests with the other designs were terminated.
We had to iterate this process to find the best operating conditions and the best bipolar plate design. After all these tests we adjusted the design of our existing stacks for field test systems. The test of larger stacks (4 kW) with this new design and operation conditions is still running in our field test systems inhouse4000.

fig.4 inhouse4000 system at the botanical garden of the city Chemnitz

Based on these new stacks and our experiences with our field tests systems we also had new conditions for the peripheral devices which required a modification or adjustment of the devices and the process. The air supply system has been reduced about 50% in size and power consumption. The humidification system was radically modified and does not need any auxiliary power. The new air supply system and the humidification system have been running since 10.000 hours in one of our field test systems. The reformat conditioning follows a complete new concept. It is a self regulating system without auxiliary power consumption. The fuel cell operating has been stabilized and its sensitivity against outside influences has been strongly reduced.
The reformer of the inhouse4000 system was revised last year, concerning efficiency, achievable gas quality and dimensions. Size and thermal mass of a reformer can be reduced by improved catalysts. Hence first of all a catalysts screening was carried out to take into account the advances of catalyst manufacturers.

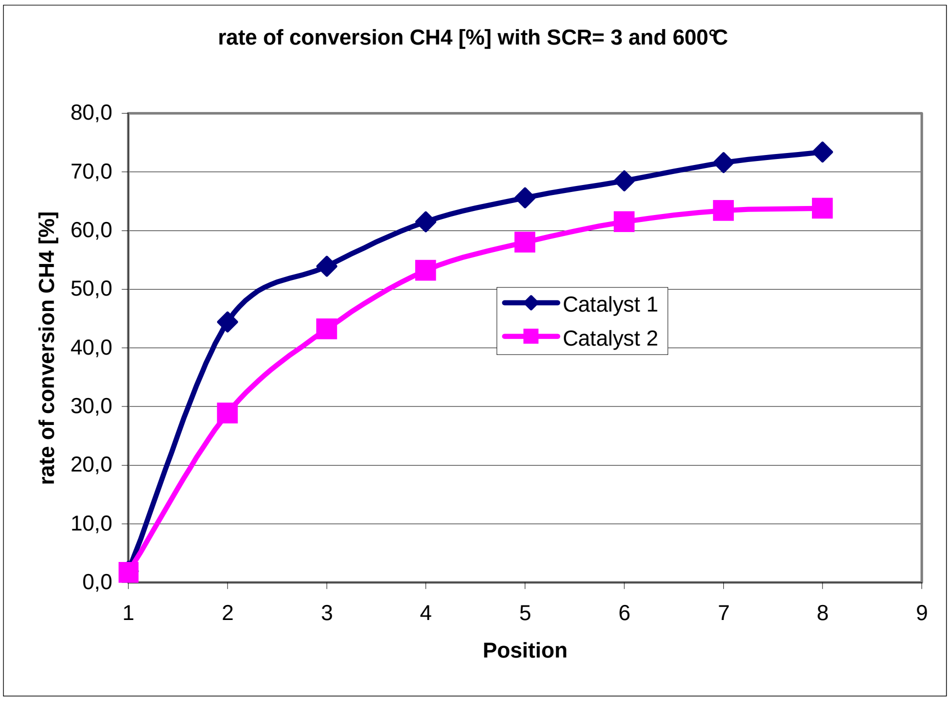


fig.5 results of catalyst screening/ CH_4 [4]

For the steam reforming process two catalysts could be identified, which enabled us to reduce the reforming temperature by 50 K. Furthermore catalyst mass and heat exchange surface could be reduced by 20%. For the new inhouse system this leads to a reformer with a performance enhanced by 25% and a simultaneous reduction in size by 20%. A similar result could be achieved within the gas-cleaning stages. The CO concentration of the reformat gas after the reformer could be reduced by 20%. The achievable gas quality is shown in fig.6

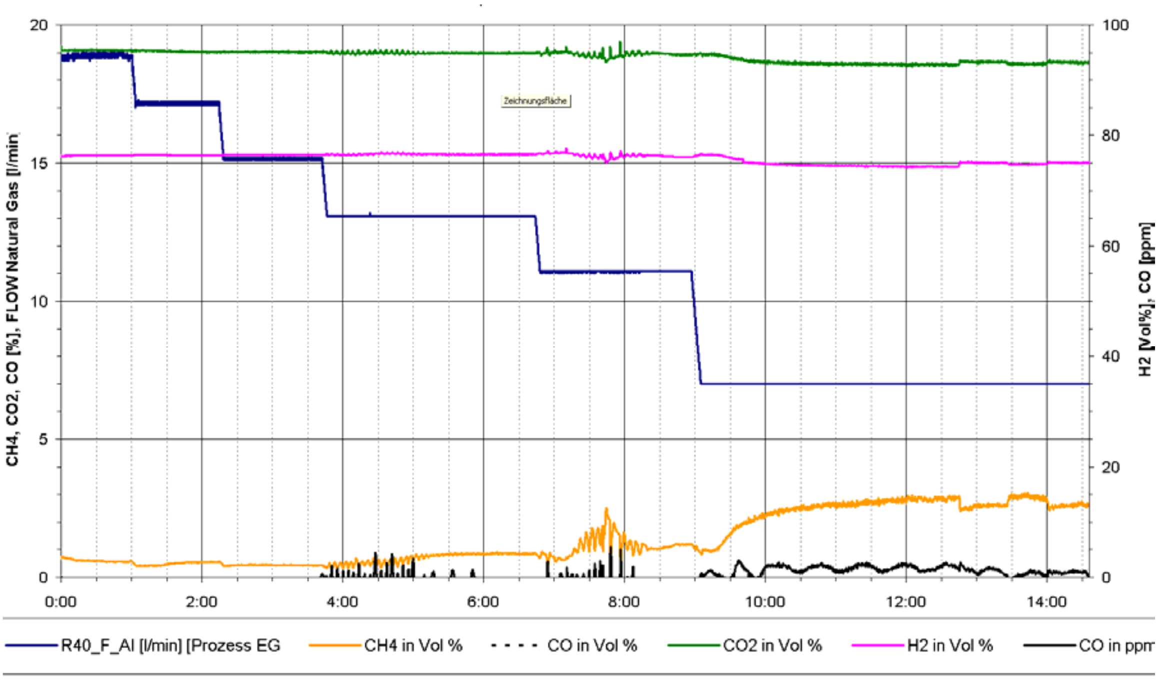
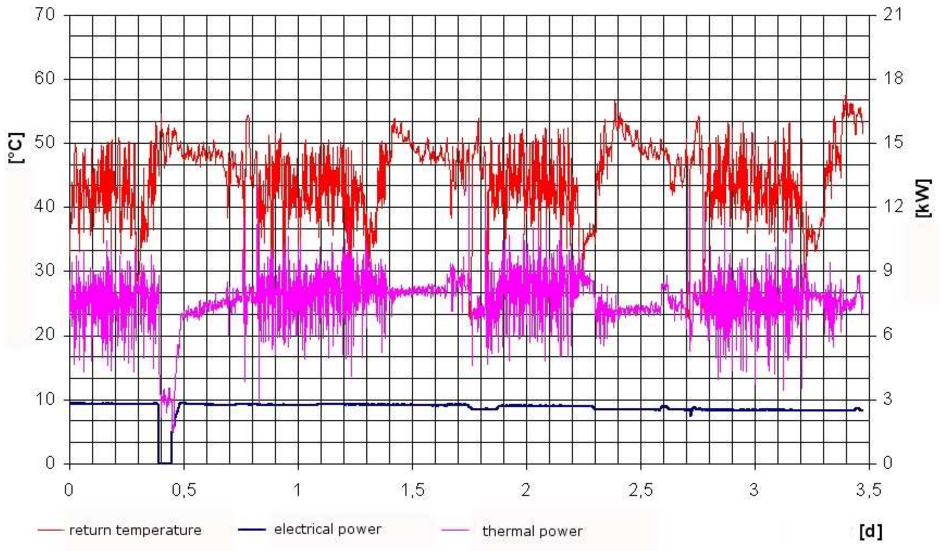


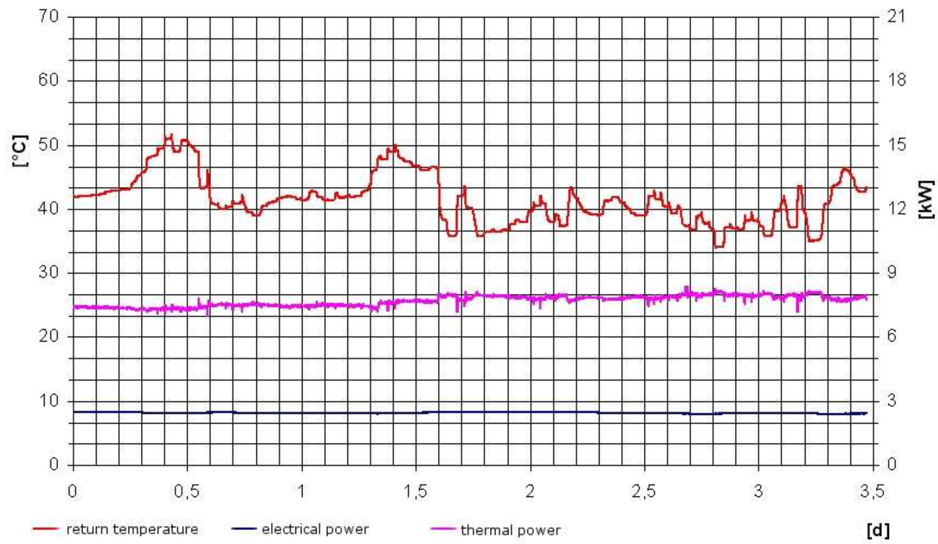
fig.6 gas quality steam reforming [4]

It can be shown that the gas quality and especially the CO concentration are stable during load changes. Another advantage could be gained by using catalytic active heat exchangers. An important aspect to improve the energy efficiency of the entire system was to reduce the heat losses to a minimum. Compared to the new, the old design showed a number of heat bridges, which were eliminated completely. Furthermore a new isolation, which features a heat transfer coefficient that is only 25% of the original value. Therefore the heat losses of the reformer and temperature inside the housing of the system could be reduced essentially. For example the temperature of the external wall of the reformer could be reduced from 90°C to 45°C. Durability and stability of the reformer through 10.000 h of operation under difficult conditions were already shown by the old design. Because of the modified catalyst the new reformer can also reform biogas. A first prototype is already built and is tested extensively at the moment. The first test shows that the reformer is applicable with biogas with CO_2 content from 25% to 55%.

A network of field test systems (inhouse4000 and older types) installed in our labs and at cooperation partners secures a lot of experiences and measure data. Due to the fact that these systems are operated under real application conditions we could eliminate a lot of construction failures and we learnt a lot about failures occurring under real applications. It is really necessary to adjust/ modify the existing conventional system. Mostly it is not possible to completely replace the existing system. It is economical useful to secure the minimum and middle load of the object. Both systems, fuel cell and normal heating system have to work together and should not influence each other. Fig. 7 shows some problems with an inhouse4000 system integrated in a conventional heating circuit. The heat production of the inhouse4000 system is very small comparing to the existing condensing boiler (2x250 kW).



a)



b)

fig. 7 problems heating circuit in inhouse4000 system [3]

Due to switching on and off of the condensing boilers, we had great temperature variations (fig. 7 a), which influenced several systems in the inhouse4000 system. The operation of the fuel cell was badly influenced. A small water storage tank helped to solve this problem (fig. 7 b). The temperatures did not change rapidly anymore.

Taking all experiences from field tests and our new developments we created a new generation of the inhouse system – the inhouse5000. This system is suitable for the energy production in

- multi family residences
- hotels
- wellness centres
- public buildings
- small and medium sized enterprises

The inhouse5000 system produces electrical and thermal power to deliver the minimum and medium load of the mentioned objects. The system is designed for a grid connected operation. It has a modular design and can be adjusted to the customer needs. Ongoing tests with biogas extend the range of applications. Cluster of inhouse systems can be installed on farms to use biogas and produce electrical power and heat for the farm.

RESULTS

With the new system inhouse5000 we solved a lot of problems. The reformer size was reduced by 20 % by increasing the reformer performance by 25 %. The energy consumption (natural gas) of the system was reduced by optimising the isolation of the reformer and reducing the anode stoichiometry. New catalyst screening helped us to reduce the carbon monoxide. The carbon monoxide part is less than 10 ppm. According to the new operating conditions of the stack and the reformer we optimised and adjusted our peripheral devices. The electrical energy consumption for peripheral devices was reduced by nearly 50 %. New materials and the optimised construction of reformer and fuel cell stack led to a significant increasing of the lifetime of stack, reformer and system. As already mentioned one inhouse4000 system is running for 10.000 hours and the test is going on. Surely we can not operate a fuel cell for such a long time. But currently we can operate a stack 4.000 – 5.000 hours in our inhouse4000 system. Saving of primary energy (natural gas) resulted in a reduction of carbon dioxide production. Comparing to conventional systems we produce with our fuel cell system approximately 75% of carbon dioxide. If we are using biogas we have a closed carbon dioxide cycle.

INTERPRETATION OF RESULTS

Sufficient innovation potentials have been revealed with the mentioned methods. Through intensive R&D it is possible to develop competitive products within the next 3 – 4 years. Inhouse5000 is the next step to a competitive product for the market. Already today we can demonstrate the potential of fuel cell systems.

basic principle	
fuel generation	natural gas steam reforming
fuel cell type	PEM fuel cell
nominal power	
electric	1 - 5 kW
thermal	2 - 10 kW
natural gas consumption	0,5 – 1,5 m³/h
temperature of heating circuit	50/ 70°C
cold start time	1 h
load change (30-100%)	15 min
current efficiency	25 – 30 %
total efficiency	60 – 90 %
dimensions(WxHxD)	700x1400x1000 mm³

fig. 8 inhouse5000 system

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Micro-CHP: Overview of Technologies, Products and Field Test Results

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SUMMARY

This paper gives an overview of selected Micro CHP technologies and products with the focus on Stirling and steam machines. First results of field tests in Germany, the United Kingdom and some other EC countries are presented and commented in the paper. The first field test results available to the date show that along with the overall positive performance there are some differences in sector performance (domestic vs small business). Some negative experiences have been gained, especially regarding the results of field tests with the Stirling engines and the free-piston steam machines and there are still obstacles for market implementation. Further projects and field tests with several kinds of micro-CHP are starting in different countries. If the positive experiences will prevail and deficiencies of the technologies or the operation can be eliminated, a comprehensive serial production and full market implementation can be started in the near future

INTRODUCTION

Decentralized power generation combined with heat supply is an important technology for improving energy efficiency, security of energy supply and reduction of CO_2 emissions. Micro-CHPs are especially interesting for such market sectors as single family houses, smaller multifamily houses and business enterprises due to their technical and performance features:

- A high overall energy conversion efficiency (e.g. in excess of 90 % for Stirling engines)
- Low maintenance requirements equivalent to a domestic gas boiler.
- Very low noise and vibration levels for installation in the home.
- Very low emissions of NOx, COx, SOx and particulates.

According to latest predictions, early adoption of domestic combined heat and power (CHP) by utilities and manufacturers could easily lead to sales and service contracts worth over £1.5bn per annum (aprox. €2.2bn) across Europe by 2010.

The objective of the study is an overview of selected micro-CHP technologies and products with the focus on Stirling and steam machines, an estimation of market potential and an overview and short assessment of first results of field tests in Germany, the United Kingdom and some other EC countries.

METHODS

The paper benefits in particular from the German experience in Stirling machines and field tests of a special Stirling engine (power range 2 – 9.5 kWel) of a German manufacturer, field tests of a Stirling engine (about 1 kWel) of a manufacturer from New Zealand in France and the Netherlands and UK field tests of a number of CHPs. Market studies [1] and the description of experiences with field tests in Germany [2] and UK [3] were evaluated to get comprehensive information for this study.

The market study focused on producers and conversion techniques with a high development status. These products are close to market introduction and either undergo CE-certification procedures (currently) or are already mature products. Background of this classification is the precondition of a timely entrance to the market with the analysed products.

Another determination of conversion techniques was made out of this classification: Stirling engine, hot air engine and combustion engine. The present study excluded fuel cells and micro-gas-turbines because of their low development status and their unfavourable and inefficient application in the small-scale housing and the trading sector.

RESULTS

Applications of micro-CHPs in Germany, the UK, France and the Netherlands have shown different results.

German market study

In the framework of the market study [1], 15 relevant products concerning micro-cogeneration were identified and analyzed. For further research on feasibility and application of micro-cogeneration the following objects were chosen:

1. Lion Powerblock (free-piston steam engine)
2. WhisperGen MicroCHP (Stirling engine),
3. Microgen M-CHP (Stirling engine)
4. Ecowill (combustion engine),
5. Dachs (combustion engine),
6. Ecopower Mini-CHP (combustion engine)

The profitability of the chosen products was analysed considering a cost benefit calculation of the product application in single consumption states and different buildings in Germany.

The results of evaluation of the products application were in most of the examples negative, a larger number of the case studies were analysed as economically inefficient. The energy costs of the different products varied in a range from 38.0 to 9.8 ct/kWh.

In the case of the Stirling-engines WhisperGen micro-CHP and Microgen micro-CHP savings of 10 % of the energy costs can be generated by using it in existing one-family-houses. Because of the less heat consumption (50 %) in new build one-family-houses the feasibility of micro-cogeneration is much lower than in older buildings. Concerning new-build one-family-houses the Stirling is therefore at its feasibility limit. The free-piston-steam-engine (e.g. Lion Powerblock) is the least cost efficient with an energy price of 0.38 €/kWh, which derogates the yearly energy costs.

As the different examples indicate a general statement on the feasibility cannot be given. It can be noticed that the feasibility is dependent on the operational mode and engine application according to the demand structure. Single task oriented products have a better feasibility, as modules designed with an additional “back-up” like a boiler or a gas-fired thermae.

German field tests with Stirling engines

A manufacturing company in Sindelfingen has already started its series-production of a gas-fired Sterling engine (Stirling 161) in 2004. In the meantime over 120 production-models of the Stirling 161 were sold.

Berlin

One of the Stirling 161 engines was made available for a test-run by the Berlin gas utility. In a state public institution building, the Kreuzberg fire station in the district Friedrichshain-Kreuzberg of Berlin, it is used to power the first Stirling CHP unit of this utility.

In association with a large owner of gas distribution systems (VNG) the company is going to check road capability and client acceptance in a three years test-run (since November 2005). According to the utility company, the first engines from the series-production run well – however not without smaller problems.

The VNG evaluated that the electrical power is underachieved, as well as a frequent fault liability which leads to higher costs in maintenance. Nevertheless the VNG gives a positive feed-back, as the Stirling is a very energy- and resource efficient way to convert energy. The future potential of the Stirling lies also in the advanced R&E, in comparison to fuel cells.

In the case of the fire station in Berlin Kreuzberg, the Stirling 161 (2 to 9 kWel; 8 to 26 kWth) was integrated in the existing building-service-engineering. The Stirling engine is parallel spliced with the central heating boiler, in such a way as to take the boiler offline in case of a non heat-requirement. The whole Stirling cogeneration unit can be taken offline with the help of an automatic balanced disc stopped valve. Because of the constant demand of energy and power (heat) this fire station is perfectly suitable for the test run.

According to performance data, the Stirling 161 can be operated full-load about 5,800 hours and part-load about 7,800 hours. The feed temperature (68 °C) in the heating circulation can be judged as less efficient which derogates the electric power. According to the producer a maximum feed temperature of 65 °C should not be exceeded.

Only after the transition to series-production in 2004 several problems with material occurred. After the appraisal of the company Solo this could be the reason for the defect of the connection rod of the aggregate at the Kreuzberg fire station. In the meantime a new method was developed for precise measurement and documentation of tolerances. There was also a change in the subcontractor/deliverance so that the piston rod is available with the requested quality and will be refitted.

Fürth

Another test run of Solo series-production Stirling 161 engines has been proceeding in Fürth. As in Berlin the test run is integrated in a measurement engineering process. In cooperation with the energy agency EAM and a district heating company in Fürth a district heating station was chosen, which is supplying heating power for a residential area (79 buildings).

Because of the combination of the district-heating-station (max. 4.5 MWh) and the Stirling-cogeneration-unit, different operations and charge states can be measured in various test-series, without having an impact on the supply of the client, as explains the EAM. Solo Stirling is said to achieve a degree of efficiency of over 90 % and its optimum in part-load.

Because of the high feed temperature from the district heating station the pre-conditions can be considered as suboptimal. Nevertheless the forecasted degree of efficiency could have been complied since the beginning of the test run. EAMs appraisal is positive saying that fundamental problems do not exist but the testing of a larger number of Stirling engines is regarded to be useful.

Otherwise an EAM project manager has some doubts regarding Stirling 161, questioning if the malfunctions like the defect of the connection rod were system immanent or only an outlier. However Solo maintains that the major part of the 120 Stirling hot-air-engines runs without any problems.

Ditzingen-Hirschlanden

An ESCO (part of a large French utility company) reports only sufficient results using the Stirling 161. In the Swabian town of Hirschlanden a Stirling CHP unit integrated in a district-heating-station is supplying 100 one-family-houses, a home for old people and a nursing home with energy (heat). The Stirling-cogeneration-unit is predominantly used to satisfy the demand of the boiler house.

The correlation between the high feed temperature and the derogation of electric efficiency has occurred also in this test-run. At a feed temperature of 60°C an electric power of 6.5 kWel is measured in Ditzingen-Hirschlanden. The absolute degree of efficiency amounts up to 79%. Because of the external combustion the interior of the Stirling engine remains free of residues. The pollutant emissions are much lower than the emission of other engine-concepts.

The ESCO specified the following disadvantages: the little operation experiences, the lack of a mass production and the still relatively high specific investment costs. The technical potentials are still not taped fully. The service and support could be optimized. But if the positive experiences will proceed, further projects with the Stirling unit will start.

Others

In June 2006, the Berlin gas utility started the second two year test-run in a one-family-house, where a gas-operated Stirling-cogeneration-unit WhisperGen micro-CHP of a New Zealand manufacturer (1 kWel; 7.5 kWth) was installed.

The same cogeneration-unit was chosen by another German utility company in Mannheim, in October 2006 over 20 households started with voluntarily participation in a test-run.

UK market study

The UK market for micro-CHP is assessed to be about 400,000 households [3] within ca 1 million households and SMEs which displace boilers per annum [5].

Unlike mini-CHP (which is regarded as a mature technology and have been applied for quite a long time), micro-CHP technology is not yet available commercially and there are rather few data about its performance in the UK.

Mini systems used in industrial and commercial applications show advantages of their performance for quite a long time. On the other hand, the direct projection of mini-CHP experience to micro-CHP level turns counterproductive due to extra complexity at the micro level and lesser tried technologies (for example fuel cell technology) which still needs to get mature.

Overall CHP are intrinsically more efficient in comparison with other generating facilities and the grid with its 30-40% efficiency [6]. Potentially micro-CHP can have the competitive edge over condensing boilers, but still considerable work has to be done to implement this into real economic and environmental benefits.

The latest research shows that micro-CHP have to improve their performance to become widely accepted in the market. Since the potential market is quite large in terms of units, there are good incentives for improvements.

By 2006, not much progress was made in micro-CHP penetration into the market. There are still very few micro-systems available, the public, architects and civil engineering community en mass are not much familiar with the technology.

Another factor of a rather poor micro-CHP market penetration was a successful campaign to promote energy-saving condensing boilers.

This said, the British government announced in 2001 that CHP was one of important ways to implement the Kyoto commitments. The UK target for installations set for the end of the year 2000 was 5,000 MWe. For year 2010 the target is rather vague, further of 12,000 to 19,000 MWe [5].

UK Carbon Trust field tests

Carbon Trust launched a trial in 2003 aimed to carry out a comprehensive analysis of the current situation and development of recommendations to overcome the barriers that impede the implementation of this energy saving technology. The trial is expected to finish by mid or late 2007 when full results will have been published. Until now only preliminary data are available.

The technologies included in the assessment were Stirling Engines, Organic Rankin Cycle Machines, Fuel Cells and Internal Combustion (IC) Engines. In total about 40 units (of those 31 micro-CHP, mostly in homes) have been assessed.

The performance indicators recorded were: overall thermodynamic efficiency, the amount of electricity generated the carbon intensity of the electricity displaced from the grid.

One very prominent result of this trial is that the first two performance indicators differ very significantly for micro- and small CHPs.

The micro-CHP units assessed show very different performance characteristics in different environments. The carbon footprint and savings being important characteristics, the capital costs needed to be evaluated as well. However, at this stage, the units are not manufactured at the scale sufficient enough to a give a final conclusion on the units capital costs.

The preliminary results of the Carbon Trust trial indicate that the performance of micro-CHPs was not that impressive as expected. Some of the reasons are:

- The units were actual efficiencies is lower than assumed by existing models.
- The amount of electricity generated is much lower than the forecast.

The reasons for the poorer than expected efficiency appear to be related to the design and operation of the units at their current stage of development. The thermal inertia of the micro-CHP units seems to be still too high compared with the conventional boilers when the units are exposed to real-life intermittent demand for heat in buildings.

In many houses the demand is to a large degree intermittent and the heat is required only for short periods but many times a day. In such circumstances the repeated warm-ups cause waste of energy which is not re-released in a useful way, hence reduced efficiency. This conclusion opens up a vast scope for improvements focused on reduction of thermal inertia and/or number of cycles.

Another factor contributing to over-estimation of micro-CHP performance is the assumption of constant average electricity demand. However the trial has proved that for the most of the time demand is much lower than average and also lower than typical micro-CHP output.

This is also combined with short periods of very high demand (when some appliances are running, e.g. kettles, electric showers, etc). So far significance of peaks and troughs has been neglected by the designers. Along with new design of low voltage networks which could cope with high levels of export, in addition to full load import when the units are not running, the micro-CHP industry needs to design units with the ability to modulate electrical output much more widely than currently.

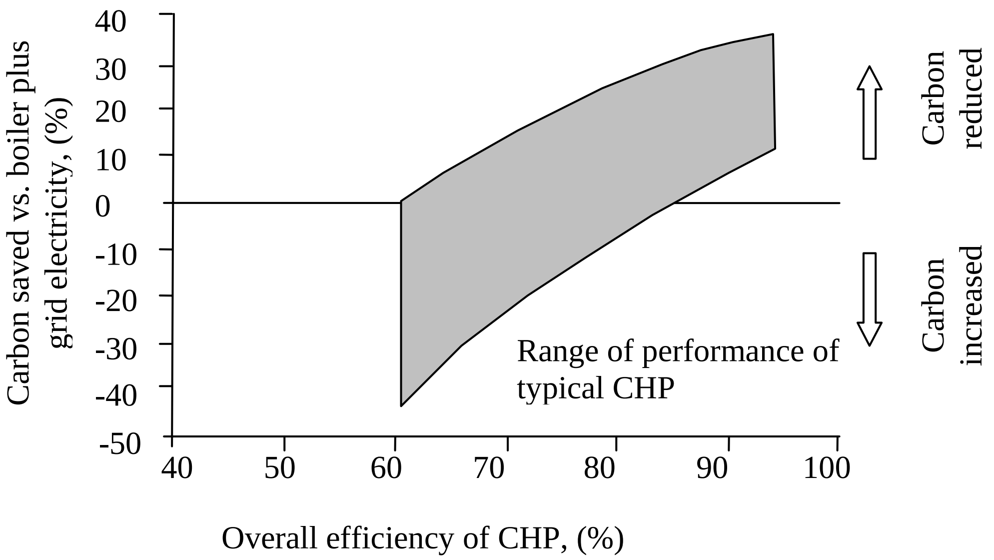

Figure 1. Performance envelope of real CHP [3]

Summarising the preliminary test evaluation of micro-CHP performance (Fig.1), the Carbon Trust [3] indicates that at the current stage of technology, these units will have limited contribution to CO_2 reductions and considerable improvements in technology are required.

Field tests with Stirling engines in France and the Netherlands

Funded by DG TREN the project DEO [4] was launched to compile a comparative study on innovative technologies of power generation in buildings, analysing pilot projects in the years 2001-2003. Partners of the DEO project were several European gas supply companies. The study was focused on the application of innovative gas technologies such as micro-CHP, gas heat pumps, gas fired household appliances and combination systems e.g. with solar heat.

One-family houses or apartments were taken as demonstration objects in the DEO-project. In those houses/apartments the operation was measured and controlled during the period of a year. For the later comparison and calculation of the energy saving potentials, other objects without the installation of the innovative technology were taken into consideration. In the frame of this study micro-CHP units were installed by leading gas supply companies in France and in the Netherlands.

A Whisper-Gen module powered a test run apartment in France and a one-family-house in the Netherlands. In the Dutch test run, the micro-cogeneration unit was supported by actions and applications of other innovative (energy saving) technology, as gas-heat-pumps, gas fired household appliance and electronic thermostatic-valves etc.

The French demonstration-run of micro-cogeneration achieved good, satisfying results. During the test-run there appeared no operational malfunctions. The specific performance of the heat supply was achieved as announced by the producer. The electricity performance was close to the predictions.

A degree 82 % of the efficiency of the installation unit (75% thermal and 7 % electric) was achieved during operation. During the monitored and controlled test-run 55 % of the power demand was satisfied by the micro-cogeneration generation. The average annual electrical power was 0.2 kW. At the test-run-object savings of 13 % of primary energy and 7 % of end use energy were achieved in comparison to a reference building.

The performance of the micro-cogeneration unit of the Dutch test run example was in contrast disappointing. Several problems in the line-operation as well as problems with the start/stop mode operation and noise emissions were observed. During the test-run period the total operation time was limited.

Overall only 278 kWh were generated by the micro-CHP unit which equals only 6 % of the energy demand. Only because of the combination of actions and applications, as per statement above, savings of 13 % primary energy and 12 % end use energy were achieved.

DISCUSION

For certain consumers, micro-cogeneration is already a comprehensive solution for energy supply, according to the present market study [1]. For broad market introduction further research and technology innovation is still necessary. It includes stimulation on the part of energy supply companies as well. As shown for Berlin test study, the potential of demand side (potential customer) for a market introduction of micro-cogeneration units exists and the customers attitude is in principle positive. Nevertheless the further establishment in terms of a market introduction of micro-CHP has to be supported actively e.g. with the help of energy supply companies research institutes, producers, networks and producers cooperation. This need of further stimulation is one of the results of field-tests [2] and [3].

The authors of the paper came to the following conclusions.

A comprehensive serial production and full market implementation can only start in the near future if more comprehensive and long time tests proceed and more positive experiences of micro-CHP are reached. The deficiencies in the technologies and operation found have to be overwhelmingly eliminated before wide implementation could be approached.

ACKNOWLEDGEMENT

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Towards sustainable energy systems – role and achievements of heat integration

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SUMMARY

This paper presents an established sustainable and integrated design methodology for the efficient heating and cooling of individual buildings and complexes. The methodology includes the design basis for combined heat and power systems, refrigeration, air conditioning and heating with pump systems. It is equally applicable for single family houses as well as large building complexes and meets a major challenge in the design of heating and cooling systems, namely, the complexity of energy and power integration. The efficient use of available heating and cooling resources for serving buildings of various sizes and designations can significantly reduce energy consumption and emissions. Although heat integration (or Pinch Technology) has been around for several decades and has been very successfully applied in large industrial applications, it has only recently been applied for improving the energy efficiency of buildings and building complexes. There is significant scope for application, as energy prices rise and energy related emissions are required to be reduced. This methodology can also be used to integrate renewable energy sources such as biomass, solar PV and solar heating into the combined heating and cooling cycles. Case studies are used to illustrate the methodology. The paper makes suggestions how the methodology can be applied for energy and emissions reduction in single buildings and complexes.

INTRODUCTION

Since 1995 the energy consumption of the EC member countries rose by 11 %, to the value of 1637Mt of oil equivalent [1]. The population of the EC member states however is growing at a much slower rate, approximately at 0.4 %/y [1]. In the UK domestic energy consumption has risen from 35.6 Mt of oil equivalent to 48.5Mt in the period 1971 to 2001, an increase of 36%, despite energy efficiency increases [2]. The increase in energy consumption per household was less, at 5 %, over the same period, better reflecting the increase in energy efficiency in individual dwellings. However in the UK it is proposed that the number of houses built over the next 10 years should increase by 200,000/y, to account for the smaller number of people per household, and the demand for housing [3]. This will have serious implications for energy consumption and greenhouse gas emissions. To date the increase in energy consumption from individuals [2], dwellings, and buildings generally has been met by the burning of carbon based fuels, or in some cases, by the increase in nuclear power. Although renewable energy has seen a large increase, the contribution towards total domestic energy use is still small (in the UK 1.0 Mt of oil equivalent compared to a total use of 47.0 Mt of oil equivalent).

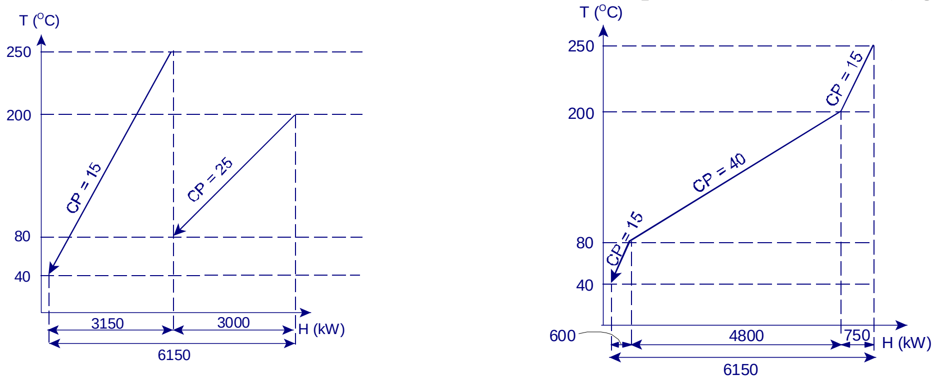
SUSTAINABLE ENERGY AND HEAT INTEGRATION

Heat Integration (or Pinch Technology/Process Integration) is an energy saving methodology that has been extensively used in the processing and power generating industry over the last 30 years. This method examines the potential of exchanging heat between heat sources and heat sinks via the use of heat exchangers and reducing the amount of external heating and cooling required. A systematic design procedure has been developed to provide the final energy reduction design of the system. The method has further been developed to specify the source of heating and cooling required (e.g. steam, hot water, cooling water) and also the potential of power production in the form of shaftwork or electrical production. More details are available elsewhere [4-9]. This methodology is based on the analysis and understanding of heat exchange between process streams through the use of a temperature-enthalpy diagram. The specific steps for drawing the curves in this diagram are presented in Figs. 1 - 3. The methodology first identifies sources of heat (termed hot streams) and sinks of heat (termed cold streams) in the process flowsheet. Tab. 1 presents a simple example.

Table 1 Sources (Hot) and Sink (Cold) streams

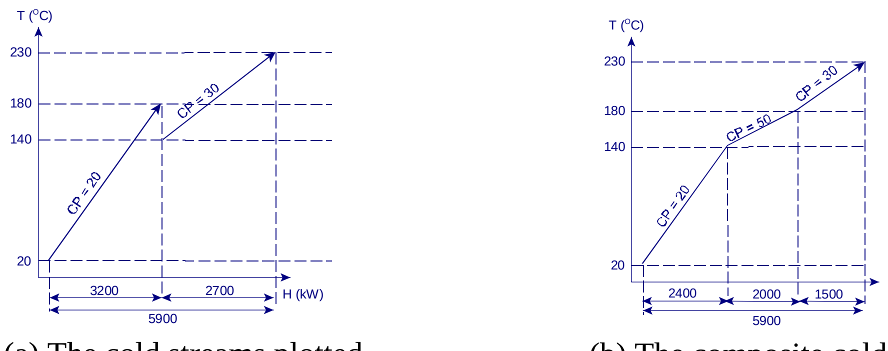
Stream	**Type**	**Supply Temp.** T_S (°C)	**Target Temp.** T_T (°C)	**ΔH** (kW)	**Heat Capacity Flowrate CP** (kW/°C)
Fresh Water	Cold	20	180	3200	20
Hot Product 1	Hot	250	40	-3150	15
Juice Circulation	Cold	140	230	2700	30
Hot Product 2	Hot	200	80	-3000	25

Sources of heat can be combined to construct the composite hot stream (Fig.1) and sinks of heat can likewise be combined to construct the composite cold stream (Fig. 2).



(a) The hot streams plotted (b) The composite hot stream

Figure 1. Composing hot streams to create a hot composite curve



(a) The cold streams plotted (b) The composite cold stream

Figure 2. Composing cold streams to create a cold composite curve

The relative location of these curves on the temperature-enthalpy diagram is dependent on the allowable temperature difference for heat exchange. The next step is to select a minimum permissible temperature approach between the hot and cold streams, ΔT_{min}. The selection of the optimum ΔT_{min} is a result of an economical assessment and trade-off between the capital and operating costs (mainly for energy usage) of the process. A large ΔT_{min} implies higher energy use and costs and lower capital costs. For increasing energy cost the optimum ΔT_{min} is reduced, meaning the heat exchanger system is allowed to recover more energy, but at the expense of more capital to pay for the greater heat transfer area. This issue has been discussed in greater detail elsewhere – [10], [8], [11]. In this example a ΔT_{min} of 10 °C was selected for simplicity. Plotting the composite curves (CC) in the same graphical space (Fig.3) allows values to be derived for maximum heat recovery and minimum hot and cold utilities. These are known as targets. In this particular case of ΔT_{min} = 10 °C, the minimum hot utility requirement is 750 kW and minimum cold utility requirement is 1,000 kW.

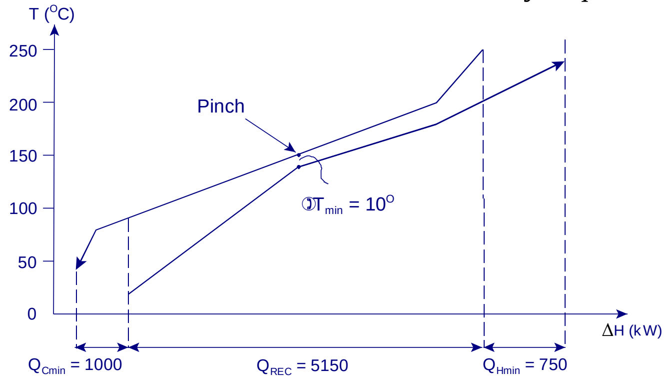


Figure 3. Plotting the hot and cold Composite Curves

In Fig.3 we can also determine the position of the Pinch. The Pinch represents the position where the hot composite and cold composite curves are at their closest (for a $\Delta T_{min} > 0$). The Pinch has provided the name for the Heat integration / Pinch Technology and has important features which make a substantial contribution to the design of maximum energy recovery systems and the design of economically efficient heat exchanger networks.

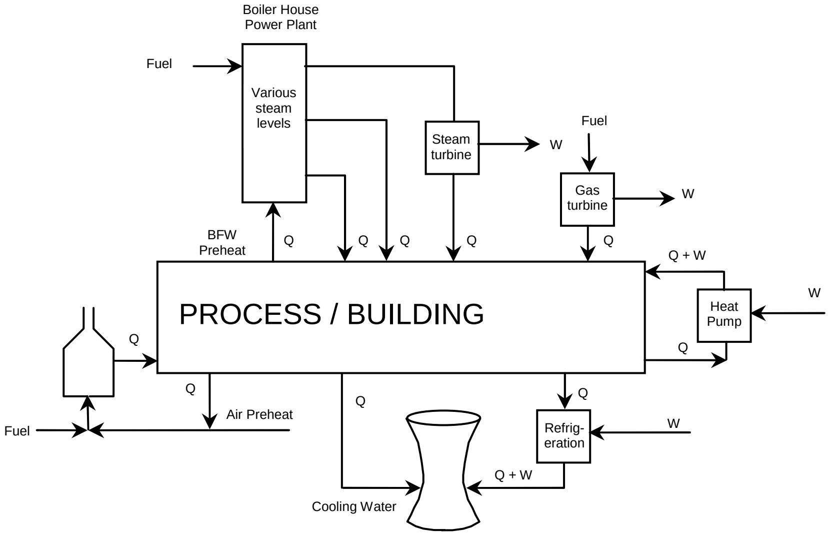


Figure 4. Potential for the choice of hot and cold utilities

Various design methods have been developed which allow these targets to be achieved in practice for both grass roots designs [4], and more importantly, for the retrofit of existing plants [12-14]. These methodologies are supported by process integration software which provides both design and retrofit support and automated design [15, 16]. In most cases there are more than one hot and one cold utility potentially providing heating and cooling requirements after energy recovery. In these situations we have to find and evaluate the cheapest and most desirable combination of the potential utilities that are available (Fig.4). To assist with this choice and to further enhance the information derived from the hot and cold CC, an additional graphical construction has been developed. This is known as the Grand Composite Curve (CC) and provides clear guidelines for the optimum placement and scaling of hot and cold utilities. The GCC together with the Balanced Composite Curves (the CC with the utilities selected added) provides a convenient tool for the optimum placement and selection of hot and cold utilities. An example of selection of utilities and its placement is shown in Fig. 5.

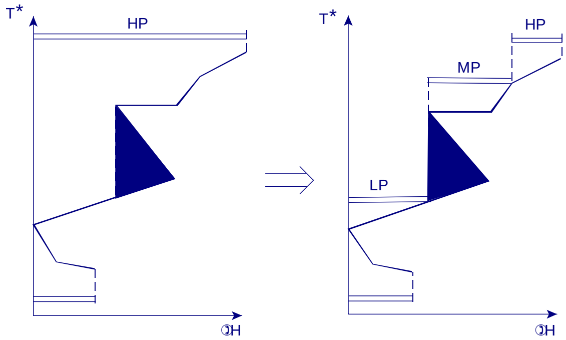


Figure 5. Placement of utilities with the help of GCC

The GCC is also a useful tool for targeting the cooling requirements in sub-ambient processes (such as air conditioning) which require some form of chilling or compression refrigeration. The GCC provides a target for the heat that has to be removed by a refrigeration process, and the temperature at which the refrigeration is needed. However, the overall process/utility system can be improved by using the heat rejected by the refrigeration system to provide low level heating to the process above ambient, thereby saving heat supplied by another utility source (such as hot water). The more detailed description has been provided elsewhere [17].

Traditional Heat Integration assesses the minimum practical energy needs for a process through a systematic design procedure involving five steps: (i) collection of plant data; (ii) setting targets for minimum practical energy requirements; (iii) examination of process changes that contribute to meeting the target; (iv) obtaining the minimum energy design that achieves the target and (v) optimisation which allows a trade-off between energy costs and capital costs.

EXAMPLES – CASE STUDIES

Hospitals are an example of building complexes which are important energy users, especially that of thermal energy. Although considerable research effort has been done to minimise the energy consumption in this sector, there are relatively few studies which apply Heat Integration (Pinch Technology) as a tool to evaluate potential energy savings and thereby reduce emissions.

Herrera et al [18] presented a study of a hospital complex which included an institute, a general hospital, a regional laundry centre, a sports centre, and some other public buildings. The use of diesel as a fuel represented 75% of its total energy consumption and 68% of its total energy cost which was 396,131 US$ in 1999. The hospital complex process diagram is shown in the Fig. 6. In the hospital complex, the heat demand is met by producing steam in boilers fuelled by a high price diesel fuel. There is no heat recovery between the existing heat sources and heat sinks. The hot streams were identified as the soiled soapy water from the laundry and the flow of condensed steam not recovered in the condensation network. The stream data are presented in Tab. 2.

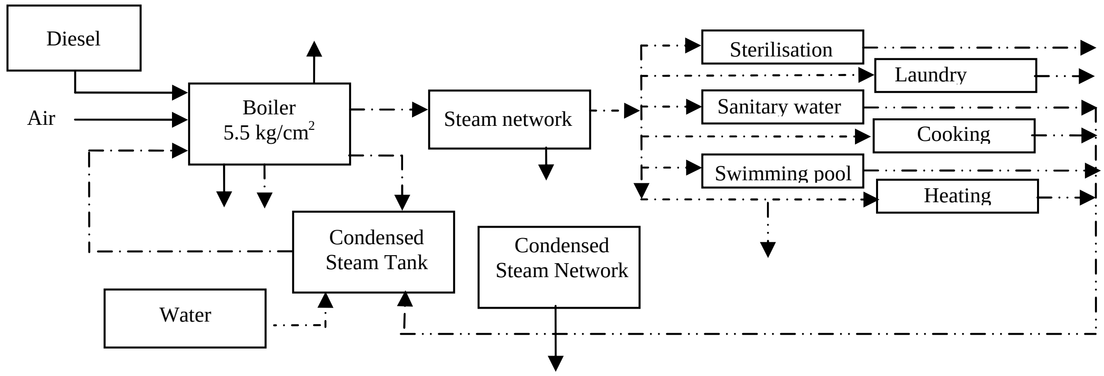


Figure 6. Hospital heat flow block diagram [15]

Table 2 presents the input data extracted from Fig 6.

Streams	ΔH [kW]	CP [kw/°C]	Tin [°C]	Tout [°C]
Hot Streams				
Soapy water (1)	23.7	0.53	85	40
Condensed steam (2)	96.32	2.41	80	40
Cold streams				
Laundry sanitary water (LS)	17.60	0.59	25	55
Laundry (L)	77.12	2.20	25	60
Boiler feed water (BF)	7.13	0.24	30	60
Sanitary water (SW)	77.12	2.20	25	60
Sterilisation (S)	12.50	0.14	30	121
Swimming pool water (SP)	151.67	50.56	25	28
Cooking (CO)	59.63	0.85	30	100
Heating (H)	100.82	14.40	18	25
Bedpan washers (B)	4.94	0.05	21	121

It needs to be noted that the CP represents mass flow m multiplied by the specific heat capacity Cp and in this case are considered constants. As can be seen from the CC (Fig. 7) the amount of external heating required, or the hot utility target, of this hospital complex is 388.64 kW. The GCC has been employed to determine the temperature levels of the utilities necessary to satisfy this requirement. This graphical method allows a more precise analysis to be undertaken in order to integrate the thermal utilities considered for the process heating and cooling requirements (Fig.8). This GCC indicates that the required heating of complex should be 388.64 kW. This means a yearly energy requirement of 12.26 TJ. The actual amount of heating provided is in fact 625.28 kW which represents the heat services that are currently transferred to the complex. This results in a potential energy savings of 38 % equivalent to saving 246,000 L of diesel/y (100,000 US$). To reduce energy demands in the form of heating to the value targeted by the analysis, the Heat Integration based analysis suggests that four extra heat exchangers should be added to the network. Two in the laundry to cover part of the heat demand, a third in the machinery rooms which help to heat boiler feed water, and the final one in the condensation tank area that heats the sanitary water. Several other issues could also have been considered to further refine the analysis, such as fouling, pressure drop and non constant heat demand.

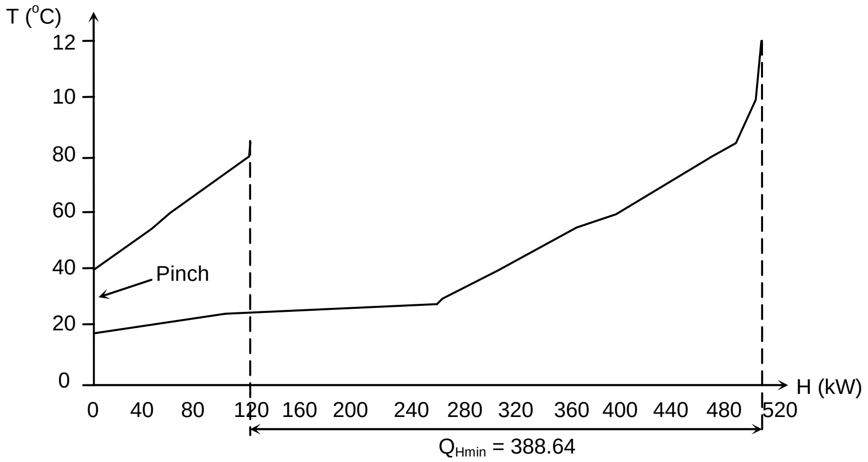


Figure7. CC for the hospital complex process streams

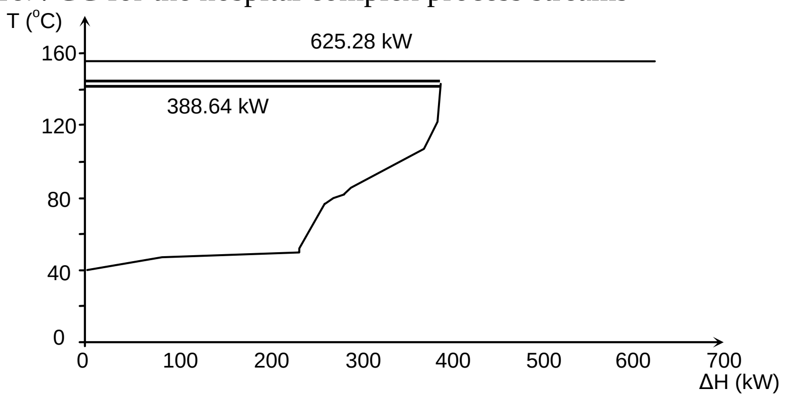


Figure 8. GCC of the hospital [15]

Kemp [9] presented a further more detailed case study of a hospital complex. The main energy demands were found to be space heating, air systems, general hot water supplies and other uses in the laundry and the incinerator. The main part of the complex was fully air conditioned. Central heating water was supplied at 80 °C, and returned at 71 °C and then reheated in a gas-fired calorifier. Tap hot water was also provided.

Table 3 Heat requirements and time distribution required during winter

Stream details and locations	No of streams	Hot streams total		Cold streams total		Times of operation
		Temp (°C)	Heat load (kW)	Temp (°C)	Heat load (kW)	
Main air supply	35	20-15	43	5-37	780	24 h: Day
	33	20-15	27	2-37	681	24 h: Night
Space heating	2			71-80	709	24 h: Day
	2			71-80	665	24 h: Night
Domestic hot water	1			5-60	244	24 h: Day
	1			5-60	48.5	24 h: Night
Autocalves	1			162	13	24 h
Kitchen	1			5-30	101	0530-0200
Laundry	3	37-30	67	18-26, 162	797	0700-1630
Incinerator	1	600-217	650			0900-1630
Boilers	1	235-60				24 h
Summary						
Maximum load			760		2,644	0900-1630
Minimum load			27		1407	0200-0530

In some areas (e.g. kitchens) the heat use varied considerably between day and night time. The identified streams are shown in Table 3. Energy use in winter for day time conditions was found to be 2534 kW (all hot utility) whereas in summer this value dropped to 1030 kW (hot utility) and 207 kW (cold utility) as space heating was replaced by air conditioning and the use of some refrigeration. Kemp [9] presented the GCC for the peak demand during winter (7-9am period). Heat integration analysis of these different time periods provided various schemes for improvement. The first involved rescheduling. This took into consideration the possibility of changing the periods of high utility demand. Suggestions were (i) Moving a cold stream active in 7-9am period to a different time; (ii) Moving a hot stream not operating in 7-9am period to operate during this period. As it was not possible to shut down the central heating or air conditioning systems for this high demand 2 h period, an alternative was that laundry operations could be rescheduled to start 2 h later but finish at the usual time. This would result in the installation of new machines at a high capital cost. For the second suggestion is would be possible to start the incinerator at 7am and run it for a further 2 hours. Retaining the current waste disposal value would result in a reduced exhaust flow rate and heat recovery rate. The daytime total heat recovery would be the same, but one less boiler would be required. It was also found that additional municipal waste could be delivered and that the incinerator could be run at full power over 24 hours. The Heat Integration methodology also investigated the benefits of introducing CHP. The GCC shows that the hospital is represented by four different temperature ranges which were the laundry and autoclaves (160-170 ^{o}C), space heating (80-85 ^{o}C), domestic hot water (up to 70 ^{o}C), and air heating (10-40 ^{o}C). The heat from the jacket cooling water and exhaust gases could be recovered and be suitable for the space heating system. Standby diesel generators already installed to supply emergency power would be a cheaper option to convert than to provide heat and power from a new diesel generator. These would then provide about 50 % of the peak heat demand. Further options would be to install a micro gas turbine of around 1.5 MW. This would provide all power requirements and the majority of the peak demands. Maintenance for a single source heat and power supplier would leave the hospital complex in a vulnerable state.

Several other case studies have been reported by Linnhoff March Ltd. [19]:

a) A study with the code name BRECSU/BDP was an attempt to define the energy-efficient hospital of the future. LM Ltd looked at all aspects of hospital heating including space heating, ventilation air heating, hot water heating and sterilisation. For hot streams available for heat recovery LM looked at ventilation extraction air, drainage and incinerator flue gas. The major opportunities on an existing building were be in the utility area. There was much detail in the study, including heat generated by the patients in the day (active) and night (sleeping). The work demonstrated that, for a complex new building like a hospital, potential existed for heat recovery between extract and inlet air systems and for heat pumping from the drainage system. The most economic form of heat recovery was from the incinerator exhaust and they also concluded that the optimum utility system would be CHP.

b) Eastbourne Hospital. The largest saving found was in the laundry where dryer exhaust could heat water and fresh air for the dryers and hot dirty wash water could heat fresh water (with storage). A heat engine-based CHP scheme was recommended but this would affect the laundry recommendations because it would provide the necessary hot water.

CONCLUSIONS

Heat Integration (or Pinch Technology) has been used for 20 years in industry throughout the world to increase energy efficiency of any processing plants that have heating or cooling requirements, and also have needs for power to provide electricity or directly drive machinery. Energy savings of over 30% have been recorded, and the methodologies developed have been incorporated into the design offices of all major producing companies. The same methodologies and design rules can also be applied in buildings or their complexes. Examples of two complexes have been given. It is clear that as energy efficiency in buildings and the reduction of emissions becomes a standard issue in existing and new complexes, further examples of heat integration application will be carried out.

ACKNOWLEDGEMENT

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Low Exergy Systems for High-Performance Buildings and Communities

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SUMMARY

There is an obvious and indisputable need for an increase in the efficiency of energy utilisation in buildings. Heating, cooling and lighting appliances in buildings account for more than one third of the world's primary energy demand. In turn, building stock is a major contributor to energy-related environmental problems. There are great potentials, which can be obtained through a more efficient use of energy in buildings.

An optimisation of the exergy flows in buildings and the related supply structures, similar to other thermodynamic systems such as power stations, can help in identifying the potential of increased efficiency in energy utilisation. Through analyses, it can be shown that calculations based on the energy conservation and primary energy concept alone are inadequate for gaining a full understanding of all important aspects of energy utilisation processes. The high potential for a further increase in the efficiency of; for example, boilers, can not be quantified by energy analysis - the energy efficiency is close to 100%; however, this potential can be showed by using exergy analysis [1], the exergy efficiency of a common gas boiler is about 8%.

This paper outlines the international co-operative work in the general framework of the International Energy Agency (IEA), the ECBCS Annex 49 "Low Exergy Systems for High Performance Buildings and Communities" [2].

INTRODUCTION

As a consequence of the latest reports on climate change and the needed reduction in CO_2 emissions, huge efforts must be made in the future to conserve high quality, or primary energy, resources. A new dimension will be added to this problem if countries with fast growing economies continue to increase their consumption of fossil energy sources in the same manner as they do now. Even though there is still considerable energy saving potential in the building stock, the results of the recently finished IEA ECBCS Annex 37, Low Exergy Systems for Heating and Cooling of Buildings [3], show that there is an equal or greater potential in exergy management [4]. This implies working with the whole energy chain, taking into consideration the different quality levels involved, from generation to final use, in order to significantly reduce the fraction of primary or high-grade energy used and thereby minimise exergy consumption [5]. New advanced forms of technology have to be implemented. At the same time, as the use of high quality energy for heating and cooling is reduced, there is more reason to apply an integral approach, which includes all other processes where energy/exergy is used in buildings. In recent years, we have made substantial progress in the development of new and integrated techniques for improving energy use, such as heat pumps, co-generation, thermally activated building components, and methods for harvesting renewable energy directly from solar radiation, from the ground and various other waste heat sources [6].

The results obtained in research projects on optimised exergy use in buildings are promising and elucidate a huge potential for introducing new components, techniques and system solutions to create low exergy built environments. The exergy conversion, e.g. heat or electricity production, plays a crucial part in possible future activities in the overall system optimisation of the entire energy system within a building.

THE EXERGY CONCEPT AND THE LOWEX APPROACH

Exergy is a concept which helps us distinguish between two parts of an energy flow: exergy and anergy. Only the exergy part of any energy flow can be converted into some kind of high-grade energy such as mechanical work or electricity. Anergy, on the other hand, refers to the part of the energy flow which cannot be converted into high-grade energy, e.g. low-grade waste heat from a power plant. Exergy can be regarded as the valuable part of energy, while anergy designates the low-value portion.

Unless a suitable use for exergy is found, e.g. waste-heat utilisation in buildings, the low-value part of the original energy flow will eventually dissipate into the environment and be irreversibly lost. Such unalterable dissipation is designated as irreversibility. The exergy content of a given flow of energy depends on the attributes, e.g. the temperature, pressure, and chemical composition, of both the substance carrying the energy (energy carrier), and the surrounding environment. The more different the attributes of the energy carrier and the environment are, the higher the exergy content of the energy carrier is. For example, high-pressure steam required for electrical power generation has a higher exergy content than warm water needed by a dishwasher [7].

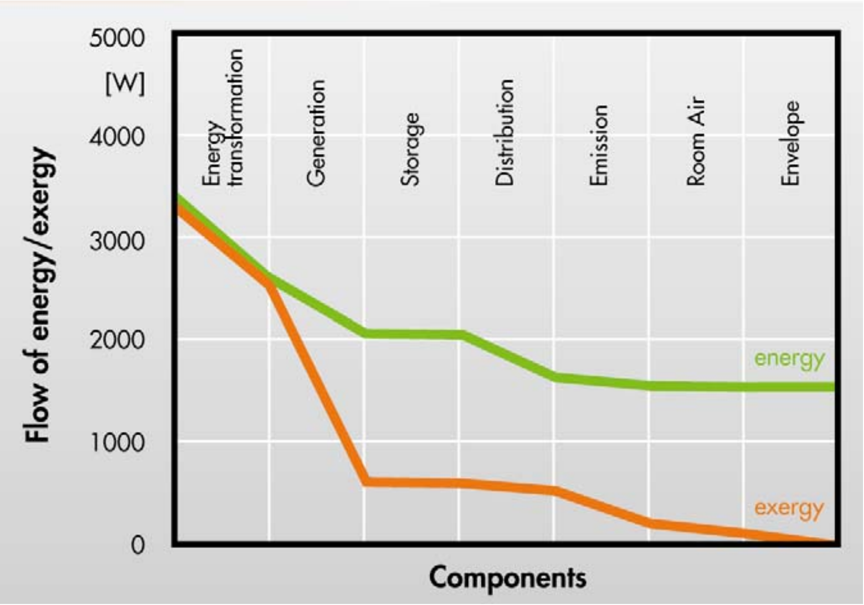

Figure 1. Results of an analysis of energy and exergy flows through a building

The Low Exergy (LowEx) approach entails matching the quality levels of exergy supply and demand, in order to streamline the utilisation of high-value energy resources and minimise the irreversible dissipation of low-value energy into the environment [5],[8]. This approach is the key concept for the work of Annex 49 on energy use and supply structures in the built environment.

SCOPE AND OBJECTIVES OF THE ECBCS ANNEX 49

The scope of this activity is to improve, on a community and building level, the design of energy use strategies which account for the different qualities of energy sources, from generation and distribution, to consumption within in the built environment. In particular, this method of exergy analysis has been found to provide the most correct and insightful assessment of the thermodynamic features of any process and offers a clear, quantitative indication of both the irreversibility and the degree of matching between the resources used and the end-use energy flows [9]. To satisfy the demands for the heating and cooling of buildings, the exergy content required is very low, since a room temperature level of about 20°C is very close to the ambient conditions. Nevertheless, high quality energy sources, like fossil fuels, are commonly used to satisfy these small demands for exergy [5]. From an economical point of view, exergy should mainly be used in industry to allow for the production of high quality products.

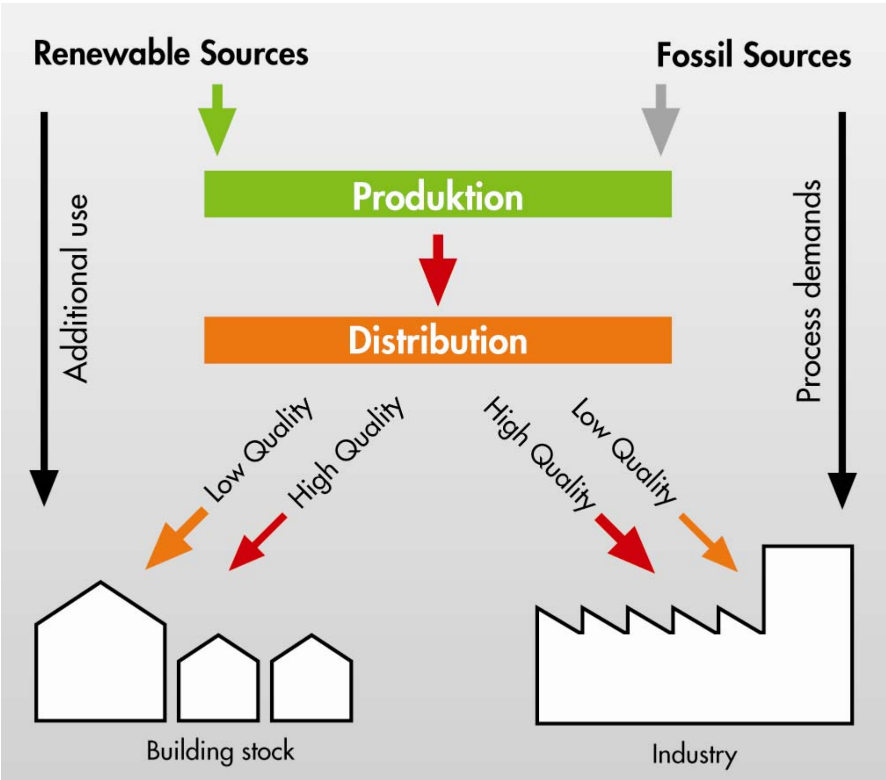

Figure 2. Desirable energy/exergy flow to the building stock and industry. In the building stock, there should be a larger share of low valued energy, whereas high quality energy should be left for other purposes, e.g. industrial processes.

It is known that the total energy use caused by buildings accounts for more than one third of the world's primary energy demand [10]. There is however substantial saving potential in the building stock. The implementation of exergy analyses paves the way for new possibilities of increasing the overall efficiency of the energy chain. Exergy analysis can support the development and selection of new types of technology and concepts with the potential of lowering exergy consumption for built environments and the related supplies. It can also quantify this potential. Up to now, considerable effort has been made to reduce the energy demand of the building stock and to increase the energy conversion factors in power stations. The new approach is not necessarily focussed on a further reduction of the energy flow through a building's envelope. When the demands for heating and cooling have already been minimised, the low-exergy approach aims at satisfying the remaining thermal energy demand using only low quality energy. This creates the potential for reducing the total amount of exergy needed by the energy supply-demand chain, and for providing a more customised distribution of exergy to consumers with different exergy requirements.

The main objective of the annex is to use exergy analysis as a basis for providing tools, guidelines, recommendations, best-practice examples and background material to designers and decision makers in building, energy production and political fields. Another important objective is to promote possible energy/exergy and cost-efficient measures for retrofit and new buildings, such as dwellings and commercial/public buildings, and their related performance analyses viewed from a community level, including the energy supplies.

The major benefit of following low exergy design principles is the resulting decrease in the exergy demand in the built environment. By following the exergy concept, the total CO_2 emissions for the building stock will be substantially reduced as a result of the use of more efficient energy conversion processes. This new concept supports structures for setting up sustainable and secure energy systems for future building stock.

The strategies developed for a better and exergy optimised building design, aimed at a future of clean, clever and competitive energy use, will help in pinpointing specific actions to reach this goal. Additionally, the exergy demand of buildings will be reduced due to new, enhanced heating and cooling systems.

IDENTIFIED RESEARCH ISSUES

The exergy concept applied to buildings and the related supply structures leads to new research topics for building stock. The ECBCS Annex 49 is addressing the following research items [2]:

- Combined exergy/energy analyses for community supply structures and buildings, especially those with changing ambient and boundary conditions. This will lead to the implementation of dynamic analyses for complex systems.
- Optimisation strategies for low exergy distribution and building technology system configurations.
- A mandatory holistic system approach to investigate the dependencies between energy production and the use of energy in buildings. This implies the feedback and the response of the building to the grid and energy production strategies.
- Integrated use of local renewable energy sources. Known and new, innovative techniques will be evaluated using new analysis tools. The results will indicate directions for new developments.
- Better control strategies for building service systems to reduce the overall exergy demand.
- Exergy as an indicator for sustainability and for long term, cost efficient solutions.
- Indoor comfort provided by placing the minimum possible exergy demand on building service systems.

STRUCTURE OF THE NEW ACTIVITY

To accomplish these objectives, participants will carry out research and work on developments within the general framework of the following four subtasks: The first subtask, "Exergy Analysis Methodologies", is aimed at development, assessment and analysis methodologies, including a tool development for design and performance analysis of the regarded systems. The second subtask, "Exergy efficient community supply systems", focuses on the development of exergy distribution, and generation and storage system concepts at a community level. A third subtask, "Exergy efficient building technologies", is based on the reduction of exergy demand for the heating, cooling and ventilating of buildings. The last subtask, "Knowledge transfer, dissemination", concentrates on the collection and spreading of information on ongoing and finished work.

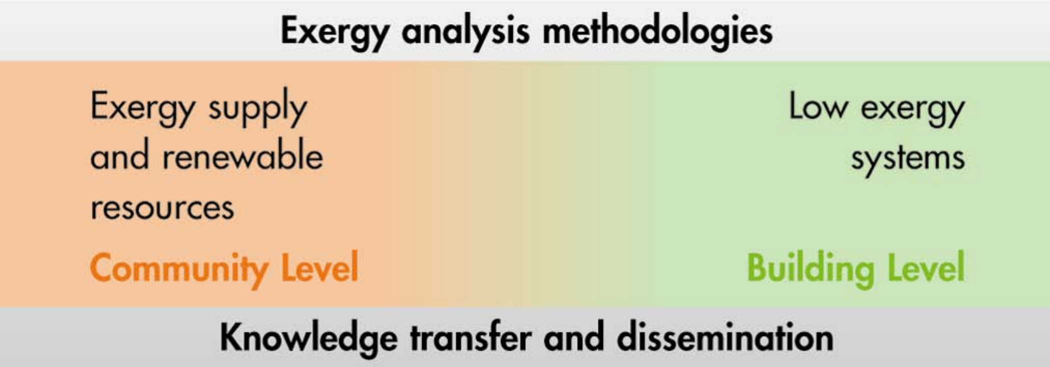

Figure 3. Subtask structure of the ECBCS Annex 49

The community and the building level are directly connected by the final energy conversion process. Nonetheless, the distribution concept for exergy has to be fixed at the community level.

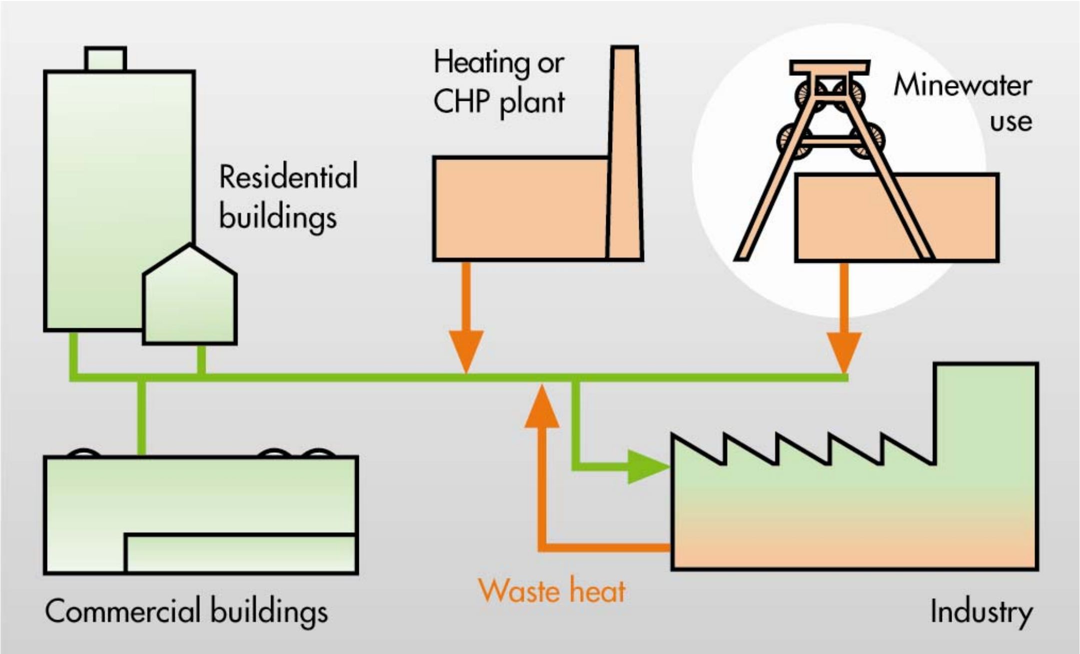

Figure 4. The integration of energy sources from our environment, e.g. the use of water from abandoned mines for heating and cooling buildings., requires exergy efficient supply systems at the community level and adapted building service systems.

EXPECTED RESULTS

The primary presentation of the annex is expected to be an IT based guidebook on how to implement advanced LowEx technology at a community level in the built environment and how to optimise supply structures to ensure low exergy demand of the system solution, while providing good comfort to the occupants and users of the buildings. Furthermore, the work will focus on analysis concepts and design guidelines with regard to exergy metrics for performance and sustainability. A collection of best-practice examples for new and retrofit buildings and techniques will show the potentials of the new approach. With this basis, recommendations for policy measures will be suggested and the aim is to conduct pre-normative work [2].

The focus of the dissemination of documents and other information is to transfer the research results to be used by practitioners. Methods of information dissemination are to include conventional methods such as newsletters and articles, as well as new media, and the Internet is to be used intensively to spread information. Workshops will be organised in different countries to show the latest project results and to provide an exchange platform for the target audience (notably, energy managers, designers, and energy service companies).

OTHER RELATED ACTIVITIES

The International Society of Low Exergy Systems in Buildings (LowExNet) was founded by participants of the completed ECBCS Annex 37. LowExNet members are working with exergy issues, supporting the work n the framework of Annex 49 and have been presenting their results and findings in a number of workshops and seminars, mainly in the framework of international conferences within the field of building technology, building physics and building services. The LowExNet group offers a platform for discussion and information dissemination on the proposed activities. To strengthen and expand the scientific collaboration in the LowEx field, a number of national (e.g. German and Dutch) and European projects have been started [11]. Furthermore, a close collaboration to the ASHRAE Technical Group 1 (TG1) on “Exergy Analyses for Sustainable Buildings” has been established.

CONCLUSIONS

The major benefit of following low exergy design principles is the resulting decrease in the exergy demand in the built environment and related energy supplies. By following the exergy concept, the total CO_2 emissions for the building stock will also be substantially reduced as a result of the use of more efficient energy conversion processes. This new concept supports structures for setting up sustainable and secure energy systems for future building stock. The strategies developed for a better and exergy optimised building design, aimed at a future of clean, clever and competitive energy use, will help to pinpoint specific actions required to reach this goal. Additionally, the exergy demand of buildings will be reduced due to enhanced new heating and cooling systems. The target is to establish a holistic approach for an affordable, comfortable and healthy built environment, while obtaining a minimum input of exergy, and implementing a substantial amount of renewable energy sources into the energy supply of buildings.

Additional and more extensive information can be found on the homepages [2] and [11].

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A Study on The Energy And Water Circulating System Shift by The Eco-systems of Buildings

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SUMMARY

The issue of sustainability has been prevailed not only in building industry but also all other industries. It has been raised that the concept of green building system should be taken into account for the design of buildings. This study is to understand characteristics of the green building system and find the solution of urban problems. Now, over 60% people all over the world live in urban areas, and their communities continue to expand. Pollution and other hazards also tend to accumulate in urban areas. Accordingly, the problems of urban environment are true microcosms of environmental afflictions happening on a global scale. Most of urban problems are caused by human and their buildings. This study was conducted by two ways. One is assessment of energy, and the other is water cycle. Energy and water is also typical factors of urban problems. To solve the urban problems, we must change or shift the wrong circulating systems on (buildings) micro scale. The object building is a recently designed as an environmental-friendly building in Korea that has eco-systems. (as earth-berm, thermal labyrinth, concrete core cooling, soil energy, water-recycle, etc.) Furthermore, comparing eco-system to normal system, difference of energy saving between eco-system and normal system were analyzed by calculation of exergy-entropy of the energy and water cycling system. Consequently, control of micro-environment (the consumption-discharge balance of the energy and the water) in buildings is a key of urban problem solution

INTRODUCTION

A city has been rapidly developed due to progress of Industrialization, so population in it is has been increased. We call it urbanization, and it has been making environmental problems more and more serious. More than 60% of people all over the world are living in urban areas, and all of them try to keep expanding their own area, so a lot of buildings are concentrated in it. Moreover, the people and buildings keep spending water and energy, and it makes excessive shifting or problems of circulating system. Accordingly, the roles of it for sustainable development became new and important issues and. Energy and water circulating systems are causes or effects from urban problems, so the aim of this study is to reduce unstable circulating system and to control the factors that cause urban problems as green building system is applied in a building unit. In brief, the purpose of study is to solve environmental problems of urban area indirectly through buildings.

METHODS

A circulating system in terms of urban area was estimated through technologies of green building, and plans for improving it were made indirectly for the purpose of this study. First of all, problems of urban ecology and environment were organized, and the problems about water circulating system were found. Second, the energy and water shifting/circulation in a building were analyzed according to green building technologies. For this, a building that green building method had been applied was chosen, and the building was analyzed through the applied systems previously. Third, the energy/water circulating system that the green building system was applied was compared and analyzed with the energy/water circulating system that it wasn't applied. Finally, improving effects of energy and water shifting systems were proven through third step for the unstable building.

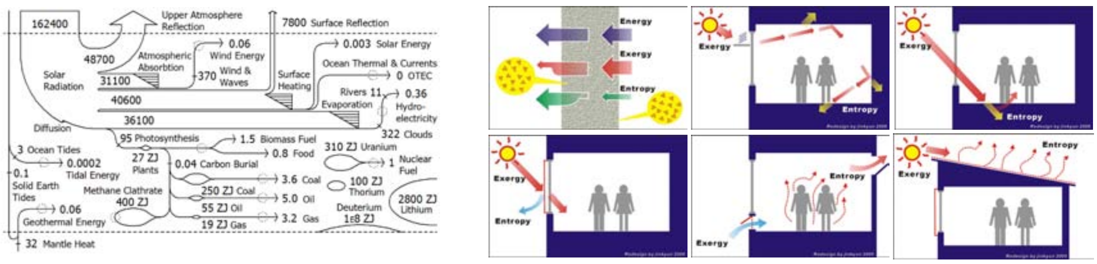
THEORIES

(1) Energy Circulating System

Exergy is originated from energy with the process of material dissolution as all materials in universe are controlled by the second law of thermodynamic. It is also existed in kinetic energy and any other dynamics because most of it is a kind of kinetic energy or dislocating energy that is related with restoration. Exergy is one of energy resources, and an element made by energy production is entropy. Equation (1) is the relationship between exergy and entropy.

$$\text{Consuming Exergy} = \text{Producing Entropy (Inflowing Energy} = \text{Exhausting Energy)} \qquad (1)$$

If the equation (1) doesn't work, the amount of error should be made up from other energy, so the energy shifting can be sequentially unstable.



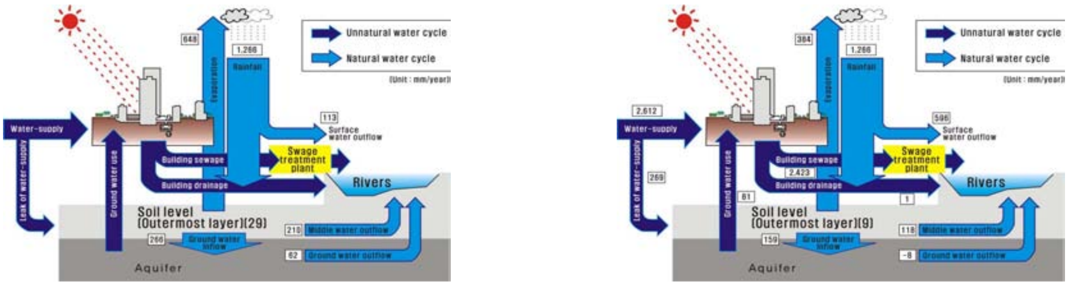
a) b)

Figure 1. Exergy-Entropy Circulation a) Shifting on The Earth, b) Shifting in A Building

Entropy is made by consumption of exergy in buildings, and energy moves to outside through the envelope of the buildings to be stable state. Generally, the shifting from exergy to entropy can be explained by four processes: ① Supply of exergy ② Consumption of exergy ③ Production of entropy, and finally ④Giving entropy off. Buildings consume exergy like electricity and fossil fuel and give off entropy. It affects on the surrounding of the buildings. For example, if it increases, the temperature around the buildings will increase, and many kinds of environmental problems will be caused like green house effect in terms of global environmental system. Accordingly, exergy has to be consumed as less as possible to minimize the environmental effects.

(2) Water Circulating System

Inconsiderable development and excessive paving destroy the water circulating system which is the basis of ecological function. The paving, that water doesn't permeate, has covered more than 70% of an urban area, and 73% of urban area is changing to desert. They are the main reason that functions of eco-system and amenity are lost. In 2002, surface water was increased as much as 483 mm/year (that was 5.3 times of 1962's one) on the other hand the groundwater was decreased as much as 107 mm/year (that is 60% of 1962's one). Moreover, flow of groundwater that goes to watercourses or rivers and shows reversing acting, so urban areas often have overflow and heat island and are accelerated to desert fast. This situation can be resolved through the recovering water circulating system before spaces of urban areas are considered. Accordingly, the alternatives that can recover the water circulating system are necessary with controlling water circulation about water consumption and release and spontaneous generation.



a) b)

Figure 2. Changes of Water Circulating System at Seoul a) in the 1962’, b) in the 2002’.

CASE STUDY : THE APPLIED SYSTEM OF THE OBJECT BUILDING

(1) Analysis of The Object Building

A campus center in E university which is still constructing now was selected for this study to analyze energy and water circulation in it according to the application of green building system. It was planned as a building in earth berm with 6 stories’ basement. The building will be better than any buildings on the ground if it can take enough daylight and ventilation in terms of energy saving, thermal comfort, and residential space. HVAC(or machine and equipment) of the building is based on the energy saving, operating cost, and environmental load as reduction of primary fuel consumption such as enough utilization of regenerative performance and various using of water resources like groundwater and dewatering.

Table 1. Summary of The Object Building And Its Building Service System

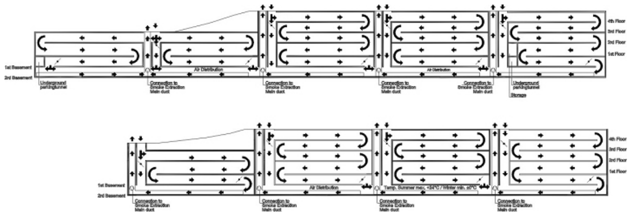
Project	E University Campus Center	Site Location	Seoul, Korea
Structure	RC + SS	Total Floor Area	About 70,000 ㎡ / Basement with 6 Stories
Services	A. Academic : small class room, auditorium, computer lab, seminar room, 24hr reading room, student gallery, theater, recreation, offices. B. Commercial: fitness, academic shop, cafeteria, restaurant, etc. C. Parking Lots		
Cooling	Hot Water Absorption Chillers + Heatpump Type Chillers		
Heating	Co-generation System + Heatpump Type Chillers + Hot Water Boiler		
Ventilation & Air Conditioning	Vairable Air Volume + CCC(Concrete Core Cooling) + CHR(Cooling Heating Radiator)		
Green Building System	Thermal Labyrinth, Geothermal Energy, Water Recyle System (groundwater, dewarering and rain water)		

(2) Applied Green Building System (Eco-system)

• Thermal Labyrinth (Air Soil Duct)

Inside soil of earth berm keeps the temperature constant during a whole year. It is one of the ground characteristics, and it can be used for the energy saving systems through preheating and precooling of outside air to Air handling unit. A double layer wall is used as an air soil duct for the building in earth berm due to the characteristic and architectural necessity, and it plays a role of cool (or hot) tube system. It decreases load of outside air by Δ11℃ (-11℃→0℃) during winter and by Δ7℃ (31℃→24℃) during summer, so operating cost and the size of machine can be possibly decreased.

a)



b)

Figure 3. a) A Building View, b) Thermal Labyrinth in The Object Building

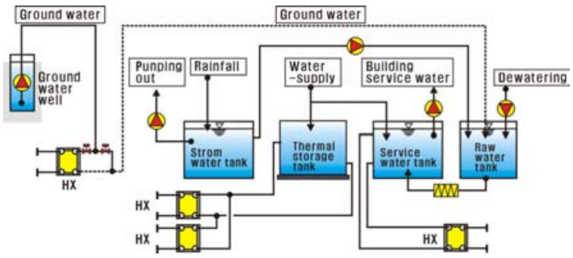
• Geothermal Energy

Pipes are installed on slab of floor to use geothermal energy, and air cooling source is made through the heat exchange with geothermal heat source. The air cooling source is supplied to CCC (Concrete Core Cooling) for 24 hours. After thermal solution (water) is passed by the pipes, the temperature of it is estimated to be around 12℃, and it has heat exchange again until it has 17℃ before it is supplied to CCC.

• CCC(Concrete Core Cooling)

It is a system that controls surface temperature of ceiling, wall or floorboard for residents to be comfortable more than they are with all-air system by cooling radiation even though indoor temperature is over comfort zone. Moreover, water is used as a thermal mass, so the amount of air volume (of AHU) is decreased. Accordingly, duct size can be reduced, delivering (fan) energy can be saved, cooling or heating peak time can be moved during a week, and thermal storage system of concrete (mass) can be combined.

a)



b)

Figure 4. a) CCC (Concrete Core Cooling), b) Recycling of Water Resource

• Recycling of Water Resource

Groundwater with around 15℃ is supplied with 80 ton (8.5 ton/h) per day not only as service water for a building but also as a cooling resource by constant water temperature during a year. The idea about it is to apply heating and cooling system to CCC that cooling source is necessary for 24 hours and to use chilled water, water source, and service water (for washing and gardening). In other words, Groundwater, dewatering, and rainwater are collected in raw water tank, and they get water treatment. After that, the water is used to Re-Cooling. Rainwater is collected in storm water tank during a long spell of rainy weather, and it is usually used for chilled water source, city water for building and landscaping. Dewatering of 150 ton is estimated to be produced.

ENERGY CIRCULATION IN THE OBJECT BUILDING

(1) Comparison with Scheme of Cooling/Heating Plants
A campus center in E university has experimental green building systems, and they are organically interconnected each other. Also, most of the exterior walls are faced on earth berm, so envelop and solar loads can be reduced. The peak load of cooling is about 4,200 kW(1,200 usRT), and the peak load of heating is about 4,000 kW in case of the building, so more than 20% reduction of loads is estimated compare to above ground buildings, and the capacity(size) of cooling/heating plant is decreased. Energy circulating system in the building was compared and analyzed with the case that eco-systems were not applied. Exergy consumption and entropy release were converted into the amount of TOE and CO_2 as a concept of energy input and output.

Table 2. Heating/Cooling System (of Eco-system and Normal System)

Sources		Eco-system		Normal System	
Cooling		App.	Capacity kW(usRT)	App.	Capacity kW(usRT)
1	Geothermal Energy	Y	120(34)	N	-
2	Groundwater & Dewatering	Y	30(9)	N	-
3	Thermal Labyrinth	Y	325(92)	N	-
4	Hot Water Absorption Chillers	Y	950(270)	Y	950(270) x 3ea
5	Electric Chiller	Y	745(212)	Y	745(212) x 3ea
6	(Water) Thermal Storage Tank	Y	2,010(527)	N	-
Total Cooling Capacity		4,180(1,144)		5,085(1,446)	
7	Cooling Tower	Y	3,628	Y	3,628 x 3ea
8	Dry-cooler	Y	500	Y	500 x 3ea
Heating		App.	Capacity kW(kcal/h)	App.	Capacity kW(kcal/h)
1	Geothermal Energy	Y	300(258,000)	N	-
2	Thermal Labyrinth	Y	412(354,300)	N	-
3	Co-generation(Waste Heat)	Y	500(430,000)	Y	500(430,000)
4	Heatpump Type Chiller	Y	960(825,600)	Y	960(825,600) x 2ea
5	Hot Water Boiler	Y	872(750,000) x 2ea	Y	872(750,000) x 3ea
Total Heating Capacity		3,916(3,367,900)		5,036(4,331,200)	

Notes) The difference of heating(cooling) capacity is the difference of load saving between a building on the ground and a building under the earth.

(2) Calculation of Energy Consumption and Release in Building
The full load equivalent hour was use for the calculation of energy consumption for heating and cooling in the building.

Table 3. Boundary Condition for Exergy-Entropy Calculation (in The Object Building)

Full Load Equivalent Hour (hour/year)			
Cooling	273	Cooling Period (90 days : June 15 – September 15)	
Heating	294	Heating Period (150days: November 1 – March 30)	
Carbon Exhausting Coefficient of Energy Sources			
Sources	Calorific value	Carbon Exhausting Coefficient (TC/TOE)	Combustion
Gas	11,000 kcal/N m^3	0.637	0.995
Electricity	860 kcal/kWh	0.787	-

Notes) Operating: 10 hours for every weekday, 6 hours for every Saturday, and no operation on every Sunday
a) TC: Ton Carbon(10^3 Carbon) b) TOE(Ton of Oil Equipvalent) 10^7 kcal

For the energy circulation in the building, exergy and entropy was calculated as the amount of TOE and CO_2 release. The consumption of exergy was 561.75 TOE for the case that the eco-system had been applied and 653.95 TOE for the case that normal system had been applied. Also, release of entropy was 974.51 Ton-CO_2 in former case and 1,202.10 Ton-CO_2 in the latter case. As a result, when the green building systems was applied, exergy was reduced by 14% and entropy was reduced by 19% in terms of energy flow and shifting.

Table 4. The Amount of Exergy-Entropy Release (Between Eco-system and Normal System)

Sources	Energy Source	Eco-system			Normal System		
		Calculation	Exergy (TOE)	Entropy (TCO_2)	Calculation	Exergy (TOE)	Entropy (TCO_2)
Cooling							
Geothermal Energy [a)]	Non	(+32,760 kW)	-	-			
Groundwater [a)]	Non	(+ 8,190 kW)	-	-			
Thermal Labyrinth [a)]	Non	(+88,725 kW)	-	-			
Co-generation	Non	Waste Heat [b)] (+136,500 kW)	-	-	Waste Heat [b)] (+136,500 kW)	-	-
Hot Water Absorption Chiller	Gas	79.3 N m^3/h x 273 h = 21,649 N m^3	22.73	53.09	79.3 N m^3/h x 273 h x 3ea = 64,947 N m^3	68.20	159.29
	Elec.	6.0 kW x 273 h = 1,638 kWh	0.41	1.18	6.0 kW x 273 h x 3ea = 4,914 kWh	1.23	3.55
Electric Chiller	Elec.	239 kW x 273 h = 65,247 kWh	16.31	47.07	239 kW x 273 h x 3ea = 195,741 kWh	48.94	141.22
Water Thermal Storage [c)]	Elec.	239 kW x 624 h = 149,136 kWh	37.29	107.61			
Colling Tower	Elec.	30 kW x 273 h = 8,190 kWh	2.05	5.92	30 kW x 273 h x 3ea = 24,570 kWh	6.15	17.75
Dry-cooler	Elec.	37 kW x 273 h = 10,101 kWh	2.53	7.30	37 kW x 273 h x 3ea = 30,303 kWh	7.58	21.87
Heating							
Geothermal Energy [a)]	Non	(+88,200 kW)	-	-			
Thermal Labyrinth [a)]	Non	(+121,128 kW)	-	-			
Co-generation	Non	Waste Heat [b)] (+147,000 kW)	-	-	Waste Heat [b)] (+147,000 kW)	-	-
Heatpump Type Chiller	Elec.	272 kW x 294 h = 79,968 kWh	20.00	57.71	272 kW x 294 h x 2ea = 159,936 kWh	39.99	115.40
Hot Water Boiler	Gas	79.3 N m^3/h x 294 h x 2ea = 46,629 N m^3	48.97	114.38	79.3 N m^3/h x 294 h x 3ea = 69,943 N m^3	73.45	171.55
Dry-cooler	Elec.	37 kW x 294 h = 10,878 kWh	2.72	7.85	37 kW x 294 h x 2ea = 21,756 kWh	5.44	15.70
Co-generation[d)]							
Gas Turbine Generator	Gas	120 N m^3/h x 7,860 h = 1,051,200 N m^3	1103.77	2578.04	120 N m^3/h x 7,860 h = 1,051,200 N m^3	1103.77	2578.04
Waste Heat	Hot Water	Colling/heating (+4,380,000 kW)	-	-	Colling/heating (+4,380,000 kW)	-	-
Generate Electricity	Elec.	320 kW x 7860 h = 2,803,200 kWh	-700.8	-2022.28	320 kW x 7860 h = 2,803,200 kWh	-700.8	-2022.28
Total Exergy-Entropy			561.75	974.51		653.95	1202.10

Notes) a) ( ): No energy shifting due to spending of natural/surplus energy
b) Waste heat from cogeneration is calculated as a surplus in terms of heat resources.
c) Electric chiller is operated for 624 hours for cool water thermal storage system during nighttime. (8 hours/day x 6 days/week x 13 weeks)
d) Co-generation is operated for 365 days/year

WATER CIRCULATION IN THE OBJECT BUILDING

(1) Comparison with Schemes of Water Recycling

The object building combines all water sources and redistributes them to maximize water utilization. It is to minimize water source waste and unnecessary sewage exhaust. Water circulating system in the building were analyzed through same ways of the comparison with schemes of energy system of cooling/heating, and the annual consumption of city water and sewage were calculated with the concept of water sources consumption and release.

Table 5. Schemes for Water Recycling (of Eco-system And Normal System)

Sources			Requirement	Production	Eco-system	Normal System
1	Service Water	Potable/Washing Water	220 ton/day	-	City Water + Grey Water Use	City Water Use
		Lavatory Water(Toilet)	330 ton/day	-		
2	Supply Water for Cooling Tower		6.8 ton/h	Scatteration		
3	Water for Gardening		100 ton/day	-		
4	Water for Firefighting		100 ton	-		
5	Groundwater		-	80 ton/day	Reusing for Grey Water	Undeveloping
6	Pumping Dewatering		-	150 ton/day		Out-flow
7	Rain Water		-	22,700 ton/yr		

notes) 1. Service water(ton/day) = 110ℓ/day-person x 0.1 person/㎡ x Occupancy area

2. Cooling tower supply water(absorption chiller) = 567.3 ㎥/h(water for cooling tower) x 0.012(rate of scatteration)

5,6. They get treatment before they move to city water tank, and after they move to city water tank, they are supplied to building.

7. Rain water is calculated as monthly rainfall is multiplied by water catchment area (16,865 ㎡).

In terms of the water circulating, consumption and discharge were calculated to the amount of water supply (grey water) and sewage. Water supply was 98,328 ton/yr in case that a green building system had been applied and 187,341 ton/yr in case that the normal system had been applied. Also, the amount of sewage was estimated by 189,251 ton/yr in former case and 249,251 ton/yr in latter case. As a result, water circulating consumption was reduced by 48% and release of it was reduced by 24% when the green building system was applied.

Table 6. The Amount of Water Service Release (between Eco-system And Normal System)

Sources	Eco-system Calculation	Supply		Release	Normal System Calculation	Supply		Release
		City Water (ton/yr)	Grey Water (ton/yr)	Sewage (ton/yr)		City Water (ton/yr)	Grey Water (ton/yr)	Sewage (ton/yr)
City Water	550 ton/day x 6/7[a] x 365day = 172,071 ton/yr	98,328	-73,743	172,071	550 ton/day x 6/7a) x 365 day = 172,071 ton/yr	172,071	-	172,071
Water for Cooling Tower	6.8 ton/h x 273[b] h/yr = 1,856 ton/yr	-	-1,856	(1,856)	6.8 ton/h x 273[b] h/yr x 3ea = 1,856 ton/yr	5,570	-	(5,570)
Water for Gardening	100 ton/day x 12 day x 8 month = 9,600 ton/yr	-	-9,600	(9,600)	100 ton/day x 12 day x 8 month = 9,600 ton/yr	9,600	-	(9,600)
Water for Firefighting	100 ton (Store)	-	-100	-	100 ton (Store)	100	-	-
Ground water	80 ton/day x 6/7[a] x 365 day = 25,029 ton/yr	-	+25,029	-	Non	-	-	-
Dewatering	150 ton/day x 365 day = 54,750 ton/yr	-	+43,800	10,950	150 ton/day x 365 day = 54,750 ton/yr	-	-	54,750
Rain-fall [c]	22,700 ton/yr	-	+16,470	6,230	22,700 ton/yr	-	-	22,700
Total Supply-Release		98,328		189,251		187,341		249,251

notes) a) Building use : 6-days of One week(7-days) b) Cooling full load equivalent hour: 273 h

c) 20% of it is discharged before they are supplied to stormwater tank / Collectable period (from march to october)

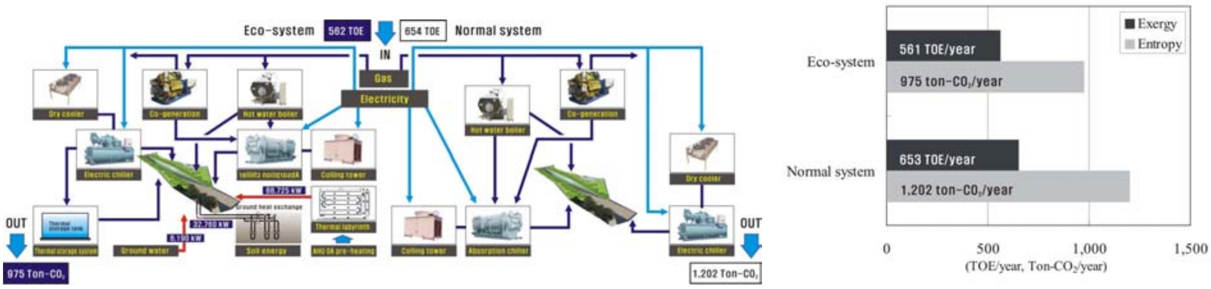
( ) : Cooling tower supply water: scatterate in the air / Gardening water: ground inflow

RESULTS

First of all, energy and water source can be saved a lot when the green building system was applied. However, more important factor is to restrict excessive consumption and release of energy and water circulation in the building. In case of consumption in the building, and exergy was reduced by 14%, and the amount of city water was reduced by 48%. Furthermore, in case of release, entropy was reduced by 19%, and sewage was reduced by 24%. If exergy consumption and entropy release are reduced, bad influence around building area can be minimized, and further, the causes of environmental problems in an urban area can be reduced.

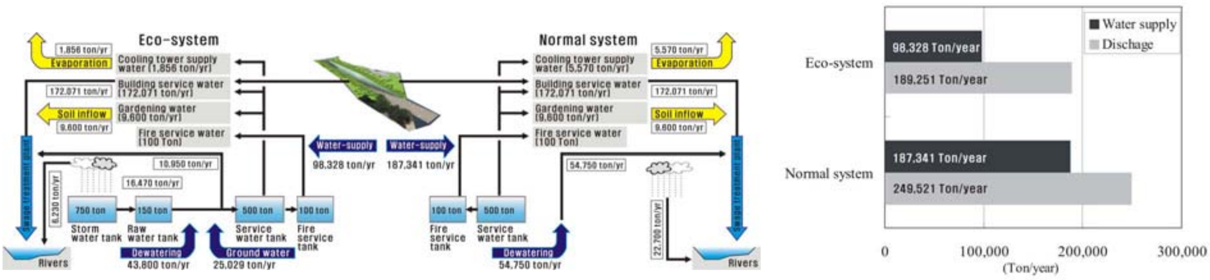
Table 7. The Result of Energy and Water Circulating System

Sources		Eco-system	Normal system	Reduction
Energy	Supplied Exergy (TOE)	561	653	(-)14%
	Released Entropy (Ton-CO_2)	974.50	1202.10	(-)19%
Water	Supplied Water (Ton/yr)	98,328	187,341	(-)48%
	Released Water (Ton/yr)	189,251	249,251	(-)24%



a) b)

Figure 5. Energy Circulation in The Building. a) Flow Diagram, b) Result Graph



a) b)

Figure 6. Water Circulation in The Building. a) Flow Diagram, b) Result Graph

CONCULUSIONS

The data in this study can be little different from the real data after the campus center in E university is constructed completely because the consumption-release calculation of energy and water circulation in this research is based on the (SD)plan of the building. However, to solve the problems of urban environment and eco-system, research of energy and water circulation in building with green building method and minimizing causes of problems is very important. There are four categories of the process about it.

(1) As a human uses a building, circulation of energy and water is caused, and it has an effect on problems of urban environment.

(2) All circulation systems have the process that is from consumption to release. It is from exergy to entropy in energy, and it is from water supply to sewage in water.

(3) Entropy and sewage can be controlled as green building system is applied and as the exergy and water supply are controlled in a building which is micro aspect. This method saves energy and minimized the environmental pollutants with improving circulating system that have excessive consumption and release.
(4) Primarily, matters' release from a building which can be environmental problems for urban area in terms of micro aspect can be reduced by controlling them in the building. In other words, the urban area and the eco-system can be recovered to stable circulating system is as macro environmental(urban) problems are solved with the removal of micro environmental problems in a building. Consequently, when a green building system is planned, the not only energy saving factors but also environmental factors have to be examined practically.

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How can we improve the efficiency of exploitation of geothermal energy

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SUMMARY

Nowadays no energy source should exist without regulation to ensure its efficient use. The regulation can be quantitative, qualitative or combined. This article compares the utilization of geothermal well in two situations. The first situation is when no regulation is used on the geothermal well head. The second situation is when the exploitation of geothermal water is regulated on the geothermal well head, i.e. quantitative regulation is considered. The regulation on the geothermal well head grants savings on the geothermal well because only as much geothermal water is consumed as needed. Therefore the lifetime of the geothermal well is elongated. In this case of regulation, the efficiency of geothermal water utilization is increased and it is nearly constant for all winter months.

INTRODUCTION

Slovak Republic is a small country, but with lots of geothermal wells. On the area of 49 035,81 km^2 there are 117 geothermal wells with the temperature of the geothermal water ranging from 18 °C to 129 °C. The geothermal energy in Slovak Republic is used for recreational purposes (thermal swimming pools), balneology, preparation of hot water, heating of buildings or greenhouses. Although we have a high energy potential of renewable energy – geothermal energy, the effectiveness of using this energy is very low. This article will show the approach of relation between the geothermal energy and its use, especially how to deal with the regulation of geothermal well head.

The main reason for the low effectiveness of geothermal energy utilization in Slovak Republic is that it is used only for one purpose or so called “at one level”, i.e. the cascaded use is not applicated. However, we can achieve increased effectiveness if we harmonize the relation between the heat demand and energy supply.

The heat demand depends on one crucial factor and this is the weather or season. The energy supply of geothermal energy is practically constant because the temperature of the geothermal water on the well head is the same for the whole year. The production of well depends on whether there is or is not a pump. Majority of the geothermal wells in Slovak Republic do not have pump and if they do, the pump is with constant speed, therefore no quantitative regulation exists.

METHODS

A more respectable approach to the use of geothermal energy is shown in this article. The main emphasis is given to the relation between the heat demand an energy supply. The relation is demonstrated at a particular geothermal well FGT-1, which is situated in a small village – Topoľniky. The characteristic values of the geothermal well FGT-1 are: the temperature of the geothermal water $\theta_o = 74$ °C and the production of the well $m_0 = 23{,}0$ l/s.

Characteristic parameters for the geothermal well are: available energy potential Q´ (kW), available energy quantum Q (MWh) and available volume of geothermal water M (m^3).

$$Q' = m_0.c_v.\rho.(\theta_0-\theta_r) \qquad (1)$$

$$Q = 24.Q'.n.10^{-3} \qquad (2)$$

$$M = m_0.n.24.3600.10^{-3} \qquad (3)$$

where m_0 is the production of the geothermal well (l/s), c_v is the specific heat of water (J/kg.K), ρ is the specific gravity of water (kg/m^3),θ_0 is the temperature of water at the geothermal well head (°C), θ_r is the reference temperature of down cooling (°C), n is the number of days (day).

The characteristic parameters of the well FGT-1 used as input data to the calculations are presented in the table 1.

Table 1 Characteristic parameters of the geothermal well FGT-1

Period	**n (day)**	**Q´(kW)**	**Q (MWh)**	**Q (GJ)**	**M (m^3)**
Day	1	5 682	136	489	1 987
Week	7	5 682	955	3 438	13 910
Month	30	5 682	4 091	14 728	59 616
Summer season	62	5 682	8 455	30 438	123 206
Winter season	**202**	**5 682**	**27 546**	**99 166**	**401 414**
Transition period	101	5 682	13 773	49 583	200 707

The reference temperature in Slovakia is $\theta_r = +15$ °C, which is the average statistical temperature and it is calculated as an annual average based on the average month temperature of the discharging waste geothermal water. In the summer the reference temperature is in the range from + 22 °C to + 26 °C and in the winter the reference temperature is in the range from + 4 °C to +8 °C. The energy quantum of geothermal well FGT-1 for winter is calculated for the reference temperature $\theta_r = +20$ °C. This temperature is much more satisfactory and available than the reference temperature $\theta_r = +15$ °C. From the table 1 it is evident, that in the winter season we have 401 414 m^3 of geothermal water and 27 546 MWh of energy quantum at disposal from the geothermal well.

The geothermal water from the geothermal well FGT-1 supplies 7 equal greenhouses. Each greenhouse is 93,12 m long and 24,77 m wide. The height to the side board of the greenhouse is 2,4 m and to the top of the greenhouse is 4,03 m. The heat requirement of one greenhouse is approximately 550 kW, so the total heat requirement from the extraction site is
7 x 550 = 3 850 kW.

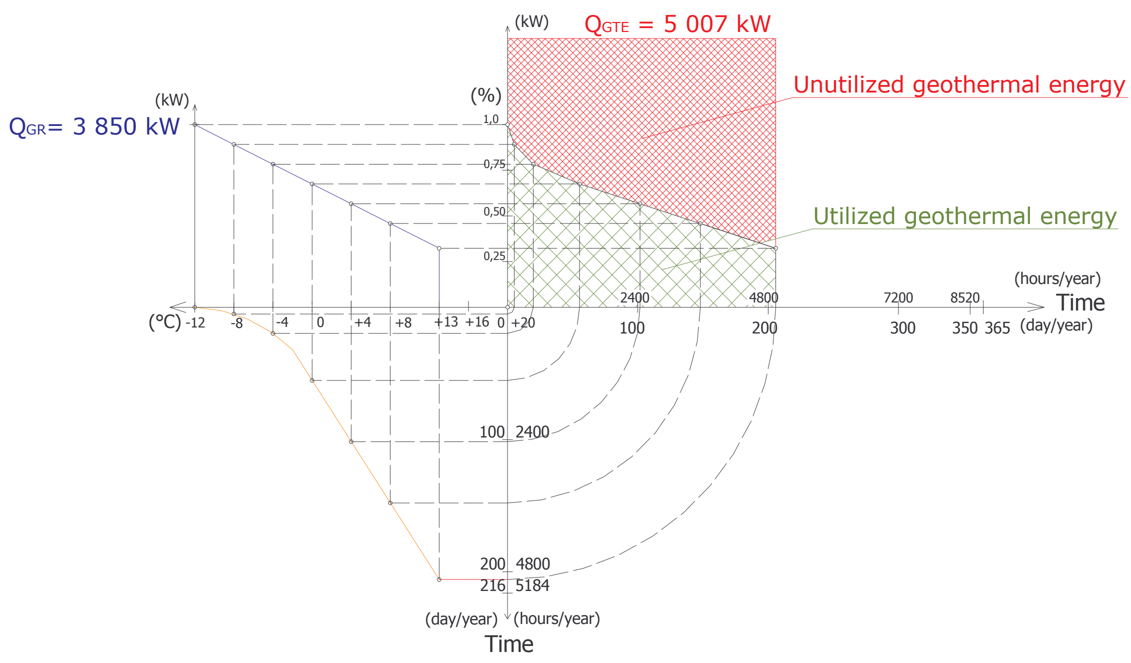
RESULTS

The table 2 presents the heat requirements of the greenhouses depending on the year season without the regulation on the geothermal well head. The map curve of the heat demand of the greenhouses without the regulation on geothermal well head in shown on the Figure 1.

Table 2 Heat requirements of the greenhouses – without regulation

Parameter	Month						
	10	**11**	**12**	**1**	**2**	**3**	**4**
Average outside air temperature (°C)	11,0	5,1	1,1	-1,0	0,9	6,0	11,1
Proportional load of heat demand (-) (empirical)	0,33	0,75	1,0	1,0	1,0	0,75	0,33
Heat demand of one greenhouse (kW)	182	413	550	550	550	413	182
Heat demand of all greenhouses (kW)	1271	2888	3850	3850	3850	2888	1271
Drop of temperature of geothermal water within one greenhouse (K)	1,9	4,3	5,7	5,7	5,7	4,3	1,9
Drop of temperature of geothermal water within all greenhouses (K)	13,2	29,9	39,9	39,9	39,9	29,9	13,2
Temperature of geothermal water after utilization in one greenhouse (ºC)	70,1	67,7	66,3	66,3	66,3	67,7	70,1
Temperature of geothermal water after utilization in all greenhouses (ºC)	**58,8**	**42,1**	**32,1**	**32,1**	**32,1**	**42,1**	**58,8**
Unutilized energy (%)	**75**	**42**	**23**	**23**	**23**	**42**	**75**

Fig. 1 The map curve of the heat demand of greenhouses without regulation

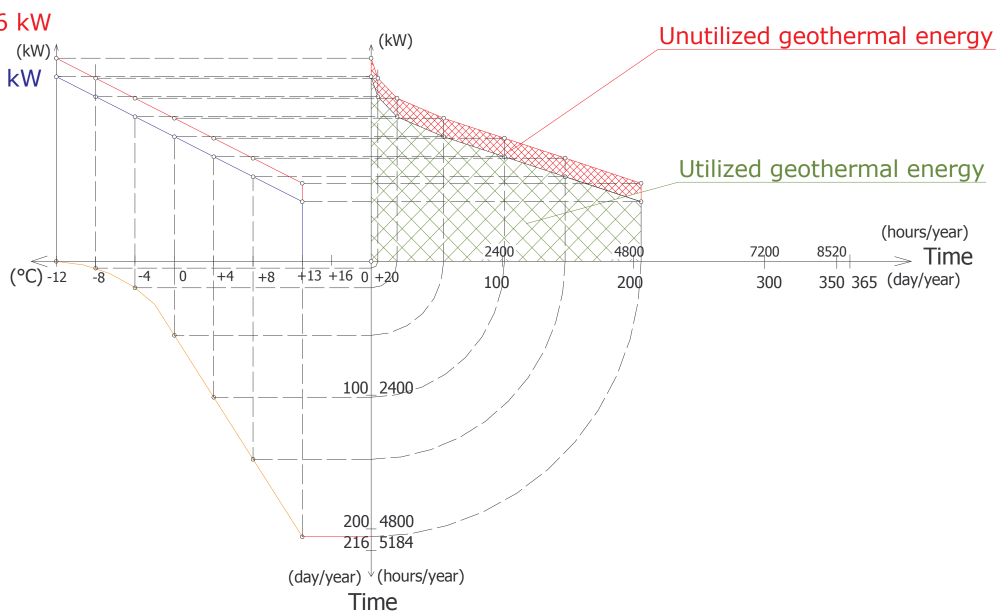


The table 3 presents the heat requirements of the greenhouses depending on the year season with the quantitative regulation on the geothermal well head. The map curve of the heat demand of the greenhouses with the regulation on geothermal well head is shown on the Figure 2.

Table 3 Heat requirements of the greenhouses - with regulation

Parameter	**Month**						
	10	**11**	**12**	**1**	**2**	**3**	**4**
Average temperature (°C)	11,0	5,1	1,1	-1,0	0,9	6,0	11,1
Proportional load of heat demand (-) (empirical)	0,33	0,75	1,0	1,0	1,0	0,75	0,33
Heat demand of greenhouses (kW)	1271	2888	3850	3850	3850	2888	1271
Production of the well (l/s)	6,4	14,6	19,5	19,5	19,5	14,6	6,4
Drop of temperature of geothermal water within all greenhouses (K)	47,4	47,2	47,2	47,2	47,2	47,2	47,4
Temperature of geothermal water after utilization (°C)	**24,6**	**24,8**	**24,8**	**24,8**	**24,8**	**24,8**	**24,6**
Available potential of geothermal energy (kW)	1393	3179	4246	4246	4246	3179	1393
Unutilized energy (%)	**9%**	**9%**	**9%**	**9%**	**9%**	**9%**	**9%**

Fig. 2 The map curve of heat demand of greenhouses with regulation



DISCUSSION

The utilization of geothermal energy is considered efficient when 90% is used and 10% held as a reserve. The table 2 in the results section shows that the geothermal well FGT-1 is not utilized to this extent even in the coldest moths of the year as December, January and February. However the utilization in the coldest moths of the year is 77%, which is considered satisfactory. The utilization in the months March and November is very low but in October and April it is insufficient. To increase the exploitation in more effective way, the regulation of the production on the geothermal well head according to the heat demand of the greenhouses in each month should be employed. The benefits of the regulation are clearly visible from the results presented in the table 3 and figure 2. It shows that the utilization of the geothermal well FGT-1 is approximately 91% and the rest of the geothermal energy serves as the reserve. In the coldest months the production of the geothermal well is reduced to the value m_1 = 19,5 l/s; in March and November it is reduced to the value m_2 = 14,6 l/s and in October and April to the value m_3 = 6,4 l/s. This approach to the utilization of geothermal energy ensures a reserve of approximately 10 % of the energy in each month.

In the stated example in the coldest months the production of the well with the regulation on the geothermal well head reaches only 85% in comparison to the measured value on the well head. This shows that in the area of the village Topoľniky more than seven greenhouses could be supplied by this well. If the production would be the full 23,0 l/s, ten equal greenhouses could be heated. The regulation of production on the well head ensures preferable map of curve for the greenhouses and the reserve of energy in each month is constant.

Another approach suggests that in the coldest months more geothermal water would be pumped out of the well than it is calculated for these months, but the overall quantity of geothermal water needed for winter season should not be exceeded. Therefore the use of this kind of regulation on the well head would provide supply of geothermal water for more than seven or ten greenhouses. In the coldest months of the year (December, January and February) the production would be increased and would be higher than it is measured on the well head. But in other months the production would be reduced according to the needs of the season.

These two examples show us that the regulation on the well head brings these advantages:

- elongation the lifetime of geothermal well in general,
- constant percentage utilization of geothermal energy during winter season,
- increase of amount of point of supply (e. g. more greenhouses heated with geothermal water).

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First Experiences with Coupled Dynamic Simulation of Building Energy Systems and the District Heating Network

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SUMMARY

New distributed simulation environment has been used to simulate the energy flows in a simple case community consisting of four buildings, power plant and a district heating network. Distributing computing task on different computers accelerates the simulation, but simultaneously some time is spent on communication. The tests indicated that a one year energy simulation of a community could be performed within 3.5-10 days of calendar time depending on the time step used. It was also found that the Internet-communication of data takes approximately 20-25 % of the total simulation time. This share may be even higher if simulated case covers frequent energy consumption peaks and simultaneously high degree of accuracy for the results is required. For optimal performance in the future, the models should be able to recognize steep gradients and adjust the time step accordingly.

INTRODUCTION

This paper presents first results of the simulations covering the whole energy chain extending from the generation site (power plant) through district heating network to the consumption site (buildings). To perform this task, a new distributed computing environment was developed. The basic idea of a distributed simulation environment has been presented in our earlier publication [1] and elaborated in new context, in which building simulators and district heating network simulator are linked. The necessary ICT-architecture was defined to handle the environment stability under consideration of different requirements (convergence, data exchange, security). Considerable effort was spent to sustain a generic approach when defining the requirements.

There is a possibility to have a benefit of using distributed simulation environment, which enables time-consuming simulations to be faster. In addition, the distributed computing with certain rules enables totally new dynamic simulations for the energy system, for example: buildings, district heating networks and power plants simulated synchronized as a whole. Furthermore, anticipating that generic service rules and connection methodology existed, it would be considerably easier in the future to expand an open architecture by linking new models. Our approach utilizes generic ICT-components and interfaces used in distributed solution environment to handle coupling challenge between different simulation platforms. The distributed solution environment can be run over the internet or LAN-networks.

SIMULATION ENVIRONMENT

The distributed generation needs new type of analysis tools to design and to optimize the energy chains from the power plants (both large and small scale) to the consumption points.

Traditional large scale production-consuming environment is less complicated than distributed and can be handled with separate modeling and simulation tools with sufficient accuracy. The suggested distributed generation environment needs new tools to handle the challenging multi-production and multi-consuming phenomena in the transmission networks to enhance the quality of the design and optimize the use of the energy system.

In this context, our simulation environment consists of building simulators and their connections to the district heating network simulator,
Figure 1. The interaction between programs and the synchronization of the simulation is performed by the Master-program. The Master is responsible for handling the data exchange, the project data and the other simulation specific features between the simulator services.

There can be several simulator-instances running on their own hardware, but for simplicity reasons there is only one of each presented in
Figure 1. The connections between different actors (marked as two-headed arrows) are implemented on the WebService technology. By definition, all actors have a WebService-interface. The approach is generic and allows new simulators to be added later on (e.g. marked shadowed in
Figure 1).

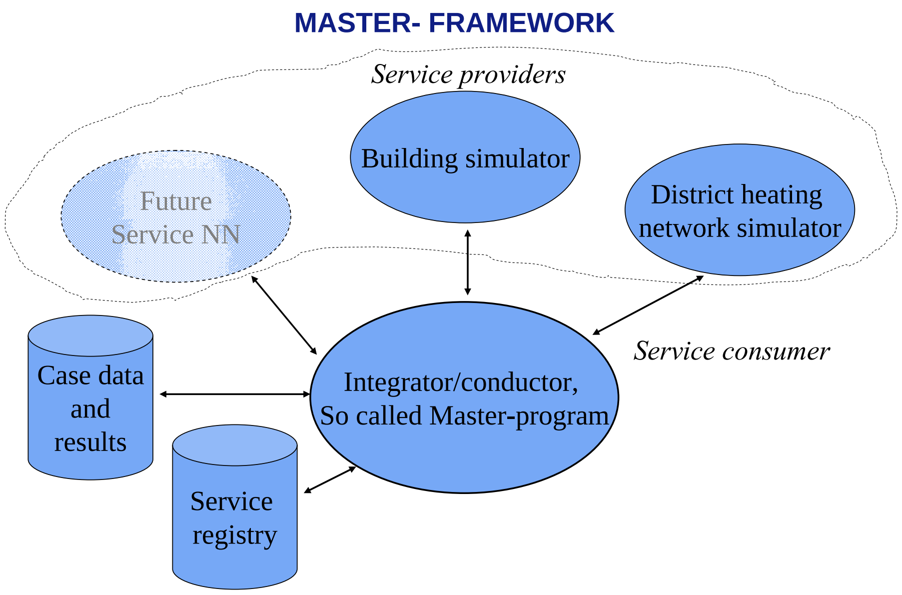

Figure 1. The framework of connections between the building simulator and the district heating network simulator.

Substation model (pT-adapter)

The building heating system as well as the eventual local heat generation system is connected to the district heating network by means of a substation. The main components in a substation are the heat exchangers but also pumps, pipes, valves and controllers are needed. From the modeling point of view, the temperatures and pressures at the coupling points are the parameters that ultimately determine the heat and mass flow and therefore the substation model is called a pT-adapter.
A dynamic simulation model for a counter flow heat exchanger is easily implemented by connecting thermally the two heat exchanging flow channels and dividing them into a number

of successive elements, in order to take into account the variable temperature difference between the two streams in different parts of the heat exchanger. But it was found that a large number of elements are needed if good predicting accuracy is required: about 40 elements are necessary to compute the heat power within 2 % accuracy.

A more accurate modeling option is to use conventional equations for thermal efficiency of a counter flow heat exchanger but then we lose the transient behavior inside the heat exchanger. This is unlikely to be a major restriction because normally plate heat exchangers are used, which have a low mass and a small water volume and therefore this kind of modeling is preferred.

To accurately simulate the behavior of commercial counter-flow plate heat exchangers, we need to know the product data, namely the heat transfer correlations and pressure loss correlations of the flow channels. A new method was developed to extract this data from the results of the manufacturer's simulation program, or from measured results. Then it is possible to size the heat exchanger for the simulations based on the design heat power and pressure losses, the result being the size and number of the heat transfer plates, as shown in Figure 2. All this information is subsequently needed when the heat exchanger is used in the actual simulations.

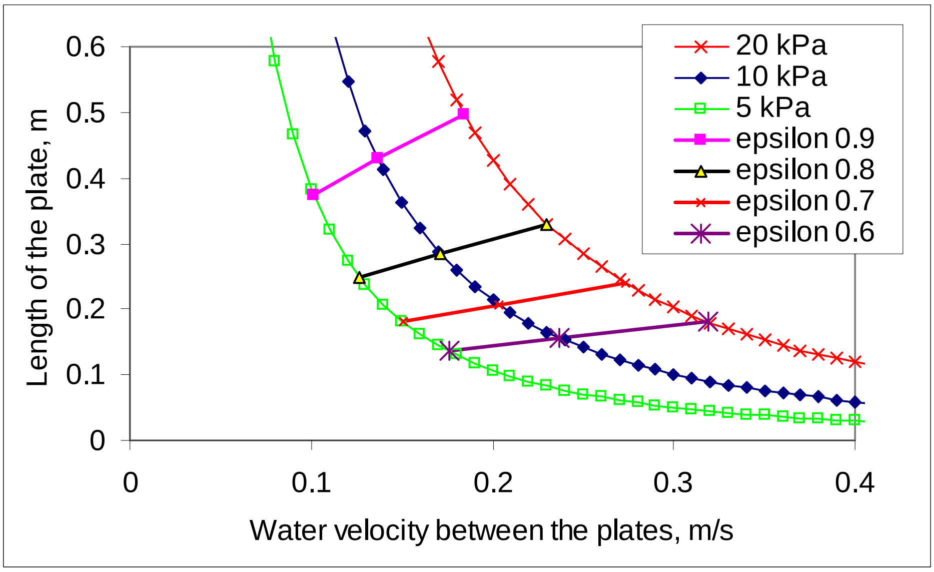

Figure 2. An example of sizing of a plate heat exchanger for district heating.

The intersections of the efficiency and pressure loss lines determine the plate dimensions and the water velocity in Figure 2. The number of plates depends on the required design heat output. The heat transfer and pressure loss properties have been extracted from the manufacturer data.

SIMULATION CASE AND RESULTS

The simulated test case is a simple district heating network consisting of four consumers with varying heat loads and a heating plant supplying the needed energy, Figure 3. Energy consumption in each building was calculated using building energy simulation software. The district heating network along with the heating plant was modelled by a network model. The building and network models communicate through a Master-application. The test application was a totally new whole simulation of a community in Nordic climate (a district heating network together with buildings). The application covered both the hydraulic and the thermal flows in the networks as well as the indoor climate conditions in the buildings.

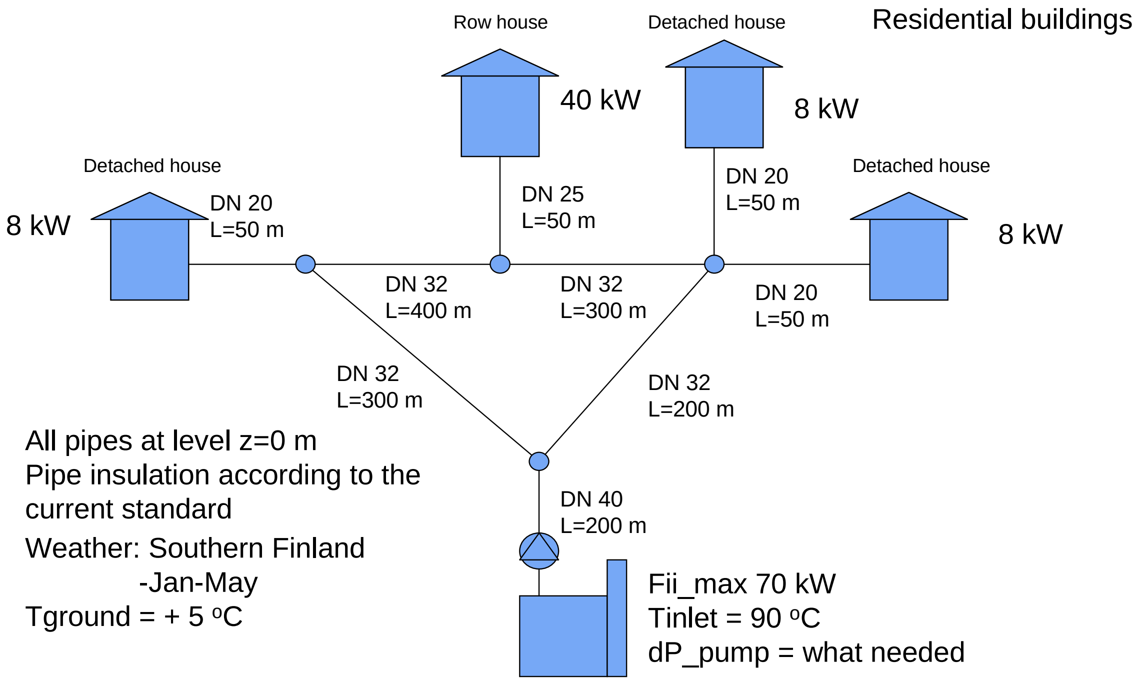

Figure 3. Schematic representation of the simulated case.

All the data produced by the models is made available for the Master-application. In this examination, only the variables needed to construct energy balances were recorded in an output file. These variables were the supply and return temperatures in the consumer and heating plant nodes, mass flows and heat losses in the network. Additionally, the total heat power demand and the variables concerning the tapping hot water, temperatures and mass flow, were recorded for every building.

The heating demand of the buildings and the heat losses in the network were compared to the heating power produced by the heating plant. Also, the consumption of an individual building was compared to the heating power received from the district heating network. The longer term results for the entire network are summarised in Table 1.

Table 1. Energy consumption for long simulation (19 weeks) and relative differences between energy balances calculated by the building energy simulation tool and district heating network simulation tool.

(kWh) -->	13 400	32 400	14 100	15 800	165 200
external time step	Building 1	Building 2	Building 3	Building 4	Power Plant
900 s	1.5 %	0.4 %	1.1 %	1.5 %	-6.6 %
600 s	1.9 %	-0.1 %	1.4 %	1.9 %	-4.9 %
480 s	1.8 %	0.1 %	1.2 %	1.8 %	-3.9 %
300 s	1.0 %	0.1 %	0.8 %	1.0 %	-2.3 %
120 s	0.7 %	0.2 %	0.6 %	0.7 %	-0.1 %
20 s	0.6 %	0.2 %	0.5 %	0.6 %	1.3 %

The results imply that reducing the time step improves the match of energy balances. Closer examination of the dynamic variation of the heating power demand for a single building showed that the load due to basic heating of a building can be calculated with longer time step, because the changes are slow. However, any hot water tapping in the buildings cause rapid changes in the consumption and should be calculated using a shorter time step to improve accuracy, see Figure 4. In the future, the models should be able to recognize steep gradients and adjust the time step automatically.

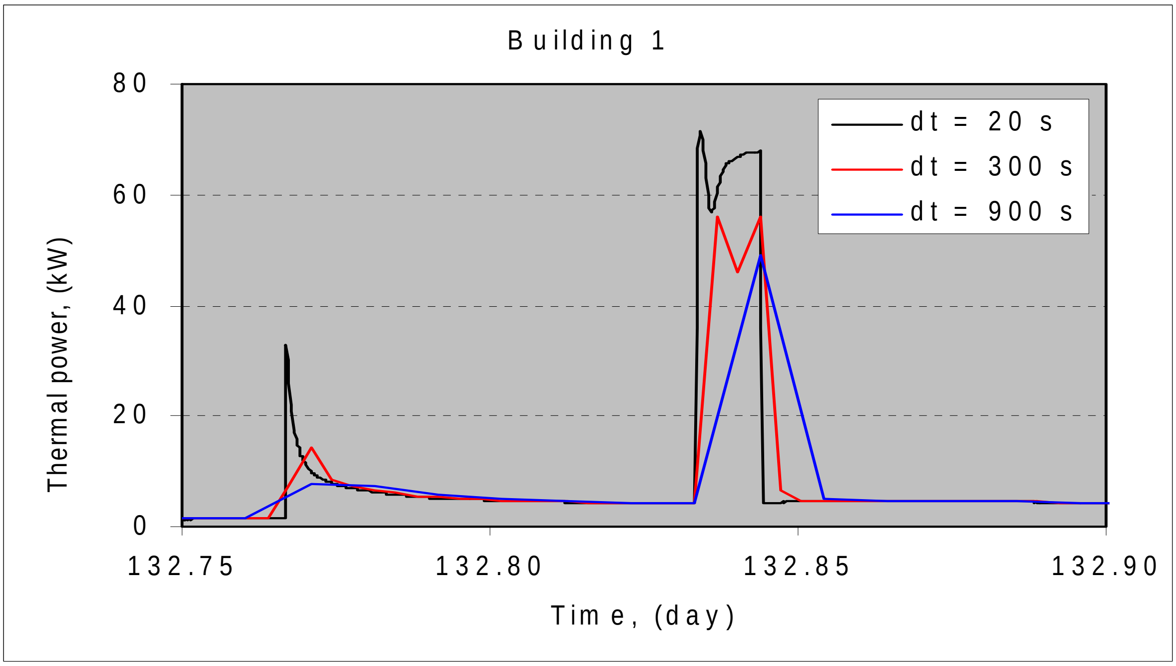

Figure 4. Example of simulated thermal load peak due to hot water consumption in Building 1 (time step is a parameter).

The performance of the distributed simulation environment

Several simulations with various external time-steps were carried out to investigate the general performance of the distributed simulation environment. The hardware set of the distributed simulation environment consists of 4 different desktop-PC:s. Table 2 shows the hardware and software configuration used to simulate the test case. The configuration was first fine-tuned and tested to have balanced load for all hardware components. It was found out that the most computing intensive part of the test case was the building simulator. The district heating network required less computational effort and the Master was easiest to compute.

Table 2. The hardware and software configuration of the distributed simulation environment.

Softaware type	Hardware configuration	Software configuration
Building simulator	Intel 2 CPU 6600	2 x VTT House
Building simulator	Intel 2 CPU 6600	2 x VTT House
District heating network simulator	PC Pentium III	1 x Matlab network simulator
Integrator	PC Pentium IV	1 x Master-program

Test simulations covered the 19 weeks period from Jan 1st to May 15th with external time-steps 20 s, 120 s, 300 s, 480 s 600 s and 900 s. The external time step refers to the data update interval in the Master-program. Both the building simulator and the network simulator may internally use shorter time step when needed. The simulation speed-factor, which means how much faster the simulation is when compared to the real time clock, varied from the 37 to 100. This means that depending on the time step, the full yearly simulation of a test community would last from 3.5 days (dt = 900 s) to 10 days (dt = 20 s).

Using distributed simulation reduces the computing time, but on the other hand some time is spent on communication between the simulators and the integrator program. This latter effect was studied by comparing the shares of the Internet communication time in the external time step. The test simulation with various external time-steps showed that the average network interaction time between the Master and the simulators was 0.361 seconds during one external time-step. This value remained nearly constant in different simulations. Figure 5 shows the average relative share of the Internet communication traffic time plotted against various external time-steps. It can be seen that relative duration of the network traffic drops quickly as the external time-step of the Master-program increases.

For example, during the hot water tapping in the building – when shorter external time steps are necessary – the Internet network traffic of the distributed simulation environment can represent more than half of the total simulation time. When these results are generalized to the community level, it means that mornings (from 7 to 10 am) and evenings (from 8 to 10 pm) are the slowest to simulate. Assuming 5 h of shorter external time step (20 s) and 19 h of longer external time-step (300 s) per one simulated day, the communication would take from 20 % to 25 % of the total simulation time.

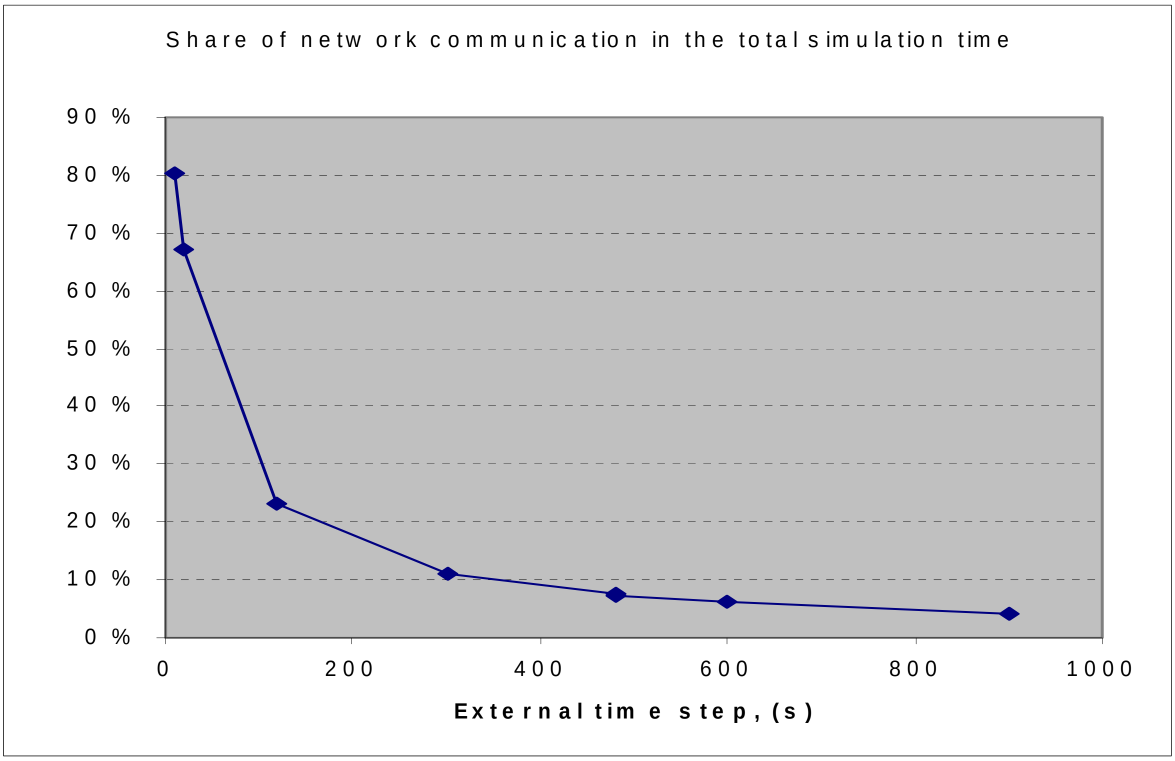

Figure 5. Share of the Internet-network communication time in the total simulation time.

CONCLUSIONS

In reality buildings connect to the district heating network in substations. There are two options to model the heat exchangers in the substation: the dynamic model is computationally heavy whereas the static one may lose some very fast transients. A method was developed for extracting the necessary heat exchanger information from the manufacturer provided data.

New distributed simulation environment has been created and used to simulate the energy flows in a simple case community consisting of four buildings, power plant and a district heating network. To carry out this task, building simulations and district heating network simulation were all performed in different computers linked through the Internet to a Master-program.

Using distributed simulation considerably accelerates computation, but simultaneously some time is spent on communication between the simulators and the integrator program.

It was found that for a typical community level energy simulation the Internet-communication of data may take approximately 20 - 25 % of the total simulation time. This is based on an assumption that more intensive computing is required during the hot water tapping peaks in the mornings and in the evenings. The time share of Internet communication may be even higher if simulated case covers numerous energy consumption peaks and simultaneously high degree of accuracy for the results is required. For optimal performance in the future, the models should be able to recognize steep gradients and adjust the time step accordingly.

ACKNOWLEDGEMENT

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A linear programming based model for strategic management of district heating systems

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SUMMARY

This paper is devoted to the development of a decision support system that could assist district heating authorities in strategic management of their asset. Such a tool is designed in order to answer a variety of questions asked by concerned stakeholders regarding the future of a district heating (DH) system about optimal plant and fuel choice, possible network extension, valorisation of heat surplus, etc.
The kernel of this system is a simulation model which determines the energy balance of a given DH network, over any type period. The time period is cut into different time steps, assuming steady state on each time step.
At each time step, a linear programming LP based approach is used in order to compute the optimal combination of heat power at the plants to meet the demand. The modelling problem is formulated on a graph and LP as the search for minimum energy production cost rooted at the sources.

INTRODUCTION

District heating (DH) is presently considered in Europe as a challenging technology, both as a great step toward sustainability of energy systems and as a powerful instrument for local energy policy managed by communities. The bases for establishing a DH supply is the possibility of obtaining higher efficiency and thereby lower heating costs when producing the heat in few large plants than when using small private boiler units. DH supply gives also the opportunity to utilise waste heat, that otherwise could not be used, especially from incineration plants. DH is also a good way to make efficient use of other local and renewable energy resources such as biomass, biogas, geothermy and to develop cogeneration. Moreover, environmental issues have lead to an even greater interest in DH supply, due to the possibility of cost-effective flue gas treatment in centralised plants, in order to limit local atmospheric pollution, or in the near future, CO_2 emissions.

The need for tools for decision making and strategic management is therefore obvious. The control and strategic management of a DH network is a complex process in which different factors like costs, availability, heat losses and environmental impacts must be taken into account. For this reason, mathematical aided models are being increasingly in use in order to identify the best decision factors in different operating situations and scenarios.
Examples of scenarios to be evaluated are : i) Refurbishment and extension of the DH system ; ii) Changing energy sources or plants ; iii) Variation of the heat demand ;
iv) Changes in control and operational management of the DH ; v) Defining new policies in heat pricing and connections. An important feature is also *ex-post* evaluation of network

operating, in particular when DH is operated by a private company, under authority of the community.

These models require appropriate mathematical description of the different subsystems which form the DH. Typical methods account for simulation by minimizing operational DH costs. A literature study has shown that several methods have been proposed for minimisation of the operational cost of DH systems, but only very few references include case studies where the suggested methods are general and could be applied to any network. Due to the great complexity of the problem, most of the references suggest methods which only regard some parts of the total problem, while other parts are not considered or treated separately. The proposed methods can, thus, be divided into two categories:

(1) Strategic and mid/long term models which includes two subgroups:

- Determination of the optimum load distribution between different heat producing units, in a well specified DH system: network topology, energy supply and heat plants are known. DH supply temperature is considered constant or predetermined and DH networks dynamics is disregarded. Approaches in this area include markedly the linear programming and mixed integer programming techniques as solving algorithms, at each time step [5] [6] [7]. This type of modelling cannot be actually considered as an optimisation approach (although an optimisation algorithm is performed), but rather a time-dependent simulation of a given DH system under specified conditions.

- Optimisation of the network structure by looking for the best construction and operational cost. These methods discuss namely the geometric optimization of the DH distribution system and the choice of the most cost-effective heat plants (to be build) and energy sources, accounting for investment and operation costs [9] [12]. Linear programming and mixed integer programming techniques are generally used as solving algorithms as well, but they incorporate a global cost optimisation over a long-time horizon.

(2) Fully dynamic optimization determining the optimum supply temperatures and system control, often related to operational studies. Some dynamic approaches, based on an aggregation of the DH network, are discussed by a number of authors [3] [4] [8], but a general method for solving the problem does not exist today.

The model proposed in this paper is aimed to emulate the DH running on a given period, either from the past, on the basis of historical data, or for scenarios of the future. The simulation model investigates the energy balance in each time step. The environmental and economic balances could be recognised. The model is a masterpiece of a strategic management tool that includes as well detailed data bases on the DH network e.g. outside temperature records, heat load, maximum heating power, heat losses, etc.

The model could assist decision makers in evaluating the impact of several strategic choices, by simulating the effect of different scenarios related to the introduction of new heating plants, new consumers, etc. Nevertheless, in a perspective of decision support, when the number of potential scenarios to be tested is high or a very large panel of actions is possible, the simulation model is not always sufficient. So it could be complemented and coupled with an optimization module in order to identify the optimal choice of plants, energy sources and

sizing of components. Anyway, the simulation model remains the kernel of the decision support tool.
For the real time control and operating, other tools based on thermo-hydraulic modelling or optimal dynamic control optimisation could be more reliable.

METHODS

The proposed model concerns a completely defined DH system, with given set of power plants and heating demand, and looks for determining the overall DH thermal balance and power flows in the DH network at each time step, under external conditions. As we will see, the model is based on a linear programming formulation. The decision variables are the heat flows in DH pipes. The constraints are of two kinds. First, thermal properties of heat plants and DH pipes will restrict the range of heat flows and second, distributed energy should satisfy consumer (substation) heating demand at any time period.

Analytical formulation

A suitable method to describe a district heating pipe system is to use concepts from network theory. We define a district heating graph G as a set of branches and nodes. A branch is assumed to be a pipe; a node could be a substation or a heating plant. Heat flow is assumed to be directed *a priori* from one node to another and the graph is oriented.

Notations

The following notations are used as subscripts

i, i'	for substations	$1 \le i \le n$
k	for heat plants	$1 \le k \le p$
j	for network branches (lines)	$1 \le j \le m$

A branch j connects two nodes, either a substation i to another substation i' or a heating plant k to a substation i. The corresponding information is provided through connectivity matrices, defined below. *Su(i)* is the set of outlet branches (successor) of node i (substation), *Pr(i)* is the set of inlet branches (predecessor) of node i ; *Su(k)* is the set of outlet branches of node k (heat plant). Heat plant nodes are assumed to be without inlet branches.

At a given time step, the following parameters are assumed to be input data

Δt	the time step length.
P_i^{apl}	heat demand in substation i.
P_k^{max}	maximum heat power of plant k.
c_k	cost of fuel used in plant k.
η_k	efficiency of plant k.
p_j^{max}	maximal heat power transferred through pipe j.
L_j	heat loss in branch j.

Unknown variables, to be determined for each time step, are the heat flows in the network

p_j heat flow entering branch j.

Heat flow leaving a branch corresponds to (p_j - L_j), accounting for heat losses from pipes. Moreover, for each substation, two heat flows may be defined

P_i^- total heat flow leaving node i $$P_i^- = \sum_{j \in Su(i)} p_j \,, \qquad (1)$$

P_i^+ total heat flow entering node i $$P_i^+ = \sum_{j\in \Pr(i)} p_j - \sum_{j\in \Pr(i)} L_j \,, \qquad (2)$$

For each substation, the heat demand from consumers must be met, so the thermal balance leads to

$$P_i^+ = P_i^- + P_i^{apl} \,, \qquad (3)$$

Principle

The proposed model to determine the state variables p_j is based on the minimization of the heat production cost of the whole district system over each time step of a work period, with respect to energy balance (both at substation and heat plants) and thermal pipe capacity constraints. While expressing the objective function and the constraints as function of heat flows, it can be seen that all equations are linear.

The problem is stated to be at each time step as follows:

$$Minimize \;\; f(p_j) = \sum_{k\in Sources} \sum_{j\in su(k)} \frac{c_k}{\eta_k} \times p_j \times \Delta t \,, \qquad (4)$$

With constraints

$$\sum_{j\in \Pr(i)} p_j - \sum_{j\in \Pr(i)} L_j = \sum_{j\in Su(i)} p_j + P_i^{apl} \,, \qquad (5)$$

$$\sum_{j\in Su(k)} p_j \le P_k^{\max} \,, \qquad (6)$$

$$0 \le p_j \le p_j^{\max} \,, \qquad (7)$$

Matrix formulation

Preliminary implementation of the model was done under MATLAB. A graph structure was designed under Matlab as a schematic network layout of the DH system topology and the LINPROG function was used for optimisation. Like many linear programming routines, LINPROG uses the standard simplex algorithm to solve LP problems and it requires data in an LP standard format which is the following:

To find out the vector X minimizing $f = {}^t c \,.\, X$ subject to $$A \,.\, X = b$$ $$A' \,.\, X \le b'$$ $$u \le X \le v$$

Thus the inputs for this function (objective function, equalities and inequalities constraints) should be presented in a matrix format. Let's remind that we consider a DH system with n consumers (substations), m pipes and p heat plants.

The cost function f is written at each time step as a scalar product of two vectors c and P, where c contains the heat production costs (cost of fuel, divided by efficiency, + other possible proportional costs) and P is the unknown vector composed from the heat flows in the DH system pipes, p_j.

$$f = {}^t c \cdot P \,, \qquad (8)$$

Calculation of energy balance at each consumer (substation) is based on the first law of Kirchoff [11]. In order to express these equations in a matrix format we may define the connectivity matrix A of the DH representative graph as follows:

$$a(i,j)=\begin{cases}1, & \text{if the pipe } j \text{ is an incoming pipe for consumer } i.\\ -1, & \text{if the pipe } j \text{ is an outgoing pipe for consumer } i.\\ 0, & \text{if it's not connected.}\end{cases}, \qquad (9)$$

The connectivity matrix is an *n by m* matrix [1]: it has one column for each branch in the system and one row for each consumer. Each column can have two non zeros entries. The connectivity matrix expresses the inflows/outflows relationship in the district heating system.

Considering the energy balances equation at each substation (eqn. 3), the equality constraints of the DH model can be easily expressed as function of heat flows as:

$$A\,.\,P = b, \qquad (10)$$

Where b is the constant term, which contains the heating demand for each node, including heat losses from inlet branches.

$$b_i = P_i^{apl} + \sum_{j\in \Pr(i)} L_j\ , \qquad (11)$$

Inequality constraints express the inability of power plants to exceed a specific heating power value. We define as well a specific matrix denoted A' to present the inequality constraints in a linear format.

$$a'(k,j)=\begin{cases}1, & \text{if pipe } j \text{ is connected to source } k.\\ 0, & \text{otherwise.}\end{cases}, \qquad (12)$$

Hence, the inequality constraints could be expressed as function of the heat flows.

$$A'\,.\,X \le b'\ , \qquad (13)$$

Where b' contains the maximal heating power of each power plant.

The LP optimization routine LINPROG is called at each time step. The function inputs can be classified into two categories, variables and static parameters according to their values over each time step (see Fig. 1). Static inputs includes A and A' and they didn't depend on time since they are related to the network configuration. Variables parameters include the cost function f as well as second term parameters b and b'.

Typical simulation can be done with an hourly time step and a running period of one year, enabling to account for combined time variations of all input data (demand, energy price, heat plant characteristics). With additional assumptions on input data (constant demand, prices, availability, efficiencies), much longer time steps can be adopted, leading to faster simulation.

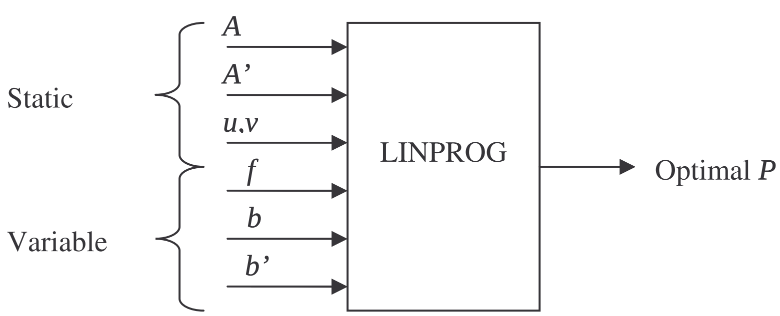

Figure 1. The optimisation routine, called at each time step

RESULTS

The aim of this section is to apply the proposed model in its hourly form to the district heating of Beaulieu Malakoff in Nantes. This district heating counts 20 km in length and 96 substations. The network provides approximately 115 GWh of energy to 16000 consumers. Incineration plants of municipal solid waste built in 1987 provides most of the heat for the network. The waste incinerator is modelled as an ordinary plant with variable maximum power and zero cost. The heat provided by an MSW incinerator is supposed to be totally free; the costs related to plant maintenance and waste transport are disregarded.
The maximum heating power of an MSW plant calculation is based on the lower heating value *LHV*(GJ/tons), the waste fuel rate *r* (tons/hour) and the boiler efficiency η thanks to this formula:

$$P_{\max} = LHV \times r \times \eta \,, \qquad (14)$$

The monthly values of LHV and waste fuel rate are used as inputs. Efficiencies are supposed to be constant and the availability of the plants is set to 100%.

The duration curve shows what proportion of heat production is attributed to each plant to satisfy the demand at any time. It may also give the start and stop moments in the heating plants.

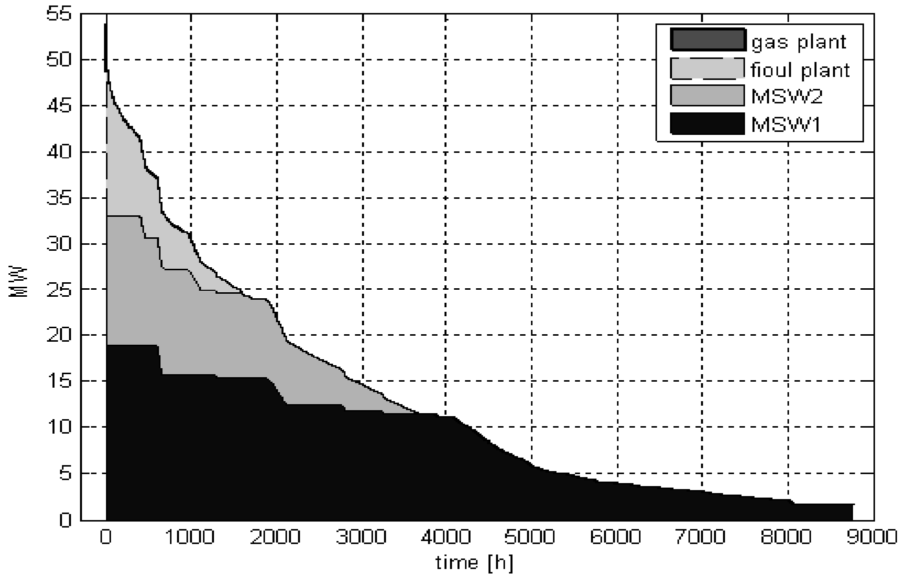

Figure 2. Load duration curve indicating the proportion of the heating plants

The duration curve as presented above shows in reality the production scheduling of the DH plants and their start and stop moments as function of heat load levels.

Table 1. Optimal heat production dispatching

Heat load range	Starting heating plant	Covering ratio
<12 MW	MSW1	67.8%
12-24 MW	MSW2	24%
24-49 MW	Fioul plant	7.9%
>49 MW	Gas plant	0.3%

Fossil basic plants are operational in peak loads; they cover the rest of the heat demand served basically by incinerators. The swap between the fossil heating plants depends obviously on the fuel cost. The model gives a priority to a boiler at a given moment if its fuel is the cheapest at that moment.

Table 2. Yearly balances (energy in GWh, costs in k€)

	Feed in primary side	Valorised heat	Heat surplus	tCO2 realised to the air	Running cost
Incinerators	235.9	105.3	95.2	60872	
Fuel boiler	10.6	9	0	2774	229.35
Gas boiler	0.02	0.015	0	4.19	

From a strategic point of view, the stakeholder could take decisions concerning heat production in the DH system in the future. For instance, assuming the same heat load for next years or using heat load forecasting techniques, the decision maker, by the aid of this model, will have an idea about the amount of energy in primary side required to cover the demand.

DISCUSSION

The mathematical model and algorithm presented in this work contributes as a first step toward the development of a fast, robust and accurate DH strategic planning tool. It can be considered as a general and flexible tool. In order to illustrate this flexibility, some examples are analysed below, considering the main features of district heating.

In term of **network characteristics**, the model is suitable whatever the topology of the network, hierarchical or looped network structure. Although the temperature of the network is not taken explicitly, DH systems with variable temperatures may be tackled. In most cases, flow and return temperatures depend on the total heat demand addressed to the network and can be computed at each time step, as an exogenous parameter (enabling to adjust input data varying with temperatures such as heat losses or heat plant characteristics). So, common network control strategies, either through fixed supply and return temperatures (and adjusted flow rate) or fixed flow rate (and adjusted supply temperature), can be considered by the model. Temporary problems taking place in the network, such as interruption or limitation of supply in a given branch, may be simulated by modifying the maximal heat power transferred through this pipe. Interconnection of a DH with other DH system(s) may also be considered easily. Pumping cost, which represents a non negligible fraction of operation costs (a few percents), can be modelled by giving a cost to heat transferred into branches where pumps are located.

Regarding the **energy demand** addressed to the network, the model may consider simplified demand profiles from consumer or very precise profiles from quite different types of consumers, depending only of the quality of available data and required accuracy. An interesting feature regarding demand is the possibility to simulate temporary and partial disconnection of some customers, using "virtual" heat plants associated to a cost of non-supply: interruption of demand is a service which can be provided by some customers to the network at a given contractual price.

Finally, the flexibility of the model concerns also **heat production plants**. A large variety of heat plant technologies are directly tackled, provided their maximal capacity and efficiency are known at each time step. This includes usual types of boilers, using fossil or renewable energy sources, whatever their cost. Waste heat, such as heat produced from domestic waste incineration plants, can be simulated, being considered as free energy source.

Availability of heat plants vs time (due to maintenance, e.g.) and time varying characteristics of the energy sources (price, lower heating value, maximum capacity) are considered through input parameters. For some heat plants, such as cogeneration, heat pumps or geothermal energy, their characteristics depend on network temperature. They may be simulated as well, as soon as this temperature is pre-determined at each time step.
The main limitation of the model is probably linked to the assumption of constant efficiency for each heat plant (considered as an input data for each time step, possibly varying with time). Efficiency depending on the load (with lower efficiency at partial load, e.g.) cannot be tackled by linear programming algorithm.

Thus, the model incorporates a lot of features, relevant for most of current district heating systems. It is being implemented as a master piece of a strategic management tool of district heating systems. In the same time, additional components are being considered for introduction in model equations, namely heat storage and complex cogeneration (including connection with electric grid).

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Effects of structural and ample composition and process activities of a substation at district heating systems on energy efficiency

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INTRODUCTION

The district heating system should be accepted as a technical system, functionally and workable to provide heat for thermo technical needs of a higher number of different and connected us ers being located on a wider area usually in urban environment. The thermo technical needs of users mean the needs for heating, ventilation, air conditioning, and preparation of consume sanitary hot water and/or for special purposes. The district heating system on defined quality level provides heating and its delivery to users. The district heating systems are complex in their contents, conditioned in structure and non-alternative in processes and they may be active during heating season or during the entire calendar year.

The DHS is projected on the bases of established and emphasized requirements in written project task, on the level of consciences and expertise of engaged designers, of material, equipment and components being present on open market; after that it is realized by contractors having references and routines and after tests, regulations, technical checks and acceptances they are given to an investor for use in accordance with culture of users. The district heating system is projected and dimensioned for needs of defined environment but also for a longer period of its development; therefore it is constantly developed, reconstructed, revitalized and maintained up to limits of technical, ecological and economic justification in lifetime.

General model of structural arrangement of the DHS, on the level of a subsystem is timely defined [8].The structure is composed of seven constitutive subsystems from No 1 to No 7, where No 4 subsystem is recognized between them, being defined for direct exchange of heat in the sense of delivery of heat to users connected actively to the DHS. Practically, the No 4 subsystem integrates all substations in the one DHS and regularly with not a low number; for example in the town of Pozarevac there are cca 300 active substations in one DHS, and in the city of Kragujevac there are cca 1500 active substations in three DHS. Indisputably there are no DHS without the No 4 subsystem but the No 4 subsystem considerably effects representative quality of the DHS as a whole [10].

SUBSTATION AT DHS

The substation is an universal time determinant that in thermo-technical has single sense and meaning. The substation in thermo-technical is understood as a space, equipment and process for exchange of heat among tube exchanger (primary carrier of heat) and collector (secondary carrier of heat) within technical and technological control. The substation is real, physical and direct limited connection of the DHS and user.

The substation space belongs to the No 7 subsystem, the primary carrier of heat to the No 2, subsystem and the primary equipment to the No 4 subsystem. The secondary equipment in substations belongs to users connected to the DHS [8].

Algorithm composition is characterized for a thermo-technical substation regardless it is about single, group or zone substation [1]. Besides others, any substation equipment is adopted to type of heat, way of heat exchange, needs of users and the DHS concept in composition, contents and capacity.

Any substation consists of a defined group entity, namely of a constitutive group to provide certain process activities [6,7,8,9,10].

Any substation has to have its own particular functional and particular work capacity, and all together makes partial functional and partial work capacity of the No 4 subsystem [10]. Indicators and parameters being featured to the k substation are presented in the figure 1.

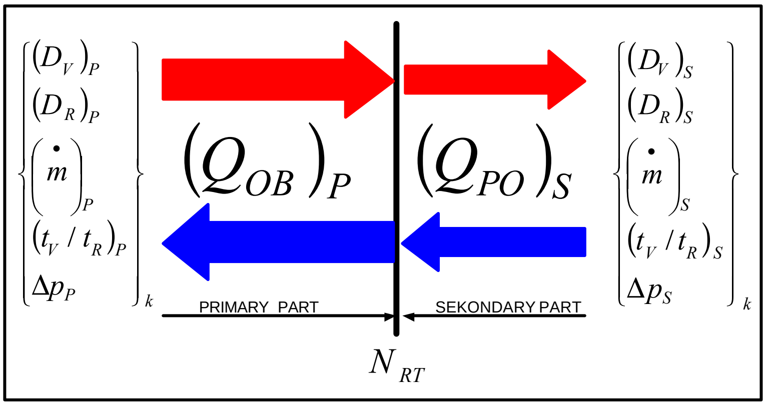

Figure 1. Indicators and parameters of autonomous substation of the DHS.

The process of heat exchanger, between the primary and the secondary heat carrier may substation, and then number, type and operating conditions of users are unchanged input parameters and quantity of heat required by users is changed input value. At any moment of exploitation period the users be recuperative done by surface heat exchanger and mixing, i.e. by elevators. In practice the first principle of heat exchange is more used.

The substation does not produce any heat but it may save some . Such saved heat has higher value than the newly produced one. The basic starting criteria for identification of substations are as follows: location of an object where the substation is located relating to the source (No 1 subsystem); disposition of the substation in the object relating to the magistrate primary pipeline; number, type, operating conditions and quantity of required heat from the subject substation; type and parameters of primary and secondary carrier of heat and way of heat exchange; presence of content components in the substation; structural component arrangement of substation; level of regularity and level of process control on activities in substation.

Users of substation of k subsystem may have status of active, passive and potentional user.

Nominal required heat power of substation has to meet the needs for heat of all users connected to that substation.

SUBSTATION AND THERMO-TECHNICAL DOUBTS

Substation is projected based on the group of reliable starting data such as data on number, type, operating conditions and required heat quantity of every user connected to the substation and for projected conditions, for example outside project heat temperature. It is tacitly considered that in that way quantitative and qualitative needs of users are provided at any moment of exploitation period. Based on final project the substation is constructed and its structural and content complete works are done. Conditionally may be accepted that structural and content composition of substation, and then number, type and operating conditions of users are unchanged input parameters and quantity of heat required by users is changed input value. At any moment of exploitation period the users of k substation require certain quantity of heat from the syste m that is provided directly through the substation. Such required quantity of heat goes from 0 to maximum, i.e. higher limited value. It is undisputedly that the heat exchanger has to provide, on the secondary side, every required heat quantity, even the maximal one. It is also undisputedly that necessary conditions for fulfill of primary task have to be realized by structural and content composition of substation. When the parity of required and provided quantity of heat is realized during the exploitation period, it is said that such substation owns particular functional capacity [10]. A serious question is made: Is it possible, on reliable and energetically effective way, to provide parity of required and delivered quantity of heat at any moment of the exploitation period by conventional, structural and content substation composition? Since it is very complex and important question, and having in mind there is no much room in this paper but to give answer to this question we further analyze the heat exchanger in the k substation only. In this analyze we start with the following assumptions: users use heat during the heating season; projected heat conditions are known for zone of users; heat exchanger is recuperative, in standard; on projected conditions the users require defined heat quantity; selection on producer, type and size of heat exchanger for projected conditions are selected; the heat exchanger has appropriate heat power; regulation of heat efficiency for heat exchanger is made on qualitative principe which means that mass flow rate of heat carriers from primary and secondary heat exchanger sides are stable.

Heat exchanger and thermotechnical doubts

In selected prospect for producer of heat exchanger it is usually mentioned data «heat power of exchanger» is presented as entire number in tens but it is not mentioned if the heat power is on primary or secondary side. Due to unavoidable disipational processes during exchange of heat in the exchanger, the unparity (NRT)P > (NRT)S is uncontested. At calculation and selection of heat exchanger it is usual to reach needed heat power which number value is not the whole number and not in tens. A projector always takes higher value, i.e. higher tens value while selecting heat exchanger. Calculation of heat exchanger is made for outside project temperature, even it is rare during the heating season, for which average outside temperature is prevailingly. Practically, heat exchanger is used during the heating season, under optimal performances. Inertia of primary side, static construction of heat exchanger and dynamism of secondary side based on required heat quantity cause absence of acceptable correlation between the required and delivered heat quantity. Practically, oscillatory unsynchronized exchange of heat is present within limited surplus and deficit state.

Changing of temperature of primary and secondary heat carriers during heating season stipulates variability of coefficient of heat flow and heat flux in the heat exchanger.

Process of exchange of heat in the heat exchanger is done through firm surfaces which they are disturbed with two heat carriers of various temperature initiating formation of deposit on such surfaces and in that way resistance to heat conduct is increased with obligatory reduce of exchange efficiency.

The primary heat carrier circulates through the whole DHS, taking in and out firm, fluid and gas contents.

A part of that content comes also to the heat exchanger, surely to the primary side, going through it or stopping there. These unwanted contents disturb circulation of the heat carrier and in that way they reduce efficiency of the exchange. Similar happens to the secondary side of the heat exchanger. The heat exchanger owns a conceived sensibility to any degradation and/or loss of system functional and/or work capacity that may directly affect the exchange efficiency [2,3].

Outside conditions may inflict needs to users for quantity of heat with even higher values than being calculated for the projected conditions, which objectively exceed real possibilities of the heat exchanger and it becomes insufficiently strong. Circumstances, conditions and function of the heat exchanger in substation point out undoubtfully that we cannot speak on acceptable and reliable energetic determinism.

SUBSTATIONS AND ENERGETIC EFFICIENCY

Heat quantity, requested by users during exploitation period, is surely indeterminist value.

Nominal heat power of the substation is a fix value during exploitation period.

Heat driving power of the substation is understood as substation heat power being equal to real exchanged heat quantity under real conditions, between the primary and secondary heat carriers, at defined time, during substation active function. We differ the driving heat power on projected work regime and on current work regime. Nominal heat power of the substation is understood as substation heat power at ideal heat exchange and it does not depend on real exchanged heat quantity between the primary and secondary heat carriers. Energetic efficiency of the substation goes from 0 to 1 and it means the relationship between the driving and regular substation heat power. During the exploitation period the energetic efficiency of the substation vary within possible values, it represents an indicator for defined k substation and it differs in values for every substation of the same district heating system. The intention is to overcome discrepancy between the requested and delivered heat quantity by constitutive group No 4.8, but it still does not mean that the substation energetic efficiency is improved. The provided heat quantity is possible to vary on three characteristic ways such as: using temperature of the primary heat carrier; changing the flow of the primary heat carrier; and changing simultaneously the temperature and flow of the primary heat carrier; but under condition that involved heat quantity is higher than the provided one, namely the requested one, since there is no sense for inverted combination. Practically, the requested heat quantity is formed from the provided one and the provided heat quantity is formed from the involved one as being presented in the figure 2.

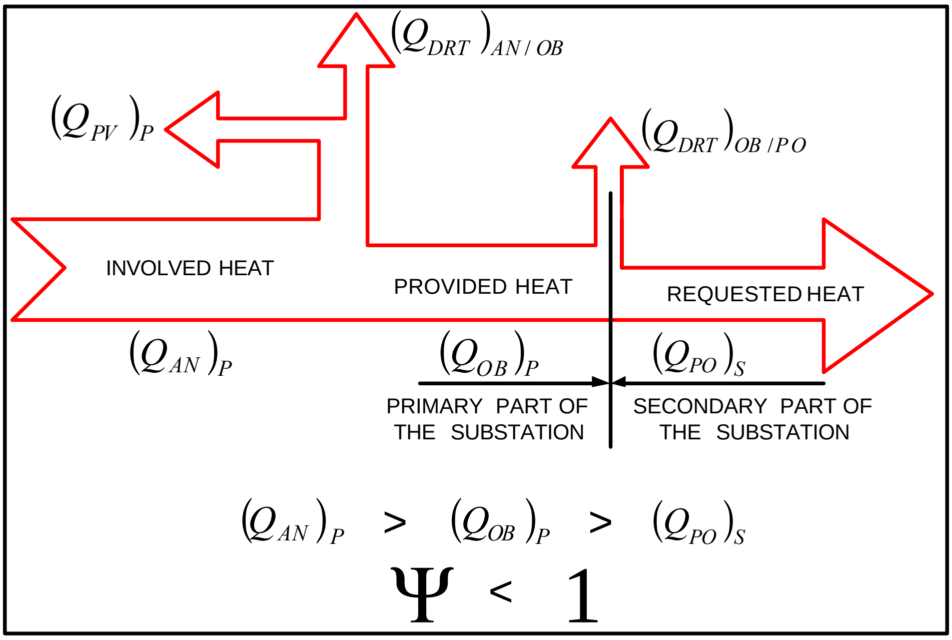

Figure 2. Diagram of heat flows in substation.

By technical-technological improvement of the structure and content composition of the substation and by improvement of the process activity it is possible to reduce the difference between the involved, provided and requested heat quantity.

SUBSTATIONS AT DHS IN THE TOWN OF POZAREVAC

The district heating system of the town of Pozarevac is unique in Serbia or wider in many specific features. The source of heat is located in thermal plant „KOSTOLAC“, where appropriate exchanging substation is constructed that practically makes the subsystem No 1, in the subject district heating system [11].

For needs in heat of places such as Kostolac, Klenovnik, Cirikovac and the town of Pozarevac, regular heat power of 315 MW is reserved. The magistrate conduit of cca 10 kilometers is made of seam steel tubes of Ø 660,4 x 7,1 mm diameter and it connects the No 1ubsystem with users from the town of Pozarevac. The primary heat carrier on the projected regime (-18 °C), is hot water of 130/75 °C temperature and with pressure of NP 16 bars. The secondary heat carrier is hot water of 90/70 °C temperature on the projected conditions. Until now for users in the town of Pozarevac, cca 150 MW of heat power has been involved, which is less than 50% of reserved regular available heat power.

The users are connected to the system through individual, group and zone substations. For now there are cca 170 of individual substations with regular heat power of 10 to 100 KW, and cca 100 of group and zone substations with regular heat power of 80 to 5300 KW. Drum or plate recuperative heat exchangers are installed into substations. The difference between regular heat power and driving heat power on projected conditions is evident of cca 30%. If it is known that average temperature for the town of Pozarevac is + 4,5 0C, than we reach a serious data that the process activity in the substations of this district heating system is performed under unfavorable energetic efficiency.

CONCLUSION

Nominal heat power of the substation is mostly increased by 30% regarding driving heat power on projected conditions, namely, increased by 60% regarding driving heat power at average heating temperature. Heat is no effectively used and the thermotechnical complex is seriously jeopardized. We cannot be satisfied with the existing condition and processes in substations of the district heating systems from the aspects of energetic efficiency. Practically, all advantages and opportunities offered by the district heating systems are not used, namely they are disgraced, according to traditional structural and content composition and process activities in existing substations. Samples of such conditions are numerous, and known, thus the following should be done to improve the conditions and eliminate the sequent: work on and application of standards and regulations in the field of district heating systems; improvement of existing and development of new components, apparatus and equipment used for substations and district heating systems; wean of projectors out of dangerous habits and procedures, based on their inertial brains; training of construction workers in the sense of development of new capabilities; development of culture of users; care of „heating“ cult, as a precondition to keep the development of thermo-technical complex and urgently, with results and global reconstructive improvement of the existing district heating system.

Special attention should be paid to innovation of the heat exchanger in the sense of its „revival“.

Energetic efficiency as a criterion parameter is very grateful in analyses as also being confirmed in this occasion.

ABBREVIATIONS, DESIGNATIONS, SYMBOLS AND INDEXES

DHS	district heating system
N (KW)	heat power
Ψ (-)	energetic efficiency of substation
No ...	numbering of subsystem or constitutive group
Q (W)	heat quantity
q (W/m²)	heat flux
R (m²K/W)	eat resistance in heat duct
A (m²)	Surface of heat exchange
k (W/m²K)	Coefficient of heat duct
c (KJ/kgK)	Specific heat capacity
t (°C)	Temperature
ΔT (K)	Temperature difference
t(s)	time
$\dot{m}$ (kg/s)	Mass flow of heat carrier
D (mm)	Tube diameter
Δ p (bar)	Dissipation pressure outlay

Indexes

P	Primary
S	Secondary
HE	Heat exchanger
L	Logarithm
DHO	Dissipation heat outlay
Z	Dirtiness
HD	Heat duct
e	external
I	internal
OPT	Outside projected temperature
H	heating
AV	average
IN	involved
PR	provided
RE	required
RT	return

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PREA Promoting Renewable Energies in Africa

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SUMMARY

PREA is a joint project between four European Universities and three African Universities as well as the International Solar Energy Society (ISES), an international NGO, that promotes renewable energy. The aim of PREA is to reduce poverty by influencing energy policy and regulations in Africa, through training and capacity-building of energy professionals, regulators, academics as well as policy- and decision-makers to enhance their skills in implementing Renewable Energy Technologies (RETs) and Energy Efficiency (EE) in buildings in Africa starting with three countries, namely South Africa, Tanzania and Uganda. Training will be offered through Workshops and a Masters Degree Course in Integration of Renewable Energy in Buildings (IREB), both made to suit local requirements in climate, economic and cultural conditions and allowing for exchange and further training opportunities.

INTRODUCTION

Africa is the only continent on this planet that sits squarely on the equator and where both the tropics of Cancer and Capricorn pass through its land mass. Thus Africa is mostly tropical but local microclimate in various parts it are modified by other factors such as relief and proximity to natural features such as mountain ranges and big water masses. Africa boasts of different forms of natural vegetation raging from dense tropical rain forests through grasslands to scrubland and deserts. The biggest desert in Africa is the Sahara. All these features give Africa a unique position in potential on the world renewable energy map.
It has 95% of the world's best sunshine and a huge potential of hydro, wind, bio and geothermal energies.

However, despite the fact that on one hand the African continent is endowed with vast resources of renewable energies (RE), but on the other hand, the energy situation in most African countries is desperate.

For example in Uganda and Tanzania only 10% of the total population has access to electricity at all, the main energy source is biomass, usually in the form of firewood, or charcoal. For the few urban and very few rural areas that are connected to electricity grid, supply failure resulting in blackouts and brownouts is a frequent and increasingly regular phenomenon.

There is, therefore, an urgent need to address RE and energy efficiency (EE) in order to improve the energy situation in an environmentally friendly and sustainable way.

One starting point to implementing RE and EE strategies in Africa is to focus on energy use in buildings, as the built environment contributes significantly to energy waste and pollution.

AFRICAN - EUROPEAN PARTNERSHIP

PREA is a joint project between four European Universities (London Metropolitan University, UK; University of La Rochelle, France; National and Kapodestrian University of Athens, Greece; University of Dortmund) and three African Universities (University of Dar es Salaam in Tanzania, Uganda Martyrs University in Uganda, Witwatersrand University in South Africa), as well as the International Solar Energy Society (ISES), an international NGO, that promotes renewable energy. It aims at a joint development and implementation of coordinated Masters' degree courses in this field that initializes the formation of a network of Southern African Higher Learning Institutions and links them with an already existing European network (TAREB) in which Dortmund University already participates together with six other European Higher Learning Institutions, three of which are also named above as participating in the PREA project. The Masters' courses are to be preceded by, and later to run parallel with a series of Workshops, in order to sensitize African government policy makers, decision makers and implementers, regulatory agencies and senior members of academic institutions about energy efficiency (EE) and application of renewable energy technologies (RETs) in buildings, as a way of fighting poverty and saving the environment at the same time.

This project is intended to run for three years during which time it is expected to have established permanent structures in form of Masters' courses at the three African universities, which are intended to continue even after expiry of the project period.

WORKSHOPS

Together with establishment of Masters' courses at each of the three African universities, organization of sensitization Workshops is one of the major activities in the PREA project. Three Workshops have so far already been conducted in the three African countries, namely South Africa, Tanzania and Uganda. They were all conducted in October 2006 with short intervals in between. All workshops had a common title, **"Sustainable and Energy efficient Building in Africa"** but were further differently subtitled to reflect areas of local focus which are slightly different from country to country among the three participating African countries.

The Workshop in South Africa took place at the Midrand near Johannesburg on 3-4 October 2006. Its subtitle was **"DMEs Energy Efficiency and Renewable Energy Targets: Addressing South Africa's Energy Crisis through Built Environment Interventions",** where DME stands for Department of Minerals and Energy. The particular local situation in South Africa addressed at this Workshop was the fact that the country did not produce enough electricity to meet the growing demand, giving rise to what is locally termed as an "energy crisis". That is why this phrase was used in the Workshop subtitle. The workshop which was supported by the Development Bank of South Africa (DBSA) was attended by 150 participants.

The Workshop in Tanzania, subtitled **"Addressing Tanzania's Energy Crisis through Design and Settlement Development"** was organized at the Landmark Hotel in Dar es

Salaam on 10-11 October 2006. It was attended by 32 official participants from various government offices and professional organizations and individual professionals. In addition 10 students also attended the Workshop where 16 papers were presented 6 of which were from project partners and 10 from other attendees.

In Uganda, the workshop was subtitled **"Promoting Sustainable & Energy Efficient Urban & Building Design practices in Uganda"**. It was organized on 13-14 October 2006 at the Hotel Africana in Kampala and was attended by 35 participants who included government officials mostly policy and decision makers, professionals including Engineers and Architects, university academic staff members and some students. Eleven papers were presented, of which six came from project partners and five were presented by government officials and professionals. The Ugandan Ministry of Energy and Uganda society of Architects were also represented and their papers were presented.

The critical situation of electricity supply and the need for new and sustainable energy resources has been underlined by the long almost daily power cuts, especially in Tanzania and Uganda in the last few years. Coincidentally, as if to underline the fact for the Workshoppers, long duration power failures occurred during the course of the events in both Tanzania and Uganda. Water shortages in Dar es Salaam were also indicative of this power problem.

As a form of side activities, some exhibitions served at these workshops made by manufacturers of sustainable energy and building products as well as service providers in the sustainable buildings industry, such as sales representatives and installers of renewable energy products, e.g. PV modules, Solar water heaters etc. One of the other highlights of the Workshops is that they served to bring together all parties involved and interested in renewable energy activities such as businesses and respective departments and individuals in local academic institutions. Certificates were issued to Workshop participants. These certificates had some extra value because in some cases such as Uganda and South Africa they could be used to gain some professional development points which is a requirement of some professional bodies from their members in those countries.

Another series of workshops is scheduled for later this year (September/October 2007), after which a project meeting will be organized in Uganda to evaluate the milestones reached in the implementation of the project and to asses its impact(s).

MASTERS DEGREE COURSES

Masters courses are to be introduced at the three African Universities in the PREA project for capacity building in education and training and to promote sustainability concepts in the design, construction and occupancy of buildings. The long term target is to train academicians for more research and further propagation of these ideas and concepts in subsequent courses even after the end of the project's scheduled time of three years. The aim is to eventually spread these ideas and concepts throughout the entire continent, by cooperation of the three African Universities and by networking with other African institutions engaged in this area.

The masters courses are supposed to use the expertise gained on a similar project in Europe called TAREB (Teaching about Renewable Energy in Buildings) but will be tailored to suit the local environment and to reflect specific demands of the country in which they are offered and taught as well as the technologies that can be easily made available there. The Masters programs will generally have some compulsory core modules and optional specialist modules some of which will be tailored to reflect local demands.

At Uganda Martyrs University, the Masters' course is planned to be introduced in phases step by step. According to Mark Olweny, the assistant Dean of Faculty of Building and Technology, who is also the local PREA project coordinator there, the project would be phased in, in two steps starting with a Graduate Diploma in Environmental Design to run either as a one year full-time course or as a two year part-time course. The part-time program, 50% of which can be taken in form of off-campus modules, is to be aimed at applicants possessing the equivalent of the basic three-year undergraduate program currently run by the same University as Bachelor of Science in Building Design and Technology (B.Sc. BDT). The second phase will be the actual Masters program will be called Master of Environmental Design (M.Sc. ED). It will consist of specialist modules and will be aimed at professionals who have either completed the full five years Bachelor of Architecture course or have upgraded their basic three year course with the Graduate Diploma. Some people with other professional qualifications e.g. in Engineering, Urban Design or Quantity Surveying will also be eligible to apply directly for the one year full-time Masters. Basic concepts in environmental design, will already have been introduced at undergraduate level, will develop students' interest in this area and serve as a "catchment area" for students and professionals.

The new Masters course at Witwatersrand University (WITS) will aim at both students and professionals. According to plans already under way at WITS, the Masters course will be associated with four separate postgraduate activities namely organization of short open certificate courses and modules in collaboration with other institutions such as Stellenbosch University, establishing new "Continuing Professional Development" (CPD) courses for established professionals, incorporation of Energy Efficiency and Renewable Energy research into existing Masters and PhD work by research and thesis, as well as introduction of taught modules into Bachelor of Architectural Sciences (BAS(HONS)). There will be two masters versions namely the Professional Masters of Architecture (M.ARCH(PROF)) and the Master of Architecture specializing in Housing (M.ARCH(HOUS)). As an unexpected opportunity the PREA project coordinator at WITS, Daniel Irurah, was requested to develop a teaching module on Renewable Energy, in the process of establishing of a new Master of Philosophy (M.Phil) on Renewable Energy due to start at Stellenbosch University later this year (July 2007). PREA has been identified as one of the key strengths of WITS in its collaboration efforts with other institutions in South Africa.

Dar es Salaam University is also quite ready to establish the new Master course. It has all the necessary manpower and teaching facilities for the course to be able to take off this year. Existing departments which are ready to collaborate in establishing the new Masters course include the Department of Architecture in the Faculty of Architecture and Planning (FAP), the Faculty of Civil Engineering and the Built Environment, the Department of Energy in the Faculty of Mechanical Engineering and Chemical Engineering and the Department of Electrical Power in the Faculty of Electrical Engineering and Information Technology.

RESULTS / DISCUSSION

Although only one year of the three-year-project has passed there could be improved already a few things in Africa that would not have happened without PREA:

- Three universities in sub-Saharan Africa have decided to implement masters courses in the area of renewable energies energy and efficient building during the project duration.
- Three workshops about sustainable energy supply and about low cost and high comfort buildings have been carried out with active participation of key actors from the three African countries.
- The network of African institutions working in the areas of energy and building could be improved , last not least by the website www.ises.org/PREA.

Thus the PREA Project has already proved to be an important event in the development of energy consciousness in Africa. Both the Workshops and master courses are developing satisfactorily. Inference from the so far already conducted Workshops is very good. From responses to questionnaires given to both participants and organizers it is indicated that people concerned are very satisfied with the stages and milestones that have been reached so far. Project websites (shown hereunder) have been established, Workshop handbooks with all the papers presented at Workshops have been published and distributed to workshop participants and other interested parties, a CD summarizing all activities has been developed and as requested by the European Commission, a PowerPoint presentation containing "publishable summary slides" has been produced and updated. The most important fact however is that through this PREA project, the issue of energy efficiency and renewable energy in buildings in Africa has obtained a forum through which it will be more specifically and efficiently addressed within an integrated building design and construction approach. Moreover, African universities have had a unique opportunity at South–South collaboration among each other and South-North collaboration with their European counterparts.

CONCLUSIONS

The PREA project, although scheduled to run for three years, is meant to have a long lasting impact in the development of a new energy consciousness in Africa. It has started with the Workshops that are aimed at sensitizing African governments' officials, policy makers, decision makers and implementers as well as regulatory agencies about the importance of energy efficiency and application of renewable energy technologies in buildings as a way of fighting poverty and at the same time preserving the environment for posterity. In short the PREA project is a catalyst for sustainable development and poverty eradication in Africa. It will help Africa achieve some of the millennium development goals sooner rather than later. In the first project year it has become evident that there is interest in capacity building in postgraduate university education and a strong demand for it. The implementation of the masters courses at the three African universities has started a sustainable development.

ACKNOWLEDGEMENT

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Websites:

Some websites associated with the PREA projects have been established by ISES and some of the other project participant universities. Some aspects of PREA have also been published by other independent publishers as well. Following hereunder is a short list of websites and on-line publications about PREA.

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http://www.ises.org/PREA

http://hermes.wits.ac.za/www/Conferences/PREA-WITS

http://grbes.phys.uoa.gr/prea/index.htm

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11 June 2007 at 10:00 - 11:30

C01
Life cycle services and commissioning

Life Cycle Models In Building Service Technology

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ABSTRACT

Customer-focused life cycle building service technologies are changing the operations and networks of companies. New requirements and value-creating mechanisms fall extensively in the various fields of business and in different stages of projects. Effective utilization of life cycle models requires new know-how and the ability to verify the added value from both the client and provider alike. Output-based procurement methods affect the risk transfer and the combination of tasks in a new way. Competitive procurement procedures and output-based long-term contracts bring about new options. Additionally, international consideration has broadened the views on development. The five life cycle service models developed within the *Life Cycle Models in Indoor Environment and Building Services* project (CUBENet) are based on an options selection framework, which also provides a platform for further development of these models. Tools supporting the implementation of the models promote their usage.

INTRODUCTION

In the modern world of fast communications, ideas travel rapidly around the world, and so do the concepts influencing the development of life cycle models in building service technology. For the present, the implementation of life cycle models is new everywhere and projects implemented are numbered. Consequently, the wider introduction of life cycle models depends on local conditions, such as how the usually traditional construction business is locally in favour of employing life cycle analyses, the demand for renovation and repairs of the building stock and, if in connection with this reproduction, the requirement of durability and sustainability may be highlighted. Furthermore, the high energy consumption of buildings or uncertainty as to their energy supply, as well as governmental regulations, might promote the use of life cycle models. For various reasons, in some countries, most of the buildings need repairs, for example, in areas rebuilt after World War II. Apart from China and some similar exceptions, deceleration in the volume of new construction and expansion in the building renovation sectors is a global phenomenon.

The idea of using life cycle models has been adopted into building construction from infraconstruction. Large-scale infraprojects demand large investments. Organizing such investment has induced the transfer into life cycle models. The leading idea is to outsource the implementation of municipal or national projects and the related operational services, i.e., to transfer the total responsibility for providing them to private actors. The conventions have spread from North America throughout the British Commonwealth of Nations and further on to other countries, to Europe in particular. Outsourcing the needs of municipal technical services renders the life cycle model of thinking more common. Life cycle models have been utilized in developing new more efficient contract forms to tighten the project schedules in order to shorten the lead-time and to accelerate the cycle of tied capital, not least in project development in Finnish construction [1] and [2].

Until now, the service tasks related to building services and energy technology maintenance have formed a part of the property and facilities management operations that have quickly been embedded as part of the building management and maintenance business and been adopted by the business widely. These concepts have spread from North America to the English-speaking world and to Europe starting from Great Britain through the Netherlands [3]. The requirements set by sustainable development have turned the attention of the actors in the construction business to the operating costs of the building and the ecological standards of the whole life cycle of buildings. Internationally, building maintenance is seen as the most important factor in the economy of a building. Therefore, energy and building services businesses have woken up to the need of independent life cycle models in building service technology and to the need of offering for sale new progressive services together with hardware supply. Finland, together with Sweden and Great Britain is one of the foremost countries in developing these new service models in Europe.

Above all, common rules and feasible functional technical solutions improving the business prerequisites are sought for the life cycle models. However, appropriate national incentives for energy efficiency have taken an interest in developing support forms or favourable financial arrangements for the life cycle service projects. Frequently, life cycle repairs can start with energy audits that are a well-established practice throughout the developed world. National support obtainable for them can vary between 20, 50, and even 100 percent [4]. Life cycle models can yield significant benefit in the operation costs of building services depending on the standard of energy-efficient or sustainable technology employed during the design and implementation phases or during repairs to achieve an efficient building performance life cycle. The total economical savings yielded can be used either in funding the construction or, as is usual, in funding the repairs and improvements of the building. However, this is still quite rare internationally as the projects implemented have external investment funding.

METHODS

The Tekes (the National Technology and Innovation Agency) CUBE technology programme project CUBENet (the *Life Cycle Models in Indoor Environment and Building Services* project*)* comprised several work packages: procurement, measurement and verification (M&V), risk management, contract documents, business models.

Each of the work packages has used scientific methods suitable for the demand of the tasks. Typically, several case studies have been carried out, particularly in testing the M&V methods in existing building stock, in the development of the risk assessment tool and the CUBENet life cycle cost calculation method, as well as in the generation of the contract model templates. Most of the work has been planned and verified during the sessions of two working groups: the first one for the building service management and the second focusing on the public works and the energy performance contract (ESPC, energy saving partnership ESP) performed by the energy services companies (ESCO).

Various building service technology models developed within the CUBENet project are, and will be, discussed and referred to in scientific literature; the implementation tool box for this theory of models is described in other papers [5].

LIFE CYCLE BUSINESS MODELS

In several industrial fields, product-related life cycle services are becoming increasingly common. The main idea in the development of the life cycle services is that the product manufacturers take further responsibility of the functionality of their product during its life cycle in their client's use. Respectively, the client uses life cycle services in pursuing cost-effectiveness, easier maintenance or better functional quality of their own processes. Life-cycle services may be aimed at the entire product life cycle or at a specific stage according to the situation and as agreed. Indeed, business-to-business (B2B) service business aims at long-term partnership with the client.

Implementing life cycle service models changes the interfaces and processes between companies, sometimes significantly. Services offered by providers replace some of the client's processes or support processes. Cooperation between organizations is facing new challenges and requires new kinds of methods and supportive tools to deal with them. The objective is to build a network-wide value creation process in which the provider not only knows the client's processes but, by using their own expertise, are also able to make them more productive in a cost-effective way. This may indeed be described as life cycle wide management of products and services all the way from product development to after-sales services and discarding of the product.

Similar change is apparent in the real estate business. In the past few years, the implemented projects have also included supporting operational services, such as maintenance, cleaning, and ensuring the energy savings in the building and in the systems. The services take shape case-specifically on the basis of the client's needs, while the contents of the contracts are tailored to the project in question. Insight concerning what should and can be included in the services is developing. Thus, it is now possible to more efficiently develop both the life cycle service user's ability to take advantage of the services, and the service provider's processes and networks to provide cost-efficient services. This way, verifiable added value can be created for the both parties, which is a prerequisite for more extensive implementation of life cycle services in the field of building service technology also.

BASICS OF LIFE CYCLE MODEL

Life cycle models of building service technology refer the procurement method, in which the chosen service provider is responsible for planning, building and maintenance of a building service system for an agreed period of time. A wider scope of obligations compared to traditional form for procurement, and the responsibility of the functionality of the systems, require that the service provider takes into account the life cycle economy and indoor air conditions of the building services.

A very wide variety of method types fall under the concept of life cycle models, depending on the service entity offered and the way the provision of a service is organized. Procurement of building services with life cycle models differs from traditional procurement methods in various aspects [6]. To begin with, including the responsibility for planning, building, and building services in one contract offers better possibilities when striving towards cost-effective solutions and services [7]. These effective mechanisms can be described as the common effect of the following basic solutions [8,9]:

- **Combining tasks.** In traditional methods, the client uses various contracts in obtaining the building services during the building's lifespan. Then, it is difficult to make the objectives of all parties meet, and often the aims of an individual party override the potential overall best for the project and the customer. By combining these tasks, service providers have a chance to remove operational borders between different actors and to develop the service entity in several different ways in the long term.
- **Goal-oriented thinking**. By defining the client's requirements and focusing on the functionality of the building services, the service providers obtain greater possibilities in choosing solutions and preparing service concepts than with traditional methods in which the client defines the technical requirements and solutions. This approach enables and encourages the service provider to provide the services in an economical way for the life cycle and to develop service-related innovations. The output specification gives the service providers increased flexibility to innovate in developing their service provision solutions for the project. This way, the client's goals become the leading idea driving the implementation of the entire project
- **Procurement method**. Procurement methods are used to ensure a good price-quality ratio for the building service acquired as the client may choose from various offers and select the best one. By using competition, it is possible to utilize a significant amount of the potential that is based on combining the building services in planning, construction and use of the building and that has been made possible by, for example, concentrating on determining the requirements at the service level.
- **Risk transfer**. In the life cycle model, the client transfers several risks to the chosen service provider. Some of the risks are shared between the client and the service provider [10]. In principle, the risk lies with the party with the best prerequisites to handle it. With appropriate risk transfer, the client creates incentives for the service provider in developing technical solutions and services.
- **Quality- and output-based service charges**. In life cycle models, the service provider's payment bases are usually not totally fixed, as the sum depends in part on the operational service level. The use of output-based specifications ensures that the client pays only for the provision of a pre-defined service. In addition, these mechanisms create incentives for the service provider to minimize the life cycle costs.
- **Long-term contract**. Mainly, the contractual relationships in life cycle models last for several years. A long-term contract enables the risk transfer and the functionality of the service charges. Moreover, long-term contractual relationships are used to seek a service provider that provides service and maintenance of building systems and is offered a chance to develop building services and integrate the data yielded in planning and building the systems into the operations in the operational stage.

Life cycle services broaden and lengthen the responsibility of the service provider as regards the functionality and maintenance of the target. For planning business also, this offers a new perspective. Life cycle services influence the business relationships between companies. The responsibility for the operational functionality of building technical systems and services forces each party to assess also from the economical viewpoint those factors and risks that are related in depth in the tendering stage when pricing the life cycle service.

DEVELOPED BUILDING SERVICE LIFE CYCLE MODELS

Various life cycle models for building service technology have been developed in the *Life Cycle Models in Indoor Environment and Building Services* project (CUBENet). The models were selected using a so-called option selection framework of thinking in which the construction project is divided into three subsets based on the size and scope of the project, the service provider's project tasks, and the payment base. These subsets are independent of each other, but by combining them, other models can be formed alongside the main models defined. Figure 1 displays the parts of the option selection framework.

The service provider can assume responsibility for planning and implementing the entire construction project or for a defined part only, such as building service planning and building service works. Further, operations within the scope of the project can be targeted to only one building or to several. The service provider's obligation may involve performing one or several tasks needed in implementing the construction project (for more details, see Figure 1). Different tasks may be assigned to the service provider with a different payments base. The payment base may be so-called common grounds, which include fixed price, unit price, invoicing, or target price. In addition to these, the service provider may be paid a service charge. In this case, the reimbursement paid to the service provider is paid partially or entirely during the use of the building. The payment base may also be combined so that, for some of the tasks, the grounds that apply are different from those that apply to others; for instance, design costs are paid according to a fixed price but building technical work is paid on the basis of an invoice, while building technical work is paid for on the basis of service charges.

On the basis of the option selection framework, five different life cycle models for building service were developed.

1. Full responsibility building life cycle service
2. Full responsibility supply of building service technology
3. Life cycle service of building service technology renovation
4. Building service technology renovation with competitive dialogue
5. Savings-funded building service technology renovation

In the *Full responsibility building life cycle service* model, the task of the service provider is to design the entire construction target and to take care of the construction and building technical works. Additionally, the service provider assumes responsibility for building technical and real estate services for an agreed period of time. The client will reimburse the service provider for the design and construction costs according to the general payment base of contract performance, whereas for the services, the client pays monthly service charges during the contract period.

In the *Full responsibility supply of building service technology* model, the task of the service provider is to design and construct the building service technology system and to provide service and maintenance for the system for an agreed period of time. The client sets requirements for the system that may concern a system feature (e.g. efficiency) or the service level (indoor air or energy service). Furthermore, the service provider assumes responsibility for providing the service agreed upon. For the entire project, the payment base may be a service charge that is paid during the contract period. In this case, the service provider assumes responsibility for the investment costs required by the project. The payment base of the investment costs may also be the so-called common base for contract work. Additionally, costs for services provided are covered by service charges.

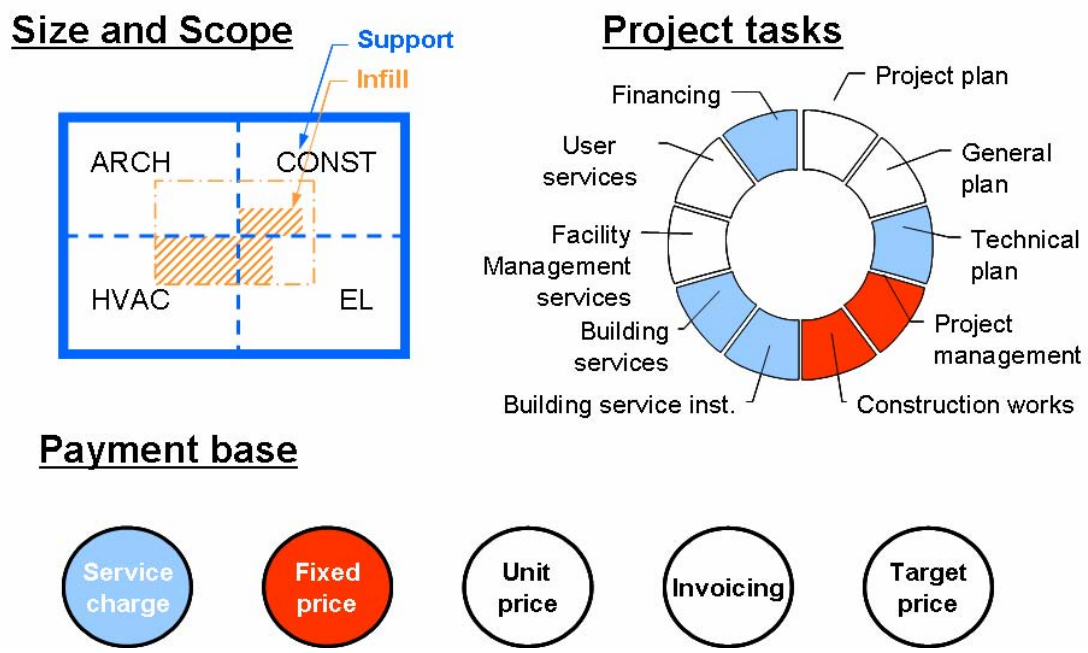

Figure 1 Option selection framework with the choices of model 5.

In the *Life cycle service of building service technology renovation* model, the objective of the project is to renovate old building and construction technology, and, at the same time, to improve the indoor air quality and conditions, and efficiency of energy use. In this model, the client submits a tender with preliminary sketches and system requirements and the service providers offer a target price. The client and the service provider guide the end design and choose the systems in cooperation. The service provider is responsible for some of the building services, and the client is responsible for the rest. The building technical services the service provider caters for are related to the end product, such as indoor air quality. The service provider may also provide some of the real estate services. For the services, the service provider is reimbursed on the basis of service charges.

In the *Building service technology renovation with competitive dialogue* model, the client builds an ensemble of several buildings, for which the conditions or energy efficiency is improved. Since the ensemble may contain buildings of varying conditions, tasks and jobs may vary with the building. In addition to the aforementioned works and services, the measures may be targeted to real estate and user services. Feasibility studies and preliminary engineering is implemented in cooperation with the client and the chosen service providers using competitive dialogue.

In the *Savings-funded building service technology renovation* model, the project is started when the client verifies or assesses the energy savings potential of the building service system. With their own measures, e.g. inspections, the client determines the savings potential and building technical systems to be renewed. In addition to those, the client can define other, non-saving, measures for the same project. The costs caused by those are reimbursed to the service provider in the normal manner during the contract performance. The service levels or periods of utilization are not changed essentially, but are maintained throughout the contract period. The costs incurred by the project may be covered in total or mainly by the savings from energy costs accumulated by the service during the contract period, verified in a manner agreed upon. The service provider takes care of the funding needed for the implementation, while the client pays the implementer a service charge based on the energy savings.

Project task	Model 1	Model 2	Model 3	Model 4	Model 5
Project planning	○	○	○	○●●●●●	○
General planning	●	○	○	○●●●●●	○
Technical planning	●	●	◐	●●●●●	●
Construction technical work	●	○	●	●	●
Building technical work	●	●	●	●	●
Building system service(s)	●	●	●○	●	●
Facility management services	●	○●	●○	○●	●○
User service(s)	●○	○●	○	○●	○
Funding	○	○	○	○	●
Ownership	○	○	○	○	○

Figure 2 Project tasks and responsibilities in life cycle models (○ = client, ● = service provider).

TOOLS OF LIFE CYCLE MODELS

Various tools, as well as contract and project models, are needed to implement the life cycle models in order to set and reach goals, compare life cycle economy, and identify and manage risks. Tools of life cycle models are presented in the article called *Tools for life cycle models in building service technology*.

DISCUSSION

The CUBENet project has acted as an ice breaker in the field of building services. It has introduced the life cycle paradigm to the experts in the field in a very serviceable manner. Strong co-work between the scientists and representatives of Finnish companies and municipalities has taken place. It has not only taken into consideration the needs of experts in building service technology, but also communicated the needs of co-operation to building managers, both in the private sector and in the municipalities. Furthermore, considering the whole life cycle of the building, co-operation between other sectors in construction, from the design team to the property and facilities management operators, is essential.

The option selection framework, used to create five life cycle models for building services, was created. The framework can be used to create appropriate combinations, depending on the size and scope of the project, the service provider’s tasks and responsibilities in a project, and the payment base. Supporting tools to help implementation of these models are presented in the *Tools For Life Cycle Models In Building Services Technology* paper [5].

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Tools For Life Cycle Models In Building Services Technology

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ABSTRACT

In construction and building services technology, life cycle models are perceived as an increasingly important way of implementing projects, and are gaining in popularity. Implementing and applying life cycle models requires supportive methods, measures, and tools. Above all, tools as well as contract and project models are needed to implement life cycle models in order to set and reach contract targets, to compare life cycle economy, and to identify and manage risks. The Finnish *Life Cycle Models in Indoor Environment and Building Services* project (CUBENet) defined the service level descriptions for setting targeted contract measurements and generated new verification methods, such as an operational verification method to be used by a building automation system. Risk assessment tables for life cycle building service schemes have been drawn up to avoid problems and conflicts; the prepared model document templates are to serve in planning the project content and form, as well as in the agreement of the contract stipulation.

INTRODUCTION

Products and systems delivery-related life cycle services are becoming increasingly common in various fields of business. In other words, in various fields, the new business concept of life cycle services are gaining in popularity as a means of establishing new business models for the concepts that are of concern. The prime idea in the development of life cycle service models is the fact that the responsibility of product providers for the functionality and performance of their products during its life cycle is increasing. The service provider guarantees the operation of the product beyond the traditional guarantee period and takes care of the product during the performance period on behalf of the client in the client's premises. Accordingly, the client uses life cycle services in striving for cost-effectiveness in their inter-company processes and management, as well as in striving to achieve lower maintenance resource allocations and better quality in their operative systems.

The trend described above can also be seen in real estate business and building services technology. In the past few years, projects implemented have also included operational services, mostly in maintenance and cleaning, but also in ensuring energy savings in the building and systems. Often, these services take shape based on the client's needs, while the contents of the contracts are tailored to the set targets of the project in hand. Currently, ideas are developing as to what should and can be included in the services. In order to expand the implementation of life cycle models and procedures, uniform principles, methods, measures, and supportive tools need to be created in the field. In this way, it is possible to develop more efficiently the life cycle service user's ability to effectively use the services, as well as to develop the provider's processes and networks in order to be able to provide services cost-effectively. The CUBENet project has included the processes of developing methods and tools to support the practical implementation of life cycle models.

LIFE CYCLE MODELS

Life cycle models of building service technology refer to procedures for obtaining systems of, and services related to, building service technology, in which the service provider chosen is responsible for planning, building, servicing and maintaining the building service technology systems for an agreed contract period. The wider scope of these tasks compared to traditional contracts, and the responsibility of the functionality of the systems, require that the service provider take into account the life cycle economy and indoor environmental conditions of the building service technical systems. The challenging purchase method, output-based incentives, and charges based on the service level, are the most essential methods for the implementation of the models.
Various tools, as well as contract and project models, are needed to implement the life cycle models in order to set and reach goals, compare life cycle economy, and identify and manage risks.

METHODS

The Tekes (the National Technology and Innovation Agency) CUBE technology programme project CUBENet (the *Life Cycle Models in Indoor Environment and Building Services* project*)* comprised several work packages: procurement, measurement and verification (M&V), risk management, contract documents and business models.

Each of the work packages has used scientific methods suitable for the demand of the tasks. Typically, several case studies have been carried out, particularly in testing the M&V methods in existing building stock, in the development of the risk assessment tool and the CUBENet life cycle cost calculation method, as well as in the generation of the contract model templates. Most of the work has been planned and verified during the sessions of the two working groups: the first one for the building service management and the second focusing on the public works and the energy performance contract (ESPC, energy saving partnership ESP) performed by the energy services companies (ESCO). Various building service technology models developed within the CUBENet project are, and will be, discussed and referred to in the scientific literature; this work on the tool box for the implementation of the models draws much from the work described in other papers [1].

SERVICE LEVEL DESCRIPTIONS

In order to set targets, service level descriptions have been created for building services technology services that are also presented as an independent part of facility services. Traditionally, maintenance tasks, which comprise technical maintenance and repairs, in building service technology have been perceived as part of property management. In future, new kinds of tasks will be numbered among the building service technology services that meet the life cycle demands.

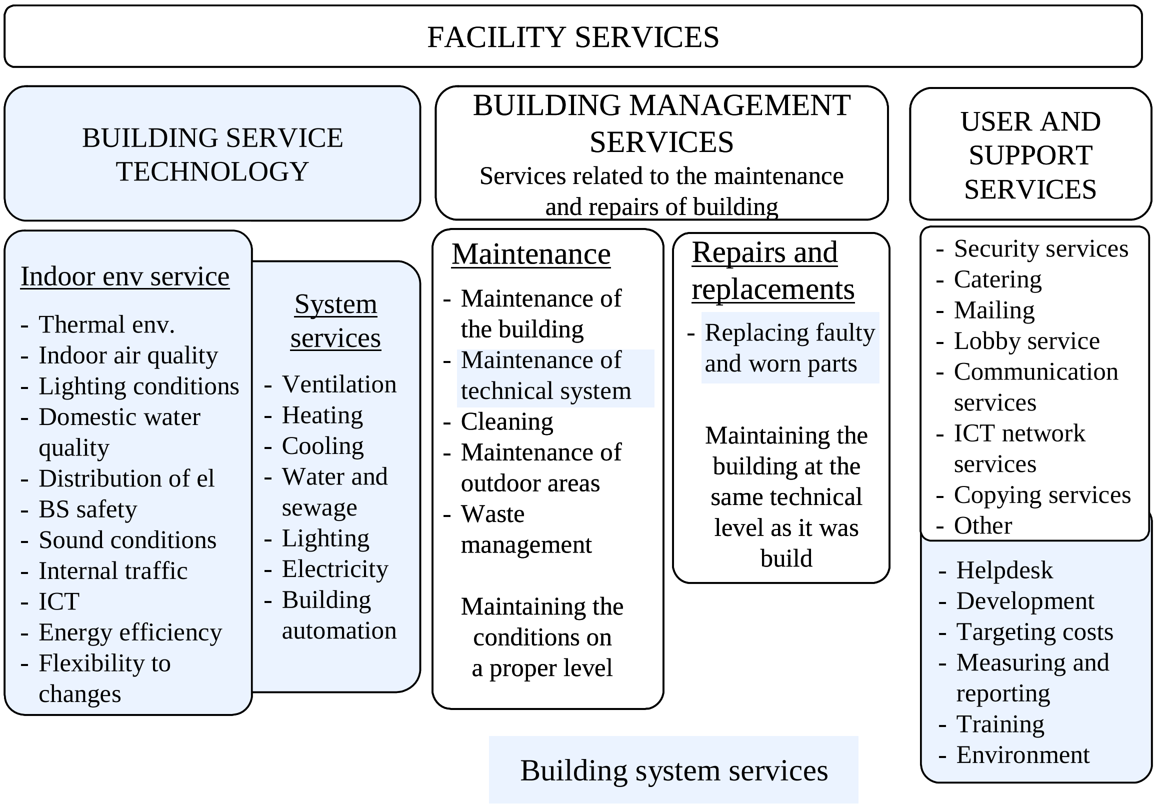

Figure 1 Building system services as a part of facility services.

In life cycle contracts, services should be presented as measurable quantities in order to be able to clearly log the tasks and obligations of, as well as the benefits yielded by, the tasks of the cooperation partners. These service level definitions can also be documented in room and system cards.

The services have been defined either on the basis of the quality of the end result or on the basis of the task by describing the operations or task to be performed. The quality of the end result can refer to, for instance, summertime room temperature with a defined acceptable range and routine to measure it. A task-based service level description is an alternative to the end result description, as it refers to, for example, the cooling power provided in the room by air conditioning. However, in this case, the service provider assumes no responsibility for room temperature.

RISK MANAGEMENT

Methods in risk sharing and assessment have been developed for risk recognition and management. To assess the success factors of projects, corresponding assessment tables have been drawn up to help recognize the special features of successful projects [2].

Systematic risk recognition and assessment is the method for managing risks and pinpointing the responsible party. The stages of risk analysis are defining the target, recognizing hazards, assessing consequences, calculating probabilities, assessing the total risk, and finally, removing, reducing, and preventing the risk. The parties involved in the implementation of the project are assessed for their ability to take responsibility for the risk; usually, the responsibility is given to the party that can handle it and to whom it naturally belongs.

Drafting a contract is a concrete tool for pinpointing the party responsible for risks. Through risk assessment, crucial matters to be included in the contracts are observed, along with the necessary insurances and potential securities.

Risk assessment in planning stage

One practical method for systematic risk assessment is a risk sharing table (or checklist, or matrix). Potential (along with apparently impossible) risks are listed and shown to the project parties. Reference [3] presents the categorisation or risks in macro, meso and micro levels (‘meta-classification approach’), [4] shows the practical risk evaluation and allocation table for a school project in Finland. With the risk-sharing table, the parties share the responsibilities for the risks each party will assume. The responsibility for an individual risk is left to either party, or to both parties, according to principle agreed by the respective parties. The client assesses the risks of the project and assumes responsibility for some of the parties involved in the project. In the call for bids, the bidder is asked to pay attention to the risks transferred to the provider and to describe how they will assume responsibility for the risks or how they will take these into account in the bid. In practice, the provider may not necessarily take the responsibility of all risks offered but will transfer the risk back to the client in the bid. The responsibility for a risk will naturally have an influence in pricing – the party taking the responsibility of the risk will set the price for the risk.

The risk assessment table helps the bidders in evaluating how they accept the risks the client is transferring to the provider. The table forms a foundation for the proposition of the bidders in order to show how well they understand the nature of the risk, how effectively they forward or share the risks in the subcontractor chain, and how they minimize the unfavourable effects caused by risks in relation to the client and the end users.

Risk management during use

After the construction phase in the project has been completed, risks are also managed with the help of components of the information system, such as the building management and monitoring system (BMS). The functionality of the technical system is monitored and measured, paying particular attention to the factors sanctioned in the contracts. The monitoring system can contain risk assessment tools, which focus the monitoring and service to the risk points of the process. Disadvantageous factors of the various parts of the system are assessed on the basis of three factors [5]. The probability of occurrence (showing how probable the occurrence of a factor is) and the probability of discovery or detection (showing how easy it is to discover or detect a factor), as well as the severity (showing how harmful this factor is) of the disadvantageous factors, are assessed separately on a scale of one to ten (or some other selected scale). The product of these three factors is the total risk number. The higher the risk number, the higher the risk caused by the disadvantageous factor. When the risk points have been recognized, the measures can be focused. The risk points of the AC system can be handled with the monitoring system (critical monitoring points), while service can be targeted to the system risk points. Additionally, measurements can be utilized in defining the numerical values. Moreover, applications can be developed to assess risks in energy consumption, which is crucial in some contract models, e.g. in energy service contracts.

PROCUREMENT AND CONTRACT MODELS

All life cycle models in building services technology accentuate cooperation between contractual parties. This cooperation starts in the bidding phase and continues throughout the service stage to the termination of the contract. In life cycle models in building services, the contract acts as the basis for cooperation. The objective of the contractual parties is not only to enter into a contract but, with the help of the contract, to accomplish successful cooperation. In addition to mutual trust, one basic prerequisite for successful cooperation is compatibility of the objectives and endeavours of contractual parties. Building and maintaining strong trust between the client and the service provider should be one of the most important objectives for both parties, even with regard to the preparation stage of the project and, naturally, to the contractual period, all the way to the termination of the contract. Trust makes cooperation possible by supporting the parties' mutual commitment and their ability to take risks [6].

The first stage of entering into a contract is an invitation for bids. In this stage, the service provider obtains the information needed to make a bid. In this stage, clients must recognize and concretize their objectives and requirements concerning the service level and quality of building service [7]. It is essential to concretize the objectives and requirements so that the objectives set for procurement can be turned into functional and technical features, requirements, and contractual obligations that are understood in the intended and in a uniform way. The parties have to identify the party responsible for the risk of functionality, and indicate whether the service provider is committing to an output or operational obligation. Moreover, when the client in a life cycle project is a public authority, purchasing must comply with the regulations and legislation pertaining to public purchases. In the case of public purchases, the public contracting authority is obliged to organize a competition for each purchase and regulate the principals and methods to be complied with in competitions [8].

Depending on the project, the contractual period in life cycle contracts usually spans several years. Due to the long-term nature, the life cycle contracts highlight the role of the contract as a tool of risk management, as well as the preconditions of changing the terms of the contract [9]. In life cycle contracts, risk management is divided into two parts: reactive and proactive [10]. In reactive risk management, the terms of the contract are resorted to due to a failure or triggering of risk. Based on the conditions, the disadvantageous economic consequences are divided between the contractual parties. In proactive risk management, contracts are used in anticipatory preparation, which decreases the probability of risks, diminishes the effects, and speeds up the recovery. At its best, the formulation of the conditions of the contract and the entire contract is really proactive risk management in which legal, technical, and economic expertise is utilized.

Also, due to the long-term nature of a life cycle contract, the implementation of the contract and cooperation are affected by conditions that can vary in a way that could not be foreseen or in a way for which the contractual parties had not agreed in a clear risk-sharing model [11]. The longer the time span, the less foreseeable the problems. The most difficult issues are unforeseeable, essential, permanent, or long-term changes in the conditions. These issues influence either the original cost structure or income formation significantly. However, the parties must prepare themselves for the changes by creating contract mechanisms to remove excessive imbalance. Nevertheless, the main starting point is that the changes do not justify changing the contract if responsibility for the risk of these changes, or for the risks themselves, has been assumed by a contractual party. Some sort of a change in the situation is an overwhelming obstacle for which contract law offers several established solutions. Generally, an overwhelming obstacle involves unforeseeable, overpowering, external obstacles rendering performance impossible. Nonetheless, most changes do not lead to impossible, but rather to essentially altered, implementation costs.

Each life cycle model in building services involves document templates and instructions pertaining to procurement and contracts. The document templates and instructions developed within the CUBENet are intended as guidelines for dealing with the matters to be covered in the procurement documents and contracts. On the other hand, models yield examples of ways of defining and agreeing upon matters. With the help of the models developed, actors can, in practice, define the methods and conditions of the contract suitable for their own needs and projects.

VERIFICATION AND MEASUREMENT

Life cycle project contracts are entered into before the building design phase has been finished, as design is a part of the life cycle delivery. This emphasizes the importance of the agreement on the feasible design criteria, the service level of the life cycle contract and the definition of methods for the measurement and verification for the final outcome of the life cycle project. The CUBENet project provides instructions on agreeing on the M&V tasks and methods, and taking implementation into account as demands of the building automation system. Further, methods and tools have been developed for a partly automatic and, if necessary, manually operated, M&V tool to be used during the performance phase of the contract. The methods developed are designed for M&V of room temperature, air quality, and energy consumption.

COMPARISON OF LIFE CYCLE ECONOMY

The importance of inspecting the economy from the life cycle viewpoint is gaining in importance as service providers provide clients with service solutions covering the entire life cycle of building services. In such projects, the service provider is responsible both building and maintaining the system. As the same body is responsible for the entire life cycle of a building service system, it is sensible to pay attention not only to the investment costs but to the operating costs as well. By developing life cycle cost management, the total costs of building service systems can be cut [12]. By paying attention to life cycle costs in planning and implementation, it is possible to decrease operating costs in the future.

When the client ponders the procurement method with which he will realize the construction project, a calculating tool is needed to help to perform a rough comparison of purchases produced with life cycle services with that implemented with a traditional method. Selection of the procurement method should be based on the selection of a consortium that is best able to provide the desired service [13]. In comparing procurement methods, it is especially important that the tasks and responsibilities of the procurement methods, as well as the risks inherent in these methods, are made comparable. What is crucial in comparing different procurement methods is an assessment of whether life cycle services create added value in the investment process and service provision [14]. A particular significant benefit of comparing procurement methods is the fact that it “forces” recognition and emphasis of the matters influencing the total economy and risks of the project. In this way, decision-making is supported by information that is as extensive and high quality as possible.

In the planning stage, life cycle costs are used in choosing various alternative engineering solutions and as a tool for guiding the plans. The reason for a life cycle cost estimate may be:

- There is a plan to install machines, equipment, or systems in a building in order to conserve energy. In this case, a life cycle cost estimate is used to see if building such systems is feasible, i.e., if the obtainable savings are larger than the investments needed.
- There are several alternatives to choose from, each acceptable for the operations of the building, but with different construction and maintenance costs. In this case, a life cycle cost estimate is used to see which alternative is the best one economically.

In comparing bids, the contents of bids obtained using life cycle costs are compared to the call for bids and conditions set therein, as well as to other bids. With life cycle calculation, bids, and operational and maintenance costs are made comparable. With the calculation, the contents of bids can be opened up and assessed as regards the reasons for potential price differences and if the contents of the bids are uniform with the content of the call for bids and with each other. In life cycle costs, all potential factors affecting the total economy of the project are taken into account, whether quantitative or qualitative.

The calculation tool for life cycle economy (CUBECost) developed within the CUBENet can be used as a tool when deciding on a project form, selecting planning solutions, and comparing bids.

DISCUSSION

The CUBENet project has acted as an ice breaker in the field of building services. It has introduced the life cycle paradigm to the experts in the field in a very serviceable manner. Strong co-work between scientists and representatives of Finnish companies and municipalities has taken place. It has not only taken into consideration the needs of experts in building service technology, but has also communicated the needs of co-operation to building managers both in the private sector and in the municipalities. Furthermore, considering the whole life cycle of the building, co-operation with other sectors in construction, from the design team to the property and facilities management operators, is essential. This has been recognized during the modelling of the building service life cycle and generating the contract tool box, as well as in the establishment of the methods for the M&V. The CUBENet project has kicked-started the development of the building service life cycle approach, in accordance with the EU energy policy that targets ever better energy efficiency and the new directives for the contracting of public works.

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Integration of HVAC Value Chains across Eight Competitive Arenas for Better Wellbeing Indoors

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SUMMARY

The aim is to advance HVAC businesses, procurement, and marketing. The Porterian concept is applied to integrating HVAC value chains. The logic of HVAC focused, 8-arena competition is as follows. Arena 1 (of wellbeing indoors): occupants exploit HVAC systems. Arena 2: owners compete with buildings. Arena 3: providers offer services for managing high HVAC system life-cycle performance. Arena 4: investors compete by realizing (renewing) new (old) buildings. Arena 5: wholes contractors and CM consultants compete in realizing buildings as wholes. Arena 6: HVAC engineers compete with solutions and HVAC contractors with systems. Arenas 7/8: HVAC products/materials suppliers and traders compete with (in)direct deliveries. The integration of HVAC value chains is in its infancy in most countries. HVAC inputs procurers (and outputs marketers) are encouraged to set challenging procurement (marketing) objectives and to attain them by adopting the three logics. Cross-national research is called for to conceptualize and to test the espoused logics further.

INTRODUCTION

Two interrelated questions are herein addressed as follows: “How to replace the current ineffective rules of separated or multi-phased competition in building markets? and “How to pioneer some new more effective ways of integrating all HVAC knowledge and value chains for producing better innovative solutions for the wellbeing of occupants indoors?” It is posited that these questions and the related replies are highly relevant as part of advancing HVAC solutions and practices as well as the life-cycle management of new buildings and building stocks as a whole in various (inter)national contexts at least at three levels as follows. **At the broadest level of (inter)national practices**, building markets are facing with constant challenges, that being the separation of designs and construction works culturally, contractually, and technically. Emerging integration efforts are influenced by political, economic, social, and cultural concerns. These efforts become highly significant when supported by governmental stakeholders [1].

At the level of buildings, many new concepts of sustainable and green buildings, spaces, or workplaces are being co-planned and adopted. In the exemplary context of Canada, there is a growing momentum in both the public and private sectors for **more sustainable approaches to buildings**. This momentum arises from the need to reduce raw materials, water, and energy use as well as waste and greenhouse gas output. However, even more important than mitigating these negative impacts is the potential of amplifying the positive impacts of occupant satisfaction and performance. The life-cycle benefits of sustainable construction appear to considerably outweigh their 2-7 % higher first costs. However, there are several impediments to implementation that are slowing the adoption rates. These impediments include (i) changing the ways of implementing buildings, (ii) adopting sustainability and still meeting each owner's needs and cost requirements, (iii) removing the resistance of stakeholders to change, (iv) eliciting action, support, and involvement by all participants in supply chains, and (v) reducing the lack of relevant skills and knowledge. In order to meet these impediments, one of the essential principles is the use of fully integrated teams who consider all aspects of buildings from cradle to grave [2]. In the context of the USA, more and more **workplaces** that are distributed, connected, adaptable, flexible, serviced, enabled, and move seamlessly between space and cyberspace are not only the source of future competitive advantage; such workplaces may now be a matter of company, business, and organizational survival [3].

At the level of HVAC solutions and systems, the first integrated HVAC value chains are readily playing decisive roles in meeting energy efficiency and first cost objectives on high performance green building projects across national building markets. The well thought-out networks of interrelated processes are being designed to satisfy the needs of building owners and users. However, there are still many disabling structural features that must be overcome such as extreme specialization within functionally stove-piped organizations and industry fragmentation, optimized to meet each individual participant's performance objectives but far from optimal from the perspective of integrated HVAC value chains, not to talk about building value chains as a whole (applying [4]).

Herein, **the aim** is to advance core strategic thinking in HVAC business management as well as in related competition, procurement, and marketing by re-defining the three existing logics and applying them to integrating HVAC value chains across many competitive arenas. For advancing the wellbeing of occupants indoors, the sub-aims are as follows: (1) To introduce the logic of HVAC related competition as eight arenas, (2) to introduce the two logics of HVAC related procurement and marketing in terms of principal routes connecting the eight arenas, and (3) to suggest a set of principles for integrating HVAC value chains and solutions across arenas, and (4) to discuss some key implications of such integration for both practitioners and scholars.

METHOD

The nature of this paper is that of **applying one of the authors' recent Porterian concepts** [5], [6] on competition in construction to defining and explaining some new ways of integrating HVAC value chains as part of building markets. Briefly, the original rationale involved the (self-)critical evaluation of the related literature as follows. Within the strategic management literature, **Porter's [7] five forces framework** offers a generic frame of reference for any firm who aims at attaining repeatedly its marketing goals. This framework applies to all high-tech, low-tech, and service businesses, including global and local building and HVAC related businesses. Indeed, Ofori [8] recalls several construction-related applications such as Yates et al.'s [9] analysis of the US construction industry and Betts and Ofori's [10] framework for strategic planning in construction enterprises. Over time, the structures, advancement, and applicability of Porter's frameworks have been criticized in both generic (e.g. [11]) and construction-related (e.g. [12]) terms.

Since the year 1994, Kiiras and Huovinen have applied the same Porter's framework to the various contexts such as a spearhead strategy for entering a new construction market in Europe [13]. Nevertheless, it was concluded that the six aforementioned references had produced only the fairly applicable concepts for practitioners by the mid-2000s. Thus, these two authors engaged themselves in designing three recent concepts or logics [5], [6].

RESULTS

The three logics of arena-based competition, procurement, and marketing are re-defined in the context of HVAC markets. This is followed by the new applications for integrating HVAC value chains when such integrators inside the upstream Arenas 1-5 involve users, owners, investors, wholes contractors, and wholes CM consultants, respectively, and when **pioneering integrators** inside the downstream Arenas 6-7 involve the focal HVAC systems engineers and contractors as well as products and materials suppliers and traders.

Three logics of 8-arena competition, procurement, and marketing in HVAC markets

The logic of HVAC focused competition as part of any building market is herein defined in a form of eight interrelated Porterian arenas. In Arena 1 of wellbeing indoors, users occupy the stocks of buildings and exploit the HVAC systems that enable to perform their purposes in business, public, and private life. In Arena 2, owners compete with their building stocks including HVAC systems on a long-term basis. In Arena 3, providers offer their services for managing the high HVAC system performance over the life-cycles (L-Cs) as part of building stocks. In Arena 4, capital investors compete through investing in the realization of new buildings and the renewal of the existing ones including HVAC systems. In Arena 5, wholes contractors and CM consultants compete through the realizations of new and to-be-renovated buildings as wholes including HVAC systems. In Arena 6, HVAC engineers compete with their solutions and HVAC contractors with their (sub)systems when buildings are realized as parts. In Arena 7, HVAC products suppliers and traders compete with their (in)direct deliveries. In Arena 8, HVAC materials suppliers and traders compete with their deliveries as well.

Table 1. Logic of 8-arena competition in HVAC markets (applying [5], [6], and [7]).

No	Competitive arena	Incumbents and entrants
1	Wellbeing of occupants using HVAC systems	Owner users and tenants
2	Ownership and trading of buildings incl. HVAC systems	Owners + traders
3	Life-cycle services for HVAC systems in buildings	L-C services providers
4	Investing in and trading of buildings incl. HVAC systems	Investors and traders
5	Realization of buildings as wholes incl. HVAC systems	Contractors+ CM consultants
6	Realization of buildings as parts such as HVAC systems	Engineers + contractors
7	Supply and trading of HVAC products	Suppliers + traders
8	Supply and trading of HVAC materials	Suppliers + traders

The logic of the route-based procurement of HVAC inputs from downstream arenas is defined in terms of those procurement routes that couple the eight competitive arenas and the ones that enable sub-procurement inside any given arena. Focal procurers have one or more routes to pull together more or less integrated HVAC value chains across markets, businesses, downstream arenas, and projects. It is axiomatic that each particular procurement route, which any procurer chooses, determines the nature and rules of competition in the next downstream arena(s).

The logic of the route-based marketing of HVAC outputs to upstream arenas is defined in terms of those marketing routes that couple the eight arenas and the ones that enable a marketer to win deals also inside its base arena. Each route is coupled with a particular kind of buying behavior of clients or procurers. It is axiomatic that each marketing route, which any client/procurer in the chain prefers or accepts, determines the rules of competition in the next downstream arena(s). Focal marketers have one or more routes for connecting their base arena to one or more targeted upstream arenas or for enabling their marketing inside the base arena. Each HVAC related firm occupies **the two principal roles** of a marketer and a procurer in its primary competitive arena (or its base arena). As a marketer it interacts via various routes with targeted clients in upstream arena(s). As a procurer it acquires specified inputs via various routes from inside the same arena and from one or more downstream arenas.

Integration of HVAC value chains as part of buildings in base Arenas 1-2 and 4-5

The integration of HVAC value chains as part of value adding building related processes in the base Arenas 1-2 and 4-5 is illustrated in Figure 1. **In Arena 1 of building uses and wellbeing occupation**, many advanced owner users and most attractive tenants aim readily at being extremely well indoors inside pioneering high-performing green buildings, workplaces, and spaces across the globe. On the one hand, they desire to manage well the environmental impacts in terms of reducing energy consumption, emissions, electricity use, water use, solid waste, wood and materials use, and land exploitation (e.g. [14]). On the other hand, future workplaces are ecosystems comprising a careful balance of people, processes, and places. The focus is shifting from places to people: supporting their performance, satisfaction, and wellbeing in indoor climates. By analyzing work people do and then creating the right combinations of spaces, processes, and tools to support it, workplace management enables to increase both effectiveness and efficiency as well as to accelerate cultural/organizational change (applying [15]).

In Arena 2 of building stock ownership and trading, many advanced long-term owners and traders aim readily at cultivating their building stocks toward high-performing green buildings. In the OECD countries, people spend the bulk of their time indoors. Indoor climate has a direct impact on their health and productivity at work. Typically, personnel's salaries form the dominant shares of the life-cycle costs of office buildings (e.g. [16]). By improving indoor climate conditions, the wellbeing and productivity of personnel can be enhanced markedly. Therein, the energy efficiency of HVAC systems and the limited need for maintenance add to the profitability of indoor climate investments. In addition, the Internet convergence requires a new approach to designing the spaces in which principal processes are conducted, from stores to offices to factories to classrooms. This requires a new kind of **convergent architecture** including HVAC systems (e.g. [17]).

In Arena 4 of capital investing, many advanced short-term investors (along long-term owner investors) aim readily at enabling their pioneering clients to stay on the cutting edge in their core businesses or public services by **maximizing the total productivity** of their buildings, workplaces, spaces, and special rooms. Handing-risk-out investors rely on wholes contractors with their integrated value chains or nets. Instead, carrying-risk-themselves investors rely on wholes CM consultants with their core competences in integrating even unique, complex building systems and system-specific value chains. For example, integrating energy efficiency throughout building design processes will yield significant first cost savings. Capital and operating cost impacts are considered in tandem in order to realize maximum value from

HVAC based facility improvements. Efficiency is also associated with healthier and higher quality work environments, especially regarding the use of outside make-up air. In addition, the pre-diagnosis of HVAC solutions as part of the life-cycle engineering of buildings enables the early identification of maintenance problems, safety issues, and premature equipment failures (aligning with [18]).

In Arena 5 of realizing buildings as wholes, many contractors and CM consultants aim at enhancing their core competences to higher levels where they can assume those integrative roles of pulling together value chains or nets. At the same time, many of them may become the providers of life-cycle services vis-à-vis targeted owners and their building stocks. In Finland, special system contracting is being developed to advance the adoption of extended building contracts with the detailed design and engineering, manufacturing, delivery, and installation of various building systems (e.g. HVAC systems), modules (e.g. clean rooms), and functional elements (e.g. wellbeing in a high-level indoor climate) [19].

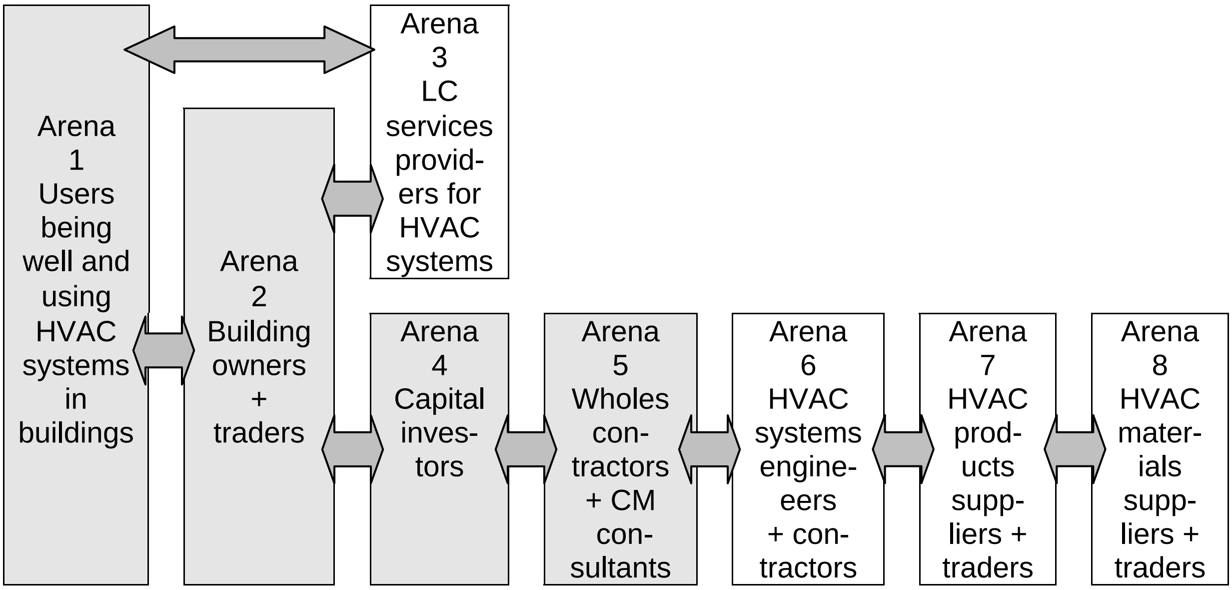

Figure 1. Eight basic routes for (a) procuring HVAC inputs from downstream arenas and (b) marketing HVAC outputs to upstream arenas within any given building market.

Integration of HVAC value chains by systems engineers across Arenas 3 and 6-8

In Arena 6 of HVAC systems realized as parts in buildings, pioneering independent HVAC systems engineers can aim at (i) integrating their engineering-based HVAC value chains in terms of the most advanced and evolving HVAC solutions, expertise, and core E&D competences, (ii) developing such chains internally and acquiring them externally in a form of (inter)national mergers, acquisitions, licensing, and collaboration, (iii) enlarging the scopes of their services to include also an array of HVAC systems life-cycle services, and/or (iv) becoming the HVAC knowledge-based members of the most competitive building-level value chains led by some of globally, internationally, regionally, or nationally leading wholes contractors. Engineering-based HVAC value chains can market their integrated solutions and competences via **four connecting marketing routes** to (a) wholes contractors and CM consultants in the upstream Arena 5 and/or (b) capital investors in the Arena 4. In addition, such integrated engineers can market their HVAC systems life-cycle services to building owners in Arena 2 and/or responsible users in Arena 1. These four marketing routes are shaded in Figure 2.

In particular, it seems that integrated HVAC engineers can attack many pervasive problems such as **the systematic over-sizing of utility systems** and the resulting oversized cooling equipment caused by overestimating first electrical demands thereby creating the fictitious sources of heat within a facility. This is compounded by the aggregate impact of multiple disciplines (owners, process engineers, electrical engineers, HVAC engineers, etc.) each adding “safety factors” to their calculations. The result is also higher operating cost due to excessive and inefficient on-off cycling and/or part-load operation. For example, **a combination** of high-performance glazing, low-pressure-drop HVAC design, and efficient right-sized lighting systems results in major reductions of the mechanical and cooling system sizes. Smaller duct sizes even allow to lower overall building heights for significant construction cost savings. This reduces the sizing of generators, resulting in additional first-costs savings. Ideally, above-standards performing buildings enable substantial savings in annual operating HVAC cost (applying [18]).

In part, engineering-based HVAC value chains can be **virtualized**, i.e. they may establish collaborative one-stop virtual engineering services such as the COVES project in Singapore. With the help of collaborative platforms and virtual communities, collaborative HVAC engineers might target to offer and to integrate services for heat ventilation and air-conditioning, fire and smoke, pollutant dispersion, and indoor air quality. Virtuality is being enhanced via e-modeling, e-analyses, e-simulations, and e-visualizations [20].

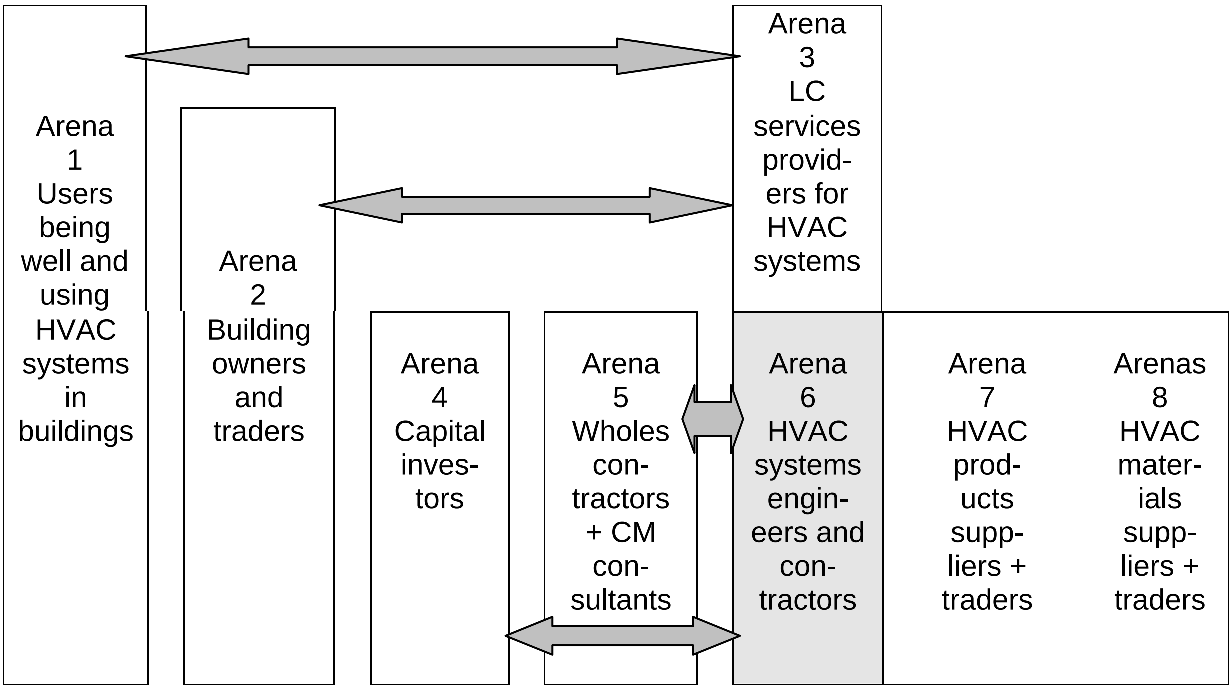

Figure 2. Systems engineers and contractors integrating HVAC value chains across Arena 3 and Arenas 6-8.

Integration of HVAC value chains by systems contractors across Arenas 3 and 6-8

In Arena 6 of HVAC systems realized as parts in buildings, pioneering HVAC systems contractors can aim at (i) integrating their contracting-based HVAC value chains in terms of the most advanced HVAC technologies, systems, products, and materials as well as core system management competences, (ii) developing such chains internally as well as acquiring and procuring them externally in a form of (inter)national mergers, acquisitions, licensing, partnering, networking, and/or competitive bidding (vis-à-vis key contractors, suppliers, and traders), (iii) enlarging the scopes of their services to include also an array of HVAC systems life-cycle services, and/or (iv) becoming the HVAC system-based members of the most competitive building-level value chain led by some of globally, internationally, regionally, or nationally leading wholes contractors. Contracting-based HVAC value chains can market their integrated solutions and competences via **four connecting marketing routes** to (a) wholes contractors and CM consultants in the upstream Arena 5 and/or (b) capital investors in the Arena 4. In addition, such integrated contractors can market their HVAC systems life-cycle services to building owners in Arena 2 and/or responsible users in Arena 1 (Figure 2).

The integration of HVAC systems targets opportunities in terms of integrating better the pieces of equipment, systems, or facilities. High variations in the equipment efficiency (e.g. cleanroom fan-filter units) can be minimized. Challenges and opportunities extend beyond swapping in and out HVAC products/components. It is not enough for pumps to be efficient. Indeed, even small improvements that enable and enhance high system integration can yield large benefits. In turn, **building automation and control systems** enable system-level energy savings opportunities. For example, variable-speed fans need to run at full speed less than 10 hours per month. Buildings with more advanced ventilation control systems are more resistant to chemical and biological attacks. The use of outside air improves the overall air quality [18].

DISCUSSION

This is a theoretical paper where the application of three logics of 8-arena competition, procurement, and marketing to the context of HVAC markets is initially supported with the selected exemplary references on the empirical status in some national contexts. Nevertheless, it is herein posited that **the integration management of HVAC value chains** is still in its infancy. Across most national practices, stakeholders are still focused on achieving partial, in-house process performance improvements. Overall, many pioneering owners, investors, wholes contractors and CM consultants as well as HVAC systems engineers and contractors have shifted to a view outside in their value chains including both first-tier and higher-tier stakeholders along upstream and downstream arenas. However, only a few efforts are spanning the entire length of any one value chain, not to talk about the life-cycles of HVAC systems embedded in building stocks and those of inherent HVAC systems (aligning with [4]). Readily, many existing barriers have been identified and the first progressive paths toward best practices are being outlined (e.g. [18]).

In turn, **pioneering HVAC inputs procurers (outputs marketers)** are hereby encouraged to set more advanced procurement (marketing) objectives and also attain them by (i) adopting the three logics, (ii) identifying the incumbents and their procurement (marketing) routes connecting the base arena with the targeted downstream (upstream) ones, (iii) anticipating changes in the focal arenas, entrants, and routes, (iv) deducing non-existing procurement (marketing) routes and/or inventing the new ones, and (v) (re)designing their procurement (marketing) strategies accordingly. Admittedly, the real usefulness of the espoused applications of three logics to integrating HVAC value chains needs to be tested, for example, in the context of building and HVAC markets in Finland and some other EU countries in the near future. Thus, collaborative cross-national research is called for in order to reveal the most advanced benchmarks for advancing best integrative practices internationally. In addition, the three HVAC focused logics may be conceptualized in more detail, arena by arena, by rolling the focus over upon each of stakeholders forming the targeted HVAC value chains.

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Identifying Suitable Procurement and Briefing Processes for Schools

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ABSTRACT

The UK government are increasingly emphasising the importance of improving the school built environment. Recent studies have demonstrated the influence school architectural design factors have on pupils' achievement. Yet, the procuring mentality continues to be influenced by the legacy of short term decisions based on cost. The long term implications of the designed solutions are not fully appreciated. Procurement strategies exclude the real experiences and needs of the users of the buildings. The briefing process is failing to capitalise on the wealth of design talent that unarguably exists. New procurement methods are presented as means of coming to terms with some of these problems. Yet, these procurement routes have generated a great deal of debate and have received mixed responses. Initial evaluations have been less than conclusive as to their potential benefits. This working paper reports on the preliminary findings from a research project exploring the potential for changed practices in the design stage in these new procurement methods. Particular interest is given to the correlation between the design intent and the design outcome. Including how the chosen briefing and delivery methods allows for sufficient exchange of skills and expertise to produce designs in line with client needs.

INTRODUCTION

This paper addresses the pressing concerns of educationalists and the construction industry about the future approach to school design and procurement and how will this variety of approaches impact on the building design.

The overall aim is to assess whether links exist between new, well designed buildings and the procurement and briefing process utilised. In addressing the aim of the study, a number of key research questions were posed, namely:

- Are users satisfied with the quality and functionality of their buildings and associated facilities, and do they equate good quality with better performance?
- Are designers achieving good school design?

The study ultimate aim is to answer the following question: *Is there an ideal procurement route when designing a school*?

PROBLEM BACKGROUND

The last decade has seen one of the largest school building programmes in UK history However, many reports (Audit Commission 2003) have shown that apart from some notable exceptions, far too many schools now operational are considered to be mediocre or worse. These schools are not providing pupils with an adequate place where to learn. Nor are they providing teachers with an adequate place where to teach.

One of the reasons behind this problem is that the briefs tend to be based on traditional models of teaching and learning. In this way, issues like ICT and personalised learning are not being taken into consideration when designing schools. Added to this, a variety of functional problems are frequently deficient in many of the new schools such as: inadequate entrances and circulation spaces; limited social areas such as dining halls; a lack of future adaptability and; a lack of design of external areas. Environmentally poor many schools have lack of day lighting in classrooms with inadequate ventilation and poor acoustics making it very difficult for teachers and pupils to interact.

The fact that all these schools are a lost opportunity to create interesting and exciting buildings that would make teacher and pupils proud is a very real concern. These schools are inadequate to the shifting pedagogy, curriculum and learning expectations of the present and the future are lacking the agility to cope with anticipated changes(Building Futures 2004). Although there is only sparse evidence that education standards can be better achieved with 'great' buildings, it can still be argued that stimulating environments can be a great motivator for teachers and students.

THE NEED

At present there is a growing recognition that the public sector must be provided with environments that provide children with good places to learn, designed to the highest quality. Schools that are sustainable. Projects that involve users, stakeholders and local communities in their design and implementation. The government is giving the issue considerable attention and several separate but interconnected initiatives have been initiated.

PRIVATE FINANCE INITIATIVE

The Private Finance Initiative (PFI) provides approximately 15 per cent of the public sector's total capital investment and is currently limited to use on projects with a capital value in excess of £20 million. Between 2000 and 2005 PFI has been the main source of funding from the Department for Education and Skills (DfES) for new or replacement schools. One of the reasons behind the use of PFI for provision of new buildings for the public sector is that the finance available for the private sector may be the only way of filling the shortfall resulting form previous decades of underinvestment in infrastructure, in particular in schools.

PFI is a particularly complex and lengthy procurement process and it has frequently been considered to get in the way of creating design quality. In practice it is a commercial arrangement and driven in no small way by commercial interests. As such those involved in PFI, including public sector parties, are predominantly concerned with risk transfer, public sector comparators, 'value for money', output specifications and affordability. Therefore, the importance of 'design quality' has at times been suppressed.

Initially the government's belief was that the involvement of the private sector in the provision and management of the project would provide greater and continuing efficiencies in the construction and operational management of the projects. Also, the private sector would bring commercial and technical expertise and innovation to the project. Overall, the involvement of the private sector would generate cost efficiency and value for money.

However, the introduction of PFI projects into schools in the UK has generated a great deal of debate and has received mixed responses. In terms of the resulting buildings, the PFI in Schools study commissioned by the Audit Commission (Audit Commission 2003) found not only that few schools came out well in terms of the buildings cost of ownership but that the PFI sample scored, were significantly worse than the traditionally funded sample.

In the PFI procurement route, clients produce their requirements in highly structured documents setting out performance standards for bidders to meet. Few, if any, drawings are produced by the client, who takes no direct responsibility for the design. Bidders have to assemble a team that includes designers, contractors and facilities managers. The need to focus on the whole life cycle of a building making sure that standards will be maintained throughout 25 years of the contract is considered to be the great strength of PFI.

The RIBA has identified several problems with PFI including the excessive cost and time necessary to bid for PFI; inadequate resourcing of the public sector client at the early stages of procurement; lack of experience by public sector client who were first time commissioners of school buildings; work project briefs; insufficient direct contact between the client and the design team during the bidding stages. The result is a reduced value for money, excessive costs, barriers to complete competition, delays to delivery with an overall lowering of design quality.

BSF

The Building Schools for the Future (BSF) programme started in 2003/2004 (Building Futures 2004). It is a coordinated national strategy driven by the DfES that over the next 15 years will result in the rebuilding or refurbishing of every secondary school. BSF is not merely a building or procurement programme. The intention is to drive reform in the organisation of schooling, teaching and learning, ensuring transformational educational change.The BSF has now entered into the delivery phase that will see all secondary schools across England rebuilt or refurbished by 2020. 38 local authorities are engaged on BSF in the first three waves of the programme, and a further 33 have been invited to prepare for waves 4−6.

In 2005/06 the government’s investment in school buildings reached £5.5 billion. This included £2.1 billion for the Building Schools for the Future programme becoming a key part of the current strategic and targeted capital programme.

The audit commission report PFI in schools (Audit Commission 2003) which assessed quality and cost in early PFI projects (up to 2002) showed that PFI schemes did not deliver high quality buildings. In this way, BSF attempts to respond to that criticisms, with a programme of improvements. However, a recent study carried out by CABE (CABE 2006) shows that, though still in the early stages, many of the BSF schools on the drawing board are facing the same problems as previous programmes.

WHAT IS GOOD SCHOOL DESIGN?

For a long time reports have been written by different sources pointing out the need for good design in public buildings (RIBA 2001; CABE 2002; CABE 2004). Some of these reports are specifically targeting the PFI procurement route arguing that this approach is not addressing the concern of the government and stakeholders of achieving good design(CABE 2002; Audit Commission 2003; RIBA 2003; UNISON 2003; CABE 2005; CABE 2006). Some reports are aiming to draw lessons from what has been delivered to date that could help improve future PFI schemes (McCabe *et al.* 2001). In a document entitled *'PFI? A question of quality'* the Royal Institute of British Architects (2003) makes key recommendations on how to improve

PFI procurement, lending its support to the scheme as long as architects are involved earlier in the process.

But what does good school design really mean? There will probably be different points of view depending on the different stakeholders. The best people to judge this are the people that have to use the building because if the end users of the school are happy it indicates success.

The Commission for Architecture and the Built Environment (CABE), is charged with promoting good design in public sector building projects. In their report (CABE 2006) that assessed the design of a representative sample of 52 secondary schools completed between 2000 and 2005 using a variation of the design quality indicator for schools (DQI) clients argued that a good school is one that has a *'sense of place'*. This can be interpreted as a building that is inspiring and welcoming, but at the same time has to be functional in a way that encourages good behaviour and is easily managed. Furthermore, one that enables ease of movement with use of wide corridors facilitating easy and logical movement without congestion. Added to this, the outside is as important as the inside, therefore there needs to be good link between the two. Flexibility is another important aspect of the building where the different spaces can be used for different purposes. Last but not least, the school has to be green and sustainable considering alternative forms of energy and be built using robust materials from sustainable resources.

People have different opinions regarding quality, but good design of buildings, specially the public buildings, should respect and enhance the environment, add value and reduce whole-life costs. Buildings should also be created as flexible, durable, sustainable developments that will benefit the user and the community as well as providing functional, efficient and adaptable spaces.

Good design is not just a question of style and taste. It involves adhering to a set of objective principles that determine whether or not a building works well for all users and the community. Good design needs to address the following issues: how a building functions, quality indicators of the building, its environmental effectiveness, its contribution to the surrounding context and its attractiveness.

INFLUENCE OF GOOD DESIGN ON LEARING OUTCOMES

Globally new pedagogies are emerging such as increasing use of ICT and virtual classrooms, tailoring of the education towards the individual child's needs, different means of grouping within schools, opening up schools to wider community and a greater integration of special needs within mainstream schools. In the face of such change in environments there will potentially be a significant impact upon teaching and learning.

However, the question is, *is there is an actual link between the environment and the potential to learn*? Whilst it is easy to calculate and minimise heat loss from a building, the target of minimising a loss of potential learning through good design is considerably more intangible. Different schools, children, cultures and context at different times will create a variety of conditions for potential learning and these needs to be given voice, challenged and defended.

There have been a number of studies (Tanner 2000; Department of Education 2002; Green *et al.* 2004; Design Council 2005) undertaken trying to establish a link between the nature and quality of the physical environment in which students learn, and the learning outcomes associated with this environments. Whilst it is difficult to achieve explicit links, some convincing evidence linking the physical school environment with teaching and learning has emerged from these studies. Similarly in offices there is much evidence concluding that the environments designers create, affect work performance (Clements-Croome, 2005)

In a research study done in America looking at how school architectural factors influenced student achievement scores in elementary schools(Tanner 2000), several design factors were found to correlate with student learning outcomes. Computers where found to be an integral part of the building design where students earned high scores. Freedom of movement within the school and among learning environments was another of the significant design patterns discovered in the study. Furthermore, a lack of expansive pathways implies higher density, which has been proven to affect learning (Bechtel 1997). Crowding and high occupancy density have been associated with decreased attention, lower task performance, behavioural problems, and social withdrawal (Wohlwill *et al.* 1985).

Students with an overall good impression of the learning environment tended to score higher in average than those expressing negative feelings. However, the overall positive impression of a school implies the presence of friendly students and teachers. Students attending schools with poorly designed and maintained outdoor spaces had lower scores. It is clear that positive outdoor spaces invite nature to blend with the school's function and form. It is important for children to have a sense of being in a natural setting (Lawson 2001). A positive outdoor area gives a feeling that the school's learning environments are "in harmony with nature".

A research study done on school building investment and its impact on pupil performance in one area of England (Green *et al.* 2004) provides evidence of a perceived positive relationship between government investment and students achievement from the view point of the building users. The findings showed that the built environment was certainly perceived as having an effect on the well being of the teachers and students.

Perhaps one of the most surprising positive illustrations of the effects of building quality on performance is in the results of a case study carried out by a UK local authority (Williams 2006), which looked at the relationship between building quality and education achievement. Twelve primary school buildings were analysed and given a quality score and the educational achievements were taken from the OFSTED results. It is quite amazing to find that there is almost a straight line correlation between higher quality and better educational achievement.

The discovery of relationships between design characteristics and academic achievement should help in future school planning by pointing to design patterns that complement learning. In any case people experience the environment through their available senses at the same time, and it is this accumulation of experience that produce a physiological and psychological effect that ultimately will provide the stimuli to children and teachers.

PRELIMINARY OBSERVATIONS

Based on the theory stated above and on the questions that have been asked several preliminary observations can be mentioned. They are the result of an empirically based study of 2 architectural studios involved in the design of schools using PFI and BSF procurement methods. The interviews aimed to get a better understanding of the realities behind the design of schools under different procurement routes. In detail, the interviews were looking at the way in which innovation, flexibility and user's participation was handled in the context of different procurement methods.

The interviews captured the concern of the architect involved in the design process. In the future, exploratory semi-structured interviews will be carried out with other professionals with experience in different procurement routes such as contractors, facilities managers and investors and a selection of clients such as principals, teachers, and DfES. Thus, information will be gathered from numerous open interviews with practitioners and public sector representatives having experience of completed schools in the UK and abroad. This will help to determine current practice and increase the understanding of the actors' participation in the design process and their expectations. Most importantly, it will help structure the study through the identification of core areas of interest.

The first stage of the interview asked what separates a good school from a bad school, followed by what are the features of a good school building. Building on these issues, these questions were followed by their own experience regarding design of schools.

From the interviews it is clear that the architects' point of view is closely linked to the building features of a good school. Interviewee 1 argued that the architecture of communal spaces, such as dinning room, and the circulation within the building are the main places that can make the difference between a good and a bad school. In this case, the following question regarding the features of a good school building was almost seen as the same question. He notes that once the basic classroom has been reasonably designed, the architect can contribute to a successful school design by providing the spaces such as the street, the entrance and the dinning hall with a unique identity. This provide each school with a unique signature and a place in the local community. Once the building has achieved a degree of design were things are working accordingly, the really exciting thing about designing a school is the potential the building has to achieve something special and unique that makes it different from any other school. The ability to include the ethos of the school into the building will positively help the school achieve its potential. He argues that this is very difficult to achieve in some of the procurement methods such as PFI. The reasons behind this are that the briefs tend to be very similar, leaving little room for new ideas in school design.

On the other hand, interviewee 2 had a more managerial view of schools where, for him, the difference is done by the teachers and the management together with the head teacher and the leadership. Therefore, buildings have a limited role to play in whether a school is a good or bad. The school is the people in it, the pupils and the teachers, where the building is a component of the school, but it is not the school. Drawing from his experience, a very low achieving school was turned around into a highly performing school with no need for a new building, but with the introduction of a new whole learning program focused on motivation and achievement. Looking at another example where a new school was finished in 2005, the 2002 OFSTED reports and exam results showed 66% A to C and in 2006 the school scored 89% A to C. But was this improvement achieved only because of the new building or was it because, when they where moving themselves from the old site to the new site, the school took the opportunity to completely change the management structure, the way they delivered education. Following the idea the interviewee puts forward an analogy by saying "… *I wonder whether it has to do with the fact that it(the building) doesn't directly influence but it makes the journey better. I think the analogy is that you can get on an old train and you still hopefully will get from A to B, but if you get in a new train you still get from A to B but it just makes it more pleasant. It makes actually the experience a better one*".

He concludes this first stage by arguing that the school is a complete fusion of management, the staff, the pupils and the learning program and the school. Therefore, a good school is one where the design of the building supports the learning and integrates the ICT technologies, as well as a healthy naturally ventilated, well lighted, acoustically responsive and accessible building.

But, can all of these be achieved using the existent procurement routes?

In the second part of the interview, architects were asked what they would consider to be an ideal procurement route for schools. In this part architects complaint about the lack of contact with the end users of the school in some procurement routes such as PFI. In this approach the contact with the end users is controlled. It is quite exhausting for the end users and unsatisfactory for the designers.

One of the interviewees argued that one of the problems with the PFI process is to be cost led and lacking continuity so there is no incentive to produce a better product, the only incentive is to maximise profits. The PFI did not generally include education in the agenda, the PFI was about buildings.

Compared to other procurement methods, the BSF programme was considered to be a good route in the way that brings the constructor, the educationist, the ICT provider and the architect together at the very outset of the project, enabling a proper learning integration into the school. This is the reason why hat is BSF programme was considered to be a good system because it incentivises the market to improve on quality because of the long term relationships. Within the BSF programme the PFI is the funding stream that enables the new school.

In the PFI context there was no incentive for the consortium to improve learning outcome. It was all about the building. In the BSF program is not about buildings, is about the integration of learning into the school.

Other advantage in using the BSF Programme is having the constructor at the beginning of the process. This enables the architect to focus on quality, aligning the cost, the design to the most efficient method of delivery. In this way better quality buildings can be achieved.

The third part of the interview looked at the different issues concerning design quality, such as innovation, flexibility and user participation. In both interviews architects agreed that PFI is not delivering innovation, in fact compared to other procurement methods, PFI is considered to be less innovative.

The reason behind this lack of innovation is because the brief is too rigid. The only way to innovate is overlapping some of the activities to get better spaces. However, as the judgement of everybody involved in the tender process is done under the same criteria designers straggle to prove within a square meter that they meet the brief. There is a temptation that you make sure to tick all the boxes that everything is per square metre. And it turns out to be a matter of ticking out the box. Though, innovation has been achieved in some PFI projects this has been despite of it not because of the fact that it is a PFI.

Risk has been considered to be one of the reasons behind the difficulty to innovate. The contracts are loaded to penalties if failing to deliver. So everything is about managing the risk, certainty of delivering within the cost and on time.

However, BSF has the potential to innovate because of that long term continuity. It really does have a potential.

CONCLUSIONS

The slightly discouraging emerging picture that is appearing is that regardless of the programme/initiative it appears as the public sector continues to be driven by short term calculations of cost. The failure to compute the emotional, social and therefore economic benefits that accrue from good design has led to procurement processes which exclude the real experiences and needs of the people who will use the buildings, objects and experiences that are designed. Briefs are issued which ask the wrong questions and thereby fail to capitalise on the wealth of design talent within the UK.

RIBA in 2005 introduced a paper entitled *Smart PFI* which emphasizes the need for longer preparation of the procurement process so more time is spent between the client and the design team besides giving the opportunity to ensure that innovations and best practice are all deployed on the project. An example of good practice is the Jo Richardson School in London.

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Systematic process for commissioning building energy performance and indoor conditions

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SUMMARY

In Finland, buildings are commissioned mostly in the end of the construction phase. However, the commissioning of building performance is a systematic process which should kick off already at the pre-design phase and it should be continued through design, construction and acception to occupancy, use, operation and maintenance. During the Cx process, it should be checked that owner's and users' needs and requirements are clearly defined and documented, and indoor and energy performance requirements are included to owner's program. In addition, it should be audited that the design solutions and installation outputs meet given requirements, and verified that the building satisfies given indoor and energy requirements in use. Commissioning should also be included as an essential part of a systematic facility management process over the building life cycle.

This paper describes a systematic process for commissioning building energy performance and indoor conditions. The Cx process is developed to be applied in Finland. In addition, checklists for each commissioning phase for a commissioning manager were developed. Also preparation of a Cx plan, and options for a Cx team and examples of task responsibilities are briefly discussed.

INTRODUCTION

Nowadays in Finland, building commissioning is not a standard procedure during the building life-cycle. Most often it is only used during building hand-over in new buildings, and sometimes as a separate measure in existing buildings.

Commissioning (Cx) process should be launched as early as in the programming phase to check that owner's and users' needs and requirements are clearly defined and documented, and that indoor and energy performance requirements are included to owner's program. In addition, it should be audited that design solutions and installation outputs meet given requirements, and verified that the building satisfy given indoor and energy requirements in use. Cx should also be included as a part of routine facility management process over the building life cycle.

Cx activities include the following goals:

- To provide safety, healthy and comfortable spaces for living and business
- To improve design quality by more effective feedback
- To assure that all building services systems are compatible with each other
- To improve energy efficiency of buildings and building systems

- To decrease operation costs
- To improve operation and maintenance personnel introductory briefing and training
- To improve documentation during the building life-cycle
- To meet customer needs and expectations and satisfy customer requirements

CX PROCESS

In Figure 1, there is the outline chart of the Cx process [1]. The Cx processes presented in [2] and [3] were the starting points for the development of this Finnish Cx process. The attached checkpoints “diamonds” link the Cx activities to the building process phases in general. At the beginning of a construction project, goals are established and the owner’s and users’ needs are determined. Then, the system requirements are set with the help of design procedures. Thirdly, the goals are implemented and performance is verified in the elaboration and construction phases. Finally, indoor climate and energy consumption is managed with new building automation and online reporting systems.

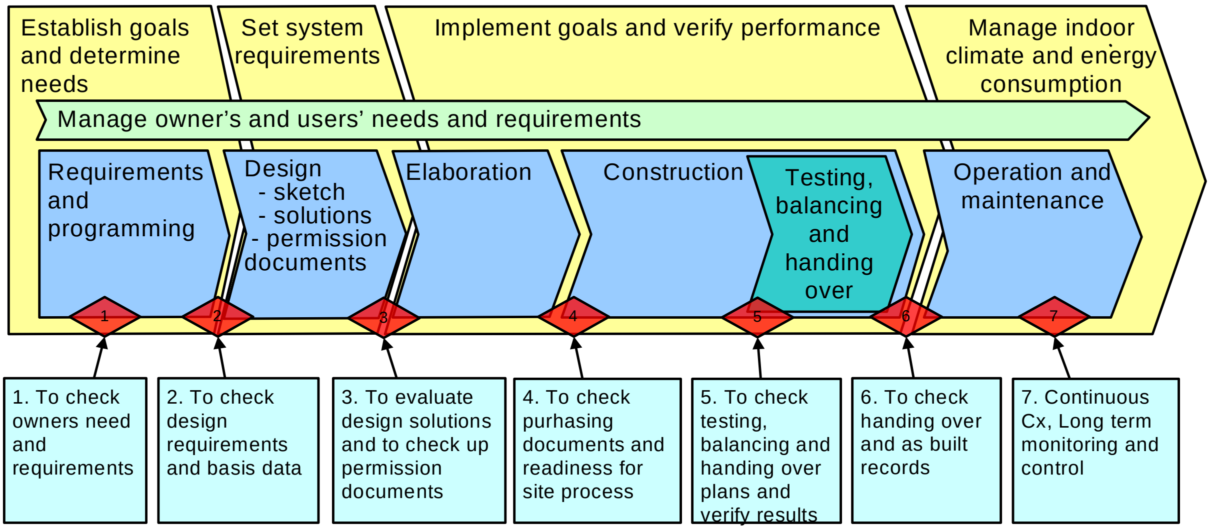

Figure 1. The Cx process.

Requirements and programming review

The objective of the phase (diamond 1) is to ensure that the owner’s and users’ needs and requirements are defined and documented. During the project planning different options to fulfill the owner’s needs are clarified, plans for the project budget are made and the goals for the next phase (diamond 2) are defined.

Key activities are to:

- Check the owner’s future strategies and action plans
- Check the owner’s and users’ needs and requirements
- Check the construction site and detailed town plan
- Check different goals and requirements and identify the possible risks

Design requirements review

The objective of this phase (diamond 2) is to ensure that the design requirements and basic data are relevant for setting up the system requirements. This information is also used to draw up contracts.

Key activities are to:

- Check design goals and requirements
- Check design contract documents
- Check that maintenance and monitoring requirements are sufficiently included

Design solutions and permission documents review

The objective of the phase (diamond 3) is to ensure that the design concepts and permission documents are correct. Indoor climate and energy consumption are based on the design concepts, proper results can be obtained only if the design concepts are relevant.

Key activities are to:

- Check design concepts
- Check building permits
- Take into account the system integration point of view

Purchase and contract documents and construction site reviews

The objective of the phase (diamond 4) is to ensure that the purchasing documents are relevant and the construction site is ready for implementation. In this connection, it is important to accept all system specific objectives with all the contracting parties. Especially, the integration perspective of different subsystems and procurements must be taken into account.

Key activities are to:

- Select and accept the systems to be implemented
- Calculate the design and construction cost levels for every system
- Agree the functional requirements for all systems
- Check that the tender documents meet bidding requirements

Functional testing and balancing review

The objective of the phase (diamond 5) is to ensure that the testing, balancing and hand-over plans are relevant. The main focus is on final tests and preparation for hand-over.

Key activities are to:

- Make sure that the systems and subsystems are functioning as agreed
- Make sure that the agreed level of indoor climate can be achieved
- Make sure that the assignment and maintenance manuals are relevant

Hand over review

The objective of the phase (diamond 6) is to ensure the hand over, also the as built records play an important role. At this phase the interoperation of all systems is crucial.

Key activities are to:

- Review all possible defects in the hand-over process and assess the repair work to be done

- Make sure that the building is faultless
- Make sure that all subsystems are tuned and operating as planned

Long term review of operation and maintenance

The objective of the phase (diamond 7) is to ensure that the indoor climate and energy consumption are managed and monitored for the optimal performance of the building during its whole life cycle. At this phase the continuous commissioning tools play an important role.

Central activities are to:

- Monitor the indoor climate, energy consumption and water consumption
- Measurements, audits and functional tests can be used when needed

CX MANAGEMENT

Before any Cx activities can be realized, a project specific Cx plan must be prepared and a qualified Cx team gathered. In the following, these issues are shortly discussed.

Cx plan

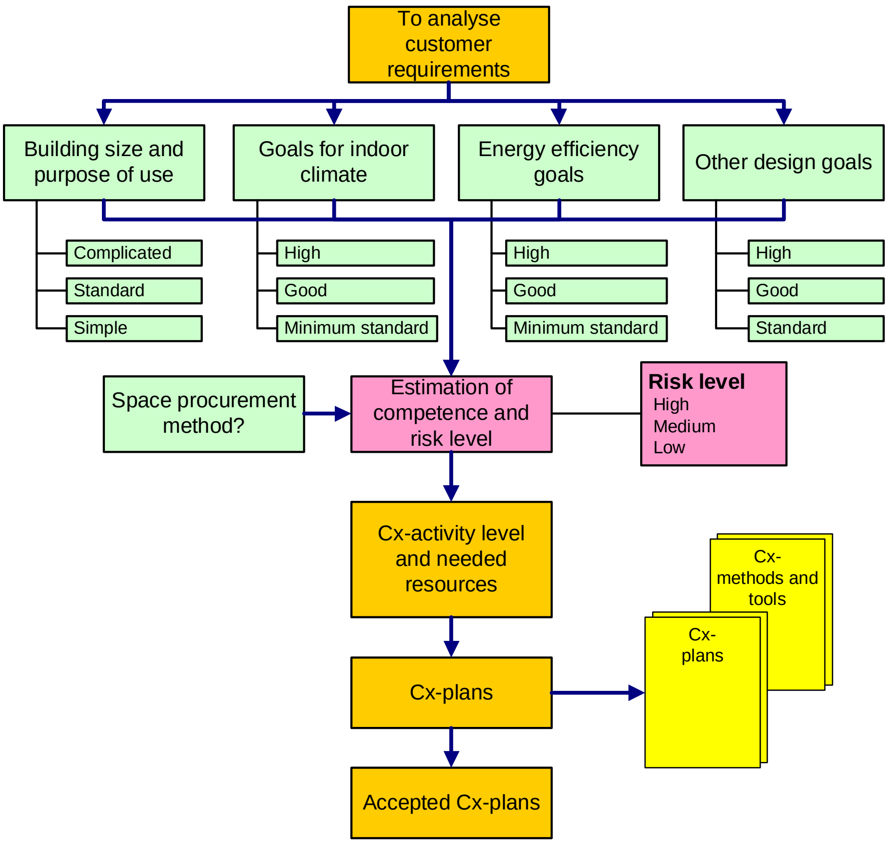

Figure 2. Preparing a Cx plan, applied from [2].

It is necessary to prepare a Cx plan according to the project phases. If on hand, general commissioning planning procedures support the Cx manager’s work. However, different building projects have different features like size, complexity, procurement method or risk profile. Altogether, the Cx manager should prepare a project specific overall plan and detailed phase plans in cooperation with the owner and the design professionals. In Figure 2, there is shown a procedure for preparing a Cx plan. There several important aspects should be taken into account.

In the implementation of the Cx activities a number of checklists and verification documents are also needed.

Cx team

Different procurement and delivery methods establish different starting points and terms to organize Cx activities. In the traditional design-bid-build projects the owner must hire also a Cx manager. In the design Build contracts the owner has only one contracting party and it is possible to transfer at least some of the Cx responsibilities to the general contractor. It's the question of the owner's strategy and decision, how detailed commissioning process he wants to implement. Table 1 gives examples for selecting the Cx manager in different projects.

According to Finnish Land use and Building code every building project must have the principal designer. Tasks and responsibilities of the principal designer are partly similar with Cx responsibilities. It is necessary to adjust the role of a Cx person with principal designer tasks. In many cases the principal designer and the Cx manager could be the same person. It's also related to the project size and complexity as well as the professional background of the principal designer.

Table 1. Options to select a Cx team manager.

Cx manager ¿ Strong role l Participation ¢	Small DBB project	DBB project	DBB project	Design & Build project	Design & Build project	DBOM project
Principal designer	l	¿	l	l	l	l
Independent Cx consult (ToVa)				¿	¢	¢
Owner s agent	l	¢	¿	¢	¢	¢
General contractor s agent	¢	¢	¢	¢	¿	l
HVAC designer	¿	l	l	l	l	l
Electrical designer	¢	¢	¢	¢	¢	¢
Construction designer	¢	¢	¢	¢	¢	¢
Automation designer	l	l	l	l	l	l
FM service engineer		l	l			¿
Supervisor	¢	¢	¢	¢	¢	¢
HEPAC contractor	¢	¢	¢	¢	¢	¢

The most building projects are rather small or medium sized. The Cx team manager may be an external consultant. On the other hand, the Cx manager can be seen as a role and he or she could also be a design professional, a FM service engineer or an owner's supervisor. In design build projects and DBOM projects the Cx manager may be a service provider's agent.

In very large and complex projects, it's a better possibility to engage the whole external commissioning team.

DISCUSSION

In the construction market some big trends point to the need of a systematic Cx process. Especially upward energy prices and increasing requirement for a good and healthy indoor air are the drivers.

The commissioning concepts and processes are not yet widely accepted and in use. The Cx concepts and procedures need to match with the actual project and quality management practices. In addition, the Cx team should match with the normal project organization. So, the key question is the role of the traditional supervisor and the building services supervisor. We cannot only add an extra Cx organization to the traditional construction project practices. The whole project management needs to be reformulated somehow.

The Cx concept has challenging goals: to achieve good indoor conditions, energy efficiency and the other owner's goals at the same time. It's necessary to implement a growing amount of pilot projects in order to prove benefit potential in practice,

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Building Performance Optimization Services for the city Boras, Sweden

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SUMMARY

In 2002 the Siemens branch office of Gothenburg and the representative of Boras started a pilot project to address how Building Performance Optimization (BPO) Services could support the city of Boras to increase energy efficiency in existing school buildings. The subject of the project was a school campus called Erikslund.

Siemens was able, in close cooperation with the management and operational staff of the school campus, to reduce energy consumption by more than 30%. These savings where achieved without major investments in equipment or changes to the building itself.

The key to success was BPO Services and making maximal use of the installed Building Automation Management System.

As a result of the pilot project the comfort and energy efficiency of a total of 12 buildings in Boras has been addressed and improved.

INTRODUCTION

A new Service was introduced in 2002 that reflected the increased interest of the market in energy efficiency, environmental responsibility and increasing energy prices: Building Performance Optimization Services. The aim of the offering was to develop a tailored service agreement together with the customer to optimize the building's performance. The offering, which is part of the Advantage Services, includes Energy Optimization, Operational and Alarm Management Services. Cost effective implementation of these services is made possible by connecting to Siemens' Advantage Operational Centre (AOC).

To give an example of how BPO Services provides value to customers, the school campus Erikslund in Boras was selected as a pilot project. In order to provide a proactive service, the branch office of Boras scanned in the 2002/2003 consumption figures of all customers with existing Siemens Service Agreements, and by benchmarking these figures, identified customer sites with high consumption. In the case of the school campus, Erikslund, the deviation between the benchmark for schools and the actual energy consumption of the campus indicated a need for more in-depth investigation into how the energy in the buildings were being utilized.

THE PROJECT

The school campus is made up of three buildings: The oldest building on the campus is Komvux, built around 1945 and dedicated to adult education. In the early 1970's the school was extended by the Erikslund building, a high school, and a gymnasium. Approximately 1'100 students are on the campus with a total heated area of ca. 12'600 m^2.

Figure 1: Class Room

Figure 2, Komvux Building

All three buildings are supplied by one central district heat substation. The majority of the heat demand of the building is covered by a radiator system; additionally most of the space is mechanically ventilated. The whole campus is controlled by a VISONK- System which is connected to a DESIGO-Insight management station. This allows Siemens to monitor and control the whole facility remotely.

Looking at the specific consumption (kWh/m^2) figures of 2002 and comparing it with benchmark figures, it indicated that how the energy was used on the campus could be improved.

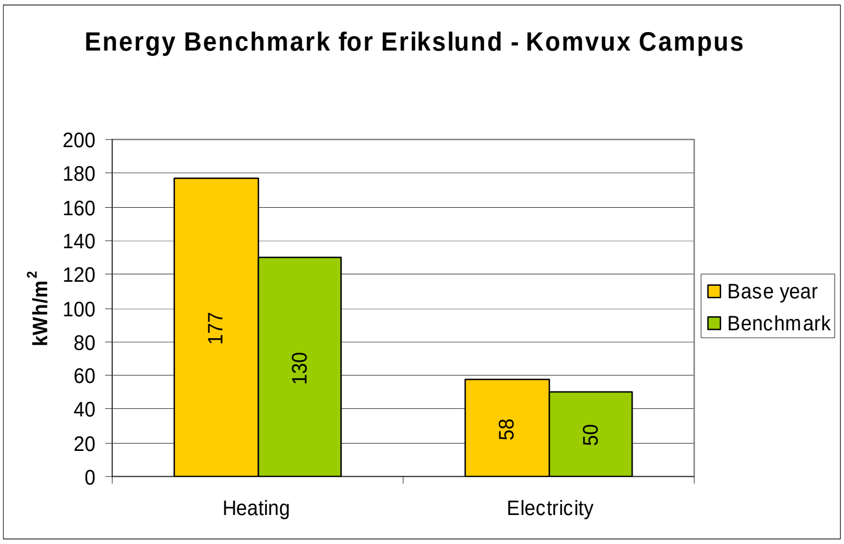

Figure 3

Based on this assessment, the available data and Siemens competence in optimizing HVAC-Systems, the responsible officials of the Boras Community were invited by Siemens to discuss the findings and to introduce the details of BPO Services. A proposal was presented how Siemens could support the owner to increase energy efficiency of the campus with the result of lower energy costs and less CO_2 emission but without effecting comfort levels. The

representative of Boras decided to commission Siemens to implement BPO Services at the school campus.
To understand BPO Services it is important to understand the concept behind it. It's based on two phases:

Phase 1: A tailor made Solution

Build a tailor made solution for the specific needs of the facility and its occupants. As described below, Siemens followed a clear roadmap together with the customer, to gather all necessary information, data and requirements.

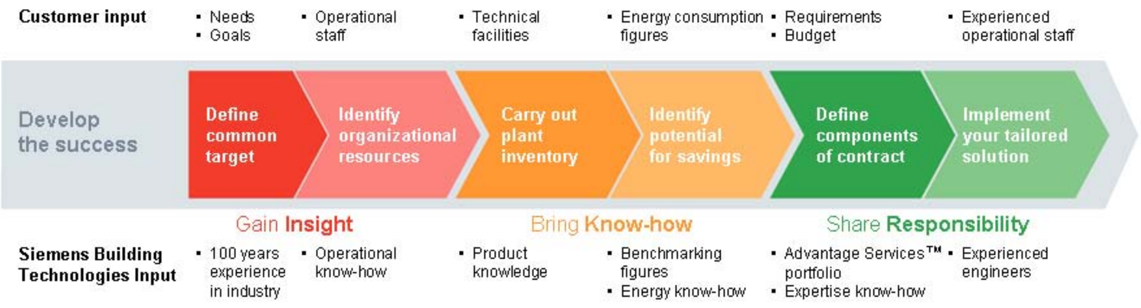

Figure 4

Meetings with the headmaster and operational staff gave Siemens insight in the operational sequences of the campus and its special needs. On the other hand these meetings were a good opportunity for the school personnel to ask questions and understand what was going to happen. At the end of this first phase both sides had a common goal and understanding.

Phase 2: Implementation

In this phase Siemens implements the solution in close cooperation with site personnel by following the process below:

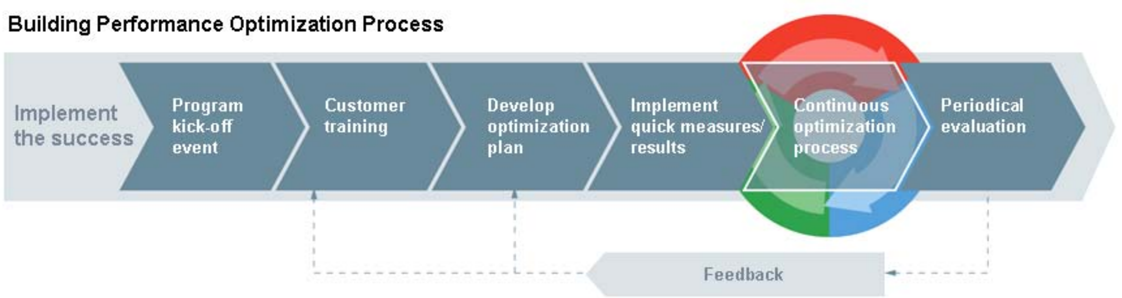

Figure 5

Before the implementation was actively rolled out, the school campus's Building Management System (BMS) was connect to the Advantage Operational Center (AOC). This is an important precondition and gives Siemens operational staff remote access to the site 24/7 to monitor and control building operation as well as helping school staff to identify and fix HVAC related problems.

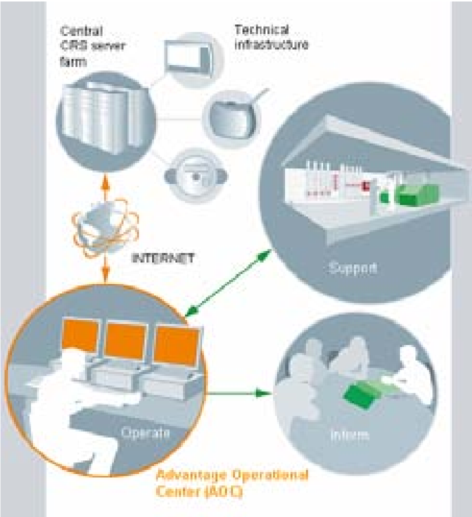

Figure 6: The AOC

After the successful connection to the site, Siemens established a metering concept with all available meters in order to monitor consumption figures. This allows monitoring of the effects that optimization measures have on the consumption and gives indices for changes of operation.
Using the AOC, Siemens staff were able to set up diagnostic reports and analyze plant operation.

In the case of the school campus room temperatures, running hours of pumps and air handling units, flow and return temperatures and the actual district heat demand was trended. An Energy Engineer together with the AOC- Operator assessed and analyzed the gathered data. With this information an on-site meeting was scheduled with the involved operational personal to share the findings and develop an optimization plan for the campus.
In a first step the Energy Engineer, accompanied by a Service Technician and the customer's operational staff viewed al the different plant rooms within the campus to have a full picture of the installation and its condition.

By sharing know-how, the findings from the analyzed data and input from the occupants, Siemens and the operational staff agreed on optimization measures written down in an Optimization Plan.

The measures which were taken can be split up in three categories:

- BMS related measures (Software)
- Organizational measures and
- Structural measures

Measures based on the BMS were to adjust control parameters, optimize set points and change time schedules to actual needed values. The basic requirement to be able to do such changes is a detailed understanding of the requirements of the specific space during a specific time. To always be up to date on these requirements, regular meetings with the headmaster were taking place to discuss needed adjustments of the control strategy.
Additionally, Siemens Engineers applied the latest control strategies to heating circuits and air handling units, e.g. the so called HEAT ECO function, Optimum Start Stop Control or the ECONOMIZER function based on the outside air enthalpy.

As examples for organizational measures, the following two are mentioned:

1. The gymnastic building was supplied by a district heating connection of some one hundred meters from the main substation of the campus to cover the energy demand for heating and domestic hot water of the showers throughout the entire year. By using the existing electric heater in the DHW tanks it was possible to switch of the district

heat connection during the summer period. This resulted in saved pumping energy and heat losses of the pipes.

2. As usual for district heat supplied sites, the customer of the utility provider has to pay a demand charge additional to the consumed kWh. This demand charge is depending on the highest value of heat that the site consumed during a certain period of time (e.g. 1h). Siemens optimized the programming of each heat consumer on site in a way that the maximum demand of heat decreased significantly and the customer now pays less demand charges to the utility provider.

Measures where smaller structural actions were needed were done as well.

BPO Services is not only taking care of optimizing BMS related equipment; a major contribution to the success of this project was the ability to look at the whole picture. Most of the class rooms in the newer building of the campus were heated by single independent ventilation units which were installed beneath the windows. A room temperature sensor controls the heating coil inside the unit and a damper the amount of outside air. During the night the damper to the outside is closed and the room set points are set back. By trending the consumption figures and after questioning the operational staff, Siemens started to check the function of the dampers. It turned out that a high percentage of the dampers were not working well, with the result that too much outside air infiltrated the space and this caused high energy consumption.

The air handling unit in the gymnastic building for example was controlled by a room temperature sensor and schedule. As is the nature of these kind of buildings where the occupancy rate varies heavily during the day, with the effect that more outside air is provided to the space than it is needed to keep the comfort level. Even in cases of very short term changes of the planed building schedule the air handling unit was not able to react to the changed conditions.

Siemens applied CO_2 and occupation sensors in order to control the air handling units based on demand rather than on a fixed set point or schedule. A close monitoring of the plant operation after adding the sensors and changing the control strategy by the AOC ensured the high performance of this measure.

After implementing the measures of the Optimization Plan the necessary data points get trended in the AOC. Each deviation form the pre-calculated budget generates alarms, which are carefully handled by the AOC Operator. Consumption reports generated on a regular basis and discussed with site personnel and the headmaster of the campus ensures long term success and gives the opportunity to extend the savings year by year. The following chart shows the development of the yearly heat consumption figures.

Discussion of the consumption figures

The monthly consumption figures for heat for 2002 (Base year) and 2006 show the savings achieved for Erikslund school campus. The weather adjusted heat consumption for 2006 was reduced by 707’000 kWh or 32% compared to 2002.

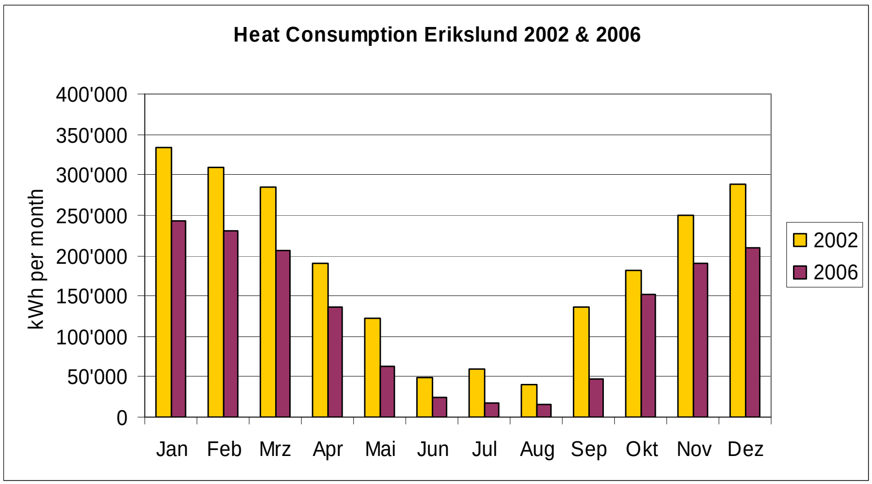

Figure 6

The chart below shows the heat consumption on a yearly base split up in a weather independent (domestic hot water) portion and in one for space heating.

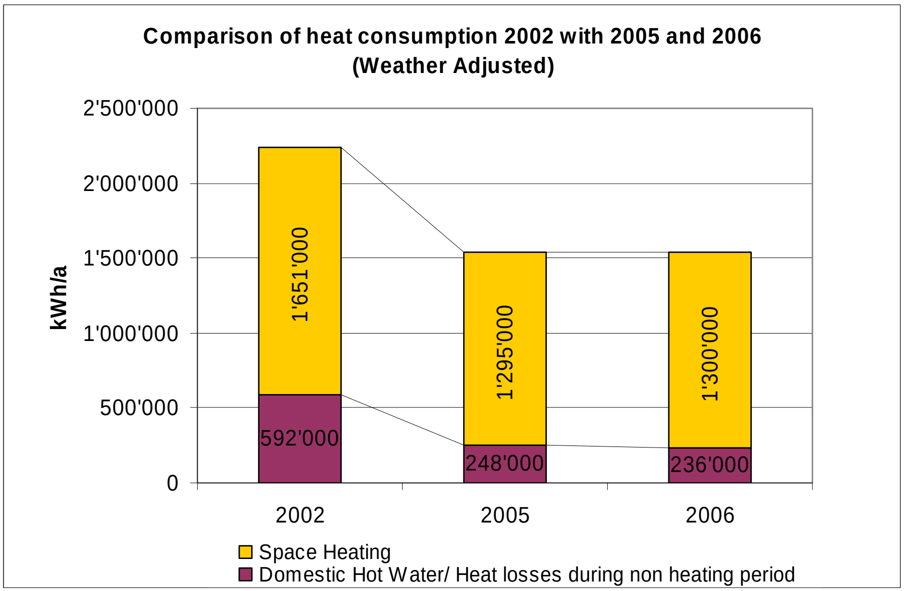

Figure 7

The figures show an interesting detail: The weather independent consumption is reduced by 60%. This is the result of changed heat supply of the gym during summer (less heat losses for district heat piping), the HEAT ECO control function that reduces heat consumption especially in between seasons, the ECOMOMIZER function of the air handling units and the adjustments of heating set points.
(The additional electricity consumption for preparing DHW in the gym is not taken into account for that graph)

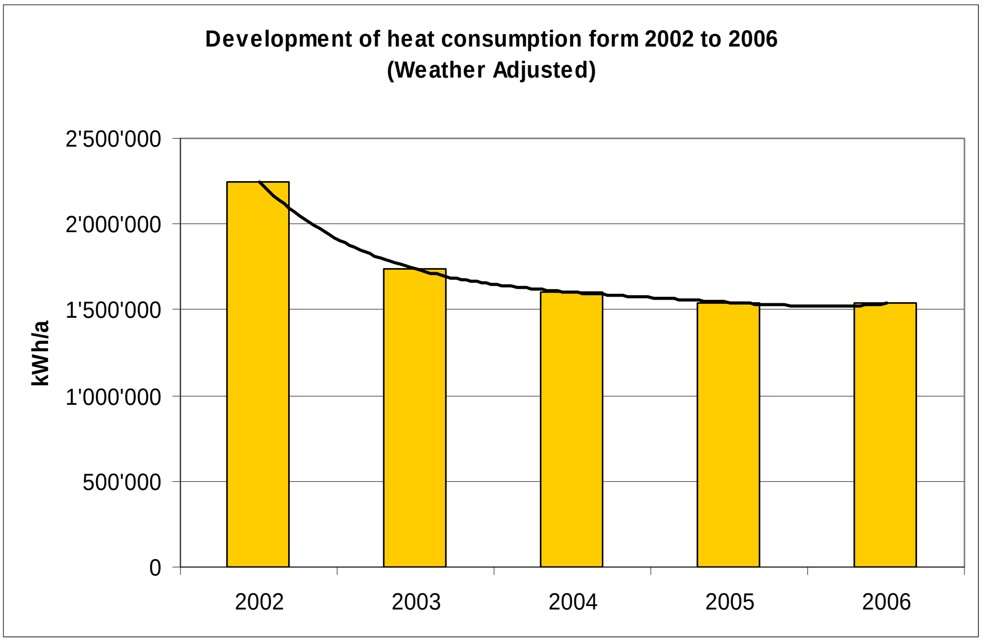

Figure 8

The chart of the heat consumption development is showing the continuing approach of BPO Services. 2002 compared to 2003 shows a big reduction in heat consumption and during the following years it was possible to prevent an increase. Due to proactive optimizing of the plant operation it was even possible to increase the savings.

It can be assumed that if the close monitoring and controlling of the site is stopped, the consumption tends to increase again. Like every process it is necessary to invest some amount of effort to keep it alive.

The development of electricity shows a similar impressive picture to the heating. The consumption is even more dependant on the actual amount of students. Komvux student figures are quite varying, whereas the high school student figures are nearly constant throughout the year.

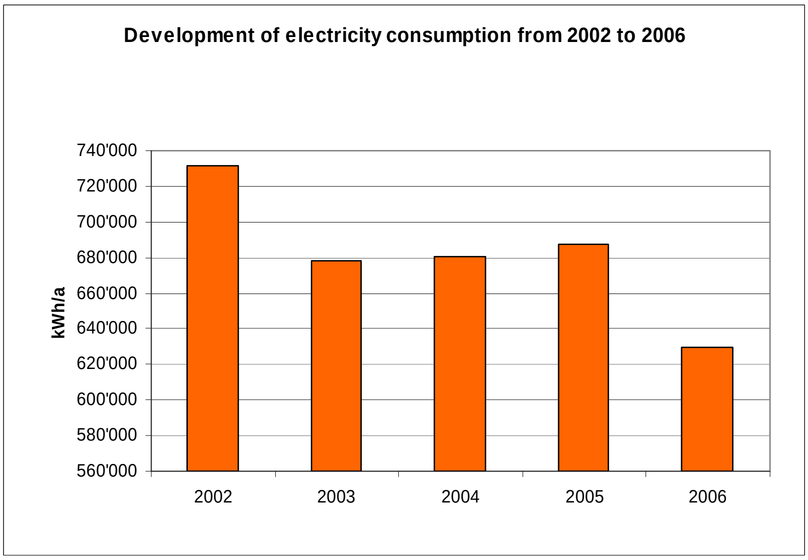

Figure 8

The savings are based on adjusted time schedules and reduced run hours of secondary equipment like pumps and fans because of a more on demand based control strategy. Important to mention is that Siemens and the headmaster of the campus meet at the beginning of each semester to adjust the time schedules to the new class schedules. The daily tracking of the schedules through the AOC is another important factor to keep the run hours at a minimum.

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Risk management for planning and use of building service systems

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SUMMARY

The design, use and maintenance of buildings and building service systems have become more and more complex, in the sense of management of information and knowledge. The systematical risk management methods and applications are sophisticated ways to evaluate and manage the risks. This paper presents three case studies for risk management. The first case describes the management of risks and success factors in building service life-cycle project, and this is application for design phase. The second case describes the systematic approach for risk management of indoor air quality of ventilation system during the use of system. The method points out the biggest risks inside the system, which may cause problems in indoor air quality, and will help to find out the most important points in the technical system, to be checked and controlled. The third case is evaluating the risks when taking the responsibility of energy consumption of the building service system.

INTRODUCTION

The design, use and maintenance of buildings and building service systems have become more and more complex, in the sense of management of information and knowledge. The systematical risk management methods and applications are sophisticated ways to evaluate and manage the risks.

When planning a new building or renovation of the existing buildings, the best possibilities to effect on the costs, performance and conditions of the building is the pre-design and design phase (figure 1). If the requirements for the building are properly set and also the possible evident risks have been evaluated, the implementation of the conditions and their matching to needs and requirements can be checked following a commissioning procedure (or, in general, mutually accepted quality control procedure) [1] (figure 2). The main reasons, why the building does not fulfil the prerequisites can be divided roughly into three parts: 1) faults in pre-design and design stage, 2) defects in implementation stages and 3) malfunctions in TAB (testing-adjusting-balancing) and in use stage. The energy performance, energy efficiency and indoor conditions, as thermal comfort and indoor air quality, depend on the proper integration of building envelope, functioning of the ventilation system, heating systems, cooling systems, BAS (building automation system) and internal and external loads (weather conditions, use, etc). The crucial matter is how the operations of these factors are integrated together, and how these factors will cohere. In the pre-design phase, the building owner should set his requirements and demands as well as possible matching the needs. The problem has been earlier, that these requirements have not been properly set. In some cases, for instance, when planning a shopping mall or commercial building, the owner probably does not know the final users, or not all of them. The needs of individual end users may vary, which would cause e.g. the use of distributed ventilation system.

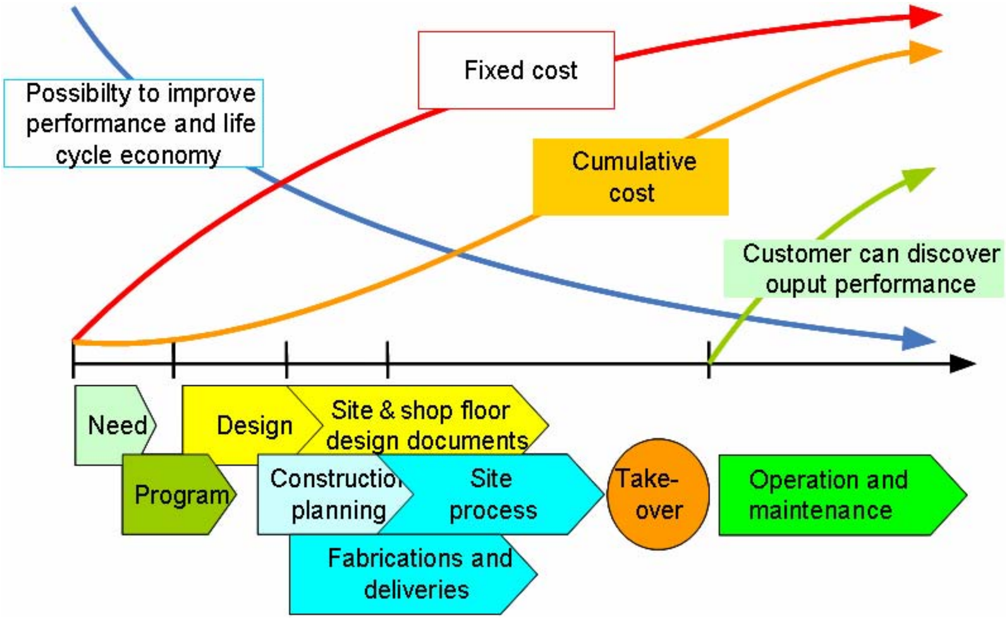


Figure 1. Building process [2] .

Different type of risks exists in the building process from design phase to operation phase. The quality and commissioning procedures are needed to support the building processes. The procedure includes the different stages of the building process. Figure 2 shows how the requirements and goals will be checked between each stages of the project [1]. Each 'diamond' contains check-lists of tasks and operations.

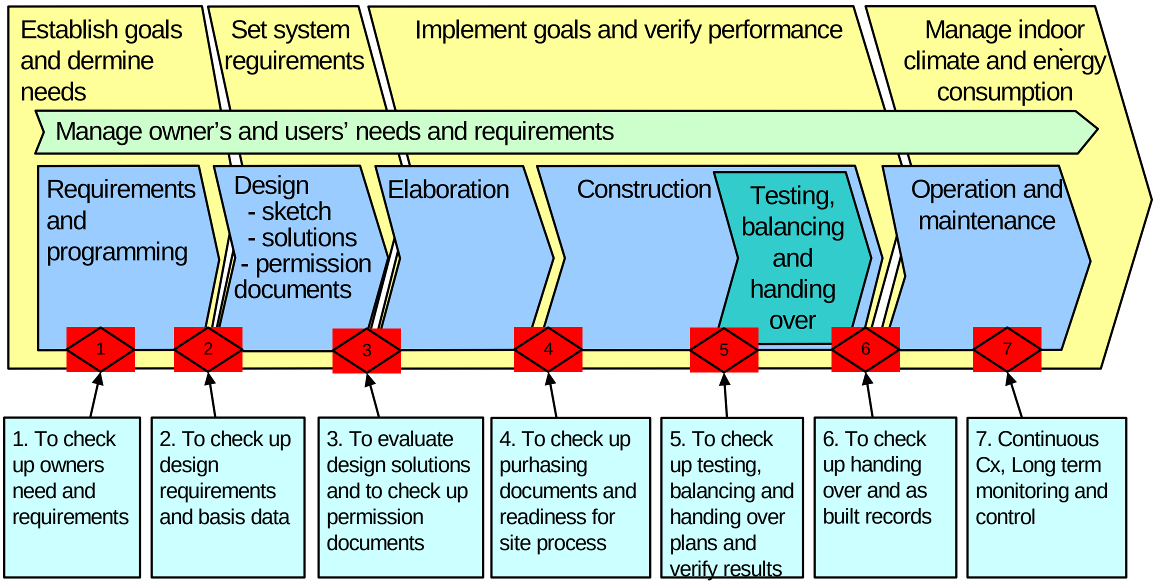


Figure 2. The phases of commissioning [1].

The objective of the paper is to present the methods and three case studies of systematical risk analysis and management, supporting the quality of the overall building process. The general systematic for identification, evaluation and sharing of risks are presented. Classification of risks, model for risk evaluation table (matrix) and risk number method are presented.

METHODS

Risk assessment

Methods in risk sharing and assessment have been developed for risk recognition and management. To assess the success factors of projects, corresponding assessment tables have been drawn up to help recognize the special features of successful projects [3].

Systematic risk recognition and assessment is the method for managing risks and pinpointing the responsible party. The stages of risk analysis are defining the target, recognizing hazards, assessing consequences, calculating probabilities, assessing the total risk, and finally, removing, reducing, and preventing the risk. The parties involved in the implementation of the project are assessed for their ability to take responsibility for the risk, and usually, the responsibility is given to the party that could handle it and to that it naturally belongs. Drafting a contract is a concrete tool for pinpointing the party responsible for risks. Through risk assessment, crucial matters to be included in the contracts are observed along with the necessary insurances and potential securities.

Classification of risks

One example of classification of risks is presented in [5]. Country related risks depend on the situation in the county, including economic risks, political risks and risk related to natural conditions. Implementation risks are risk caused by the realisation of the case, including technical, financial, contractual (caused by the juridical content), personnel risk (caused by actions of employees) and risks caused by methods of implementation. Force Majeure risks are unforeseen risks, which are not possible to remove.

In the project CUBENet [11], the risks were classified for practical building project. Implementation risks include source information, and quality of this information (e.g. conditions of structures in renovation project), technical risks in construction work, investment costs, timetable of construction work and quality of technical plans. Risks related of way of action during contract period include quality source information (e.g. energy consumption of previous years), availability and cost of finance (including interest rate), guaranteed savings and commissioning, maintenance costs (including changes in energy price and cost of maintenance investments). The risks related to partners of the project include economic status of parties and organisational issues (e.g. changes in personnel).

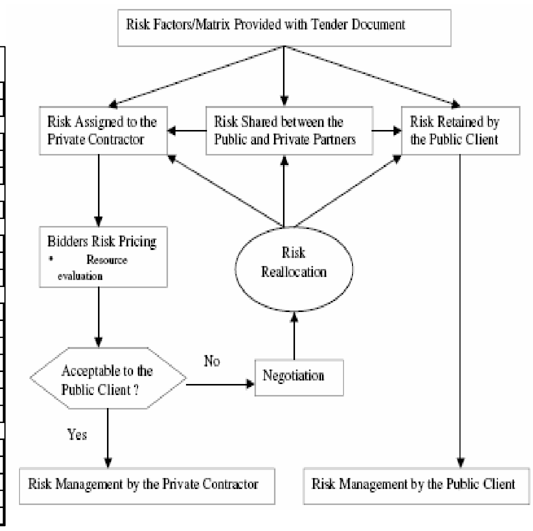
Risk assessment in planning phase – risk matrix and success factor methods

One practical method for systematical risk assessment is a risk sharing table (or check-list, or matrix), figure 3. The potential (along with apparently impossible) risks are listed and shown to the project parties. Reference [4] presents the categorisation or risks in macro, meso and micro levels (‘meta-classification apprach’). Reference [5] shows the practical risk evaluation and allocation table for PPP (Public-Private-Partnership) project in Finland. With the risk sharing table, the parties share the responsibilities for the risks each party will assume. The responsibility for an individual risk is left to either party or to both parties according to principle agreed by parties. The client assesses the risks of the project and assumes responsibility for some of the parties involved in the project. In the invitation for bids, the bidder is asked to pay attention to the risks transferred to the provider and to describe how they will assume responsibility for the risks or will take these into account in the bid. In practise, the provider may not necessarily take the responsibility of all risks offered but will transfer the risk back to the client in the bid. The responsibility for a risk will naturally have influence in pricing – the party taking the responsibility of the risk is setting price for the risk.

The risk assessment table helps the bidders in evaluating how they accept the risks the client is transferring to the provider. The table forms a foundation for the bidders' proposition in order to show how well the bidders understand the nature of the risk, how effectively the bidders forward or share the risks in the subcontractor chain, and how the bidders minimize the unfavourable effects caused by risks in relation to the client and the end users.

. Macro level risks	Risk allocation		
	Client	Bidder	Shared
Political and government policy			
_ Unstable government			
_ Expropriation or nationalisation of assets			

B. Meso level risks	Risk allocation		
	Client	Bidder	Shared
Project selection			
_ Land acquisition (site availability)			
_ Level of demand for project			
Project finance			
_ Availability of finance			
_ Financial attraction of project to investors			
_ High finance costs			
Residual risk			
_ Residual risks			
Design			
_ Delay in project approvals and permits			
_ Design deficiency			
_ Unproven engineering techniques			
Construction			
_ Construction cost overrun			
_ Construction time delay			
_ Material/labour availability			
_ Late design changes			
_ Poor quality workmanship			
_ Excessive contract variation			
_ Insolvency/default of sub-contractors or suppliers			
Operation			
_ Operation cost overrun			
_ Operational revenues below expectation			
_ Low operating productivity			
_ Maintenance costs higher than expected			
_ Maintenance more frequent than expected			



a) b)

Figure 3. a) An example of risk allocation table [3] and b) Flow-chart of negotiations [4].

Critical success factors and sub-factors have been analysed based on method in reference [7].

Risk management during use – risk number method

After the construction phase in the project has been completed, risks are also managed with the help of the information system, such as building management and monitoring system (BMS). The functionality of the technical system is monitored and measured, paying particular attention to the factors sanctioned in the contracts. The monitoring system can contain risk assessment tools which focus the monitoring and service to the risk points of the process. Disadvantageous factors of the various parts of the system are assessed based on three factors [6]. The probability of occurrence (showing how probable is the occurrence of factor) and probability of discovery or detection (showing how easy to discover or detect the factor is) as well as the severity (showing how harmful this factor is) of the disadvantageous factor are assessed separately on a scale of one to ten (or some other selected scale). The product of these three factors is the total risk number (example in table 1). The higher the risk number, the higher the risk caused by the disadvantageous factor. When the risk points have been recognized, the measures can be focused: the risk points of the AC system can be handled with the monitoring system (critical monitoring points), and service can be targeted to the system risk points. Additionally, measurements can be utilized in defining the numerical values (severity based on measured value). Moreover, applications can be developed to assess risks in energy consumption, which is crucial in some contract models, e.g. in energy service contracts.

Table 1. Criteria of the drawbacks.

Severity, S	Numerical value	Probability of detection, Pd	Numerical value	Probability of occurrence, Po	Numerical value
No influence (Category 1)	1	Drawback always detected	1	Occurrence of drawback improbable	1
Category 2	2 ,3	Category 2	2 ,3	Category 2	2 ,3
Category 3	4,5,6	Category 3	4,5,6	Category 3	4,5,6
Category 4	7,8,9	Category 4	7,8,9	Category 4	7,8,9
Health risk	10	No detection of drawback	10	Very probable	10
Severity S Describes the seriousness of the examined drawback factor significance from the point of view of the observed feature		Probability of detection, Pd presents how probable it is to detect the drawbacks by other means than by measuring		Probability of occurrence, Po describes how probable the occurrence of the examined factor is in the system	
The result of these three factors is total risk number: $RPN = S \times Pd \times Po$					

RESULTS

Case study 1 - risks and success factors of ESCO contracting models

New procurement models and service concepts, e.g. energy service concepts, have been developed actively. PPP (Public-Private-Partnership) and PFI (Private-Finance-Initiative) models have become more common in areas of transportation, health-care, power and energy and buildings [8]. The experiences of these projects and possibilities for Finnish applications in construction sector have been reported ([9], [10]). The concepts for building services, e.g ESCO/EPC-services, have been presented in [11]. The first case study describes the management of risks and success factors in building service life-cycle project, e.g. ESCO. The main focus is in the planning phase. These concepts need risk management and risk sharing between client and energy service companies. The sharing of risks will be agreed in practice with contract documents. The risk and success factors have been evaluated based on risk allocation matrix (example in figure 3), success factor tables and interviews. The interviews were done separately for clients and service providers.

The following factors were evaluated the most important risks for the client in ESCO services: 1) control of timetables, 2) success in demolition work, 3) discovery of additional savings and 4) re-utilisation of old devices with new ones. The main risks for the service providers are: 1) initial information (given by the client), 2) trust and familiarity with the client, 3) success and correctness in planning, 4) success and fluency in construction works, 5) identification of ‘grey areas’, which are not easy to control based on the available information, and 6) knowledge of contractual issues.

The main success factors for client using ESCO model are: 1) the traditional ESCO is possible without investment, 2) in case investment is needed for construction work, the investment cost is smaller than in case of traditional, 3) service contracts guarantee extra savings and higher quality of buildings, 4) after 1st contract period, competition between service providers is possible, 5) in case savings are not realised, this is decreasing the fee for service provider. The main success factors for service providers are: 1) fluency in cash flow due to long contract period, 2) investigation of correct basic information, 3) success in evaluation of risks and forwarding the risk aspects in contracts.

In the analysed ESCO projects, there were no big conflicts between the opinions in sharing the risks between the client and service provider, therefore possibilities for successful sharing of risks exists. The concept of risk sharing table will help the discussions about existing risks and will speed up the contract negotiations. The concept of success factors was seen as a tool for strategic planning and selection of successful models for delivery, not as a tool for single building project.

Case study 2 - risk management of indoor air quality of ventilation system during the use of system

The second case describes the systematic approach for risk management of indoor air quality of ventilation system during the use of system. The effects of the component level performance on the system overall performance and existing risks were evaluated using risk number approach. The method points out the biggest risks inside the system, which may cause problems in indoor air quality. This approach will help to find out the most important points in the technical system, to be checked and controlled during the use and maintenance. Figure 4 presents the ventilation system, table 2 evaluation criteria and table 3 risk numbers of the drawbacks [12]. The number values are based on expert opinions, and should be updated by experiences. In this case, the control and service should be focused especially on pre-filter and supply air duct, to guarantee the high quality of the ventilation air.

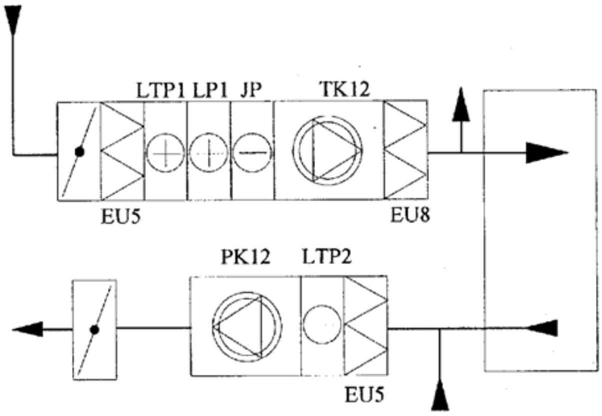


Figure 4. Ventilation system in case study.

Table 2. Evaluation criteria for risk analysis in case study.

Severity, S	Numerical value	Probability of detection, P_d	Numerical value	Probability of occurrence, P_o	Numerical value
No influence on quality of supply air	1	Always detected/noticed	1	Improbable	1
Small decrease in quality of supply air	2 ,3	In most cases detected	2 ,3	Small	2 ,3
Decrease in quality of supply air	4,5,6	Sometimes detected	4,5,6	Possible	4,5,6
Bad quality of supply air	7,8,9	Seldom detected	7,8,9	Probable	7,8,9
Bad quality of supply air has influence on health	10	No detection of drawback	10	Very probable	10
Severity S describes the seriousness of the examined drawback factor significance from the point of view of the observed feature		Probability of detection, Pd, presents how probable it is to detect the drawbacks by other means than by measuring		Probability of occurrence, Po, describes how probable the occurrence of the examined factor is in the system	
The result of these three factors is total risk number: $RPN = S \times P_d \times P_o$					

Table 3. Risk numbers for drawbacks in ventilation system case study.

Part of system	Drawback	Severity S	Probability of detection Pd	Probability of occurrence Po	Risk number RPN
inlet grid (at roof)	smell	5	6	4	120
inlet chamber	microbes	4	8	8	256
prefilter (EU5)	microbes	4	8	7	224
	VOCs	5	8	6	240
	fibres	2	9	4	72
	particles	7	9	8	**504**
	leakage	4	4	4	64
heat recovery device	not evaluated				
heating coil	VOCs	5	8	3	120
cooling coil	microbes	5	8	2	80
supply air fan	VOCs	5	8	3	120
fine filter (EU8)	microbes	8	8	5	320
	VOCs	5	8	5	200
	fibres	4	9	2	72
	particles	7	8	4	224
	leakage	8	4	2	64
supply air duct	microbes	8	8	7	**448**
	VOCs	5	8	5	200
	particles	7	8	6	336
control dampers	not evaluated				
material of sound damper	microbes	8	8	5	320
	fibres	4	8	8	256
Air inlet valves	particles	8	4	8	256

Case study 3 - evaluation of the effects of the quality of building on energy consumption

The third case deals with evaluation of the effects of the quality of building on energy consumption. The quality of structures and HVAC is evaluated from point of view of effects on energy consumption. Case simulations and sensitivity analysis can be added to risk evaluation procedure. The energy performance of the building is depending strongly on the efficiency of the ventilation systems. The example shows the influencing factors and ways to reduce heating energy consumption. Three factors are considered: 1) air flow rates and running times of ventilation, 2) heat recovery of ventilation and 3) air tightness of the building envelope. This case is evaluating the risks when taking the responsibility of energy consumption of the building service system.

Figure 5 shows an energy balance of a building in cold climate conditions. In this example the biggest losses are caused by ventilation. If building owner or facility manager has set targets for energy consumption, they must have possibility to monitor the distribution of energy. By generating energy balance chart (e.g. Sankey-diagram) it is possible to observe, which are the main factors influencing on heat and electricity usage.

Even in well-performing buildings the energy consumption may vary within relatively wide range. The estimated consumption of district heating (DH) of a new school in Southern Finland was 26,3 kWh/m^3/year (real building, calculations based on Finnish Building Code, part D5). The biggest component of heat losses is ventilation 25,7 kWh/m^3. Changing the operation time from daily 12 hours to 10 hours decreases the heating energy consumption to 23,1 kWh/m^3. Increasing the efficiency of heat recovery device from 50 % to 60 % decreases the heating energy consumption to 20,1 kWh/m^3, and finally improving the air tightness of the building from 0,1 change/hour to 0,05 change/hour decreases the heating energy consumption to estimation 18,1 kWh/m^3. This decrease of 32 % in heating energy consumption (DH) is realised by optimising operation hours, selecting more energy efficient heat recovery device and realising the air-tight building. This case simulation is showing the influence and risks of these factors on heating energy consumption ('severity' is presented as energy consumption). The most important influencing factors should be analysed in a systematic way showing the influence of the factor on consumption. This analysis could be presented in form of table or Sankey-diagram including basic value and sensitivity to influencing factors (e.g. heat recovery device has influence 0,3 kWh/m^3/1 % change in efficiency).

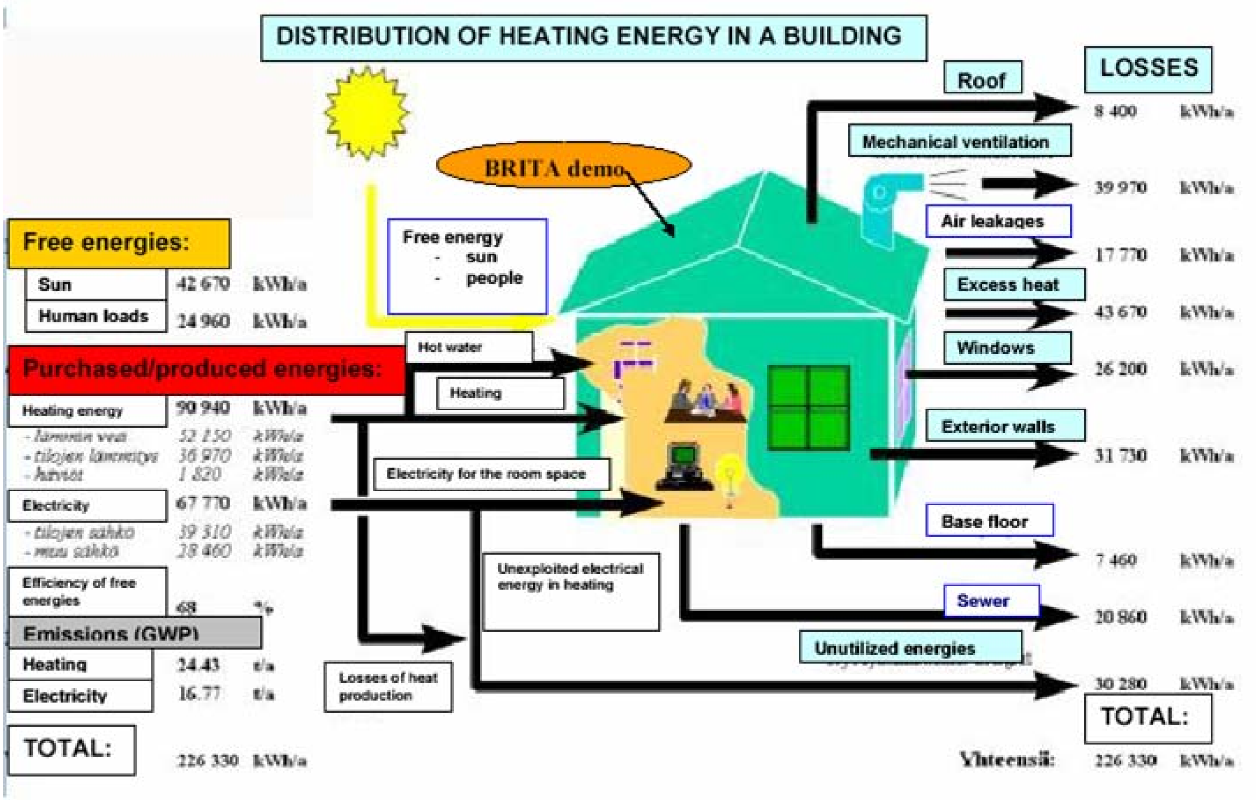


Figure 5. An example of the energy balance of the building.

DISCUSSION

The systematical risk management methods and applications are sophisticated ways to evaluate and manage the risks during the life-cycle of the building and the associated technical systems. Three presented cases show possibilities to utilize methods in pre-design, design and operation phases. The risk table approach is a checking-list type of method, which is a good tool for assessment of all the possible risks existing in the building project. The risk table can be used as part of invitation for bids, annex of contract documents or check-list during the negotiations. The risk number approach is a simple tool for pointing the drawbacks or risk components of the complicated systems. The strength of the risk number approach is evaluation of drawbacks based on three sub-factors (severity, probability of detection and probability of occurrence), which all are evaluated separately, based on experts opinion or measured values. This is giving more confidence on evaluation. The simulation tools can be used as part of risk assessment, when evaluating the effects of the quality of the building structures and HVAC components on the energy consumption and costs.

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Methods and Metrics to Control Energy Use, Indoor Climate and Life-Cycle Costs of Buildings

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SUMMARY

According to several studies there is a remarkable potential for decreasing energy consumption of buildings by optimization the use of the building services and systems.

The energy consumption, indoor climate and life-cycle costs of the buildings are dependent on the performance of the building services and systems. Development of information and communication technology, building automation and simulation tools have created good opportunities to improve control and monitoring techniques of buildings.

INTRODUCTION

Simulation models are often used in designing building services. Advantage is not usually taken of existing simulation models in the operation and maintenance phase of the building. Model has to be revised due the changes during construction phase. The calibration is needed to update the information to the simulation model concerning actual usage of the building. Calibration has to be cost effective before it can be widely used. The building automation system should be used as much as possible to aid the calibration process.

The aim of the study is to develop the methods to calibrate and to use the simulation model in the operation and maintenance phase of the building. This research will focus on studying newly constructed buildings, where information and simulation models have been utilized in the designing process.

The methods to be developed will be tested in 4 pilot buildings in the near future. In this paper the state-of-art situation of the project is presented.

METHODS

Building Functionality Indicators and Measurements

Building functionality metrics is a problematic area of study but often a manageable one with right tools and sufficient amount of expertise.

The functionality of a building and building services can be described with a set of verified indicators by which for example energy consumption, indoor air quality, reliability of operation and life cycle costs can be inspected. The need to inspect building functionality can be justified with the difficulty of determining the state of operation or possibly trying to improve the performance of a building without first inspecting the current and time dependent performance of the building [1].

There are a number of different facility services such as cleaning services, catering services etc. This study focuses mainly on building services and HVAC-systems.

Means by which the state of operation can be monitored are for example tracking energy consumption time wise, benchmarking (referring to similar buildings) or defining and using target levels. Often it is wise to combine these aspects to obtain better results. The importance of tracked history data on building functionality cannot be emphasized enough.

Indicator levels can be divided into sub-groups [2]. The most important of which are
- Building-scale inspection level (e.g. building energy consumption)
- System-scale inspection level(e.g. specific fan power for the air handling units)
- Component-scale inspection level (e.g. COP for the water chillers)
- Independent variables (e.g. building usage, the weather)

Indicator types on the other hand can be divided into
- Energy-indicators
- Indoor air indicators (e.g. PPD and PMV of a building space)
- Money indicators (e.g. net present value of an air handling unit (AHU) with initial investment and operation costs taken into account)
- Environmental indicators (e.g. environmental impact of oil-heating compared to district heating
- Overall efficiency (e.g. overall amount of energy to maintain a certain level of indoor environment, can be used as benchmarking tool)

Building functionality indicators can be applied in numerous ways. Two dependent variables can be compared in order to define saving potential in operating costs. E.g. building energy consumption can be plotted as a function of building specific energy consumption.
From Figure 1 it can be seen that the buildings in the upper right side of the diagram have large energy consumptions and large specific energy consumptions. Therefore these building have large saving potentials and it is cost-efficient to focus on the energy economical improvement of these buildings. This so called functionality fourfold table is an illustrative representation by which also state of operation in general can be monitored.

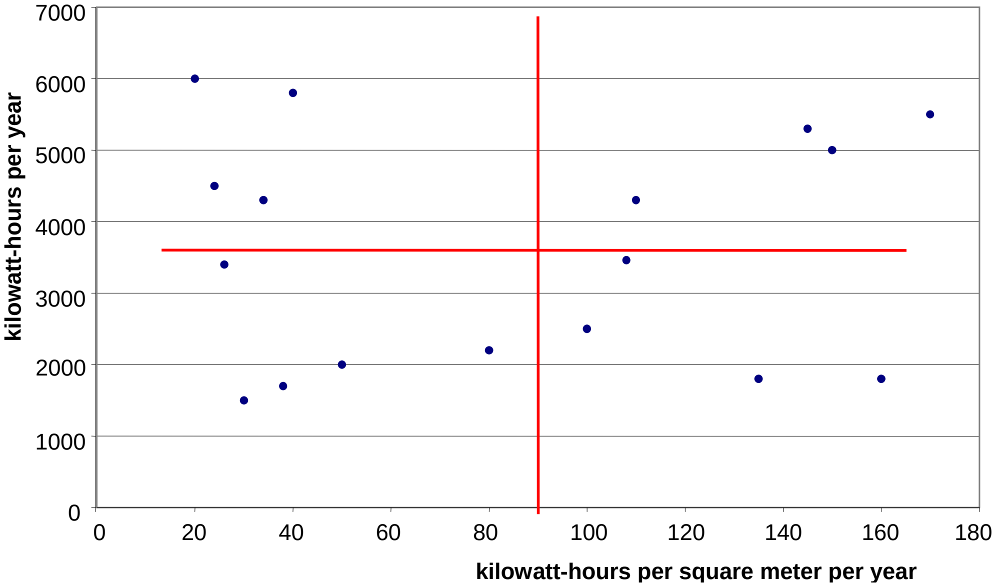

Figure 1 Heating energy consumption vs. specific heating energy consumption

Dynamic target levels with real-time measurement data from the building automation system (BAS) can be utilized, since e.g. the target level of indoor air temperature is not the same throughout the year. User queries integrated into the occupant's intranet can be a powerful tool in trying to restrict the indoor air quality problem zones in the building.

It is often possible to use the BAS in order to obtain functionality data and indicators. However the BAS being insufficient can independent measurements be used.

Whole Building Simulations

Office buildings are usually modeled with simulation programs. Interoperability with Information Foundation Classes (IFC) makes building of the model easier and faster than earlier, especially to create building geometry, when the model was composed using inadequately developed user interfaces. The simulation model is seldom used in evaluation the performance of building in maintenance phase. Simulation model should be calibrated to use it in the performance evaluations.

The simulation model can be composed with several ways; the building can be modeled using single calculation zone or the rooms can be modeled separately. Zones can also consist of several rooms. Less important rooms like corridors and entrance halls can be combined into one space. Windows, electric devices etc. can also be combined in construction of the simulation model, to shorten the execution times. Influence of coarseness of model will be studied in the project.

Nowadays, only the building geometry can be taken from IFC-model. In the future, also the HVAC-components will be included in IFC-model. The work to implement the HVAC-components in IFC is under development in IAI and in LBNL [3].

Calibration of the Simulation Model

Calibration of the simulation model is explained in ASHRAE Guideline 14 [4]. Calibration should include the following actions

1. Produce a Calibrated Simulation Plan
2. Collect Data
   - Obtain Building Plans
   - Collect and Review Utility Data
   - Prepare for Data Collection as Defined in the Simulation Plan
   - Conduct On-Site Surveys
   - Interview Operators and Occupants
   - Conduct Spot and Short-Term Measurements
   - Collect Weather Data
3. Input Data into Simulation Software and Run Model
4. Calibration of Simulation Model Outputs to Measured Data

As shown above, the calibration procedure seems time-consuming and difficult. Collecting the needed information takes most of the time used in calibration. A way to shorten the calibration process could be replacing a part of the calculation parameters with default parameters.

Collecting data is the most time-consuming work in the calibration. Design data has to be gathered from several documents in normal case. Information of the building core is in construction plans, pipes and ducts are in HVAC-plans, performance values are in maintenance manuals, and weather data have to be requested from meteorological stations, many parameters need in-situ surveys and occupant interviews. Internet based maintenance manual would help the information gather, if all the information was transferred there.

When all the information is gathered, construction of the model can be started. There are many ways to make the input work. For example volume flows of air handling units may be given in average values, or in scheduling values of them to the model. Even measured volume flows can be fed to the model. The internal gains can be given in average values per floor-area or by modeling lighting and other electric equipments with certain input power combined with the time tables. The utilization factor of the building has found out to be one on the most difficult indicator to specify. It could be defined by measuring CO_2 or moisture emissions balance on the supply and exhaust air. This will be tested in out project.

The measured output values can be verified with the calculation values. Two most important values to compare are specific heat losses of building envelope and ventilation and heat loss of air infiltration.

Specific heat losses can be calculated from measurements from the BAS. Also the air infiltration has to be taken into account. The measurement can be conducted, when supply or exhaust fan is off, measuring pressure difference between outdoor and indoor air simultaneously. The measurement can be done e.g. in one ventilation zone of the building.

Building Information Models

The use of Building Information Models (BIM) is rapidly increasing. Building service companies see the potential in information technology because it reduces re-work and makes the building models more reliable, and reduces building and maintenance costs [5]. When the shell and the zones of the building are once planned by an architect, it is sensible to utilize the model in planning later. It shortens the modeling time and makes the models more reliable and easier to maintain.

Another way to use building information models is to use it as a information databank. The monitored performance assessment results of the building can be saved to the information model where it can be accessed whenever it is needed. Unreliable paper documentation is not needed and the information is up-to-date more likely than with the paper copies.

The performance of the building services should be evaluated regularly during the life cycle of a building. The first evaluations are done in design phase when the HVAC-components are selected and the use of the building and indoor climate of the zones are planned. The next evaluation is done during the building is taken into use. This is usually called commissioning, but in broader sense the term can mean all the performance assessments done during the life span of the building.

The use of the information models is tested in several projects in evaluations of the building performance in National University of Ireland and Lawrence Berkeley National Laboratory in USA [6-12]. The core of the information model is IFC, but the evaluation results can be saved in broader information model, which consists of measured and calculated performance data. These “additional parts” can be tied to the IFC-model using programs, which have an access to IFC-model and which can reprocess it. The time-series are normally saved in XML-format or other data format. IFC-compatible C++/Java/etc. programs can be used to reprocess the data and to visualization of it. The middleware programs were used because all the simulation programs used in USA are not interoperable with IFC.

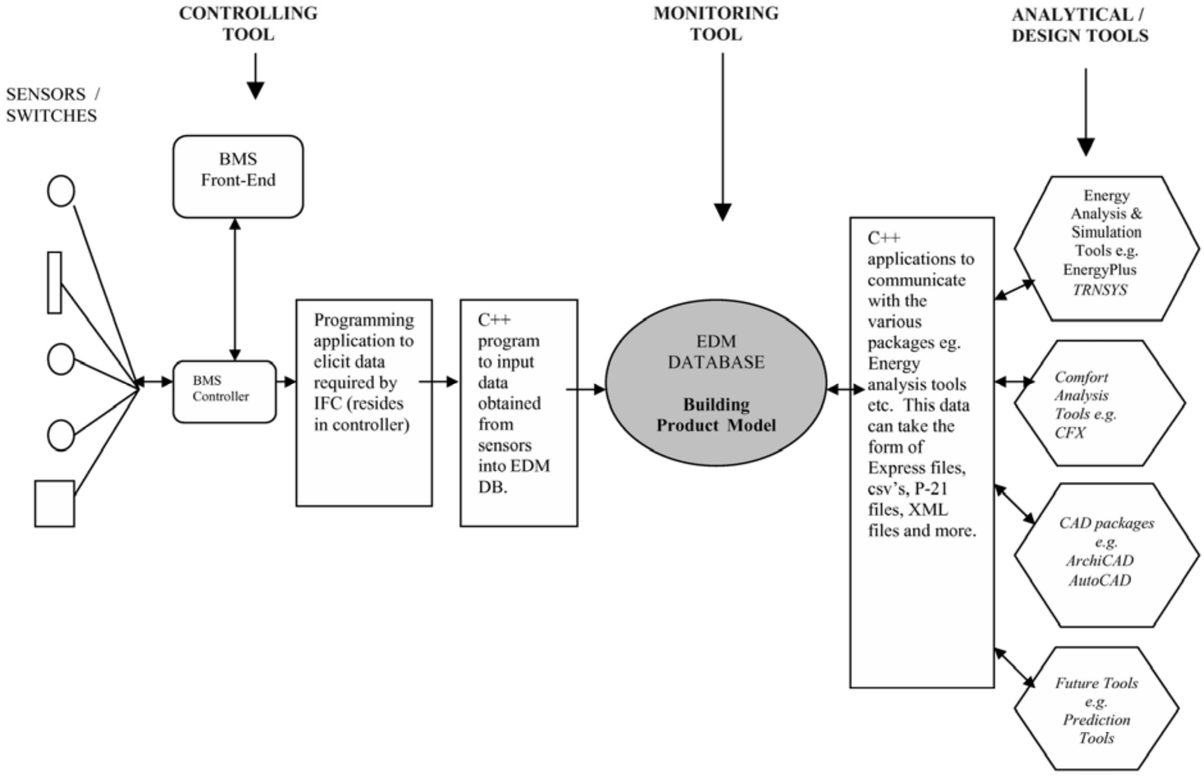

Figure 2 Use of building information model in evaluation of the real and simulated building performance [10]

Figure 2 presents, how building information models can be used in evaluation of the building energy performance using building information models and simulation programs [10]. Express Data Manager 4.5TM (EDM) is a program, which is used to save output values from the building automation and simulation program and to verify measured and calculated performance metrics e.g. the total energy consumption of the building.

Utilizing Building Automation System

BAS is seldom utilized as effectively as possible in maintenance of the building HVAC-systems. That is partly because of great amount of information in the database of BAS, which is not usually post-processed for advanced use. The building automation systems usually save the information in frequency of couple of minutes, which is too short a time concerning energy and environmental analysis. Hourly average values would be more usable in simulations and verifications of the measured and calculated values.

Many metrics can be calculated from information of the BAS. An efficiency of the heat recovery system, the COP of cooling system, the specific fan powers of fans can be calculated easily − if the measurements are designed well. The planning of the in-situ measurements should be done carefully. For example the efficiency of a heat recovery unit needs measurements of the air-side volume flow as well as temperatures before and after the recovery system. The COP of a cooling system needs measurements of the electricity use for cooling and measurements for cooling energy of the rooms and the ventilation system. The specific fan powers can be calculated if the volume flows and electrical power consumption of the AHU is measured.

Figure 3 presents the measured, simulated and benchmarks metrics. They are not identical because they are based on different parameters, simple and advanced models and the actual use of the building is seldom identical to the design parameters. Comparison of the simulated and measured values should provide a way to evaluate total performance of the building or any part of HVAC-system.

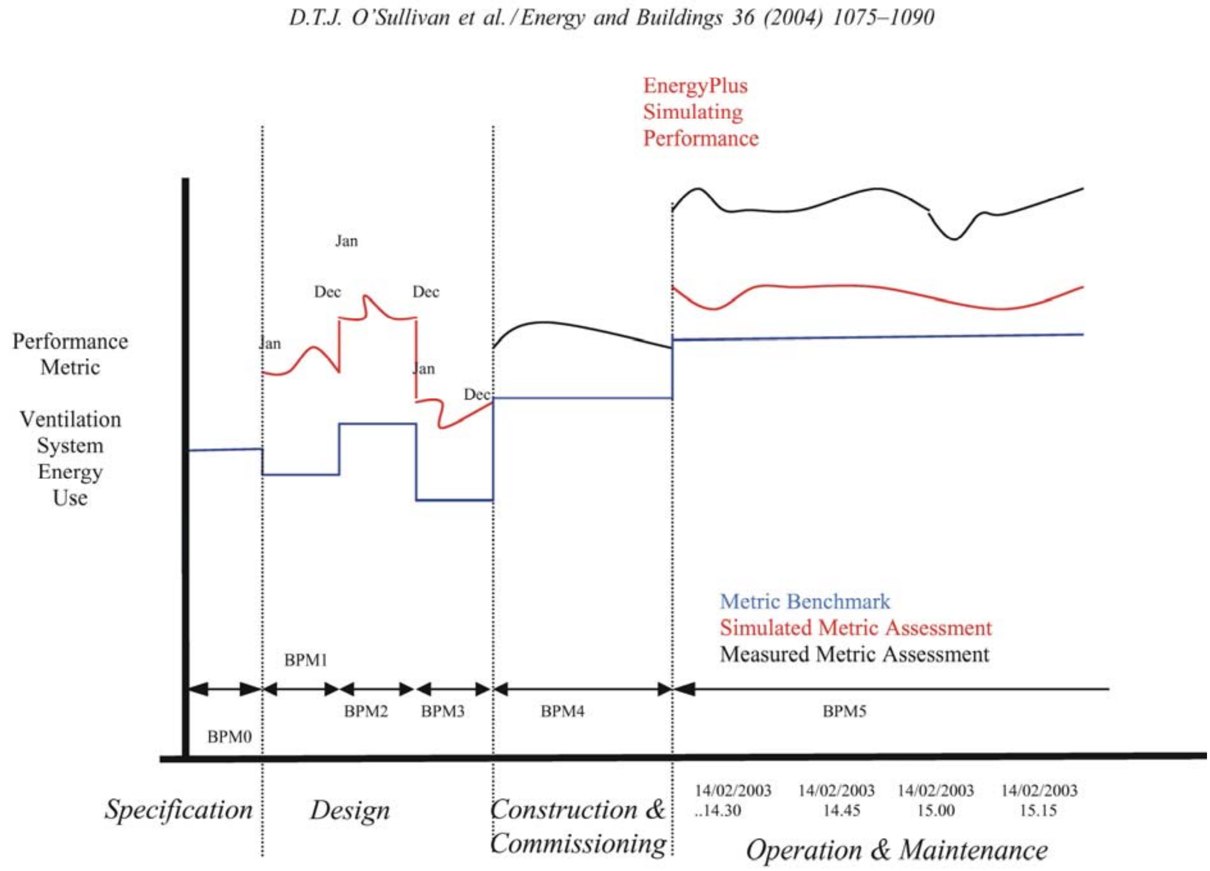

Figure 3 The evaluation of the performance during the life span building of building [10].

RESULTS

A school building was studied concerning indoor climate and energy use. Some of the measurements were done concerning calibration of a simulation model. The results of the building automation are also available.

A specific heat loss of the envelope and of the ventilation of the building was measured using manometer and temperature measurement of the heat transfer liquid. The manometer was installed on to the control valve of the network. The temperature of the entering and the returning liquids were monitored at the same time. Also the outdoor temperature and relative humidity were measured using a self-functioning combined temperature and RH logger.

The ventilation volume flows were also measured using pitot-static measurements and pressure drop measurements of the control units. The volume flows of the supply air were measured in all the five main air handling units of the building.

The tightness of the building envelope was also evaluated using supply fans. The pressure difference between the indoor and the outdoor air was measured and the volume flows of the supply air and the leakage through the return ducts were measured at the same time.

The indoor climate was measured concerning indoor temperature, operative temperature, and draft in two class rooms.

The CO_2-concentration of supply and return air was measured in one of the AHUs. That is because the method will be tested to determine the utilization degree of the building. There is not any easy way to measure the heat gains from people in the school building.

All the measurements are useful, when the simulation model will be calibrated. Especially volume flows and tightness of the building influence a great deal on the heat use of the building. The calculated and measured specific heat uses can also be verified to be sure that the calculation model is correct concerning the heat losses.

The measurement and the simulation results will be available later.

DISCUSSION

The functionality of a building and building services can be described with a set of verified indicators such as energy consumption, indoor air quality, reliability of operation and life cycle costs. The need to inspect building functionality is apparent due to well-known difficulties in determining the state of operation or improving the performance of a building. Building functionality indicators can be applied in numerous illustrative ways. Often it is possible to take advantage of the BAS in order to obtain functionality data and indicators.

The evaluation of the building performance for the maintenance purposes needs systematic methods to make it cost effective. The building information models and simulation programs are used increasingly in design of the building. They could also be used in the maintenance of the building for database to save of simulations and measurement results. They can also be used to visualize results and to make verification of simulated and real performance and indoor climate more effective.

Use of the building information model (BIM) makes modeling easier and faster. Construction of the whole building simulation model does not need so much manual feeds, so there is more time to check, that calculation parameters of HVAC-system are correct. In the future, the control parameters of HVAC-system can be transferred directly to the simulation model using HVAC-IFC, which is under development.

Calibration of the simulation model is needed, if it is used as reference value to measured performance energy and quality indoor climate. Design information, weather data, building automation and is-situ measurements are needed in calibration of the simulation model. If calibration is systematically done, it should be done in some weeks, and the model can be used in the energy performance analysis for decades. The calibration can be done more automatic, if some of the building automation results can be used directly as input of simulation.

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A Life-Cycle CO_2 Assessment Procedure in Refurbishment of Old Non-Domestic Buildings for Sustainable Energy Use

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SUMMARY

This work is taken part of REVIVAL (Retrofitting for Environmental Viability Improvement of Valued Architectural Landmarks), an energy project supported by European Commission, under THERMIE program, started officially in April 2003 and it will be finished in 2008. REVIVAL is a demonstration on refurbishment of non-domestic buildings, post-war architecturally values, to improve their energy efficient performance and well internal comfort conditions for occupants, especially in summer, with low energy consumption and using innovative measures and best practice. A life cycle CO_2 assessment procedure has been applied and concerned to the emissions due to embodied energy in building materials and transportation. There are six refurbishment demonstration sites, which are included in REVIVAL project in five EU countries. Our paper aims to present a study focusing on CO_2 stock analysis and taking in account GHG emissions on a building site – a secondary school complex, located in France. The REVIVAL target "*WP 10: CO_2 building stock budgeting*" is run under the coordination of Mr. G. Gaudin (engineer, CEO of Gaudin Ingénierie office and technical coordination of REVIVAL), Mrs. T. de Queiroz Gaudin (architect, D. Sc., professor of Federal University of Rio de Janeiro and researcher in REVIVAL project) and, the technical concepts have been realized by Mr. L. Key, engineer from Central University of Nantes and M. Sc., from Munich Technical University.

INTRODUCTION

In most European cities, there is a vast stock of existing buildings, some them of important architectural values. Considering that the refurbishment of existing buildings, it is a viable option for achieving comfortable and energy efficient buildings, REVIVAL (Retrofitting for Environmental Viability Improvement of Valued Architectural Landmarks) is an energy demonstration project supported by the European Commission, under THERMIE program, that includes the energy efficient refurbishment and monitoring of six existing non-domestic buildings in five European countries [1]. This project contributes to both the general objectives of the thematic area "Energy, Environment and Sustainable Development" and the specifics of the "Eco-buildings" Target. The main global objective of REVIVAL is to demonstrate that refurbishment applied to this particular subject buildings type, tertiary buildings from post-war and pre-energy conscious, can show lower life-cycle CO2 emissions, improvements in energy performance and better cost-effectiveness than the original building or an equivalent new building or demolition [2].

REVIVAL project includes upon refurbishment works of six tertiary buildings, architectural values, in five EU countries: one educational building, in France; two hospitals in Italy and in Greece and three office buildings in Netherlands, in United Kingdom and, in Greece. All sites have common characteristics as: poor insulation, over-provision of glazing in large façades, inefficient plants and degraded materials and fabric. Aiming to provide good indoor comfort conditions and low energy consumption, the projects incorporate both passive strategies (daylight, natural ventilation and solar) and renewable components such solar water heating, solar absorption cooling and photovoltaic panels. In order to achieve this project, there are specific technical and scientific objectives to demonstrate that the measures can improve the quality of indoor environment.

Regarding the CO_2 emissions, an assessment procedure of life-cycle CO_2 has been studied to be applied in each site, considering the GHG emissions due to embodied energy in building materials and due to material transportation to site, personal and vehicles for companies and building site machinery. The aim is to apply a life-cycle CO_2 assessment procedure in the refurbishment site, to make comparisons concerning CO_2 emissions during the refurbishment, the demolition and with the new refurbished building. Due to achieve this focus, in the REVIVAL project there are specific technical and scientific objectives, taking part of targets, named works packages.

In this way, this work focuses the work package concerning to life-cycle CO_2 analysis, titled "*WP 10: CO_2 building stock budgeting*", taking in account the analysis of GHG emissions due to embodied energy in building materials, transportation and vehicles on a site. This paper presents one part of an application in a large secondary school complex, located in France [3]. The results demonstrate a comparison of GHG emissions (ton equivalent CO_2) from the beginning of the works and during the building refurbishment, now in its fourth year.

GREEN HOUSE'S GAS (GHG) EMISSIONS AND LIFE-CYCLE CO_2

A control of Green House's Gas emissions can not be reached without a strong act in the building sector. In a global way the aspects from material's extraction and demolition are considered in the building area. Considering that the CO_2 emissions mechanisms are complex, the work about GHG emissions can not be realized against other priority of a building such as: materials extraction, manufacture of components, transportation of materials, transport of persons, architectural quality, functionality, health, comfort, construction, waste and demolition.

Nowadays the industrial buildings are the most energy-lover sector [4]. Rather acrimonious discussions its real impact on the emissions. The discussions are not ended. Regarding gases emissions, carbon dioxide equivalent (CO_2-equivalent) for a greenhouse gas, measure used to compare the emissions from various greenhouse gases based on their global warming potentials (GWPs), it is derived by multiplying the mass of the gas by its associated GWP. For example, the GWP for methane is 21. This means that the emissions of one million tone of methane; it is equivalent to emission of 21 million tones of carbon dioxide. CO_2 is considered the gas which has the shorter life-time and it is the easiest to eliminate. On the other hand, industrial gases are much more harmful (important life-time) and there is difficulty to eliminate them. See table 1.

Table 1. Impact of different gases: global warming potential [5].

GWP	CO_2	CH_4	N_2O	HFC	PFC	SF_6
Life length (years)	50 to 200	12	114	HFC23:257		
Global warming potential (at 20 years)	1	56	280	460 to 9100	4400 to 6200	16300
Global warming potential (at 100 years)	1	21	310	140 to 11700	6500 to 9200	23900

OBJECTIVES OF THE CO_2 BUILDING STOCK BUDGETING TARGET

The objective of the *CO_2 Building Stock Budgeting* target is to determine a life-cycle CO_2 analysis for a period (for example, on a 20 years), associated with building materials, transportation of materials and personal, which can be applied to these cases: (a) partially deconstructed and totally refurbished building; (b) total pull down and erection of a new building; (c) building maintained in state.

In the first case (a), CO_2 emissions are due to: deconstruction a part of the existing building, erection of the new parts of the building, transportation of wastes and building materials. Running of the new building on 20 years it is considered: energy and lighting. In the second case (b), CO_2 emissions are due to: complete pull down of the existing building, erection of the whole new building, transportation of wastes and building materials. Running of the new building on 20 years it is considered: energy and lighting. Finally, in the third case (c), CO_2 emissions are due to: energetic consumption on a period (for example, 20 years), such as: heating, ventilation, auxiliaries, air conditioning (possible), lighting, upkeep and maintenance, demolition for a period. The real stake of the *CO_2 Building Stock Budgeting* target concept is to determine which case produces less CO_2 emissions. As refurbishment is most of the time necessary, we can asses the total life-cycle CO_2 emissions of a procedure and try to demonstrate its environmental added value.

Analysis of a demolition building

In case of a demolition of an existing building and its substitution by a new one, it is not enough the new building be less energy lover/m² to observe reduction of CO_2 emissions. It is necessary to add the supplementary consumptions as: demolition of the existing building, transport of wastes and materials, erection of buildings and manufacturing of the components.

CASE STUDY – SECONDARY HIGH SCHOOL

The case is an educational buildings complex – Chevrollier High School, located in Angers city, in France. Since the 60's, none refurbishment has ever been done to the buildings of Chevrollier school. On the other hand, none ventilation system has ever been installed in the high school. Some areas, such as the workshops, were built with asbestos. Windows and steel frame were degraded. There was the same problem concerning to the stability of the concrete. Table 2 shows the surfaces before and after refurbishment. Figure 1 presents the general plan of Chevrollier site and Table 3 presents the works progress [3] and [6].

Table 2. Surfaces of Chevrollier refurbishment: before and after works.

Work phases	Surfaces (m²)
Before refurbishment	37.000
Refurbishment	33.000
Demolition (workshops)	4.000
New buildings	9.000
Total surface	42.000

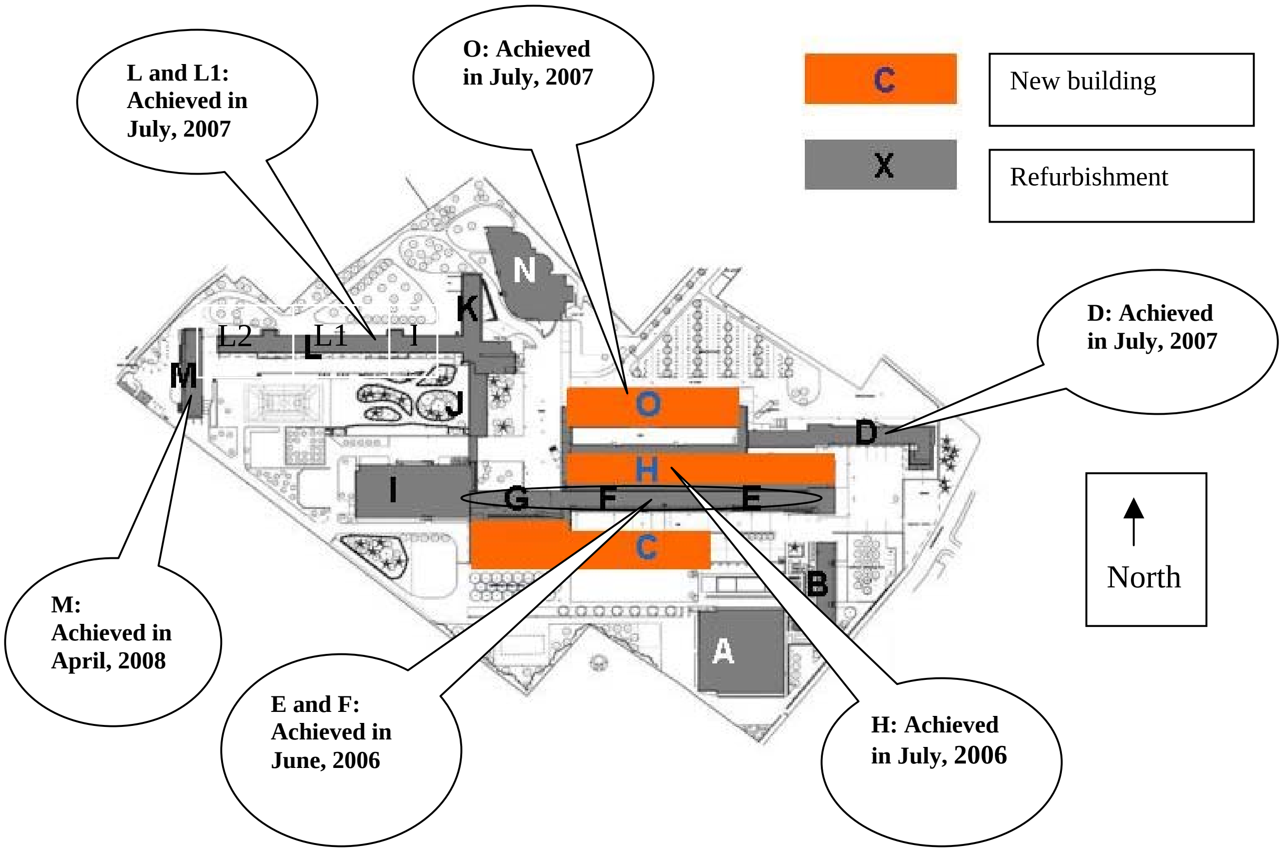

Table 3. Works progress.

Buildings	A	B	C	D	E	F	G	H	I	J	K	L	L 1	L 2	M	N	O
Advance (%)	/	100	100	40	100	100	100	100	100	100	100	0	0	0	0	/	20

Figure 1. Plan of Chevrollier site.

The buildings C, E, F, G and J are used as classrooms. The building H and O are used as workshops and the buildings K and L are the residential buildings for students and teachers. The buildings A (gymnasium) and N (cooking/catering) are not concerned to revival projects. The refurbishment works are completed without stopping the scholarship of the pupils.

METHOD – GHG ACCOUNTING AND CALCULATION PROCESS

In the case of Chevrollier site, studied in this paper, applying the sustainable refurbishment was obviously a necessity because: the school must be kept in state of use during five years of refurbishment and the existing site location is a social stake. Indeed, Chevrollier School is located between the town centre, quite classy, and popular suburb. It is essential to preserve this social coeducational system.

After a short description of the technical design of the solution for GHG accounting, the global results are presented by graphical form. For the first time since the beginning of the

CO_2 building stock budgeting target, GHG emissions figure are able to be exploited, interpreted and extrapolated.

Before refurbishment

Before the refurbishment, the causes of CO_2 emissions were electricity and gas consumption. Concerning to gas consumption, it has realized an average of data for four previously years. In order to obtain the annual emission, it has applied the technical documentation about CO_2 emissions from boilers. Due to electricity consumption, it was used the quantities gave by ADEME [5].

Based on electricity and gas consumptions for year 2003 (the start of works), the following table (4) analyses the amount of GHG emissions before refurbishment.

Table 4. GHG emissions before refurbishment [6].

2003	Consumption (KWh)	GHG emissions (Kg eq. CO_2/KWh)	Source	Total GHG emissions (ton eq. CO_2)
Electricity	880.000,00	0,053	EDF	46,64
Gas	4.200.000,00	0,19	ADEME	798,00
Total				844,64

For future calculations, the figure 850 ton eq. CO_2 /year before refurbishment is retained.

GHG emissions due to transportation on the building site

The method to calculation is based on the methodology realized by Jancovici [7] in ADEME [5], to audit the GHG emissions per Km in Kg eq. CO_2. There are communications between: (a) monthly collected data; (B) GHG emissions data base; (c) GHG emissions calculation results. The audited amounts of kilometers are multiplied by given GHG emissions values (in Kg eq. CO_2/Km), depending on transportation type and brand.

In this study, GHG emissions due to transportation concern the following vehicles categories: personal vehicles, companies vehicles, building site machinery and material transportaion. GHG due to waste transportaion have yet been neglected in the present study, but it will be considered in the progress of the works. For database requirements, the way is filled for each type of vehicles from the main source based in ADEME [5].

GHG emissions due to embodied energy in building materials

The design of an acceptable calculation process to evaluate GHG emissions coming from embodied energy in building materials has been the main research task for this work. The calculations provide different results since 2003, when the refurbishment has started.

Calculation process

No standard data collection is available in order to audit the consumption of building materials. There is technical documentation received per month of all materials used for the building site, from the enterprises involved in the refurbishment works, as manufacturers, constructers. Much computer programming has been realized in order to ease the transcription of paper bills into standard excel worksheet linked to a well informed database of all kinds of materials bought for the building site. Therefore, calculation process is in this case much more complex.

Database requirements

The French institutions as ADEME [5] and AIMCC [8] provide data documentations about life-cycle of materials. According to norm XP-P-01-010 of AIMCC [8], the materials have an analysis of their life cycle about their environmental impact [9]. When a product is not referenced by AIMCC, the impact of its realization on climate change is calculated with the amount of raw material in it and its technical documentation. GHG emissions pro ton of raw material are provided by ADEME. This way of calculation provides the minimum embodied energy for the realization of the product.

RESULTS

Global audit

The following graphic (figure 2) gives overall results of the audit campaign, concerning GHG emissions and realized in Chevrollier refurbishment since the start of works. The total value of mean 3850 ton eq. CO_2 since the beginning of the refurbishment may seem too high, but it is exactly the contrary. Due to a few lack of environmental data, this value is underestimated. However, this graphic is relevant, especially when it is damped and added to prediction on GHG emissions for the life time of the building.

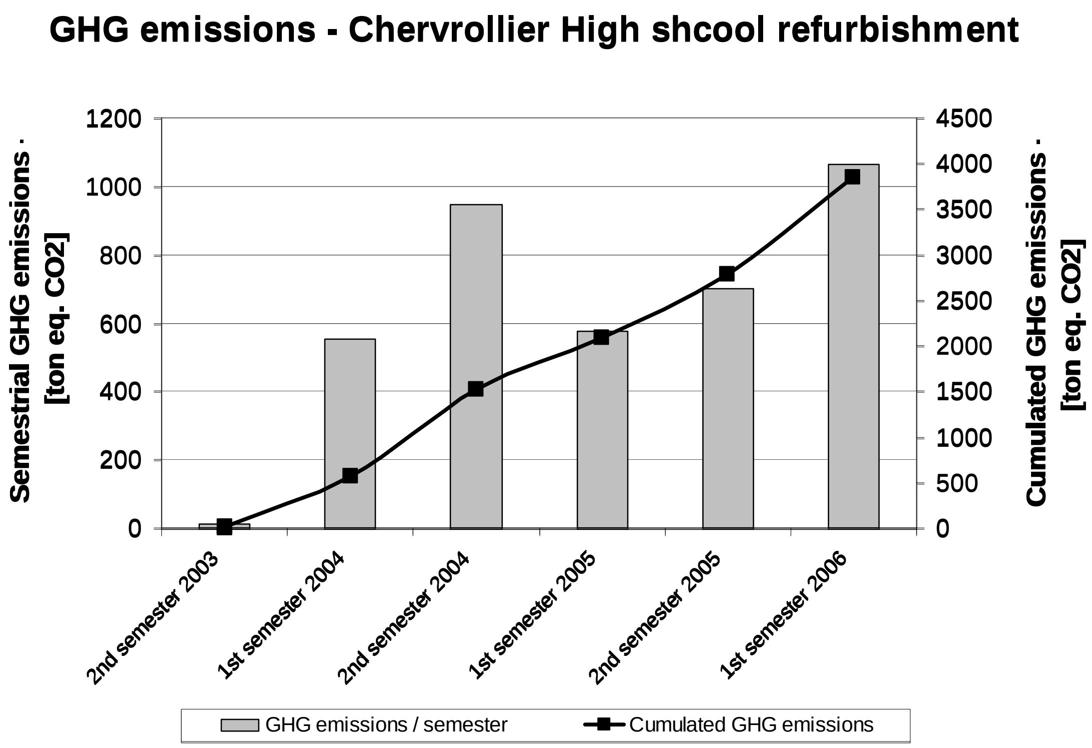

Figure 2. Global results of GHG emissions audit in Chevrollier site.

Source of GHG emissions

Besides the consideration on emissions due to transport and embodied energy, it is relevant to estimate the amount of GHG emissions of each of the four main building tasks achieved on Chevrollier refurbishment site such as: (a) lot 1: road network; (b) lot 2: electricity; (c) lot 3: heating; (d) lot 4: structural work and building shell.

Prediction

A simple regression has been run on the results in figure 2. The upper bound of the prediction gives a global GHG emissions figure of 5438 ton eq. CO_2 for the building site until the end of first semester 2007. See figure 3.

On one hand, in this graphic of Figure 3, the value of 5438 ton eq. CO_2 must be compared to former GHG emissions for Chevrollier high school. On the other hand, we must evaluate the importance of this amount of GHG emissions in the life time of the new high school.

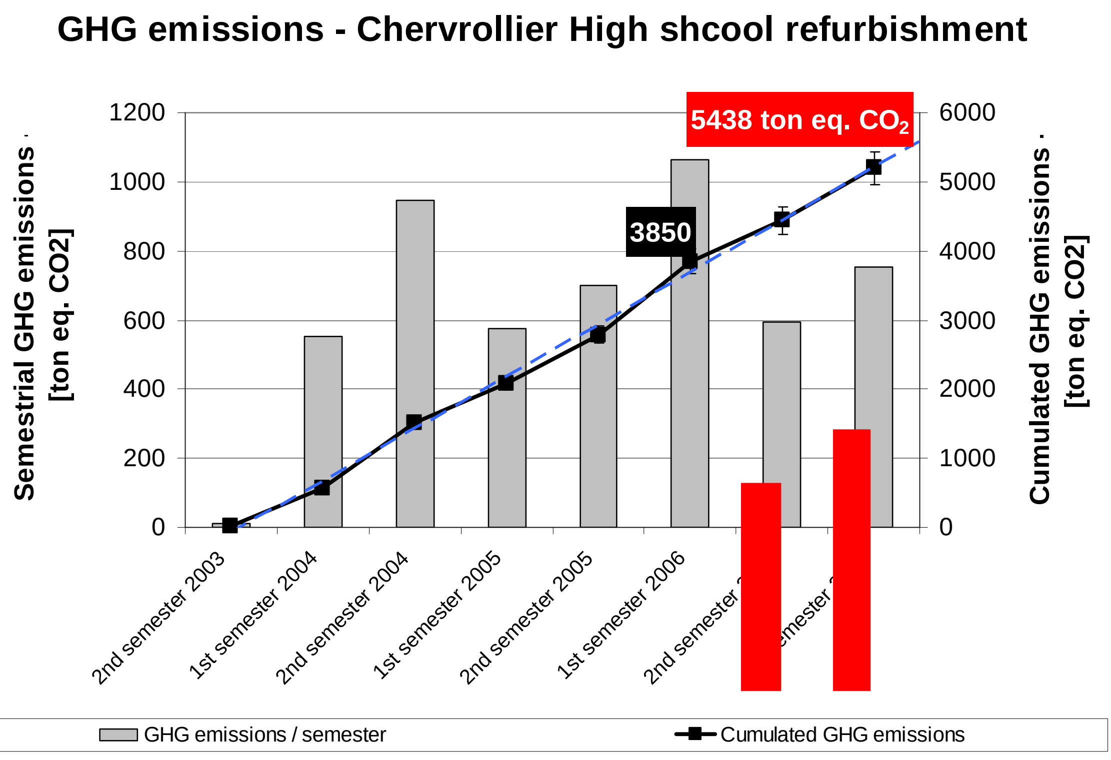

Figure 3. Predicted results of GHG emissions audit in Chevrollier site until first semester 2007.

DISCUSSIONS OVER RESULTS

We can notice that in order to match with a return on GHG investment smaller than 20 years (theoretical life time of the refurbishment building), the improvement of Chevrollier high school meet a GHG reduction goal per year of about 35%. We can observe that without improvement in GHG emissions, refurbishment still represents 32% of annual emissions on the 20 years life time building. However, 5438 ton eq. CO_2 is a very high value and in this case, the amount of GHG emissions due to refurbishment would count for more than 50% of the annual emissions for the building life time.

These results are impressing, but more study will be realized to give a good and scientific interpretation. Actually, the prediction of 5438 ton eq. CO_2 concerns both old building refurbishment and new building construction. The next step of the study is to evaluate the amount of GHG emissions specifically due to refurbishment by studying the time scale of the project.

CONCLUSION

CO_2 Building Stock Budgeting is related to the owners who have management of a large building park. The finale objective of CO_2 Building Stock Budgeting target is to define a budgetary policy, which enables the owners to really reduce CO_2 emissions.

If governments' targets for reduction in CO_2 emissions are to be achieved, refurbishment and renewable technologies should be considered for the existing large stock of typical tertiary buildings in most European cities.

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Examples of the Characteristics of European and ASEAN ESCO concepts

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SUMMARY

Presumably, the means of saving energy are not the very same while the climate and culture in Finland compared to those in the ASEAN region are very different. The expert interviews in Finland, in Singapore and in Malaysia proved rather similarities than differences in the energy saving partnerships. Starting the processes with the energy audit, a need of transparent contract models, surprisingly common energy saving measures, difficulties in development and implementation of measuring and verification methods and relaying so far on a handful of active operators are shared. Practise of energy labelling, efforts for defining the national baseline and the accreditation of the ESCOs are strong in Singapore. The Finnish ESCOs apply the latest EU legislation into the ESPCs and seek how to solve the use of multiple ESPCs in the ageing building stock in municipalities all over the country. Ageing buildings form an energy saving potential in both countries.

INTRODUCTION

An energy performance contract (ESPC) project (energy saving partnership ESP) is a partnership between the customer and an energy services company (ESCO). The ESCO conducts a comprehensive energy audit and identifies improvements that will save energy at the facility. In consultation with the customer, the ESCO designs a project that meets the customer's needs. The call for bids precedes the contract. The customer evaluates the competing ESPCs based on demonstrated capabilities to manage the general scope of work, terms, and conditions for firm-fixed-price task orders for performance-based energy savings. Theoretically, the ESPC can be classified as a form of the Life-Cycle Procurement Contracting such as the EPC (also called Third Party Financing), the PFI (Private Finance Initiative), the PPP (Public Private Partnership), the BOT (Build–Operate–Transfer), etc. The key elements of them are the financing, the service level agreement, the financial and energy technological risk assessment and the measurement and verification. The bankers, the ESCOs and the building owners seek for a beneficial win-win-win business concept.

ESPCs allow the customers to accomplish energy retrofit projects in their facilities without up-front capital costs and without special appropriations to pay for the improvements. After the contract period ends, all additional savings benefit the customer. The primary factor motivating the use of ESPC by governments or municipalities has often been the difficulty of obtaining funding for required energy upgrades and infrastructure renovation through the normal budgeting process. These improvements may be needed to replace outdated or poorly performing equipment and to meet externally imposed energy conservation goals. In general, if a project is economically and technologically feasible and is large enough to interest an ESCO, it can be implemented through the funding of an ESPC. To customize the scope of contract to suit local markets, the development and implementation of multiple ESPC projects over a large geographic area, or the call for bids covering techno-economical approach for one or more defined site-specific projects can be used.

Worldwide, including Finland and Singapore, the energy performance related business has been no mainstream field in construction. A handful of new ESPC projects have been started per year nationally. It is a so called radical product without established markets. Nevertheless, the ESPCs match certain customer needs better than any other contract forms. Energy contracting is based on both legislation and energy policies which especially calls upon the public sector to take a leading role in the sphere of energy efficiency. The US and European administrations as well as those within the Commonwealth of Nations have enacted some type of enabling legislation permitting hospitals, day-care centres, schools, universities, etc. to use the ESPC. The ESPC constitutes a successful contracting model in reducing carbon dioxide emission and its negative impact on the climate, as well as in lowering of energy costs. It is therefore not surprising that the model is increasingly finding many imitators.

Not only insufficient funds but also the shortage of specialised labour or the lack of resources in the municipal management might lead to exploit the ESPC, especially in the small communities. The time it takes to implement an ESPC project can be much shorter than the time consumed waiting for directly funded projects. In some cases, the democratic municipal decision making might cause troubles, if the local councillors do not understand the benefits gained by the ESPC. In contrary, quite often, the short term monetary values alone count most when comparing the bids, especially if no emphasis has been put on sustainable values.

The ESPCs can yield significant benefits in the operation costs of building services depending on the standard of energy efficient or sustainable technology employed during the design and implementation phases or during repairs, to achieve an energy efficient building performance life-cycle. Although the total savings yielded can pay the repairs and improvements, still, these so called guaranteed ESPCs are no norm as the feasible projects tend to have external investment funding (the shared ESPCs). Using financing secured by the ESCO, by the customer or by a third party or even, by a combination of them, the ESCO installs the improvements and guarantees that they will result in a specific level of annual cost savings, sufficient to pay for the project or the debt over the term of the contract, often including performance-period services such as maintenance of the equipment. The ESCO must remedy any performance problems and reimburse the owner for any savings shortfalls.

The ESPC can be understood as a tool box for implementing energy efficient and sustainable technology and related services. It has launched a new approach to building services as a turn-key delivery − with the added value of performance guarantees, lasting longer than what the guarantee requires; from a couple of years up to 25 years, instead of the one year guarantee period for the performance of the equipment within a traditional construction contract. The ESCO guarantees that the equipment will perform as specified and that the specified standards of service (temperature, humidity, etc.) will be maintained for the life of the contract. An important difference from the traditional design–bid–build process is that in an ESPC, the ESCO provides a single point of accountability, from energy audit through design, construction, and commissioning. This eliminates any uncertainty about who is responsible if design or construction issues arise.

The loan is often translated into low-risk financing of quite low interest rates, in the case of a governmental or municipal lender. The ESCO’s credit is seldom as good and thus the ESPC programs might be structured so that the customer obtains the financing, not the ESCO. In summary, shared ESPCs lowers the risk to the lender and allows lower financing costs, and sometimes simplifies the process. Guaranteed or shared ESPCs benefit the building owners or occupant companies which is the target of the ESCOs, who in addition, seek for profitable business concepts for themselves. Interest rates can be fairly lower if the risks perceived by the lender are mitigated. Recently the interest rates in financial markets in general have been low enough to offer various business opportunities, influencing also on the demand for the ECPSs. It is of interest to note that sometimes the ECPS could be left without financing because the bankers are not convinced of the feasibility of the project due to lacking knowledge of the concept of ESPC because of the very technical nature of it.

Periodic measurement and verification of savings (M&V) provides the facility owner with assurance that the specified performance and guaranteed savings will be realized. An M&V report describes in detail the results of a program of measurements, inspections, engineering calculations, and comparisons with energy baselines, carried out to estimate the level of savings being delivered by the installed equipment. Standardised M&V methods have been introduced. They can base on calculations and measurements. Common energy conservation measures comprise lighting retrofits, high-efficiency heating and air conditioning equipment, insulation upgrades or replacement of windows with energy efficient window types, sophisticated energy management and intelligent control systems, boiler and chiller replacements, installation of heat recovery and improvements in steam and hot water distribution systems. In addition to the hardware related measures, systems design and end-user empowerment tools have been used such as peak load management, optimisation of HVAC operations and shifts, adaption of systems to demand or user training from briefing the caretakers to the guiding courses for the end users (e.g. nurses in hospitals), and not to mention the instructions for the end users how to operate the building energy efficiently, e.g. opening the windows when too warm or switching off the lights when the room is not occupied [*]. Few ESPCs promote specific advanced technologies such as air or ground source heat pumps, photovoltaics, and biomass or alternative methane fuels as well as solar thermal concentrating systems and new energy storage methods.

The building project or the renovation of a building can be accessed via many modern concepts necessary in the operations of the construction sector: commissioning procedures, total building performance considerations, creation of life-cycle services, new construction contract models and integrated facilities management as well as financing instruments to come. General trends of construction business affecting energy performance contracting can be summarised as: the capital and "renovation" intensity[1] and spend is going up, the knowledge intensity[2] why design should be given to agile designers, the labour intensity[3] and the globalisation of labour market, the win-win-win procurement[4] challenging how to measure and the need or the fairness of incentivisation, the consolidation of companies[5] by developing new contract models, a growing awareness[6] of usability and empowerment of occupants, an ever more sophisticated business concepts[7] creating markets for radical products and services, immaterial finance[8] instruments and unawareness of their risks and a risk management of end-user behaviour[9].

METHODS

Via expert interviews were gathered current knowledge of the ESPCs in Finland, Malaysia and Singapore. The methods for the personal interviews varied: in Finland a theme interview addressed to the 21 experts who had worked on the six recent major ESCO projects [1], a structured questionnaire addressed to the 30 experts within the 12 ESCOs and an energy audit consultant in Singapore, and the 12 experts in Malaysia. However, the last mentioned set of questions worked rather as a source of inspiration than as a strict guideline for the discussions. Realization of these interviews resembled theme interviews. In Finland and in Singapore each ESCO was interviewed separately while in Malaysia group interview method was employed. The key characteristics out of the ESP best practises are summarised in this context.

Both in Finland and in Singapore, the interviews got a freely flowing and open form while the interviewees were keen on the subject and on telling of their aims and efforts. As known the group interview was not the best pick for the method, because during the discussions one interviewee might take the lead and dominate the answering. This could not be prevented totally in this case either.

Locations of Studies

The land area of Finland, in the Temperate Zone, is relatively large and Singapore, again, is a relatively small island in the Tropics, in front of the Malaysian peninsula occupied by Malaysia[1]. In those democracies, the consensus on national interests matter together with the transparency in government's activities and in national financing. The importance of energy and its role under careful administration in all three countries did not live up to expectations of cultural differences − Finland, a homogeneous Christian society; Singapore, a multinational society with several regions; Malaysia, a Muslim society[2].

Finland and Singapore depend mainly on energy imports while Malaysia has own oil resources. Finland has hydro and nuclear power and fairly good chances to benefit from such renewable energy sources as wood, air and earth grounded heat pumps, wind and waste. In contrary to the Finnish multiple set of energy sources, Singapore roughly said is solely oil dependent. In Singapore the chances to use renewables, solar energy in particular have been found limited excluding the potential of waste. In Finland as in Europe in general quite often energy supply and distribution systems are competitive. Building owners have to choose between competing technical systems of oil, electricity or renewables while in Singapore and in Malaysia electrical cooling dominates.

BUSINESS LANDSCAPE

Both in Finland and in Singapore, efforts for energy performance and co-operations between government bodies, scientific institutions and private energy sector are active and versatile encouraging ESPCs. The Malaysian experts are keen on following the same progress. The practice of energy audits has been established by many governments [1] encouraging the national operators to carry out energy audits; in Finland 40 per cent of the costs are granted, in Singapore 50 and in Malaysia 100 per cent. In all studied countries, the energy audit procedure comes first and then the energy performance project follows. The minimum project comprises the management of system and equipment choices with the life-cycle costs calculations, while the maximum concept covers design, contracting, evolution and integrated facilities management of the whole building and the outdoor area. In Singapore, the internationally operating ESCOs could use several financing instruments: self finance, good credit from banks or financing from clients while some ESCOs found financing risky and told that they might face difficulties in covering the financing of the guaranteed ESPC. However,

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[1] The land area of Finland is 360000 km^2 and population of 5.3 million; the land area of Singapore is 647.5 km^2 and population of 4.4 million; the land area of Malaysia 329 750 km^2 and population 25.8 million.

[2] Religious minorities exists

not only the major Singaporean ESCOs but others as well, operating also in the neighbouring countries: Philippines, Thailand, Vietnam, etc. had committed guaranteed ESPCs. That was neither the case in Finland nor in Malaysia. In Finland, the shared ESPCs dominated the markets where also the municipal financing instruments were used. In Finland, the ESCOs can be granted a public investment aid of 20-25 per cent. In Malaysia, the financing of ESCOs could benefit from the United Nations aid for developing countries and get five year loans for a zero interest rate. An interesting practise obtained by some ESCOs in Singapore proposed an MOU of the upcoming ESP (memorandum of understanding) to the clients based on a preliminary energy audit to make sure of their motivations. After the MOU has been signed the energy audit was finalised.

The trust between the ESCO and the client was found important in Finland as well as in Singapore and in Malaysia. The contracts drawn up without a national contract model were not necessarily very detailed. The contracting was rather based on mutual trust between the client and the ESCO than on written paragraphs of the contract. Thus, the need for contract models was obvious in all countries. Court cases have not been avoidable in total. In addition, contract models establish a certain commonly acceptable commercial usage based on correct business ethics and on fair play. All interviewed shared the difficulty of no common language between the ESCO and the client concerning the service descriptions and the service level requirements (fees, complains and sanctions). In Finland, no one mentioned of the problematic changes due to the end user actions during the ESPCs whereas in Singapore end-user behaviour was quite often considered as a risk. A cultural difference can cause this difference. In Finland, according to common practise, the business partners negotiate the contract and that after it is realised very firmly as written, sometimes even without any flexibility in changed situations. The newly developed template for the stipulation of the ESPC contract in Finland includes the clause of the necessary changes in contract after changed situation. Still, it is recommended to keep the contract in the first place. The Singaporean business culture is evidently one of the most Western economies in Asia if not the most Western. However, the Asian business is more than the Western, and definitely more than the Finnish one based on negotiated terms during the contract. No wonder that the clients of the ESCOs find it correct to negotiate even the baseline of the ESPC during the contract.

Both in Singapore and in Finland the ESPCs were structured on project by project basis after the outcome of the energy audits or according to the targeted savings or energy efficiency defined in the tender documentation. The scientists work for the nationally universal procedures and especially in Singapore on the definition of the national baseline together with the development of energy benchmarking which were not emphasised in Finland, at least by the interviewees. The M&V of the ESPC was also under development in both countries because of improving the problematic quality control of the ESPC. However, to find the best calculation or measurement methods is not easy. The best pick of selected parameters caused troubles as much as the surprises found in lacking HVAC systems and their functionality when measuring the performances. The lively international co-operation can be used in defining the measurements needed for contract stipulation and the M&V.

Legislation and governmental policies, efforts for transparency of business activities, accreditation of ESCOs, as well as energy labelling assist in promotion of energy efficiency and sustainability. In Finland, when implementing the latest European directives, described later within this context, the transparent procurements on public works was considered as a two-edged sword opening the tenders better than earlier to competition while on the other hand, a challenge to the authority of the new directive emerged by some well established effective consortiums of ESCOs and their clients. Also a topical subject was how to organise the neutrality of the consultants working under the contract of an ESCO. Lack of competitive providers for larger ESCO projects was a problem in Finland. The market was not mature enough for numerous large players and smaller providers have not enough assortment to offer for large tenders.

An increasing number of Finnish municipalities have considered outsourcing of their municipal technical services as a good solution for their problems to make those services available. No particular interest in involving the ESPCs in these life-cycle procurement contracts was noticed. However, as the ESPCs theoretically can be classified as a form of life-cycle procurement contracts, the more common any of these contracts become the better the idea of the ESPC will be embedded. This thinking can be considered parallel to the more general thinking in Europe of intentions to extend the classic approach of an ESP focussing on energy efficient equipment and control engineering to the development of life-cycle building services contracting models. In fact, the latest progress of the ESPCs enforced this thinking in Finland, within the CUBENet project[3] [6]. It has been asked if a life-cycle ESPC could be possible in the future. Developing such an innovative model would be a cornerstone in increasing energy efficiency and in offering an attractive financial alternative for the public purse. In addition this offers an opportunity to attract new customers in industry and trade, in office buildings and in the housing industry.

The Finnish municipalities seek also means how to develop and implement multiple ESPC projects covering several small buildings, for example day-care centres or village primary schools. Models of life-cycle contracting of that kind have been found from Sweden [7] and from Great Britain [8].

In Singapore, the American companies established there work together with the Singaporean operator for the interests of both parties. To put it very simple, in addition to their high class scientific work, the Singaporeans have interpreted also the American models of energy performance concepts and modified them suitable for their effective way of operation and economy. Likely, the natural progress of the ESPCs or other forms of life-cycle procurement contracts will continue to follow this feasible line of development. Like in Finland also in Singapore the aging building stock, particularly housing, needs renovation and holds a large energy saving potential. However, so far the efforts to encourage the ESPCs have been on public or commercial buildings, hotels and industry.

In Malaysia, the industrial buildings were in focus in this first wave of implementing ESP. The organisations encouraging the ESPC [11] provide material and courses of the technology, methods and economy of energy performance for local ESCOs and building owners. The interviewed in all countries considered the co-operation with facilities management (FM) organisations and the ESCOs essential. However, the mixing roles of building services and FM services were found confusing. Even, some Asian FM operators thought of their position threatened due to the ESPC.

Motivation of Energy Savings

Since the energy crisis in the 1970's, energy savings and energy efficiency have been continuously the top topics in the research and development of technology (RTD) in the Western economies, in Europe including Finland, while a similar progress has started in

[3] Future Life-cycle Building Services, 2003-2007 (www.sisailmatieto.com/cubenet/) within Tekes (the National Technology and Innovation Agency) technology programme CUBE

ASEAN quite recently. Europe is dependent on imported energy and energy efficiency is an important part of the European energy policy. As recently the prevention of creation of carbon oxide is common around the globe, among other things, the increasing use of renewables has become more important than ever within the European energy policy. In Finland, the ESCOs are in the line of these persistent efforts to save energy. In Singapore, in the democracy of progressive policies, the government is behind all efforts to promote energy efficiency and has launched own programmes to encourage the implementation of ESCO projects. Lack of motivation of clients in energy performance contracting is one major problem in the Malaysian business environment. In all studied countries the private sector favoured the ESCOs while they allow also other additional financial benefits.

The vitality of the energy performance business bases on a win-win-win situation within an ESP. The service provider wins when the customer and its customer, the end-user wins. This is not a traditional business arrangement while earlier businessmen have tried their best to maximise their own profit. Obstacles can be found in any business, why not in this new business. In the first place, due to the modern technical nature of the ESPC third parties such as financiers do not trust on or fully understand this type of profit making. In order to create a positive business climate for energy performance business new thinking or attitudes are needed. Common efforts give right to shared profit. Motivation counts. The ESPCs face hard competition from alternative investments. The Singaporean clients originated from sophisticated financial markets could be characterised as having quite high expectation on the ROI (return of investment). Not the plain saved kilowatts out of energy effective performance motivate the Finnish clients to conclude the ESPCs either.

The European directive of building performance might improve the motivation of energy savings further. The energy certificate to be issued will give a base to argue about energy monitoring and energy management based on the activities of ESCOs. A problem faced in the context of any radical product or service and also in the introduction of the ESCO services, is how to prevent the bad reputation of some false operators to destroy the honest businesses. In Malaysia, in particular, the interviewees mentioned of this problem. The UN money was not found only a blessing while it could draw ESCOs and industrial partners who are more after easy money than energy performance. Accreditation is an asset for preventing problems. In Singapore, an Accreditation Programme for ESCOs was launched in 1995 and it is actively continued since then (www.esu.com.sg). In Europe, Austria has developed the Austrian Logo for Certificated Energy Contracting. Still, in the first place it promotes the effective image of the ESCOs.

The idea of green building comprising the energy efficient technologies has reached wide audience in the Western world (cf. e.g. www.greenbuilding.org). Despite the interest in the sustainable development or in the life-cycle cost analysis, the realisation of the green technology in Finland does not necessarily correspond to the interest in the subject. In the first place, the building providers and developers have aimed energy efficiency simply because of the prevention of high heating costs [12]. Consequently, the green values did not come up during the Finnish interviews either. However, some strong groups exist, who stand behind the sustainable measures to all intents and purposes working on tools for profitable green development [12] and green building labeling [14]. The green city is targeted also with the activities for energy efficiency in Singapore (cf. Energy Smart Building Label, www.esu.com.sg). In Singapore a couple of the ESCOs highlighted the meaning of the green technology and the responsibilities towards the future generations of the condition of the global nature. Thus, they pushed forward the use of renewables and green technology within a long time frame of the ESPC meanwhile they believed the business potential of this thinking. Giving all operators their due, the result of the interviews showed that the Finnish clients of the ESCOs were quite motivated in energy savings while the ESCOs in the ASEAN countries were more cautious of the motivation of their clients.

The empowerment of individual occupants and why not also the occupant companies, in energy saving is challenging in general, and the cultural differences might make it difficult to apply knowledge of it internationally. In either of the countries, the ESCOs have not taken the end user involvement in their set of services if some very end-user oriented energy audits are excluded. It was not directly asked, but the interviewer could not help taking the view of lacking knowledge of the means to empower the end-user for the advantage of the ESPCs and unawareness of the meaning of the end user behaviour for the fact in all countries.

Building Trust by Related Activities
In Europe, municipalities and government organisations are bid to the European energy directives and the legislation on public works contracts. Private sector can operate within the framework of financial interests. However, good public practices might move in on the private business venture. In the European context two recent EU (European Union) directives important to the ESPCs have been launched: The European Legislation on Public Works contracts, public supply contracts and public service contracts[4] [3] and The European Legislation on Energy Performance of Buildings [4]. Most recently the European Commission made a Proposal for a directive on energy end-use efficiency and energy services, which also introduces various modifications of ESP.

As mentioned, in Finland, these directives have been already taken into the consideration in the contract modeling of the ESPCs and other life-cycle contract models for the building service sector [4], [6] and [9]. In Singapore, the development of establishing a transparent contract process is strong because the business operators hunger for it, and the government requires it. In Malaysia, the market should be developed further to find consensus of the ESPC contract models in other sectors than in the industry [10].

The directive of the energy performance of buildings concerns setting of energy performance requirements, making available energy performance certificate on built, sold and rented buildings, inspection of boilers, inspection of air-conditioning systems and independent experts. The only one of the recommendations not adaptable for national purposes is the availability of energy performance certificate. European Commission wants to awaken

[4] Certain rules applicable to all European public works contracts comprise first of all the contract award criteria based either on the lowest price only or, where the contract is awarded to the most economically advantageous tender, on various criteria linked to the subject-matter of the contract in question, such as: quality, price, technical merit, aesthetic and functional characteristics, environmental characteristics, running costs, cost-effectiveness, after-sales service and technical assistance, delivery date, and completion date. The contracting authority should specify the relative weighting it gives to each of the criteria. The legislation highlights combating fraud and corruption, electronic auction, rules on publication and transparency and to put traditional and electronic means of communication on equal footing.
Four alternatives of procedures were included. In an open procedure1, any interested economic operator may submit a tender. In the case of restricted procedures2, any economic operator may request to participate and only candidates invited to do so may submit a tender. In a negotiated procedure3, the contracting authority consults the economic operators of its choice and negotiates the terms of the contract with them. Within the competitive dialogue4 a contracting authority may make use of it for complex contracts if it is not able to define by itself the technical solutions to satisfy its needs or is not able to specify the legal and/or financial make-up of a project. Large infrastructure projects would seem to lend themselves to this type of dialogue.

awareness among Europeans of the possibilities to save energy and to reduce emissions from green house gases, thus, it is not only the matter of energy efficiency but use of domestic energy sources and cultivate sustainability.

The costs for the provision of the Energy Performance Certificate in Finland are estimated to be relatively high[5] [10]. When implementing of the directive the Finnish government looks after means for a lean production of the certificates for example by preventing unnecessary authorisation and the establishment of extra organisation for it which does not advance energy efficiency. This differs from the ASEAN efforts as well as the North American and the British where authorisation in general is more common than in Finland − in a country without a system of post graduate registration and qualification of the practicing architects and engineers. However, it is worth of mentioning that the authorisation of the energy audit experts exists also in Finland. Further on, universities and polytechnics as well as private educative organisations will be obliged to take care of the training of the providers.

The Ministry of Environment and the Ministry of Trade and Industry estimates that the influence of the energy performance certificate directive will be large on the renovation activities the level of which already is high. Further on, after the directive has been implemented as a whole the new born activities will yield savings depending on the realisation of implementations. Energy savings are estimated not to cover the renovation costs or have a remarkable influence on the reduction of green house gases.

Energy Saving Measures in Different Climates

Both in the Northern Europe and the tropical ASEAN difficult climatic conditions are faced. In Finland, some periods in winter are very gold, causing high rated temperatures for HVAC technology, and in Singapore the air humidity and temperature are high all year round. The energy saving measures were expected to be different while carrying out the interviews on the ESPs in two different climates causing contradictory aims for the performance of technology: in one case, extra heat has to be removed by cooling the space, and in the other, heat has to be gained by heating the space. In the first place, the ESPCs in Finland comprise often the adding of heat recovery and in Singapore a removal of extra chillers.

Nevertheless, in the both climates the balance between effective energy use and good indoor air quality is targeted without forgetting the aims of sustainable development and taking into account the worries of global warming. Means for building service technology are many and in addition, interestingly enough the very similar technology available globally can be applied differently. The building automation, often implemented by the ESCOs is self evidently the very same all over despite the climate. Insulation is applied either against cold or heat as well as air-conditioning for high indoor air quality follow universal design criteria. All are concerned of the energy spending in lighting of empty spaces and the optimisation of the interplay between artificial illumination and natural lighting. Meanwhile, it is not clear enough to say how the sustainable or green technology can solve the indoor environment in different outdoor conditions. However, the same technology is considered and tested internationally.

ACKNOWLEDGEMENT

[5] During the first 10 years 240-360 million euros when 22-25 per cent of housing and private sector service buildings are included and the number of the number of 400-500 new independent experts are needed [10].

This paper draws from the results of the EC-ASEAN project 64, Development of a Business Model for Building Energy Performance Services in the ASEAN region[6] and within the Finnish national project CUBENet, in co-operation with the EU 6th FP project Eurocontract, European Platform for the promotion of Energy Performance Contracting[7], and the IEA Annex 46[8], Holistic Assessment Tool-kit on Energy Efficient Retrofit Measures for Government Buildings.

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Energy Optimization Services in a Belgian Hospital; Facts & Results

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SUMMARY

The presented paper will describe the outcome of a service agreement to optimize the performance of an installed Building Management System (BMS).

In an existing building (a polyclinic) with a BMS, many common control and comfort settings have been questioned and adapted resulting in astonishing energy savings. A thermal reduction of 35,7% and an electrical reduction of 15,7% are achieved after a two year optimization period. These figures resulted in a very fast pay back time of 11 months for the common cost of the audit and service agreement.

The impact of the figures is even more impressive when one looks at the initial benchmark results. On a heat consumption base the hospital under investigation wasn't performing bad, the fact we could even improve this figure really creates opportunities for efficiency optimization services at customers with a BMS. Towards the electrical consumption one can clearly see the impact of uncontrolled computerization and fully air conditioned buildings.

We hope that this paper will help the technical manager of a facility to experience the optimization potential of a well maintained and controlled BMS.

INTRODUCTION

In the beginning of 2005 we introduced the Building Performance Optimization Services on the Belgian market.

As an environmental concerned organization we want to investigate the impact of the energy optimization services potential for a customer whose building is equipped with a BMS. The outcome of this study is important to quantify possible energy- and emission savings in other buildings who are or could be equipped with a BMS.

The customer who participated in this study is a community hospital of the city of Aalst namely the 'ASZ-Aalst'.

The ASZ-Aalst represents a group of three hospitals:

ASZ - Aalst Hospital group			
ASZ - Aalst		m²	# beds
	Hospital	34264	368
	Polyclinic	5025	0
ASZ-Geraardsbergen			
	Hospital	13125	160
	Eldery people clinic	2872	60
ASZ-Wetteren			
	Hospital	7377	101
	Polyclinic	2084	0

Table 1: Overview of the ASZ-Aalst hospital group.

Figure 1: The polyclinic in front of the main hospital

From these three hospitals, only the newly build polyclinic, located on the ASZ-Aalst territory is having a modern BMS. For this reason, all BMS related improvement measures are implemented at the polyclinic.

PAPER

In 2004 the technical director of the ASZ-Aalst hospital group wants to know the potential impact of energy optimization measures related to the ASZ-Aalst hospital. A basic audit shows the major discovered energy opportunities.

Table 2: Energy audit results

	Savings		
Calculated Measures - Main hospital	Energy (kWh)	€	CO2 (ton)
Advanced cooling & compressor control	300000	27000	225,0
Standardisation and optimal control of comfort level & BMS	850000	46580	419,9
Replacement of the existing window panes (bad shape)	1330000	46550	465,5
Total main hospital	2480000	120130	1110,4
Calculated measures - Polyclinic	Energy (kWh)	€	CO2 (ton)
Advanced cooling & compressor control	50000	4500	37,5
Standardisation and optimal control of comfort level & BMS	125000	6850	61,75
Total polyclinic	175000	11350	99,25

Besides a major architectural optimization, the replacement of the current window panes, huge opportunities are waiting in regards to the BMS.

As the polyclinic is entirely equipped with a BMS, we suggest to primarily optimize the small day clinic and to use the optimization results to plan further investments in the other hospitals of the ASZ-Aalst group where no accurate BMS is currently available.

Investments in a BMS are financially important decisions, so the ability of a technical director to prove the financial (environmental) benefits towards upper management is a very important aspect to defend the associated investments.

A first important step in the workout of the paper is the explanation of the task flows and the basic philosophy behind the BPO service model.

Figure 2: Integrated approach of the BPO model[1]

As the graphical representation of the BPO model shows, we want to help the customer improving the all round performance of his BMS. This results in the short quote: 'Helping your building work for you'.

As this paper is focusing on the energy performance optimization of the model a correct alarm handling and online follow up of the installation is very important.

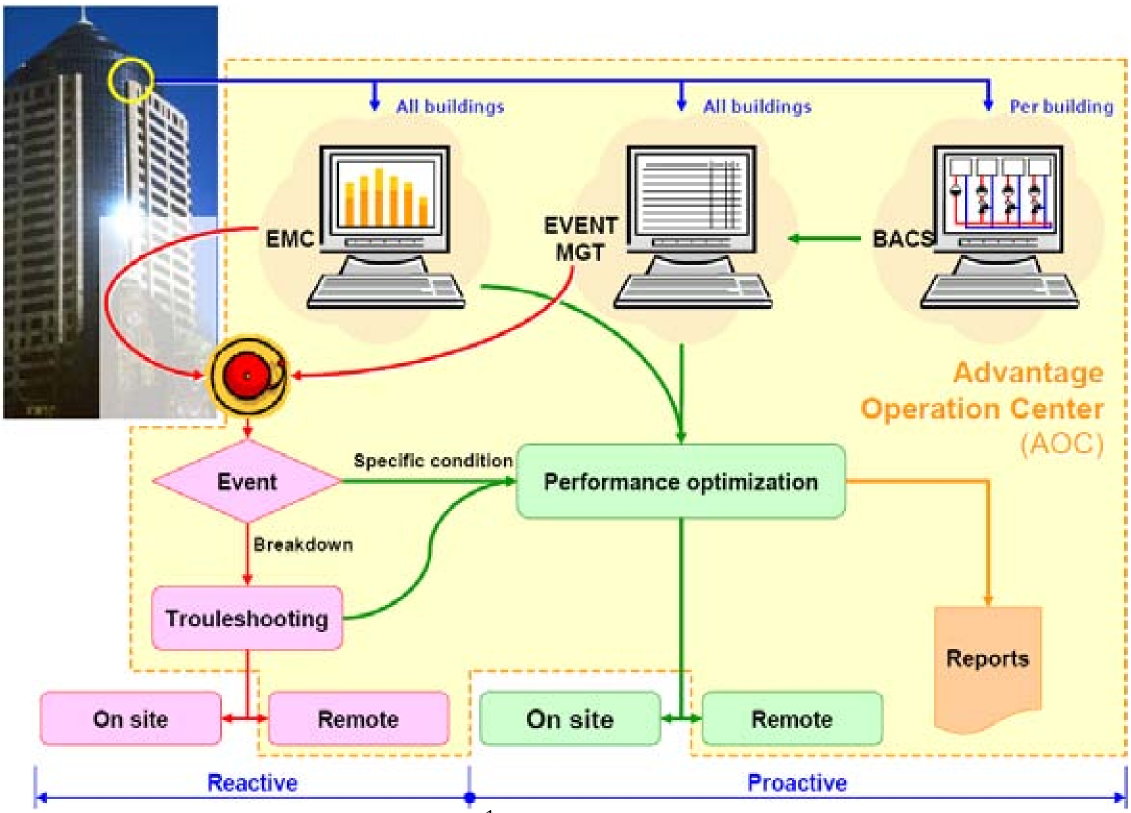

Figure 3: Task flows behind the AOC[1]

In an Advantage Operation Center (AOC) we have a complete online view of the customer. Any process or consumption abnormality is triggered immediately to the responsible. As most of the programming measures are implemented on site, the follow up and proactive tuning can take place from any location which is a major managerial & environmental advantage.

Figure 4: The EMC software label

Part of the AOC is the EMC (Energy Monitoring & Controlling) tool which gives us the opportunity to check & follow up online the current consumption (= achieved optimization results).

A second important step in the optimization process is the implementation of a basic metering structure. A general electricity-, heat- and water meter are installed on site although no separate domestic heat - and electricity for cooling meter are available. As we have an idea on the domestic heat consumption (basic thermal consumption during hot summer periods), we can use the standard HDD (Heating Degree Day) correction functionality of the EMC software tool on the data. Unfortunately towards cooling we can’t use the CDD (Cooling Degree Day) correction as there are no reliable means to quantify this consumption, in the near future a separate electricity consumption meter for the cooling machines will be installed.

Once the metering structure is up and running we started the implementation of the different FIM’s (Facility Improvement Measures).

All the measures can be addressed in one of the following main classes:

- Comfort
- Heat distribution
- Cooling production & distribution
- Lighting

Comfort

The first thing we analyze in the optimization process is the current comfort level of the building. As all rooms own their personal comfort adjustment (PCA) panel, extreme temperature settings can be encountered. Another important setting is the difference between the SPV (set point value) for heating and cooling. In the polyclinic both settings were extremely mixed up. Comfort temperatures were varying between 19°C and 26°C, a majority of the room settings had a gap of 1°C between the heating and the cooling set point.

This resulted in a first implemented measure: the standardization of the comfort temperatures. In the current situation all the rooms have a SPV for heating of 21°C and a SPV for cooling of 24°C. The personal comfort adjustments are reduced from +/- 3°C towards +/- 2°C. Another change towards the PCA panel is the reset of the adjustment at the end of each comfort state. In the past, once you increased the temperature these settings remained the same until someone changed them again. Now, after each working day when the controller changes from comfort to standby mode, the deviation is resetted to 0°C.

Besides these programmable changes, the customer invested in IR protection glass foils. These foils diminish the solar influence on room temperature and increase the thermal resistance of the windows.

Although the first measure obviously limits the personal comfort influence of the individuals in the room, both measures resulted in a dramatic fall of comfort complaints.

Heat distribution

The flow temperature of the different heating circuits is automatically adapted towards the outside temperature, reducing the average temperature of these circuits. Each pump is activated by an underlying demand to reduce the overall amount of running hours and heat input into the building.

A further step is the implementation of an OSTP (Optimum Start Stop) algorithm. This algorithm is self learning and adapts automatically the start up - and stop time of the heating/cooling installation as a function of outside temperature, - humidity, inside - and comfort temperature by interpolating the current situation to past situations.

Cooling production & distribution

The cooling compressors in the past were released based on outside temperature and extreme comfort settings. Now the machine will only be released when one of the rooms achieves an inside temperature of 24°C. By running trends throughout the different rooms in the polyclinic we could shift the air conditioning release temperature from 5°C to 15°C outside temperature. Furthermore the flow temperature of the cold water is adapted as a function of outside temperature.

While in the past the machine was working on a fixed 7°C, now we work between 8°C and 12°C.

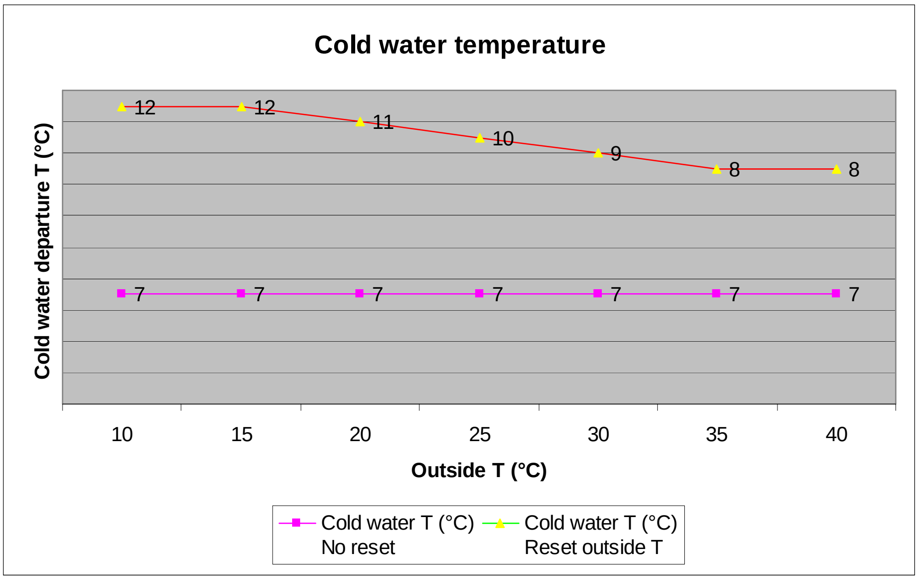

Figure 5: Cold water temperature reset.

As for the heating distribution circuits the same strict pump release conditions and OSTP settings are applying.

Lighting

The light circuits of the hospital are further separated into outside (window) and inside circuits. By means of an outside light intensity measurement we are able to switch of certain circuits. Also during the night there is a control on the minimum amount of lights to guide the employees through the building. For each room an overwork timer function is integrated to allow a general switch off of the lights at 7:00pm. In case someone is still working, by pushing the light button once he is able to work for another hour before the lights will be switched off again.

RESULTS

We will spread this section of the paper over three topics

- Presentation of the impact of the measures on the energy consumption (heat, electricity)
- Presentation of environmental impact & performance of the polyclinic in Aalst, in comparison with other hospitals
- Presentation of the financial return on investment

Impact of the optimization measure on the heat consumption

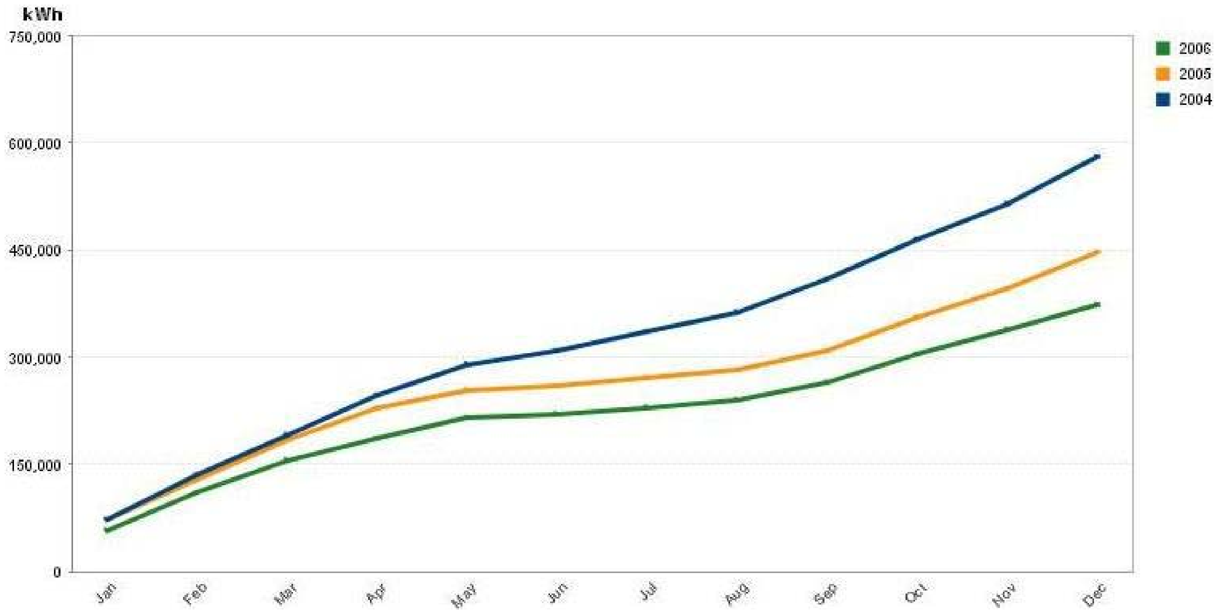

Figure 6: Evolution of the thermal consumption (HDD corrected)

The first graph shows a clear diminution of the thermal consumption throughout 3 consecutive years. A thermal HDD corrected profit of 35,7% is reached over a two year period.

Table 3: Evolution of the thermal consumption (HDD corrected)

Fuel report - HDD corrected			
Year	2004	2005	2006
	kWh	kWh	kWh
Consumption	580074,5	446126,0	372925,3
cumulative saving		133948,5	341097,7
%profit comp. To 2004		23,1%	35,7%

Impact of the optimization measures on the energy signature

The energy signature is showing the thermal consumption as a function of outside temperature.

Three aspects are important when we analyze the energy signature:

- The signature itself should be as low as possible
- The correlation coefficient of the temperature depending points (for the polyclinic this was the interval between 18°C and -10°C) should be as close as possible to 1.
- The rest heat above 18°C (domestic heat) should be as low as possible.

While analyzing the graph we can immediately see that the curve has lowered entirely by more than 10000 kWh (01.2004; 3.68°C → 69824 kWh – 12.2005; 3.91°C → 52096kWh). Also the correlation coefficient (calculated in excel) between consumption and temperature for temperatures below 18°C has increased (2004: 0,95 → 2006: 0,99). On a monthly basis we see an almost linear controlling algorithm.

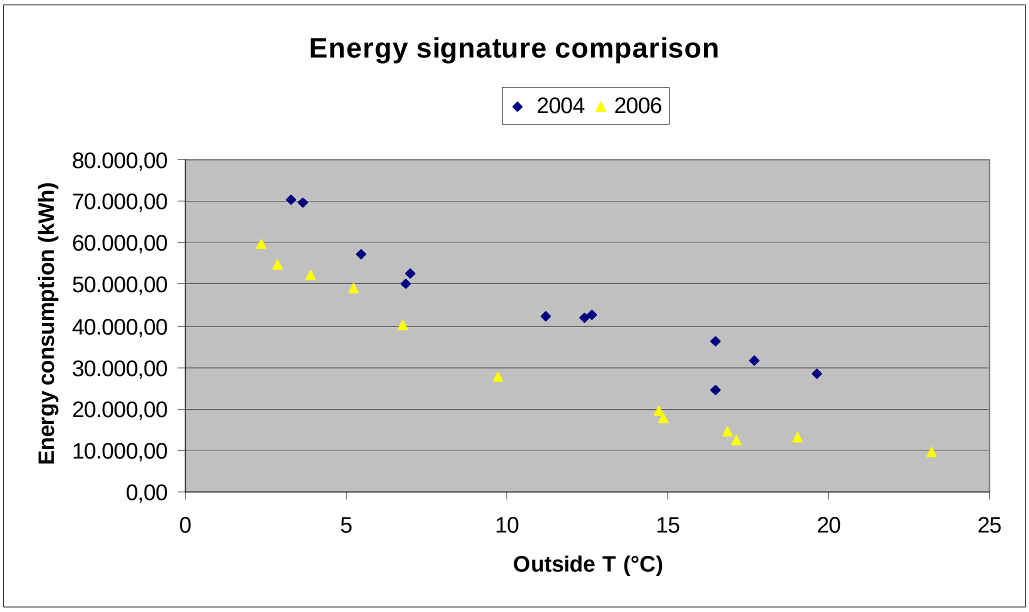

Figure 7: Energy signatures (Before - After optimizations)

Towards the rest consumption for temperatures higher than 18°C, we can also see a dramatic diminution by more than 15000 kWh (08.2004; 19,63°C → 28480 kWh – 09.2006; 19,05 → 13190 kWh). A lower heat consumption during summer and mid season will automatically affect the cooling load on the building.

Impact of the optimization measures on the electrical consumption

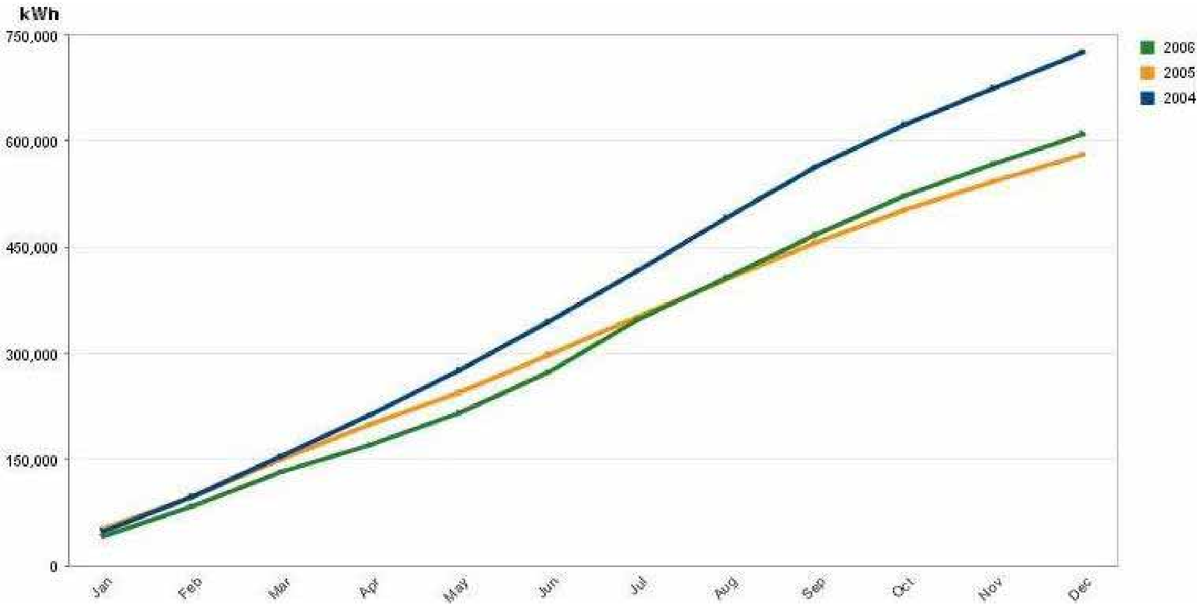

Figure 8: Evolution of the electrical consumption

Table 3: Evolution of the electrical consumption

Electricity report			
Year	2004	2005	2006
	kWh	kWh	kWh
Consumption	725440,0	580928,0	610576,0
cumulative saving		144512,0	259376,0
%profit comp. To 2004		19,9%	15,8%

On the graph and table we can see the impact of the optimization measures on the electrical consumption. In contradiction with the heat consumption we don't see a three year consecutive diminution of the consumption. The answer to this statement can be found in the temperature shift of 2005 in comparison with 2006. The temperature during the typical cooling months (May – October) raised on average with 1,14°C (or 6.8%) . As we have no separate meter registering the electricity consumption for cooling so we can't present a CDD corrected electrical consumption. In the final year comparison we can see a yearly electrical consumption increment of 5.1% for 2006 in comparison with 2005. On average we have lowered the electricity consumption by 18% in comparison to 2004.

After analyzing the thermal and electrical consumption one can see that the audit savings were underestimated.

Impact of the optimization measures on the CO_2 emission

Table 4: Evolution of the CO2 emission

CO2 report			
Year	2004	2005	2006
	kg	kg	kg
Consumption	736070,0	578025,6	573970,0
cumulative saving		158044,4	320144,5
%profit comp. To 2004		21,5%	22,0%

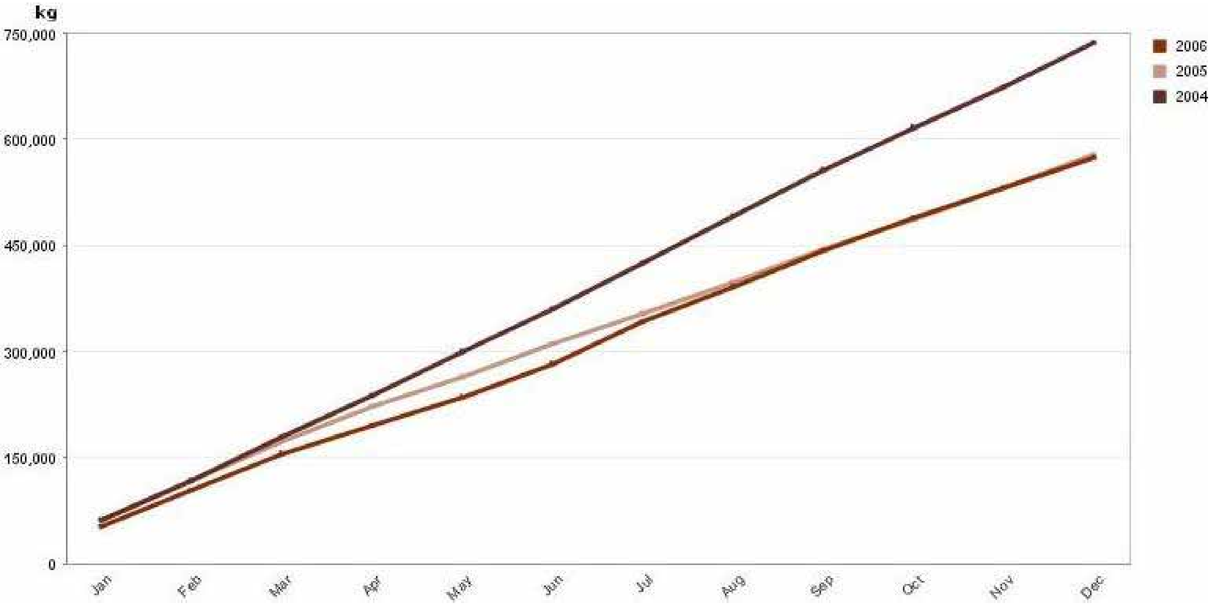

Figure 9: Evolution of the CO2 emission

An immediate effect of an overall diminution of the energy consumption is a reduction in CO_2 emissions. Over a two year period we reached a CO_2 emission drop of 22% representing 320,15 tons of CO_2.

Presentation of the environmental performance of the ASZ-Aalst in comparison to other hospitals

Table 5: Senter Novem benchmark figures[2]

Senter Novem consumption limits					
Specific fuel consumption (m³gas/m²)			Specific electricity consumption (kWh/m²)		
20%	50%	80%	20%	50%	80%
16	50	85	26	95	164

Table 6: Specific consumption figures Polyclinic related to Senter Novem benchmark figures

Specific fuel report - HDD-corrected			Specific electricity report		
2004	2005	2006	2004	2005	2006
m³gas/m²	m³gas/m²	m³gas/m²	kwh/m²	kwh/m²	kwh/m²
11,5	8,9	7,4	144,4	115,6	121,5

Conclusions

In Belgium no official benchmark figures for hospitals are available. So we worked with the official Dutch figures supplied by Senter Novem.

Looking at the thermal consumption figures one can clearly see that the polyclinic was and is still performing well as we have a lower consumption than 20% of all interrogated hospitals. Towards the electrical consumption we get another picture. The consumption is going towards the 80% limit of all interrogated hospitals. A possible explanation of these figures can be found in the fact that there are no residing patients in the polyclinic. No patients mean less thermal consumption for domestic purposes. On the other hand the polyclinic has a denser electrical load because of all the electronic equipment installed for clinical research purposes (scanners, PC etc…), in comparison with an average hospital. Also the fact that the hospital is fully air conditioned is affecting the electrical performance.

Financial returns on investment

Table 7: Financial analysis of the BPO project

Polyclinic ASZ-Aalst	Thermal profit (€)	Electrical profit (€)		Investments (€)	Pay back time (years)
2005	4688,2	13006,1	Audit	10000	
2006	7250,2	10337,8	Services	21000	0,9

Conclusion

We have calculated the savings towards the base year 2004. The BPO services were ended last December and the customer now only pays a fixed cost to supervise his installation from the AOC. The overall payback time for this 2 year optimization project is refunded in less than one year. Besides the on site optimizations the AOC proved it's value in maintaining and optimizing the BMS by means of a web application (see the results of 2006).

Equations

Equation 1: HDD corrected year comparison formula used by EMC software[3]

January: $kWh_{\text{HDD corrected Jan}} = kWh_{\text{actual Jan}} * (HGT_{\text{Ref Jan}} / HGT_{\text{Actual Jan}})\ldots$

October: $kWh_{\text{HDD corrected Oct}} = ((\sum kWh_{\text{Actual Jan + … + Oct}}) * (A/B)) - \sum kWh_{\text{bHDD corrected Jan + … +Nov.}}$

$A/B = (\sum HGT_{\text{Ref Jan + … + Dec}} / \sum HGT_{\text{Actual Jan + … + Dec}})$

The formula above is used by the EMC software to calculate the HDD corrected heat consumption.

The reference year we use is the average Belgian climate at Ukkel over a 30 year period (1970 – 2000).

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Advancing The Management of Firms And Their HVAC Related Businesses for Better Wellbeing Indoors

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SUMMARY

The aim is to advance HVAC related business management by introducing a platform of 52 construction-related business-management concepts published between the years 1990-2005 (based on the pioneering review) and exploring their applicability to managing HVAC related businesses. 30 (58 %) concepts belong to research in construction management and economics, 9 (17 %) project management, 8 (15 %) real estate development, and 5 (10 %) industrial management. Among the eight schools of thought on business management, the combined share of 13 organization-based, 11 Porterian, 10 dynamism-based, and 10 knowledge-based concepts is 84 %. Concept applicability is explored by coupling each concept with the HVAC related context of application, i.e. all HVAC related firms, engineers, contractors, product suppliers, or life-cycle services providers. No applied research tradition in business management exists as part of construction-related management research. Nevertheless, it is posited that this platform enables business managers and researchers to advance their ways of managing a HVAC related business. Many dynamism-based concepts will be among the ones that will turn out to be most profitable by the year 2010.

INTRODUCTION

The background involves the conduct of **the first-ever review of a population** of 52 construction-related business-management concepts published in English between the years 1990-2005 [1]. The sub-results have been introduced as part of the CIB W55 and W65's symposiums in Singapore [2], Toronto [3], Helsinki [4], and Rome [5], and as part of the 1st and 2nd ASCE Specialty Conferences on Leadership and Management [6], [7] during the years 2003-2006. **Construction** involves design, implementation, servicing, and life-cycle aspects of (capital) investments in natural resources usage, energy supply, telecommunications, transportation, infrastructure, manufacturing, and general building concerns. A firm population belongs to seven business-scope groups: (i) technology-intensive contracting, (ii) construction-related contracting, (iii) process engineering, design, and consulting, (iv) construction-related design and consulting, (v) building products and construction materials supply, (vi) real estate ownership, development, and management, and (vii) life-cycle (incl. FM) services.

HVAC related firms and their businesses belong mainly to construction-related contracting, design and consulting, building products supply, and life-cycle services. Today, the advancement of HVAC firm and business management is triggered by owner and user-based driving forces across building markets. Pioneering investments in **green buildings** target reduced energy consumption and their associated costs, increased occupant productivity and worker retention, increased market values, and reduced health liability risks due to better indoor air quality. Green buildings can save more than 250 % of their up-front costs over the course of their 40-year useable life cycles [8]. In turn, **workplace management** is enabling increased effectiveness and accelerated cultural/organizational change [9].

Thus, the following research problem is herein addressed: "What business-management concepts (to be found in the literature) can readily enable managers to advance the ways of managing of their firms and HVAC related businesses for better wellbeing indoors in building markets?" **The aim** is to advance core thinking in HVAC related business management by introducing a platform of 52 construction-related business-management concepts published in English between the years 1990-2005 and exploring the applicability of this platform to actually managing HVAC related businesses. The sub-aims are as follows: (1) To introduce the eight schools of thought on generic business management, (2) to introduce the 52-concept platform and its relatedness to each of eight schools of thought, (3) to explore initially the applicability of 52 concepts to managing a HVAC related business and readily to point out to some concepts that seem to be the most useful ones, and (4) to discuss some promising implications for HVAC related business managers and scholars.

METHOD

Herein, the outcomes of the pioneering review, i.e. a population of 52 construction-related business-management concepts published between the years 1990-2005 is introduced and **the scope of the applicability of these concepts** is initially explored by coupling each potential concept with the perceived primary HVAC related context of application, i.e. all HVAC related firms, engineers, contractors, product suppliers, or life-cycle services providers. In addition, the applicability areas in HVAC related business management of 52 concepts are roughly presented in the text as the concepts' units of analysis.

The two sub-reviews were conducted in the years 1999-2005 by applying **Hart's [10] literature-review guidelines**. The qualitative literature-review method, i.e. the replicable ways of searching, browsing, in-/excluding, retrieving, inferring, moderate coding, describing, and analyzing the construction-related conceptual data have been specified, used, and documented in detail in Huovinen 2003 [1]. **The coherent nature** of managing a business was maintained by focusing on the research on firms and business units that are based in one of the OECD countries. The exception was to allow the references originating from Singapore and Hong Kong to be included due to these authors' British Commonwealth heritage. Over the 16-year review period, three publication channels have been relied upon, i.e. a population of 25 journals, a population of nine publishers, and 22 conference proceedings. Notably, any new concepts that the nine book publishers may have published during the years 2003-2005 are not yet included in the results. The author will submit the three lists of the publication channels on request.

RESULTS

Eight school of thought on generic business management

Business management (BM) concepts are herein seen as abstractions representing objects or phenomena such as managing a HVAC related business [11]. In BM literature, many distinguished authors have replied to the question "What is the primary way (element) of managing a firm's business that will enable managers to set challenging goals and also to attain them?" Based on the fairly comprehensive literature review [1], this author has regrouped the research traditions into **the eight schools of thought on generic BM** as follows: (i) Porterian, (ii) resource-based, (iii) competence-based, (iv) knowledge-based, (v) organization-based, (vi) process-based, (vii) dynamism-based, and (viii) evolutionary BM. It is the core ideas of the founding scholars that make each of eight schools of thought distinct. By the year 2007, it is assumed that the number of business managers **applying Porter's frameworks** is still exceeding the combined number of practitioners relying on the seven other schools in the OECD countries. However, not even the Porterian school can show the empirical evidence of sustained good performance [12].

Platform of 52 construction-related business-management concepts

Overall, **no applied research tradition in BM** exists as part of construction-related management research in the OECD countries. The authors of 52 concepts have added neither very high theoretical contributions, nor deep practical values onto the original generic BM concepts. A real 16-year platform may consist of 65-70 concepts. In this paper, the identified platform is promoted toward HVAC related business managers and interested scholars alike as follows. **The identified theoretical platform** consists of 52 construction-related BM concepts published within 48 references between the years 1990-2005. The authorship consists of 42 (individual, pairs or teams of) authors. There are 72 individual authors. **Within construction-related management research**, 30 (58 %) concepts are primarily related to construction management and economics, 9 (17 %) to project management, 8 (15 %) to real estate development, and 5 (10 %) to industrial management. **Among eight schools of thought on BM**, the combined share of 13 organization-based, 11 Porterian, 10 dynamism-based, and 10 knowledge-based concepts is 84 %. The temporal pattern is still emerging (Table 1).

Table 1. Relatedness of 52 construction-related business-management concepts, published between the years 1990-2005, to the eight schools of thought on business management.

School of thought	Concepts published btw. 1990-2002 (1st review)		Concepts published btw. 2003-2005 (2nd review)		All concepts	
	No.	(%)	No.	(%)	No.	(%)
Organization-based school	9	(24 %)	4	(29 %)	13	(25 %)
Porterian school	11	(29 %)	0	(0 %)	11	(21 %)
Dynamism-based school	7	(18 %)	3	(21 %)	10	(19 %)
Knowledge-based school	7	(18 %)	3	(21 %)	10	(19 %)
Process-based school	0	(0 %)	4	(29 %)	4	(8 %)
Competence-based school	3	(8 %)	0	(0 %)	3	(6 %)
Resource-based school	1	(3 %)	0	(0 %)	1	(2 %)
Evolutionary school	0	(0 %)	0	(0 %)	0	(0 %)
Sum	38	(100 %)	14	(100 %)	52	(100 %)

A range of 54 home-base contexts includes 19 (35 %) worldwide or global, 12 (22 %) UK, 10 (19 %) US, 4 (7 %) Finnish, 3 (6 %) generic, 3 (6 %) Swedish, 1 (2 %) Australian, 1 (2 %) German, and 1 (2 %) Swiss context. **Among business-scope groups**, 19 (37 %) concepts address construction or building, 15 (29 %) project or contracting business, complex product systems, or combined engineering, purchasing, and construction (EPC) projects, 8 (15 %) real estate development and FM services, and 5 (10 %) capital investments-based businesses, 4 (8 %) design and consulting services, and 1 (2 %) building products supply. So far, no applied concepts have been designed for managing a HVAC focused business.

13 concepts for managing a HVAC business in organization-based ways

Generic organization-based BM school proposes that managers can achieve high long-term business performance through organizational solutions in various business contexts such as those of globalization, multi-markets, and multi-stakeholders (e.g. Hedlund 1994 [13]).

13 (25 %) construction-related organization-based BM concepts are coupled with the HVAC contexts of application in Table 2. Originally, capability building is enabled by Davies and Brady, a project-based organization is matched with products by Hobday, PM-centred organizations are advocated by Sauer et al., PM offices are supported by Kendall, one-size-fits-all approach is opposed by Turner and Keegan, a business system is designed by Huovinen, and an organizational model is offered by Artto for project-oriented firms. A systems-change strategy is designed by Leinberger for real estate firms. Partnering pillars are designed by Bennett and partnering is customized by Cheng and Li. Virtual CM firms are designed by Kiiras and Huovinen. Successful contractors are envisioned by Flanagan. A collaborative model is made by Huovinen and Hawk for building products suppliers. The theoretical advancement of 13 concepts seems to be fairly modest. Instead, the practical relevance is fairly high. It seems that Hobday's [14] principles of project-based organizations (PBO) are truly applicable to managing complex HVAC value chains and integrating their high value HVAC systems. A pure PBO consists only of strategic management and HVAC projects. It is probably best applicable among innovative firms.

Table 2. Potential organization-based concepts for managing HVAC related business (n = 13).

Author	Year	Business management concept	Context of application
Davies and Brady	2000	Capability building and interaction	HVAC related firms
Hobday	2000	Project-based organization	HVAC related firms
Sauer et al.	2001	PM-centered organization	HVAC related firms
Kendall	2003	Support to PM office management	HVAC related firms
Huovinen	2004	Organization-based management	HVAC related firms
Leinberger	1993	Systems-change strategy	HVAC related firms
Bennett	2000	Seven pillars of partnering	HVAC related firms
Cheng and Li	2002	Customized model of partnering	HVAC related firms
Artto	1999	Organizational model for PM	HVAC contractors
Turner and Keegan	2000	Project-based organization	HVAC contractors
Kiiras and Huovinen	2004	Virtual CM company management	HVAC contractors
Flanagan	1994	Successful company (in year 2000)	HVAC contractors
Huovinen and Hawk	2003	Client-supplier relationship model	HVAC product suppliers

11 concepts for managing a HVAC business in Porterian ways

Generic Porterian BM school, i.e. a chain of Porter's frameworks, proposes that superior business performance can be achieved by managing a chain of causalities. Managers can ex ante formulate and then implement a longitudinal strategy for their dynamic businesses, (a) by enhancing their understanding of a firm's initial internal and external conditions as well as their ability to make true choices, (b) by formulating and implementing a set of cross-sectional strategies in innovative and integrative ways, and (c) by learning to identify chances (or ”luck”) and to evaluate their roles in inputs, processes, causal relations, and outcomes [15].

11 (21 %) construction-related Porterian BM concepts are coupled with the HVAC contexts of application in Table 3. Originally, planning concepts are recommended by Betts and Ofori. Product differentiation is addressed by Langford and Male. Low cost, differentiation, and speedy response is linked by Rapp. All Porterian modes are recommended by Kale and Arditi. A model for architectural practices is designed by Winch and Schneider. Strategies for A/E firms are applied by Veshosky. Strategies for quantity-surveying practices are redefined by Jennings and Betts. A customer-based project-success metrics is designed by Pinto et al. Five competitive arenas are chained by Huovinen. Value chains for real estate services are applied by Roulac. The applicability of these 11 concepts varies a lot. Typically, Pinto et al. [16] have adapted **Porter's value-chain model** to establishing a project supplier's success metrics. Besides time, budget and performance, a client-driven model enables to make better project decisions. Superior HVAC systems go hand-in-hand with superior services.

Table 3. Potential Porterian concepts for managing HVAC related business (n = 11).

Author	Year	Business management concept	Context of application
Betts and Ofori	1992	Applying Porter's frameworks	HVAC related firms
Langford and Male	2001	Applying Porter's 5F framework	HVAC related firms
Rapp	2001	Adapting Porter's 5F, value chain	HVAC related firms
Kale and Arditi	2002	Mode and scope of competition	HVAC related firms
Winch and Schneider	1993	Four generic strategies (matrix)	HVAC engineers
Veshosky	1994	Applying Porter's 4 strategies	HVAC engineers
Jennings and Betts	1996	Strategy model with IT support	HVAC engineers
Pinto et al.	2000	Value chain of a project supplier	HVAC contractors
Huovinen	2001	International competitive strategy	HVAC contractors
Roulac	1999	Real estate value chain	L-C services providers
Roulac	2001	Strategy for competitive advantage	L-C services providers

10 concepts for managing a HVAC business in dynamism-based ways

Generic dynamism-based BM school proposes that managers can manage successfully their businesses even in highly unstable, even chaotic markets (e.g. Hamel and Prahalad [17]). **10 (19 %) construction-related dynamism-based BM concepts** are coupled with the HVAC contexts of application in Table 4. Originally, dynamic construction BM is enhanced by Chinowsky (with Meredith), de Haan et al., Huovinen, and Langford and Male. Project BM is advanced by Meklin et al. Real estate focused concepts are offered by Barrett, Mitchell-Ketzes, and Osgood. Some concepts are more theoretically advanced while the others are more pragmatic. It seems that Lampel's [18] extension of the core competency theory [17] is highly applicable to managing HVAC related businesses. Core competencies support core HVAC project processes (instead of core products in the original concept).

Table 4. Potential dynamism-based concepts for managing HVAC related business (n = 10).

Author	Year	Business management concept	Context of application
Chinowsky	2000	7 areas of strategic management	HVAC related firms
Lampel	2001	Core competency-based model	HVAC related firms
De Haan et al.	2002	Market-strategy-capabilities fit	HVAC related firms
Huovinen	2005	Recursive global business system	HVAC related firms
Meklin et al.	1999	Project business management	HVAC contractors
Langford and Male	2001	Model of strategic management	HVAC contractors
Huovinen	2002	Global competitiveness of a firm	HVAC contractors
Barrett	2000	Linking a firm's business and FM	L-C services providers
Mitchell-Ketzes	2003	Linking workplaces to business	L-C services providers
Osgood	2004	Strategy alignment model and map	L-C services providers

10 concepts for managing a HVAC business in knowledge-based ways

Generic knowledge-based BM school proposes that managers can nurture their intellectual capital and manage learning processes for the creation of advantages and the transfer processes for the leveraging of them (e.g. Nonaka and Takeuchi [19]). **10 (19 %) construction-related knowledge-based BM concepts** are coupled with the HVAC contexts of application in Table 5. Originally, learning organizations are advocated by Hawk, Davies and Brady, and Love et al. Knowledge concepts are offered by Langford and Male, Robinson et al., Huovinen, Borner, and Walker. Alliances are enhanced by Love et al. Project-portfolio management is enabled by Anell. The theoretical advancement and the applicability vary.

Table 5. Potential knowledge-based concepts for managing HVAC related business (n = 10).

Author	Year	Business management concept	Context of application
Hawk	1992	Continual learning system	HVAC related firms
Davies and Brady	2000	Organizational learning cycle	HVAC related firms
Love et al.	2000	Learning organization model	HVAC related firms
Langford and Male	2001	4 ways of knowledge management	HVAC related firms
Robinson et al.	2002	Knowledge management model	HVAC related firms
Huovinen	2003	Knowledge-based management	HVAC related firms
Borner	2004	Project and success-oriented KM	HVAC contractors
Walker	2005	Knowledge competitive advantage	HVAC related firms
Love et al.	2002	Construction alliance model	HVAC related firms
Anell	2000	Project portfolio management	HVAC related firms

Four concepts for managing a HVAC business in process-based ways

Generic process-based BM school proposes that managers can manage their businesses as sequenced, deliberate and/or emergent processes over time in various contexts such as internationalization, organizational change, and renewal and, thus, achieve high long-term business performance (e.g Beer and Nohria [20]). **4 (8 %) construction-related process-based BM concepts** are coupled with the HVAC contexts of application in Table 6. Originally, project-based approaches are linked with businesses by Anderson and Merna as well as Morris and Jamieson. World-class FM services processes are designed by Kaya et al. and Rogers. The concepts diverge a lot in terms of theoretical advancement and applicability.

Table 6. Potential process-based concepts for managing HVAC related business (n = 4).

Author	Year	Business management concept	Context of application
Anderson and Merna	2005	Business development process	HVAC related firms
Morris and Jamieson	2005	Linked corporate/project strategies	HVAC related firms
Kaya et al.	2004	World-class FM framework	L-C services providers
Rogers	2004	High performance business unit	L-C services providers

Three concepts for managing a HVAC business in competence-based ways

Generic competence-based BM school proposes that managers can manage organizational competences in ways that enable to attain their business goals and sustain above-normal rents (Sanchez and Heene [21]). Only **3 (6 %) construction-related competence-based BM concepts** are coupled with the HVAC contexts of application in Table 7. Originally, competences are addressed by Huovinen and Langford and Male. Capability assessment is enabled by Trejo et al. The theoretical advancement and the applicability are diverging a lot.

Table 7. Potential competence-based concepts for managing HVAC related business (n = 3).

Author	Year	Business management concept	Context of application
Huovinen	1999	Recursive competence-based firm	HVAC related firms
Trejo et al.	2002	Capability assessment framework	HVAC related firms
Langford and Male	2001	International(ization) strategy	HVAC contractors

One concept for managing a HVAC business in resource-based ways

Generic resource-based BM school proposes that managers can achieve high long-term performance in fairly low-dynamism businesses. In Barney's [22] words, above-normal profits can be sustained based on managing a firm's recourses. **1 (2 %) construction-related resource-based BM concept** is coupled with the HVAC context of application in Table 8. Originally, four resource categories are differentiated by Lowendahl. The categories seem to be highly applicable to managing HVAC engineering and design businesses.

Table 8. Potential resource-based concept for managing HVAC related business (n = 1).

Author	Year	Business management concept	Context of application
Lowendahl	1997	Management of 4 resource types	HVAC engineers

No applied concepts for managing a HVAC business in evolutionary ways

Generic evolutionary BM school proposes that managers can act upon the internal and external forces that affect their destiny over time, i.e. to manage intraorganizational variation, selection, retention, and competition processes in ways that enable to achieve high business performance. Strategy shapes destiny and strategy is shaped by destiny (Burgelman [23]). No applied construction-related concepts exist readily for managing a HVAC related business.

DISCUSSION

It is posited that **this theoretical 52-concept platform enables** both business managers and management researchers to choose and advance their ways of managing a HVAC related business as part of building markets. It is envisioned that many dynamism-based concepts will be among the ones that will turn out to be **most profitable** by the year 2010. Thus, it is suggested that HVAC focused business managers (i) dynamize their BM concepts to exploit flexibility among members belonging to their HVAC value chains and (ii) add many new value-based ways of integrating such value chains, offerings, systems, and services.

Further, it is posited that **each of eight schools of thought on generic BM** deserves to be considered each time a HVAC related researcher will stand at the 8-school crossroad and make a choice about the future direction. For example, CLIMA 2007 related researchers could co-set the research goals jointly with involved HVAC focused business managers to craft the first concepts for managing integrated HVAC systems contracting, HVAC systems life-cycle services, HVAC technology-intensive engineering and design services as well as prefabricated HVAC products and materials supply businesses.

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Note: The author will submit a list of 48 references containing a population of 52 construction-related business-management concepts published between the years 1990-2005 on request.

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Preliminary Step in Collecting Data for Commissioning of Existing Buildings (Characterization of buildings, systems and problems)

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SUMMARY

This paper deals with the current results in the survey for collecting data for building commissioning. The first aim of the survey was to make an overview of the most typical buildings, HVAC equipments and their related problems. The second aim was to establish the criteria for both choosing the buildings in the further research and establishing the existing building commissioning tools. The survey is carried out by developing the questionnaire for the building caretakers. The current results could give some of the criteria for commissioning of the existing buildings. The paper gives the list of the buildings, equipments, and problems, for which the commissioning tools should be developed.

INTRODUCTION

Commissioning tools for existing buildings should be developed to be useable in the most common cases of buildings. Actually, commissioning tools should give the solution for the most typical problems in the most common buildings. Therefore it is important to make an overview of the most common buildings in Norwegian building facilities. A questionnaire is developed as the preliminary step in an overview of the building facilities. The idea of the questionnaire is to give as much as possible straightforward answers. In addition, the idea of the questionnaire is to check which and how much of the building data can be obtained for our further studies.

METODS

The questionnaire is developed as a tool for the survey, and consists of eight groups of the questions. Since the questions are defined to be as straightforward as possible, the answers should be either “yes”, “no”, or some numbers. Together with the questionnaire a short guide is attached, too. The questionnaire has to be filled out by the person who is in charge of the building administration, e.g. it could be the caretaker.

The first round of the survey was to send the questionnaire to the caretakers. After getting the answers, a second round was performed based on the unanswered questions. The second round was performed by sending the e-mails to the persons, which filled out the questionnaire.

The survey is still in progress, because it is necessary to have as good as possible overview of building facilities in Norway.

The content of the questionnaire

Building information
These questions have the aim to get general data about the buildings. The most important questions from this group are about: net total area, year of construction, year of the last renovation and if a renovation is planed in the next two years [1].

Building equipment
This group of the questions should show the most used equipment and type of HVAC system in the buildings. Most of the questions in this group should be answered by ticking off. The questionnaire gives the possibility to tick off the most common HVAC equipments, but there is also possibility for additional equipment. Besides the information about equipment, this group of questions asks for: the number of substations, the number of the particular type of HVAC system, and the age of the oldest HVAC equipment in the building. The guide for fill out the questionnaire gives the explanation of HVAC system based on the following definition: "An HVAC system is functional assembly of equipment for air treatment. The assembly of equipment is connected by a net-work that is determined by the system nature (for example radiators are connected by pipes, air-distribution units by ducts, etc.). An HVAC system is one in which energy is transformed from a source of energy through a distribution net-work to spaces to be conditioned [2]."

Indoor environment
These questions deal with the most common problems in the buildings. The most typical problems concerning the indoor environment are suggested in the questionnaire, but also there is a possibility for additional problems. The suggested problems are noted generally, as the following: too warm or cold, unstable temperature, draft and noise. There is no emphasis on when problems occur. These questions should be answered by ticking off, too.

Documentation
The questions treat the building documentation. All the terms in this group of questions were defined based on the Norwegian translation of the commissioning terms [3]. The aim of this part of the questionnaire is to show how much of building documentation are available. The questionnaire asks for the following documentation: design intention, ordinary operation step, as-built records, and (TAB) testing, adjusting and balancing documentation.

Energy use
These questions aim at energy use and the attitude of the building owners in practicing energy efficiency. A question from this group asks about using *Enova* energy efficiency tools. Enova SF is a governmental agency established to deal with the implementation of energy efficiency and renewable energy policy in Norway. This is a pro-active agency that has the capacity to stimulate energy efficiency by motivating cost-effective and environmentally sound investment decisions. Enova SF advises the Ministry in questions relating to energy efficiency and new renewable energy [4]. The questions about energy efficiency attitude among building owners are important for an overview if the buildings need some additional tools for energy efficiency. There are four questions in this group, and they ask about the following: energy bills during the last two years, practicing the energy efficiency measures in the building, practicing some of the authority measures in the energy efficiency, and using the Enova tools.

Measuring
Since access to a building is necessary to measure, these questions ask about it.

Maintenance
These group of the questions aims at maintenance in the buildings. The aim of this part is to connect the problems and maintenance service. There are questions either the building owners have in company an employed maintenance service or a hired company for maintenance service. There is a question if the caretakers attend training for HVAC equipments. In addition, there is a question about the responsibility of a hired company for the maintenance.

Commissioning
The last question is if they would use some of commissioning tools. A short explanation about commissioning is given additionally to this question. Commissioning is a systematic process of ensuring that all building facility systems perform interactively in accordance with the design documentation and intent. It implies accepting the different methods, guidance and procedures. Process eliminating the need for costly capital improvements, and can give short payback time [5].

RESULTS

The following building owners took part in the survey: Forsvarsbygg (the Norwegian Defense Estates Agency), Telenor (Governmental telecommunication company), and NTNU (Norwegian University of Science and Technology) campus. The questionnaire was completely answered by *32* building caretakers from *15* towns in two months through two rounds. Some of the caretakers sent photos, which are given in Table 1.

Table 1. Photos of some of the buildings participating in the survey.

NTNU-Main building (96 years old)

NTNU-Realfabygget(137 HVAC system)

NTNU-Materialteknisk

Military office building - Rena

Telenor Kolstad - Bergen

Military office building - Bodø

Based on the answers, some of the starting criteria for future research can be established. The age of buildings that participated in the survey is from *1* to *96* years. The answer to this question should be straightforward, but in some cases there are few years of construction. The reason for this is because the buildings have been extended through the entire life time, and there is no any regularity how it happens. The question about the year of the last renovation was not answered in *21* cases. The reason is that the caretakers do not know the exact date.

Using the available data about year of the last renovation, the rate of renovation during the building life time can be explained. The relation is shown in Figure 1.

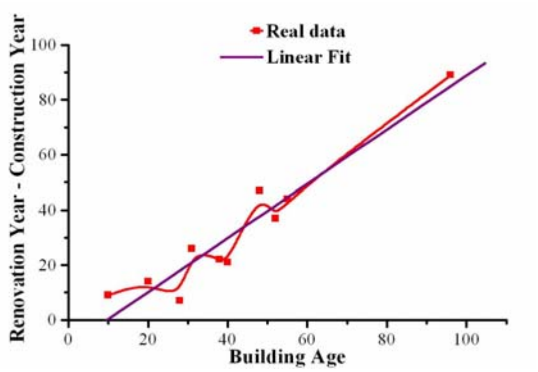

Figure 1. The rate of building renovation during the life time

Figure 1 shows that the buildings had the first renovation approximately *10 years* after the construction. The equation of the fitted line in Figure 1 is:

$$Last\ renovation\ Year - Construction\ Year = -9.52 + 0.98\ Building\ \ Age\,. \qquad (1)$$

The multiplier of the building age is *0.98*. This implies that the building owners invest in the buildings throughout the entire life time. Therefore the building commissioning would be necessary on a continuous basis.

The age of HVAC system is important information for the building commissioning. According to the answers, the relation between the age of building and the age of the HVAC system can be established as in Figure 2.

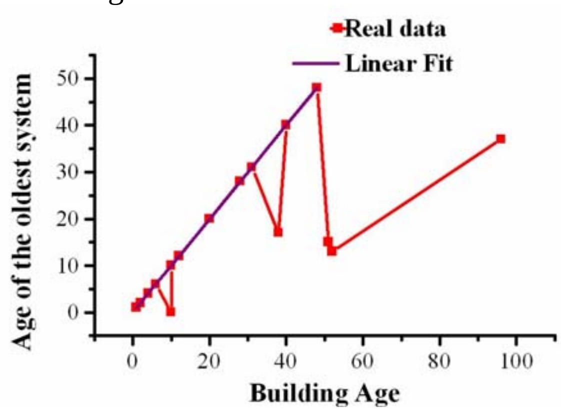

Figure 2. Building Age vs. Age of the oldest system

For *41%* of all buildings there is a linear relation between the age of building and the age of the oldest system. Figure 2 shows that the buildings, which age is less than *50 years*, have at least one of the HVAC equipment of the same age as the building age. The complete data about the year of the construction, year of the last renovation and the age of the oldest system were obtained for *18* buildings. Using this data, the relation between the system age and year of renovation can be established as in Figure 3.

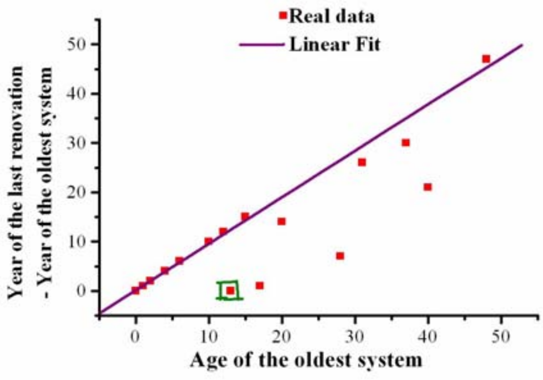

Figure 3. System age vs. Renovation

Figure 3 shows that for 11 buildings there exists a relation between the difference of the year of the last renovation and the year of the oldest system, and the age of the oldest system. Figure 3 shows that the HVAC systems are not replaced after each renovation. This shows that the HVAC systems might be renovated, but not changed completely, because some of the components are still kept. Only in the case of one building, with the green square, (Kaserne Setermoen) the HVAC equipment has been changed completely when the renovation was undertaken. Since most of the buildings are renovated partly, the estimation of the retrofit benefit would be necessary. Therefore the commissioning protocol of a suggested improved measure is necessary.

The answers about the HVAC systems are important for defining the conditions for which the commissioning tools will be developed. *84 %* of all the buildings have a central heating source. 59% of the buildings have both the central heating system and the district heating. Since there is BAS (Building Automation System) in *91 %* of the participated buildings in the survey, the further study will treat buildings with BAS. The obtained results can be used to establish the relation between the number of controllers and the number of HVAC systems as it is shown in Figure 4.

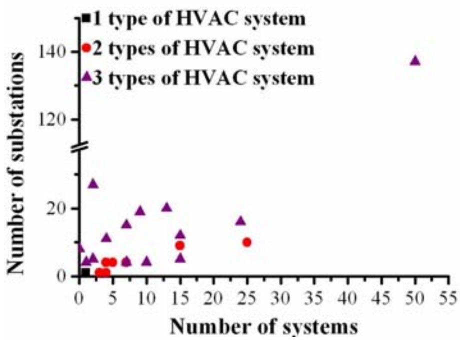

Figure 4. Number of substations vs. Number of systems

Still it is not possible to give the strict criteria for the number of the systems and the number of the substations, but there is a trend that several different types of HVAC systems imply higher number of the substations. The buildings with all three types of HVAC systems (ventilation, heating and cooling) have a higher number of substations than the other, as Figure 4 shows. BAS gives a possibility for the indirect measurements. Therefore a possibility of use the BAS data is valuable for the commissioning procedures.

The question about number of systems was not answered in *22 %*. Some of the answers were obtained through the second round. However, some of the participants in this survey have in the building data base the exact data about the number of the systems (a good example is NTNU Gløshaugen campus). The operation management at NTNU campus uses the data base program LYDIA [6]. Also, they have their own data base in Excel format, which has been developed over seven years.

The results concerning the number of HVAC systems and net total area are shown in Figure 5 and 6.

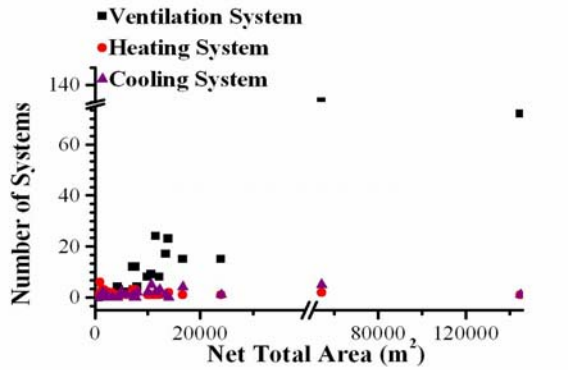

Figure 5. Number of Systems vs. Net Total

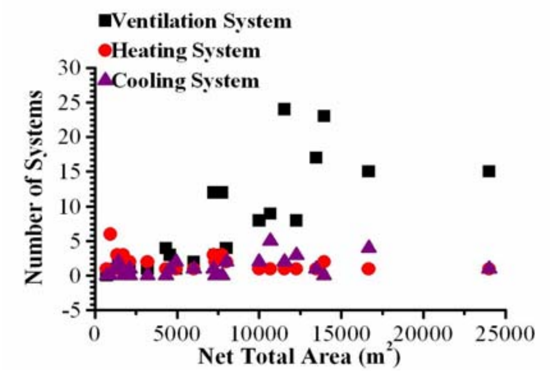

Figure 6. The Subplot of Figure 5

Figure 5 shows a certain relation between the number of the particular type of the system and net total area, but still it is not possible to establish any criteria. In addition, it is obvious that a larger net total area of the building implies a higher number of the ventilation systems. Figure 6 is subplot of Figure 5, so it gives an enlarged view of the narrow area in Figure 5. In Figure 6 is easier to note that the number of the heating and the cooling systems are mostly the same regardless of net total area. Information about building area and the position of HVAC system are important for developing the building models. In addition, the knowledge of the BAS measurements gives a possibility for better understanding of a building performance. Therefore the relation among both the building net total area, number of systems and the number of the substations if beneficial for establishing the commissioning protocols for the building estimation.

The fourth group of the questions should show which building documentation can be obtained. Most of the caretakers have never heard of the term "design intention". This is the reason why only in *19 %* of the answers were "yes". As-built records and TAB documentation can be obtained from *78 %* of the buildings participated. The ordinary operation step documentation exists in *91 %* of buildings. The above information about documentation implies the commissioning criteria that the future studied buildings should have as-built records and TAB documentation and ordinary operation step available.
Energy bills for the last two years are available in *81 %* of the buildings. In most of the buildings, the owners practice some measure for energy efficiency, either on their own way or following the authority measures aimed to the energy efficiency. Enova tools are used in *41 %* of the buildings. These results show that the owners of the buildings practice the energy efficiency measurement in some way, but they also need several tools for that purpose.

Each building in the survey has their own maintenance service employed in the company and all of them attend some courses about the equipment in the building. *69 %* of buildings hire outside company for maintenance. The hired companies have the responsibilities for all HVAC equipments only in *19 %* of the cases, because they are hired only for the particular equipments. This implies the conclusion that for some issues in the building only the employed staffs are in charge. Since there are a few levels of the maintenance services, a discrepancy between their strategies can appear. Consequently, some faults or issues can appear in a building, and therefore a commissioning protocol for the maintenance service can be useful.

For developing and application of the commissioning tools in their buildings, all the participants except one are interested. This is also proven by the fact that in all the buildings we can do research and measuring.

DISCUSSION

For the future study and for developing the commissioning tools, it is necessary that the specific type of equipment exist in at least *50 %* of the buildings. The overview of the most used equipment is given in Table 1.

Table 1. Summarization of the most typical building, HVAC equipment and problems

Type of building	**Num.of Buildings**	**Type of HVAC**	**Equipment**	**Problem**
Office building	13	Ventilation	Water heating coil	Draft
			Cooling coil	
		Heating	Radiator	
		Cooling	Air cooling	Too warm
				Too cold
				Unstable temperature
Hospital	1	Ventilation	Water heating coil	Noise
			Cooling coil	
		Heating	Radiator	
			Floor Heating	
		Cooling	Air cooling	
Barrack	3	Ventilation	Water heating coil	
		Heating	Radiator	Too cold
			Floor Heating	
Multipurpose	12	Ventilation	Water heating coil	Draft
				Noise
			Cooling coil	Bad air
				High air temperature
		Heating	Radiator	Too warm
				Too cold
				Unstable temperature
		Cooling	Air cooling	Too warm
				Unstable temperature
Mess hall	1	Ventilation	Water heating coil	
		Heating	Radiator	
Repair shop	1	Ventilation	Water heating coil	
		Heating	Radiator	
			Floor heating	
			Ceiling Heating	
		Cooling	Air cooling	
Training center	1	Ventilation	Water heating coil	
		Heating	Radiator	

The future study in developing the existing building commissioning tools will deal with the occurred problems for the particular type of building and the appropriate HVAC equipments. Based on the answers, for the most used equipments in the particular type of buildings, the occurred problems are found. For each of these problems in the particular type of building and for the particular type of HVAC equipment, commissioning tools should be developed. The summarization of the most typical problem and equipment in the particular type of building is given in Table 1. The questionnaire is still in progress, and list below will be extended.

CONCLUSION

The survey is a useful tool to start research for the building commissioning. The presented results give a few criteria for the commissioning of the existing buildings. In the future study, the building documentation will be required. The age of both building and equipment will not be strict criteria, but persistence in building renovation would be a required criteria. Therefore, the commissioning protocols should estimate the retrofits benefits. Since BAS give a possibility for the indirect measurements, the future studies will consider buildings, which have HVAC systems with BAS. Even though the relation among both the number of the systems, net total area and the number substations is not established, the further work should find such a relation, since it is important to utilize BAS data for a good understanding of the building performance. The actual summarization the most typical building, HVAC equipment and problems should be spread in the future.

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Commissioning in Existing Building Using Computer-Based Tools

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SUMMARY

This paper deals with a ventilation system with a recovery wheel and several related operation parameters. The first aim was to develop a commissioning tool using computer-based tools. The second aim was to find a better energy management control strategy for the ventilation system. The problem in the ventilation system could imply both too warm air and poor air quality. The simulation and optimization tools used in the study are *EnergyPlus*[1] and *GenOpt*[1]. To evaluate the performance of the ventilation system components, it is important to establish the relations between the components. The result of the study suggests measures for retrofit in overcoming a part of the problem of too warm air. In addition, the results indicate energy savings from *15* to *30 %*.

INTRODUCTION

A survey among Norwegian building facilities shows that there is a problem of too warm air and poor air quality in the buildings [1]. This study gives an example how to start an analysis for developing a commissioning tool. In addition, few retrofit measures are tested using *EnergyPlus*.

The heat recovery efficiency is usually studied separately from buildings themselves. Considering significant influence of the heat recovery wheel on the indoor air temperature, it is necessary to study together with the building and HVAC equipment as mentioned in [2]. This study shows how to deal with such a challenge by using computer-based tools. The existing energy management control strategies are mostly heuristic, so there is a necessity for systematically examining the existing energy management control functions and improving them [3]. The study develops the control strategy for the supply-water and tests it.

The building model is developed in *EnergyPlus*. The building has a ventilation system with a heat recovery wheel and a water heating coil. The simple configuration of the equipment is chosen for establishing a basic overview of such a ventilation system. The weather data for Trondheim (Norway) are used in this study.

BUILDING AND VENTILATION SYSTEM DESCRIPTION

The model in this study is developed in *EnergyPlus*. The base of the building is shown in Figure 1. Figure 1 shows the building zoning such that the exterior zones extend six meters inside from the exterior surface as recommended in [4]. Such a way of zoning implies two

[1] Copyright by Lawrence Berkeley National Laboratories, Berkeley, CA, USA

exterior zones, which are ventilated. Hatched zones are ventilated (Figure 1). The ventilated zones have a common ventilation system. In Figure 1, "N" implies the north direction.

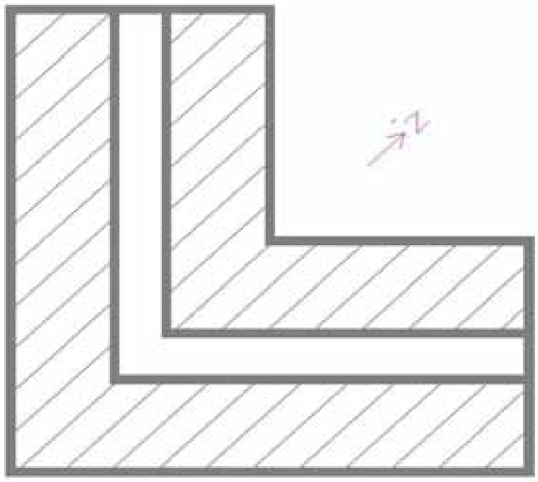


Figure1. Ventilated zones are hatched in building base

The building facades are shown in Figure 2 and Figure 3. The total area of the building is *1350* m^2. The attic and the core zone (non-hatched zone) are assumed to be unconditioned zones. This study treats only the external zones at the first floor, the hatched zones in Figure 1. The total area of these zones is *546* m^2.

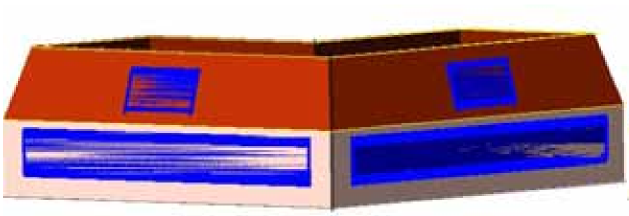


Figure 2. South side of the building

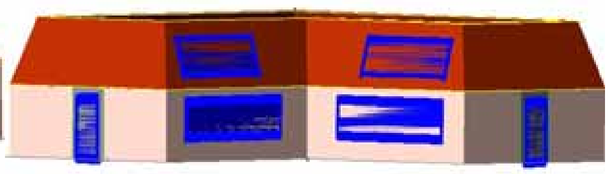
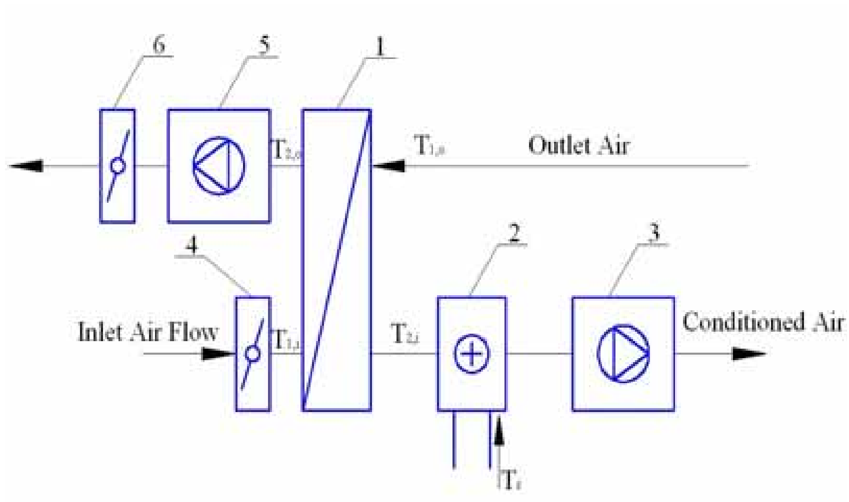


Figure 3. North side of the building

A sketch of the ventilation system is shown in Figure 4. One of the objectives of this study is to find the mutual influence between the heat recovery wheel and the water heating coil. The air flow through the system is *3.3* m^3/s, the larger zone consumes *2* m^3/s, while the smaller consumes *1.3* m^3/s. The amount of air is chosen according to ASHRAE recommendations for air changing per hours in office buildings [5].



Legend:
1. Heat Recovery Wheel
2. Heating Coil
3. Supply Fan
4. Damper
5. Exhaust Fan
6. Damper

- T_s – The supply-water temperature of the heating coil,
- $T_{1,i}$, $T_{2,i}$ – The inlet air temperatures before and after heat recovery wheel,
- $T_{1,o}$, $T_{2,o}$ – The outlet air temperatures before and after heat recovery wheel

Figure 4. Ventilation system

The only internal loads are assumed to be the loads from occupants. The density of people is *0.1 persons/m²* [6]. This implies *32* persons in the larger zone and *22* persons in the smaller zone.

The supply-water temperature of the heating coil is shown in Figure 5. Figure 5 is obtained from BEMS (Building Energy Management System) for the actual building.

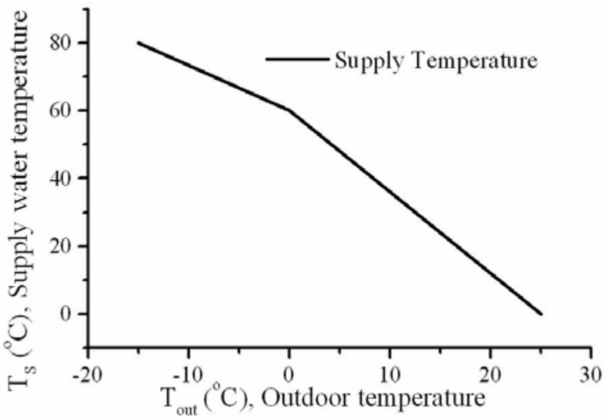


Figure 5. Supply-water temperature curve

The curve in Figure 5 is the same for all the equipment that are using heating energy, except that the heating coil does not operate when the outdoor temperature is above *5°C*. Therefore, the outdoor temperature range from *-15°C* to *5°C* is actual for this study, and the baseline energy use is calculated in the same outdoor temperature range. This outdoor temperature range implies 190 days for the weather data for Trondheim.

METODOLOGY

To find a better energy management control strategy for the ventilation system, it is important to find the mutual relationships between the important parameters. This study finds these relationships by establishing an optimization problem and post-processing the optimization results. Before the optimization problem is defined, the heat recovery model is involved.

EnergyPlus **model for the heat recovery wheel**

The sensible energy efficiency of the heat recovery unit, ε , in *EnergyPlus* is:

$$\varepsilon = \frac{T_{2,i} - T_{1,i}}{T_{1,o} - T_{1,i}}. \tag{1}$$

The notation in Equation 1 is according to Figure 4, where $T_{1,i}\left({}^{o}C\right)$ is the air temperature of inlet air before the heat recovery wheel, $T_{2,i}\left({}^{o}C\right)$ is the air temperature of the inlet air after the heat recovery wheel, and $T_{1,o}\left({}^{o}C\right)$ is the air temperature of the outlet air before it enters the recovery wheel.

The capacity of the heat recovery wheel is:

$$\dot{Q} = \dot{V} \cdot c_p \cdot \rho \cdot \left(T_{2,i} - T_{1,i}\right), \tag{2}$$

where $\dot{Q}\left(W\right)$ is the heat recovery rate of the wheel, $\dot{V}\left(m^3/s\right)$ is the volumetric air flow rate, which is defined by the user in *EnergyPlus*, $c_p\left(J/kgK\right)$ is the specific heat capacity of air, and $\rho\left(kg/m^3\right)$ is the air density. The surface area of the heat recovery wheel is not taken into consideration in this *EnergyPlus* model.

Optimization Problem

The optimization problem is defined in *GenOpt*. Since the aim is to increase the heat recovery rate of the wheel, the objective function is defined to be the maximum of the heat recovery rate of the wheel, Q_{rec}. Therefore the optimization problem is defined as:

$$\max Q_{rec}, \tag{3}$$

$$\text{and the variables are } \begin{bmatrix} UA \\ T_S \end{bmatrix}. \tag{4}$$

The variables in this problem are: the *UA* (*W/m*^{2}) value of the heating coil and $T_S\left({}^{o}C\right)$, the supply temperature of the heating coil. The *UA* value is multiply of the overall heat transfer coefficient, *U* (*W/m*^{2}*K*), and the area of a heat exchanger. The water flow rate of the heating coil is not considered in this optimization problem, because the defined value of the maximum flow rate in *EnergyPlus* is only a limit for the simulation. The water flow is calculated at each time step, so that the load requested of the coil is met at a flow rate which is less than the defined flow rate. This implies that changing the maximum flow rate will not change any results. The entire optimization process is performed for the volumetric air flow of *3.3* m^3/s. In addition, the same optimization process is repeated for the different values of the outdoor temperature. The range of the outdoor temperatures in this optimization problem is from -15^oC to 0^oC.

The optimization problem is shown in Figure 6. The optimization variable is the *UA* value, but *NTU*, the number of transfer units, is more used in practice to express a characteristic of heat exchangers. The *NTU* value is defined as:

$$NTU = \frac{UA}{\dot{m}c_p}, \tag{5}$$

where $\dot{m}(kg/s)$ is water mass flow rate and $c_p\ (J/kgK)$ is the specific heat capacity of water.

Figure 6 gives the optimization results for the volumetric air flow of *3.3* m^3/s (*11880* m^3/h). The results are obtained for the range of the outdoor temperature from *-15*^oC to *0*^oC. For the particular couple of *NTU* and the supply-water temperature, the heat recovery rate of the wheel can be obtained from Figure 6. This heat recovery rate gives a particular value of the inlet air temperature after the heat recovery wheel, $T_{2,i}$, for a particular value of the outdoor temperature.

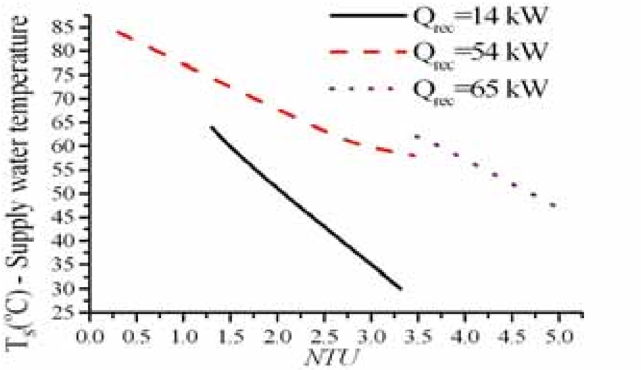


Figure 6. Curves of constant wheel capacity

Figure 6 shows that if the supply-water temperature of the heating coil increases and the value of *NTU* increases, then the heat recovery rate of the wheel increases. Similar figures could be established for the heat rate of the heating coil, Q_{coil}, and zone indoor temperature, T_i.

Post-processing of the optimization results
By using the optimization results from the above problem, the appropriate relations for minimizing energy use can be established.

The optimization results are used to establish two relations:
1. Heat recovery rate of the wheel vs. heat rate of the heating coil, and
2. Indoor temperature vs. heat rate of the heating coil.

The above relations are established for the different values of the outdoor temperature, T_{out}, and in this case they are: *-15*^oC, *-10*^oC, *-5*^oC, and *0*^oC. The results are shown in Figure 7. All the results are valid for the optimal combinations of the optimized parameters.

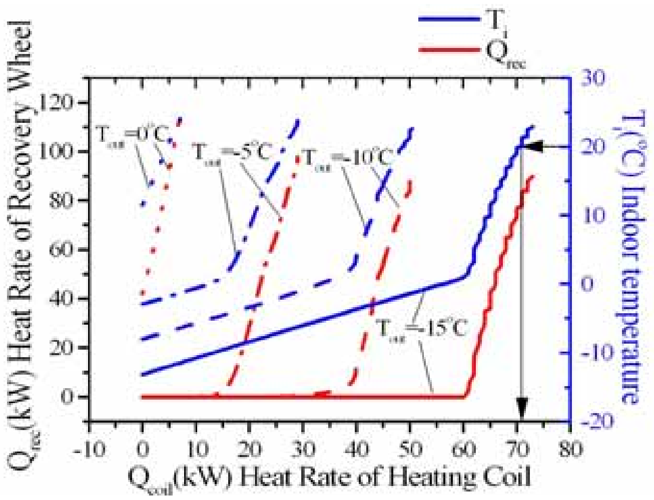


Figure 7. Heat recovery rate of the wheel and
Indoor temperature vs. Heat rate of the heating coil

Figure 7 shows that the relationships between the heat recovery rate and the heating coil rate are different depending on whether the heat recovery wheel recovers energy or not. For example, for $T_{out} = -15^oC$, the heat recovery rate of the wheel is negligible until the heat rate of the heating coil reaches *60 kW*. In addition, the slope of the indoor temperature line is different depending on whether the recovery wheel recovers energy or not. If the heat recovery wheel recovers energy the slope is steeper. This appears because when the recovery wheel recovers the energy, then the reached indoor temperature is higher. Figure 7 shows that when the outdoor temperature increases, the heat rate of the heating coil is negligible comparing to the heat recovery rate of the wheel.

RESULTS

The results in Figure 7 show that it is possible to use a lower supply temperature and to shut down the coil when the outdoor temperature is higher than 0^oC. Based on the optimization results, it is possible to obtain the new values for the new supply temperature curve. Both the new and the current supply temperature curves are given in Figure 8. The consequences of such a measure on energy consumption are shown in Figure 9.

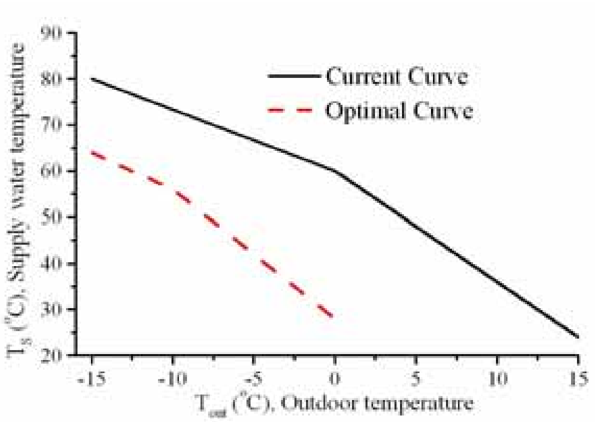


Figure 8. Optimal and Current supply temperature

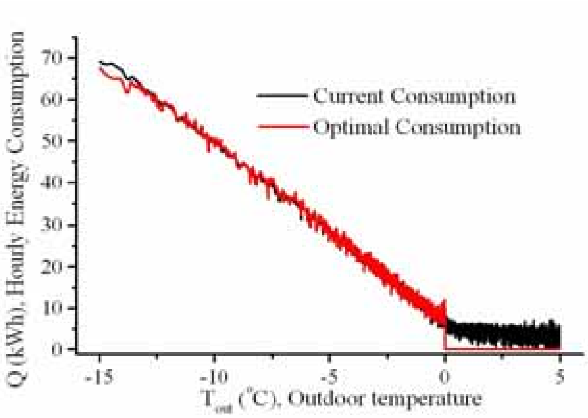


Figure 9. Hourly energy consumption

Figure 8 shows that it is possible that the supply temperature is lower, while the achieved indoor temperature is the same. Figure 9 gives the hourly energy consumption for the outdoor temperatures from -15^oC to 5^oC. In addition, when the coil uses the supply-water temperature as in Figure 5, it works until the outdoor temperature is 5^oC. While, when the coil uses the new curve, it works until the outdoor temperature is 0^oC. The results in Figure 9 are normalized by the outdoor temperature. The highest energy saving by using this measure is archived after the coil is shut down. The energy consumption of the coil is slightly lower when using this optimal supply temperature curve. The reason is that the same amount of the energy is necessary to achieve the desired indoor temperature regardless to the level of the supply temperature.

The energy saving can be determined based on [7] by using an equation, which compares the difference between the continuous end-use baseline and continuous post-retrofit energy use measurement. In this study the energy saving is calculated as:

$$ES = 100 \cdot (Q_b - Q_{pr})/Q_b \tag{6}$$

where $ES\ (\%)$ is the energy saving, $Q_b\ (kWh)$ is the total baseline energy consumption, and $Q_{pr}\ (kWh)$ is the total post-retrofit energy consumption. The energy savings are calculated in the same way for both heating and electric energy. In this study the post-retrofit consumption is possible energy consumption, if the measure would be implemented.

Figure 9 shows that the system shut down implies the highest energy saving, so different cases are tested to estimate when it is possible to achieve the highest energy saving. The cases are the following:

- case 1: the current supply temperature (the supply temperature as in Figure 5);
- case 2: the new supply temperature (the supply temperature as the red-dashed curve in Figure 8) is applied and the coil is shut down when $T_{out}=0^oC$;
- case 3: the current supply temperature and the coil is shut down when $T_{out}=0^oC$;
- case 4: the new supply temperature, the coil is shut down when $T_{out}=0^oC$, and the entire system is shut down in four hours after *20:00*;
- case 5: the current supply temperature, the coil is shut down when $T_{out}=0^oC$, and the entire system is shut down in five hours after *20:00*.

The resulting energy savings from the above measures are given in Table 1. Table 1 gives the energy savings in both the heating and the electrical energy. The energy consumption in Table 1 is the total energy consumption when the outdoor temperature ranges from -15^oC to 5^oC. The heating energy, Q_{coil}, is the energy consumed by the heating coil, while the electric energy, Q_{el}, is the energy for the fans, pump, and the motor for the heat recovery wheel.

Table 1. The energy savings implied by each measure

Case	Q_{coil}(MWh)	Q_{el}(MWh)	Heating Energy Saving (%)	Electric Energy Saving (%)
1	55.6	26.2	0	0
2	44.7	26.2	19.6	0
3	47.0	26.0	15.5	0
4	38.2	21.7	31.0	17.1
5	38.2	20.5	31.0	21.8

The largest part of the total power consumption is made by the fans. Therefore there is no big difference in the electric consumption between the first three cases. When the entire system shut down is applied, there is also a significant energy saving in the electric energy. The energy saving is *19.6 %* when the new curve is applied for the supply temperature. In addition, it is possible to save *15.5 %* of energy by using the current curve for the supply temperature, but shut down the coil when the outdoor temperature is higher than 0^oC. The highest possibility of the heating energy saving of *31 %* is possible when the entire system is shut down during non-working hours.

The results of the study show that the supply-water temperature, as in Figure 5, is high when the outdoor temperature is above 0^oC. The supply-water temperature of the heating coil could be controlled in different way than in Figure 5. In addition, the heating coil works until the outdoor temperature is 5^oC, while the heating coil should be shut down then according to the results. This can be proved from Figure 7, which shows that for the outdoor temperature of 0^oC the heat rate of the heating coil is negligible compared to the heat recovery rate of the wheel.

CONCLUSION

The paper gives the better energy management control strategy for the ventilation system developed by using the computer-based tools. In addition, it defines the optimization problem, and gives the procedure how to post-process the optimization results to get the useful relationships. The results show that the current controlling strategy is one of the causes for too warm air. The results show the possibilities for energy savings from *15* to *30%*. The study did not treat the size of the heating coil, because it was not available. The future work should treat the size of the equipment, and should apply similar measures on a real building.

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C02
Simulation of building systems

Reduction of Space Heating Energy through Minimisation of Life Cycle Cost Using Combined Simulation and Optimisation

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SUMMARY

In the current study, minimisation of Life Cycle Cost (LCC) for a single family detached house is achieved by combined simulation and optimisation. The house has a typical Finnish construction with initial U-values in accordance with the Finnish National Building Code C3 of 2003. The house is heated by a direct electric heating system. The implemented approach is coupling the IDA ICE 3.0 building performance simulation program with the GenOpt 2.0 generic optimisation program to find optimised values of five selected design variables in the building construction and HVAC system. These variables are three continuous variables (insulation thickness of the external wall, roof and floor) and two discrete variables (U-value of the windows and type of heat recovery of ventilation air) which are defined within set limits.

The assumptions for the LCC data are: number of years under study (20 and 50 years), escalation of electric energy price (1 and 5%) and real interest rate (2.94 and 4.90%). The general solution for the LCC minimisation goes towards reducing the energy cost for space heating by investing in the insulation of the house and using better windows. It suggests lowering the U- values for the external wall, roof, floor and the window from their initial values to be equal to, and sometimes lower than, those for the so-called “Low Energy House”. The optimisation results indicate that 28 to 49% reduction in space heating is achieved compared with the reference case (Standard House 2003).

Keywords: life cycle cost, heating energy, simulation, optimisation

SIMULATION AND OPTIMISATION

Simulation tools are increasingly used in analysing building performance. If the designer decides to improve the building envelope and/or HVAC system parameters, he usually makes a guess for the values of the design parameters to be modified and runs the simulation many times trying to find the effect of the changes on the simulation output and to conclude a relation between the studied parameters. This is an inefficient procedure in time and labour. Besides, the relation between the parameters may not be simply understood, especially when there are many parameters to be studied, due to nonlinear interaction between the parameters.

To overcome such difficulties, it is possible to do automatic simulation-optimisation using search techniques that require little effort and time. GenOpt, a generic optimisation program can be used for such a purpose. GenOpt 2.0 [1] is an optimisation program for the minimisation of a cost function that is evaluated by an external simulation program. It is developed for optimisation problems where the cost function is computationally expensive and its derivatives are not available or may not even exist. GenOpt can be coupled to any simulation program that reads its input from text files and writes its output to text files. The independent variables can be continuous variables (with lower and upper bounds), discrete variables, or both. It is to note that minimisation of the cost function could represent minimisation (or maximisation) of any specified objective function (e. g. annual energy consumption, indoor air quality, equipment efficiency etc). However, optimisation is not easy: The efficiency and success of any optimisation is strongly affected by the properties and the formulation of the objective function and by the selection of an appropriate optimisation algorithm. Refs. [2 and 3] indicate that GenOpt has been implemented in the optimisation of building design and HVAC system parameters.

GenOpt has an open interface on both the simulation program side and the optimisation algorithm side. In this study GenOpt is combined with the IDA Indoor Climate and Energy 3.0 program (IDA ICE 3.0). IDA ICE 3.0 is a whole-building dynamic simulation program which makes simultaneous performance assessments of all issues fundamental to a building design: shape, envelop, glazing, HVAC systems, controls, light, indoor air quality, comfort, energy consumption etc. The accuracy of the IDA ICE 3.0 was assessed through the IEA solar heating and cooling programme, Task 22, sub Task C [4]. IDA ICE 3.0 has been chosen by a group of specialist as one of the major twenty building energy simulation programs which are subjected to analysis and comparison [5].

The task in the current study is to minimise the Life Cycle Cost (LCC) of a single family detached house having Finnish construction by combined simulation and optimisation (IDA ICE 3.0 and GenOpt 2.0).

DESCRIPTION OF THE HOUSE AND HVAC SYSTEM

The dimensions and construction of the detached house are generally based on those for the RET project [6]. The dimensions are indicated in Fig. 1.

The U-values for the external wall, roof and floor are 0.25, 0.16, 0.15 W/m^{2}K, respectively. Among other layers in the construction, following are the thickness and thermal conductivity λ (W/mK) of the insulation: 122 mm mineral wool (λ = 0.035) in External wall, 299 mm Blow-in wool (λ = 0.050) in Roof, and 165 mm Polyurethane (λ = 0.026) in Floor. The windows are 3 layers glass having U = 1.4 W/m^{2}K. The U-value for external doors is 0.35 W/m^{2}K. The loss factor of the building's thermal bridges is 8.3 W/K. Those U-values are in accordance with the Finnish National Building Code C3 for 2003 [7]. Therefore, this construction will be called here "Standard House 2003".

The heating system is a direct electric heating system where heating energy is supplied by two means: electric radiators inside the rooms and an electric heater in the air handling unit (AHU). The AHU heater keeps the supply air temperature at 18 ^{o}C when the incoming outdoor air temperature is lower than this limit. There is a thermostat on the electric radiators to keep the temperature of air inside the zone at a lower set-temperature of 21^{o}C. The ventilation system (always ON) supplies fresh air to the house by a supply fan (62 dm^3/s) and an exhaust fan (67 dm^3/s). The exhaust air flow = 0.65 air change per hour. There is a heat exchanger for heat recovery from exhaust air (efficiency = 70%). Good air tightness of the house is considered: the air leakage value n_{50} = 1.0 l/h. The internal heat gain is 5.1 W/m^2 of the floor area. The location of the house is in Jyväskylä in middle of Finland. The hourly weather file for 1979 is considered.

The building with the above mentioned specifications for the construction and HVAC system is considered as a **reference case** in this study.

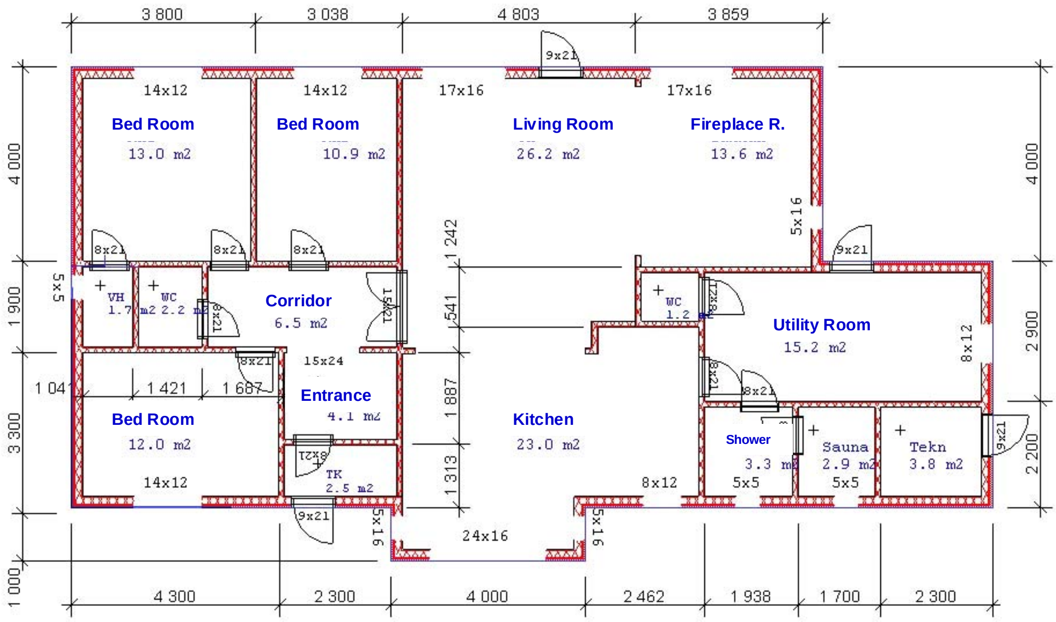

Fig. 1. Dimensions of the single family detached house (internal height is 2.5 and net floor area is 142 m^2) [6].

CALCULATION OF LIFE CYCLE COST

The life cycle cost (LCC) is the sum of the investment and running costs for the building and the system over a specified life span. In the current investigation, the absolute value of the LCC is not calculated but the difference in the LCC for any case (dLCC) from that for the reference case. This way, it is not necessary to include data for all components of the building and system but only the differences produced by variation of specified parameters between the reference case and any other case. Besides, variations in the maintenance cost are not considered. Thus the LCC difference dLCC (€) calculated at present date is

$$\text{dLCC} = \text{dIC} + \text{dOC} + \text{dRC}$$

where dIC is the difference in the investment cost for specified items, dOC is the difference in the operating cost and dRC is the difference in the replacement cost.

The difference in the operating cost is due to the difference in electric energy consumption of the heating system $\text{dOC} = a\, e_p\, \text{d}E$ where e_p is the electric energy price at present date (€/kWh), dE is the difference in annual electric energy consumption E of the heating system (kWh), a is the discount factor which takes into account the effects of inflation and escalation of electric energy price $a = (1-(1+r_e)^{-n})/r_e$, n is the number of years under study, r_e is the real interest rate including effect of escalation of electric energy price $r_e = (r-e)/(1+e)$, e is the escalation in electric energy price (%/a), r is the real interest rate $r = (i - f)/(1 + f)$, f is the inflation rate (%/a) and i is the nominal interest rate (%/a).

The difference in the replacement cost is due to replaced items in specified years $dRC=dR_o/(1+r)^k$, where dR_o is the difference in the replacement cost at present date, k is the year number at which the replacement takes place.

The following data are assumed for the LCC calculations:
Nominal interest rate i = 7% and *inflation rate f* = 2%. Accordingly, the *real interest rate r* = 4.90%. Two values are assumed for the *escalation in electric energy price e* = 1% and a high value of 5%. Two values are assumed for the *life span n* = 50 and 20 years. Two discrete variables are considered (window type and heat recovery type), which have lower life than 50 years. Therefore, two replacements are assumed to take place in year number k =17 and 34 when n = 50 years. When n = 20 years, no replacement costs are considered. The electrical energy price e_p for direct electric heating is 0.0767 €/kWh at 1.4.2005 [8]. *Insulation price* are according to data from Isover Company : Prices for mineral wool in the external wall, blow-in wool in the roof and Polyurethane in the floor are, respectively, 5.63, 3.25 and 26.8 €/m^2 per 10 cm thickness. The change from windows having U-value of 1.4 to 1.0 W/m^{2}K is considered to cost 397 € for all of the windows of the house. The change of the heat-recovery exchanger from a plate type (efficiency = 70%) to a rotary type (efficiency = 80%) is considered to cost 730 €.

DESIGN VARIABLES AND OBJECTIVE FUNCTION

Five design variables are considered in this study: Three continuous variables and two discrete variables. The continuous variables are additional insulation thickness dX of the existing insulation material in the external wall, roof and floor. The discrete variables are: U-value of the window (with two options, 1.4 and 1.0 W/m^{2}K) and type of heat recovery (with two options, plate type with 70% efficiency and rotary type with 80% efficiency). Table 1 shows these variables. Constraints on the continuous variables are indicated as minimum and maximum bounds. The initial and the minimum values correspond to the reference case which means that the optimisation starts from the Standard House 2003.

Table 1. Variable parameters and bounds.

	dXextwall (m)	dXroof (m)	dXfloor (m)	Uwindow (W/m^{2}K)	Heat Recovery
Min (m)	0	0	0	1.0	Plate type (Eff= 0.7)
Max (m)	1.0	1.0	1.0	or	or
Step (m)	0.05	0.05	0.05	1.4	Rotary (Eff=0.8)
Initial value (m)	0	0	0	1.4	Plate type (Eff= 0.7)

The task is to find optimised values of the specified five variables which result in a minimum value for the objective function (dLCC). The combined simulation tool (IDA ICE 3.0) and optimisation tool (GenOpt 2.0) are used to find the optimum values in each case. IDA ICE is iteratively called by GenOpt. The objective function result is checked and values of the design variables are changed in each iteration till a solution is reached. The required annual energy for space heating is calculated by IDA ICE 3.0 based on the hourly input weather file for one year. The used optimisation algorithm is the GPSPSOCCHJ [1]. This is a hybrid global optimisation algorithm that initially does a Particle Swarm Optimisation for all variables and then switches to the Hooke-Jeeves Generalized Pattern Search algorithm to refine the continuous variables.

OPTIMISATION RESULTS

Five cases are studied, and as indicated by table 2, with the following assumptions: Two life spans n = 20 and 50 years, two values for the escalation of electric energy price e = 1% and 5%. These four cases are with a real interest rate r = 4.90%. Case 5 is same as case 3 but with r = 2.94% (nominal interest rate i = 5% and inflation rate f = 2 %). It is to note that 50 years is closer to the building life, while 20 years is closer to the system life. The optimisation results are indicated in table 2. It shows that minimum value of dLCC is reached (negative value) as a result of the reduction made in the electric energy cost for heating (negative value) due to the investment made (positive value) in the insulations, windows and heat recovery and associated replacement costs and as shown in Fig. 2. Highest reduction in LCC is obtained in case 4 when e = 5% and n = 50 years.

Table 2. Studied cases and optimisation results.

Case	1	2	3	4	5
No. of years	20	20	50	50	50
Escl. in electric price	1 %	5 %	1 %	5 %	1 %
real interset rate	4.90 %	4.90 %	4.90 %	4.90 %	2.94 %
dLCC, €	-2102	-4388	-4680	-18437	-8722
dOC, €	-4229	-8042	-8201	-27747	-14371
dIC+dRC, €	2127	3655	3521	9309	5649
Components of dIC+dRC					
dCost Walls, €	983	1393	1556	2683	1966
dCost Roof, €	747	1136	1315	2809	1838
dCost Floor, €	0	0	0	1973	0
dCost Windows*, €	397	397	650	650	650
dCost HtRecovery*, €	0	730	0	1195	1195
Space heating					
kWh/m^2a	71	63	65	50	58
ratio to reference case, %	72	64	66	51	59
Additional insulation thick.					
dXextwall, m	0.150	0.213	0.238	0.409	0.300
dXroof, m	0.156	0.238	0.275	0.588	0.384
dXfloor, m	0.000	0.000	0.000	0.050	0.000
Uextwall, W/m^2K	0.121	0.099	0.093	0.064	0.080
Uroof, W/m^2K	0.107	0.091	0.085	0.056	0.072
Ufloor, W/m^2K	0.150	0.150	0.150	0.116	0.150
Uwindow, W/m^2K	1.0	1.0	1.0	1.0	1.0
Eff. Ht. Recovery	0.7	0.8	0.7	0.8	0.8

* including those for replacement.

Relevant values for the reference case (Standard House 2003) and values from the INDUCON building concept report [9] for the Low Energy House and Minimum Energy House are indicated on the y-axis of Figs 3 and 4 for comparison with results obtained by this study.

The IDA ICE 3.0 calculations show that the required space heating is 99 kWh/m^2a for the reference case compared with 100 kWh/m^2a for the Standard House 2003 as defined by [9]. The optimisation results indicate that 28 to 49% reduction in space heating is achieved compared with the reference case. The highest is for case 4. Fig. 3 shows the required space heating for the studied cases. It shows that for the studied cases, the space heating for the optimum house is lower than that for the Low Energy House.

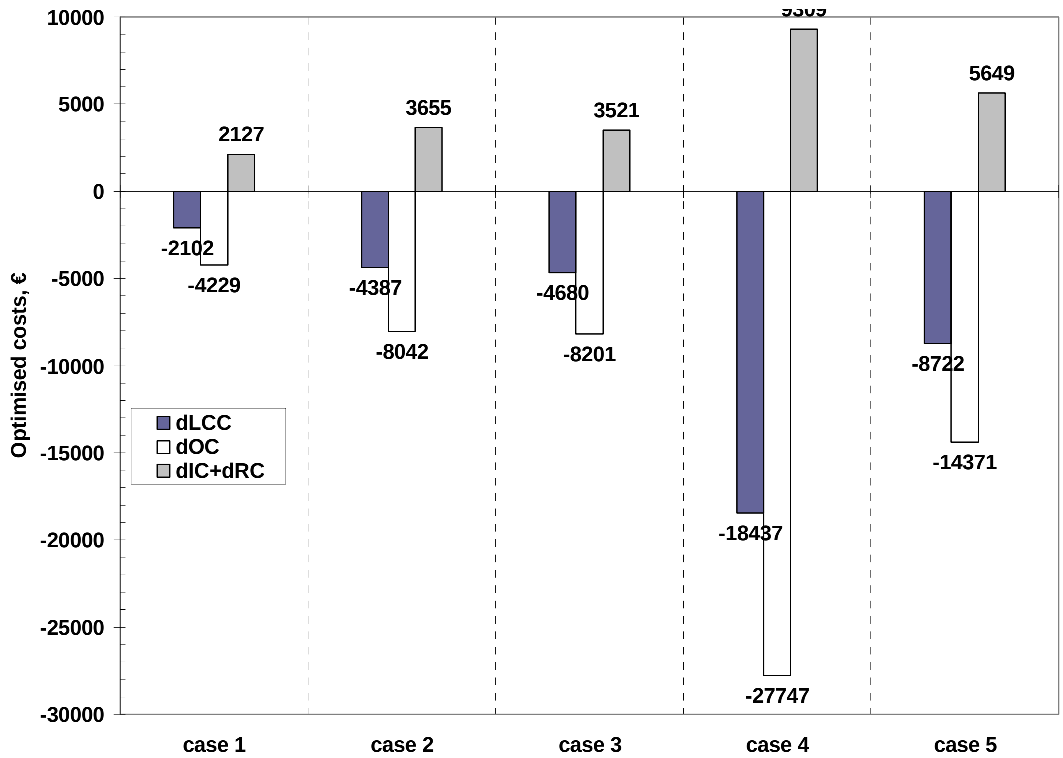

Fig. 2. Difference in LCC (dLCC), energy cost (dOC) and investment and replacement costs (dIC+dRC) for the studied cases.

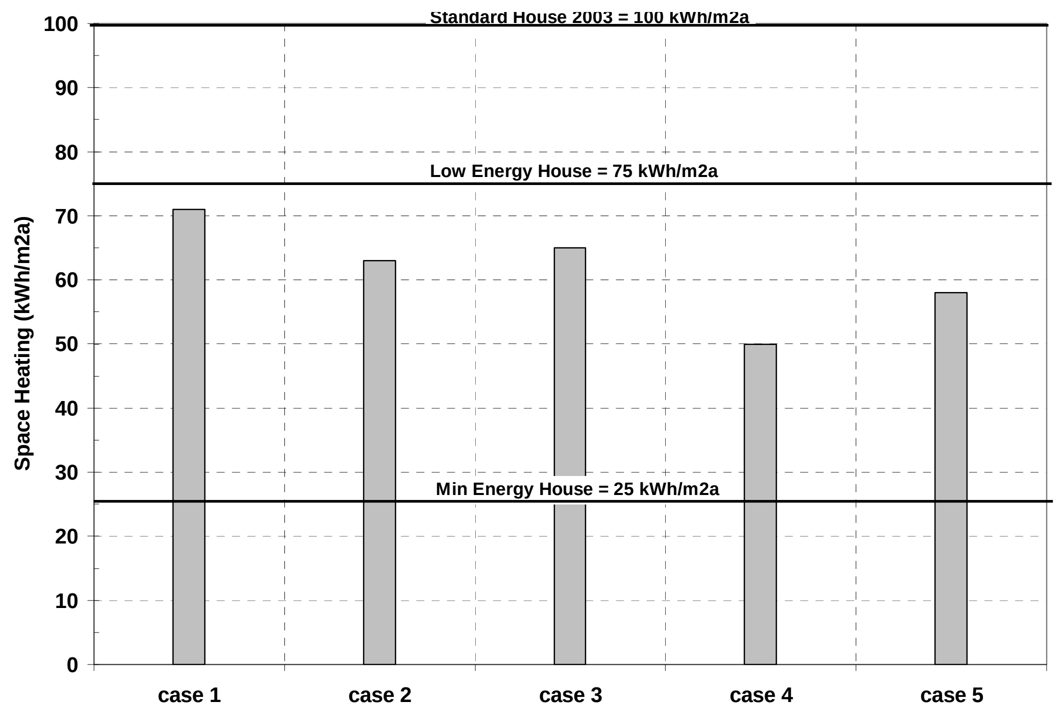

Fig. 3. Space heating of the optimised house for the studied cases.

The results for the U-values are shown in Fig. 4. The optimised U-values for the external wall $U_{extwall}$ are much lower than that for the Standard House 2003. The optimisation suggests increasing the thickness of the mineral wool insulation in the external wall so that $U_{extwall}$ is lower than that for the Low Energy House and even lower than that for the Minimum Energy House for most cases. The U-value of the roof U_{roof} comes close to that for the Low Energy House (which is also the Minimum Energy House) and is even lower for two cases. Due to the high price of Polyurethane as insulation in the floor, the optimisation keeps U_{floor} at its initial value according to the Standard House 2003 (which is equal to that for the Low Energy House) for four cases, while it drops to a lower value for case 4, when e = 5% and n = 50. Polyurethane high price is due to its good characteristics (low thermal conductivity, good compressive strength and resistance to moisture ingress).

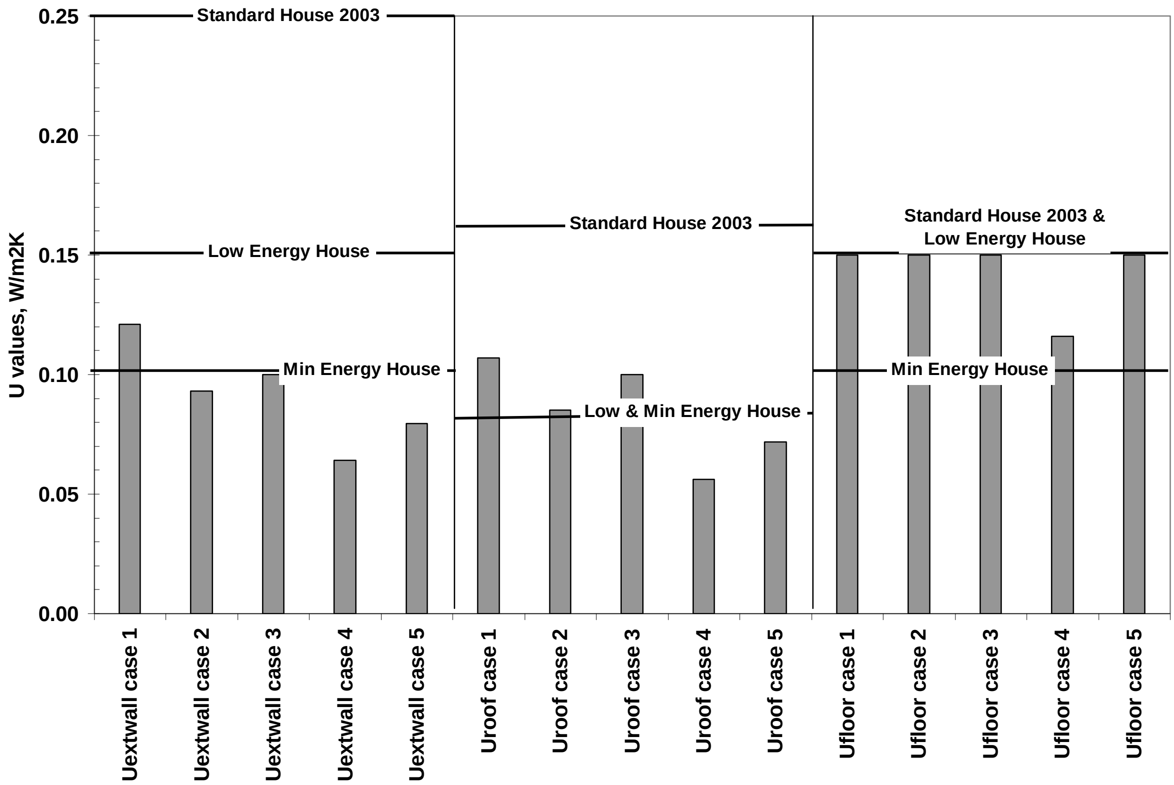

Fig. 4. Optimised U-values for the external wall, roof and floor for the studied cases.

For the two studied discrete variables, table 2 indicates that the optimised U-value for the windows U_{window} is 1.0 W/m^{2}K for the five cases, which is equal to that for the Low Energy House. While for the heat recovery, the rotary type with efficiency of 80% is selected for three cases when e = 5% or r = 2.94%. For case 5, it is concluded from table 2 that assuming the real interest rate r = 2.94% allows making higher investment which produces higher saving in electric energy and higher reduction in dLCC compared with case 3 (r = 4.90%).

CONCLUSIONS

This investigation shows the advantages gained from the implemented approach of combining the IDA ICE 3.0 simulation program with the GenOpt 2.0 generic optimisation program in the minimisation of the life cycle cost of a typical Finnish detached house. The house is electrically heated. This approach enabled finding optimised values of selected design variables in the building construction (insulation thickness of the external wall, roof, floor and U-value of the windows) and HVAC system (type of heat recovery unit) which achieved minimum LCC value for the studied cases.

With different assumptions for the following LCC data; number of years under study (20 and 50 years), escalation of electric energy price (1 and 5%) and real interest rate (2.94 and 4.90%), the general trend of the solution goes towards investing in the insulation of the house and using better windows. It suggests lowering the U-value for the external wall, roof, floor and window to be close to, and sometimes lower than, those for the Low Energy House. The change of the heat recovery system to a more efficient type seems to be feasible for the studied cases when the escalation in the electric energy price is 5% or the real interest rate is 2.94%. The space heating energy for the optimised house is between 50 and 71 kWh/m^{2}a which makes a reduction between 28 and 49% compared with the reference case (Standard House 2003).

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Field Testing of Data Driven Multizone Model Calibration Procedure

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SUMMARY

Field tests of a proposed efficient air flow model calibration method were performed on two classroom/office buildings. Models developed using CONTAM multizone software were tuned via an iterative procedure that sought to maximize the fraction of correctly predicted interzonal flow directions. Site measurements during a concentrated period of testing, including HVAC air flows, envelope leakage, and site weather data were used to update the multizone models. Following an initial group of mandatory measurement, CO_2 tracer gas experiments were conducted to permit independent assessment of model quality using ASTM Standard D5157-97. In one case, the procedure was quite effective, improving the number of correct interzonal flow directions from an initial value of 52% to a final value of 81% and significant improvement was also indicated by the ASTM method. Tuning produced only minor improvement in the second building. Greater difficulties in acquiring site measurements and the greater complexity of the second building are possible reasons that performance was less satisfactory in that case.

INTRODUCTION

In recent years, the public, governmental agencies, and the HVAC industry have shown increased interest in understanding and predicting air and contaminant movement through buildings for purposes of indoor air quality evaluation and the analysis of extraordinary incidents. A number of documents available to the public [1, 2] give generic guidance on making buildings resistant to airborne chemical and biological releases. However, prediction of air and contaminant flow for a particular building of interest is necessary for risk assessment, development of emergency procedures, and system design for building protection against both intentional and accidental events.

Modeling with various types of software can be a quick and cost-effective way to analyze the consequences of an event of interest. Multizone models, although limited by the use of the well-mixed zone assumption, are considered the most practical and useful choice for modeling air and contaminant flows in whole buildings [3]. However, due to the many simplifications and assumptions inherent in multizone methods, an uncalibrated model may be a poor representation of a particular real building. Although highly desirable, calibration can be a time consuming and expensive process.

Field tests were performed as part of a project to develop a method for rapidly developing and tuning multizone air flow models of real buildings constructed with the widely used CONTAM software [4] via relatively simple, inexpensive measurements taken at the site.

This paper summarizes the findings of those tests with respect to the degree of model quality improvement achieved as a result of the tuning process.

METHODS

Equipment and Measurement Techniques

The procedure requires a variety of measurements and the appropriate instruments to perform them, including indoor temperature (hand held thermometer), diffuser air flow rate (flow measuring hood), differential pressure for leakage tests (wireless pressure sensor), interior air flow direction (smoke bottle), AHU flow rates (fan inlet airflow sensor or pitot traverse), and weather conditions (portable weather station). Additionally, transient CO_2 tracer gas tests were performed in an effort to validate the procedure (portable infrared CO_2 gas analyzers).

Test Sites

Calibration exercises were performed for two buildings. The first (RB-1) is a three story (plus basement), 3,810 m^2 building mainly comprised of office spaces, but also including conferences areas and a snack bar. The building is air-conditioned by three air handling units (AHUs). AHU 1 serves three large conference spaces on the first floor with ducted supply and return. AHU 2 serves the remainder of the first floor and the entire second floor. The first floor has ducted supply and return, while the second floor has ducted supply and plenum return. AHU 3 serves the third floor with ducted supply and return. Bathrooms are connected to an exhaust system and receive makeup air through transfer grilles.

The second building (RB-2) is a larger, more complex five story (plus basement), 8,500 m^2 structure comprised of classrooms, office spaces, and conferences areas. The area excluding the basement, which was not modeled, is 6,920 m^2. The first two floors house classrooms and medium-sized lecture halls, while the third through fifth floors house faculty, staff, and graduate student offices. Each floor is served by its own AHU via ducted supply and return. The first floor also has two large lecture halls with dedicated single-zone AHUs, but these are isolated from the rest of the building and were not considered in the modeling.

Model Tuning Procedure

The tuning procedure was initially developed from “virtual building” testing in which a simplified multizone model was tuned to match a reference model through the use of simulated measurements (i.e., the use of data values from the reference model to update the simplified model) [5 - 7]. The approach is similar to the heuristic method described by Musser et al [8], differing mainly through the use of a more structured technique based on iterative improvement of a suitable performance metric.

One possible quality measure is ASTM Standard D5157-97 [9]. However, this standard requires the use of tracer gas testing and was deemed too difficult and costly for widespread application by building modelers or owners. After considering various options, the fraction of correctly predicted interzonal flow directions (or equivalently, the relative pressurization of zones) was selected as the principle quality metric. A flow direction map of a building can be developed rapidly and inexpensively using simple hand-held flow visualization devices such as smoke bottles.

The first step in the procedure is initial model development. This model is based on construction documents and data values from literature (e.g., component leakage, terrain coefficients, etc) and requires no site data, although sources such as commissioning reports may be of use, if available.

At the beginning of the site measurement phase, a weather station should be set up to log ambient conditions throughout the test. The first iteration of building measurements includes mapping of interzonal flow directions and measurement of main air flows such as supply, return, and outside air at air handlers and, if possible, envelope and shaft leakage. The model is updated with these data.

Prior to the first model update and after each update, predicted flow directions are compared with measure directions to determine the value of the performance metric and to identify areas of discrepancy within the building. Air flows to zones connected by flow path with incorrectly modeled flow direction are measured (Figure 1), the model is updated, and new predictions of interzonal flows are generated. This process is repeated until a stopping criteria is reached, which may be the point of diminishing returns in improvement of the predicted flow direction distribution, the completion of all possible measurements, or some other condition determined by the analyst.

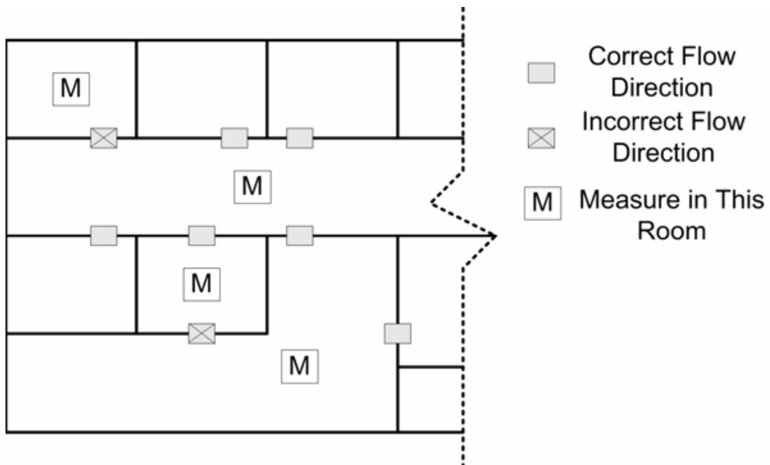


Figure 1. Example of measurement guidance based on location of incorrect airflow directions

The final step in tuning is an analytical process in which optimal (in the sense of reducing error) values of difficult or impossible to measure model parameters are determined. To date, this processs has been applied to four parameters: average exterior leakage, average interior leakage, average shaft leakage, and terrain coefficient (used to calculate wind pressure on the envelope in CONTAM). Any of these parameters that have been measured previously are excluded from this step.

Validation Methodology

Two key questions arise in the validation of the tuning procedure. The first is whether the measurement and model correction process leads to improvement in the correct flow direction performance measure. The second is whether improvement in this metric parallels improvement as measured by an independent standard such as ASTM D5157.

ASTM D5157 employs six statistical measures (correlation coefficient, regression slope, regression intercept, normalized mean square error, fractional bias, and fractional variance) to compare modeled and observed tracer concentrations at a given sampling location. Satisfactory ranges are defined for each parameter. How to apply the standard to multiple sampling points is not discussed in the standard. In the present case, the approaches used were 1) to tabulate the percentage of all metrics (six per sampling location) that fell within the satisfactory range at each stage of correction and 2) to examine the distribution of values of a particular metric at different points in the tuning process.

In RB-1, releases were performed in all three AHUs in turn and measured at all AHU returns (three locations) with a repeat of each release. In RB-2, releases were performed at the AHUs serving floors one, three, and five and measured at all five AHU returns. The quantity of CO_2 injected was intended to raise concentration in the area served by the release AHU by1600 ppmv in order to obtain an easily measured signal.

RESULTS

RB-1 Model Tuning

Tuning of the model of RB-1 was completed in seven iterations, as outlined in Table 1. It should be evident from the decreasing number of measurements at later iterations that the measurement time per iteration decreased significantly as the process progresses.

Table 1. Summary of RB-1 Tuning Process

Iteration	Description
0	Model developed using design document data
1	Interzonal air flow directions and supply, return, and outdoor air flows of all AHUs are measured. Recording of weather conditions (temperature, wind speed) begins.
2	122 diffuser measurements based on location of incorrect airflow directions
3	24 diffuser measurements based on location of incorrect airflow directions
4	10 diffuser measurements based on location of incorrect airflow directions
5	6 diffuser measurements based on location of incorrect airflow directions
6	2 diffuser measurements based on location of incorrect airflow directions
7	Interior leakage, exterior leakage, shaft leakage, and terrain coefficient & exponent are adjusted based on minimization of regression equation.

Figure 2 shows the progression of the percentage of correct interior airflow directions and the percentage of satisfactory ASTM metrics for the seven iterations described in Table 1. A single point in the ASTM data set represents, in this case, one hundred and eight data points: 2 sets of releases x 3 releases per set (at AHU 1, 2, and 3) x 3 measurement points per release x 6 statistical metrics per measurement points. The percentage of correct interior airflow directions is initially 52% and increases to a final value of 81% over the course of the tuning exercise. The greatest improvement occurs during iterations one (to 63%) and two (to 72%). The automated tuning process (iteration 7) produced no change in this metric, probably because the only parameter changed was shaft leakage value.

In the base model, 31% of ASTM D5157 metrics were within satisfactory ranges. The first iteration of tuning increased the total to 41%, but subsequent iterations, evaluated in this way, produced little change. The lack of improvement in later iterations is very likely the result of there having been a small number of measurement locations in the field test, and the placement of those sampling points in the AHU returns. The concentration of tracer returning to these locations represents the average of all connected zones, therefore, changes in flows between zones are likely to have little effect on what is observed there. Sampling points within spaces would be needed to provide a more refined indication of the effect of local flow adjustments. Previous application of ASTM D5157 during “virtual building” tests permitted evaluation of the ASTM statistical metrics in every zone. In these tests, improvement in quality as measured by the ASTM standard paralleled quality as indicated by improvement in correct flow directions and continued throughout the tuning process [5-7]

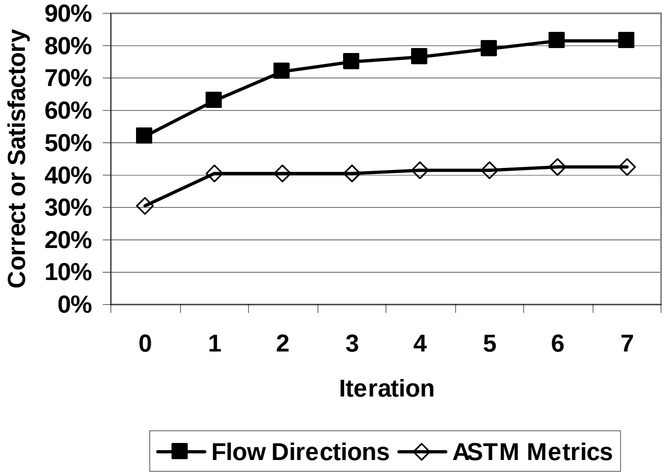


Figure 2. Correct interzonal flow direction and satisfactory ASTM metrics for RB-1 model tuning.

The use of a binary satisfactory/not satisfactory criterion for evaluating model quality using ASTM D5157 does not account for the possibility that unsatisfactory metrics move closer to the satisfactory range as a result of tuning. They may remain unsatisfactory as defined by the standard, but nevertheless are improved. One way to investigate this issue is to compare the distribution of values of a metric with the ASTM D5157 criterion over the course of several tuning iterations. For example, Figure 3 shows a cumulative distribution plot of correlation coefficient values from iterations 0 and 1 (during which essentially all of the change in ASTM metrics occurred). Lines indicating perfect and minimally acceptable values are shown for reference. . This result was the same for all statistical measures (except normalized mean square error, which showed no change), so this way of analyzing the data lends credence to the previous analysis.

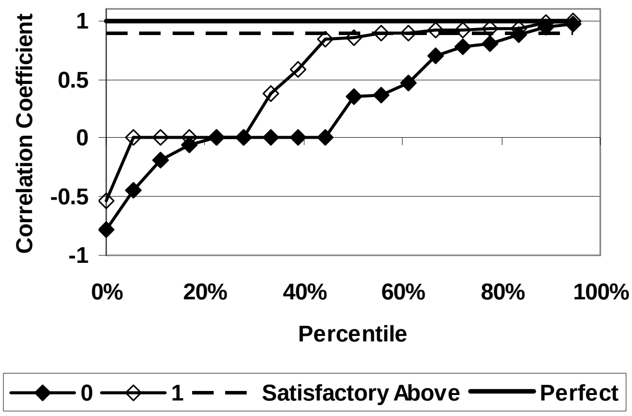


Figure 3. Percentile distribution of correlation coefficient for RB-1 tuning for iteration 0 and iteration 1.

RB-2 Model Tuning

The model of RB-2 was tuned in four iterations, described in Table 2. Iterations 4a and 4b are Alternative applications of regression to determine unmeasured parameters. Iteration 4a included estimation of exterior leakage while 4b did not.

Table 2. RB-2 tuning steps

Iteration	Description
0	Model developed using design document data
1	Interzonal air flow directions and supply, return, and outdoor air flows of all AHUs are measured. Recording of weather conditions (temperature, wind speed) begins.
2	229 diffuser measurements based on location of incorrect airflow directions
3	52 diffuser measurements based on location of incorrect airflow directions
4a	Interior leakage, exterior leakage, shaft leakage, and terrain coefficient & exponent are adjusted based on minimization of regression equation.
4b	Interior leakage, exterior leakage, shaft leakage, and terrain coefficient & exponent are adjusted based on minimization of regression equation. Measured exterior leakage is used

Figure 4 shows the progression of the percentage of correct interior airflow directions and the percentage of satisfactory ASTM metrics.

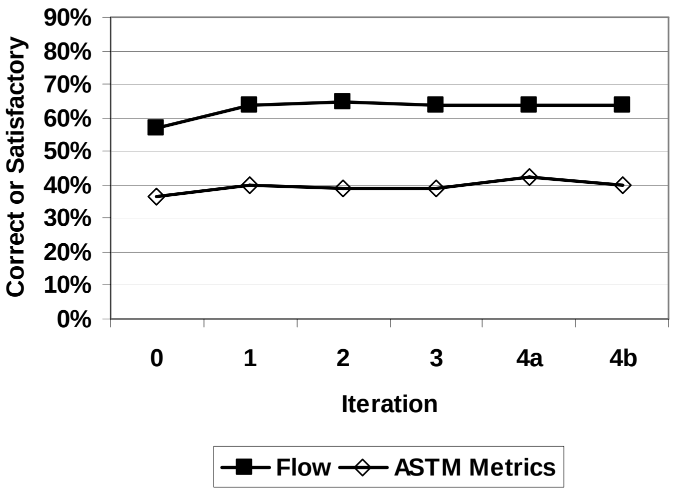
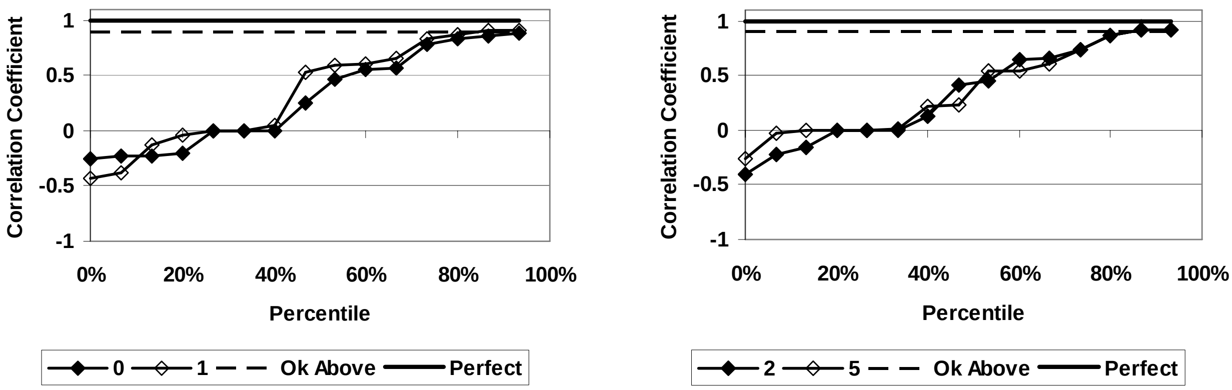


Figure 4. Correct flow direction and satisfactory ASTM metrics for RB-2 model tuning.

The intial percentage of correct interior airflow directions was 57%. The first iteration increased correctly modeled directions to 64%. Subsequent efforts to tune the model achieved only an additional 1% improvement. Automated tuning did not change the interior, exterior, or shaft leakage, only the terrain coefficient and wind pressure exponent. Since there was not much wind at the time of the test, it is not surprising that this change had little effect.

The overall change in the percentage of satisfactory ASTM metrics also was small, starting at 37%, and increasing to 40% after iteration 1. Regression based tuning of leakage and terrain data increased the satisfactory metrics to 42%. Use of measured envelope leakage characteristics resulted in slightly worse performance (40% satisfactory) than use of a predicted value.

Distributions of correlation coefficient calculated as part of the ASTM D5157 analysis (Figure 4) also show modest improvement from iteration 0 to iteration 1 and little change thereafter, for the same reasons noted with respect to building RB-1. It should be noted in Figure 5 that in this application of the proposed method correlation coefficient at some locations improved as a result of tuning while at others it degraded.



(a) (b)

Figure 5. Percentile distribution of correlation coefficient for RB-2 model tuning. (a) Iterations 0 and 1. (b) Iterations 2 and 5.

DISCUSSION AND CONCLUSIONS

The tuning of smaller, simpler building RB-1 was successful in terms of improvement in both predicted interzonal flow directions and as indicated by ASTM D5157. The application of the tuning process to the larger more complex building RB-2 produced little evidence of improvement. One possible explanations for the lack of success in tuning the RB-2 model are that the model failed to capture certain important features of the building that were not affected by any of the model modifications made on the basis of measured data. A second possibility is that factors like distribution system leakage that were not measured played a role. A third possibility is that control errors occurred during testing that put the system in some other status than was assumed.

Main air flow measurements assigned to the first iteration produced the largest improvements in both cases. Diffuser measurements based on the location of incorrect interior airflow directions produced smaller, local improvements in air flow direction for RB-1, but had little impact on RB-2. As noted, improvement in ASTM D5157 metrics after iteration 1 was predictably small because of the limited number of tracer measurement points and their location in AHU return air streams. Using regression equation optimization to tune difficult to measure leakage and wind parameters was the least effective tuning measure.

It was observed that the characterization of model quality can vary significantly with the measure used. In the tuning of the model of RB-1, the airflow direction metric improved significantly throughout the tuning process, while the interpretations of ASTM D5157 improved mainly during the first iteration. Results of RB-2 tuning demonstrated that an aggregate measure of quality such as the total number of satisfactory ASTM D5157 metrics can disguise finer scale changes.

Although ASTM D5157 is quite detailed, it leaves unresolved several significant issues regarding what it measures and how it should be applied. First, it defines quality at a particular sampling point relative to a particular tracer release. If the release or sampling location changes, the assessment of model quality will also change. Second, it defines quality in terms of six different metrics and does not indicate whether a model can be satisfactory if one or more of these measures falls outside the satisfactory range. Finally, it does not address the issue of how to judge model quality in a complex building in which multiple sampling locations are used. All of these issues should be addressed in future development of this standard.

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An integrated model-based approach to building systems operation

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SUMMARY

This presents a research effort toward an integrated model-based approach to energy-efficient operation of building systems. The ingredients of the corresponding concept are as follows: i) A comprehensive building state model underlines all operative entities and activities in the life-cycle of the building. This model includes information on building context (e.g. weather conditions), building topology, components, and systems, as well as building occupancy (user presence and actions); ii) The model is updated real-time via a sensory infrastructure including sensors for outdoor and indoor environmental conditions, occupancy presence and actions, state changes in control devices; iii) The building state model is provided to multiple applications pertaining to facility management and control systems. Such applications use various tools (including building performance simulation, trend analysis and learning algorithms) in order to anticipate the state of building and indoor climate as a result of alternative control options.

INTRODUCTION

Complex buildings are frequently affected by shortcomings in the configuration and operation of building service systems, resulting in sub-par indoor climate conditions and poor energy performance. To address these concerns, this contribution presents a research effort toward an integrated model-based approach to energy-efficient operation of building systems. The ingredients of the corresponding concept are as follows: i) A comprehensive building state model underlines all operative activities in the life-cycle of the building. This model includes information on building context (e.g. weather conditions), building topology, components, and systems, as well as building occupancy (user presence and actions); ii) The model is updated real-time via a sensory infrastructure including sensors for outdoor and indoor environmental conditions, occupancy presence and actions, state changes in control devices; iii) The building state model is provided to multiple applications pertaining to facility management and control systems. Such applications use various tools (including building performance simulation, trend analysis and learning algorithms) in order to anticipate the state of building and indoor climate as a result of alternative control options. To demonstrate the viability of the proposed approach, a 1:1 two-cell office space has been set up in our laboratory. The facility is equipped with systems for heating, ventilation, lighting, and shading. The state of weather conditions is monitored via a weather station. Indoor environment data are collected via sensory units measuring temperature, relative humidity, air flow speed, illuminance, etc. Changes in the configuration of the cells and workstations (e.g. location of furniture and partition elements) as well as occupancy information are captured via an optical location-sensing system. These data are updated regularly and provided to the building systems control unit along with users' feed-back regarding their indoor climate preferences. The building systems control unit uses building performance simulation to predict the implications of alternative control device states and identifies (through comparison and ranking of the simulation results) and brings about a configuration of the control device states that is preferable both in terms of energy efficiency and indoor climate.

We first describe two instances of automated model actualization, namely the actualization of a context mode (sky luminance distribution pattern) that can be used for real-time simulation-based lighting systems control, and the actualization of the room configuration model via location sensing. We then describe the principles and application of a model-based systems control strategy using the example of lighting controls. Subsequently, we illustrate the development of a test-bed for the integrated realization of a model-based multi-system energy-efficient building systems control strategy.

APPROACH AND RESULTS

To realize a model-based building control strategy, the building must possess a model of itself, including both context and components. To illustrate this point in this section, we first use the case of dynamic generation of sky luminance model that can be used for real-time simulations toward lighting systems control. We then illustrate the process of room geometry and configuration model generation and updating via location sensing.

Sky-Scanning

The sky model is generated on a real-time basis using calibrated digital photography. Toward this end, the building's weather station is augmented with a digital camera with a fish-eye converter. From images, the sky luminance model is extracted in terms of distinct luminance values for 256 sky patches (Figure 1). Digital images of the sky are continuously taken, analyzed and calibrated real-time to construct the sky model for the simulation application. The calibration is necessary, as the camera is not a photometric device. It is possible, however, to derive reliable photometric data from properly calibrated digital images [1]. The approach to calibration involves measuring global horizontal illuminance data simultaneously with digital sky photography.

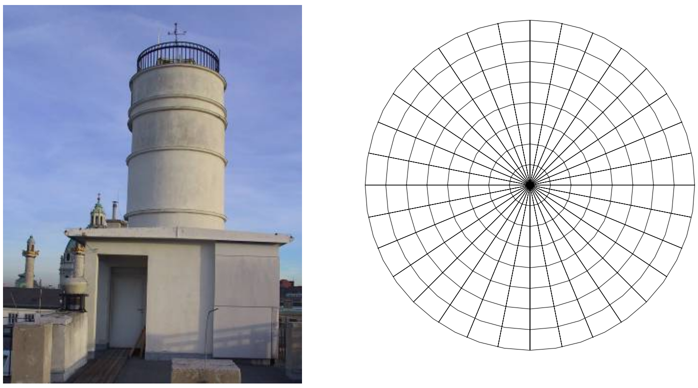

Figure 1. Weather station and the pattern for sky luminance mapping.

The external illuminance data is obtained from the building's weather station. For each image, the initial estimate of the illuminance resulting from all sky patches on a horizontal surface can be compared to the measured illuminance. The digitally-derived sky patch luminances can be corrected to account for the difference between measured and digitally estimated horizontal illuminance levels [2].

Location Sensing

In previous studies, we showed how a simulation-based lighting systems control [3] [4] may be supported by provision of dynamically updated room models that are generated based on the vision-based location-sensing system VIOLAS [5]. Thereby, it was shown how a set of networked pan-tilt cameras in a facility can regularly provide updated models of room geometry and objects within the buildings that can be used for facility management and building control applications. VIOLAS involves five major blocks, whereby each block has a distinct role in context data extraction (Figure 2).

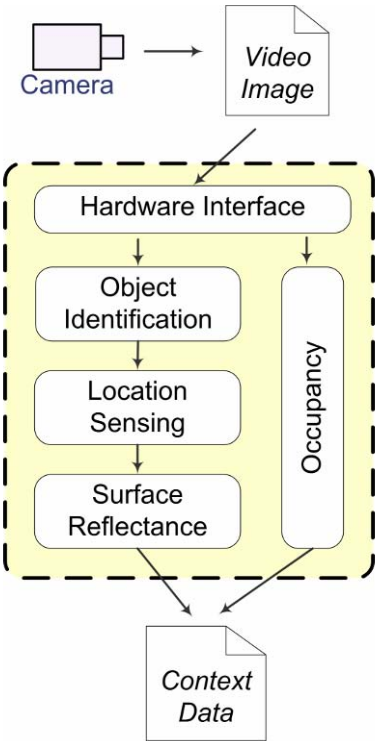

Figure 2. "Sensing core" in VIOLAS.

The Hardware Interface block permits multi sensor data acquisition. In our first attempt, due to the possibilities introduced by the usage of integrated computing, netcams equipped with pan/tilt units were used. However, netcams also generate relatively low-quality data due to data compression resulting in low resolution and blurred images without sharp details. Moreover, pan-tilt netcam solutions are rather bulky, expensive, and involve moving elements. Due to these circumstances, we proposed to study the possibility of improving the VIOLAS system by means of digital cameras with fisheye lenses. Already employed within VIOLAS, the location sensing module is adapted from TRIP system (cp. [6] and Figure 3). A circle on the target plane generates an ellipse on the image plane of the camera. From the known parameters of the ellipse, it can be back-projected to the original circle enabling the extraction of the orientation and the position of the target plane with respect to the camera origin [7]. In order to implement this algorithm, barcode-like tags with circular marks are used. Any regular black-and-white printers can be used to reproduce these tags enabling low-cost and low-maintenance tags without the necessity of power input. Identification is achieved by the codes printed around the circular mark (reference circle). Contrasting with the TRIP system that uses ternary coding, VIOLAS uses binary coding, where the tags are divided into

16 equal sectors (Figure 4) resembling pie slices. The presence or absence of the black mark on the sector denotes the 1 or 0 coding respectively. The pattern of "0111" code sequence defines the start bit sequence, and is not repeated in the code string to avoid ambiguity.

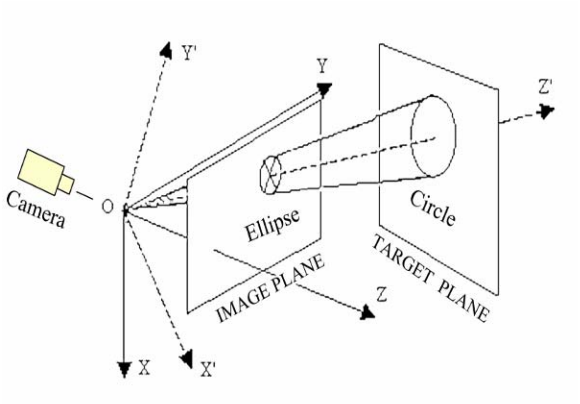

Figure 3. "Pose from circle" algorithm. (*X*,*Y*,*Z*) denotes the coordinate system of the image plane whereas (*X'*,*Y'*,*Z'*) denotes the coordinate system of the target plane. The outcomes of the algorithm are the parameters of the transformation between two coordinate systems.

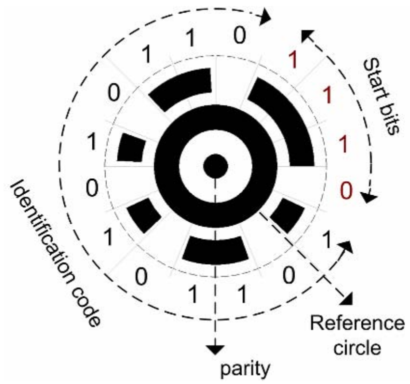

Figure 4. Tag structure is illustrated with a sample tag coded with 0111-011010101101 data string (even-parity = 1). Identification number corresponds to 1709 in decimals.

With the remaining 12 sectors we are able to encode 2031 distinct tagged objects. The TRIP system divides the location sensing procedure into two phases. In the first phase, "target recognition", where the tags are detected, parameters of the reference ellipse (projection of the reference circle on the camera image) are extracted and the identification numbers are decoded. In the second phase, "pose extraction", the locations of the tags are computed from the outputs of the first phase [6]. VIOLAS improves the original TRIP method by incorporating two additional algorithms (Figure 5). An "adaptive sharpening algorithm" [8] on the input image prior to "target recognition" (Figure 5) is applied, resulting in an enhanced output image. As the pixel resolution of the tag images decreases with increasing distance between the tags and the camera, code identification becomes difficult, despite the fact that the tags are being detected and reference ellipses are being correctly extracted. In order to resolve this problem, "edge-adaptive zooming" [9] is applied locally to spurious tags from which the code could not be deciphered or validated. Edge-adaptive zooming, rather than its equivalents such as bilinear and cubic interpolation, enhances the discontinuities and sharp luminance variations in the tag images. This procedure is recurring until the "target recognition" is successful or the zoomed image region loses its details (Figure 6). In the latter case, a false alarm or an unidentified tag will be retrieved.

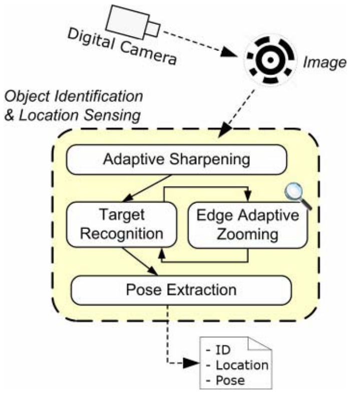

Figure 5. Algorithm flow of object identification and location sensing in the "sensing core".

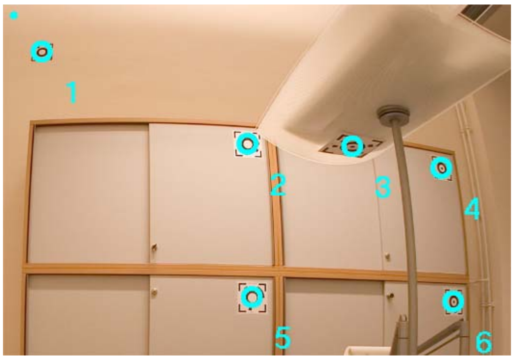

Figure 6. Example of a successful tag localization.

Model-Based Control

To demonstrate a model-based building systems control strategy, we consider the case of lighting and shading systems control in an office space (see Figure 7) [10]. This office's two windows are equipped with automatically controllable blinds. Artificial illumination is provided by two free-standing luminaires. The room model entails information about room geometry, furniture, the location and size of windows as well as the physical properties of room components (such as reflectance and transmittance) as well as the position of virtual sensors that monitor pertinent performance parameters (such as illuminance levels or glare indices). The room is equipped with the aforementioned location-sensing system, which automatically tracks changes in the position of moveable furniture elements (including the aforementioned luminaires). Presence of the people in the room (at the workstations) is monitored with occupancy sensors. User preferences (e.g. desirable illuminance levels, relative weights for objective function) can be communicated to the lighting control application via occupants' and facility manager's computers. The room model provides the basis for the system's internal representation and is updated dynamically using an optically-based location-sensing system.

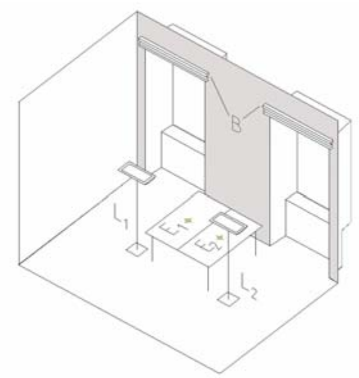

Figure 7. Illustration of the test space (L_1, L_2: Luminaires; B: Blind; E_1 and E_2: virtual illuminance sensors).

Control Devices and Control State Space

Three control devices are considered, namely the two luminaires (L_1, L_2) and the window blinds (B). In the control scenarios considered in this paper, the two blinds are controlled simultaneously. As a primary indicator of lighting performance, we considered the two mean illuminance levels E_{m1} and E_{m2}. Since all these devices affect both E_{m1} and E_{m2}, a central control instance (C) is required to coordinate the three devices toward the most preferable control state. The overall behaviour of the control system is determined through a utility function (UF). The overall objective of the control process is to maximize UF, which, in our test runs, pertains to preferences for illuminance levels, cooling load, and electrical energy consumption. The following scenario involving a control cycle is repeated regularly (in this case, every 15 minutes). At every time step $t_{i,}$ each device (i.e., L_1, L_2, B) submits to the controller application C a list of candidate device states for time step t_{i+1}. In the present case, each device submits four alternative options. These options are: the device's current position, the 2 adjacent positions, and a fourth – randomly chosen – option from the rest of the device's control state space. Thus, the control application considers the resulting overall option space involving a maximum of 64 distinctive control states. To illustrate the working of the above-described control method, we documented the operation of the system in the course of fifteen days (fourteen days in May and one day in June 2005). In this case the control systems reassessment of the desirable control state occurs regularly every 15 minutes. The following figures illustrate this operation in terms of system recommendations and its performance. Given this paper's space limitation, figures 10 to 12 exemplify data only for one day, namely 14th May 2005, in terms of: measured external global horizontal illuminance and control system's predictions of the illuminance levels E_{m1} and E_{m2} as a result of the control system's recommended shading and luminaire states (Figure 8); recommendations of the control system for dimming positions of the two luminaires and the blind position (Figure 9).

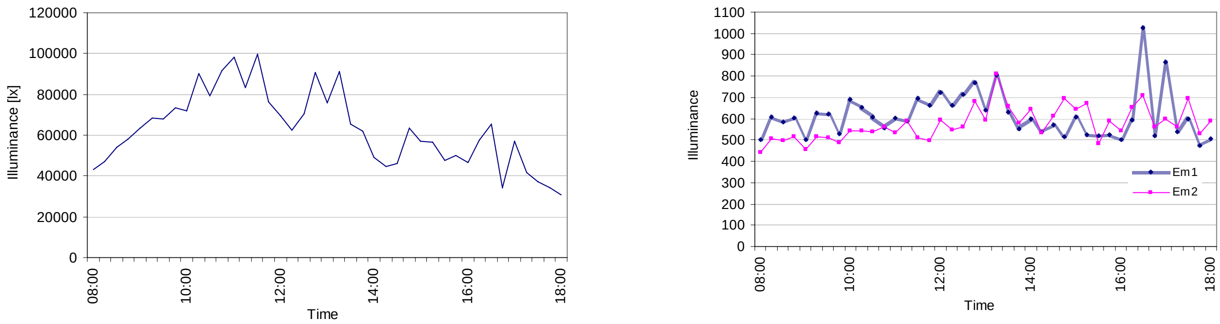

Figure 8. Prevailing external global horizontal illuminance levels in the course of one day and task illuminance levels (E_{m1} and E_{m2}) as the result of system's operation.

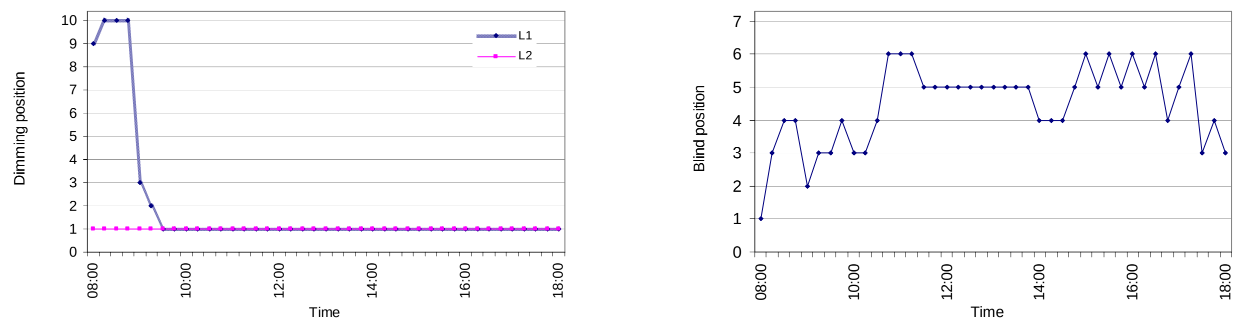

Figure 9. System's recommendations for the luminaires' dimming states and the blind positions.

Development of the Test Bed

The test bed is being started "from scratch" as a full-scale mock-up of two office rooms in a test booth located in the building physics laboratory at TU Wien. The current objectives of the test bed are to enable replication of the results of the above mentioned lighting control study and extension of the lighting control model to include a system for heating, ventilation and air-conditioning (HVAC). Current plans for test bed equipment installation in the existing light-weight structure include: *i)* HVAC system; ii) Electrical lighting system; *iii)* Active daylighting system; iv) Enclosure systems; v) Furniture systems; vi) Control systems; and vii) Information systems. An overview of the test bed layout is shown in Figure 10, below.

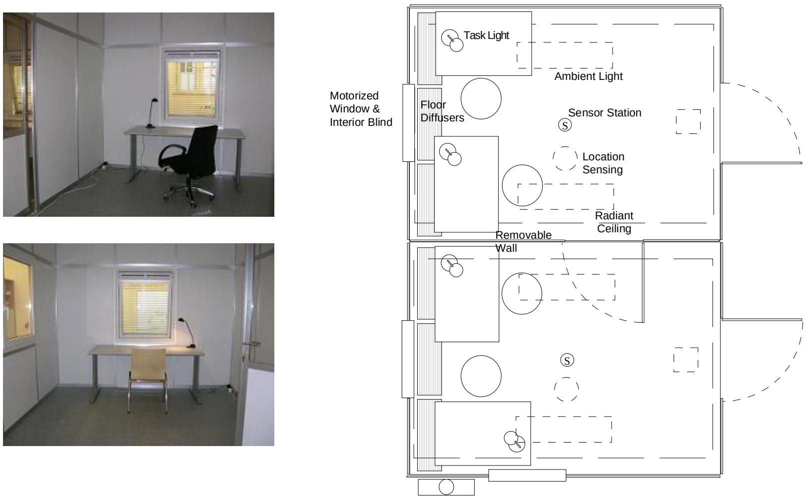

Figure 10. Interior and plan views of the test bed under development.

Simulation-assisted control involves two types of information, those typically transmitted over local area networks, such as data from local servers and weather stations, and data exchanged between the controller and the actual test bed, e.g. for sensor – actuator interaction. Most commercially available building systems for automation communicate on a LON-bus [11]. The test bed uses both bus systems for an authentic demonstration of the model-based control strategy. The test bed can be seen as a continuation of previous experiments in simulation-assisted control. Table 1 shows an overview of control parameters and illustrative control states.

Table 1: Illustrative represententation of the control state space of the test bed.

Control Parameter	Control Variable	Actuator	States (Values)
Natural Ventilation	Window Tilt Position	Window Actuator (% open)	1. 0% 2. 100%
Daylighting	Blind / Louver Position (vertical & horizontal)	Blind Actuator (shading rate, descending)	1. vertical & fully down 2. horizontal & fully down 3. vertical & 2/3 down 4. horizontal & 2/3 down 5. vertical & 1/3 down 6. horizontal & 1/3 down 7. fully pulled up
Electrical Lighting	Dimming Level of Ceiling Fixtures (power output in Watts)	Lighting Actuator (% dimmed, descending)	1. 0% ; 2. 20%; … 10. 100%
Room Air Temperature (mechanical heating)	Supply Air Temperature (dry-bulb temperature, °C)	Throttle at supply air terminal (% closed)	1. 10%; 2. 40%; 3. 70%; 4. 100%
Room Supply Air Volume (mechanical ventilation)	Fan speed (power output)	Fan actuator (% running)	1. 0%; 2. 50%; 3. 100%

CONCLUSIONS

We described a test bed for the prototypical realization of an integrated (multi-system) model-based building control strategy. A comprehensive self-actualizing building model is at the core of the implementation. It encompasses information regarding building context, components and processes. Regularly updated, it provides a real-time source of information for facility management and building systems control operations. Ongoing research is expected to expand the already developed simulation-based lighting and shading control systems to cover further environmental systems for heating, cooling and ventilation. Moreover, innovative user-interface systems are being tested to facilitate an intuitive interaction modus between occupants and the building's environmental control systems.

ACKNOWLEDGEMENT

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Summer season temperature control in Finnish apartment buildings

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SUMMARY

In modern apartment buildings with good insulation and heat recovery, often having large windows and high internal heat gains, overheating may be a serious problem not only in summer season, but also during heating season. This study determined the design curves for the assessment of room temperatures and cooling demand in apartments. The design curves are based on many simulations and they show the effects of window airing, solar protection glasses, window size and orientation, cooled supply air and room conditioning. Determined graphs allow to check quickly the thermal comfort in apartments and this will be a step forward from today's practice not to do such check. They suit well for preliminary design and are useful for architects and HVAC-consultants not using indoor climate and energy simulation software.

INTRODUCTION

Modern apartment buildings are well insulated, heat recovery is used in ventilation and they often have large windows and high internal heat gains especially in well-equipped apartments. These factors may cause overheating not only in summer season, but also during heating season. In the building process, it is common that the architect decides quite independently important choices as regards temperature control. Such issues are the shape and solar protection of building, window size, orientation and glass properties and the selection of structures. After that, the task of HVAC-consultant is to design heating and ventilation. This means in practice the sizing of heating and ventilation system according to architect's choices, not the indoor climate design.

Occupant questionnaires have shown that the building process described gives good results for heating season, but in the summer, apartments are often overheated. This problem is seldom noticed when apartments are for sale, but usually afterwards, when it is not easy to fix the problem. Temperature control has also a concern of public health, which will be stressed with increasing frequency of heat waves. In the summer excess mortalities caused by high temperatures are between 100–200 cases per year [1] in Finland.

Questionnaire survey conducted as a part of a wide European HOPE project [2] studied thermal comfort in apartments in summer and winter. Thermal responses in 7-point scale are shown in Figure 1. The responses of 3, 4 and 5 show acceptable conditions. Good thermal comfort was reported by 73% of occupants in the winter and by 60% in the summer. In the summer about 40% of occupants report that temperature is too high and even in the winter 11% reported too high temperature. There was high variation in the responses depending on the orientation of the apartment and presence of balcony glazing.

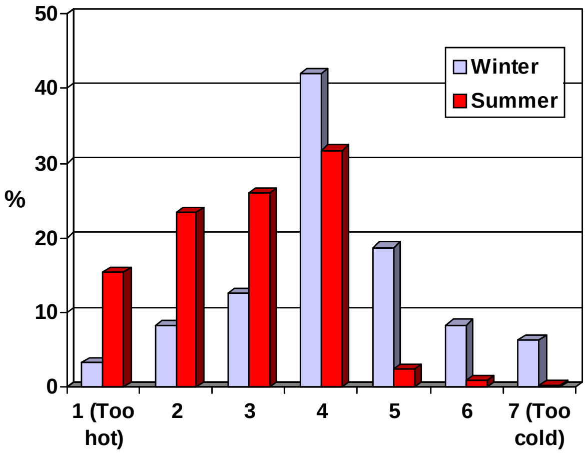


Figure 1. Thermal comfort reported by occupants in 7-point scale (n=600) in winter and summer.

The main problem in the design of apartment buildings is that the temperature simulations are usually not conducted for typical flats. However, the results of the simulations should be taken into account in the choices of architect and in the design and sizing of ventilation and cooling. In this study we tried to overcome this problem by conducting a large number of simulations for typical cases and presenting the results in the form of design curves which can be used for the design of thermal comfort in all typical apartment buildings. These design curves give a simple design tool for architects and HVAC-consultants.

METHODS

We simulated two apartments from relatively new apartment building, construction year 2001, representing typical construction solutions and size of apartments. The building consisted of 10 apartments with two bedrooms (70,5 m^2) and 33 apartments with one bedroom (60,1 m^2). On apartment of both types was modeled with IDA-ICE indoor climate and energy simulation software, Figure 2 and 3. All apartments were equipped with mechanical supply and exhaust ventilation with heat recovery. Total design supply airflow rate was 50 L/s and exhaust 60 L/s in all of apartments.

Figure 2. A photo of the simulated apartment building.

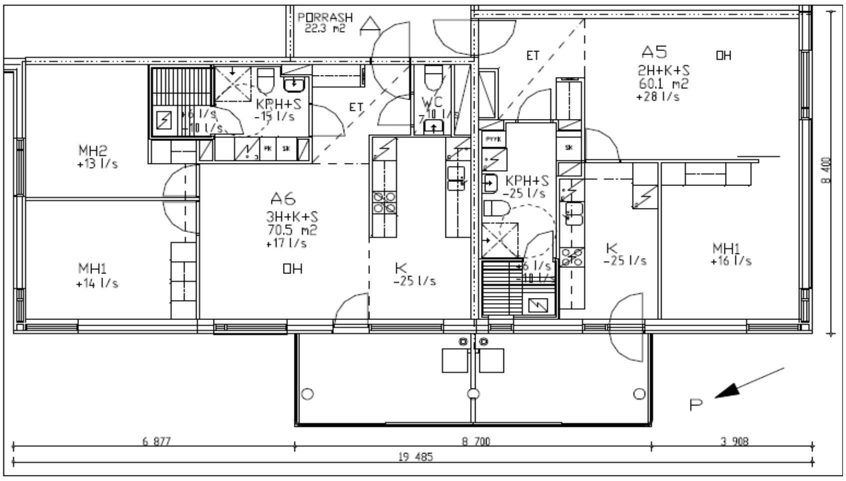
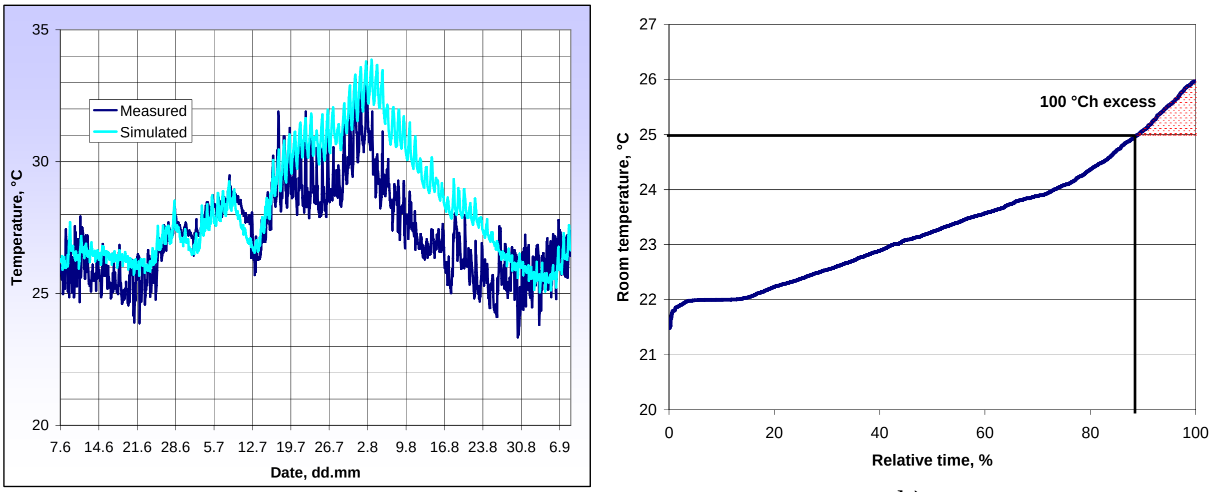


Figure 3. Simulated apartments with 2 and 3 bedrooms.

Summer time temperature measurement data from one flat was used for the verification of the IDA-ICE model, Figure 4, a). This was not the attempt of detailed validation as the occupancy data and the use of window airing was not known. However, such comparison indicates how realistic temperatures are predicted by the model.



a) b)

Figure 4. a) Typical measured temperature from the summer of 2003. The simulated temperature shows some discrepancy as window airing is not taken into account. b) Accepted temperature exceeding for good indoor climate category S2 during three months in the summer.

In the parametric simulations the effect of 22 building and HVAC-factor was studied. Based on these simulations, the most important factors were chosen for sensitivity analyses. Three basic cases were studied:

1. the reference case corresponding to measured apartment building without cooling
2. cooling of the supply air
3. room conditioning unit in the apartment

For these basic cases sensitivity analyses were conducted for the following parameters:

- orientation of the building (four main directions)
- window area equal to the reference case, and increased by 1,5 and 2 times

- with and without window airing
- with solar protection glasses and with common clear panes
- 5 W/m^2 and 10 W/m^2 internal heat gains

The results are presented as simple graphs, from which the effect of each factor can be seen for two defined indoor climate classes. Indoor climate classes were defined because exact key figures are needed for the comparison of design alternatives. For that purpose the target values of Finnish indoor climate classification [3] and the weather data of 2003 was used. This approach follows the principles given in new indoor climate standard prEN 15251 [4] and leads to categories:

- Basic category S3, where room temperature is usually below 27°C
- Good indoor climate category S2, where room temperature is usually below 25°C

For both indoor climate categories the exceeding of the limit temperature by 100 °Ch degree hours is accepted during the summer period from June 1st to Aug 31st, Figure 4, b).

RESULTS

Design curves for room temperature and cooling demand

Design curves are intended for the assessment of room temperatures and cooling demand in apartments and can be used by architects and HVAC-consultants. They show the effects of window airing, solar protection glasses, window size and orientation, cooled supply air and room conditioning. All design curves are given for the four main directions and as a function of the ratio of window area to floor area.

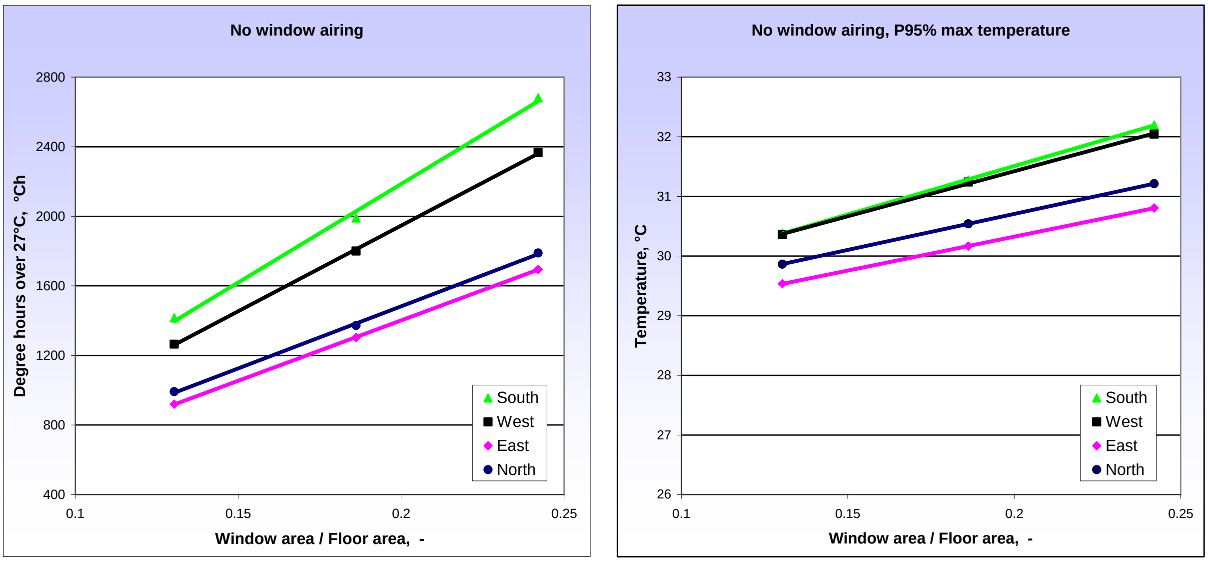
Design curves show, would the indoor climate category achieved in the studied case or what should be changed to achieve the category. Room conditioning curves are for the good indoor climate category S2 and all other curves for the basic category S3. Design curves are not sensitive to the apartment size and may be used for sizing of full range of apartments. However, they are sensitive to internal gains. 5 W/m^2 applies for common apartments (including occupants, lighting and equipment). In well equipped apartments with relatively small floor area the gains may be higher. In that case 5 W/m^2 curves may still be used if the part of gains exceeding 5 W/m^2 will be removed by increasing the airflows in the case of cooled supply air or cooling power in the room conditioning case respectively.

All cases are calculated with solar protection by blinds between the outer panes in three pane window. Possible shading by neighboring buildings, etc. is not taken into account, i.e. all curves apply for fully exposed building. Solar shading may have significant effect especially in apartments on lower floors. This can be roughly estimated, if in the case of clear panes the curves of solar protection glass are used.

In the graphs, the degree-hours exceeding the room temperature of 25 or 27 ºC are shown. For the estimation of maximum room temperature, P95% values of the room temperature frequency distribution curves are given (this is considered as reasonable estimate of maximum room temperature during hot days). Room temperature assessment should be done according to the room showing the highest temperature (= highest window are to south or west). If the room has windows facing to two directions, the window areas may be summed and the higher result can be used, or the result may be interpolated by weighting with window area.

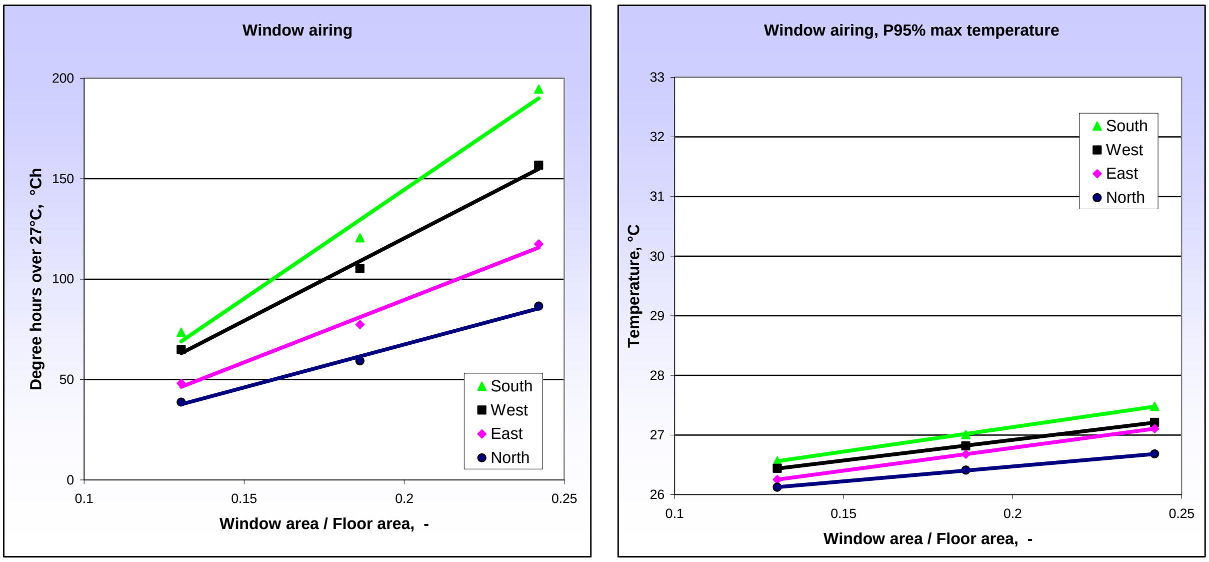
The effect of window airing and solar protection glass (no cooling)

The overheating corresponding to current building practice is shown in Figure 5. The results apply for the building with clear window panes and if window airing is not used. In that case the apartments are hot independently of window area; room temperatures in the hottest days are over 30°C. Solar protection will decrease the temperatures, but this is not sufficient measure as alone, because the rooms are still hot, about 29…30°C. Effective cross ventilation by window airing will reduce room temperatures to acceptable level, Figure 6. With reasonable window areas the basic indoor climate category S3 is achievable. In the case of solar protection glasses, about 1.5 times higher window areas in south and west facades may be used compared to ones in Figure 6.



a) b)

Figure 5. a) The degree hours exceeding the limit temperature of 27°C, and b) the estimate of max room temperature during the hottest days, for the building with clear window panes and without window airing.

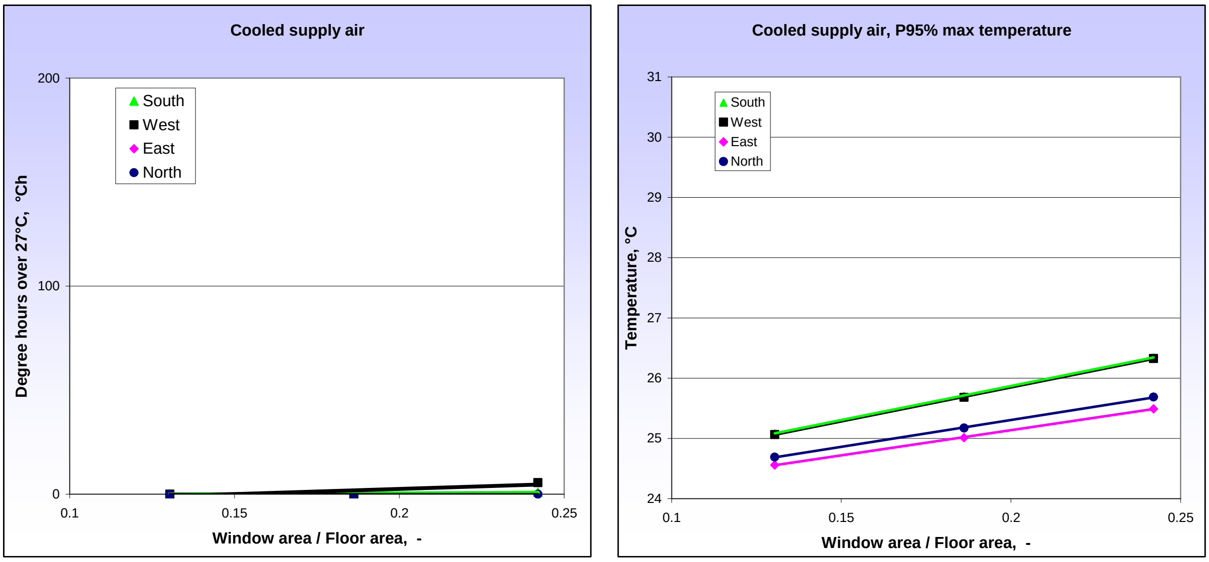


a) b)

Figure 6. The effect of cross ventilation. Clear window panes and open windows (with size of 0.6 m^2) on two facades.

Cooling of supply air

Cooled supply can satisfy a relatively low cooling need of 10…15 W/m^2. It is important to insulate supply air ducts. The following results apply for the supply airflow rate of 0.7 L/s per m^2 and temperature control so that the minimum supply air temperature to the room is 16°C (should be lower from the air handling unit). Supply air temperature is controlled according to exhaust air temperature. Results show that cooled supply air is not sufficient in the case of windows with clear panes. The basic category S3 is achievable, if solar protection glasses are used in south and west facades together with cooled supply air, Figure 7 (window airing is not used).

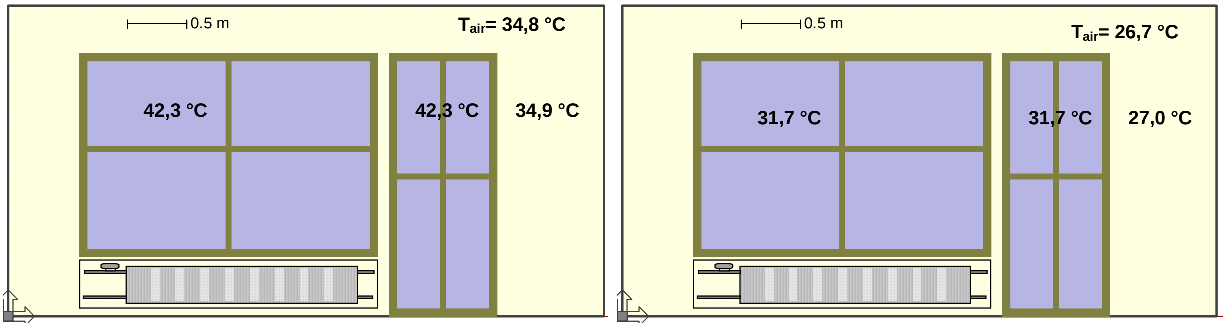


a) b)

Figure 7. Cooled supply air (supply air temperature to the room of 16°C and airflow rate 0.7 0.7 L/s per m^2) and solar protection glasses with solar heat gain coefficient of 0.33.

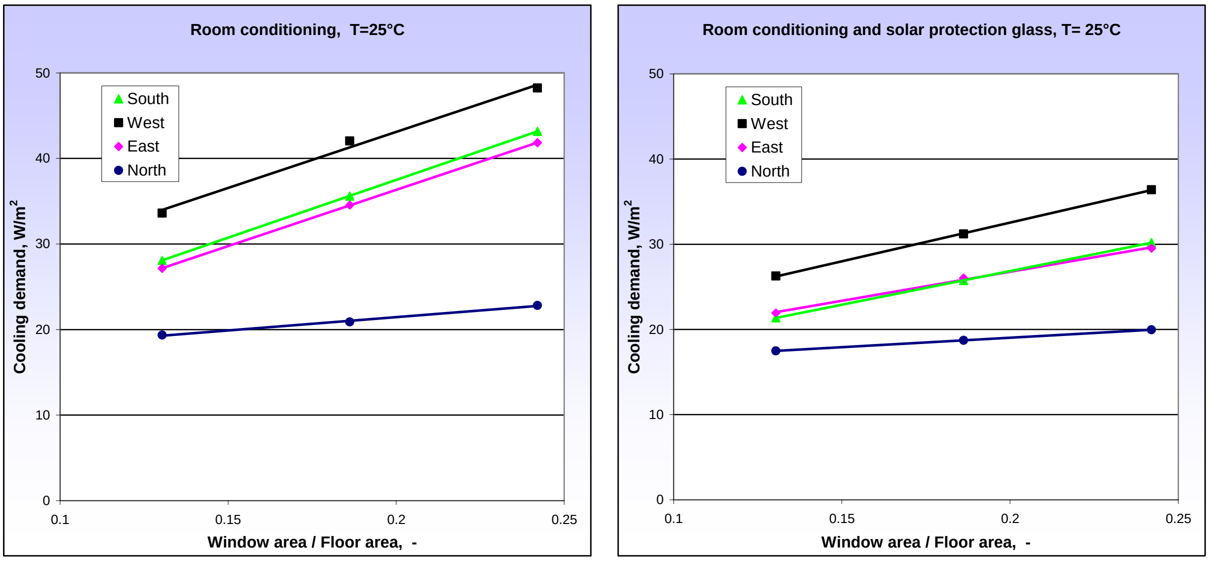
Room conditioning

Room conditioning devices give all possibilities for temperature control. In common apartments, good results are often achieved with one centrally located device (in the living room or in the entrance-hall in open plan apartments) if bedrooms' doors are kept open during the daytime. Due to higher investment cost, better indoor climate category S2 is recommendable to guarantee sufficient temperature control in all cases. For the reduction of cooling demand and the surface temperature of windows, the use of solar protection glasses is recommended on south and west facades. An example of the effect of solar protection glass on the surface temperature of the window is shown in Figure 8. Cooling demands are calculated room by room with the curves given in Figure 9.



a) b)

Figure 8. An example of the effect of solar protection glass on the surface temperature of window. a) 20 W/m^2 cooling and clear window panes, b) 20 W/m^2 cooling and solar protection glasses.



a) b)

Figure 9. Cooling demand for rooms with clear panes a), and with solar protection glasses b).

DISCUSSION

The results show that the current buildings tradition leads to strong overheating in apartments. The basic indoor climate category S3 (room temperature of the 27 ºC with 100 ºCh excess) is achievable with effective cross ventilation and relatively small window size. Cooling of the supply air used together with solar protection glasses is enough for S3 category at internal gains level not exceeding 5 W/m^2. This solution is sensitive to internal gains, and for higher gains level the temperatures were out of the comfort range. For the room conditioning case, the graphs show the cooling load corresponding to the good indoor climate category S2. Determined graphs allow to check quickly the thermal comfort in apartments and this will be a step forward from today's practice not to do such check. They suit well for preliminary design and are useful for architects and HVAC-consultants not using indoor climate and energy simulation software.

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Development of Fault Diagnosis Method for HVAC equipment using Support Vector Machine

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SUMMARY

Fault diagnosis method for HVAC equipment was developed with Support Vector Machine (SVM) and BMS data. By this, the early detection and correspondence of unusual operating became possible. It aims for the reduction in cost and time to spend on fault diagnosis. Normal data and fault data were generated by the physical model of AHU as a training data for SVM. Trained SVM is validated with data obtained from a real building after introducing artificial faults. Results of the diagnosis of actual measurement data of unusual operating, it was apparent that it had some diagnosis ability.

INTRODUCTION

In late years environmental problems such as global warming have become international issues, and measures are needed for reduction of greenhouse gas. In the field of building engineering, particularly effective measures to reduce environmental load are expected. From this, to secure good building as social overhead capital, aiming at the sustainable building that attached great importance to the life cycle management is important issues. On realizing such a social demand, the role that HVAC equipment serves as is big, and maintenance and management of HVAC equipment are important.

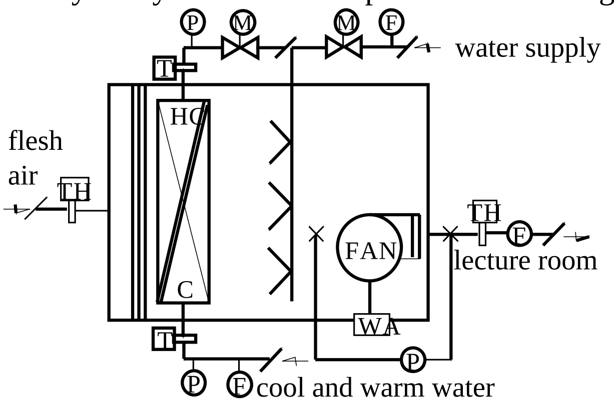
Studies such as measures or maintenance to fault and deterioration of HVAC equipment were made to some extent. However, as for these studies, the history is still superficial, and it is the present conditions that leave some problems[1)]. For example, about fault diagnosis method, it was developed the diagnosis method how provided data by a simulation model made based on knowledge and actual measurement data were analyzed statistically. However, it is necessary for a database in various states of HVAC equipment to be prepared in this method beforehand, and, with that in mind, making of the most suitable parameter to perform fault diagnosis is demanded[2)]. In this case it is difficult to perform fault diagnosis from directly after the completion of a building because the operating situation of HVAC equipment must be measured till data reliable are provided.

A purpose of this study is to establish algorithm of Fault diagnosis for HVAC equipment which can be used from directly after the completion. Since the method enables early diagnosis and early correspondence of deterioration or fault of HVAC equipment, it will prevent waste of costs such as time and expense to spend on fault diagnosis.

METHODS

The Equipment of Study

In this study, an air handling unit (AHU) was the equipment of study. The AHU was installed in Waseda Research Park Communication center (WRPC), which is educational facility located in Honjo City Saitama in Japan.
Sensors were installed to record physical conditions around the AHU. Figure 1 shows placement of the sensors. AHU includes a fan and a coil which are characteristic facilities of the air-conditioning system. In addition, it is used for various buildings. For these reasons, it is very likely that the developed method has high versatility.



P:Pressure
F:Flow quantity
W:Electricity
A:Electric current
M:Control of a valve
T:Temperature
TH:Temperature and Humidity

Figure 1. Sensor layout drawing.

Support Vector Machine[3)4)]

Support vector machines (SVM) are a set of related supervised learning methods used for classification and regression. They belong to a family of generalized linear classifiers. Equation (1) is an identification function of linear threshold node. Linear threshold node calculates two output values for an input vector.

$$y = sign(\omega^T X - h) \tag{1}$$

where y is the mean value of output vector and ω is the parameter for synapse load, X mean value of input vector, h is the threshold value.
Function sign (u) is a symbolic function that takes 1 when u >0 and takes -1 when u < 0. When think about this geometrically, this operation is equivalent to an identification plane dividing input space into 2. A training sample that is the nearest the classification plane which is called a support vector, and the distance up to this support vector is called a margin. A special property of SVM is that they simultaneously minimize the empirical classification error and maximize the geometric margin. Hence it is also known as the maximum margin classifier.
As a result, SVM can show high classification performance for unlearned data.

Study Process

A flow of the study is as follows.
1) Modeling of AHU.
   The modeling of AHU was performed in dynamic simulation program HVACSIM + (J). Each parameter of AHU was set from a maker book or actual measurement data.

2) Generation of normal and fault data by a model of the AHU.
   Four groups of dataset are generated.
   a) Dataset of normal operating state for training SVM.
   b) Dataset in unusual condition for training SVM.
   c) Dataset of normal operating state for SVM.
   d) Dataset in unusual condition for SVM.
   Parameters were modified to represent unusual conditions.
3) Training of the SVM.
   Train SVM by dataset a) and b).
4) A classification of simulation results and inspection of the effectiveness.
   Dataset c) and d) were classified by trained SVM to confirm diagnosis performance.
5) A classification of actual measurement data and inspection of the effectiveness.
   BMS data got by actual measurement were classified by trained SVM to confirm diagnosis performance.

Data Acquisition

Table 1 shows an inspection system and generating methods of the unusual state. An artificial unusual state occurred in AHU and data at the time of unusual operating were measured. After that inspection of the effectiveness of this developed method was performed. The inspection data were BMS data collected with a sensor installed in WRPC.

Data are obtained from a real building after introducing artificial faults.
It shows that are procedures and results of actual measurement as follows. A purpose of actual measurement was collected as data to inspect developed methods.

Table 1. Inspection system and generating methods of unusual state.

Condition	Method
Filter blockage	Filter part blockage
Coil blockage	Coil part blockage
Unusual air supply temperature	Squeeze a valve of cool and warm water
Unusual Fan belt tension	Tension adjustment of fan belt

Table 2. An inspection condition and an actual measurement schedule.

Condition	Actual Measurement Schedules
Supply air temperature is too high or low	06/08/21 09:00~15:00
Fan belt is too tight or slacken	06/08/21 15:00~17:00
Filter blockage (half-open)	06/08/22 09:00~15:00
Coil blockage (half-open)	06/08/22 09:00~15:00
Filter blockage (full-open)	06/08/23 09:00~15:00
Coil blockage (full-open)	06/08/23 09:00~15:00

This is an actual measurement procedure and occurrence methods of unusual states as follows.
1) AHU filter blockage
   A board of the area obstructive to 1/3, 1/2, 2/3 was established on a filter part to cause blocking of a filter.
2) AHU coil blockage
   A board obstructive same as A on a filter part was established to disturb heat exchange.
3) Unusual air supply temperature

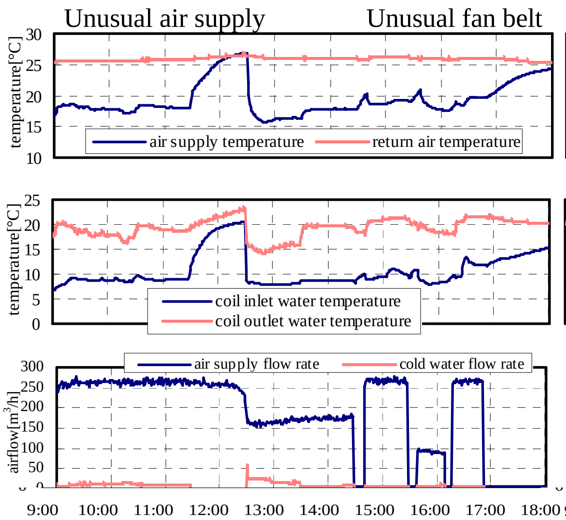
A valve of cool and warm water sent to AHU was squeezed to half-open, all shut, and predetermined air supply temperature was not provided. 1), 2), 3) interlaced, and actual measurement of filter confinement and coil confinement in a half-open state of a valve was performed.

4) Unusual AHU fan belt

Fan belt tension of the AHU inside was regulated.

Actual Measurement Result

Results about data item to use for modeling in HVACSIM+(J) and input to SVM was rounded up along the actual measurement schedule. There are the times when a value is 0 and the times when a value goes up and down momentarily in figure 3-5, that is because operating of AHU which stopped at one time.



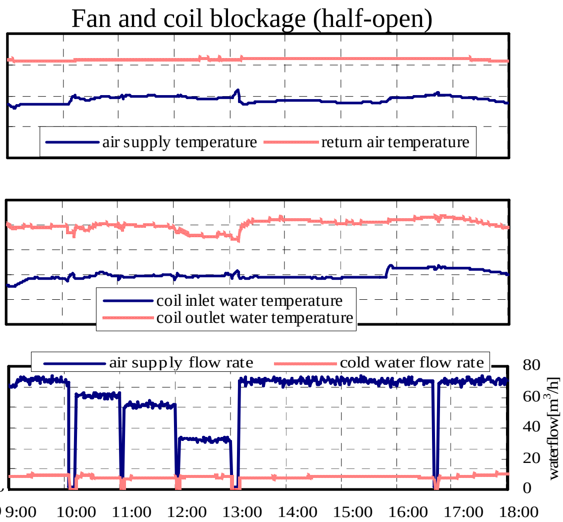


Figure 2. Actual measurement results in 8/21. Figure 3. Actual measurement results in 8/22.

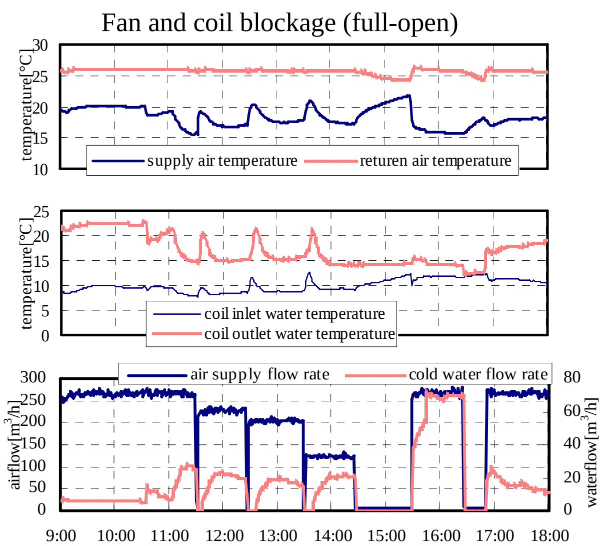


Figure 4. Actual measurement results in 8/23.

Modeling of AHU
HVACSIM+(J)

HVACSIM is a dynamic air-conditioning system simulation program developed in former NBS (National Bureau Standards) of the American Department of Commerce[5)]. The thing which various models were added to in this HVACSIM was HVACSIM+(J) and AHU was modeled with Type602"Cooling or Dehumidifying Coil".

Input and Output Values of a AHU Model

HVACSIM+(J) is a simulation program designed to calculate the output from the input data given based on set parameters. Figure 6 displays the concerning input and output values of a model of AHU. The input values are given as interface variable which every second changes into. In this study, outlet air temperature, outlet air absolute humidity and outlet water temperature were used among the output shown in a figure 6.

Input		Output
Air mass flow Inlet air temperature Inlet air absolute Absolute humidity Water weight flow Inlet water temperature	→	Outlet air temperature Outlet air absolute humidity Outlet water temperature Sensible heat load of coil All heat load of coil

Figure 5. The input and the output of an AHU model

Simulation Result
The result of simulation demonstrated that the coil outlet water temperature and the dynamic outlet dry-bulb air temperature became the values that were near to results data which was displayed in figure 6. It did not agree to the results data about the outlet air absolute humidity.

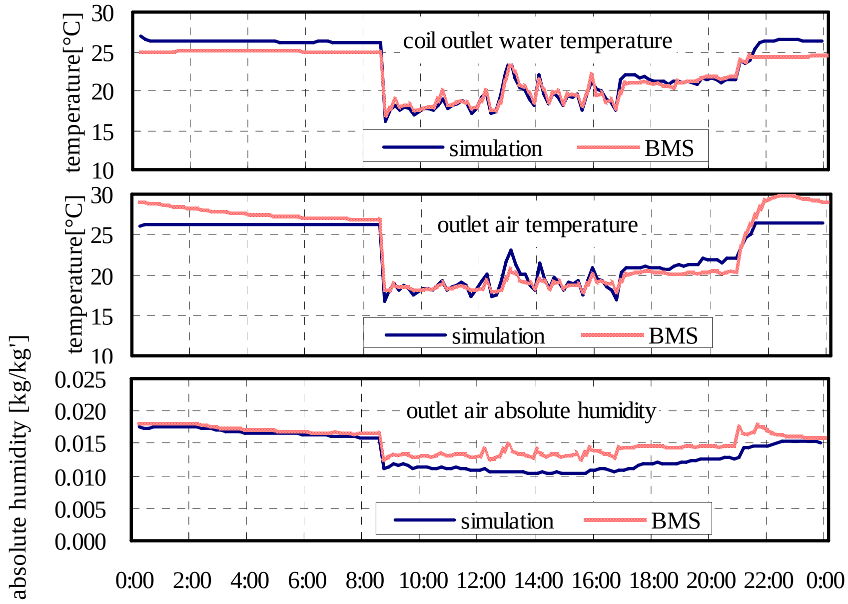


Figure 6. Simulation results

Normal and Fault Data Generation

Normal data and fault data for learning and diagnosis performance confirmation of SVM with a simulation model of AHU were generated.
Data in normal operating state are values of the simulation result that mentioned above. The various kinds of fault were occurred from a change of fouling factor. And the unusual data were generated. Fouling factor is the value to express a heat transmission inhibitory degree of fouling which stuck to heat transmission, and 0 is in condition not to be stained at all. Fouling factor is generally obtained from dividing thickness of a fouling by the thermal conductivity. However, fouling factor used in HVACSIM+(J) is no dimension, and a unit is not determined. Fouling factor is calculated from thickness and thermal conductivity of fouling in HVACSIM+(J), and this is not given as a parameter, and the reason is because HVACSIM+(J) treats fouling factor for numerical value to merely express degree of fouling. Put simply, a user of HVACSIM+(J) can decide a value of fouling factor so that precision of a model improves freely. A user can decreases temperature of coil outlet water and temperature of raise air supply by changing fouling factor.

RESULTS

Learning and Diagnosis of Simulation Data by SVM

Diagnosis of normal operating and unusual operating were performed by SVM based on provided data from the AHU model. Improved computing speed without finding complete diagnosis performance and aimed at diagnosis that had precision in tolerance level. Much time suffers from a calculation when they classify it in search of complete classification performance. In addition, noises to be included in data are not permitted. This is not practical and does not really match a purpose introducing from immediately after the completion.
At first, before a classification of actual measurement data, a classification of normal or fault data generated by simulation was performed. The fault simulation data which fouling factor was changed into and normal simulation data in operating time (9:00~18:00) were given as learning data, and similar data on other days were given as test data.

SVM could classify with probability of 100% in this condition, and classification performance was confirmed.

Adjustment of learning data and selection of input variable

Because the effectiveness of SVM was able to confirm, classification of actual measurement data was performed next. Fouling factor was set to 2.6 or 2.7, and the fault data of learning data were classified. That is because when fouling factor is equal to or less than 3 more than 2, SVM was not able to diagnosis it at all. Therefore fouling factor was a state among two or three, and a classification was performed. When data generated with 2.6 or 2.7 were used as learning data, diagnosis performance was the highest. As for this, a value of fouling factor is not for 2.6 and 2.7 to be most suitable in constant. It was shown that this value was most suitable for actual measurement in this study. When this developed method is introduced, facilities designer, HVAC equipment maker and facilities manager can decide a level of unusual operating and can regulate diagnosis precision by setting the fouling factor.

As the input (feature vector) to give SVM, the input of a model (air flow rate, water flow rate, coil inlet temperature and air inlet temperature) and the output of a model (coil outlet temperature, air outlet temperature) were given. However, SVM which gave all these six input may not get high diagnosis precision for unlearned data. That because it may includes the non effective features for the models. Therefore variable selection was performed by R-square method. As a result, It was shown that model which all variables were given or model removed outlet air temperature comparatively has high performance. It is thought that the reason is because the accuracy of outlet air temperature of a simulation model is worse than coil outlet temperature of a simulation model.

Diagnoses Results of Actual Measurement Data

Figure 7-12 are diagnosis results. For reference, actual measurement data of outlet air temperature appear in the same figure.

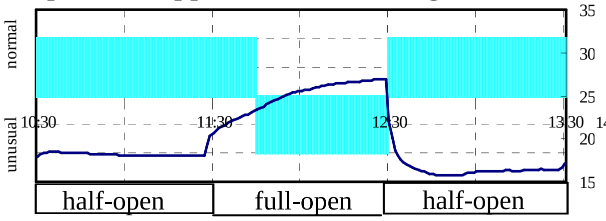


Figure 7. Unusual air supply temperature.

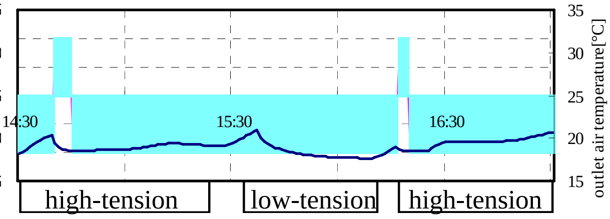


Figure 8.Unusual fan belt tension.

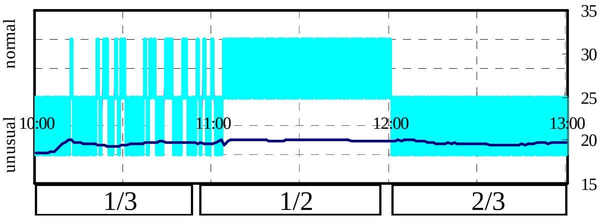


Figure 9. Filter and coil blockage (half-open).

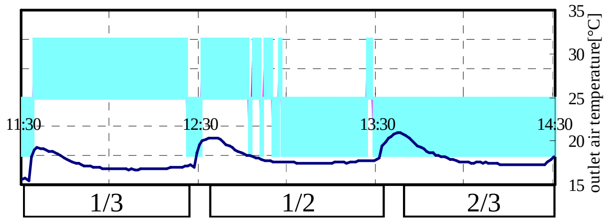


Figure 10. Filter and coil blockage (full-open).

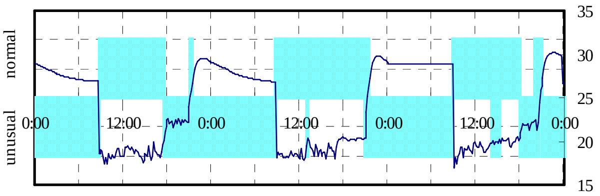


Figure 11. Normal operating state.

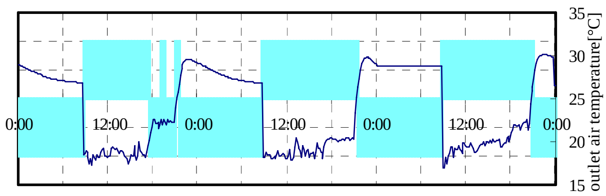


Figure 12. Normal operating state.

DISCUSSION

Normal and fault classification results were replaced each other between actual measurements. A correlation was seen between fault degree and classification.
However, an enough classification has not been possible. Because the each actual measurement time was short. "Filter blockage (full-open)" and "Coil blockage (full-open)" in particular was not able to be diagnosed under steady state. In addition, a diagnosis was difficult because precision of a model was not complete and supply air absolute humidity and return air absolute humidity were not given as the input of SVM.
The situations were observed that a diagnosis result to be fault even when actual measurement of fault operating was not performed. It is because that HVAC operating state is not always normal at the beginning and end of the control and when it was stopped. This tendency was

observed on BMS data of normal operating day with the same learning data as presented in figures 11 and 12.
The diagnosis result every actual measurement item are mentioned as follows.
1) Unusual air supply temperature
As for the diagnosis result at the time of unusual air supply state, it was able to be classified only in the case when of all shut state a valve, but it was not able to be classified in a half-open state a valve.
2) Unusual fan belt
In the case of unusual fan belt SVM was able to classify fault data.
3) Filter blockage (half-open) and Coil blockage (half-open)
Because a change in 2/3 blockage was the biggest, SVM was able to classify fault data. However in the case of 1/3, 1/2 blockage state, SVM did not succeed.
4) Filter blockage (full-open) and Coil blockage (full-open)
Even filter and coil blockage (full-open) became diagnosis performance that was 2/3 confined situations. In the case of 1/3 blockage state, SVM failed classification. In the case of 1/2 blockage state, classification of SVM shows good result, when the state of system becomes almost steady.

As a result of BMS data diagnosis of unusual operating, there was a difference every actual measurement item, but it was confirmed that developed method had some diagnosis ability. However, for improving classification precision, long-term measurements have to be implemented and AHU model representations have to be raised.

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Thermodynamic Modeling and Optimization of Air Handling Units

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SUMMARY

Air handling unit (AHU) is defined as a self-contained unit that the conditions of air vary while passing through it and reach to the desired temperature and humidity. To perform variations in weather conditions various processes such as heating, cooling, humidification, dehumidification and mixing are applied. In this research thermodynamic modeling and mathematical optimization of air handling units approaching minimum energy consumption is achieved. The objective function for optimization is pressure drop of air crossing coil per cooling and heating load of the system. This function comprises all thermal and geometrical parameters of the coils such as coil surface area, number of rows, fin spacing and air side pressure drop of the coil. The objective function is minimized using Lagrange multipliers method. The optimization results are composed of minimum pressure drop, optimum area, optimum number of rows and fin spacing. The effects of varying the cooling and heating load, fin efficiency and the surface area of the coil on fan power consumption are investigated as well.

INTRODUCTION

The major utilities in residential, commercial and industrial buildings are electrical distribution networks, air conditioning systems, steam systems and compressed air systems. The utilities must operate at the highest possible overall efficiency for minimization of the building's energy requirement. Generally, the energy consumption is minimum when the sources such as electrical energy perfectly track the loads. These loads might be cooling or heating loads, which are directly affecting the human comfort conditions. When there is mismatch between the source and load, energy losses will be higher [1]. One of these utilities is air handling unit in which there are a few studies about the energy efficiency about it. An AHU is the primary equipment in an air system of a central hydronic system. It handles and conditions the air and distributes it to various conditioned spaces. AHUs utilize various types of equipments, arranged in specific order, so that space conditions can be maintained. AHUs may consist of a supply fan and coil section with a chilled water or direct expansion coil, preheat or reheat coil, heating coil section, filter section, mixing box, or combination of mixing box and filter [2,3]. In AHUs one of the major components in energy consumption of system are coils. The pressure drop of air passing the system depends on thermal and geometrical parameters of coil. In this research heating coils are modeled by log mean temperature difference and cooling and dehumidifying coils are modeled by log mean enthalpy difference. Since then their optimization proceeds. Attempts show that the investigation of AHUs has been performed considering only one component of the AHU. Robinson [4] investigated experimentally damper control characteristics and mixing box effectiveness of air handling units. The results from tests indicated that the mixing effectiveness of the mixing box was a function of the damper position. Liu et al. [5] developed an air filter pressure loss model for fan energy calculation in air handling units and showed to be highly consistent with available experimental data. Roulet et al. [6] investigated real heat recovery with air handling units. Their research addressed real energy recovery with air handling units from a theoretical point of view and presented results of measurements on 13 units. Sanaye and malekmohammadi [7] developed a new method of thermal and economical optimum design of air conditioning units with vapor compression refrigeration system. The objective function for optimization was total cost per unit cooling load of the system including capital investment for components as well as the required electricity cost.

THERMODYNAMIC MODELING

1. Heating coil modeling

The total heat transfer rate from a heating coil is:

$$q = U_o A_o \Delta T_m , \qquad (1)$$

where U_o is overall heat transfer coefficient, A_o is total outside surface area of the coil and ΔT_m is log mean temperature difference. Considering the tube material to be copper having high thermal conductivity, so the thermal resistance of copper tubes is ignored and the overall heat transfer coefficient based on the outside surface area of the coil U_o can be calculated as

$$U_o = \frac{1}{\frac{A_o}{A_i h_i} + \frac{1-\eta}{h_o\left(A_o/A_F + \eta\right)} + \frac{1}{h_o}} , \qquad (2)$$

where h_i is inner surface heat transfer coefficient, h_o is outer surface heat transfer coefficient, A_i is inner surface area of the coil, A_F is total surface area of fins and η is fin efficiency.
McQuiston [8,9] developed the correlation between Chilton-Colburn j factors and parameter JP for dry coils which is as follows:

$$\begin{aligned} j_s &= \left(\frac{h_o}{G.c_{p,a}}\right)\mathrm{Pr}^{2/3} = 0.00125 + 0.27JP \\ JP &= \mathrm{Re}_D^{-0.4}\left(\frac{A_o}{A_p}\right)^{-0.15} \\ \mathrm{Re}_D &= \frac{G.D_o}{\mu} \end{aligned} , \qquad (3)$$

In equation (3), A_p is total outside surface area of tubes, G is mass velocity, D_o is outside diameter of tube and μ is fluid viscosity.

$$G = \rho . V_{\min} , \qquad (4)$$

where ρ is fluid density and $V_{\min}$ is air velocity at minimum flow area. For hot and chilled water at turbulent flow inside the tubes, the inner surface heat transfer coefficient h_i, can be calculated by Dittus-Boelter [10,11] equation:

$$\frac{h_i D_i}{k_w} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{n} , \qquad (5)$$

In equation (5), D_i is inside diameter of tube and k_w is thermal conductivity of water. When the tube wall temperature $T_t > T_{bulk}$, $n = 0.4$ and when $T_t < T_{bulk}$, n=0.3. Here T_{bulk} indicates the bulky water temperature beyond the boundary layer.

2. Cooling and dehumidifying (wet) coil modeling

In cooling and dehumidifying coil, both heat and mass transfer occurred. With dehumidification, the air side surface is wetted with liquid water. Since water vapor transfer does not depend on temperature difference alone, it follows that the method generally used in the analysis of dry air side surface does not suffice. In this paper, a modeling procedure for wet cooling coils will be presented by adopting the method of Threlkeld [12] which bases the analysis on enthalpy potential. The total heat heat transfer rate for cooling and dehumidifying coil is:

$$q = U_{o,w} A_o \Delta h_m , \qquad (6)$$

where $U_{o,w}$ is overall heat transfer coefficient for wet coil and Δh_m is log mean enthalpy difference. If the thermal resistance of copper tubes is ignored, the overall heat transfer coefficient based on the outside surface area of the wet coil $U_{o,w}$ can be calculated as

$$U_{o,w} = \frac{1}{\dfrac{b_R A_o}{A_i h_i} + \dfrac{b_{w,m}(1-\eta_w)}{h_{o,w}\left(A_o / A_F + \eta_w\right)} + \dfrac{b_{w,m}}{h_{o,w}}} , \qquad (7)$$

In Eq. (7) the combined wet surface coefficient $h_{o,w}$ must be calculated by Eq. (8). In equation (7), η_w is wet fin efficiency. The quantity b is the slope of Eq. (9) which represents saturation enthalpy of air as a function of temperature over a small temperature range.

$$h_{o,w} = \frac{1}{c_{p,a} / (b_{w,m} h_o) + y_w / k_w} , \qquad (8)$$

The quantity b_R should be evaluated at the mean water temperature inside tubes t_R while the quantity $b_{w,m}$ should be evaluated at the mean water film surface temperature.

$$h_{as} = a + b.t_{as} , \qquad (9)$$

where h_{as} is enthalpy of saturated moist air and t_{as} is temperature of saturated moist air.
In Eq. (8) the term y_w / k_w is usually small, so that an estimate of water film thickness is not critical [13]. The convection heat transfer coefficient h_o in Eq. (8) can be calculated by Eq. (3). The inner surface heat transfer coefficient h_i is determined by Eq. (5).

OPTIMIZATION

The objective function for optimization is pressure drop of air passing coil per cooling (heating) load of the system. Air side pressure drop over a finned-tube coil is expressed by [14]

$$\Delta P = \frac{G^2 v_i}{2}\left[\left(1+\sigma^2\right)\left(\frac{v_o}{v_i}-1\right)+f\,\frac{A_o}{A_{\min}}\frac{v_m}{v_i}\right], \qquad (10)$$

where v_i is air specific volume at the inlet of coil, v_o is air specific volume at the outlet of the coil, v_m is mean specific volume and f is friction coefficient. In Eq. (10) the parameters are:

$$\sigma = \frac{A_{\min}}{A_a}, \qquad (11)$$

$$G = \frac{\dot{m}_a}{A_{\min}} = \frac{\rho \dot{V}_a}{\sigma A_a}, \qquad (12)$$

where $\dot{m}_a$ is mass flow rate of air and $\dot{V}_a$ is air volume flow rate.

$$f = 0.124\,\mathrm{Re}^{-0.21}, \qquad (13)$$

The ratio of free flow (or minimum flow) area to face area σ is usually 0.54 to 0.6 [9]. The face area A_a is determined by selecting a proper face velocity of 2.5 to 3 m/s .

$$A_a = \frac{\dot{V}_a}{V_a}, \qquad (14)$$

Substituting Eqs. (11)- (14), in Eq. (10), ΔP can be expressed as:

$$\Delta P = \rho\left(\frac{\dot{V}}{A_{\min}}\right)^2\left[\left(1+\left(\frac{A_{\min}}{A_a}\right)^2\right)\left(\frac{v_o}{v_i}-1\right)+0.124\left(\frac{\rho \dot{V} D_o}{\mu A_a}\right)^{-0.21}.\frac{A_o}{A_{\min}}.\frac{v_m}{v_i}\right], \qquad (15)$$

Except A_o and $A_{\min}$, all other parameters of Eq. (15) are known for a coil. Therefore air side pressure drop of a coil, ΔP can be expressed as

$$\Delta P = f_1\left(A_o, A_{\min}\right), \qquad (16)$$

Total outside surface heat transfer area of heating and cooling and dehumidifying coils can be determined from Eqs. (1) and (6) respectively. Substituting inside and outside surface heat transfer coefficients in Eqs. (2) and (7) and by using Eqs. (1) and (6), the total outside surface heat transfer area can be expressed as

$$A_o = f_2\left(\frac{A_o}{A_i}, \frac{A_o}{A_p}, \frac{A_o}{A_F}\right), \qquad (17)$$

Total outside surface area of a coil as a function of number of rows and face area expressed as

$$A_o = F_s A_a N \,, \qquad (18)$$

In equation (18), F_s is coil core surface area parameter and N is number of rows. Using Table 1, coil surface ratios are defined as the following functions in terms of number of rows and fin spacing.

Table 1 Finned- tube coil construction parameters [9]

Fin Spacing S_F (fins/m)	A_o/A_p	A_o/A_i	A_F/A_o	F_s
315	7.85	7.95	0.873	9.91
394	9.68	9.68	0.896	12.07
472	11.54	11.40	0.913	14.21
551	13.46	13.17	0.925	16.37
591	14.47	14.03	0.928	17.48

$$\begin{aligned}
&\frac{A_o}{A_i} = 0.8694 S_F + 0.9872 \\
&\frac{A_o}{A_p} = 0.9442 S_F + 0.2584 \\
&\frac{A_o}{A_F} = 0.001 S_F^2 - 0.0322 S_F + 1.3399 \\
&F_s = 1.0795 S_F + 1.2698
\end{aligned} \qquad , \qquad (19)$$

Substituting Eqs. (17)- (19) in Eq. (16), the objective function, ΔP is obtained as

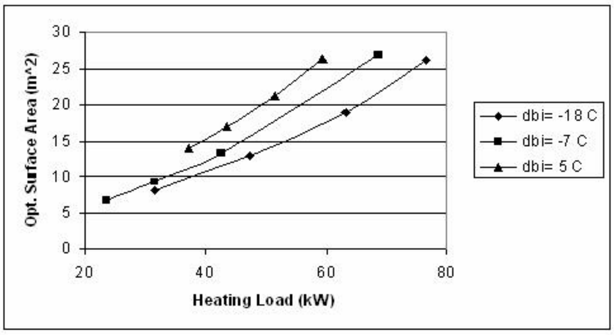
$$\Delta P = f_3(N, S_F)\,, \qquad (20)$$

To find the system optimum design parameters, the objective function (20) is minimized by Lagrange multipliers method [15,16]. The boundary conditions of problem are number of rows and fin spacing. Fin spacing is usually expressed in fins per meter, and it often varies from 315 to 591 fins/m (1.6 to 3.1-mm fin spacing) for coils used in air conditioning systems. The number of tube rows in heating coils varies from 1 to 4 rows and in cooling and dehumidifying coils varies from 4 to 8 rows [9].

RESULTS

1. Heating coil

Figure 1 shows the influence of heating load on optimum surface area and pressure drop of heating coil in 4200 m^3/hr air flow rate. Since the conditions of air entering the AHU can be different, optimum design parameters is determined in three different entering temperatures namely, $-18°C$, $-7°C$ and $5°C$. In the optimization process of heating coil, the entering water temperature to coil is always $82°C$ and leaving water temperature from it is $71°C$. In Fig 1, by increasing the heating load, the optimum surface area of coil increases. If two different inlet temperatures are considered, at a constant heating load, optimum surface area of greater temperature of entering air is higher than optimum surface area of smaller inlet temperature. To justify this result one can say that by increasing the inlet temperature of air, log mean temperature difference decreases and by using Eq. (1) and considering constant heating load, optimum surface area of coil increases. In this Figure, by increasing the heating load, optimum pressure drop of coil increases, since by increasing the heating load, optimum surface area of coil increases and relevant with Eq. (10), pressure drop is proportional to surface area of coil. Also at a constant heating load, optimum pressure drop of greater inlet temperature is higher than the optimum pressure drop of smaller inlet temperature. This result can be justified by considering proportion of pressure drop with surface area.



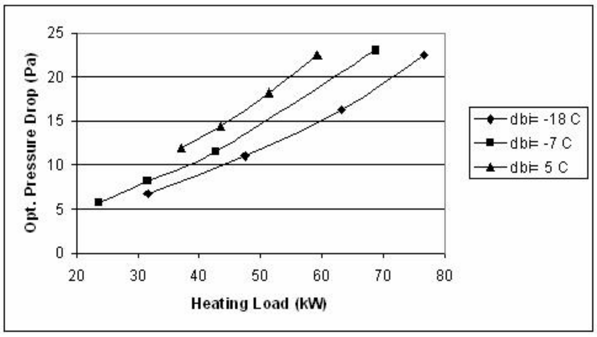
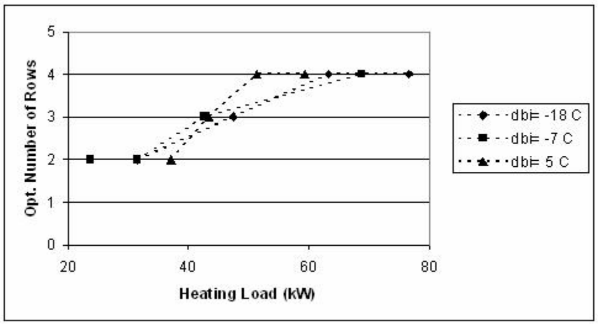


Figure 1 The influence of heating load on optimum surface area and pressure drop



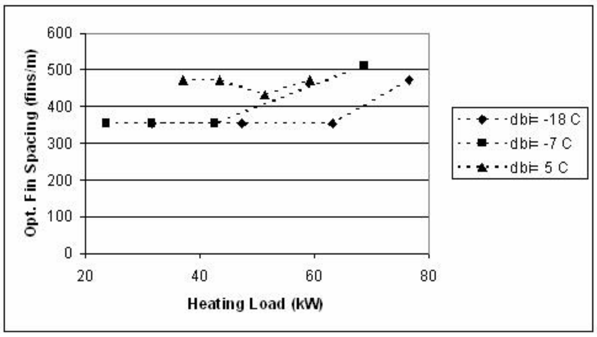


Figure 2 The influence of heating load on optimum number of rows and fin spacing

Figure 2 shows the influence of heating load on optimum number of rows and fin spacing of heating coil in 4200 m^3/hr air flow rate. In this Figure, at the inlet temperature of $-18°C$, it is observed that by increasing the heating load, primarily the number of rows increase and fin spacing remains constant, then the number of rows remain constant and fin spacing increases. At the inlet temperature of $-7°C$, by increasing the heating load, at first number of rows and fin spacing remain constant and then both of them increase. At the inlet temperature of $5°C$, by increasing the heating load, fin spacing decreases since with decreasing fin spacing, the number of rows increases, as a result of this, consequently optimum surface area of coil increases. In Figure 2 in some cases with increasing heating load, the number of rows or fins remains constant, because in that heating range, coil can have further heat transfer without increasing number of rows or fins.

2. Cooling and dehumidifying coil

In Figure 3 the influence of cooling load on optimum surface area and pressure drop of cooling and dehumidifying coil in three different conditions of air is shown. In the optimization process of cooling and dehumidifying coil, the entering water temperature to coil is always $7°C$ and leaving water temperature from it is $13°C$.

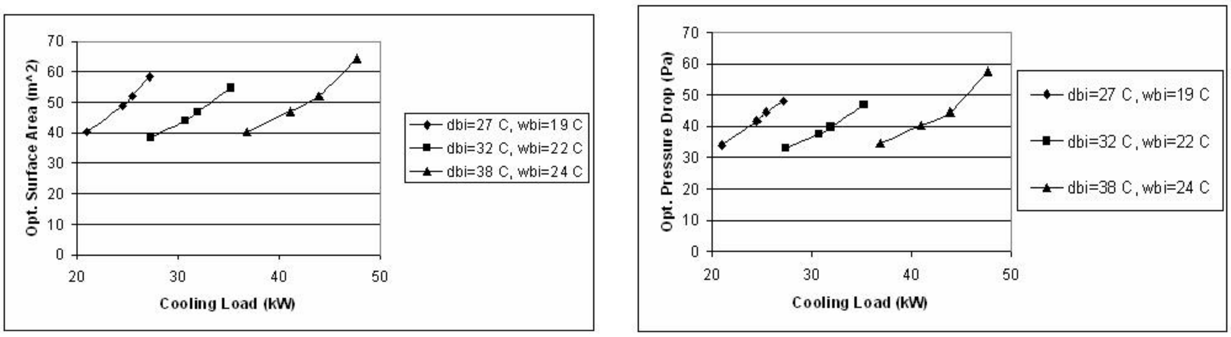


Figure 3 The influence of cooling load on optimum surface area and pressure drop

In Figure 3 by increasing the cooling load, optimum surface area of coil increases. Optimum pressure drop of passing air is proportional to outside surface area of coil and with increasing the cooling load, it increases. If two different conditions of entering air are considered, at a constant cooling load, optimum surface area of greater dry bulb temperature of entering air is smaller than optimum surface temperature of smaller dry bulb temperature of entering air. To justify this result, one can say that by increasing dry bulb temperature of entering air, log mean enthalpy difference increases and by using Eq. (6) and considering constant cooling load, optimum surface area of the coil decreases.

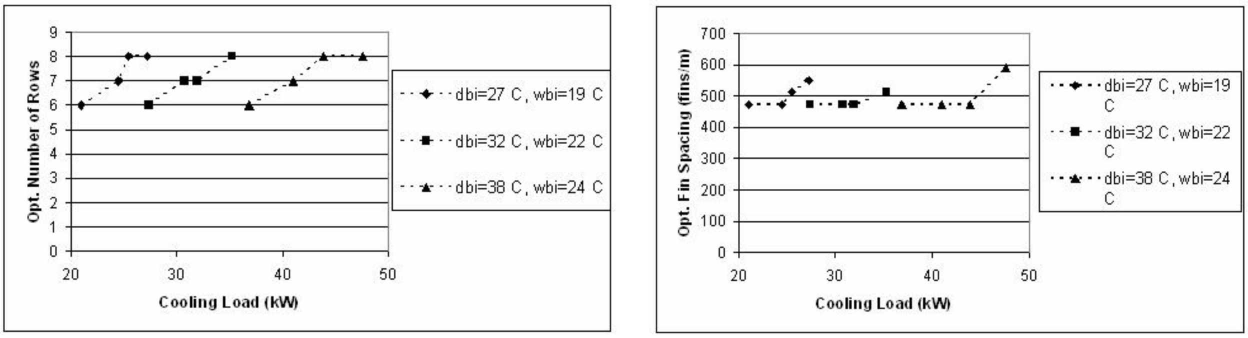


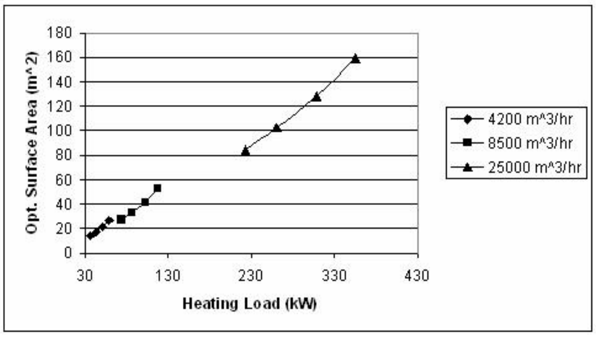
Figure 4 The influence of cooling load on optimum number of rows and fin spacing

In Figure 4 by increasing the cooling load may one of the following cases may occur and at all cases with increasing the cooling load, optimum surface area increases.

a. Number of rows remains constant and fin spacing increases.
b. Both number of rows and fin spacing increase.
c. Number of rows decrease and fin spacing increases.
d. Number of rows increase and fin spacing decreases.
e. Number of rows increase and fin spacing remains constant.

3. Influence of air flow rate on optimum surface area of the coil

Figure 5 show the influence of air flow rate on optimum surface areas of heating coil and wet coil. In both coils by increasing air flow rate, cooling or heating load increases with the subsequent of increase of optimum surface areas of coils.



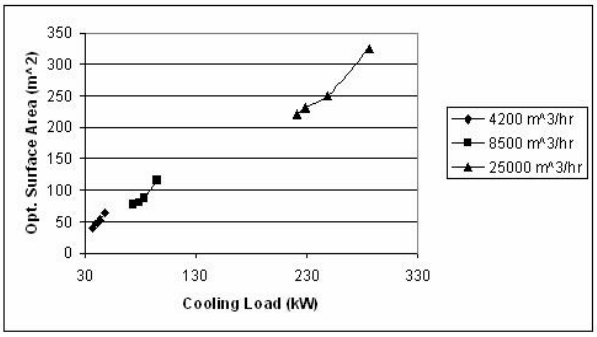


Figure 5 The influence of air flow rate on optimum area of heating and wet coils

4. Influence of fin efficiency on optimum surface area of the coil

In Figure 6 the influence of fin efficiency on optimum surface areas of heating coil and wet coil is shown.

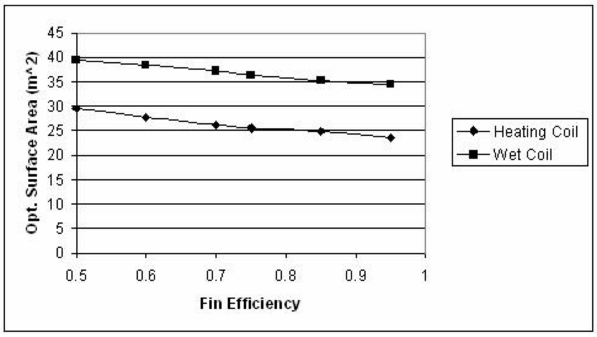


Figure 6 The influence of fin efficiency on optimum areas of heating and wet coils

At a constant heating or cooling load, by increasing fin efficiency, optimum surface areas of heating coil and wet coil decrease. To justify this result, one can say that by increasing fin efficiency, overall heat transfer coefficient increases and at the constant load, by using Eqs. (1) and (6) the optimum surface areas of coils decrease. In another word, by increasing fin efficiency, heating resistance of the coil decreases and it may have the same heat transfer in a smaller heating load.

DISCUSSION

In this paper, a method for thermodynamic modeling and mathematical optimization of air handling units having cooling and heating coils is proposed. The design parameters include thermal and geometrical parameters of cooling and heating coils. On the basis of the previously discussed results the following conclusions have been extracted.

1. The optimum conditions of coils composed of the surface area, pressure drop of passing air, number of rows and fin spacing in different conditions of air and different heating or cooling loads are determined.

2. In heating coils, by increasing the heating load, the optimum surface area of coil increases. In two different inlet temperatures and at a constant heating load, optimum surface area of greater inlet temperature is higher than optimum surface area of smaller inlet temperature.

3. In wet coils, by increasing the cooling load, optimum surface area of coil increases. In two different inlet conditions of air and at a constant cooling load, optimum surface area of greater dry bulb temperature is smaller than optimum surface area of smaller dry bulb temperature.

4. In both heating and wet coils by increasing air flow rate, optimum surface areas of coils increase.

5. At a constant heating or cooling load, by increasing fin efficiency, optimum surface areas of both heating and wet coils decrease.

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Integrated HVAC systems in Central Europe climate

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SUMMARY

Sustainable building heating/cooling design concept in climate conditions of Central Europe aims to minimise building energy performance as well as embodied energy and environmental pollution and at the same time to achieve healthy indoor environment. As a result of building science and practice development in last years we have to solve new issues related to changes in a modern building design like need for summer cooling also in Central Europe conditions, different ratio of internal and external loads or problems with room ventilation resulted from infiltration rate minimising.

As the output of radiant system is limited by acceptable surface temperature, often asked questions are – "Is this system suitable for Central Europe climate? The purpose of this paper is to evaluate indoor environment and system energy performance in case study, focusing on office rooms, to evaluate properly different heating/cooling energy transmission systems like chilled/heated ceilings, floors and walls and their operation control strategy.

INTRODUCTION

Recent low-energy building heating and cooling design concept in climate conditions of Central Europe is based on minimizing energy consumption as one of the critical criterions in building design aiming to decrease total energy consumption and environmental pollution at the same time with preserving or improving indoor air quality.

Sustainable approach to a building and its energy system design, as next step after the low-energy approach, leads to evaluate besides the operational energy also energy embodied in the building and the system material (Kabele et al. 2006).

The typical need for climate conditions of Central Europe (Czech Republic) is space heating during heating period (approx 230 days a year). There were no active cooling systems considered in traditional buildings. Due to low internal gains, high thermal inertia and optimised glazing ratio this building could be operated without any active cooling.

Figure 1. Integrated heating/cooling ceiling system with capillary mats

Current modern buildings, with high internal gains, high galzing ratio, low mass and well-insulated walls (U value less than 0,3 W/m2K) and windows (U value less than 1,8 W/m2K) must be nowadays equiped with an active

cooling system to follow the comfort requirements also in summer period. In many cases, in these buildings the cooling load finally exceeds the heating load.
Traditonal approach to the technical solution of a heating /cooling system in such a buildings was to design two independent systems (radiator heating and split units cooling). One of the modern technical solutions, which could be considered as a sustainable one, is an integrated heating/cooling ceiling system. Capillary mats, embedded in the gypsum plaster layer of the ceiling structure are supplied by the heating/cooling water (four-pipe system) (figure 1). Ceiling surface transmits energy into the heated/cooled room via radiation and convection heat transfer modes. Comparing to a traditional system, this technical solution integrates two systems into one and reduces air change rate need to hygienic minimum. The strong point of this solution is a significant reduction of the used material or rather embodied energy in a heating, cooling and ventilating system (Roulet et al. 1999).
The main problem is in the technical limits of this system (Petráš 2001). During the heating operation, output is limited by a hygiene limit reducing highest intensity of radiation on the skull-cap up to 200 W/m2. During the cooling operation it is limited by surface temperature, which should not drop below the dewpoint temperature (surface condensation).

PROBLEM DESCRIPTION

The main purpose of this paper is to investigate integrated heating/cooling system performance during typical Central Europe climate conditions with office operation load profile. This task arose from common practice, when several problems with system application occured despite following all the common recommendations (Kabele et al. 2002).
The questions to be answered were: Is the integrated ceiling heating/cooling system able to secure compliance with comfort requirements during the whole year operation in modelled case? Are the existing design recommendations in terms of maximum heating/cooling output of the ceiling applicable particularly in climate conditions of Central Europe?

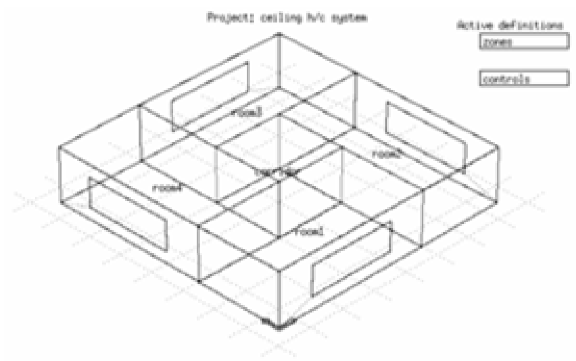

Figure 2 .ESP-r model of the building

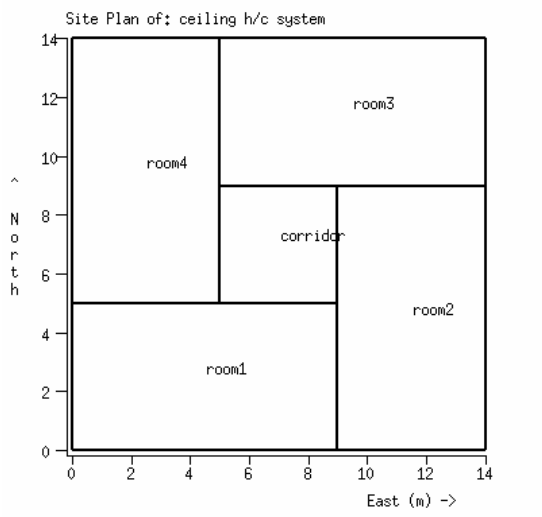

Figure 3. Site plan of the model

RESEARCH METHOD

We used problem analysis followed by computer simulation of an annual building energy performance on a case study to analyze selected parameters, that may have an influence on the possibility of system application. ESP-r, an energy system performance simulation program, was used for this purpose (ESRU 2004).

MODELING AND SIMULATION

Model Description

A five - zone model was created for this purpose (Figure 2, Figure 3). The model contains four equal zones with following dimensions 5 m x 9 m x 3, each facing different cardinal point, and a corridor in the centre with following dimensions 4 m x 4 m x 3m. Each of the zones has a window 5 m x 1.6 m in a longer exterior wall. Medium-heavy constructions were considered with the value of overall coefficient of heat transmission according to Czech building regulations (ČSN 730540 2005). For an external wall U = 0.239 W/m^{2}K, for an internal wall U = 1.561 W/m^{2}K and for a window it is 1.198 W/m^{2}K.

Room dimensions in model follows reccomended values for the proportion of hight to depth of the room from the point of day lighting (h/d = 1.66), typical in the Central Europe geographical conditions. The size of glazing surface represents 30 % of the façade surface.

No heat flux through ceilings and floors was assumed.

Heating and cooling system is radiant low temperature heating/high temperature cooling system with capillary mats placed inside the layer of gypsum plaster in a ceiling construction and defined by heating capacity controlled according to established practice in a range of 0-130 W/m^2, cooling capacity 0-80 W/m^2 in each of the rooms (Jeong et al. 2004). This technical solution is carried out in the model by placing a cooling and heating capacity to the axis of gypsum plaster layer. (R.K. Strand and K.T. Baumgartner 2005) The active ceiling construction contains layers according to Figure 4

The control of the system is running as basic heat and cool controller according to sensors, located in each of the rooms, sensing dry bulb temperature. Set point for heating is 22°C; for cooling 26°C. There is no humidity control considered.

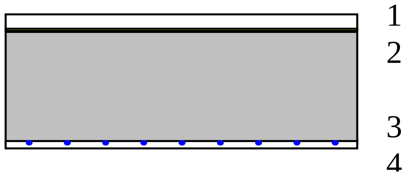

Figure 4. Active Ceiling/Floor construction:
1 Flooring, 2 Polyurethane foam board, 3 Heavy mix concrete, 4 Gypsum plaster with capillary mats

Ventilation. Mechanical ventilation 0,7 h^{-1} besides the infiltration 0,3 h^{-1} is considered. Supply air temperature is 20°C

Occupation and casual gains. Sensitive and latent heat load by persons is 7,8 W/m^2 during working hours (weekdays 7:00-18:00), sensitive heat load by equipment is 15 W/m^2 and lights 25 W/m^2 during whole day.

Simulation

The whole year period was studied using Prague (Czech Republic) ASHRAE IWEC climate files (Figure 5). An integrated building simulation was used, with time step 1 hour and initial period 11 days. Shorter time step has been tested during model tuning, but didn't have any important impact on the results and so the picked one hour step ensured sufficient accuracy. During simulation, no major problems were detected by the program. The discussion about the results was focused on heating/cooling energy consumption. PMV and PPD parameters were used to evaluate thermal comfort (Yang 1997, ČSN EN ISO 7730 2005). The third thing to follow was the possibility of condensation on the ceiling surface during the cooling period.

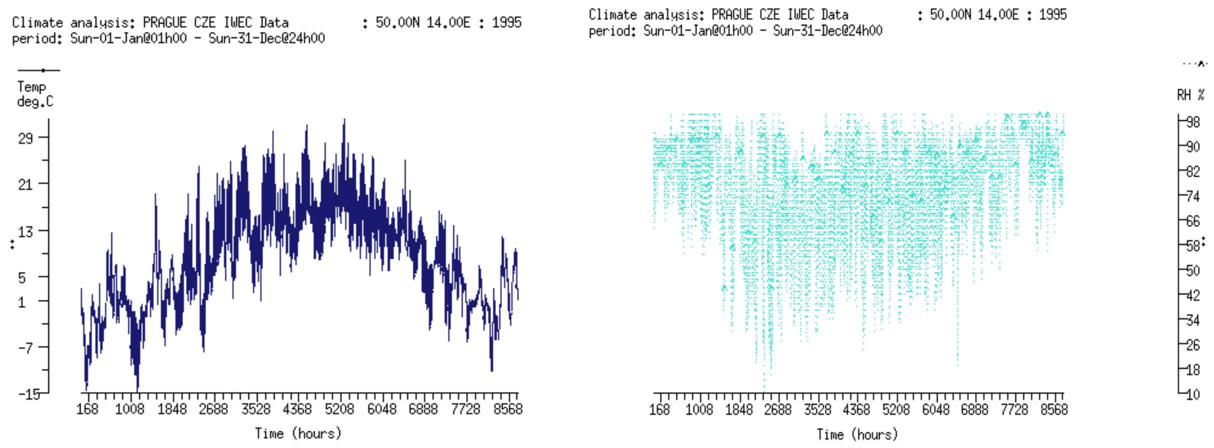

Figure 5 Annual ambient air temperatures(a) and ambient air relative humidity(b) in Czech Republic

SIMULATION RESULTS AND DISCUSSION

In following tables and diagrams there are selected simulation results from ESP-r, related to the problem.

Energy

From the point of annual energy consumption the results show a significant impact of the internal heat loads which decrease energy demand for heating and increase energy demand for cooling comparing to unoccupied space. The effect of an aspect in both heating and cooling energy demand in all of the solved cases is also considerable.

Table 1 Annual energy h/c consumption

Zone	En. for Heating	Operating time	En. for Cooling	Operating time
	[kWh]	[h]	[kWh]	[h]
room1	283	264	-12211	4455
room2	436	392	-11049	4146
room3	253	282	-11415	4247
room4	328	301	-13007	4633
Total	**1301**		**-47682**	

Thermal comfort – resulting temperature

To get to know if the system capacity is well designed we focused on the interior temperatures first

The temperature curve confirmed the ability of the heating/cooling system capacity to guarantee set temperatures within all of the zones during nearly the whole year. Set point for cooling was exceeded only in several hot days during summer with maximum value 31.5°C.

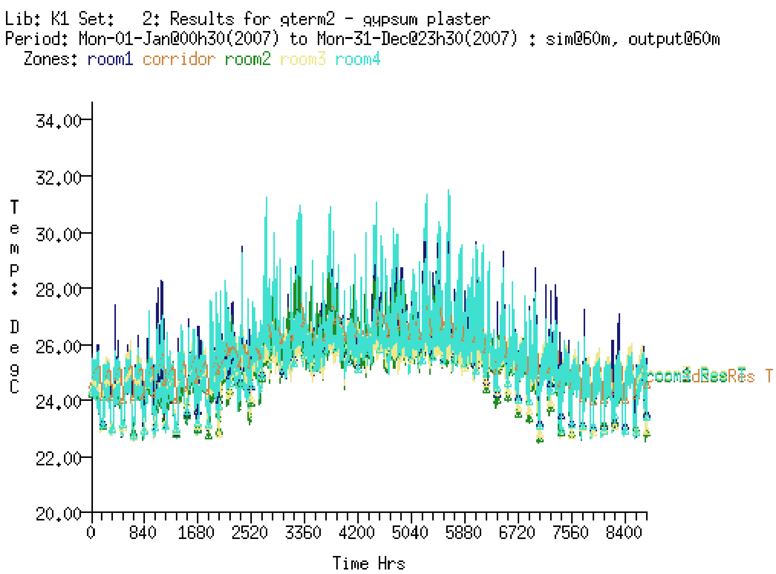

Figure 6 Annual resulting temperatures in all zones

Table 2 Resultant temperature extremes

zone	Maximum [°C]	Minimum [°C]	Mean value [°C]	Standard deviation
room1	29.7	22.5	25.3	1.2079
room2	28.5	22.5	25.1	1.1186
room3	27.8	22.5	25.1	1.0106
room4	31.5	22.5	25.4	1.3316
corridor	27.5	23.9	25.7	0.85953

Thermal comfort – PMV, PPD

Thermal comfort evaluation is based on PMV and PPD classification of heated/cooled spaces. PMV is defined by six thermal variables from indoor-air and human condition that is air temperature, air humidity, air velocity, mean radiant temperature, clothing insulation and human activity. The value of PMV index has range from −3 to +3, which corresponds to human sensation from cold to hot, respectively where the null value of PMV index means neutral to maintain the PMV at level 0 with a tolerance of ±0.5 to ensure a comfortable indoor climate. The PPD index is a description of estimated thermal comfort and a function of four physical parameters: dry bulb temperature, mean radiant temperature, relative humidity and air velocity, and parameters connected to the occupant such as clothing level, metabolic rate and external work. Comfort evaluation is based on activity level 1.2 met with clothing level equal to 0,7 clo (ASHRAE 2005, CSN EN ISO 7730).

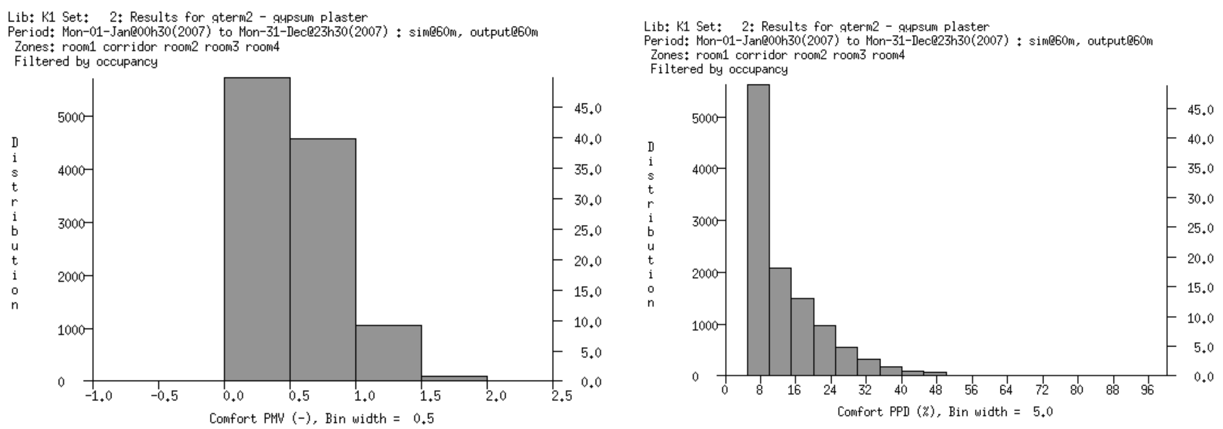

Figure 7 Annual distribution of PMV and PPD index

In this case PMV index appeared from -0.5 to 2.0. Index PPD in 48% of the time is up to 10% which means that during this time the number of dissatisfied occupants will not exceed 10%. Index PPD during 99% of the time is up to 50%.

From the point of heating there is no problem with thermal comfort in all of the examined cases; minimum value for PMV during heating period is -0.5, which means better than slightly cool.

On the other hand several problems with thermal comfort during cooling period were detected. In this case the maximum PMV index reached 2.0, which means warm.

Active ceiling surface temperatures and condensation

The active ceiling surface temperatures coming out from the simulation are in figures. Figure 8 shows temperature difference between active ceiling surface temp and dew point temperature during the year, to identify critical days, marked with circle. Detailed analysis of critical time is on the figure 9.

The possibility of surface condensation occurred in a range of one or two hours during one critical day of the year in summer, when the exterior relative humidity was very high.

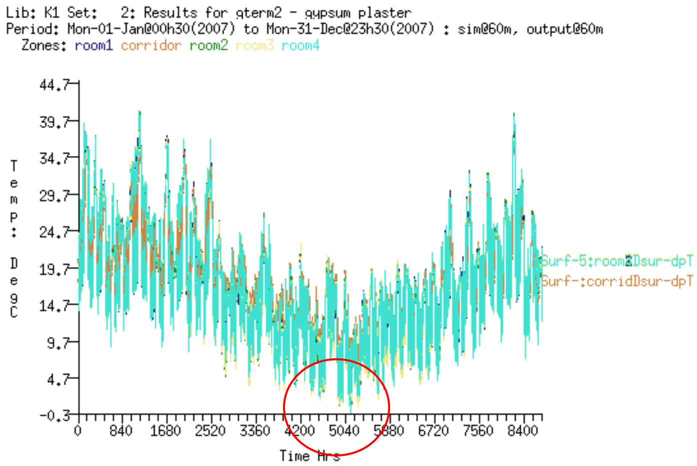

Figure 8 Difference between active ceiling surface temp and dew point temp during the year

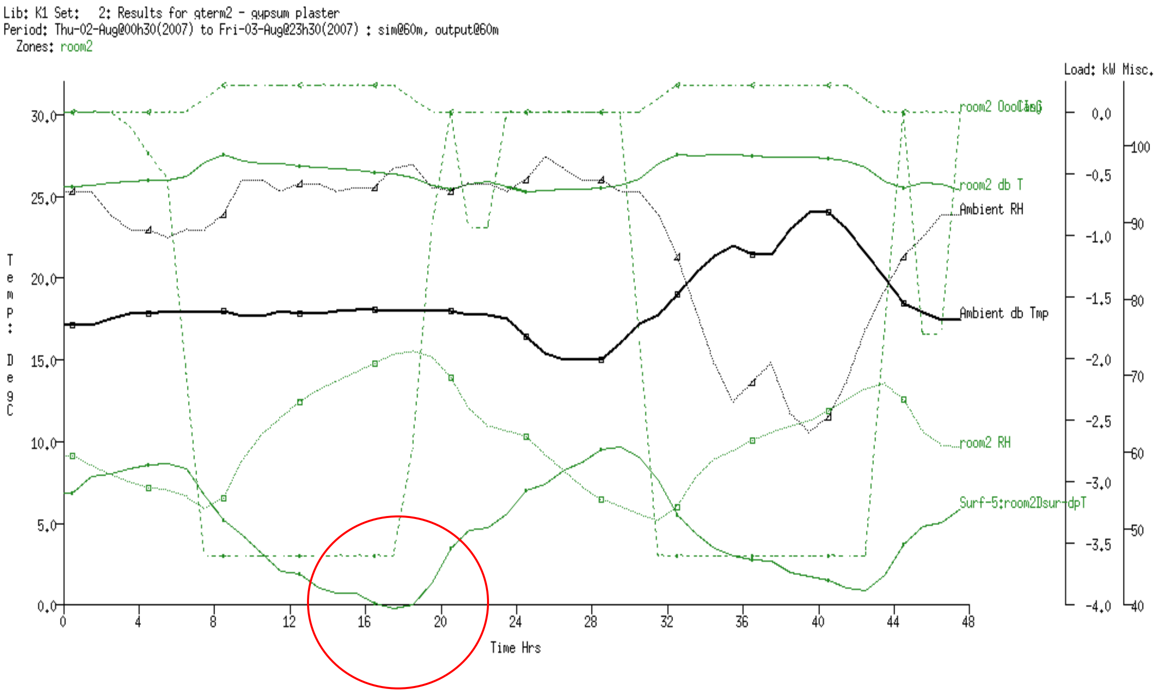

Figure 9 Integrated view on energy and environmental performance during critical days OccCasG - internal heat gains, CoolInj-cooling system injection, AmbientRH-ambient air relative humidity dbT- zone dry bulb temperature, Ambient dbTmp- ambient dry bulb temperature, room1RH- zone relative humidity, Surf-5:room1Dsur-dpT difference between cooling surface temperature and dew point.Critical hours marked with circle

CONCLUSION

The simulation results are concentrated on looking for the answer, if in a modern building, built according to valid Czech standards with respect to energy efficiency, in Czech climate conditions, is possible to use an integrated heating/cooling ceiling system. The question has been analyzed and parameterized. Based on this analysis, a case study, describing typical example of this technology application was created to predict thermal behaviour of the room heated/cooled with this system and to describe thermal comfort behaviour in time during a whole year operation. ESP-r, a modelling tool was applied. The case study was focused on the evaluation of the impact of the room orientation and internal loads describing typical office room use. System is running in switching operation mode which means, that during the whole year both heating and cooling energy sources are available and the technical solution of the system enables to switch heating mode to cooling mode (and vice versa) automatically according to the control system.

In all of the cases there are no problems with the heating. The system can reliably guarantee the required temperature during the whole year. At the same time the simulation shows that common designed heating/cooling capacities (130 and 80 W/m^2) of the ceiling surface are appropriate. Several problems are detected with the cooling, when the designed capacity cannot cover the temperature requirements and occasionally a short-term condensation can occur. This means that the application of this integrated system is limited by its capacity. Especially in the buildings with higher internal gains and connected cooling demand this application is disputable. Following the effect of building orientation individual control of the zones is recommended.

The results from above and the conclusions made from them are valid for the conditions of this simulation.

The future work in this field will be focused on sensitivity analysis on other parameters affecting energy and environmental behaviour of an integrated system aiming to create general design conditions of integrated heating/cooling systems in Czech Republic.

ACKNOWLEDGMENTS

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Optimal Design Method for Buildings & Urban Energy Systems Using Genetic Algorithms

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SUMMARY

In this study, a new optimal design method for buildings and urban energy systems is proposed. Also its applicability is analyzed using a simple case study. Genetic Algorithms (GA) are adopted for this optimization. This method has two optimization steps, equipment capacity and system operation. These optimization steps are calculated simultaneously. The results showed that the proposed method has the potential to be applied to very complex energy systems and achieve significant improvements.

INTRODUCTION

Energy conservation in the building sector is now in great demand given the increased attention throughout the world to environmental preservation of the Earth. Thanks to advances in technology for building facilities, many choices are available when designing an energy system in a building or in an urban area, such as cogeneration systems, triple-effect absorption refrigerators, air source heat pumps, etc. Although so many choices exist, it is difficult to make a quantitative evaluation about which energy system is best, i.e. lowest cost or minimum environmental impact. Reasons for such problems are thought to include:

- Numerous combinations of equipment can be considered when designing systems.
- Operational efficiency of the system is highly dependent on how it is used.
- Prediction of building energy demand at the design stage is very difficult.

In order to resolve these problems, some research on optimization methods for energy system operation using linear programs are proposed [1]. Under such methods, the energy input/output relationship is assumed to be linear. However, with technological progress such as “inverter control” for machine operation, functions concerning energy input/output cannot be considered linear. Also, energy efficiency is highly dependent on equipment size and the operational load factor. Therefore, a new optimal method which can be applied to nonlinear problems is required for improved optimization.
Since the last decade, optimization methods for solving nonlinear optimization problems have advanced steadily. Write and Hanby [2] applied the direct search method, but there may be some inaccuracy if boundary conditions are encountered. On the other hand, the Genetic Algorithms (GA) introduced by Holland [3] have been applied to a diverse range of scientific, engineering, and economic problems. GAs are very suitable for handling complicated optimization problems with nonlinear, discrete and constrained search spaces. Huang et al. [4]

adopted GA for heating, ventilating and air-conditioning (HVAC) control problems. Also Obara et al. [5] applied this method to the control problems of energy systems consisting of fuel cells, thermal storage, and heat pumps, etc. Write et al. [6] applied GA to investigate multi-objective (building energy cost and occupant thermal discomfort) problems to identify the optimal pay-off characteristics. Hongwei et al. [7] applied GA to mixed integer and nonlinear programming problems in an energy plant in Beijing, and made a detailed economic investigation by changing the economic and environmental legislative contexts.

In this paper, a new optimal design method for energy systems is proposed. This method optimizes both equipment capacity and operational control planning simultaneously by using GA. It will be helpful for engineers who design energy systems in buildings or urban areas. Moreover, the application validity of the method was examined by means of a simple case study.

INTRODUCTION

Structure of Optimal Method

Figure 1 shows the process flow of the optimal design method proposed here. There are four steps to achieve optimal solution.

1) Selection of the basic systems (each system is characterized by the difference in combination of equipment composing systems).
2) Optimization of equipment capacity composing each energy system. (Using GA)
3) Optimization of the operational process of each energy system. (Using GA)
4) Selection of the best design by comparing each local optimal solution.

In the 1st step, enumerate selection among the basic energy systems prepared beforehand is conducted, rather than optimal selection. These form the energy system database. The 2nd and 3rd steps are the optimization stages. These optimization problems, both equipment capacity and system operation, cannot be resolved separately due to the strong relationship between capacity and operation. Thus, these two steps are solved simultaneously. In the final 4th step, the best energy system which scores the highest rating among all local optimal solutions of each base system is selected as the optimal solution.

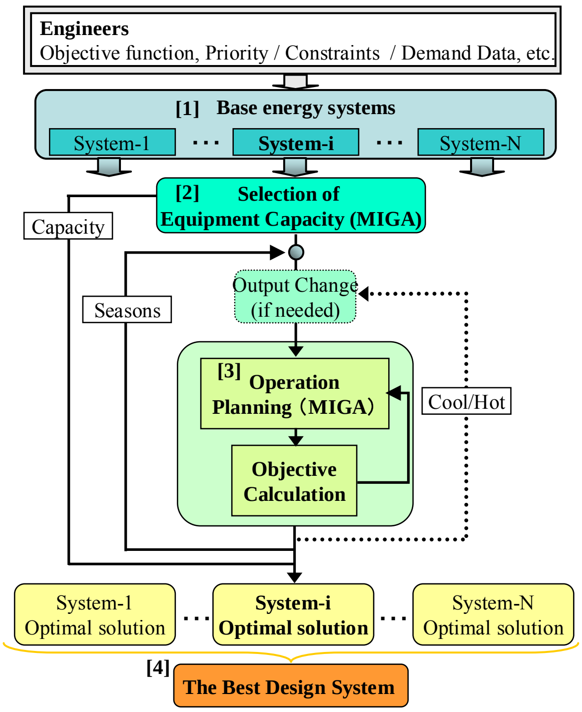

Figure 1.Optimal Design Process using MIGA.

Input Data

The main data used in these optimal calculations are as follows:

Energy Demand Data

Default data from the "Computer Aided Simulation for Cogeneration Assessment & Design III" (CASCADE III) released by The Society of Heating, Air-Conditioning and Sanitary Engineers of JAPAN is used. Figure 2 shows the data applied in this paper.

Equipment Data

Figure 3 shows an example of the equipment performance. With recent technological progress, such as "inverter controls" for equipment operation, functions concerning energy input/output have become nonlinear. A database of equipment performance is prepared and used in the calculation programs.

Application of Genetic Algorithms

GA is a method of solving optimization problems by imitating the evolutionary process based on the mechanics of Darwin's natural selection. Since its introduction by Holland in 1975 [3], GA has been applied to a diverse range of scientific, engineering and economic problems. In GA programs, the optimum solution candidate, which is termed an "individual", is considered to be a living entity. As per the evolution of organisms, each individual's information is ぅ described by sign rows called "chromosomes", which match the individual and the chromosome one to one. GA performs genetic operations such as selection, crossover, and mutations to the chromosomes of each individual, and the subsequent fitness (objective function) of each individual is calculated for evaluation. The individual with the highest fitness value of all the enquired individuals becomes the optimum individual.

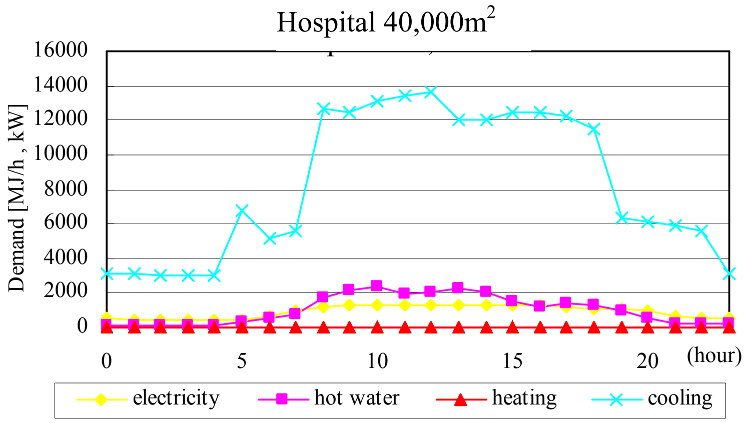

Figure2. Demand data (Aug.)

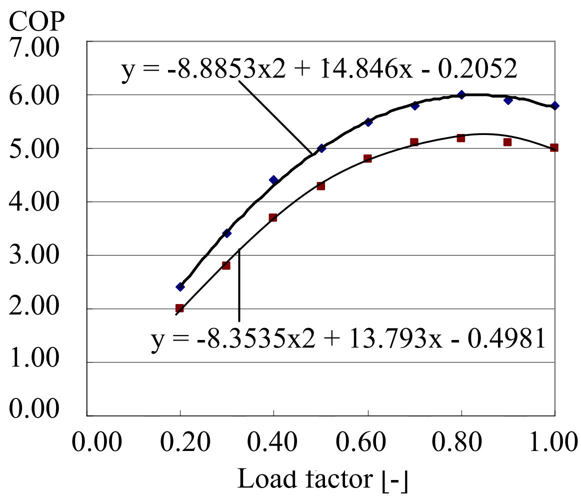

Figure3. Equipment Data (for example)

Moreover, a more efficient GA called a "Multi-island Genetic Algorithm (MIGA)" is used in this research [8]. MIGA involves a distributed genetic algorithm. The outstanding feature of this method is that the population in one generation is divided into several sub-populations called "Islands", and the genetic operations are performed independently on each sub-population. This independency enables the calculation to avoid converging partial optimal solutions. The exchange of individual information, termed "migration", is carried out periodically between sub-populations. Figure 4 shows ideal diagrams of the MIGA description for the operational control problem. Each cell in the "chromosomes" corresponds to a time division, and a number of operation load rates would be arranged in each cell.

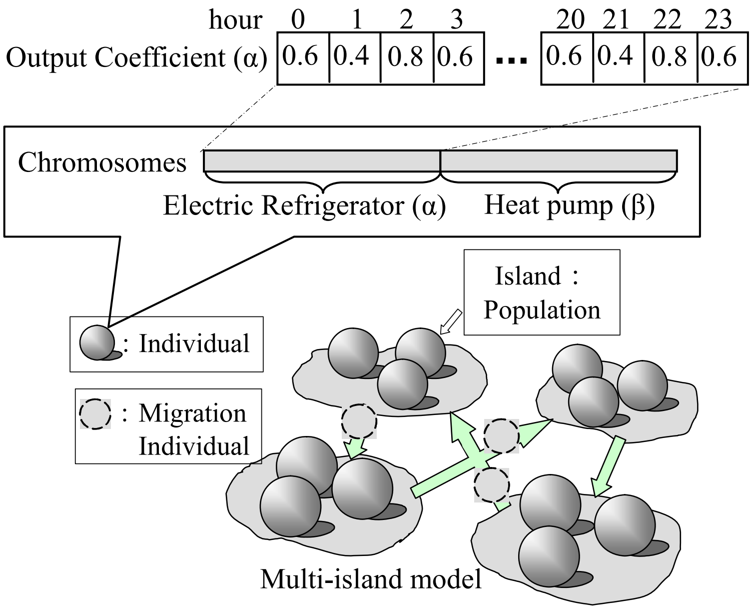

Figure 4.Diagrams of MIGA operations

OPTIMIZATION ANALYSIS

To examine the validity of this optimal design method, analysis for a unit building was investigated experimentally. Before this analysis, a simple-case analysis was conducted [9]. In this case, only the operation planning was optimized by using MIGA. Figure 5 shows a comparison between the optimal solution based on MIGA and the "correct" solution, which was derived by calculation of all operation plans. Here, the optimal solution from the MIGA shows good agreement with the "correct" solution, even though it has a few discrepancies.

In this paper, MIGA is used in two stages, capacity optimization and operational optimization. An analytical outline is described below.

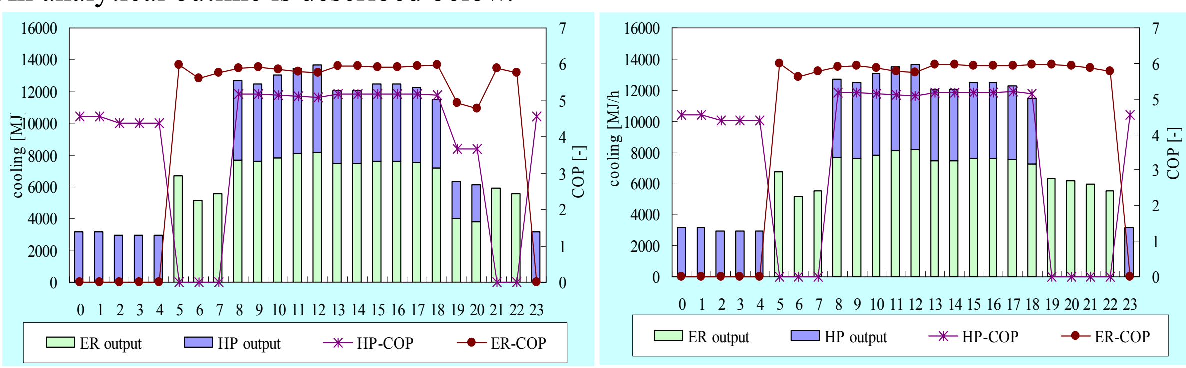

Figure5. Comparison between optimal solution by MIGA (left) and "correct" solution (right)

Analysis Conditions

1) Objective Building: Hospital in Tokyo, Total floor area; 40,000 m^2

2) Analysis term: Representative 1day in Aug. (24h)

3) Energy System: The objective energy system is shown in Fig. 6. In this study, only one system is analyzed. There are two heat pumps for hot water supply (HPw). Operation optimization problems are in Cooling Demand (CD) and Hot Water Demand (WD).

4) Variables: Shown in Table 1. Further details are as follows:

- Equipment Capacity: Set 5-6 types of capacity to each type of equipment composing the energy system
- Operation Plans: Combination of the 24 coefficient numbers for hourly output of the electric refrigerator [CDer (α)] and HPw1 [WDhp1 (β)] are optimized by using GA.

5) Objective Functions: The amount of CO_2 exhaust and energy consumption converted to primary energy.

6) Constraints: The main constraints applied to this analysis are as follows:

- If Heating Demand (HD) is 0 [MJ] for 24h, heat pump for cooling/heating ("HP" in figure 6) is automatically changed to "cooling" operation mode.
- To avoid extreme hourly changes in equipment load, there is a penalty when the hourly load difference is over 60%.
- To avoid frequent ON/OFF operation, the minimum running time is set to 2 hours. Here is also a penalty when this rule is broken.

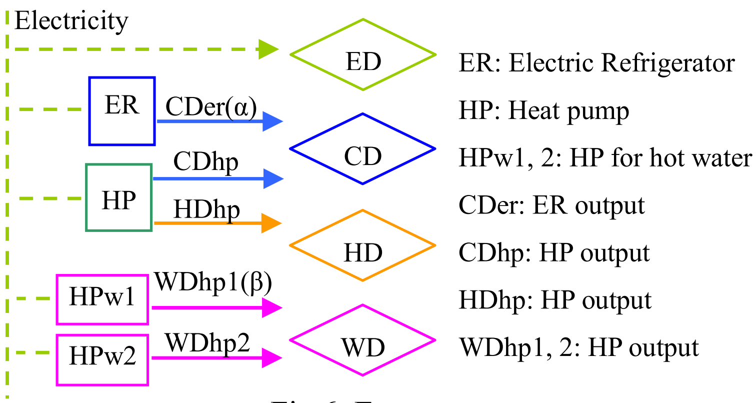

Fig.6: Energy system

Table 1: Variables in optimization

Capacity	
ER	1600, 1900, 2200, 2500, 2800 [kW]
HP1	700, 1000, 1300, 1600, 1900 [kW]
HPw1	450, 500, 550, 600, 650, 700 [kW]
HPw2	50, 100, 125, 150, 175, 200 [kW]
Load Factor	
Cooling (ER)	α_i(i=0_23): (0 , 0.2, 0.4 , 0.6 , 0.8 , 1.0)
Hot water (HPw1)	β_i(i=0_23): (0 , 0.2, 0.4 , 0.6 , 0.8 , 1.0)

Parameters

Table 2 shows the MIGA parameters in this study. To determine the accuracy of the optimization, parameters are decided by checking the effect of the parameters on the optimal solution. Figure 7 shows the effect of the generation number on the optimal solution. 250 is selected as the number of generations because a stable solution can be obtained after almost 200 generations in Fig. 7.

Table 2: MIGA parameters

MIGA parameters	Capacity selection	Operation planning
Size of Sub-Population	5	8
Number of Island	2	3
Population size	10 (5×2)	24 (8×3)
Number of Generation	30	250
Rate of Migration	0.5	0.5
Interval of Migration	5	5
Rate of crossover	1	1
Rate of mutation	0.01	0.01

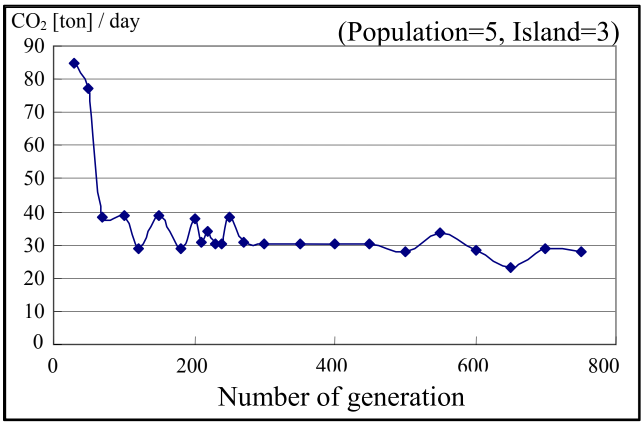

Fig. 7: Effect of the number of generations on the optimal

RESULTS & ANALYSIS

Results of this research are given in Table 3 and Fig. 8. The optimal solution for operation is illustrated in the chromosomes shape.

Cooling Operation

In the capacity optimization step, ER: 2,500 kW and HP1: 1,600 kW are selected as heat sources for cooling which can realize efficient operation. The total capacity is 4,100 kW (=14,760 [MJ/h]), which represents about 10% over the peak for the Cooling Demand (≈13,660 [MJ/h]). Here, smaller equipment such as ER: 2,300 kW or HP: 1,300 kW can be selected under the same constraints. However, if those smaller heat sources are selected, the operational load will be around 100% of its capacity. This results in lower efficiency compared with the optimal result, at which equipment can be operated at 70~90% of capacity. This is because the functions of the equipment efficiency are nonlinear, and this method can take into consideration nonlinear characteristics.

Also in the operation planning step, each equipment is loaded to about 80~90% of each unit's capacity for the peak hours, not that one equipment is loaded 100% of its capacity and another

Table 3: Optimization results.

Selected Capacity			
ER 2,500 (kW)	HP 1,600 (kW)	HPw1 600 (kW)	HPw2 100 (kW)

Selected Operation Plans

hour	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
α	0	0	0	0	0	1.0	1.0	1.0	0.6	0.4	0.6	0.6	0.6	0.4	0.4	0.4	0.4	0.4	0.4	1.0	1.0	1.0	1.0	0
β	0	0	0	0	0	0	1.0	1.0	1.0	0.2	0.2	1.0	0.2	0.2	0.2	0.2	1.0	0.4	0.6	1.0	1.0	0	0	0

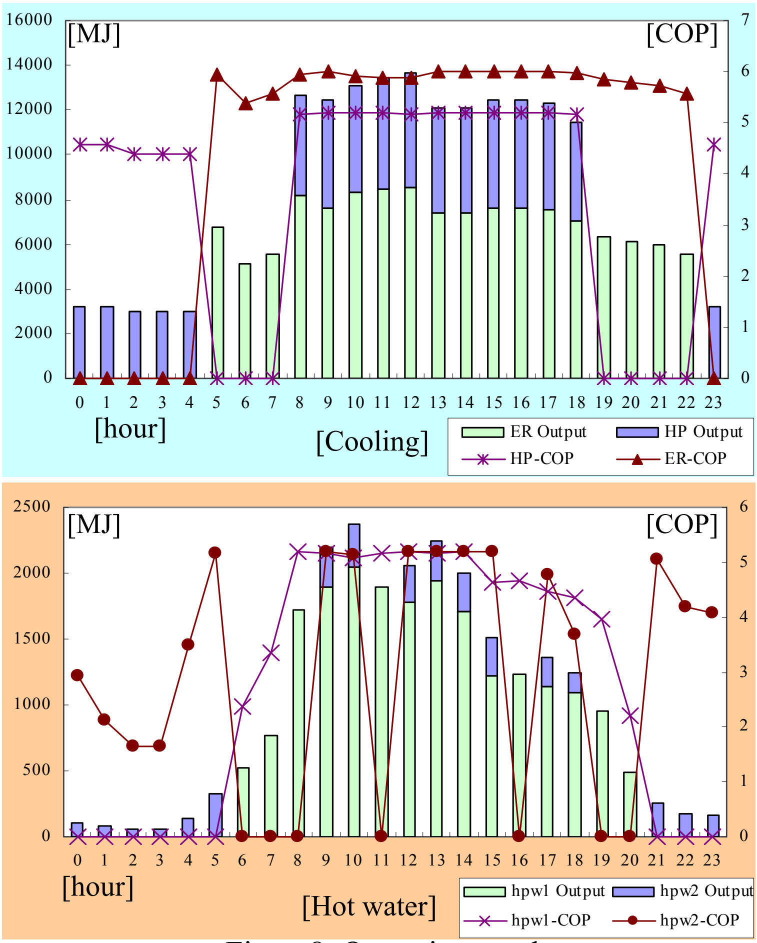

Figure8. Operation results.

is loaded to meet the rest of the demand. Here, the characteristics of nonlinear functions are also taken into account. As a result, a higher COP can be realized.

These results show the advantage of the proposed optimal design method, which can optimize both equipment capacity and operation planning simultaneously. Additionally, for hours with a low Cooling Demand (23~4), only smaller equipment is selected. In this way, the COP can be kept at a high level. And also there is detailed discussion about hot water source operation in previous paper [10]. The total amount of CO_2 exhaust is 23.11 [t/day]. This figure represents about 135 [kg/year·m^2] bearing in mind energy demand distribution for each season. Compared with public data for the amount of CO_2 discharged from existing hospitals in Tokyo, which is about 140-190 [kg/year·m^2], this optimal solution shows good efficiency.

CONCLUSIONS

A new optimal design method for energy systems which can optimize both equipment capacity and its operation planning is proposed. By conducting an experimental analysis, through the application for this method to a building, the validity of the method has been confirmed.

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Calibration of Building Simulation Model by Using Building Automation System – a Case Study

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SUMMARY

A thermal simulation program called IDA-ICE was used in the evaluation of the energy efficiency and indoor climate of a pilot-building in Jyväskylä, Finland. The simulation model was calibrated concerning the heating and cooling energy, because the performance of the building was evaluated using results of the calibrated simulation model as reference to the measured performance. The calibrated simulation model was built for continues commissioning. This paper explains the calibration process and its results. The measured and calculated energy use of the building corresponded to each other as the result of the calibration.

INTRODUCTION

The purpose of the study was to evaluate, how calibrated simulation model can be used in assessment of the building energy performance. The heat use and electricity use of the HVAC-system were evaluated as well as the indoor temperatures.

In the year 2003 the first version of the simulation model was established and the pilot-building was constructed. The measured and simulated results differed from each other about 30 % concerning the heat use and 15 % concerning the electricity use in the first year. The results are presented more exactly in sources [1] and [2].

The simulation model was calibrated utilizing building automation system (BAS) and building energy management system (BEMS). The building automation system was used in determining the ventilation air flow rates, the set point temperatures of the room devices and of the supply air. The air flow rates, which were underestimated in the original model, were corrected to the calibrated model. The measured air flow rates and temperature of the supply air and the electricity use of the lighting and the office devices were used as time dependent input parameters in the simulation model. The building energy management system was utilized in determination of the electricity use and the heat use of the building. The input files needed in the simulations were generated from BAS and BEMS.

The indoor temperatures were evaluated using duration curves concerning utilization time of the building. The duration curves were also compared to the simulated temperatures.

Basic Information of the Building

The case building is situated in Jyväskylä, 350 km from the Finnish capital, Helsinki. The building gives accommodation for the Department of Information Technology of Jyväskylä

Polytechnic. The construction works was finished by May 2003 and the building was taken into actual use in August 2003.

The five-storey building consists mainly of classrooms. The building has also an assembly hall, a library, an auditorium, a lunchroom, a kitchen and a bomb shelter, which are situated in the ground floor. The working rooms of teachers are situated in the fifth floor.

The floor area of the Pilot building is approximately 9 500 m^2 and its exterior volume is 38 700 m^3. Over 1 000 students are studying and more than 40 teachers and other personnel are working in the building.

The Principle of the HVAC System

The classrooms and the working rooms of the case building have heating and cooling panels detached from the ceiling. Floor heating is used all over the ground floor. The conventional radiators are not used for heating.

The building is heated by district heating and it is cooled by electricity using two refrigeration compressors and one storage tank and in addition free cooling by outdoor air is used. The ventilation system has combined heating-cooling-free cooling coils and a rotating heat recovery unit. HVAC-system has also a mechanism, which recovers the indoor overheat from the cooling panel's liquids to the supply air. The innovative patented system is advance low temperature heating system. The principle of the HVAC-system is explained in reference [3].

The building is conditioned using an integrated heating, ventilation and cooling system. The heating and cooling liquids are transferred through a 3-pipe system with common return liquid. The HVAC-system is presented in **figure 1**.

The building has variable air volume (VAV) ventilation system [4], which is controlled with the both indoor air temperature and carbon dioxide level [5].

EMS and BEMS

The case building has separate energy building management system (EMS) and building automation system (BAS).

BES measures hourly heat, electricity and cold water uses. Measured values are reported monthly, daily and hourly using tables and figures. Heat use, which is weather corrected by degree day method, is also available. Besides the control of the HVAC-system, building automation system measures temperatures in 20 of the total 50 rooms, temperatures of liquids, position of valves, rotation speeds of the motors of fans and pumps as well as chamber pressures in air-handling-units. Also timetables of the AHUs and set points and set point curves can be viewed on the BAS.

BAS can be used over internet, using a password protected user interface. A part of the measured data is saved to the BAS server of the case building, and it can be transferred trough the Internet using three file formats, HTML, CSV and XML. CSV-protocol was used in the evaluation of the performance of the building.

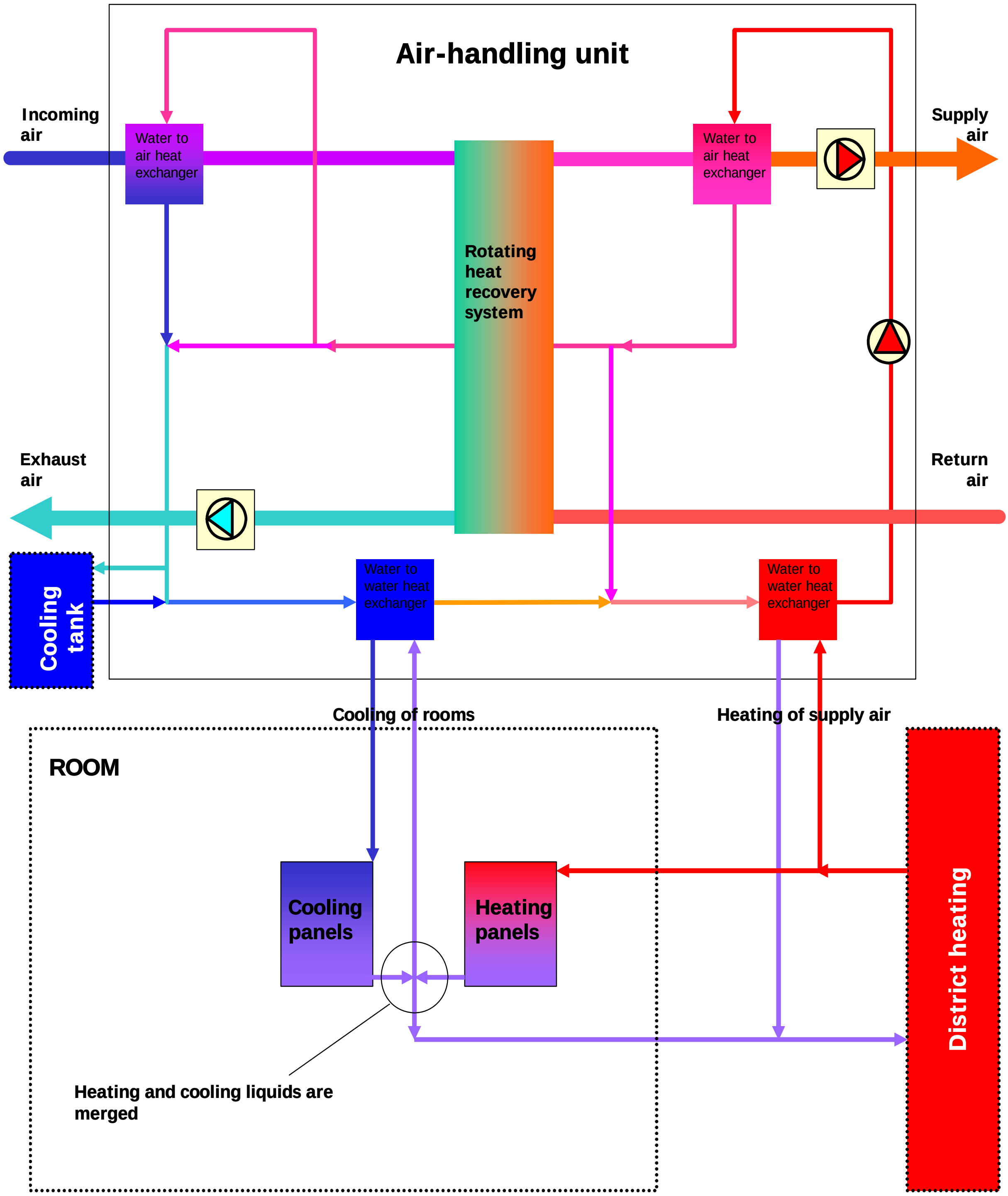


Figure 1 The HVAC-system of case building

METHODS

Simulation Program

IDA-Indoor Climate and Energy is a program designed for studying the indoor climate of zones of a building and for simulating energy consumption of a building [6]. IDA-ICE is an extension to an IDA simulation environment and an advanced user can simulate any system with the aid of the general functionality of the IDA simulation environment.

In normal circumstances, the building to be simulated consists of one or more zones and a primary (heat producing) system and one or more air handling units. The weather data is supplied by weather data files or it can be synthetically generated. The shading of the surrounding buildings is also evaluated.

Predefined building components and other parameter objects can be loaded from a database. This can also be used to store personally defined building components. The mathematical models of IDA-ICE are described in terms of equations using a formal language named NMF, which can be used to make new modules or to replace existing program modules. The actual function of the components and relationship between the components models are modeled using NMF-files.

Calibration of the Simulation Model

The calibration of the simulation model of the case building included the following steps:
1. Developing a model which determines the volume flows of VAV-system using the rotation speed and pressure rice over the fans in calculation
2. The internal heat gains from the electric devices were determined using hourly electricity measurements from the BEMS
3. The leaking of the building was evaluated using information from the literature
4. The set temperatures of the supply air as well as the room temperatures were determined using the BAS
5. The set temperatures of the entering heating and the cooling liquids were also determined using the measurements of the BAS
6. The simulated output values, for instance for the indoor air temperature, the temperature of return liquids, and the energy consumptions, were compared to measured ones to evaluate the validity of the input parameters

The first version of the simulation model of case building was made by a Finnish HVAC engineer company Olof A. Granlund. They modeled the building with a simulation program RIUSKA [7], which is based on DOE 2.1 E.

Case building was modeled again by the researchers IDA-ICE and the model was calibrated using the measurements of building automation system. The principle of the calibration process is presented more precisely in the reference [8] concerning the commissioning and in the reference [9] concerning simulation model itself.

The volume flows to each of the rooms were calculated based on the calculated indoor temperature and the calculated carbon dioxide emission from the people. However, the calculated total volume flows differed much from the measured ones, because the exact patent protected control strategy and the real occupancy level were not known. Due to that the measured and calculated heat uses differed quite much from each other.

At the next phase the volume flows of each of the VAV-flow AHUs were taken directly from BAS-measurements based calculation model to the simulation program. The volume flows were divided in the rooms in the proportion of the floor areas, because the exact volume flows of the rooms were not saved to database of the BAS. At the same time the electricity use of the rooms were determined from the hourly electricity measurements and divided in the rooms with the same principles as the division of the volume flows. The electricity use of the HVAC was reduced from the total electricity use.

RESULTS

Use of Energy for Heating

Figure 2 presents the measured and calculated heat use of the case building in the first and second year of the building in use. The measured heat use was 30 % more than the simulated one during the first year. The difference was mainly due to the difficulty to determine the actual volume flows of VAV-system.

The heat use of the second year was calculated after several corrections to the model. The calibration process included procedure explained in the previous chapter. The most important measure was to determine the variable air volume flows using a model. It was found out comparing the first estimation of the volume flows afterwards, that volume flows were underestimated with 30 % in the first year.

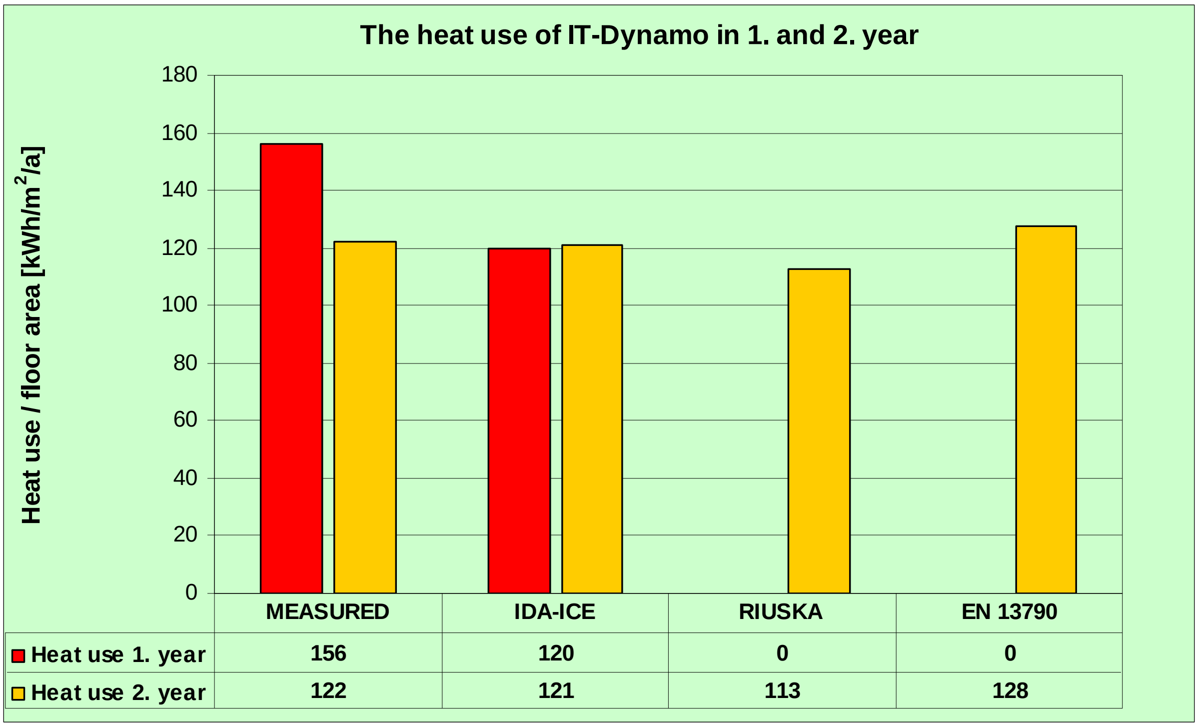


Figure 2 The heat use of the case building during the first and second year in use

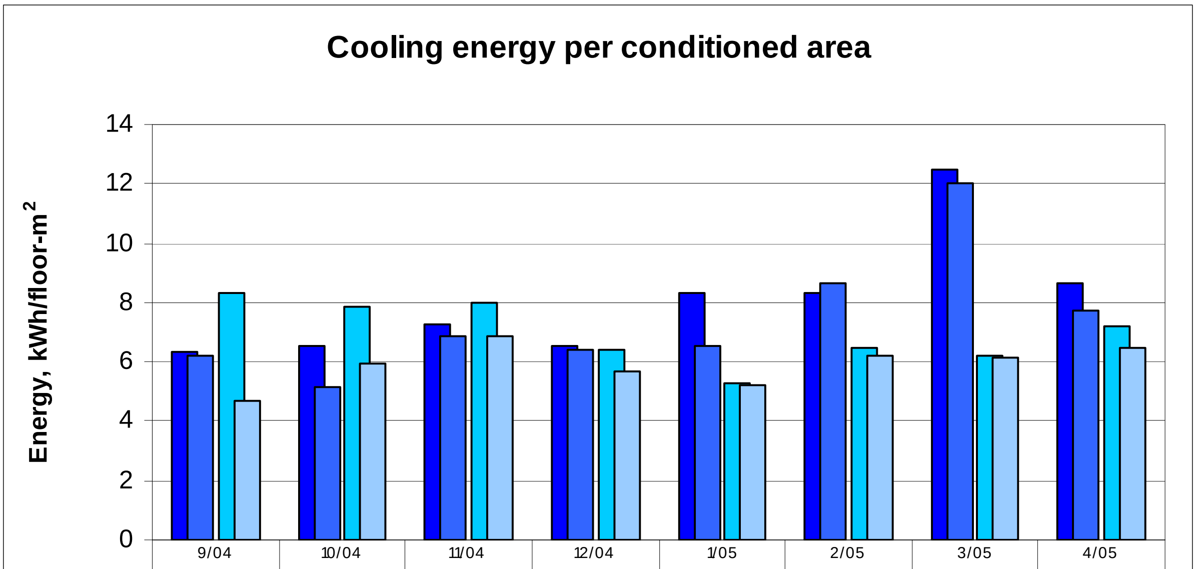
The heat use of the second year was calculated using three different calculation programs, two simulation program IDA-ICE and RIUSKA and a standard calculation method (EN 13790) [10]. The results of RIUSKA were not calibrated, and that is why the results of calibrated model of IDA-ICE correspond best to the measured heat use in the second year. The result of standard calculation overestimated the heat use with several percents.

The Electricity use of HVAC-system

Evaluation of the electricity use HCAC-system including fans, pumps, cooling devices etc. was difficult due to the same problems as in the determination of the heat use. The electricity use of the fans was dominant concerning the electricity use of HVAC. That is why the determination of the volume flows [4] was important also to find out the exact electricity use of fans and therefore the whole HVAC-system.

In the first year the measured electricity use of HVAC was 47 kWh/floor-m^2/a as while the calculated one was 35 kWh/floor-m^2/a. The measured and calculated electricity use of HVAC -system was according to both the measurement and the simulation 40 kWh/floor-m^2/a in the second year. The difference between the measured and calculated values disappeared because of the calibration.

Besides the determination of the volume flows, the electricity of the cooling devices was able to be determined more exactly in the second year than in the first year. That was done developing a mathematical model based on measurements. The electricity use of the cooling system was according to electricity measurements 12 kWh/floor-m^2/a. The energy production of the free cooling system was according to the measurement based model 36 kWh/floor-m^2 and according to simulations 33 kWh/floor-m^2/a. The whole panel cooling energy use of was according to the measurements and the simulations 48 kWh/floor-m^2/a at the same time, and therefore 75 % of the cooling power was produced using the free cooling. The measured and the calculated total cooling energy need are presented in **figure 3**. The total and free energy needs were underestimated a little in the simulations compared to the measurements in spring 2005.



	9/04	10/04	11/04	12/04	1/05	2/05	3/05	4/05
Total measured	6	7	7	7	8	8	13	9
Free measured	6	5	7	6	7	9	12	8
Total calculated	8	8	8	6	5	6	6	7
Free calculated	5	6	7	6	5	6	6	6

Figure 3 The measured and calculated energy use for cooling in the case building between

Approximately 23 kWh/m^2/a of the total electricity use of the HVAC-system 40 kWh/m^2/a was consumed by the fans in the ventilation system. The electricity use for the cooling was 12 kWh/m^2/a. The rest of the electricity 5 kWh/m^2/a was consumed by the pumps and the control devices. The electricity use of the lighting, PC:s and other office devices was 83 kWh/m^2/a in the second year, which was two times higher compared to the consumption of a similar buildings in Finland.

Free Energy Savings

An estimation of the energy production of the free cooling system was measured to be 286 MWh and simulated to be 259 MWh between 09/2004-08/2005. The total COP of the cooling system is estimated to be 1.2, which is typical COP-value of the whole cooling system. The total COP includes all the power losses in the production of the cooling energy. It is not the same as COP of a refrigerator, which is normally between 2 and 3.

The savings of heat use were 162 or 107 MWh according to measurements and calculations respectively. In the calculations the lowering in efficiency of the heat recovery unit was taken into account. The total energy savings was about 14 % in heat use, 25 % in electricity use and 20 % in the average according to the measurements. The same values were a little lower according to the simulation results, but the differences are because the deficiencies in the model. Savings are calculated also in euros in a year and current value of the savings in ten years calculated with an interest rate of 5 %. The investment costs of the system are not remarkable more than in a regular system according to manufacturer of the system. The results are presented in **Table 1**.

Table 1 Comparison of calculated and measured savings in the case building

	Measured			Simulated		
	Electricity	**Heat**	**Total**	**Electricity**	**Heat**	**Total**
Total energy use in MWh	1140	1159	2299	1150	1150	2299
Energy saving in MWh	286	162	448	259	107	366
Saving in %	25	14	20	23	9	16
Saving in euro	22895	8107	31002	20704	5358	26062
Current value in euros / 10 years	176793	62596	239389	159869	41375	201244

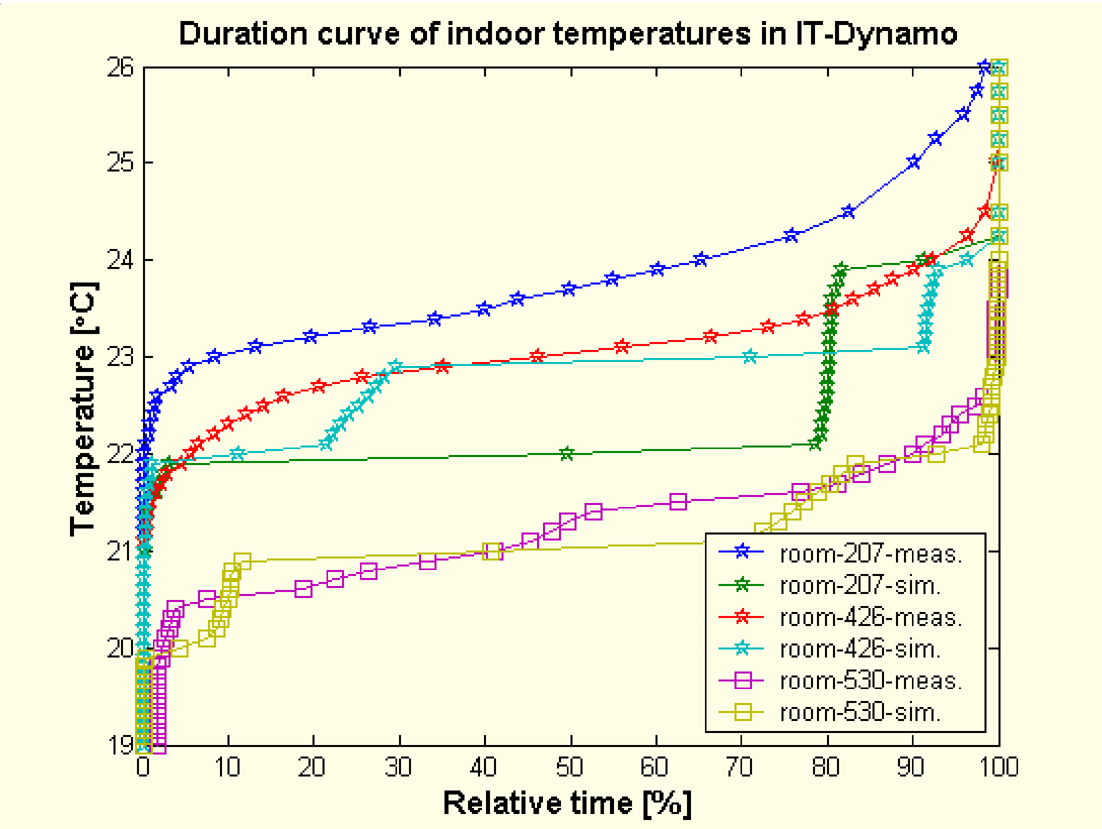


Figure 4 Duration curves of the indoor temperatures in the case building

Evaluation of the Indoor Climate

The indoor temperatures of the class rooms and the working rooms were estimated using duration curves. The duration curves presented in **figure 4** were calculated concerning the time, when the building has been in actual use. The temperature has remained between 21 °C and 26 °C in the class rooms and between 20 °C and 23 °C in the working room. The calculated and measured temperatures correspond quite well to each other in a little working room 530, but differed about 1 °C from each other in large classrooms 207 and 426 between 1.9.2004 and 30.5.2005.

DISCUSSION

The energy performance and the indoor climate of an educational building were evaluated using measurements results and simulation model. The calibration of the simulation model was done based on measurements of the building automation system and energy management system. The calibration process took two years, but after that the measurement and simulation results to correspond to each other.

The main reason for differences between measurement and calculation results were the difficulties in determining the exact volume flows, the control strategy and the actual the utilization factor of the building. The lack of the information of the exact control strategy was found out to be the main reason for difficulties to calculate the exact volume flows, the indoor temperatures and the energy use for cooling. When the volume flows and indoor set temperature was taken from measurement results of BAS, the calculated heat use and electricity use of HVAC-system correspond to each other quite well.

The simulation model should be kept as simple as possible when assessing a real system. The complicated models are difficult to calibrate, when the calculation parameters are more difficult to find out. The duration of the calibration process should also be much shorter, one month or two at the most, to make the calibration cost-effective.

LITERATURE

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Modeling Of Non-Linear HVAC system using SIMBAD

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ABSTRACT

The term heating, ventilating and air conditioning (HVAC) covers a wide range of equipments. An air conditioning system maintains desired environmental conditions within the zone. In most cases term heating, ventilating and air conditioning (HVAC) systems, a central air supply provides air at the controlled temperature and flow rate for heating or cooling the space. The main objective of this project was to develop an accurate, dynamic and non-linear digital model of an HVAC system using SIMBAD that could be implemented in the MATLAB/Simulink environment [13]. HVAC system models have been implanted in many proprietary and public domain programs. However, our goal was to implement a model in MATLAB/Simulink, a software system and tool box widely used to study dynamic response and control systems. This paper documents the model for the benefit of those interested in studying HVAC control. The model was also verified by comparing its output to the dynamic response of an experimental HVAC system (see [16]for a discussion of the experimental facility). Our point of departure was the SIMBAD toolbox from CSTB Paris, used in the MATLAB/Simulink environment [12].

INTRODUCTION

The SIMBAD toolbox provided many of the components of HVAC systems used in the modeling. The components of the system are modeled using standard Simulink block diagrams and models compiled in C code. The main characteristics of the SIMBAD tool box is the definition of connection vectors. The main two vectors used are :

The air vector : Air dry bulb temperature [C], Air humidity ratio [kg/kg], Static pressure [Pa], Mass flow rate [kg/s]

The water vector : Water temperature [C], Static pressure [Pa], Mass flow rate [kg/s]

SIMBAD is obviously in a state of evolution (and like most software will probably always be in that state). Therefore we developed or modified models to reflect our needs, to consider North American versus European design practice, and to introduce options for multi input multi output control.

MODELING

The literature is quite rich about dynamic modeling of HVAC systems using linear approximation. This research is mainly concerned with the non-linear modeling and simulation of various HVAC components and the overall HVAC system.

Components that are modeled here are : Dynamic heating and cooling coil, Electrically heated boiler, Variable speed pump, Two-way convergent air duct, Outside and return dampers, Three-way mixing valve, Variable speed fan, PI controllers.
The first four component models came from the SIMBAD toolbox. The next three were developed in- house and the PI controller model was taken from the Simulink controls tool kit.

Modeling of heat exchanger
Almost every thermal environmental system involves heating or cooling of the atmospheric air. It forms the part of both water and air cycle. The parameters for our heat exchanger modeled are: 4 pass, 23 run, plate fin and counter flow type heat exchanger. The tube material is copper and the tube core is 0.01 m in diameter. The fins have a pitch of 472/m,are 0.25 mm thick and are made of aluminum. These parameters are by way of example. Any rational set of parameters can be used in the model.

Simulink elements and five programs in "C" language calculate specific variables such as: Geometry,

Exchange coefficients, Pressure calculations, Heat transfer, Temperature (C programs), Pressure (C programs), Humidity ratio (C programs), and Mass flow rates (C programs).

Temperature calculations
This section deals with the calculations of output air temperature. The equations used are :

$$T_{ao} = T_{ai} - Q_s / (C_{pi} * m_a) \qquad (1)$$

T_{ao} = Outlet air temperature (C), T_{ai} = Inlet air temperature (C), Q_s = Sensible heat transfer rate (kW), m_a = Mass flow rate (kg/s).

The sensible heat transfer rate is the heat transfer rate which is solely manifested in raising the temperature of the incoming air.

$$C_{pi} = C_{pa} + C_{pg} * G_i \qquad (2)$$

C_{pi} = Inlet air specific heat capacity (kJ/kg K), C_{pa} = Specific heat capacity of air (kJ/kg K), C_{pg} = Specific heat capacity of steam (kJ/kg K), G_i = Humidity ratio of inlet air (kg/kg).

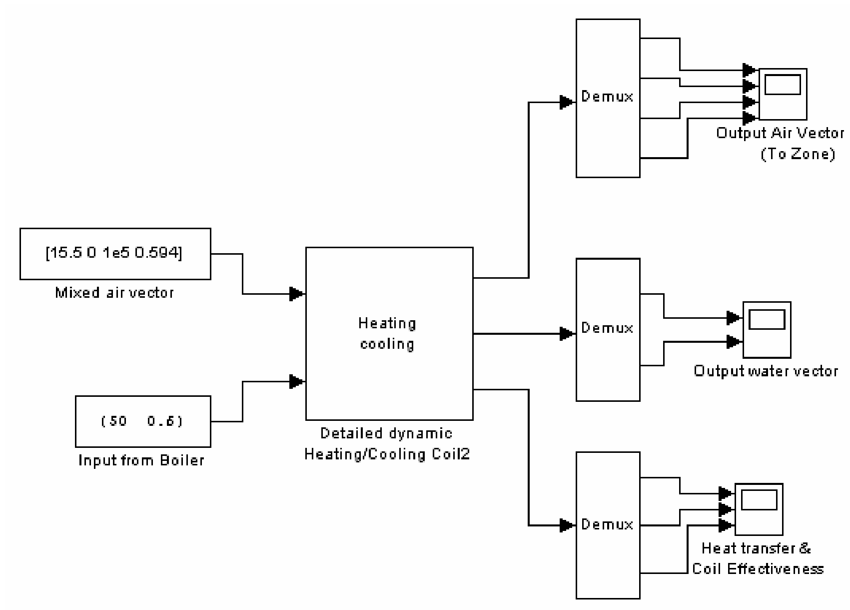

Figure 1 Dynamic model of the heat exchanger

$$Q_s = Q_a * SHR \qquad (3)$$

Q_s = Sensible heat transfer rate (kW), Q_a = Heat transfer rate on air side (kW), SHR = Sensible heat ratio (kg/kg).

$$Q_a = (T_{ai} – T[0]) / R_{st1} \qquad (4)$$

T_{ai} = Inlet air temperature (C), R_{st1} = Air-side thermal resistance (K/kW), T[0] = Initial coil temperature (C), R_{st1} depends on the type of regime, either dry or wet . Here it is assumed to be a dry regime.

$$R_{st1} = B_f * R_{st} \qquad (5)$$

R_{st} = Coil total thermal resistance (K/kW), B_f = By-pass factor for dry coil.

$$R_{st} = 1.0 / [Effect * m_{cp}] \qquad (6)$$

Effect = coil effectiveness, m_{cp} = minimum capacity rate (kW/K)

$$C_{air} = C_{pi} * m_a \qquad (7)$$

C_{air} = Air capacity rate (kW/K), C_{pi} = Inlet air specific heat capacity (kJ/kg.K), m_a = Mass flow rate of air (kg/s).

$$C_{water} = C_{pw} * m_w \qquad (8)$$

C_{water} = Water capacity rate (kW/K), C_{pw} = Water specific heat capacity (kJ/kg.K), m_w = Flow rate of water (kg/s).

Pressure Calculations

Here, we are introducing one independent variable **u,** and another dependent variable, **x** . Both u and x are not used in the model. It is used here to explain the model clearly.

$$AD = (273.16 / (273.16 + T_{ai})) / 1.293 \qquad (9)$$

AD = Air density (kg/m^3), T_{ai} = Inlet air temperature (C).

$$u = (1 / AD) * m_a \qquad (10)$$

$$x = u^2 * \text{Resistance coefficient} \qquad (11)$$

u = Volumetric flow rate (m^3/s), x = Pressure drop (Pa), m_a = mass flow rate of air (kg/s) and the resistance coefficient is an input parameter into the heat exchanger .

$$P_o = P_i - x \qquad (12)$$

P_o = Output pressure (Pa), P_i = Inlet pressure (Pa).

Humidity ratio calculations :
The equation used to calculate humidity ratio is :

$$G_o = G_i - (Q_a - Q_{sensi}) / (HFG * m_a) \qquad (13)$$

G_o = Outlet air humidity ratio (kg/kg), G_i = Inlet air humidity ratio (kg/kg), Q_a= Heat transfer rate on air side (kW), Q_{sensi} = Sensible heat transfer rate (kW), HFG = Latent heat of vaporization of water (kJ/kg), m_a = Mass flow rate of air (kg/s).

Mass flow rate calculations
There is no change in the mass flow rate of air through the heat exchanger. On the water side there are only two variables-temperature and flow rate. As the flow rate remains constant, the only calculation is that of temperature.

$$T_{wo} = T_{wi} + Q_r / C_{water.} \qquad (14)$$

T_{wo} = Water output temperature (C), T_{wi} = Water input temperature (C), Q_r = Heat transfer rate on water side (kW), C_{water} = Water capacity rate (kW/K).

$$Q_r = (T_{ci} - T_{wi}) / R_{st2\,.} \qquad (15)$$

T_{ci} = Initial coil temperature (C), R_{st2} = Water + metal thermal resistance (K/kW), T_{wi} = Water inlet temperature (C).

$$R_{st2} = 0.001 * R_{st} \qquad (16)$$

R_{st} = Total coil thermal resistance (K/kW). The value for R_{st} comes from program header files.

$$C_{water} = C_{pw} * m_w \qquad (17)$$

C_{pw} = Specific heat capacity of water (kJ/kg.K), m_w = Mass flow rate of water (kg/s).

Electrical Boiler
The boiler modeled here is an electric heat boiler. The design of electric boilers is largely determined by the shape and heat release rate of electric heating elements used. The example boiler has a capacity of 14 kilowatts. The heater is controlled through a PI control loop that tracks the set point temperature under transient conditions as illustrated in figure 2. The boiler output water temperature can also be calculated using steady state calculations.

The boiler model has two main sections, the thermal cycle and hydraulic cycle. The thermal cycle model calculates the output temperature of the water and the hydraulic cycle model calculates output pressure depending on the flow rate.

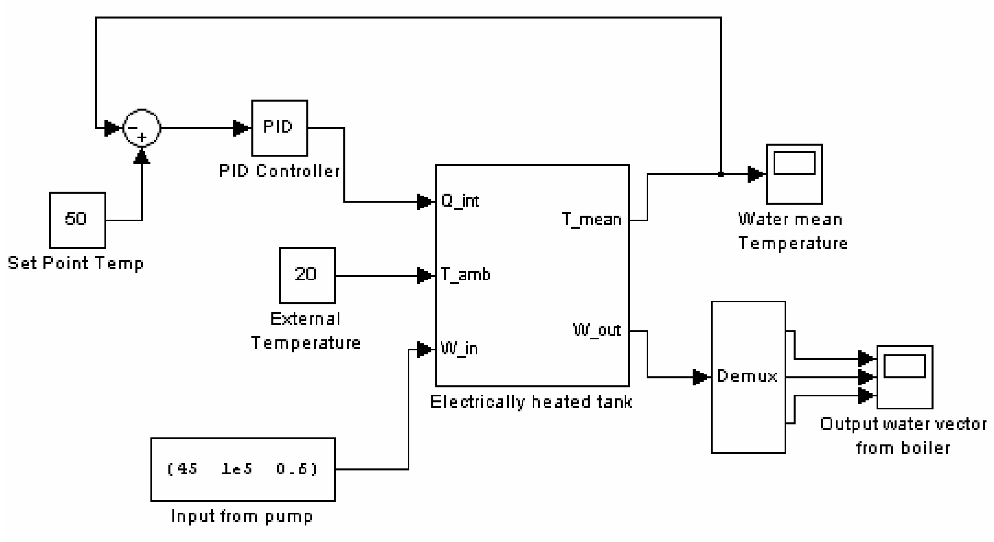

Figure 2 Electrically heated boiler

Thermal Cycle

The inputs to the thermal model are:

Q_{int} = Command from the controller. This entity depends on the required hot water temperature.
W_{in} = Input water vector
T_{amb} = Ambient temperature (C)
The inputs are used to define a column matrix. The matrix will have 3 entities, A, B and U. These three entities will be used to calculate outlet water temperature both in steady state and in transient conditions.

Defining the matrix :

$$A = ((m_{wi} * C_w / MC) + (Ah_{tank} / MC))* \text{Gain} \qquad (18)$$

m_{wi} = Inlet flow rate of water (kg/s), C_w = Specific heat capacity of water (kJ/kg.K) Ah_{tank} = Surface heat exchange product (kW/K), MC = Total thermal mass (kJ/K).

$$MC = mC_p\_water + mC_p\_metal \qquad (19)$$

mCp_water = Total thermal mass of water (kJ/K), mCp_metal = Total thermal mass of metal (kJ/K)

$$mC_p\_water = V\_water * \rho_{water} * C_w \qquad (20)$$

ρ_{water} = Density of water (1000 kg/m^3)
V_water = Volume of water in the boiler (m^3), C_w = Specific heat capacity of water (kJ/kg.K),

$$mC_p\_metal = V\_metal * \rho_{copper} * C_{pm} \qquad (21)$$

V_metal = Volume of metal on the boiler (m^3), C_{pm}= Specific heat capacity of metal (kJ/kg.K), ρ_{copper} = Density of copper (8000 kg/m^3).

As per Simbad, the value of the gain is constant at –1. As shown in the equation, the values of C_w and Ah_{tank} are normalized with MC.
The second element in the matrix is B. It has three entities, B1, B2, B3.

$$B1 = m_{wi} * C_w / MC \qquad (22)$$

$$B2 = Ah_{tank} / MC \qquad (23)$$

$$B3 = 1 / MC \qquad (24)$$

The vector B can be defined as

$$B = [\ B1 \ \ B2 \ \ B3\ ] \qquad (25)$$

The last and final element in the matrix U is also a vector consisting of

$$U = [\ T_{wi}\ T_a\ Q_{int} * P_{nom}] \qquad (26)$$

T_{wi} = Inlet water temperature (C), T_a = Ambient temperature (C), Q_{int} = Controller input, P_{nom} = Total capacity of the coil (14 kW)
Subsequently the matrix will have three outputs: **[A B U].** Here we have considered transient calculations. The matrix defined above forms the input for the transient calculations. For this purpose, we are defining two variables x and y.

Where

$$x = \sum (B * U) \qquad (27)$$

and

$$y = A + x \qquad (28)$$

Now the output water temperature is calculated by the formula

$$T_{wo} = A * \int y \qquad (29)$$

The output water temperature, in the transient calculation is determined by integrating the inlet water temperature, with matrix A, over time, until the set point temperature is attained.

Hydraulic Cycle :
The hydraulic phenomena calculates output pressure depending on the pressure drop during the nominal flow rate of the fluid. In this model we have assumed the pressure drop to be zero.

Pump

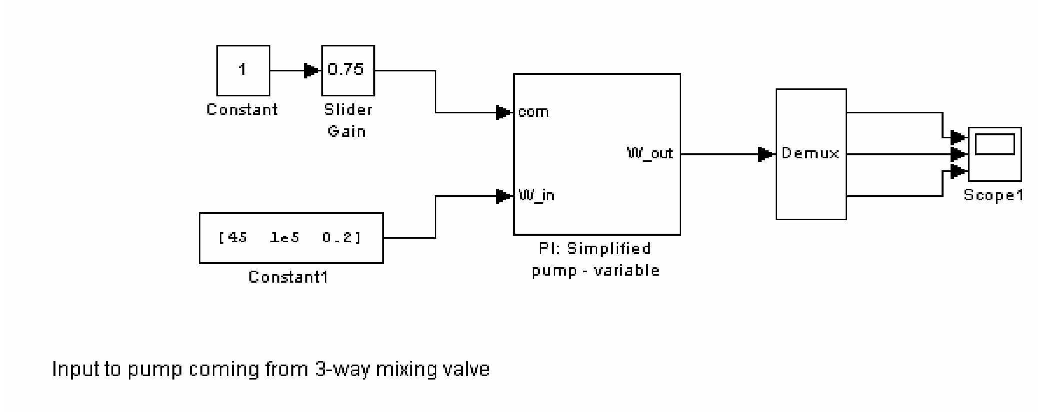

Figure 3 Variable speed pump

One of the main methods for transporting thermal energy into, out of, and within a building is with *water*. The pump inputs the required flow rate into the water circuit. The flow rate from the pump depends on the command input given at the 'com' input port as illustrated in figure 3. The pump speed can be varied to vary the flow rate of the fluid as required up to the maximum flow capacity of the pump.

The equation is given as

$$F_w = Com * \text{Maximum flow rate. (kg/s)} \qquad (30)$$

Maximum flow rate is an input function of the pump.
It is predominantly a linear function. The output water vector will have the same temperature and pressure as the input; only the flow rate will be changed depending on the command.

Two-way convergent air duct
The two-way convergent air duct gives the resultant air vector due to mixing of two air vectors - the fresh air vector from the atmosphere that comes through the outside air damper and the return air vector from the zone coming through the return damper. The flow rates of the two vectors are added and no head loss is considered.

Mixing temperature

$$T_{mix} = (T_{a1} * m_{a1} + T_{a2} * m_{a2}) / (m_{a1} + m_{a2}) \qquad (31)$$

T_{a1} = Temperature of outside air (C), T_{a2} = Temperature of return air (C), m_{a1} = Flow rate of outside air (kg/s), m_{a2} = Flow rate of return (kg/s.

Mixing humidity

$$U_{mix} = (U_1 * m_{a1} + U_3 * m_{a2}) / (m_{a1} + m_{a2}) \qquad (32)$$

U_1 = Humidity ratio of outside air (kg/kg), U_3 = Humidity ratio of return air (kg/kg) m_{a1} = Flow rate of outside air (kg/s), m_{a2} = Flow rate of return air (kg/s)..

Mixing pressure

$$P_{mix} = (P_1 * m_{a1} + P_2 * m_{a2}) / (m_{a1} + m_{a2}) \qquad (33)$$

P_1 = Pressure of outside air (Pa),P_2 = Pressure of return air (Pa), m_{a1} = Flow rate of outside air (kg/s), m_{a2} = Flow rate of outside air (kg/s)

The values of both P1 and P2 are equal at the entrance of the mixing box. The resultant flow rate will be the sum total of both the individual flow rates.

The following components were designed and built in-house. The same components in SIMBAD were not working in accordance with the components in the experimental facility.

Three-way mixing valve

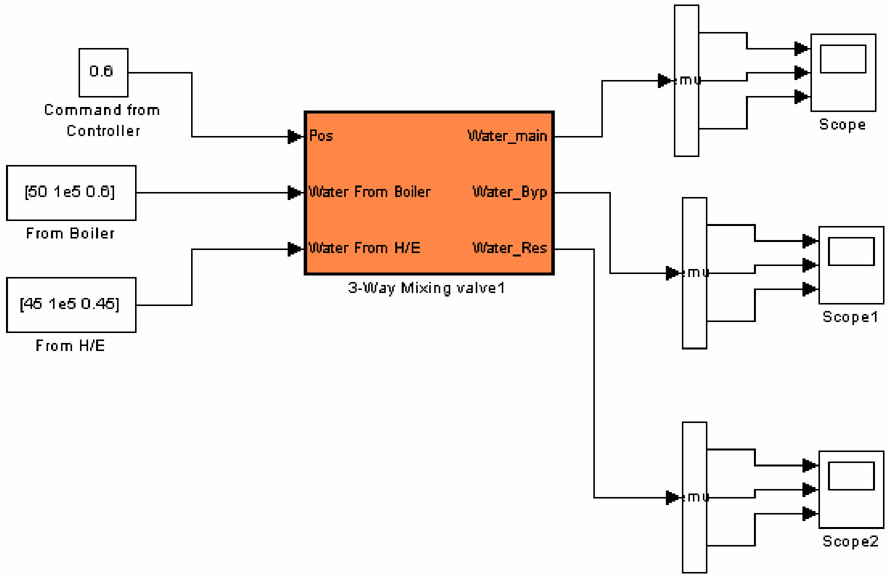

Figure 4 Three way mixing valve

The valve has three inputs and three outputs. The three inputs are: Command signal from the controller depending on the set point temperature, Input port for water coming from boiler, Input port for water retuning from the heat exchanger, The three outputs are: Main water vector going into the heat exchanger, and Water vector going into the bypass circuit.
The resultant water port mixes water vectors from the heat exchanger and bypass circuit. The bypass water vector and the water vector are mixed internally. Thus the bypass port is used for monitoring purposes and it is not connected anywhere in the system.

The valve has three parts:
Flow fraction determination: Depending on the command from the controller, which in turn is determined by the set point temperature, a look up table between the command and flow fraction determines the percentage of total flow going into the heat exchanger. The command from the controller is in effect the percentage opening of the valve.

Flow calculations: The input to this function is the flow fraction in the form of percentage of the total flow going into the heat exchanger. The flow fraction in the by pass circuit is determined by formula::

$$m_{w\ (bypass)} = m_w - m_{w\ (main)} \qquad (34)$$

m_w = Total flow rate of water from the boiler (kg/s), $m_{w\ (main)}$ = Flow rate of water into the main circuit (kg/s), $m_{w\ (bypass)}$ = Flow rate of water into the bypass circuit (kg/s).

Temperature calculations: The main purpose of this block is to calculate the resultant water temperature after mixing between return water and bypass water vectors. Inputs to this block are flow fraction, temperature of water returning from the heat exchanger and temperature of water in the bypass vector. The flow rate of the resultant water vector is the summation of the two water vectors of bypass and return vector from the heat exchanger; no head loss is considered.
The resultant temperature of the water is determined by the relation:

$$T_{res} = (T_{he} * m_{w\_he} + T_{bypass} * m_{w\_bypass}) / (m_{w\_he} + m_{w\_bypass}) \qquad (35)$$

T_{he} = Temperature of water returning from heat exchanger (C), T_{bypass} = Temperature of water in bypass circuit (C), m_{w_he} = Flow rate of water returning from heat exchanger (kg/s), m_{w_bypass} = Flow rate of water in bypass circuit (kg/s)

Actuator

The actuating action required here is linear motion of a piston. Depending on the set-point temperature, the valve regulates the flow of water into the heat exchanger. The actuator modeled here has a transfer function :

$$F(s) = 0.005 / (3s+0.005) \qquad (36)$$

Now the inverse laplace transform of the above transfer function is :

$$f(t) = 1/600 * e^{-1/ 600*t} \qquad (37)$$

From the equation the time constant of the actuator is τ = 600 s. The time constant indicates that the response decays exponentially every 600 seconds.

Dampers :

Dampers are used for the control of airflow to maintain temperature and/or pressure. An important entity in the design of dampers is the dimensionless entity loss coefficient, C_d. Over most of the operating range, the loss coefficient is an exponential function of blade angle for both parallel and opposed blade dampers. That is,

$$C_d=k_1.e^{k2.\theta} \qquad (38)$$

Where k1 and k2 are constants, and θ is the blade angle in degrees.
Now the fraction of full flow is calculated by using the formula

$$\text{Fraction of full flow} = \sqrt{\frac{C_{do}}{f\ .\ C_d + (1-f).\ C_{do}}} \qquad (39)$$

C_{do} = loss coefficient for the wide-open damper, C_d = loss coefficient, a function of blade angle, f = open damper resistance as a fraction of the total system resistance

The values of C_{do} and f are obtained from ASHRAE fundamentals (1997). Since return and outside dampers are used in tandem, one control loop is used to control both dampers depending on the set point temperature.

Actuator :

The main task of the actuator in a damper is to precisely open or close the damper blades depending on the command from the controller. Since the action required is circular motion of the damper blades, the actuator modeled here is a motor having the transfer function,

$$F(s) = 5 / (3s+5) \qquad (40)$$

Now the inverse laplace transform of the above transfer function is :

$$f(t) = 5/3 * e^{-5/3t} \qquad (41)$$

From the equation the time constant of the actuator is $\tau = 0.6$ s. This time constant indicates that the response decays exponentially every 0.6 seconds.

Variable speed fan

The fan modeled here has one input and two outputs. The input to the fan is the air vector coming from the heat exchanger . The outputs are: Air vector going into the zone and the total power imparted by the fan on the air.

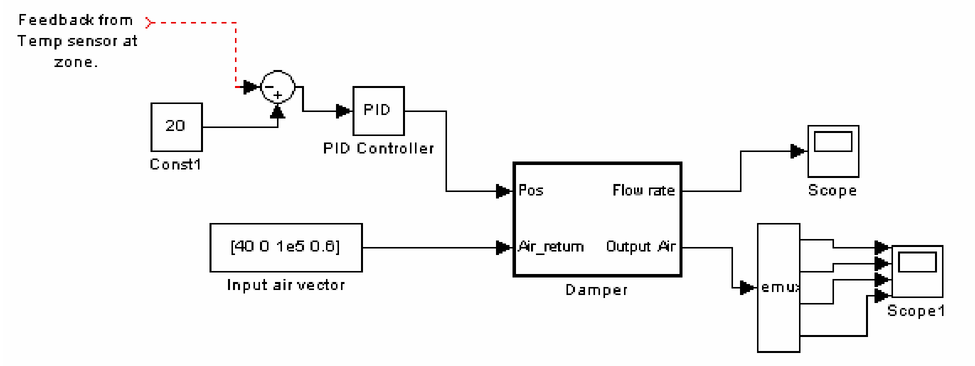

Figure 5 Damper

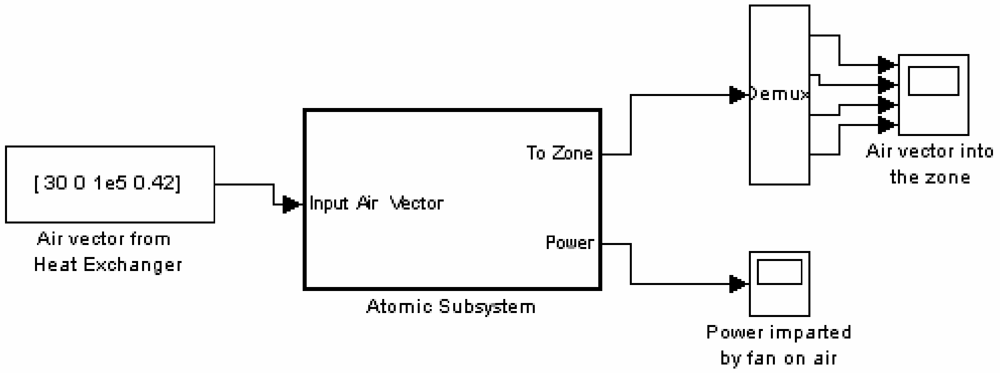

Fig 6 Variable speed fan

The fan model has two lookup tables : The relationship between volume flow and pressure loss and the relationship between volume flow and RPM. Depending on the mass flow rate of the input air vector the volume flow rate is calculated by:

$$Q = m_a / \rho \qquad (42)$$

Q = Volume flow rate (m^3/s), m_a = mass flow rate (kg/s), ρ = Density of the air (kg/m^3)

The volume flow rate thus obtained will be used to get the RPM from the second lookup table and the pressure loss from the first lookup table.
The inertial effects of the fan impeller are calculated to make the model dynamically more accurate. The value of the RPM from the lookup is used to calculate the angular velocity (radians/s).

$$\theta' = RPM * 2 * \pi / 60 \qquad (43)$$

The angular acceleration is:

$$\theta'' = \theta' / t. \qquad (radian/s^2), \text{ with t=time constant taken as 10 s} \qquad (44)$$

Now the total energy imparted by the fan to the air is given by:

$$W = Q*(P1-P2) + m * \theta'' + c * \theta' \qquad (45)$$

W = Total energy imparted by the fan on air (kW), Q = Volume flow rate (m^3/s), P1 = Set point pressure in the zone (Pa), P2 = Pressure coming into the zone after all losses (Pa), m = Mass of the fan impeller (kg), c = Coefficient of bearing friction, θ” = Angular acceleration (m/s^2), θ’ = Angular velocity (m/s)

Speed change time constant
The action required here is to change the circular motion of the fan impeller. Depending on the set-point temperature the fan regulates the volume of air flow into the zone. The motor has a transfer function:

$$F(s) = 1 / (10s+1) \qquad (46)$$

Now the inverse laplace transform of the above transfer function is :

$$f(t) = 1/10 * e^{-1/10t} \qquad (47)$$

From the equation the time constant of the actuator is τ = 10 s.
The time constant indicates that the response decays exponentially every 10 seconds

RESULTS

Steady State

The following table gives simulation test results for steady state testing.

Twi	Two	Ts	Tai	Tao	m_w	m_a	Q_a	Q_w	EB
(C)	(C)	(C)	(C)	(C)	(kg/s)	(kg/s)	(kW)	(kW)	
55	13.86	18	10.96	17.99	0.0298	0.7191	5.086	5.058	0.027
55	27.39	25	13.46	25.12	0.0829	0.8197	9.624	9.571	0.053
55	35.76	30	20.96	30.04	0.1569	0.1.38	12.66	12.62	0.03
55	38.41	35	25.37	35.00	0.2185	1.5662	15.18	15.15	0.030
55	42.27	40	30.09	39.99	0.2851	1.5249	15.2	15.17	0.031
55	46.491	45	35.97	45.00	0.3543	1.3898	12.63	12.61	0.027

Steady state energy balances were calculated using EES Software.

Dynamic testing :

Dynamic tests were performed with step changes given to a set point temperature. The temperature set point was changed from 40 ^{0}C to 50 ^{0}C .

The following graphs show the comparison between experimental data and the simulation data. In all the cases the temperature set points were 40^{0}C (initial value) and 50^{0}C (final value).

CONCLUSIONS

The primary objective of this project was to develop a Simulink model of an HVAC system to allow simulation in a software environment that will allow the study of intelligent control systems –MATLAB/Simulink are the preferred tools for studying dynamic response and control. This paper documents the models.

A second objective, achieved and reported here was the experimental verification of the models. Agreement is good. This builds confidence that users can apply the model to study control problems. The “code” is available upon request – first visit our web site at http://www.engr.colostate.edu/nnhvac/.

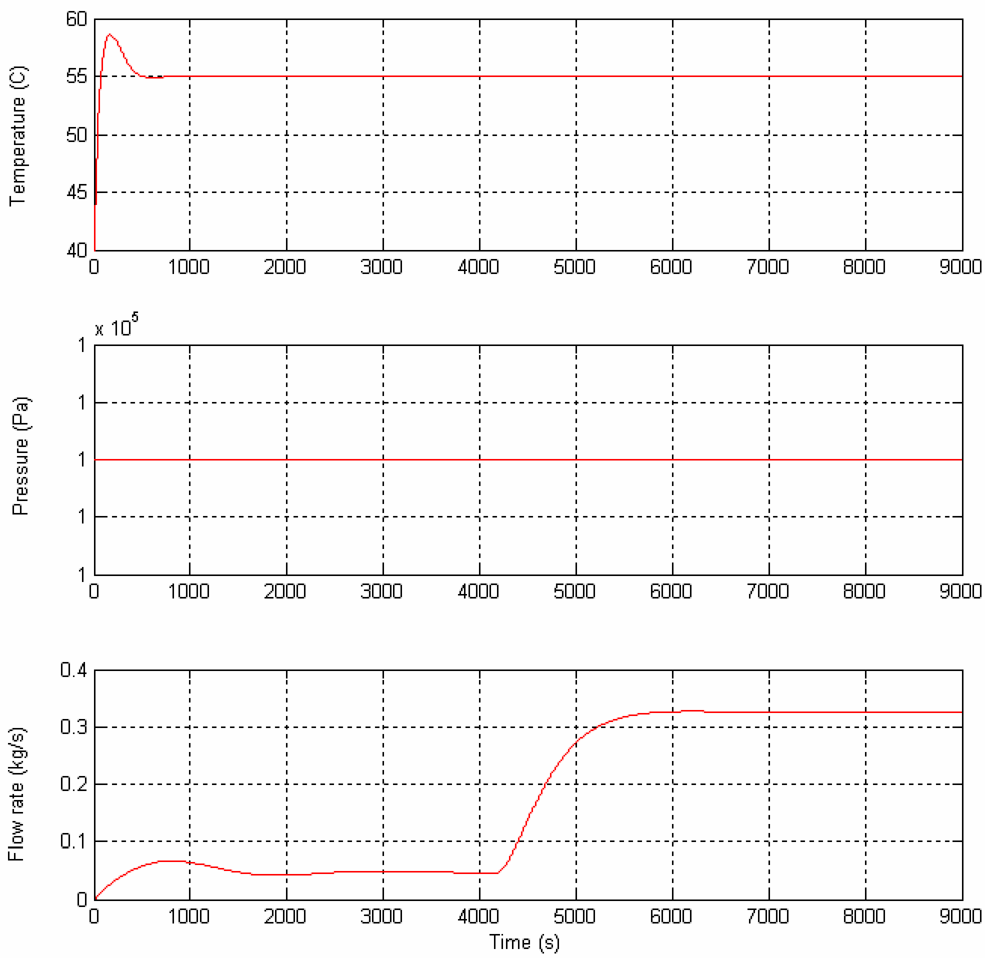

Fig 7 Water vector in main circuit of the 3-way mixing valve

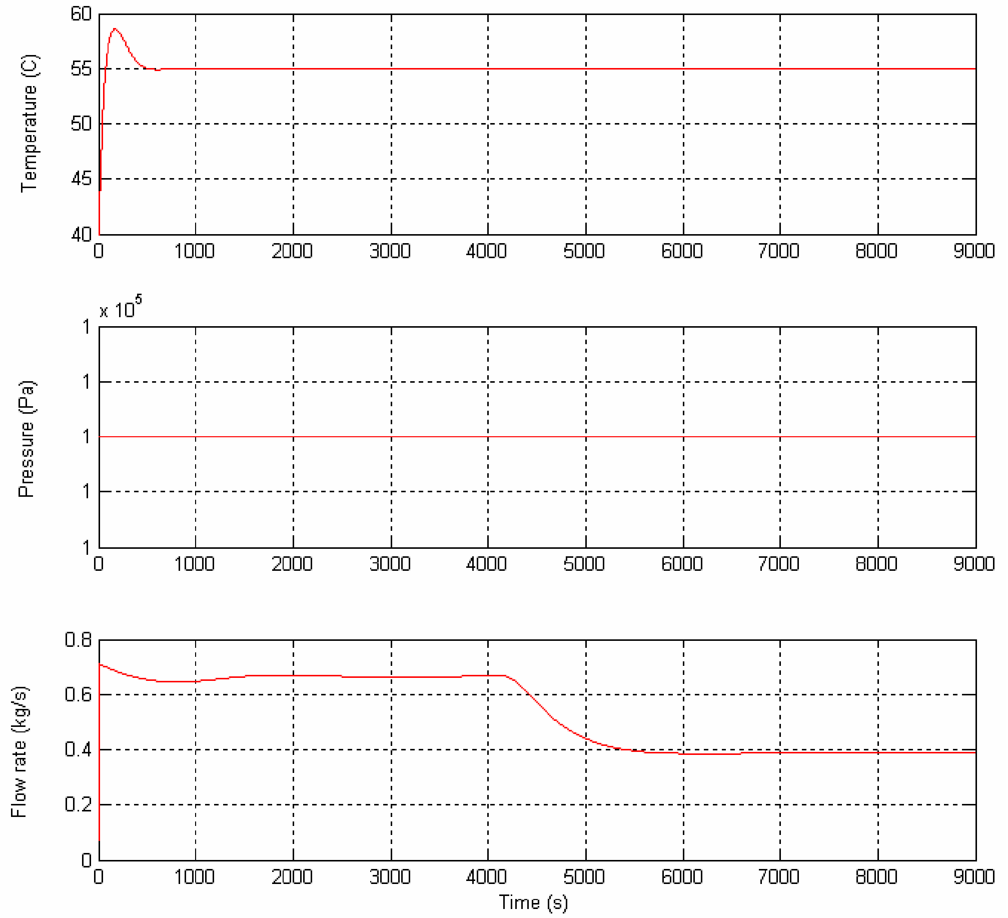

Fig 8 Water vector in the bypass circuit of 3-way mixing valve.

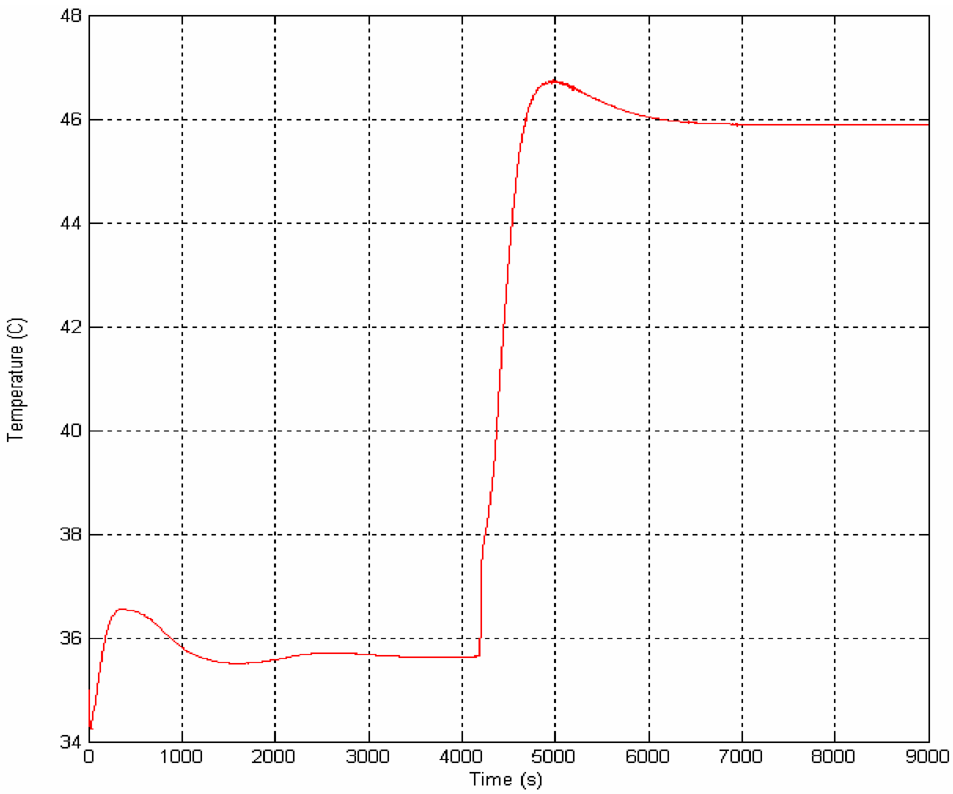

Fig 9 Temperature of air entering the heat exchanger.

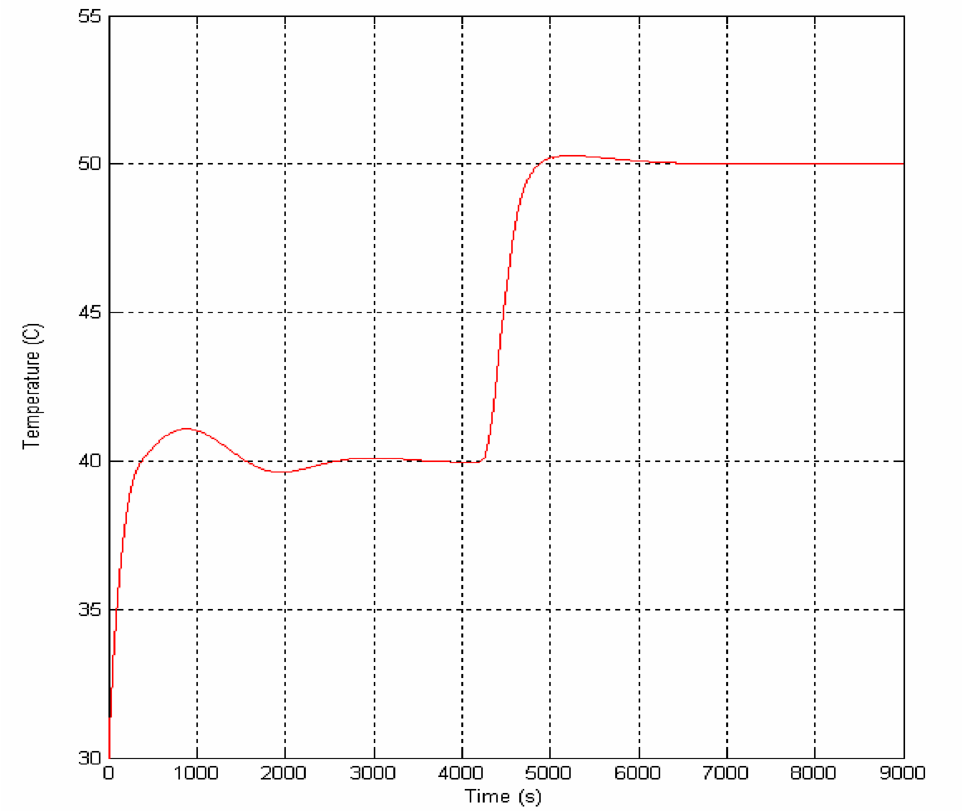

Fig 10 Temperature of air leaving the heat exchanger.

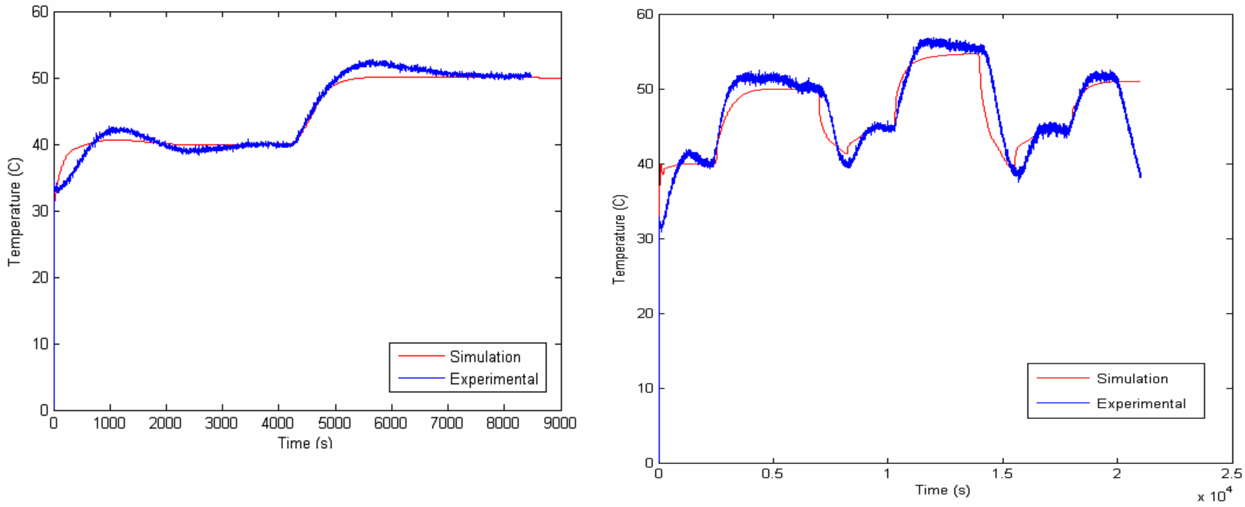

Fig 11 Graphs showing comparison between experimental data and simulation data for the same set point temperatures.

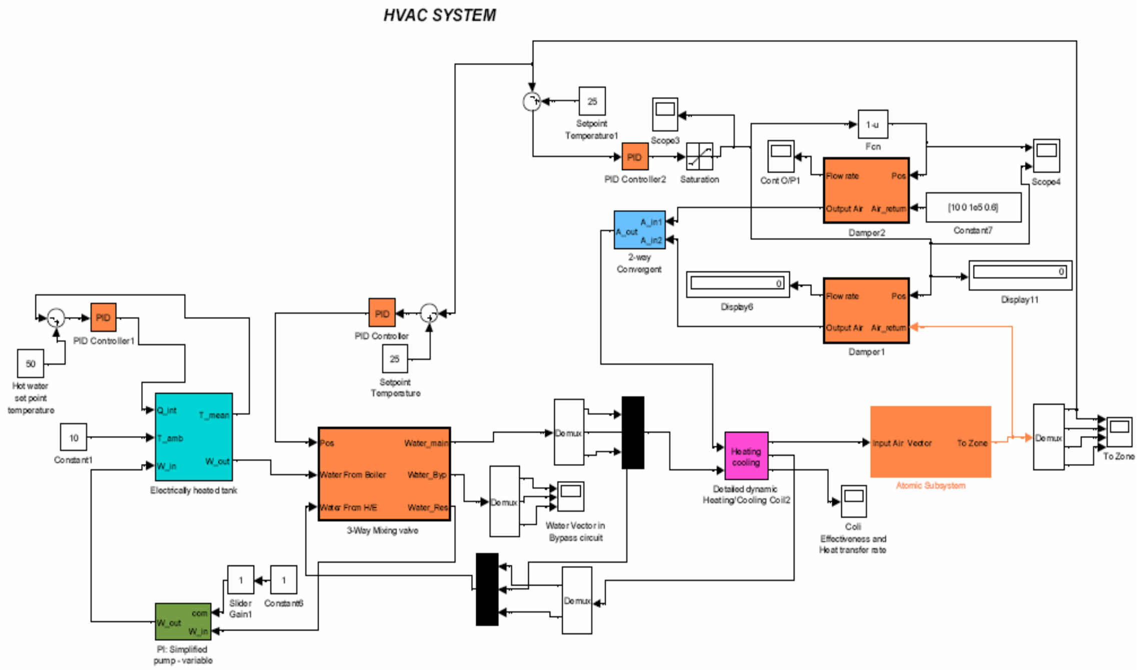

Fig 12 Figure showing simulation model of complete HVAC system

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Development of HVAC System Simulation Tool for LCEM (Life Cycle Energy Management)

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SUMMARY

The importance of Life Cycle Energy Management (LCEM) has been recognized from the view of life cycle energy saving of sustainable buildings. The purposes of this research are proposal of an LCEM framework and development of a prototype of HVAC system simulation tools for LCEM. In this paper, necessity of energy simulation tools for LCEM is discussed, and the outline and solution method of the simulation tool are shown. In addition, method of system integration using the developed tool and details of component models that comprise HVAC systems are shown.

INTRODUCTION

Everyone can easily recognize the importance of the maintenance and operation phase that consume much energy when sustainable construction is considered from the viewpoint of countermeasures against global warming. Proper management is indispensable, however, in the life cycle from the planning and design phase to the operation phase of the facility itself to realize truly appropriate maintenance.

This study is conducted to establish the concept of the basic framework of life cycle energy management (LCEM) and develop an energy simulation tool for required air-conditioning systems (hereinafter referred to as "air-conditioning system simulation tool"). This paper describes the framework of the LCEM tool, necessity of the air-conditioning system simulation tool, and the outline of the tool. In addition, it describes changing equipment characteristics into mathematical formulae, verification of mathematical formula precision, and concrete simulation tool.

LCEM EXAMINATION COMMITTEE

The LCEM Examination Committee (Chairman: Shuzo Murakami (Keio University)) was organized under the Public Buildings Association, Japan to establish the concept of the basic framework of LCEM and develop an energy simulation tool for air-conditioning systems in

2003. In this section, we summarize some achievements[1)2)] of the LCEM Examination Committee.
The LCEM Examination Committee consists of one study group and four working groups. The study group manages and adjusts the whole committee, and the working groups are in charge of the following work:

(1) Equipment Performance Survey Working Group: This group surveys the characteristics of the air-conditioning equipment and creates experimental formulae.
(2) Air-Conditioning Simulation Tool Development Working Group: This group develops the entire tool including equipment.
(3) Case Study Working Group: This group indicates concrete manner of utilization in each phase from the planning phase to the operation phase.
(4) Input Condition Setting Working Group: This group organizes and offers the thermal load conditions required for the tool.

This organization is managed through cooperation by the Land, Infrastructure and Transportation Ministry, academic experts, energy companies, consultants, construction companies, etc. The completed tool is distributed for free through the Land, Infrastructure and Transportation Ministry in principle. Distribution of the prototype tool was started for the first time in July 2006.

CURRENT SITUATION OF ENERGY MANAGEMENT

Energy management is performed mainly for the operation phase currently, but the following unsolved problems remain:

(1) Inefficient operation is performed daily using excessive equipment unsuitable for the actual status of operation in many cases.
(2) In equipment manufacturing and trial operation, it is checked mainly whether the rated specifications are satisfied. The performance in the off-peak period and the energy-saving performance throughout all periods are not checked.
(3) Operation is continued while the design intent and construction intent are not reflected in many cases.

It is estimated that the above problems are caused because the setting of energy-saving targets throughout the life cycle, corresponding control indicators, check methods, etc. are not made clear.

NECESSITY OF LCEM AND SIMULATION TOOL IN LCEM

Consistent energy management from the planning phase to the operation phase is indispensable to solve various problems related to energy management described above. And it is important to develop an energy simulation tool to achieve appropriate LCEM. It should macroscopically predict and evaluate the energy performance of the air-conditioning system, be able to grasp defects of the air-conditioning system with a rough sieve, and be easily handled by workers including operators in each phase in the life cycle. The following three points are regarded as important especially from the viewpoint of LCEM:

(1) The user can easily handle the tool, understand the contents, and modify and expand the contents.

(2) The table calculation method is simple, and calculation processes are not black box in most cases.
(3) The tool can calculate status values in the partially loaded status, and the user can understand the annual and periodical performance of each equipment/sub system.

SOLUTION USING OBJECT CELLS METHOD

The developed tool emphasizes practical utility. Accordingly, most equipment characteristics consist of experimental formulae to facilitate comparison and collating with actual equipment. The user can manipulate the GUI (graphical user interface) using only the basic functions of general-purpose spreadsheet software, and obtain the equilibrium state among equipment using the "repeated calculation" function.
The general-purpose spreadsheet software "Excel" is used for creation and solution of these programs. Macro programs are to be used only in annual calculation in principle.

The developed tool obtains the operation status values of the air-conditioning system using a solution called object cells method.

Table calculation is generally performed while mathematical formulae and numbers are input to cells on the sheet as shown on the left in Figure.1. In this technique, mathematical formulae constructing an equipment model are input to two or more cells as shown on the right in Figure1, and such cell group is handled as one piece of equipment (that is, object).

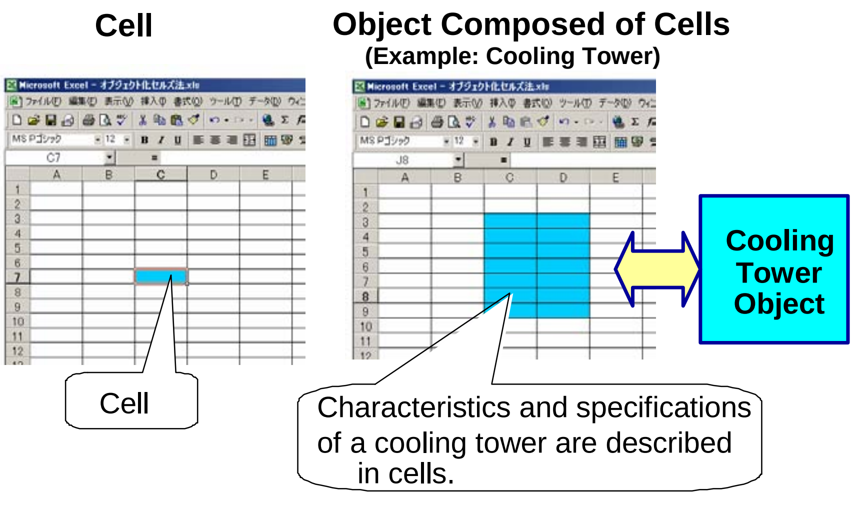

Figure 1 Cell and object consisting of cell group

CONSTRUCTION OF LCEM AIR-CONDITIONING SYSTEM

At first, prepare required objects in the object menu shown on the upper left in Figure 2. Next, "select", "copy" and "paste" required objects using the Excel functions to construct the air-conditioning system. As soon as required objects are pasted, the tool starts calculation among the objects. They are graphical operations. This form of individual, distributed, and mutual calculation without using the main program is regarded as object-oriented programming using the GUI.

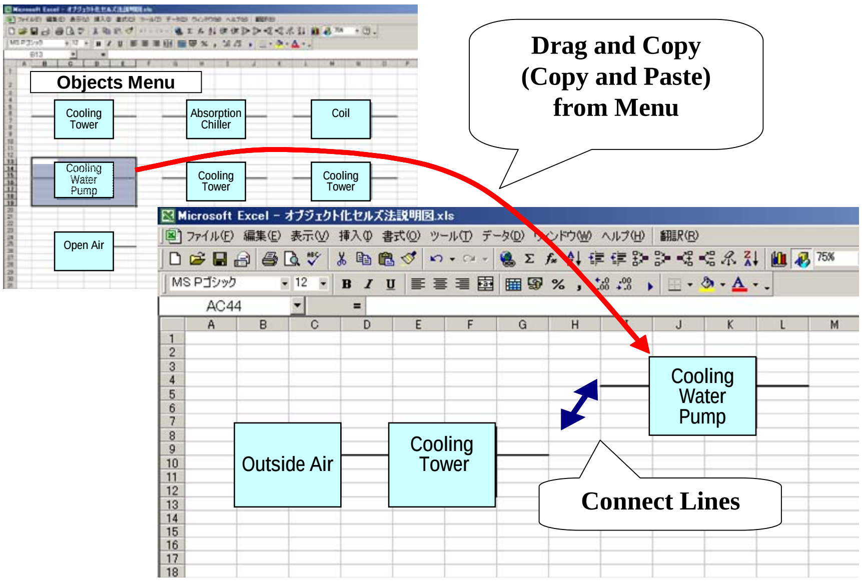

Figure 2 Object menu and system construction method

Figure 3 shows an example of objects. An object is composed of “Communication section”, “Control section”, “Method section” and “Property section”.

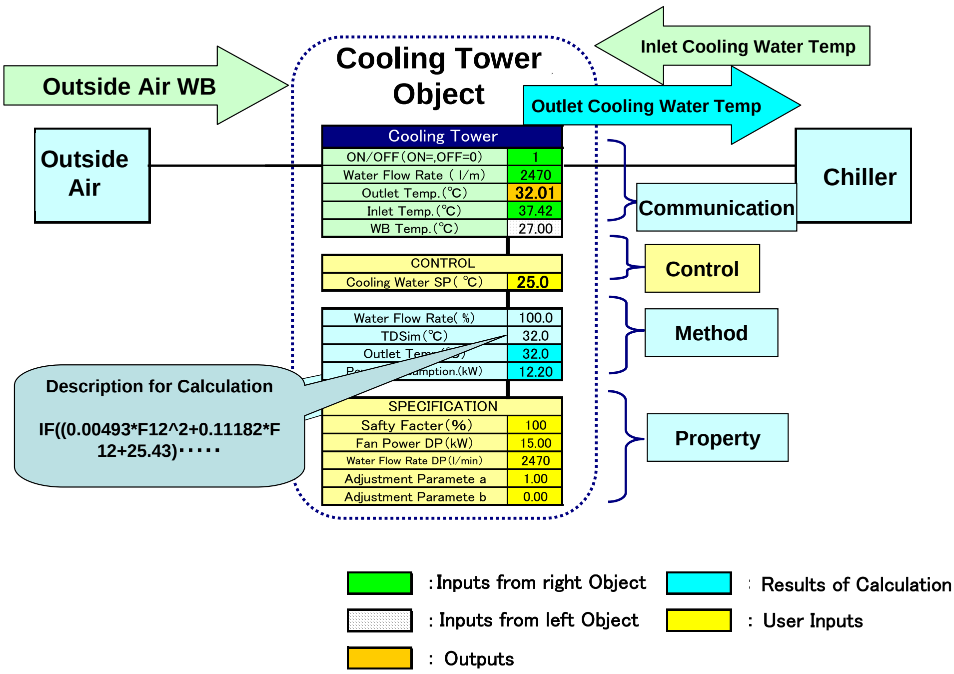

Figure 3 Example of object (Cooling Tower Object)

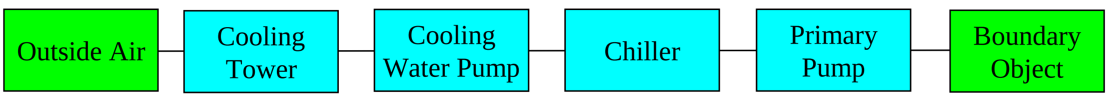

Figure 4 System construction example

Figure 4 shows a heat source subsystem where the outside air object, cooling tower object, cooling water pump object, chiller object and chilled water pump object are connected in the same way. When the outside air wet-bulb temperature and the temperature and flow rate of chilled water are given as the boundary conditions, the tool simulates the sub system.

In the case of Figure 3, the cooling water supply temperature and the cooling water return temperature are cross-referenced to each other by each cell in the table calculation. These items perform circular reference to each other, and can cause an error. However, the tool allows circular reference and offers convergent solution by repeated calculation when the user selects “Tools” -> “Options” -> “Calculation Method” in table calculation, and turns ON the “Repeated calculation” check box shown in Figure 5.

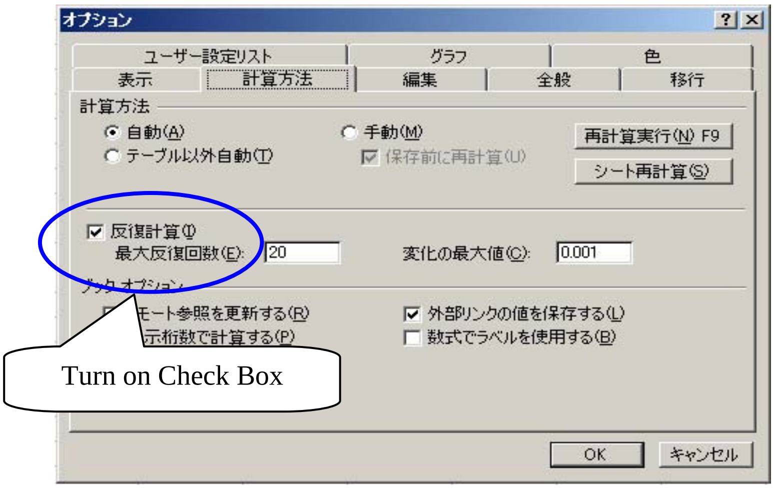

Figure 5 Circular reference and repeated calculation

All the user has to do is to connect objects when constructing a system using the object cells method. As a result, the user can easily create a large system.

DEVELOPMENT OF OBJECTS

In this section, details of some objects are described. The characteristics of the object can be changed by replacing the mathematical formulae in the objects.

Cooling tower object

Convergent calculation is required generally for the cooling tower. The developed tool, however, obtains the cooling water supply temperature T_d using an assembly of coefficients shown in quadratic expressions. The cooling tower consists of the parameters (1) to (3) below indicating the cooling performance and the parameters (4) and (5) related to operation.

(1) Cooling water return temperature (temperature of the cooling water flowing into the cooling tower)
(2) Cooling water volume
(3) Outside air wet-bulb temperature
(4) Cooling water temperature set value
(5) Cooling tower margin coefficient

In the model, the cooling water supply temperature is obtained from the experimental formula below:

$$T_d = T_{ds} \times C_2 \times C_3 \times C_4 \qquad (1)$$

$$T_{ds} = a_1 WB^2 + b_1 WB + c_1$$
$$C_2 = a_2 T_{dr}{}^2 + b_2 T_{dr} + c_2$$
$$C_3 = 1 - (1 - C_4)\frac{T_{dr} - WB}{27 - a_3}$$
$$C_4 = a_4 V_d{}^2 + b_4 V_d + c_4$$

where T_{ds} is the function of outside air wet-bulb temperature, T_{dr} is the cooling water return temperature, C_2 is the cooling water return temperature influence coefficient, C_3 is the cooling water volume influence coefficient and C_4 is the cooling water volume influence coefficient.

The cooling tower fan power ratio C_{tPW} is obtained from the formula below:

$$C_{tPW} = \frac{T_{dr} - T_{dset}}{T_{dr} - T_d} \qquad (2)$$

where T_{dset} is the set temperature when the fan is forcibly stopped, and T_d is the cooling water supply temperature in the free cooling status. Simple linear approximation is adopted here.

Figure 3 shows the cooling tower object.
The cooling tower object is located inside the dotted line frame in the center, and outside air object and cooling water pump object are connected on the both sides.
The cooling tower object is divided into the communication part, control part, method (calculation) part and property part from the top.

(1) In the communication part, the tool receives data from and sends data to adjacent objects. The user can easily create a large system only by considering connection to adjacent objects because communication is allowed only with adjacent objects.
(2) In the control part, the user determines the control condition.
In the cooling tower, the user sets the cooling water supply temperature.
(3) In the method (calculation) part, the equipment characteristics formula is written. The tool calculates the cooling water supply temperature and fan electric power using the communication, control and property data, and then displays the calculation result in the communication part.
(4) In the property part at the bottom, the user inputs the rated specification of the equipment and correction coefficients for easily correcting the characteristics formula.

Gas adsorption chiller/heater object

The object includes examination of cooling water fluctuating volume control and chilled water fluctuating volume control that are adopted recently, and offers models handling five influence coefficients - cooling water temperature, chilled water temperature, chiller load factor, cooling water volume and chilled water volume.
Products having the same characteristics are generalized. The gas consumption G_{ref} is calculated by multiplying the gas consumption ratio g_{ref} by the rated gas consumption G_{ref_r} as shown in the formula below:

$$G_{ref} = g_{ref} \times G_{ref\_r} \qquad (3)$$

The gas consumption ratio g_{ref} consists of five parameters, "C_1: Load factor q", "C_2: Cooling water return temperature T_d", "C_3: Cooling water volume ratio v_d", "C_4: Chilled water supply temperature T_c" and "C_5: Chilled water volume ratio v_c". Each parameter is 1.0 and g_{ref} is 1.0 at rating.

$$g_{ref} = C_1 \times C_2 \times C_3 \times C_4 \times C_5 \qquad (4)$$

$$C_1 = a_1 q^2 + b_1 q + c_1$$
$$C_2 = a_2 T_d^2 + b_2 T_d + c_2$$
$$C_3 = a_3 v_d^3 + b_3 v_d^2 + c_3 v_d + d_3$$
$$C_4 = a_4 T_c^2 + b_4 T_c + c_4$$
$$C_5 = a_5 v_c^2 + b_5 v_c + c_5$$

The cooling water return temperature T_{dr} is obtained analytically as shown in the formula (5) based on the fact that the exhaust heat increases or decreases in accordance with the ratio of the coefficient of performance at partial load and at rating $C_7 = COP/COP\text{-}r$. Here, T_d is the cooling water supply temperature, and V_d is the cooling water volume.

$$G_{ref} = C_6 \times C_7 \times V_d \times (T_{dr} - T_d)$$
$$T_{dr} = G_{ref} / (C_6 \times C_7 \times V_d) + T_d \qquad (5)$$

C_6: Correlation coefficient of cooling water heat quantity and gas heat quantity
C_7: Ratio of coefficient of performance at partial load and at rating

OBJECT VERIFICATION

We have verified the validity of the precision of calculation models using the air-conditioning operation data in an office building. We have confirmed that the calculation model has sufficient precision in any object in energy performance evaluation.

Cooling tower object

Figure 6 shows the calculated cooling water supply temperature in the X axis, and the measured cooling water supply temperature in the Y axis for comparison. Calculated values basically agree with measured values even if the input conditions change.

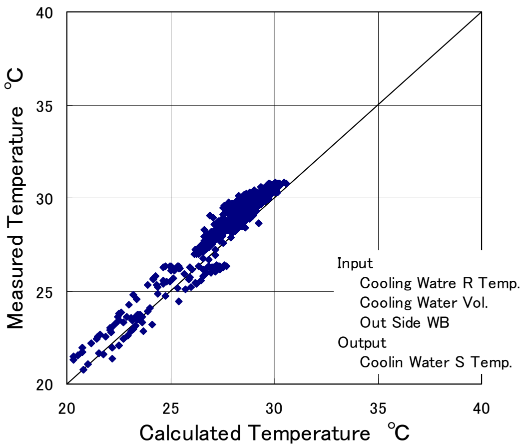

Figure 6 Cooling water supply temperature verification result

Gas adsorption chiller/heater object

Figure 7 shows the gas consumption in the chiller in the same way as the cooling tower, and Figure 8 shows the cooling water exit temperature (cooling water return temperature) from the chiller. Calculated values agree well with measured values, and the precision is sufficient in practical use.

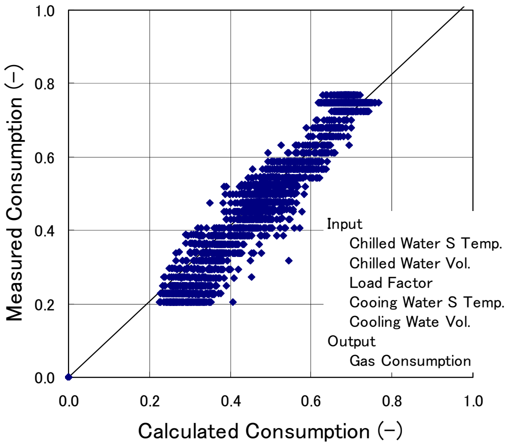

Figure 7 Gas specific-consumption verification result

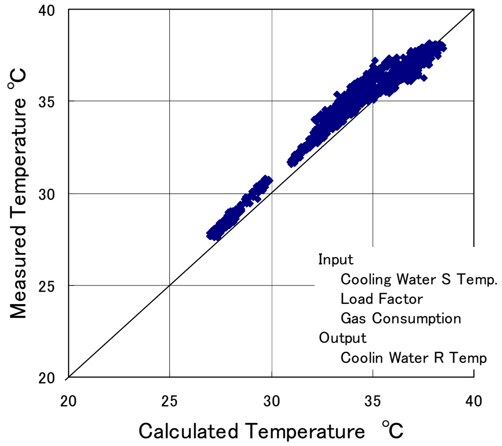

Figure 8 Cooling water return temperature verification result

CONCLUSION

We have developed an air-conditioning simulation tool that adopts the object cells method, is easily handled, and is available in every phase from planning to operation for application to life cycle energy management.

This paper indicates incorporated mathematical formulae for equipment characteristics, and verifies that the tool offers sufficient precision in energy management.

ACKNOWLEDGEMENT

We would like to thank the members of the committee for this work.

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A tool for integrated simulation to evaluate the performance of ventilation system

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SUMMARY

To properly evaluate the performance of ventilation system, the interaction among ventilation rate, indoor air-quality, and cooling/heating load should be analyzed. In this study, an integrated simulation tool based on TRNSYS will be suggested to analyze the performance of ventilation system. Some modules to calculate the changes in indoor concentration of pollutants with ventilation system and to decide the operation mode of ventilation system were newly developed. In addition, the developed modules were coupled with building load and heating/cooling simulation modules. To verify the developed integrated simulation tool, the comparison study between simulation and field study were accomplished. The simulation tool developed in this study can be used to predict the performance of ventilation system with high accuracy.

INTRODUCTION

Utilization of both healthy construction materials that produce less contaminants and ventilation system are essential to maintain the indoor air quality. Furthermore, when a ventilation system operates to improve the indoor air quality of a residential building, heating and cooling load should also be solved at the same time.

In order to comprehensively evaluate the performance of a ventilation system, experiment and field measurement over a long period of time are required. However, due to time and cost restrictions, simulation is generally used to analyze a ventilation system with heating and cooling loads.

Simulation tools are often used for assessing indoor air quality according to ventilation as well as calculating the heating/cooling load, which can be categorized both CFD and network simulations.

(1) The CFD simulation(1) is widely used due to its capability of describing in detail the generation and diffusion of contaminants from construction materials for a unit space. However, it has severe restriction for portraying reduction and discharge characteristics of contaminants over a long period of time or movement of contaminants between rooms.

(2) Conventional network simulation tools(2,3) generally focus on the movement of contaminants between rooms and changes in contaminant concentrations according to ventilation (e.g. CONTAMW, COMIS), or characteristic analysis of the building and heating/cooling loads separately (e.g. TRNSYS)(4). Therefore, conventional network simulation tools alone have a limitation in comprehensively portraying the correlated performances of ventilation system.

This study aims at developing an integrated simulation tool capable of comprehensively simulating the indoor air pollutant level, operation of the ventilation system for controlling the contaminant concentration, changes in the indoor contaminant concentration according to ventilation rate, as well as changes in the indoor thermal environment and heating/cooling load.
In this paper, the concept of the integrated simulation tool proposed in the study will be described, and the simulation tool will be validated with comparison between simulation and field measurements results. Moreover, the indoor air-quality control characteristics and heating/cooling load characteristics of ventilation system with various control logics will be discussed using the proposed integrated simulation tool.

CONCEPT OF THE INTEGRATED SIMULATION TOOL

Building, heating and cooling load simulation-TRNSYS

The integrated performance evaluation tool for ventilation system developed by this study is based on the TRNSYS(TRansient System Simulation) capable of dynamic simulations. TRNSYS was developed in 1975 by the Solar Energy Laboratory at the University of Wisconsin in the U.S. to analyze the solar energy collector, and it is currently being used in Europe as well as other parts of the world. Based on its modular structure and outstanding expandability, TRNSYS allows the user to create the mathematical modules based on the derivation from experiments or theories. Furthermore, the source codes of the basic modules are made available and can be modified so that the interaction among the system elements can be simulated in detail according to the purposes.

Development of simulation modules for describing indoor contaminant generation and contaminant removal characteristics of ventilation system

Since the basic TRNSYS does not offer the modules to simulate the changes in indoor air quality and changes in heating/cooling load according to the ventilation rate and indoor air sterilization methods(ventilation, air-cleaning), some new modules were developed to simulate the generation of a contaminant from construction materials as well as the changes in CO_2 concentration.
Indoor air quality methods can be classified as source control, removal control and dilution control. The source control means the controlling the amount of contaminant emission from the construction material. Removal control corresponds to the elimination of the contaminant in the air with air cleaning unit, and dilution control alleviates the contaminant concentration through ventilation. The developed modules can describe all the three indoor air control methods. Table 2 shows the indoor air control method adopted in this study.

Table 2 IAQ control method

mode	control method
VM1	mechanical ventilation only
VM2	mechanical ventilation with air cleaning unit
VM3	air cleaning only

CO_2 and HCHO(formaldehyde) were selected as control subjects under the considerations of the current indoor air quality regulations in Korea.

Prediction of the indoor contaminant concentration under the initial condition

In order to determine the initial concentration of the indoor pollutant prior to operation IAQ control system, contaminant generation from the construction materials or the persons in the room with the passage of time and the dilution effect due to infiltration should be considered[5,6].

Equation 1 is used for predicting the amount of CO_2 generated indoors.

$$C_{cp} = \{(C_{cf} + SC \times C_{c1}) \div VOL_z \times 10^6) \times (VOL_z - VOL_{\inf}) + C_{cout} \times VOL_{\inf}\} \div VOL_z , \qquad (1)$$

where C_{cp} : CO_2 concentration at time-step prior to ventilation, C_{cf} : CO_2 concentration at former time step prior to ventilation, C_{c1} : CO_2 emission per person, C_{cout} : CO_2 concentration in outdoor condition, S_C : presence schedule of dweller, VOL_z : air volume of the analyzed room, VOL_{inf} : air-volume induced by infiltration

Equation 2 is used for predicting HCHO concentration at each time step.

$$C_{hp} = \{(C_{hf} + S_h) \times (VOL_z - VOL_{\inf}) + C_{hout} \times VOL_{\inf}\} \div VOL_z , \qquad (2)$$

where C_{hp} : HCHO concentration at time-step prior to ventilation, C_{hf} : HCHO concentration at former time step prior to ventilation, S_h : HCHO emission at time step, C_{hout} : HCHO concentration in outdoor condition, VOL_z : air volume of the analyzed room, VOL_{inf} : air-volume induced by infiltration

Prediction of indoor contaminant elimination with ventilation mode

(1) IAQ control mode : ventilation(VM 1)

Indoor concentration levels of CO_2 and HCHO after turning on the ventilation unit can be calculated based on the assessment of the contaminant concentration before ventilation (Equation (1) and (2)) and after ventilation (Equations (3) and (4)).

$$C_{cv} = C_{cp} \times (1 - ACH_{vm}) + C_{cout} \times ACH_{vm} , \qquad (3)$$

$$C_{hv} = C_{hp} \times (1 - ACH_{vm}) + C_{hout} \times ACH_{vm} , \qquad (4)$$

where C_{cv}: CO_2 concentration after ventilation, C_{cp} : CO_2 concentration prior to ventilation, HCHO concentration at per time-step prior to ventilation, C_{cout} : CO_2 concentration in outdoor condition, C_{hv} : HCHO concentration after ventilation, C_{hp} : HCHO concentration prior to ventilation, C_{hout} : HCHO concentration in outdoor condition, ACH_{vm} : air-volume induced by ventilation system

Figure 1 shows the logic of the indoor air control module for ventilation mode proposed by this study. The operations algorithm is designed to turn on the ventilation system when either of the two contaminants (CO_2, HCHO) concentration exceeds the existing permissible level in Korea.

Since the air volume of outside air induced by ventilation system varies according to the operation mode and signal, it is reflected in ACH_{vm} of equations (3) and (4). Therefore, if the signal of the ventilation unit is off, the value of ACH_{vm} becomes 0. Moreover, the air volume of outside air introduced by infiltration or ventilation for each time-step can be reflected in ACH_{vm}. The signal of the ventilation system consists of high and low, night and off modes.

(2) IAQ control mode : ventilation + air cleaning (VM 2)

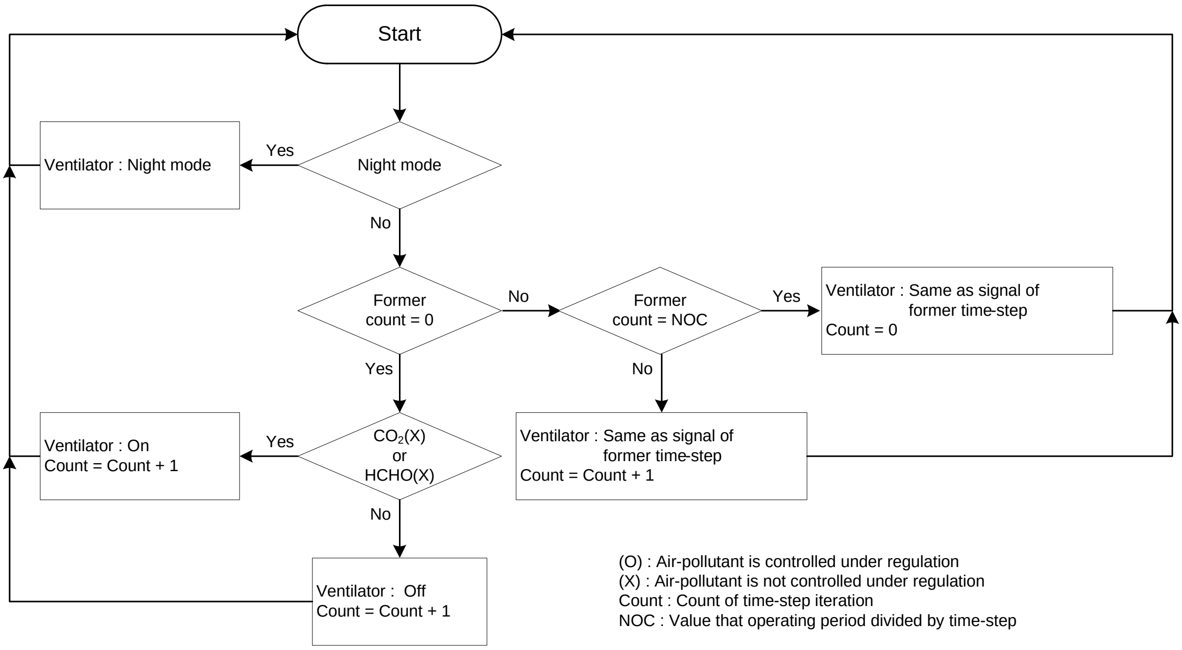


Fig. 1 Operation algorism of indoor air control (VM1, mechanical ventilation only)

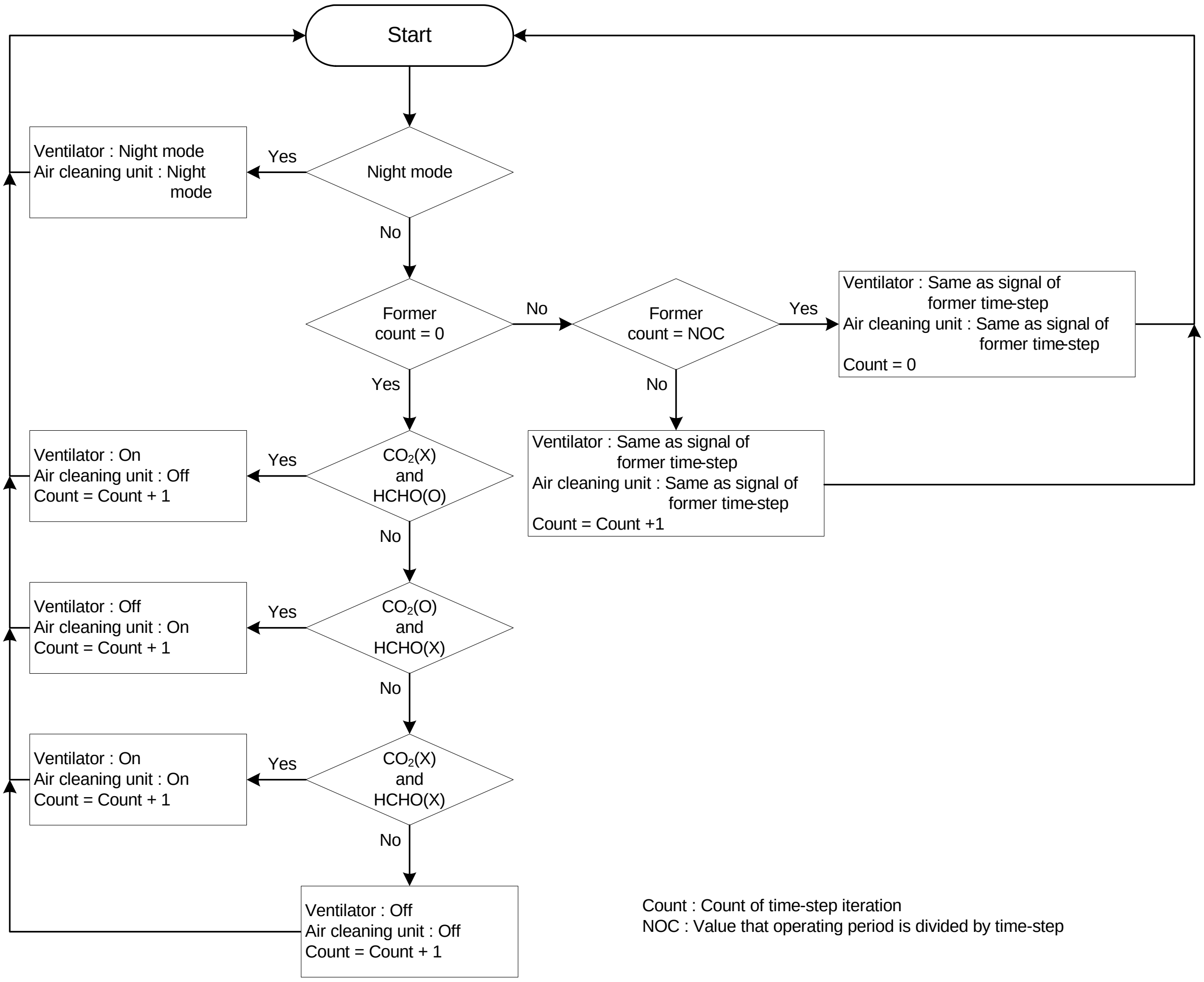


Fig. 2 Operation algorism of indoor air control (VM2, mechanical ventilation + air cleaning)

Figure 2 shows the algorithm for determining the operation signals of the ventilation and air-cleaning units. In this mode, two operation modes are operated independently or simultaneously. When HCHO concentration of the indoor air is below the permissible level and CO_2 concentration level is above the permissible level, the ventilation system will be operated. Conversely, if CO_2 level is below the permissible level and HCHO is above the permissible level, only the air-cleaning unit will be operated. If both CO_2 and HCHO concentration levels exceed the permissible level, both the ventilation and air-cleaning units will be activated.
The CO_2 concentration after the ventilation or air-cleaning unit activates can be expressed as Equation (3) and the change of HCHO concentration level can be defined as Equation (5).

$$C_{hv} = \{C_{hp} \times (1 - ACH_{vm}) + C_{hout} \times ACH_{vm}\} \times \left((1 - \eta_c) \times \frac{CHM_{fm}}{VOL_z} \right), \qquad (5)$$

where C_{hv} : HCHO concentration after ventilation, C_{hp} : HCHO concentration prior to ventilation, C_{hout} : HCHO concentration in outdoor condition, ACH_{vm} : air-volume induced by ventilation system, η_c : filtering efficiency of air-cleaning unit within one time circulation

HCHO is diluted when the ventilation unit is operating, and removed in case of an air-cleaning unit is operated. Consequently when the ventilation unit is off, the HCHO concentration will be controlled by the air-cleaning unit. In other words, when only the air-cleaning unit is operating, it does not affect the indoor CO_2 concentration.
The removal efficiency of the filter to evaluate the contaminant removal performance of the air- cleaning unit is generally defined as the ratio of contaminant concentrations in the initial generation state to contaminant concentration level in certain period after the generation stops and the air-cleaning unit operates[7]. However, contaminants are constantly generated indoors in an actual condition, and the contaminants are removed continuously when the air-cleaner is operating. Hence the concept of one-pass removal efficiency is often used in numerical simulation. One-pass removal efficiency is the ratio of the difference between the inflow gas concentration (S1) and outflow gas concentration (S2) divided by the inflow gas concentration (S1) as shown in Fig. 3.

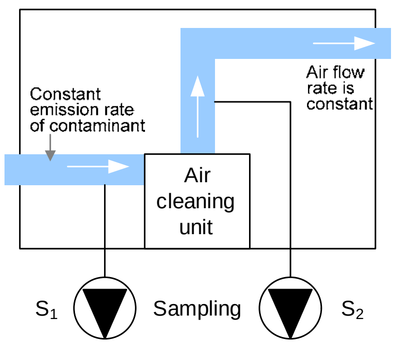


Fig. 3 Schematic of one-pass removal efficiency of air cleaning unit

(3) IAQ control method：Air-Cleaning(VM 3)
When only the air-cleaning unit is operated, the ventilation unit signal is turned off and the control algorism of VM3 is identical to the VM2 except for fixing the airflow of the ventilation unit (ACH_{vm}) at 0.

Validation

In order to examine the validity of the developed simulation modules for predicting the contaminant removal characteristics of IAQ control system, the comparison analysis between the field measurement of the ventilation system and the prediction with the developed module were accomplished. Based on the results of comparison study, the logics of the developed module were modified.

(1) Field measurement

The ventilation system was installed in an actual multi-residential house to practice performance evaluation field tests. The analyzed house was newly built and elapsed 3 month after the completion. No one was occupied in the analyzed house at that time. As shown in Fig. 4, the analyzed space is limited as living room, kitchen and dinning room. And all the exits to adjoining rooms were sealed off to eliminate air flow. A ventilation system with a rated capacity shown in Table 3 was installed under the ceiling of the living room. After activating each operation mode of the ventilation system such as VM1(ventilation), VM2(ventilation+air-cleaning), VM3(air-cleaning only), the changes in indoor contaminant concentration was measured according to the standard test method for indoor air-quality suggested by MOE (Ministry of Environment Republic of Korea). HCHO emission rate was measured two times every day. First measurement was conducted 30 minutes prior to operation of ventilation system and 30 minutes after quitting the ventilation system. The measurement condition and results are shown in Table 4, Table 5, respectively.

Fig. 4 Analyzed house

Table 3 Air-flow rate of ventilation system

Ventilator	Air cleaning unit
70[CMH]	180[CMH]

*One-pass removal efficiency of air cleaning unit is set at 0.35(35%).

Table 4 Measurement conditions

Outdoor condition	Dry B. Tem. : 10～15℃, Relative Humi. : 22～36%
Indoor condition	Dry B. Tem. : 25℃, Relative Humi. : 30%
Infiltration rate	0.2[ACH]
Outdoor concentration	HCHO : 0.036[mg/m^3], CO_2 : 350[PPM]

Table 5 Measurement results

Operation mode	Concentration(HCHO) Before operating(Base concentration) [mg/m^3]	Concentration(HCHO) After operating [mg/m^3]
Ventilator only	0.236	0.083
Ventilator and air cleaning unit	0.224	0.062
Air cleaning unit	0.367	0.113
Indoor emission rate of HCHO	0.00812[mg/hr・m^3]	

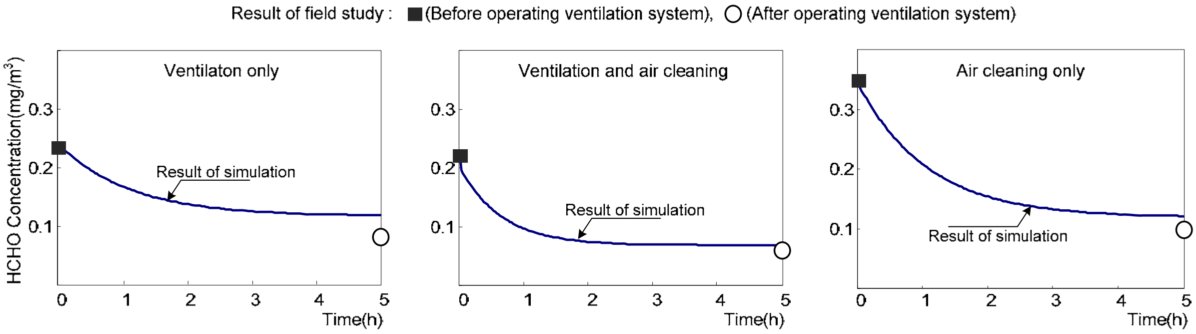


Fig. 5 Comparison between field measurement results and simulation results

(2) Validation of simulation modules

The comparison simulation for evaluating the validity of developed simulation modules were accomplished based on the field measurement conditions and results.

Figure 5 displays the results of the field measurement and numerical simulation. As shown in Fig. 5, the simulation results have a good agreement with the measurement results for the the ventilation+air-cleaning and air-cleaning mode. Meanwhile, in case of ventilation only mode, both results are somewhat different, however, the difference can be regarded as insignificant when the field measurement errors are considered.

In order to perform more accurate comparison and verification, the contaminant concentration should be measured with a shorter interval than this time field measurement.

Integrated simulation tool to predict the performance of ventilation system

In order to predict the performance of ventilation system such as indoor air control characteristics and building heating and cooling load effect, the integrated simulation method was suggested in this study. The integrated simulation method is based on the TRNSYS and the developed IAQ control module is added to make up for the limitation of TRNSYS.

Fig. 6 displays the layout of integrated simulation tool embedded in TRNSYS. Fig. 7 shows the information flow in the integrated simulation. Basic TRNSYS simulation module deals with outdoor air characteristics, indoor thermal environment, building load, heating/cooling system and energy. The new modules such as contaminant concentration prediction module, ventilation system signal and ventilation rate determinant module, heat recovery efficiency calculation module, and the module to link the ventilation system and the heating/cooling system were added in the integrated simulation.

A total of seven cases are analyzed in this study (see Table 7). Case 1 shows the results for the air current between rooms occurred by infiltration only, and Case 2 and 3 corresponds to the results for the ventilation system installed in living room and whole room respectively. Here, with the state of door opened and closed, the cases are divided into two sub-cases. The cases with -1 show the condition that the door between the rooms is opened. -2 represent the cases which the door is closed. Table 4 shows the air flow rate of ventilation system that is installed in each room. In whole rooms, the air change rate is set at 0.9times/h constantly. It is based on the results of measurement.

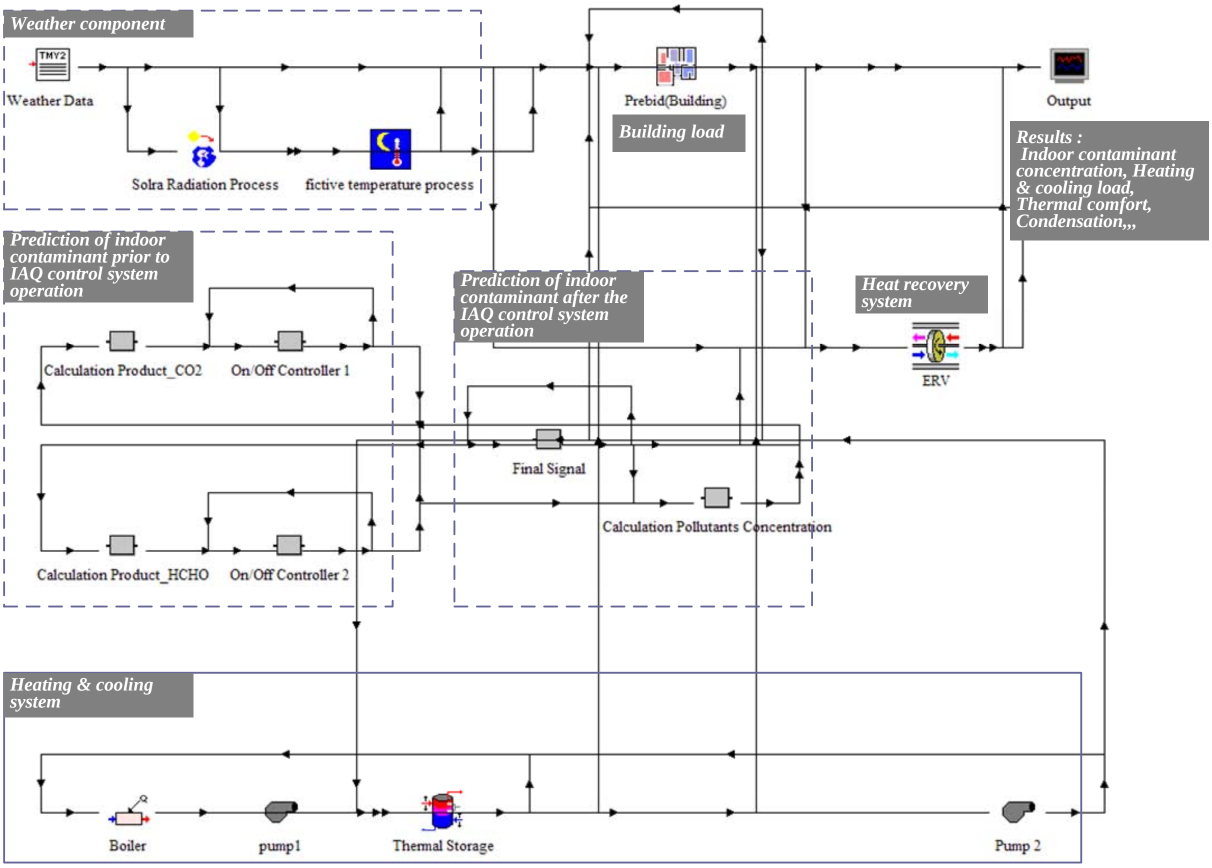


Fig. 6 Layout of the integrated simulation tool embedded in TRNSYS

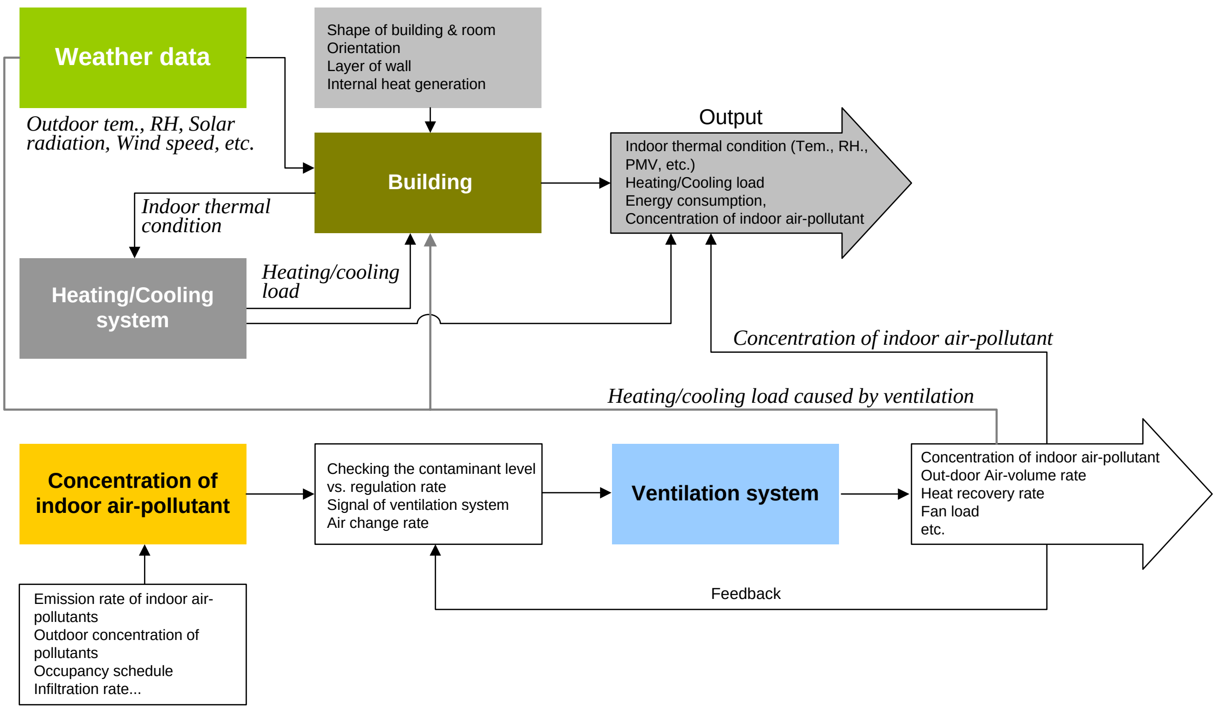


Fig. 7 Schematic of integrated simulation tool

CASE STUDY

In order to demonstrate the usefulness of integrated simulation tool, fairly detailed case study will be presented in this section.

Detailed simulation conditions

The simulation conditions such as analyzed space, infiltration rate, indoor contaminant emission rate, outdoor air conditions and the ventilation system operation conditions were identical to the field measurement conditions and results. The efficiency of heat exchange unit installed in ventilation system is about 70% in the cooling mode and 75% in the heating mode. The latent heat exchange efficiency is about 40% for both heating, and cooling under the airflow is set at 120 CMH. Moreover, the movements of contaminant between rooms were not considered for simulation as like the field measurement. The time-step for maintaining the on/off state of ventilation system was set at 30 minutes, which is identical to the field measurement condition, and the time-step for simulation was set at 1 minute 30 seconds. The detailed conditions and cases for integrated simulation are illustrated in Table 6, 7, respectively.
A total of seven cases are analyzed in this study (see Table 7). Case 1 corresponds to the present regulation of ventilation rate for residential building in Korea. And the Case 2 represents the installed ventilation system in field measurement in this study. Case 3 corresponds to the cases that satisfy the regulation level of both indoor CO_2 and HCHO in Korea. In Table 7, the cases with C show the constant air flow ventilation mode and F represent the feed-back control which the ventilation system is operated when the CO_2 or HCHO level is exceed the regulation level with operating time step (30 minutes).

RESULTS

Indoor Air Quality Control

Among the analyzed cases, only the case 3 satisfy the minimum requirements for CO_2 and HCHO level in Korea when the ventilation unit operates alone for 24 hours. In other words, the CO_2 and HCHO level will be controlled under the minimum requirements only if the air change rate should be 1.0 times/hour or more (Fig.8(a)). Moreover, even if the operation mode is set at feedback control as in Case 2-2 and Case 3-2, the ventilation system was actually operated in 24 hour constant air-flow mode. However, when the ventilation and air-cleaning units were simultaneously operated with 0.9 times/hour of 24 hour constant air-flow mode(Case 2-3) the two contaminants below the permissible level. In Case2-4 where feedback control was applied, there were some times when concentration rose above the permissible level with the ventilation system off, but the overall control characteristics were determined to be positive(Fig.8(b)).

Heating and Cooling Load

Among the analyzed cases, heating and cooling load are investigated only if the cases 3-1 and 2-4 since they satisfied the indoor air quality control guideline. Three cases were examined: Case 3-1(ventilation unit alone (1.4 ACH)), 24 hour constant air-flow), Case 3-1 with heat exchange unit, and Case 2-4(ventilation unit + air-cleaning unit (0.9 ACH, feedback control).
Indoor and outdoor conditions were identical for all three cases, and the difference among the loads for the three cases as shown in Fig. 9 can be considered as the ventilation load.
As shown in Fig. 9, the difference in the cooling load according to the ventilation can be regarded as minimal. However, in case of the heating load, when the heat recovery unit was

added there was about 75% reduction compared to the results of Case 3-1. Moreover, in case of the heat recovery unit was added to the Case 2-4, there were heating load reductions of 85% and 35% from Case 3-1 where there was no heat recovery unit and Case 3-1 + heat recovery unit, respectively. These results suggest that operating the ventilation system with air-cleaning unit can reduce the heating and cooling load as well as the amount of ventilation required for controlling the indoor air quality.

Table 6 Simulation conditions

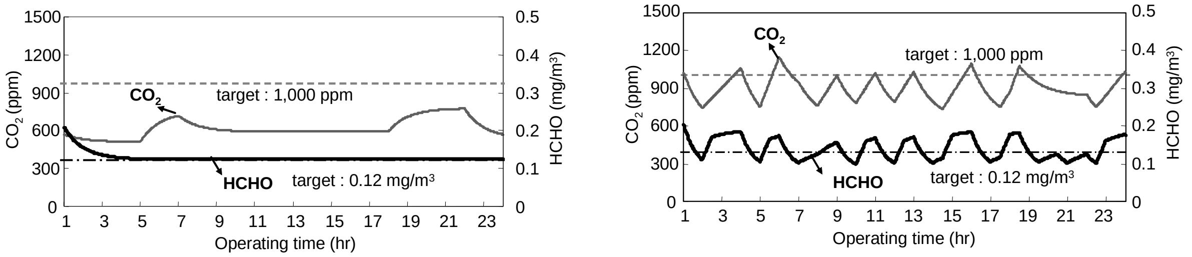
Weather data	Standard weather data(Seoul)	
Set-point	Cooling	Heating
	Dry B. Tem. : 26℃ Relative Humi. : 60%	Dry B. Tem. : 20℃ Relative Humi. : 40%
Period	June～September	December～March
Internal heat generation	Human : sensible 65[W/person], latent 55[W/person] Lighting : 20[W/m^2] , Appliance : 20[W/m^2]	
Contaminant concentration rate	CO_2 : indoor 0.021[m^3/hr·person], outdoor 350[ppm] ** HCHO : indoor 0.21[mg/m^3] **, outdoor 0.036[mg/m^3] **	

*Schedules of internal heat generation are set.
** Field measurement results
*** Regulation level of indoor CO_2 and HCHO level in Korea are set at 1000[ppm] and 0.12[mg/m^3] respectively

Table 7 Simulation cases

	ACH	Ventilation method	Air flow control
Case 1	0.7*	Ventilation only	C
Case 2-1	0.9	Ventilation only	C
Case 2-2	0.9	Ventilation only	F
Case 2-3	0.9	Ventilation＋air cleaning	C
Case 2-4	0.9**	Ventilation＋air cleaning	F
Case 3-1	1.0***	Ventilation only	C
Case 3-2	1.0	Ventilation only	F

C : constant airflow, F : feedback control means that ventilation system is operated when the indoor CO2 or HCHO level exceed regulation level in Korea.
*Present recommendatory ventilation rate in residential building in Korea.
**Air change rate that IAQ was maintained under regulations by ventilation＋air-cleaning mode with feedback control (simulation result).
***Air change rate that IAQ was maintained under regulations by ventilation mode with 24 hour constant air flow (simulation result).



(a) 1.0 ACH Constant Airflow (Case 3-1)

(b) Ventilation (0.9 ACH) + Air-cleaning: feedback control (Case 2-4)

Fig. 8 IAQ control results

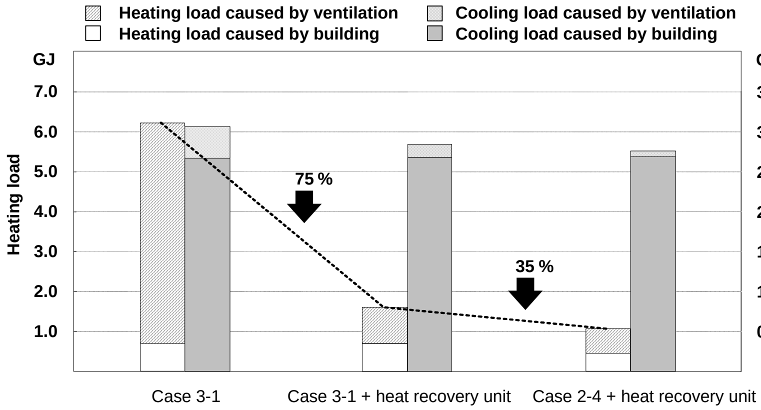


Fig. 9 Heating and cooling load

CONCLUSION

In this study, integrated simulation was proposed to evaluate the performance of ventilation system. The developed tool is based on TRNSYS and integrates the indoor air quality control characteristics for each operation mode and the changes in the heating &cooling load according to ventilation system operation.
The validity of the developed simulation module was verified by comparing the results of the field measurement of the air quality control performance of the ventilation system and the results predicted by IAQ control module of ventilation system developed from this study. In addition, an integral simulation tool was developed for simultaneously examining the indoor air quality control characteristics and heating and cooling load characteristics according to different ventilation methods.
Integrated simulation tool for evaluating the performance of ventilation system that considers the indoor air flow and contaminant concentration changes between rooms will also be presented in the near future.

ACKNOWLEDGEMENTS

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Whole Building CFD Simulation of a Swedish Low-Energy Building

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SUMMARY

When taking measures to increase energy effectiveness it is important not to deteriorate the indoor climate. In the present study a Computational Fluid Dynamic (CFD) model was built to investigate temperature and airflow patterns in a Swedish low-energy building. A full-scale model of the house has been generated in the commercial code Icepak 4.1. The commercial CFD code Fluent 6.2.16 is used for the numerical solution. The RNG k-ε model is used as turbulence model. Four cases were simulated representing Winter, Summer and Autumn climatic conditions, the last with both low and high airflow rates. The results are compared with measured air temperatures and velocities. The results show small temperature gradients in the rooms and small temperature differences between floor levels. The air velocities are generally less than 0.15 m/s in the whole building.

INTRODUCTION

The energy demand of the built environment could be decreased either by increasing the effectiveness of the external power or by taking efficient measures on the demand side, e.g. minimizing the heat losses through walls and windows, or by ventilation. In Gothenburg, Sweden, twenty terraced low-energy houses were built with the aim of minimizing the losses. When taking measures to increase energy effectiveness, the indoor climate must not be deteriorated. Earlier measurements made on the object of this study emphasized the thermal environment [1], [2]. Building energy simulation has earlier been used to predict the thermal comfort in the building [3] but this tool does not give a proper description of the airflow and temperature pattern within specified zones. This paper aims to describe a model of a low-energy building and use the outcome for whole-building Computational Fluid Dynamic (CFD) simulations.

Studied object

The object of this study consists of a ground floor, an upper floor and an attic with a total floor area of 120 m^2 and total volume of 340 m^3. A detailed description about the object can be found in Karlsson, 2006 [4]. The ground floor consists of a kitchen, a living room and a toilet. On the upper floor there are three bedrooms with high ceilings, and a washing room with a toilet (Figure 1). Both floor levels have contact with the attic through the stairwell. The houses are well insulated with low *U*-values and low-emissivity windows and utilize an air-to-air heat exchanger to minimize the heat losses due to ventilation. The supply air terminal devices are placed in the three bedrooms and in the living room. The exhaust air terminal devices are placed in the toilets on both floor levels. The ventilation duct is mounted in the framing of joists, causing the supply air terminal devices on the upper floor (e.g. the bedrooms) to be placed on the floor but near the ceiling on the ground floor. One of the twenty houses was used as a test house during the first two years of occupation. Measurements from this house have been used in this paper.

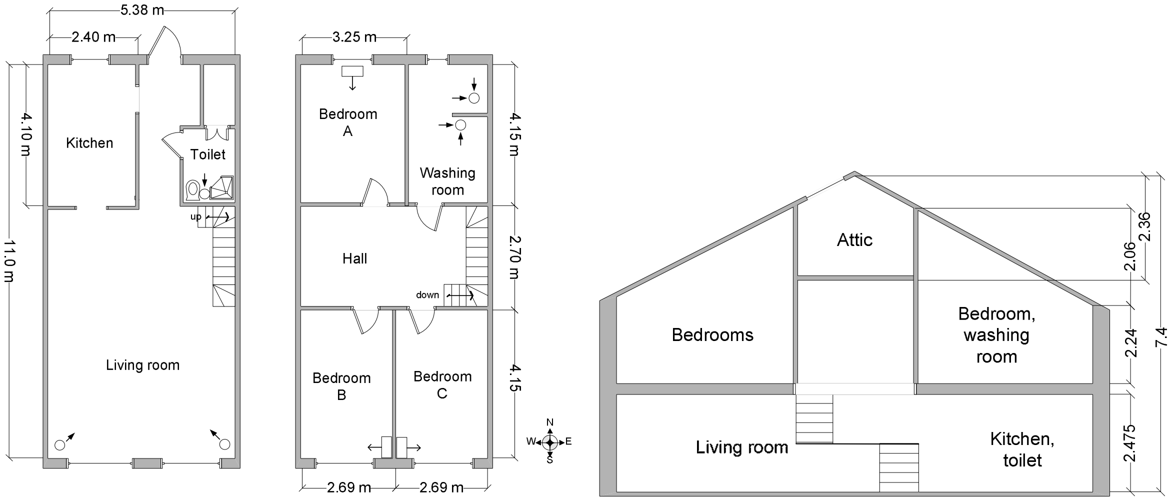

Figure 1. Ground and upper floors of the terrace house.

MATHEMATICAL MODELLING

A steady state three-dimensional model is considered for analyzing the flow and heat transfer in the whole building. The flow is assumed to be turbulent. The following assumptions are also considered: thermo-physical properties of the air are assumed to remain constant except for the buoyancy term of the *y*-momentum equation, i.e. the Boussinesq approximation, see Gray and Giorgini [5]. The radiation heat transfer is included. The radiative surfaces are assumed to be grey-diffuse with constant emissivity and the participating fluid, i.e. air, does not interact with the radiative process directly. Based on the above assumptions and using The Renormalization Group k-ε model (RNG) for the turbulence modeling the governing equations are obtained as follows:

$$\frac{\partial U_i}{\partial x_i} = 0\,, \tag{1}$$

$$\frac{\partial (U_j U_i)}{\partial x_j} = -\frac{1}{\rho}\frac{\partial p}{\partial x_i} + \nu\nabla^2 U_i + \frac{\partial}{\partial x_j}\left(2\nu_t S_{ij} - \frac{2}{3}k\delta_{ij}\right) + g_i\beta(T_0 - T)\,, \tag{2}$$

$$\frac{\partial (U_j T)}{\partial x_j} = \nabla\cdot[(\alpha+\alpha_t)\nabla T]$$

$$\frac{\partial (U_j k)}{\partial x_j} = \nabla\cdot\left[\left(\nu + \frac{\nu_t}{\sigma_k}\right)\nabla k\right] + \nu_t S^2 + \beta g_i \alpha_t \frac{\partial T}{\partial x_i} - \varepsilon\,, \tag{4}$$

$$\frac{\partial (U_j \varepsilon)}{\partial x_j} = \nabla\cdot\left[\left(\nu + \frac{\nu_t}{\sigma_\varepsilon}\right)\nabla \varepsilon\right] + C_{\varepsilon1}\frac{\varepsilon}{k}\nu_t S^2 - C^*_{\varepsilon2}\frac{\varepsilon^2}{k} + C_{\varepsilon3}\left(\beta g_i\alpha_t\frac{\partial T}{\partial x_i}\right)\frac{\varepsilon}{k}\,, \tag{5}$$

where $\eta = Sk/\varepsilon$, $\eta_0 = 4.38$, $\beta = 0.012$, $C_\mu = 0.0845$, $C_{\varepsilon1} = 1.42$, $C_{\varepsilon2} = 1.68$, $\sigma_\kappa = \sigma_\varepsilon = 0.7178$, $\mathrm{Pr}_t = 0.9$, $\alpha_t = \nu_t/\mathrm{Pr}_t$, $Sk/\varepsilon C^*_{\varepsilon2} = C_{\varepsilon2} + \frac{C_\mu\eta^3(1-\eta/\eta_0)}{1+\beta\eta^3}$, $S_{ij} = 0.5\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right)$, and $S = \sqrt{2S_{ij}S_{ij}}$.

The radiation model used in the present study is the Discrete Ordinates (DO) radiation model, which takes into account the heat transfer exchanges between different objects by solving a so-called a radiative transfer equation (RTE) for a finite number of discrete solid angles, each associated with a vector direction fixed in the global Cartesian system (x, y, z). The solution method used for RTEs is the same as the flow and energy equations; for more information see Fluent 6.2.16 [6]. Due to the size of the model, the RTEs are solved once every ten iterations in order to reduce the impact on the available computational resources and the CPU time.

Geometry and boundary conditions

The CFD model of the house is built to resemble the real object as closely as possible. However, the stairwell is not included in the CFD model to simplify the mesh generation. The test house was only sparsely furnished during the measuring period, as is the CFD model. All doors, except the doors to the toilet and the toilet/washing room, are assumed to be open and are removed in the model. A small gap of 5 cm is left below the toilet doors to allow airflow into the toilets.

The heat fluxes through the building structure are calculated based on measured climatic data and designed U-values adjusted for no convection heat transfer at the inside surface. The solar irradiation is included in the model as a heat flux applied to the floor surface in the living room and the two small bedrooms (B and C) upstairs. The average absorbed insolation over two days is used to consider the thermal inertia of the building. Table 1 summarizes the heat fluxes used for the simulations. The no-slip condition is used at the surfaces, which implies that the velocity vanishes at the surfaces. For near wall-treatment the standard wall function proposed by Launder and Spalding [7] is used. The radiation surfaces are assumed to be grey-diffuse with a constant emissivity equal to 0.9 except for the supply and discharge boundaries where the emissivity is set to 1.0.

Table 1. Heat fluxes applied to surfaces in the model for the simulated cases.

	Heat flux [W/m^2]		
Construction	*Winter case*	*Summer case*	*Autumn case**
External walls	-0.78	-0.52	-0.78
Main door	-7.21	-4.8	-7.21
Windows	-7.17	-4.78	-7.17
Floor in the kitchen and entrance hall	-1.3	-1.86	-1.3
Floor in living room	0.7	-0.84	0.7
Floor bedrooms B & C	3.3	4.9	3.3
Ceiling	-0.6	-0.4	-0.6

**The boundary conditions were used for both low and high airflow rate cases.*

In the living room and bedroom B, respectively, a human representative is standing that emits 55.6 W. In the living room there is a TV represented by a block that emits 50 W. On the upper floor, a 40 W heat source is applied in bedrooms A and C. Further, in the kitchen a heat gain of 120 W representing a fridge and a freezer is present.

The airflow through the devices used in the simulation is based on measured results [2], found in Table 2. The supply temperature is changed between 19°C in the Summer case to 21ºC in the Winter case. The Autumn case is simulated for both low and high airflow rate levels, while for the Winter and Summer cases the low airflow rate level is considered. The inlet turbulence intensity is assumed to be 10% and is applied to all supply terminals. Those values are implemented in the model using a constant airflow with a specified temperature and a uniform velocity profile normal to the supply device. A zero-gradient boundary condition is applied at the exhaust, assuming similar conditions in the flow direction (i.e. a fully developed flow). The outlets in the washing room on the upper floor are defined as an outflow.

Table 2. Airflow rate through supply and exhaust devices used as boundary conditions in the simulation model.

Device	Velocity [m/s], low airflow rate	Velocity [m/s], high airflow rate
Supply device in bedroom *A*	0.19	0.28
Supply device in bedroom *B*	0.10	0.15
Supply device in bedroom *C*	0.16	0.24
Supply devices in living room	0.88	1.32
Exhaust device in toilet on the ground floor	2.81	4.22
Outer exhaust device in toilet on the upper floor	-	
Inner exhaust device in washing room	-	-

Computational grids

ICE Pack 4.16 is used to generate the 3-D unstructured grid. Different grid densities are checked and finally the model composed of 694,544 hexahedral cells is found to be sufficient for the present study, as a trade-off between accuracy and efficiency. The grid distribution is effectively controlled by clustering the mesh towards the walls and edges in such a way that the wall function can be applied properly, see Figure 2. With this grid, it takes about 30 hours of CPU-time on a desktop computer with a 2.66 GHz processor to get a fully convergent result.

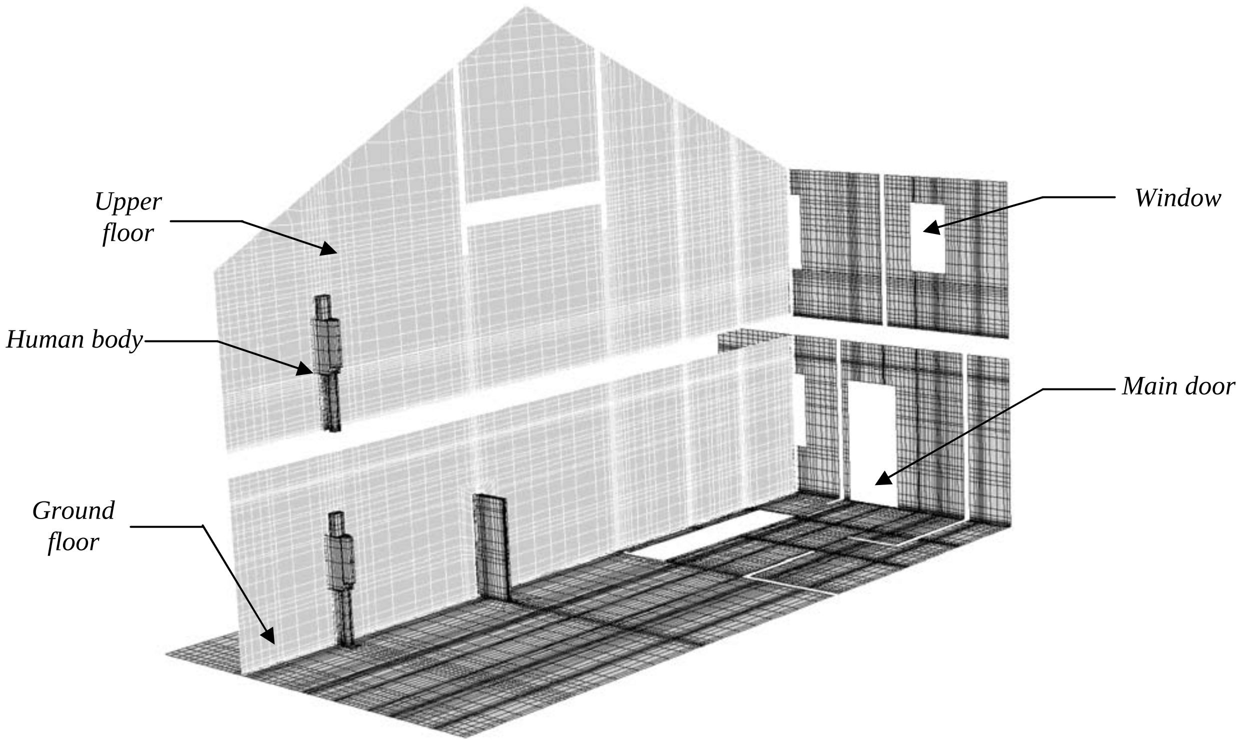

Figure 2. A perspective of the computational grid used in the model

Numerical accuracy

FLUENT 6.2.16 is used to numerically simulate the flow and thermal behaviour of the air in the whole building. The governing equations are solved with a segregated scheme. The momentum, energy, turbulent kinetic energy and turbulent energy dissipation equations, and the radiative transport equations (RTE) are discretized spatially with a second-order upwind scheme and for the pressure the body force weighted scheme was used. The pressure-velocity coupling algorithm SIMPLE is used to solve the pressure term. The following under-relaxation values were used for the pressure, momentum, turbulent kinetic energy, dissipation rate of turbulence, temperature and radiation, respectively: 0.5, 0.05, 0.5, 0.5, 1.0 and 1.0. The simulation is declared converged when the parameters and the residuals no longer change with successive iterations or the residuals from all the variables (except for the temperature) in all the cases were less than 10^{-3} and for the temperature less than 10^{-7}.

RESULTS

The simulated values for the Autumn case show temperatures within the building ranging from 19.1ºC to 32.4ºC, where the latter is found near the humans and the TV. The average measured temperature in the living room during the same period is 25.4°C and for the kitchen 24.6°C. This can be compared to the values in the third column in Table 3 where the average simulated temperature for the Autumn case is found. The simulated values are slightly lower than the measured values, which can be explained by the fact that the simulated value includes temperatures at the window surfaces. Table 3 shows the average simulated indoor air temperatures.

Table 3. Average indoor simulated temperatures on the ground floor and the upper floor including the attic, respectively.

Simulated temperatures (°C)	Winter case	Summer case	Autumn case, low airflow rate	Autumn case, high airflow rate
Average air temperature ground floor	19.36	23.69	24.49	23.62
Average air temperature upper floor and attic	19.48	22.62	22.73	21.92
Difference	-0.12	1.07	1.76	1.7

The simulated air velocities for Winter conditions in the *z*-direction in the door opening between room *B* and the upper hall has been compared to measured values and show a similar pattern. However, the measured velocity magnitudes are rather low, and hence the measured values are uncertain.

Figures 3 to 6 show the temperature and velocity contours for the Summer and the Winter cases in the *xy*-plane at z = 5.5 m (mid-plane). The Summer case was simulated with a supply air temperature of 19°C, which is not always possible to achieve during real summer conditions. The temperatures in the real object will therefore be higher during a hot summer.

An air circulation pattern is found in the attic and there is air exchange between the attic and the rest of the house. High air velocities are found on the ground floor near the kitchen and in the upper hall. The airflow in the upper hall near the floor and ceiling comes from rooms *A* and *B*, respectively. Almost the same flow pattern and velocity magnitude have been observed for the Winter case, while the temperature level is approximately 3°C lower in the Winter case than in Summer case. In the Summer case the average temperature on the upper floor is higher than on the ground floor, while for the Winter case the situation is the opposite.

Figures 4 and 6 show that there is an airflow through the stairwell to the attic, which is in agreement with results from constant concentration tracer gas measurements that showed that the flow of fresh air to the attic was considerable [2].

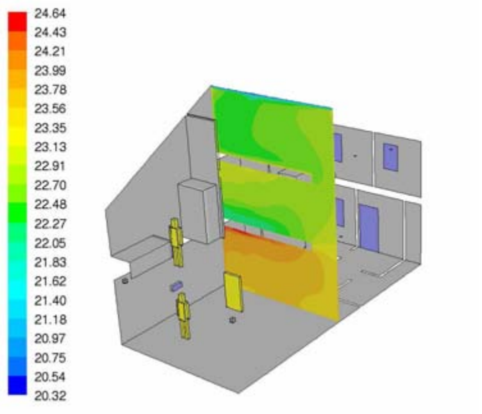

Figure 3. Contours of temperature in the *xy*-plane at *z* = 5.5 m (mid-plane) for the Summer case.

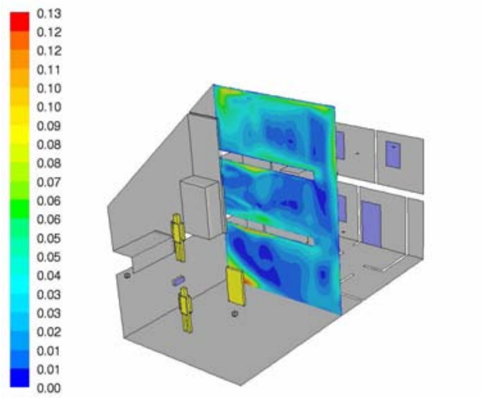

Figure 4. Contours of velocity magnitude in the *xy*-plane at *z* = 5.5 m (mid-plane) for the Summer case.

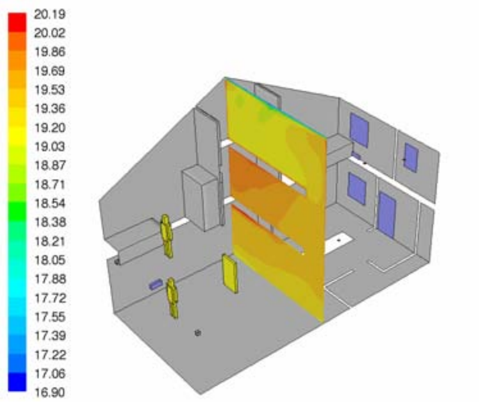

Figure 5 Contours of temperature in the *xy*-plane at *z* = 5.5 m (mid-plane) for the Winter case.

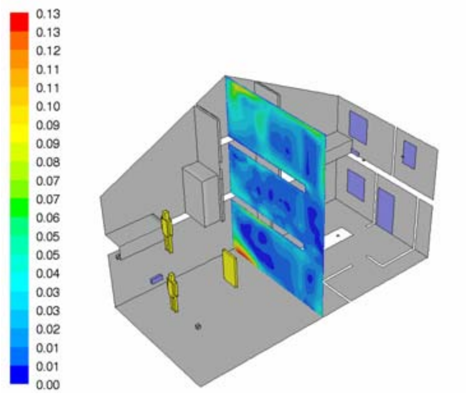

Figure 6 Contours of velocity magnitude in the *xy*-plane at *z* = 5.5 m (mid-plane) for the Winter case.

Figures 7 and 8 show the contours of the velocity magnitude for both the Summer and the Winter cases in the *yz*-plane at *x* = 1.5 m. The predicted airflow around and above the human representative can be compared to calculations of plume velocities according to empirical findings for a heat source. The airflow to and from rooms *A* and *B* into the hall is also shown and it is more pronounced in the Summer case.

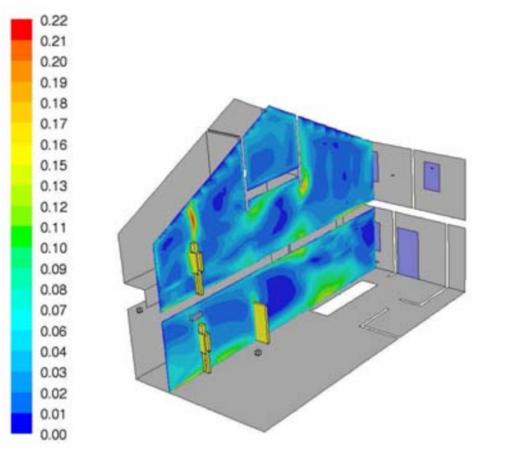

Figure 7 Contours of the velocity magnitude in the *yz*-plane at x = 1.5 m for the Summer case.

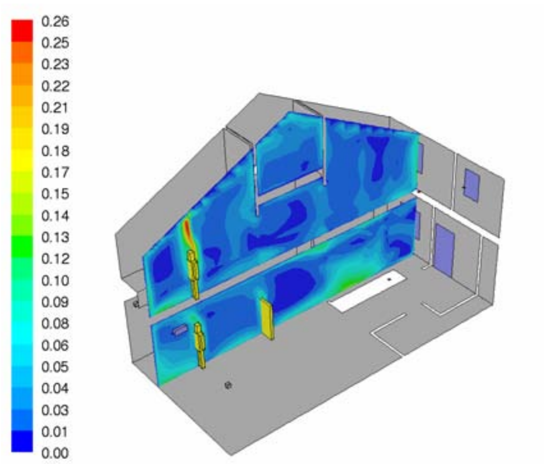

Figure 8 Contours of the velocity magnitude in the *yz*-plane at x = 1.5 m for the Winter case.

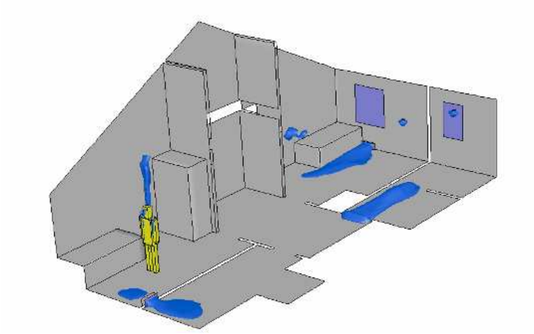

Figure 9 The iso-velocity 0.15 m/s on the upper floor for the Summer case.

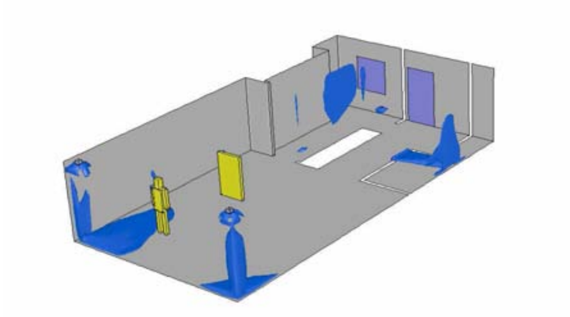

Figure 10 The iso-velocity 0.15 m/s on the ground floor for the Summer case.

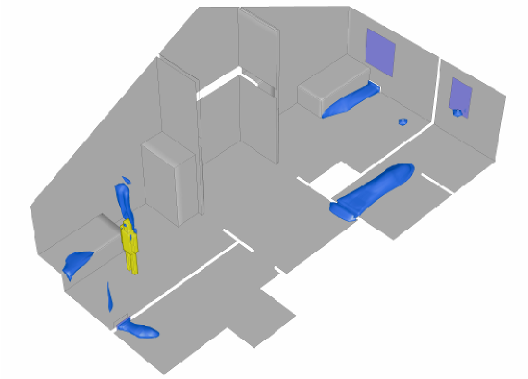

Figure 11 The iso-velocity 0.15 m/s on the upper floor for the Winter case.

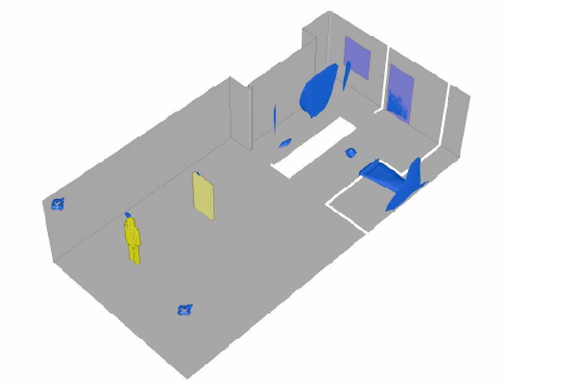

Figure 12 The iso-velocity 0.15 m/s on the ground floor for the Winter case.

Figures 9 to 12 show the iso-velocity 0.15 m/s contour for the Summer and Winter cases for both the ground and the upper floors. The air velocity 0.15 m/s is the limit to avoid discomfort due to draught during the cold season. This value can be increased to 0.25 m/s in the Summer time without causing draughts. As it is shown in the figures the air velocity close to the supply device in rooms *A* and *C*, above the human body's head on the upper floor, in the kitchen and at the air gap below the toilet doors on both floors is 0.15 m/s. The toilets are the location that the occupants of the low-energy buildings mention connected to draught in interviews about the indoor climate [8]. The downdraught from the supply devices in the living room is more pronounced in the Summer case compared to the Winter case. The air velocity is in general lower or equal to 0.15 m/s in occupied zones in all simulated cases and consequently, the risk of local discomfort due to the draught is very low

DISCUSSION

The object of this study is a low-energy house that was built with the aim of showing that it is possible to live in low-energy houses despite the cold Swedish climate. In low-energy buildings, the social load affects the thermal environment substantially. Fairly small heat loads will increase the temperature locally. It is important that heat is distributed all over the building to prevent over-heating and that surplus heat can be removed properly. The layout of a low-energy building is therefore important for providing a suitable airflow pattern. CFD simulation is a valuable tool for predicting airflow patterns within the whole building as is been shown in this paper. The results from the simulations show that the temperature differences are small and the air velocities are rather low, generally less than 0.15 m/s in the whole building. The results presented here concern temperature and airflow patterns in only a few cases. The model will be further developed to provide results about the indoor climate with other boundary conditions to represent various households and technical conditions.

ACKNOWLEDGEMENT

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CFD Modelling of Ventilation in an Educational Institute

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SUMMARY

The role of computational modelling as a development and engineering tool has recently become more significant. The major advantage is that modelling enables the testing of different geometries without having to build expensive test constructions. When a whole new building is in question, modelling is the only way to see inside it before it's finished.

The aim of this work was by using numerical simulation to investigate ventilation and heating system in an educational institute, which is under thorough renovation work. The main emphasis was put on studying whether the designed system is sufficient in extreme conditions especially in a three-story lobby and two industrial kitchens.

Only details important for ventilation were included in the models describing the room geometry. As the result of the simulation several modifications were made to the design in order to ensure, that the temperature remains in convenient level and velocities don't rise too high.

INTRODUCTION

The role of computational modelling as a development and engineering tool has recently become more significant. The major advantage is that with the help of modern modelling tools testing of different geometries is possible and no expensive test constructions are needed. Modelling is often also the only way to see inside the device; it is either too complicated for measuring equipment or it may even not exist yet. If the device in concern is a whole new building, there really are no other options than modelling.

The aim of this work was to investigate heating and ventilation in an educational institute by using numerical modelling. The building itself is under thorough renovation work, during which only its bearing outer walls were kept. Not only were its inside parts reconstructed but also totally new parts were added. Naturally its ventilation and heating system is reconstructed during this year too.

Since the major part of the vocational institute is used by the students of the catering college, the ventilation faces extreme conditions. The real challenge especially for the heating, however, is an eight meters high and 13 meters wide window in the lobby. This window in the summer time is exposed to direct sunlight, but during a cold winter day the outside temperature might be as low as -35 degrees centigrade. While the temperature and air velocities in the large three-story lobby must be remained in convenient level, hot fumes and smoky smells from the fully operating educational and commercial kitchens must be kept out of the hallways and removed.

Main emphasis was though put on finding out whether the designed system is sufficient to fulfil the requirements given. Besides the lobby two different kitchens were modelled. In order to provide as accurate information to the modelling software as possible the models representing the real geometry were made in close cooperation with the architects and designers. On the other hand, one of the most important tasks was to familiarize these experts with simulation so that its capabilities are known and can be used in further projects.

METHODS

Computational fluid dynamics

Computational fluid dynamics (CFD) is nowadays widely used in different kind of engineering applications. When modelling project is started the aim of the simulations must be specified, since it further defines what the geometrical model should be like and what kinds of boundary conditions are needed.

Geometrical representation based on real measures of the device in concern is created in a specific pre-processor. Depending on the goal of the simulation the actual geometry is either modelled accurately or simplified, when unnecessary details can be left out. The resulting model is then filled with computational mesh, which is needed in finite element based solution method of the equations describing the flow. Before calculating the actual solution for the problem, material types, mass flows and temperatures describing the situation in question must be given to the solver. After the final solution is achieved, the results can be post processed for example using numerical reports or contour plots so that the main findings can be easily seen. In this work commercial state-of-the-art computer software for modelling fluid flow and heat transfer was used.

Turbulent flows are characterized by fluctuating velocity fields. These fluctuations are too computationally expensive to simulate directly in practical engineering calculations. Instead, the instantaneous governing equations can be averaged or otherwise manipulated to remove the small scales, resulting in a modified set of equations that are computationally less expensive to solve. However, the modified equations contain additional unknown variables, and turbulence models are needed to determine these variables in terms of known quantities. [1]

In turbulence models the central issue is how the eddy viscosity is computed. The mixing-length zero-equation turbulence model uses a simple relation to calculate turbulent viscosity. The viscosity of the fluid is changed from a constant value to one defined by

$$\mu_t = \rho \ell_m^2 S \,, \tag{1}$$

where μ_t is turbulent viscosity, ρ density, ℓ_m the distance the turbulent mixing takes place and S the modulus of the mean strain rate tensor. This model, known also as the algebraic model, is very economical and suitable for most design purposes in cooling solutions [2]. Since the degree of turbulence and its details were not the major interest, this very simple model was used in the simulations presented in this paper.

As a heat transfer equations a simple balance equation for sensible enthalpy is written as

$$\frac{\partial}{\partial t} = (\rho h) + \nabla \cdot (\partial h \vec{v}) = \nabla \cdot [(k + k_t)\nabla T] + S_h, \tag{2}$$

where t is time, ρ is density, h is enthalpy, v is velocity, k is the molecular conductivity and k_t is the conductivity due to the turbulent transport. Source term S_h includes any volumetric heat sources, which might be defined in the model. [3]

The radiation heat transfer plays significant role especially if velocities are fairly slow. This is often the case in heating and cooling solutions, since the velocities are intended to keep in a fairly low level to avoid the feeling of draught. The radiative heat transfer equation for an absorbing, emitting and scattering medium at position $\vec{r}$ in the directions $\vec{s}$ is

$$\frac{dI(\vec{r},\vec{s})}{ds} + (a + \sigma_s)I(\vec{r},\vec{s}) = an^2 \frac{\sigma T^4}{\pi} + \frac{\sigma_s}{4\pi}\int_0^{4\pi} I(\vec{r},\vec{s}\,')\Phi(\vec{r},\vec{s}\,')d\Omega', \tag{3}$$

where I is the radiation intensity, a is the absorption and σ_S the scattering coefficient, n is the refractive index, s Stefan-Boltzmann constant and Ω solid angle. T is the local temperature in the domain. In the code used this equation is treated by writing the intensity of the radiation as a series, from which a source term can directly be added to the balance equation of enthalpy [4,5].

Modelled cases and boundary conditions

In this work two kitchens and large lobby were modelled. One of the kitchens is used by the students of the catering college during the academic terms; the other is the main lunch kitchen of the whole institute. It serves also the à la carte restaurant, which can be booked throughout the year for large special occasions like weddings. This restaurant is in direct connection to the lobby, which sets strict requirements for the ventilation.

For both kitchens all fixed cabinets and shelves as well as kitchen appliances, such as stoves, ovens and dishwashers, were included in the models as simple block and plate type objects. These together with the walls of the rooms formed the outlines of the geometrical model.

Heat was generated from several different sources. Kitchens were modelled in conditions corresponding to the sunny summer afternoon, which means that the solar heating coming through the windows was taken into account by providing the outer temperature and the heat transfer coefficient for the window. Single lighting devices were not modelled; instead the thermal power of the lamps was estimated to be half of the illumination power, which was set as a planar source in the ceiling. Stoves and small appliances were modelled in the similar manner, but the heating powers of larger devices were given as volumetric sources. When all the devices are operating, there also are several people inside the room. Persons were modelled as separate blocks obstructing the air flow and with small thermal power. Total heating power in the simulations of the educational kitchen was 25 kW and in the à la carte kitchen 28 kW. Example of the geometrical model and heat sources are presented in figure 1.

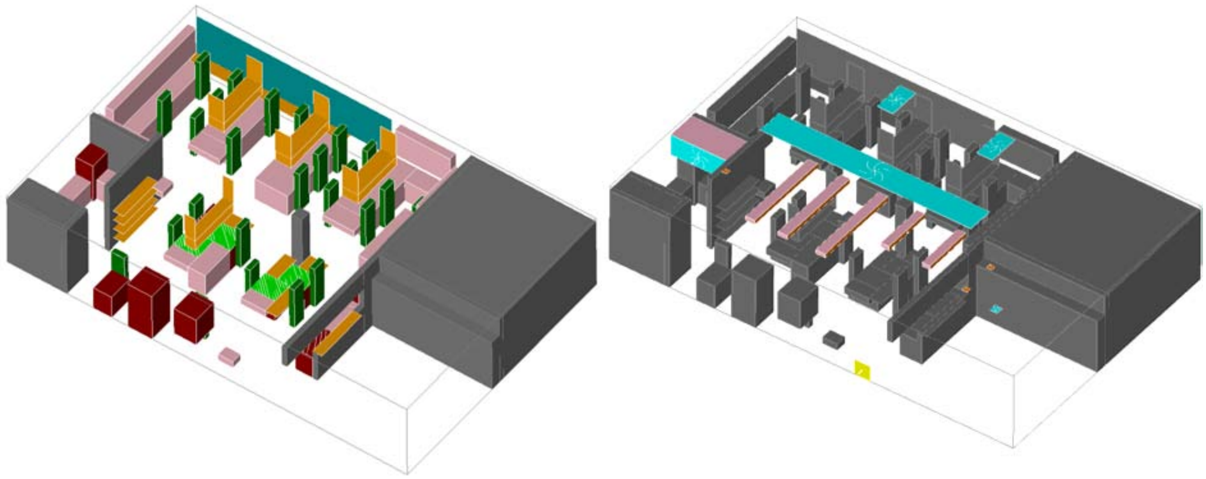

Figure 1 Model example of the educational kitchen (left) and details of the ventilation (right). Planar heat sources are coloured green and volumetric sources red. Blocks describing the students are dark green. Window is blue. Fixed cabinets are pink, shelves yellow and internal wall structures grey. Low velocity units are coloured turquoise, grease filters in the ventilation ceiling and valves orange. Grille in the door is bright yellow.

Air was provided to the kitchens mainly in two ways: through low velocity unit in the ceiling and supply air hoods. The hoods were placed above critical kitchen units and were also used to effectively exhaust the hot fumes. In addition to this in the educational kitchen air was removed via ventilation ceiling equipped with grease filters. The amount of incoming air was given to the program by fixing volume flows and temperatures. Outflow was designed to be slightly larger, and it was defined by fixing velocities. Grilles in the doors are used to balance the mass flow. This is meant to secure, that flow direction is towards the kitchen, not out of it. In the à la carte kitchen were two openings: the one through which the food is served and the one through which the dirty dished are returned. These were left fully open so, that the air could freely flow either in or out.

Lobby was modelled in circumstances, which correspond to warm midsummer afternoon and cold winter day. All three stories of open space and staircases in between were included, but no furniture was modelled.

During the wintertime there are several radiators, which heat the lobby. Furthermore, computers provided by the institute are available for the students throughout the year on the third floor. These all were taken into account as planar heat sources. Illumination was modelled as in the kitchens. Outer temperature was affecting the inside climate through the heat transfer coefficients of the outer walls and roof as well as windows and foundations. In the winter the glass of the large window on the second level up to three meters was heated with a constant power.

Ventilation is very complex and there are multiple air terminal devices in every floor. In the ceiling of the first floor, next to the demo kitchens, air is provided through four supplies, which gently blow the air in two directions along the roof. In the main floor swirl diffusers also slightly circulate the flow, but they were modelled only as supplies with appropriate swirl. On the upper floor there are two large rectangular diffusers, which also function as exhausts. They are turned off in the winter. To control the effect of the large window to the room temperature two fan coil units are installed in the floor between the second and the third level. In addition to these there are two large grille diffusers next to the large window on the second floor and in the hallway of the third floor. Air is removed from the lobby via several single valves in the ceiling and in the walls. The amount of air in the lobby was controlled and fed to the program by setting volume flows and velocities and temperatures for all together 23 supplies and 16 exhausts.

RESULTS

Simulations results were illustrated by multiple contour plots and numerical reports of the velocities and temperatures in the modelled domains.

In that part of the educational kitchen which is closer to the window temperature remains quite cool despite the heavy thermal load on the other side of the room. The flow pattern from the low velocity air units disappears almost totally before it reaches the working level and velocities do not rise too high down near the floor either. Air is heated up by the kitchen appliances before is exits the room via grease filters. Under the ventilation ceiling the temperature is clearly warmer than near the windows, but all warm air is immediately removed from the room. This is illustrated in figure 2 by the path lines released from the low velocity unit and coloured by temperature. Based on the simulations results the ventilation system for this kitchen was totally adequate and no changes to the original plan were made.

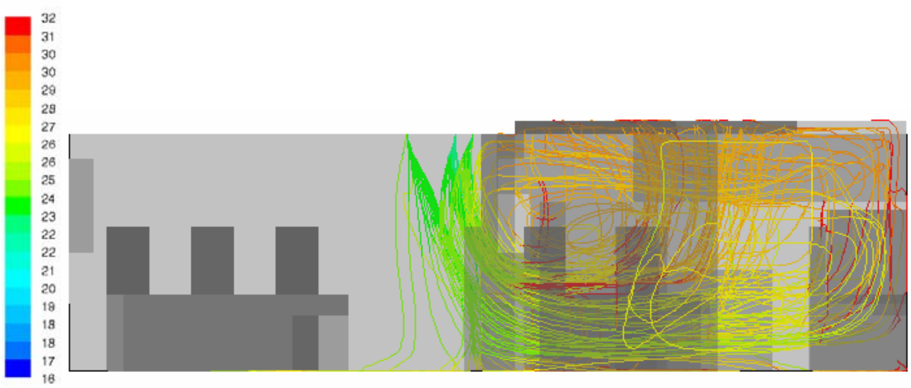

Figure 2 Pathlines released from the low velocity air supply and colored by temperature (^{o}C).

In the à la carte kitchen those appliances which have the greatest thermal power are quite close to each other. In this area major part of the food is prepared and cooked, and therefore the proper ventilation is essential. The simulations showed, however, that part of the hot fumes escaped under the supply air hood instead of being removed. The reason for this was that the screen between the two most important compartments was not as high as the ventilation designer had thought. A simple change was made to the dimensions of the supply air hood, which extended it over the screen. At the same time one large low velocity unit was split into two, since the original unit created too cool conditions in the baking area of the kitchen. The other part of the unit was transferred near the screen to provide more fresh air to the problem area. New results proved that the change made was efficient; almost all hot air escapes and the temperature level throughout the kitchen is more equal. Contours of temperature in the à la carte kitchen in the baking and cooking area are presented in figure 3 before and after the modification.

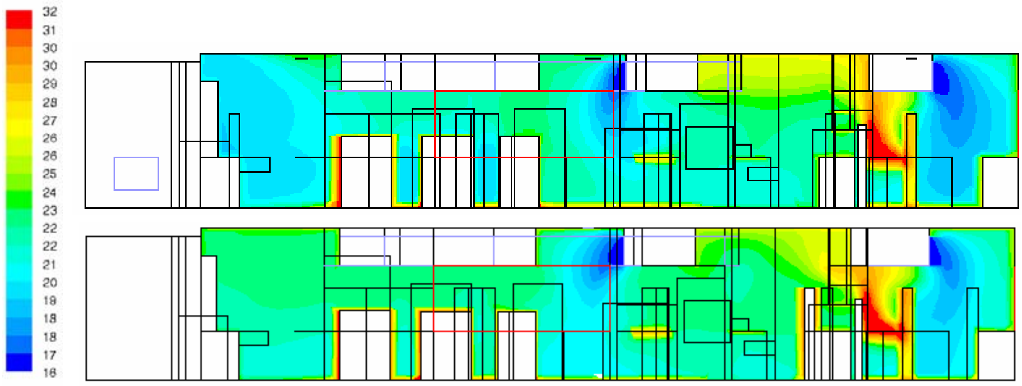

Figure 3 Temperature (^{o}C) in a cross cut of the à la carte kitchen before (top) and after (bottom) the modifications made to the low velocity unit and air supply hood. Baking area is on the left and cooking area is on the right.

In the lobby the overall temperature level and temperature difference between the floors were of special interest. Based on the simulations in the summer the difference between the bottom and the top floor was too high. In addition to this the temperature on the first floor was too high compared to the outside temperature, which might feel uncomfortable when entering the hall. Therefore the aim of the second simulation was to raise the temperature level in the two bottom floors without changing the already quite warm temperature on the third floor. This was done by raising the inlet temperature of all other air supply devices except the fan coil units in the ceiling of the top floor. To be sure that the ventilation is still adequate, 25 groups of four people with small thermal power were placed in the main floor to present the guests of a large celebration. Resulting temperature field is much smoother and the temperature difference in the lobby is only a couple of centigrade at its largest, as can be seen in figure 4.

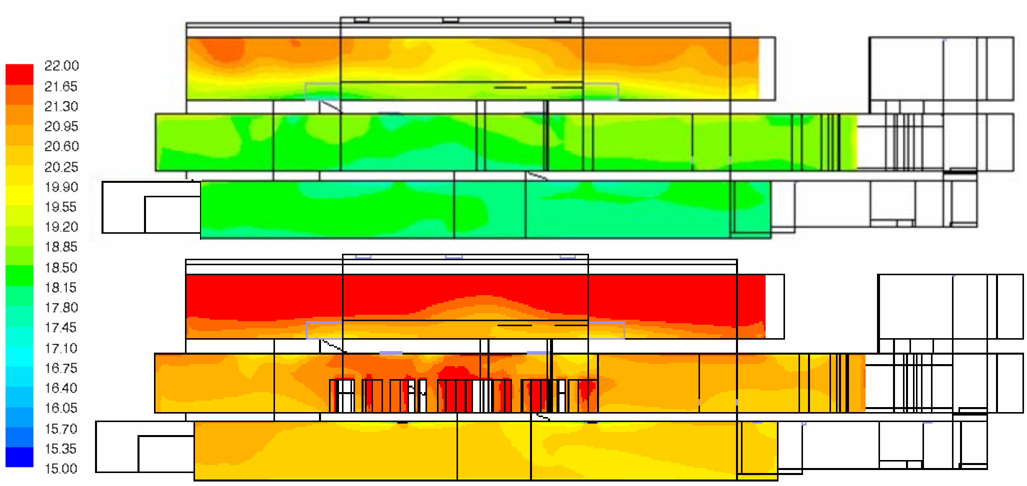

Figure 4 Temperature (^{o}C) in a cross cut of the lobby before (top) and after (bottom) the modifications made to the ventilations system in order to even out the temperature differences between the floors.

During a cold winter day, when the radiators were turn on, the temperature in the hall was too high. Even the cold windows did have hardly any effect at all. The reason was found to be in the exaggerated heating powers compared to the outer temperature, which was corrected to the simulation model too. Heating capability was designed noting, that some cold air will eventually be carried into the hall. The effect of cold air was now also included in the model as heat sinks near the cold windows. These changes decreased the overall temperature levels in the whole lobby, and the differences between the floors are still very moderate.

Finally the bottom part of the large window in the lobby was replaced by a normal plain glass window. The purpose was to study whether the quite expensive heating system is necessary or not. As the result the largest differences in the temperature were now in the main hall of the second floor. The effect is one degree of order and it stretches all the way from the window to the staircase in the back. Especially near the floor the air is colder, which might together with the air flows feel as draught. Heating keeps the surface temperature of the glass in adequate level, whereas without the heating it is close to zero centigrade. Even though the temperature difference in the hall was not that remarkable, conditions near the heated glass feel more pleasant to a person who stands near the window. Surface temperatures of all windows in the lobby are presented in figure 5.

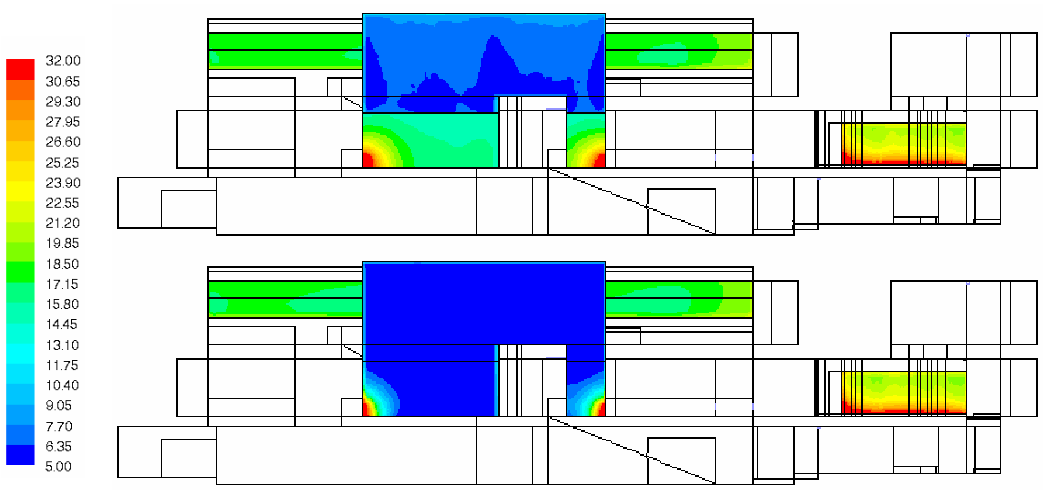

Figure 5 Surface temperature (^{o}C) on the windows of the lobby when the bottom part of the large window is heated (top) and when the whole window is made of ordinary glass (bottom). Warm areas in the corners are due to radiators located next to the window.

DISCUSSION

Since the modelling tools were not familiar to all participants of the simulation project in advance, the possibilities and restrictions of simulation were discussed over in several meetings, during which also the results were gone through in detail. Even though CFD is a powerful tool in obtaining information on large variety of even highly detailed devices, one must also keep in mind its limitations. No result can be more accurate as the geometry description and boundary conditions provided to the software. The desired goal determines what the model should be like; if the purpose is to simulate a whole building and find out the trends of the temperature and velocity profiles, there is no use to include small items such as a single casserole or door knob. CFD is at its best when cases can be compared after slight modifications, like it was made in this project.

The main purpose in this work was to secure, that working conditions are acceptable throughout the building. Temperature levels and the differences between the floors and compared to the outside temperature must not be too high. Velocities should stay on the moderate level to avoid the feeling of draught. Based on the simulation results several modifications were made to the heating and ventilation system. Most of the findings would not have been noted until the renovation work would have been totally completed. With the help of CFD some weaknesses from the design could be pointed out and corrected before the final choices were made and system constructed on site.

ACKNOWLEDGEMENT

The authors would like to thank Jyväskylä Vocational Institute and Haahtela Rakennuttaminen Oy as well as Jeven Oy for providing the funding of this work. The help of the staff of Arkkitehtitoimisto Pertti Nousiainen Ky, Insinööritoimisto Chydenius Oy and Jeven Oy in obtaining the boundary conditions for structures and devices is highly appreciated. Without the help and knowledge of fellow colleagues in Process Flow Ltd Oy this work would never have been finished.

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Experimental unsteady characterization of thermal building performance

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SUMMARY

The objective of this paper is to present the results of a study that determinates the building's heat loss, a system thought up itself that permits to obtain the energy consumption by square meter of the dwellings (W/m^{2}K) in unsteady. The system was composed by vertical fan heaters that maintained the dwelling in uniform conditions of temperature. So much interior temperatures were measured like outsides in each one of the walls of the dwelling. Measurements in 8 types of dwellings were carried out, 7 dwellings correspond at building object of the study, and the last dwelling was carried out in another building. Experiences in the unsteady were carried out, obtaining the value in the steady-state. And it was obtained that the consumption was affected by the type of glazing.

INTRODUCTION

Casas Bioclimáticas, S.L. is dedicated to the promotion and construction of dwellings since its foundation in the year 2001. It forms part of the business group that heads Promociones y Construcciones M.D., S.L., with an experience in the sector of more than 25 years.
The business is dedicated mainly to the residential building, in its great of new dwellings promotions majority (one-family, semidetached and flats), mainly in Spain (in the province of Valencia). The differential element of these dwellings situates in the criteria of sustainability, bioclimatic, environmentalism and energy efficiency, besides the use of the automated one, utilized in the construction of the dwellings.

This company concerned about improving the energy efficiency of its buildings signed in the 2006 a contract of investigation with the research team Group of Design and Development of Sensors with the objective of analyzing the energy consumption of the dwellings.

The energy consumption of a specific building depends mainly on the building type, climatological conditions, building construction, annual hours of use, installations for heating, cooling, production of domestic hot water and lighting [1]. According to the Danish Environmental Protection Agency, the residential energy use per capita varies widely among European countries, from 150–350 kWh/capita in south Europe [2]. Energy is mainly used for space conditioning (heating and cooling primarily in southern Europe), sanitary hot water production, cooking, lighting and electrical appliances (i.e. refrigerators, washing machines and entertainment equipment).

Recently, in Spain, under the umbrella of the Directive 2002/91/CE, a new building technical code (CTE) has been developed [3]. The CTE has as an objective to obtain a rational use of the necessary energy for the use of buildings, reducing its energy consumption and utilizing for it sources of renewable energy. Thus the CTE establishes the requirement to incorporate energy efficiency criteria and the solar, thermal or photovoltaic use of energy in the new buildings or in those that be going to restore. The Basic Document contains four basic energy demands: limitation of the energy demand, where the values limit for the building's envelope are established (facades, glasses, covers, etc.); energy efficiency of the installations of lighting, where they are set for the first time in the Spanish regulation, some requirements to comply for these installations above all for buildings of the tertiary sector; the demand relating to the solar contribution of domestic hot water obliges that the production of domestic hot water be carried out with a contribute obligatory of thermal solar energy that will vary between a 30% and a 70% in function of the daily volume predicted of hot water demanded; and the minim contribution photovoltaic of electric energy, that establishes that in the new buildings of the tertiary sector of a determined surface.

The CTE implements two different indicators for the energy assessment of buildings, both of which are indirect indicators and far away from the absolute quantification of the building energy performance in terms of the kW h m^{-2} $year^{-1}$ of primary energy consumed. In its simplified option, the main indicators are the stationary heat transfer coefficients from the different components of the building's envelope, without properly taking into account solar irradiation. In its general option, the implemented indicator is the energy demand of the object building envelope in relative value to the building envelope energy demand of a variable reference building, being both energy demands evaluated with a dynamic code, called LIDER, that has been specifically developed for the CTE, and that presents important limitations in relation to its capabilities to properly assess the energy performance of different building design approaches.

Casals [4] coincides that building performance indicators implemented in the CTE are completely inappropriate. The indicator of the simplified option, the stationary heat transfer coefficient, is obviously absolutely inappropriate both for assessing the effect of many building energy issues, as well as for providing quantitative information about the building's energy consumption.

The need for a more realistic evaluation of thermal building performance in the local climatic conditions is undoubtedly necessary for a more accurate assessment of the thermal performance and for better energy efficient design.

METHODS

The building

Figure 1. Building's elevations.

The residential building is situated on the outskirts of Valencia's city, in the village of Paterna, will be destined for residential use, by means of system of rental submitted at subsidized housing and focussed on people younger than 36 years; have been built 44 dwellings, being 4 adapted for people with mobility reduced, and all they with lower surfaces to the 70 m^2 useful; and 44 parking spaces, being 2 adapted for people with mobility reduced. A very exact measure of the energy consumption of the dwelling could be obtained owing to the clearing of the building. In addition there would be that to indicate that this consumption would be able to be seen reduced in a 60% on account of the losses in case that the adjacent dwellings were conditioned they would be for the facade toward the outside.

They were analyzed some of the building's dwellings, a four-story building, whose situation they are stood out in the figures 2 and 3. In the figures 4 and 5 is shown the type of external walls, as for the dwellings analyzed as for the dwelling situated on the outskirts of Valencia's city, in the village of Tavernes Blanques.

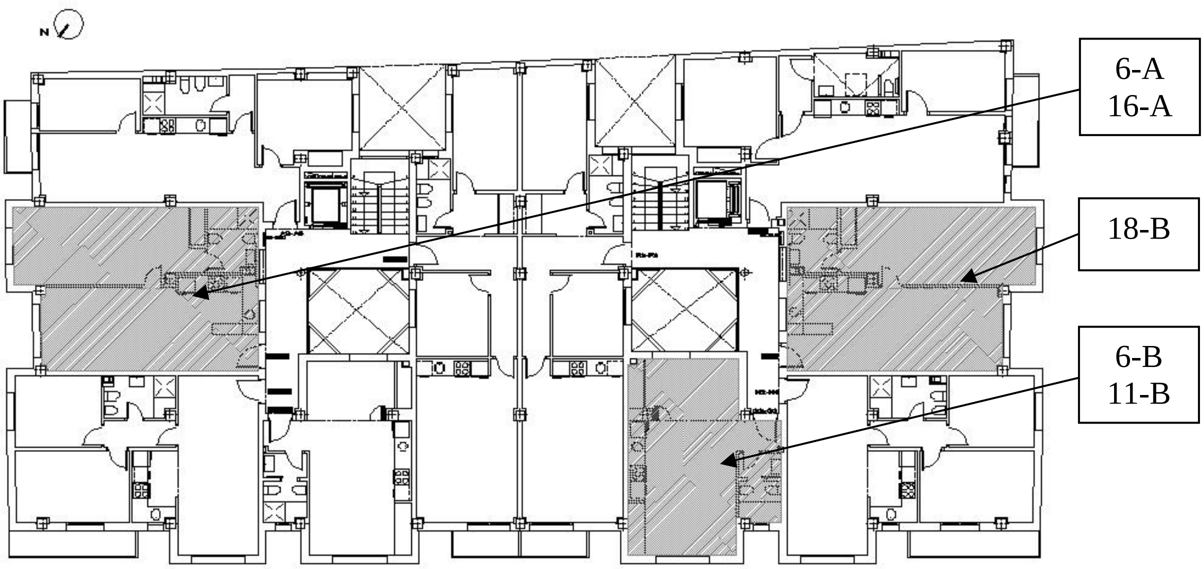

Figure 2. Type floor, which includes first, second and third floors.

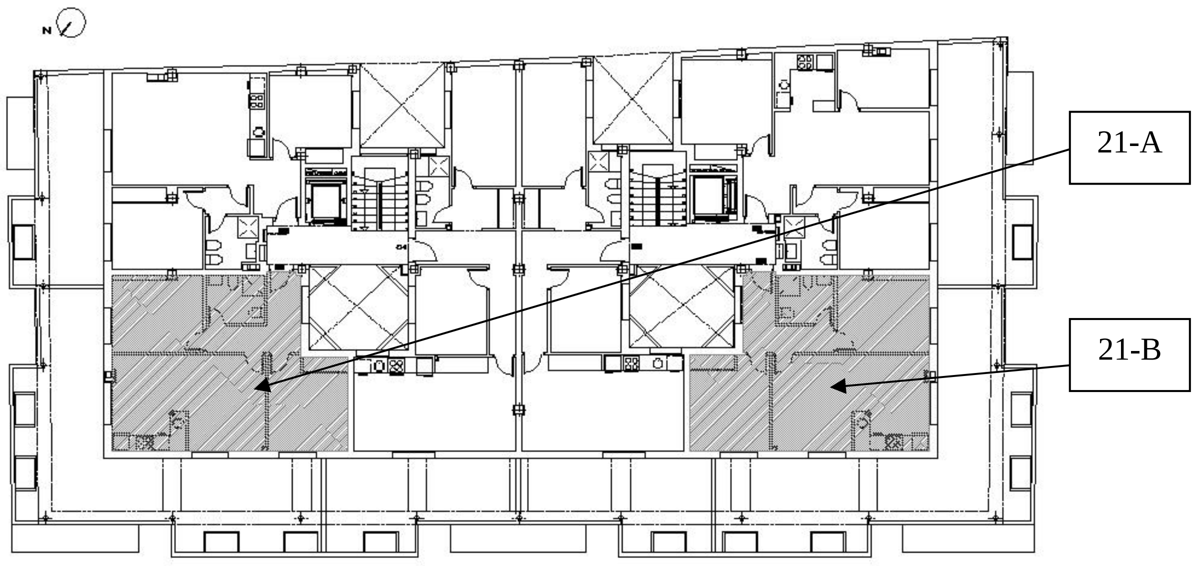

Figure 3. Top floor, housing 21-A y 21-B.

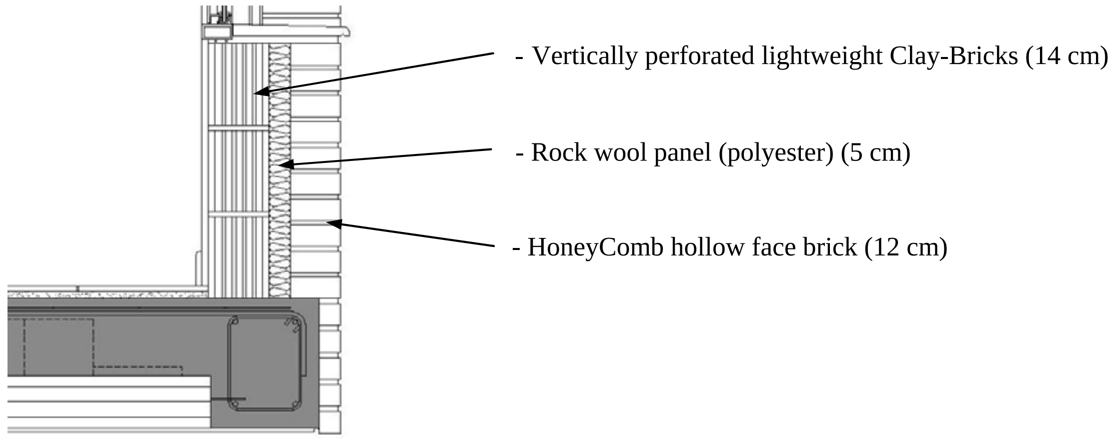

Figure 4. Section of residential building’s external walls.

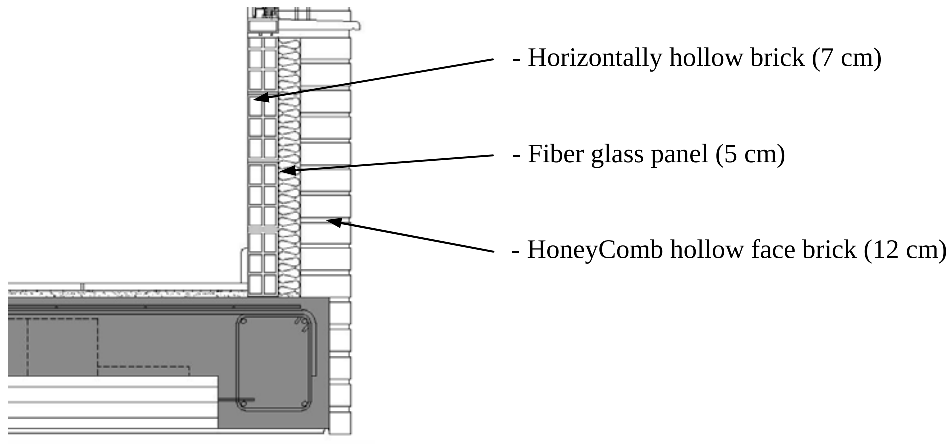

Figure 5. Section of residential building’s external walls situated on Tavernes Blanques.

With the objective to maintain a gradient temperature vertical fan heaters distributed by the entire house were utilized whose characteristics are shown in the table 1.

Table 1. Vertical fan heaters’s characteristics.

Maximum power (W)	2500
Voltage (V)	230
Power selector	2
Protection	IP 20
Thermostat	High precision

Instrumentation

The basic concept was to use existing meters and sensors and an existing data collection system already installed by the housing company in order to monitor the performance.
Data for the gradient thermal obtained from thermo couples placed on the external walls and a thermo couples portable in the closeness apartments.

RESULTS

Measurements in 8 types of dwellings were carried out, 7 dwellings correspond to the building described in the previous section, and the last dwelling was carried out in another building, which was found partly occupied, although all of the adjacent flats were found empty.

The measurements were carried out from 1 December 2006 until 1 January 2007. The results obtained were energy consumption, and exterior and interior temperatures, in relationship with the time. The instrumentation has been described in the previous section. Having differences among the different rooms' temperatures of the dwellings, it was obtained a value average that was weighted in function of their volume. Finding differences of adjacent dwellings' temperatures, it was obtained a value average was praised in function of the surface of the dividing wall.

For the calculation of the energy consumption by square meter of dwelling is utilized the following expression:

$$\varphi = \frac{Q}{t \cdot S \cdot \Delta T}, \qquad (1)$$

where ΔT is the average difference of temperatures and S is the dwelling's surface, t is the time passed between measurements, and Q is the heat consumed by the heating system. In the figure 6 they are shown for 4 dwellings the energy consumption by square meter of dwelling (W/m^2K) versus the time.

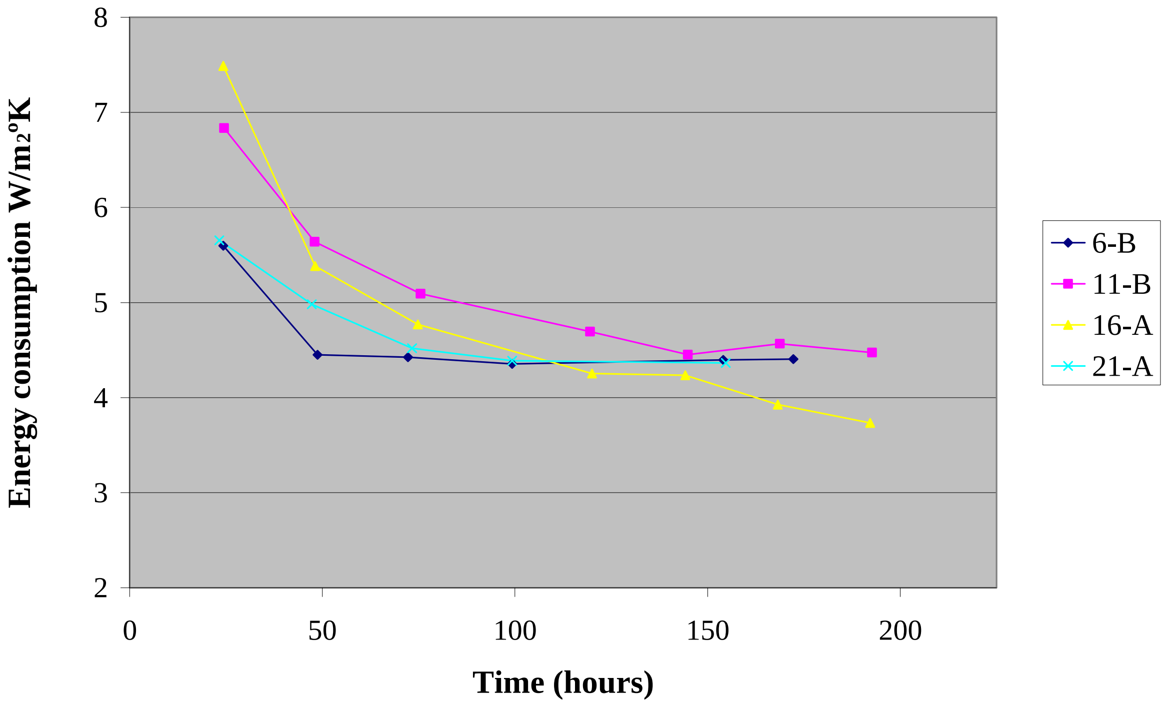

Figure 6. Energy consumption by square meter of dwelling (W/m^2K) versus the time.

Experiences in the unsteady were carried out, and obtaining the value in the steady-state by the representation of Ln φ against the time's inverse. In the figure 7 they are shown for 4 dwellings the Ln φ versus time's inverse.

In the figure 8 the results obtained are shown for all the dwellings analyzed after obtaining the intersection in the axis of the straight obtained adjusted by linear minimum mean square error.

It was discussed that the losses by windows could be important. In the table 2 they are shown the surface of windows of each one of the dwellings. It was detected that the type of window in the dwelling situated in Tavernes Blanques corresponded with the type Aluminium frame / double glazing 22 mm fold-up window with vertical axis. While in the dwellings situated in Paterna the window corresponded with a window track of Aluminium frame / double glazing 18mm, whose losses were difficult to estimate.

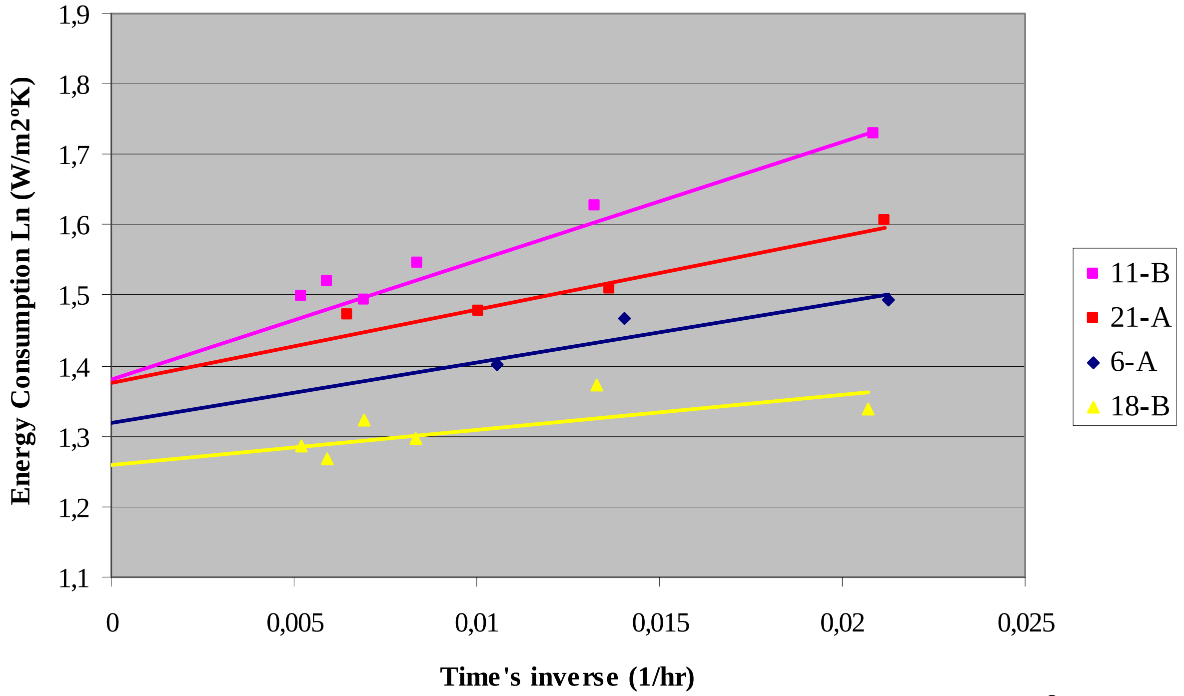

Figure 7. Logarithm of the energy consumption by square meter of dwelling (W/m^{2}K) versus time’s inverse.

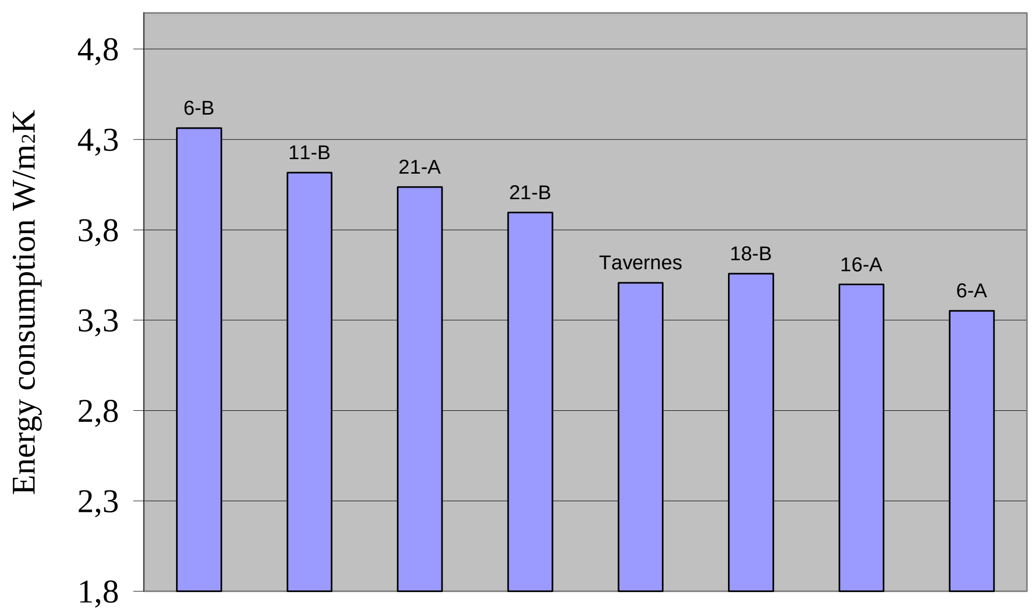

Figure 8. Energy consumption by square meter of dwelling.

Table 2. Window's surface.

dwelling	Window surface (m^2)
6-A	5,58
6-B	5,14
11-B	5,14
21-B	13,72
21-A	13,72
18-B	5,58
16-A	5,58
TAVERNES	11,64

Supposing that walls' losses should be similar in all the dwellings the value of the heat transfer coefficient was calculated (U_{window}) linear minimum mean square error by means of the following expression:

$$\varphi_{cerramiento} = \frac{\varphi_{vivienda} \cdot S_{vivienda} - U_{ventana} \cdot S_{ventana}}{S_{vivienda}}, \qquad (2)$$

To obtain the adjustment was supposed that the $U_{tavernes}$ was 3,1 W/m^2K, being reached an U_{window} equal to 3,6759 W/m^2K. The results of walls' energy consumption of the dwelling are shown in the figure 9.

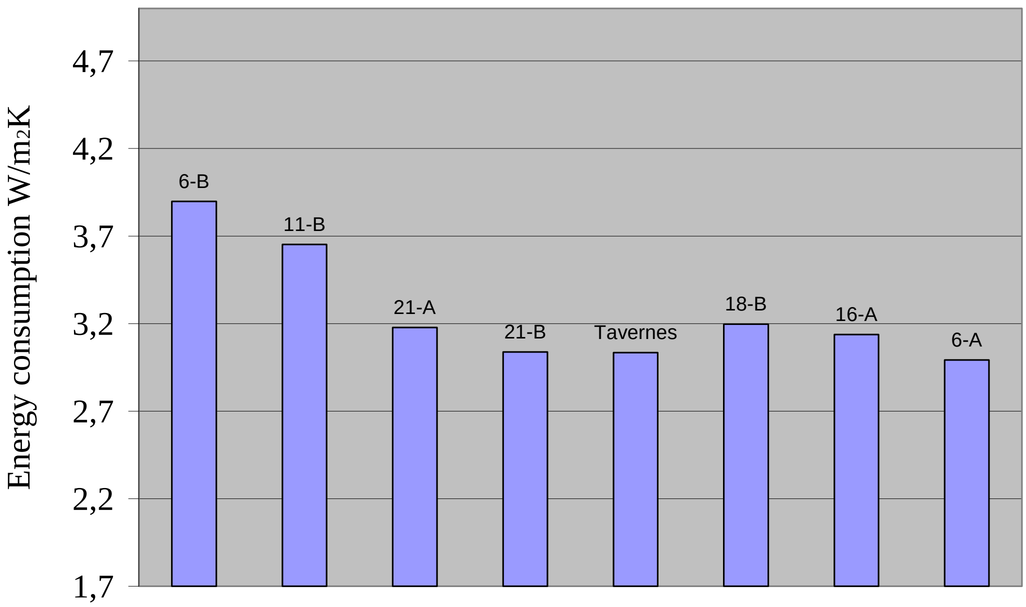

Figure 9. Walls' energy consumption by square meter of dwelling.

Observing the figures 8 and 9 can be appreciated that dwellings, that in principle, had better thermal behaviour when is analyzed the exterior wall is observed that its characteristics are very similar that other dwellings.

Supposing that the heating only is utilized 8 hours by day, and knowing that the city of Valencia have 515,9 heating degree days with base temperature 15-15. It is obtained that the consumption of the dwellings varies between 18 and 14 kW h m^{-2} $year^{-1}$, some values that compared to results of other works [1,2], for 800 heating degree days they are observed that they vary between 80 and 25 kW h m^{-2} $year^{-1}$. Those results indicate that the values obtained are inside the building of high energy efficiency. In case that the adjacent dwellings were conditioned as the losses would be practically by the facade toward the outside, then the consumption would vary between 13 and 3 kW h m^{-2} $year^{-1}$.

DISCUSSION

The thermal behaviour of different dwellings of a building has been analyzed. Experiences in the unsteady were carried out. And it was found that if the logarithm of the energy consumption by square meter of dwelling was represented (W/m^2K) with respect to the time's inverse (1/hr) could be obtained the value in the steady-state in the intersection with the axis.

It was discussed that the losses by windows could be important. It was compared with a dwelling with fold-up window with vertical axis. While in the other dwellings the glazing corresponded with a window track, being obtained the value of the heat transfer coefficient (U) of the window.

Representing the energy consumption of the exterior walls of the dwellings, after rejecting the losses by the glazing, was obtained that the differences were lows among dwellings. Even thus a pair of dwellings exists that themselves they are not found inside the rank. This would be able due to solar profits of the other dwellings through the glazing, or would also be able due to the predominant wind direction in the experience's execution that would be able to increase them lost in these dwellings.

ACKNOWLEDGEMENT

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Thermal Comfort Level and Energy Consumption in School Buildings in the South of Portugal

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SUMMARY

In this work a numerical model, that simulates the building's thermal response and evaluates the internal thermal comfort and air quality, in transient conditions, is used. This software, after being validated in a school building with complex topology in Winter and Summer conditions, is used in the thermal project of school buildings, located in the South of Portugal, with low energy consumption level.
In this study a new school building is projected and is evaluated an existing one, in Winter conditions. The new school building, with 118 compartments, will be used for teenager students, while the existent building, with 25 compartments, is used for infant students. In both school buildings the internal air temperature and the global thermal comfort levels, using the PMV index, that students are subjected, are calculated.

INTRODUCTION

In the Algarve region, located in the South of Portugal, with a climate with Mediterranean characteristics, is very important to develop school buildings, adapted to this region, that can promote the increase of the comfort conditions for students, teachers and other occupants, as well as to promote the decreasing of energy consumption level for the buildings. In general, in Winter conditions the external air temperature values are relatively lower and the solar radiation are relatively higher, in Summer conditions the external air temperature values are very higher and the solar radiation is also very higher and, in Spring and Autumn conditions, the external air temperature values and the solar radiation are higher. The teenager students are, in general, in holidays during the Summer conditions, nevertheless most of the infant students are in holidays only in August.
In general, in all school buildings, in Winter conditions, renewable resources are used, like direct solar radiation, in order to increase the internal air temperature level, nevertheless in Summer conditions is important to use adequate shading devices to reduce this direct solar radiation. The compromise of these two situations implicates a detailed study to the building thermal behavior, considered the several buildings structure details, the orientation in new buildings and inclusively the surrounding buildings.
In this study, the thermal results of a new and of an existing building, for two distinct students ages are analyzed in Winter conditions. In the first one, that will be built in the future for teenager students, the best building orientation that can promote acceptable thermal conditions is analyzed, while in the second one, used by infant students, the present thermal conditions are evaluated, including the influence of surrounding buildings, and some improvements in order to increase the internal thermal comfort and air quality conditions are

promoted. In all situations, in this study, only is considered the natural air renovation, the occupation cycle is not considered, the windows and doors are closed and the internal curtains are open.

NUMERICAL MODEL

The multi-nodal buildings thermal behavior model (see details in [1]), that works in transient conditions, is based in energy and mass balance integral equations. The energy balance integral equations are developed for the air (inside the several compartments and ducts system), the different windows glasses, the interior bodies (located inside the several spaces) and the different layers of building's main bodies and ducts system, while the mass balance integral equations are developed for the water vapour (inside the several spaces, ducts system and in the interior surfaces) and air contaminants (inside the several spaces and ducts system). In the resolution of this equations system the Runge-Kutta-Fehlberg method with error control is used. The model considers the conductive, convective, radiative and mass transfer phenomena. The conduction is verified in the building's main bodies (doors, ceiling, ground, walls, etc.) and ducts system (fluid transport) layers. In the convection the natural, forced and mixed phenomena are considered, while in the radiation, verified inside and outside the building, the short-wave (the real distribution of direct solar radiation in external and internal surfaces) and long-wave (heat exchanges between the building's external surfaces and the surrounding surfaces and among the internal surfaces of each space) phenomena are considered. In the radiative calculus the shading effect caused by the surrounding surfaces and by the internal surfaces is considered.
The input data in the software are the geometry (introduced in a three-dimensional design software), the boundary condition, the materials thermal proprieties and other conditions (like the external environmental and geographical conditions, the initial conditions, the several heat and mass load, the occupation cycle, the occupants' clothing and activity levels and the air ventilation topologies).

SIMPLIFIED BUILDING GEOMETRY

The analyzed modern school building (see Figure 1), occupied by teenager students, is divided in three floor levels (the first floor, the second floor and the roof) and three blocks (West and East Block used for lessons and North Block used for the nourishment and the administration). Each floor is formed by different compartments, building's main bodies and windows glasses. In relation to the main bodies all existing external bodies that promote shading effects were also considered. In this analyzed modern school building, that will be built in Albufeira, with three floor levels, divided in 118 compartments, 1986 building's main bodies and 231 windows glasses, are considered.
The second building (see Figure 2), occupied by infant students, divided in 24 compartments, 498 building's main bodies and 42 windows glasses are considered. This building, built in Olhão, has three classrooms, for 3, 4 and 5 years old children, and other spaces for offices, administrative, WC, teachers and non teachers staff and meeting room.
In Figures 1 and 2, the grid generation used in the numerical simulation of the school building for teenager and infant students, respectively, are presented. This numerical grid, used in the internal and external direct solar radiation determination, was spaced 30 cm in both directions.

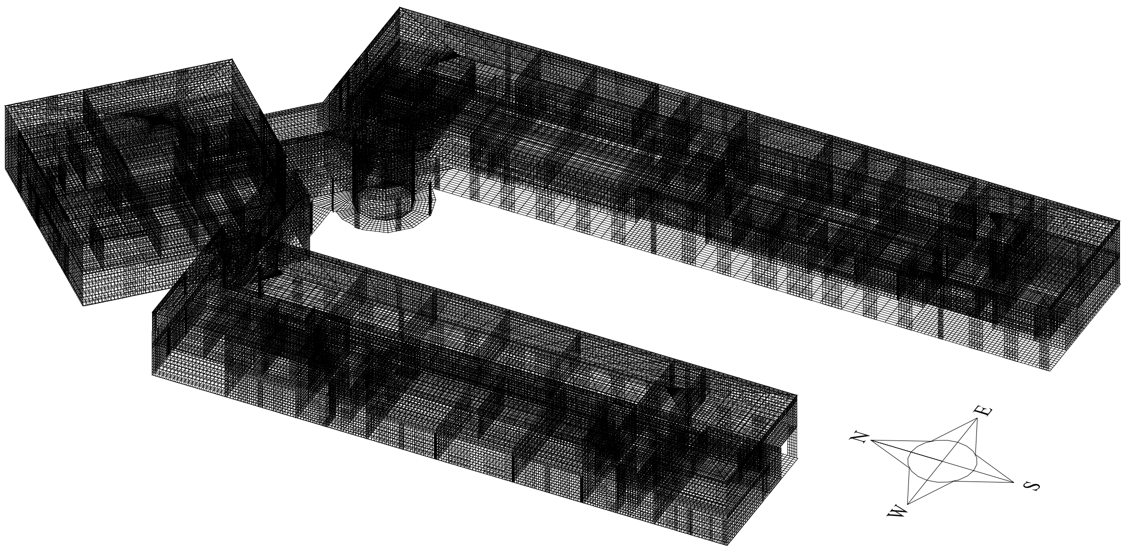

Figure 1. Grid generation in the new school building for teenagers, used by children between 15-18 years old. This geometry is based in the “Escola Secundária de Albufeira”.

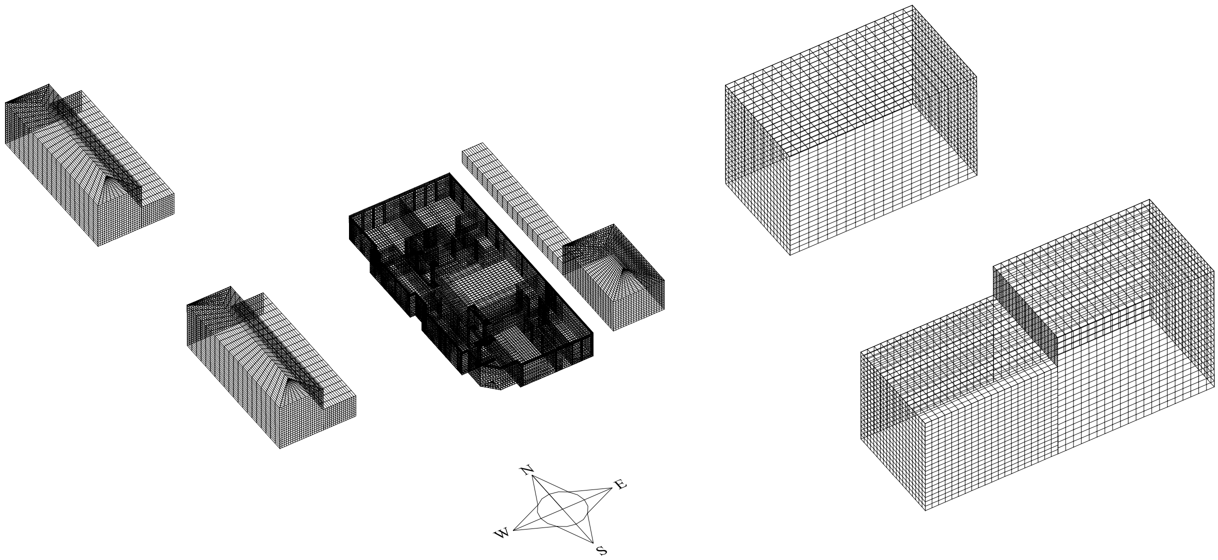

Figure 2. Grid generation in the school building for infants, used by children between 3 to 5 years old. This geometry is based in the “Jardim-de-Infância Nº1 de Olhão”.

RESULTS AND DISCUSSION

In this study, the software that simulates the school building’s thermal response, with complex topology, in transient conditions, and evaluates the indoor thermal comfort and air quality levels was used. After being validated this numerical model in Winter (see [2]) and in Summer (see [3]) conditions, two different buildings situations, for teenagers and infants, are analyzed.

In these simulations the doors and windows were closed and the internal shadings devices were open. The air exchange rate by natural renovation of 0.71 h^{-1}, determined experimentally in [4], and the non-occupation are also considered in the simulation.

The new and the existing buildings, used for teenagers and infants, respectively, were studied in Winter conditions. In both school buildings, in order to evaluate the real building thermal inertia, the previous 5 days were also simulated.
In this simulation, in order to promote acceptable thermal comfort level, was decided to use the Category B of [5].

School Building for Teenagers

The school building for teenagers was subjected to a preliminary study in order to analyze the influence of the building orientation in the thermal behavior. In accord to the energy consumption decrease and thermal comfort increase philosophy, was decided to orientate the building in the South-North direction, with the classroom windows turned East and West. The idea of this philosophy, mainly in Winter conditions, is to use the solar radiation as heating system.
The evolution of internal air temperature and thermal comfort level, for the classrooms, the laboratories, the offices and the corridors are presented, respectively, in figures 3, 4, 5 and 6. In figure 3 the classroom number 18, with windows turned North, is located in the first floor, while the other classrooms are located in the second floor: number 60 in the West block with windows turned West, number 61 in the West block with windows turned East, number 103 in the East block with windows turned West and number 115 in the East block with windows turned West.

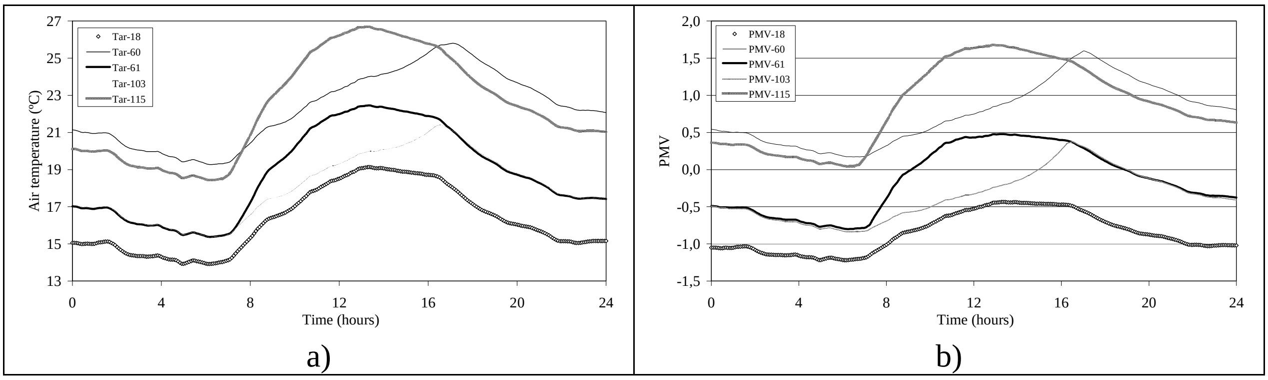

Figure 3. Internal air temperature a) and thermal comfort level b) evolution for the classrooms.

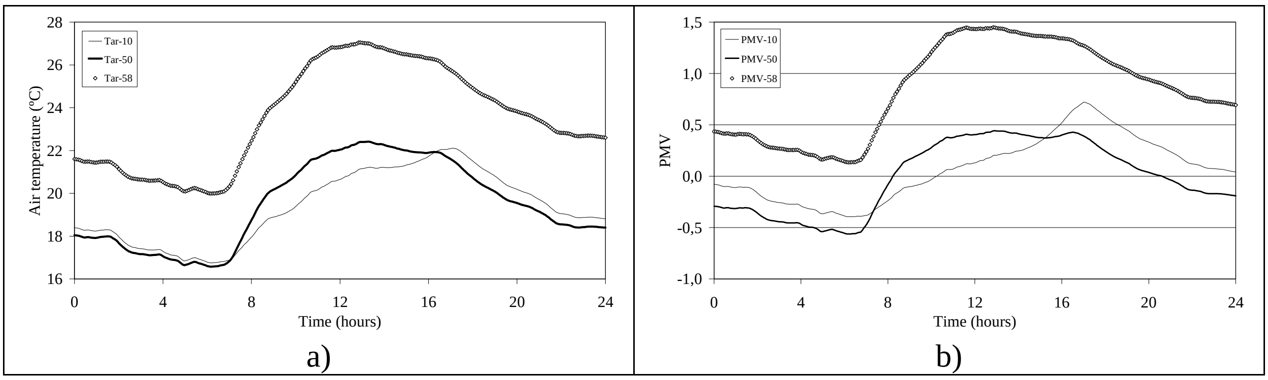

Figure 4. Internal air temperature a) and thermal comfort level b) evolution for the laboratories.

In figure 4 the laboratories 10, 50 and 58 are located, respectively, in the first floor in the West block with windows turned East and West, in the first floor in the East block with windows turned East and West and in the first floor in the East block with windows turned East, West and South.

In figure 5 the offices are located in the second floor in the North block, with windows turned West. The window of the office number 81 is shaded by the West block, while the windows of the office number 82 are not shaded.

In figure 6 the corridor number 30 is located in the first floor, while the corridor number 90 in the second floor. Both main corridors, located in the East and North blocks, are equipped with windows turned South-West.

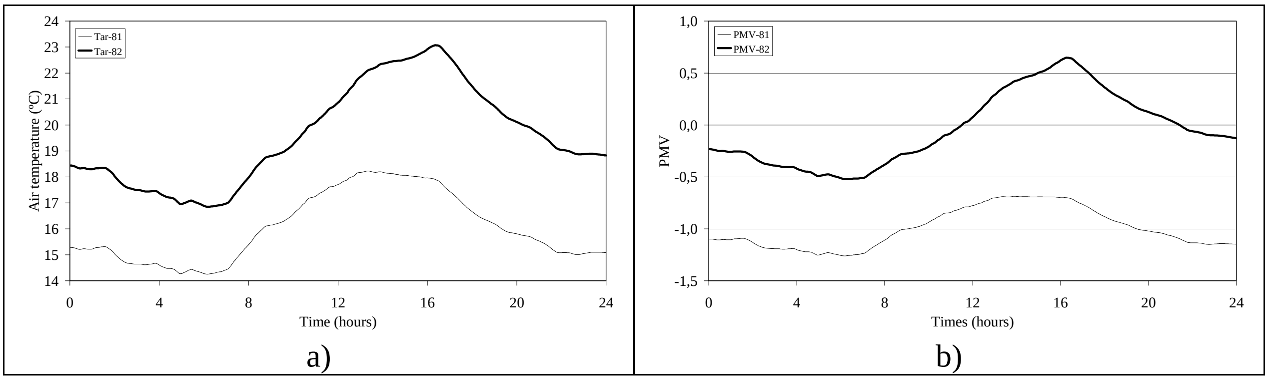

Figure 5. Internal air temperature a) and thermal comfort level b) evolution for the offices.

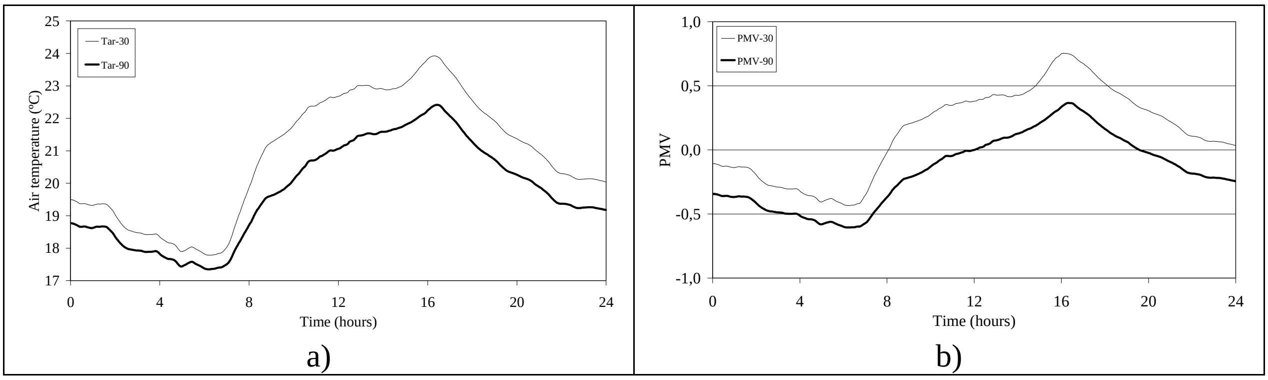

Figure 6. Internal air temperature a) and thermal comfort level b) evolution for the corridors.

In accord to the obtained results is possible to conclude that:

- In classrooms with windows turned East the air temperature increases mainly in the morning, while in the classrooms with windows turned West the air temperature increase mainly in the afternoon. Nevertheless, the shading effect caused by the East block in the morning and the West block in the afternoon is very important;
- The classrooms with windows turned East, shaded by the East block in the morning, the thermal comfort levels are in accord to the Category B of [5]. In the classrooms with windows turned West, shaded by the East block in the afternoon, the thermal comfort level is also in accord to the Category B of [5].
- When the windows are not shaded by the East block, uncomfortable thermal comfort conditions, by positive PMV values, are verified. When the West windows are not shaded by the West block, uncomfortable thermal comfort conditions, by positive PMV values, are also verified. Thus, it is recommended to use internal shading devices in compartments with windows not localized between the blocks.
- In the classroom with windows turned North (see room number 18 in figure 3) the minimum thermal comfort level is not obtained. Thus, is recommended in this compartment to introduce a window turned to West direction;

- In the laboratories, in the first floor, with windows turned to East and West, with some external shading devices caused by columns and blocks, acceptable thermal conditions, are verified. Nevertheless, when the laboratories are built with windows turned East, West and South (see room number 58), uncomfortable thermal comfort levels are obtained by positive PMV values;
- The conclusions verified in the small rooms, like work and help laboratories rooms, are similar to the previous verified conclusions;
- The offices, located in the North block, subjected to shading devices due to the West block are thermally uncomfortable, while the others not subject to shading devices are thermally comfortable;
- In the main corridors, with windows subjected to solar radiation, thermal comfort conditions are verified, while in the others this is not verified;
- Similar conclusions are verified in the other spaces. Nevertheless, the other spaces (like auditorium, canteen, students' room, teachers' room and library) with high occupation level are suggested to be analyzed in a numerical simulation with occupation.
- In other spaces, located in the second floor in the North block, like secretary and offices with windows subjected to shading devices, the introduction of a heating system is recommended.

School Building for Infants

In this study two kinds of simulations were made: with and without surrounding buildings. In figure 7 the evolution of the air temperature value, in compartments with windows turned to West (room 18) and East (room 11), with and without surrounding buildings are presented.
In the results presented in figures 8 and 9 the surrounding buildings effects are considered. The evolution of the air temperature and thermal comfort level for the classroom is presented in figure 8. Finally, in figure 9 the air temperature evolution in the atria, corridors and playground are presented.
The classroom number 3 is equipped with windows turned South, the classroom number 13 is equipped with windows turned East and North, while the classroom number 14 is equipped with windows turned West and North.
The atrium number 2 is equipped with windows turned to South and West, the corridor number 25 is interior and playground number 10 is equipped with windows turned to East.

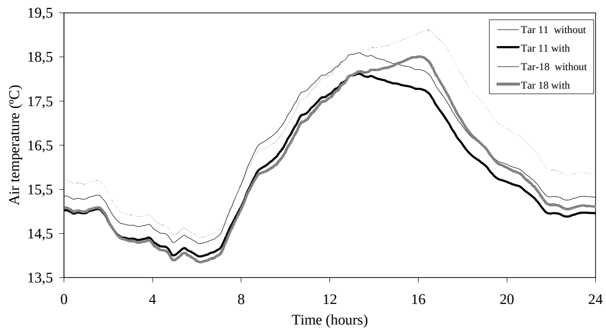

Figure 7. Internal air temperature with and without surrounding buildings.

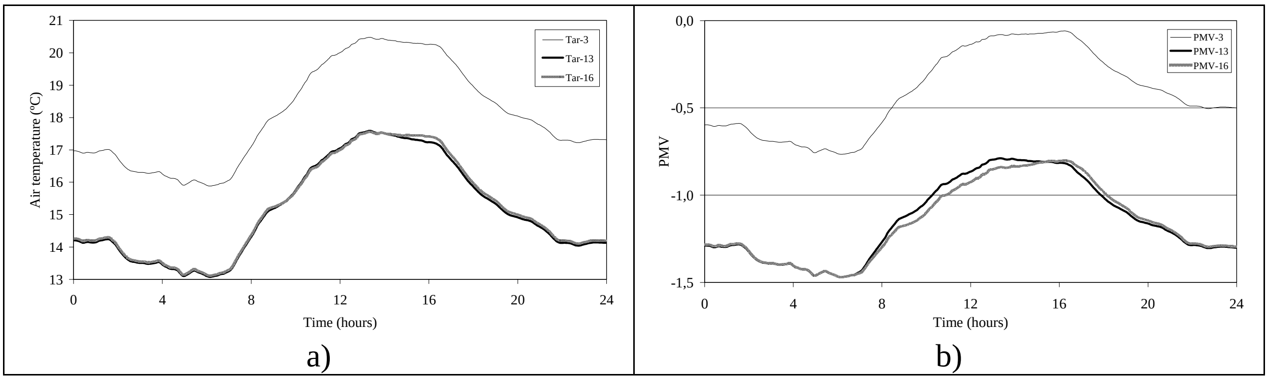

Figure 8. Internal air temperature a) and comfort level b) evolution for the classrooms.

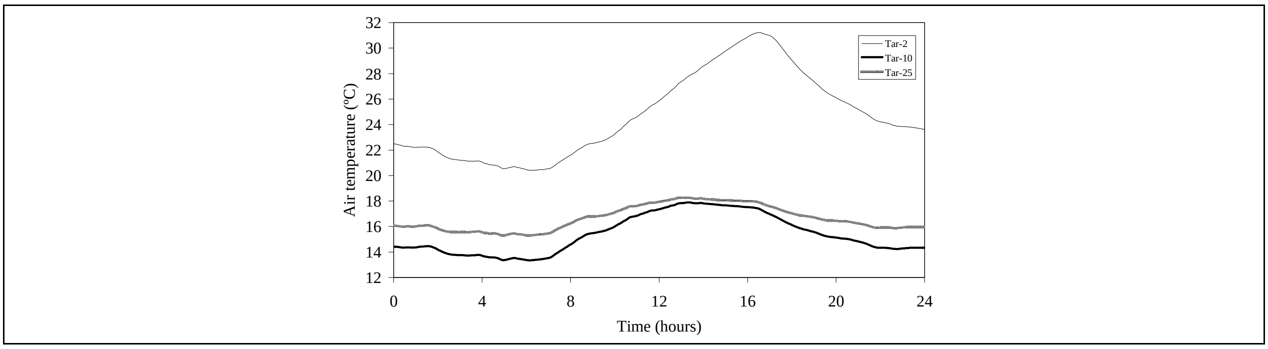

Figure 9. Internal air temperature for the atria, corridors and playground.

In accord to the presented results is possible to conclude:

- It is important, in a numerical simulation, to consider the surrounding buildings. In general, the introduction of the surroundings buildings decreases the internal air temperature level. The main difference in compartments with windows turned East is verified around 9 a.m., while in compartments with windows turned West is verified around 17 p.m.;
- In the classrooms with windows turned South, thermal comfort levels by negative PMV values are verified, nevertheless, in classrooms with windows turned North the thermal comfort levels are not verified;
- Finally, the air temperature in the atria area, verified during the day, is very high. Thus, in future works is suggested to consider the atria spaces as air collectors and to use this air to heat the classrooms with windows turned North.

CONCLUSIONS

In this work a numerical model, that simulates the building's thermal response and evaluates the internal thermal comfort and air quality, in transient conditions, is used in the thermal project of two school buildings, located in the South of Portugal, used by teenager and infant students.
It was verified in school buildings for teenagers, in classrooms, help laboratories and work rooms with windows located between the blocks, in general, are subjected to shading devices in the morning (caused by the East block) and in the afternoon (caused by the West block). The rooms with windows not located between the blocks are thermally uncomfortable by positive PMV values. It is suggested to introduce, in these windows, internal shading devices. Other particular situations as uncomfortable conditions are verified in spaces with windows

turned South, by positive PMV values, in spaces with windows turned North, by negative PMV values, and in spaces with windows shaded by other blocks, by negative PMV values.
In the school building for infants was verified that the surrounding buildings must be considered in the numerical simulation. It was also verified that the classrooms with windows turned South are thermally comfortable, nevertheless the classrooms with windows turned North are thermally uncomfortable. It is suggested in future simulations to consider the warm air in the atria to increase the thermal comfort level in classrooms with windows turned North.
In future works, more simulations are recommended, with inclusion of the occupants and ventilation, and other validation tests, mainly considering the surrounding buildings shape. The presence of occupants in the simulation will increase the thermal level and the introduction of the air ventilation decrease this level. Nevertheless, in spaces with high occupation level is important to introduce these changes.

ACKNOWLEDGEMENT

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Numerical Models for Simulation of Space Thermal Energy Demand

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SUMMARY

The paper presents soft computing methods developed for simulation and prognosis of space heating energy consumption using statistic and neural networks models.

The statistical model is a nonlinear one, based on weighing of the input variables: the outdoor temperature, the indoor temperature, the wind velocity, the supply temperature at the exit of the substation, the average fluid flow rate one hour ago and the outdoor temperature T_τ at a previous time. A detailed analysis that is referring to the heat transfer through the inertial structural elements is developed.

Artificial neural networks (ANNs) are computational systems that recognize patterns based on the data used to train them and may be used to make predictions of the output value for new different input data. The training and prediction results of proposed ANN in comparison with statistical model are presented. For both models, predicted and experimental values of heat load demand are well matched and highlight the success of applying statistic and neural networks models in predicting.

The methods were verified by comparing the modeling results with acquired data via a monitoring system from the District Heating Company of the city of Iasi (Romania).

INTRODUCTION

Sustainable development and increase of energy efficiency are important objectives for European energy research area. About 250 district heating systems work in Romania. Even if a lot of consumers had preferred individual heating, 54% of urban population still uses district heating networks. Documents regarding national saving energy policy stipulate that if 1 Euro is invested for increasing the energy efficiency, then a decrease of 1.26 Euro for acquisition of primary resources is realized. Implementation of national strategies regarding district heating production and distribution [1] is in progress, and financial support for numerous projects concerning rehabilitation and modernization of thermo energetic domain already generated hundreds of millions of Euros investments.

The most important objectives for the Romanian district heating system rehabilitation policy are the improvement of the energy efficiency and technical performance and the diminution of costs and pollutions. The recommended measures require institutional, technological, social, market, and financial decisions. Some of the most important are [2], [3]: process control should be updated according to the actual level of technology, the district heating enterprises should implement an integrated monitoring system involving a database together with regular and systematic control of the network and the substations, the customers should be given technical means and appropriate incentives to control their heating.

It may be noticed that monitoring the thermal energy consumption is essential for an efficient management. In Romania, some control systems exist at the heat source and in the substations, but only a few at the customers. This results in an imbalance of the production and the real need of heating, because little information is available from the customers' side.

In the last years, monitoring system started to be developed in Romania, so detailed information concerning the consumers' heat demand are available. It is time for a new step, the transformation of the district heating systems into demand driven systems. Therefore, soft computing programs are needed and this is the aim of this study. The paper presents soft computing methods created for simulation and prognosis of space heating energy consumption using statistic and artificial neural networks models. Such models allow evaluating the heat load dynamic of buildings. The validation was done by comparing the modeling results with acquired data via a monitoring system from the District Heating Company of the city of Iasi (Romania).

METHODOLOGIES FOR THERMAL ENERGY DEMAND SIMULATION

In order to match heat load demand to the thermal energy generated by a power plant it is essential to know heat load profile. Mathematical models are appropriate to predict heat request of buildings determined by climate and specific construction features. It is all it can be done in design stage, but when a district heating system works, subjective factors as family income, cultural background, ability to pay the bills may bring important changes to the daily heat load profile.

A method of formulating energy load profile for UK domestic buildings has been identified by Yao R. and Steemers K. [4] . They proposed to classify the energy demand into two types of determinants: a behavioral and a physical one. Their work focused on the behavioral determinants influenced by people's occupancy pattern, age, number of persons per household. Interesting results were obtained for energy consumption of domestic hot water, domestic appliances and lighting. For space heating load, a control algorithm related to occupancy pattern has been embedded within the thermal model based on physical factors such as building thermal characteristics, orientation, internal air temperature, control and local climate. Cluster analysis method has been applied based on different scenarios of occupancy patterns and the results are encouraging. Unfortunately, it is difficult to extend these results to Romania, not only because building characteristics are different, but mainly because pressing social problems generate different decisions of the inhabitants.

The prognosis of hourly heat load profile is very important for decentralised power production in "micro-CHP" installed in buildings, a modern solution that has been developed in the last years. For instance, A. Nystedt, J.Shemeikka, and K.Klobut [5] elaborated a computer program for calculating the heat demand of a building supplied by a micro-CHP and connected to the district heating network in order to sell the excess heat. The mathematical model is based on classical engineering equations without disturbing human factors. The study recommends the use of these self-sufficient buildings in East and Central Europe, where a lot of blocks of flats are connected to a district heating network and at the same time to a natural gas network.

A realistic evaluation of the fuel consumption necessary for a District Heating System releases on knowing the total heat request of the users. Barelli L., Bidini G., Pinchi E. M. [6] developed a model of the thermal daily and hourly load of a neighbourhood, based on experimental data. The statistical simulation model uses the power installed in the thermal power plant, the seasonal operation hours, the timetable of the heating service distribution and the outdoor temperature as input parameters. The calculated values differs from the corresponding real data of about 5.5%, a value that was considerate acceptable by the authors.

NOMENCLATURE

a	thermal diffusivity [m^2/s]	T	actual temperature [K]
A	area [m^2]	T_τ	previous outdoor temperature [K]
h	convection heat transfer coefficient [W/m^{2}K]	T_{SP}	temperature at the exit of the substation [K]
k	thermal conductivity [W/mK]	V	wind velocity [m/s]
Nu	Nusselt number	x	spatial coordinate [m]
q	heat flux [W/m^2]	δ	thickness [m]
q_a	average mass flow rate [kg/s]	τ	inertial time [hours]
Q	heat load [W]		
R	thermal resistance [m^{2}K/W]	*Subscripts*	
Re	Reynolds number	i	indoor
t	time [s]	o	outdoor

This paper studies two mathematical models for simulation and prediction of the heat load demand of buildings: a statistical one and an artificial neural network one. The models release on six input parameters: the outdoor temperature T_o, the indoor temperature T_i, the wind velocity V, the supply temperature at the exit of the substation T_{SP}, the average fluid flow rate one hour ago q_a, the previous outdoor temperature T_τ.

The statistical model for calculation of the heat load demand (1) is a nonlinear one, based on weighing of the input variables by means of six constants $k_1 ... k_6$ which may be calculate for each building by regression analysis

$$Q = k_1 + k_2 T_o + k_3 \left(T_i - T_o\right) V^{\frac{4}{3}} + k_4 T_{SP} + k_5 q_a^{\frac{4}{5}} + k_6 T_\tau \qquad (1).$$

The first two terms and the last one illustrate the heat transfer through the building envelope, walls, ceiling, glass, roof and floor. The previous outdoor temperature T_τ introduces the thermal inertia that depends on the structural characteristics of the building. A detailed analysis that is referring to the inertial time spent for heat transfer through the inertial structural elements is presented in the following paragraph. The third term $k_3(T_i - T_o)V^{4/3}$ represents the heat required for air infiltration losses. Generally, the buildings from Romania are characterized by reduced window tightness so that the losses corresponding to the heating of the cold infiltrate air can represent an important component. The fourth term $k_4 T_{SP}$ is used as a corrective term, because the supply temperature at the exit of the substation varies when the values settled for the control loop change. The term depending on the average fluid flow rate q_a indicates that the previous average fluid flow rate has a direct effect on the heat load demand. The heat convection through the distribution pipes and radiators depends on the correlation $Nu = f\left(\mathrm{Re}^{4/5}\right)$, that is equivalent to $Q = f\left(q_a^{4/5}\right)$.

The temperature changes affect the indoor conditions of the buildings after a specific time τ, called inertial time, in this paper. This time parameter depends on the structure of the exterior walls namely on the thickness and thermal conductivity. Because of the heat transfer through inertial building elements, the actual heat load demand at a specific time t will be influenced by the outdoor temperature at a previous time t- τ. The delay may be calculated using the differential heat equation for a plane wall without thermal energy generation:

$$\frac{\partial T}{\partial t} = a \frac{\partial^2 T}{\partial x^2} \ . \tag{2}$$

In order to find the temperature distribution through the wall $T(x,t)$, it is necessary to specify an initial condition and two boundary conditions for asymmetrical thermal load. By the Bernoulli integration method, an exact solution of the equation (2) has been obtained and finally the following relation for the heat flux on the inner surface of the wall [7]:

$$q = -k \frac{\partial T}{\partial x}\bigg|_{x=\delta} = \frac{T_i - T_o}{R} - \sum_n D_n h_i \exp\left(-a m_n^2 t\right). \tag{3}$$

The infinite series solution can be approximated by the first term of the series and thus the heat flux relation becomes

$$q = \frac{T_i - T_o}{R} - D_1 h_i \exp\left(-a m_1^2 t\right). \tag{4}$$

For a given wall structure, the D_1 parameter depends on: the initial temperature distribution through the wall, the constant indoor temperature, the instantaneous outdoor temperature.

The initial temperature distribution into the wall is calculated within the equation (5) and illustrates a steady-state heat transfer characterized by temperatures T_i and T_τ

$$f(x) = T_i - \frac{T_i - T_\tau}{R}\left(\frac{1}{h_i} - \frac{x}{k}\right), \tag{5}$$

where T_τ is the outdoor temperature corresponding to the initial conditions, respectively outdoor temperature at the previous time t- τ and R is the thermal resistance calculated with the equation

$$R = \frac{1}{h_i} + \frac{\delta}{k} + \frac{1}{h_o}. \tag{6}$$

The m_1 constant depends on: the thermal conductivity k, the thickness δ and the convection heat transfer coefficients h_i and h_o.

The final form of the heat flux equation due to thermal inertia is:

$$q = \frac{T_i - T_o}{R} - const\left(T_\tau - T_o\right)\exp\left(-a m_1^2 t\right). \tag{7}$$

In table 1, the values of the parameter time τ for different building structures are calculated. Thermal insulation consists of extruded polystyrene with thickness δ = 0.084 m and thermal conductivity k = 0.044 W/mK.

Table 1

Wall structure	Thickness [m]	Thermal resistance [mK/W]	Inertial time [hours]
Reinforced concrete	0.29	0.3095	8
Vertical cavity bricks	0.24	0.487	8
Clay bricks	0.4	0.887	12
Reinforced concrete + thermal insulation	0.20+0.084	2.076	9
Vertical cavity bricks + thermal insulation	0.24+0.084	2.396	9

In this study, numerical simulation procedures based on the statistical model (1) are developed for a reinforced concrete structure of 0.29 m thickness, a type of building common in Romania. For this type of wall the inertial time is 8 hours.

Data acquired in the period 1-31 December 2006, with a monitoring system from the District Heating Company of Iasi (Romania), has been used for simulation and prediction. The constants k_1, k_2, ...k_6 were estimated by regression analysis using the *nlinfit* function from Statistics Toolbox of MATLAB.

The second approach is focused on the artificial neural networks (ANNs) modeling. ANNs are built from a large number of very simple processing elements, neurons that individually deal with pieces of a big problem. A neuron multiplies an input by a set of weights, and nonlinearly transforms the result into an output value. The power of neural computation comes from the massive interconnection among the neurons and from the adaptive nature of the parameters (weights) that interconnect them. The neurons are connected in different topologies. Typically, an ANN has an input layer, at least a hidden one and an output layer.

A backpropagation ANN is the most common type of neural network and it undergoes its learning phase by calculating the error between the predicted and measurement values. The ANN topology chosen for modeling incorporates a hidden layer that is used to establish the relationships between the input variables and the output to minimize the error between the measurement and predicted output. The hidden layer has 27 neurons, the activation function is tanhsigmoid and the learning rule used is delta rule.

In the training phase the correlative patterns between various inputs and the corresponding outputs for different patterns of input data were identified and learnt by the ANNs. The epoch size is the number of training cases the model goes through before it readjusts the weightings of the processing elements that determine the value of the output. Once the NNs has been trained, the weights are frozen, the testing set is fed into the network and the network output is compared with the desired output in order to validate the ANNs performance.

RESULTS

Several buildings were chosen for this study. Some results obtained with the statistical model (1) based on data from six representative days are presented in Table 2. The error parameter was determined with the equation

$$r = \left| \frac{Qm - Qe}{Qm} \right| \cdot 100\% \tag{8}$$

where Qm and Qe represent the measured and the estimated thermal energy demand for an entire day. The mean square error (R) and the relative error parameter (r) for the simulation model were calculated for every building.

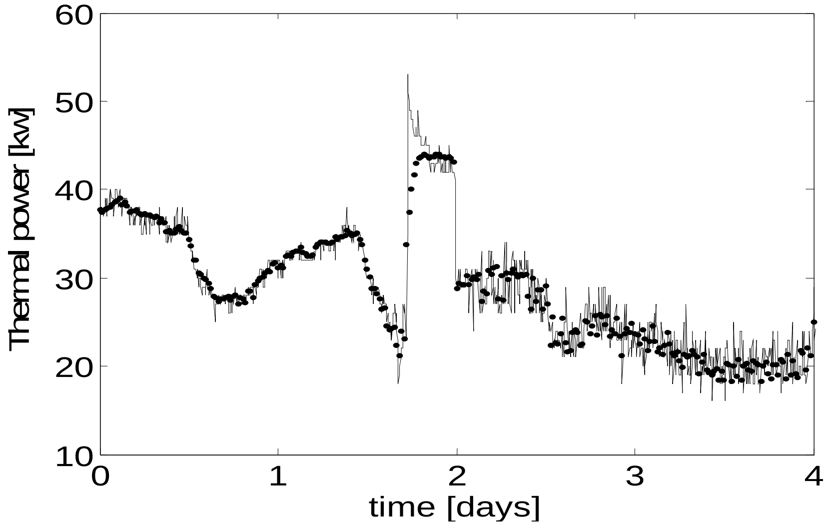

Fig. 1. The output of the trained ANNs (♦) and experimental data (—) for building B1.

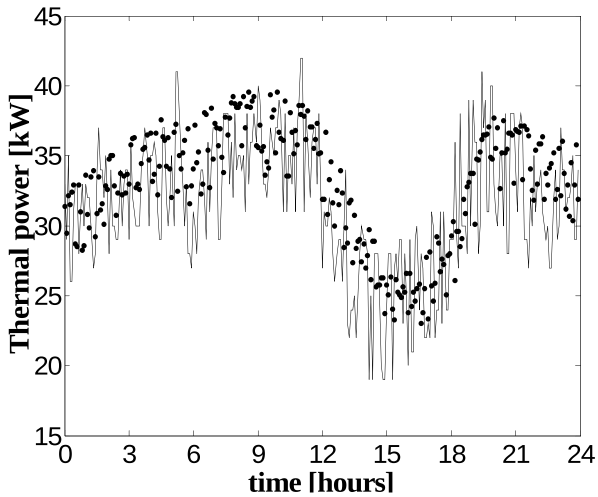

Fig. 2. The predicted values (♦) with SM and experimental data (—), 2nd Dec. 2006, building B2.

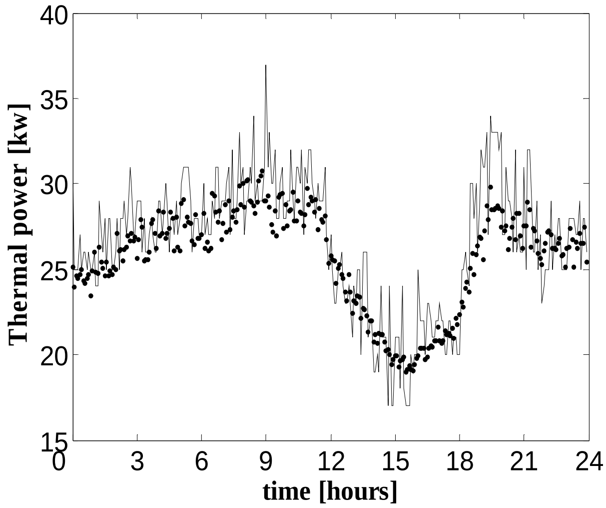

Fig. 3. The predicted values (♦) with ANNs model an experimental data (—), 2nd Dec. 2006, building B1.

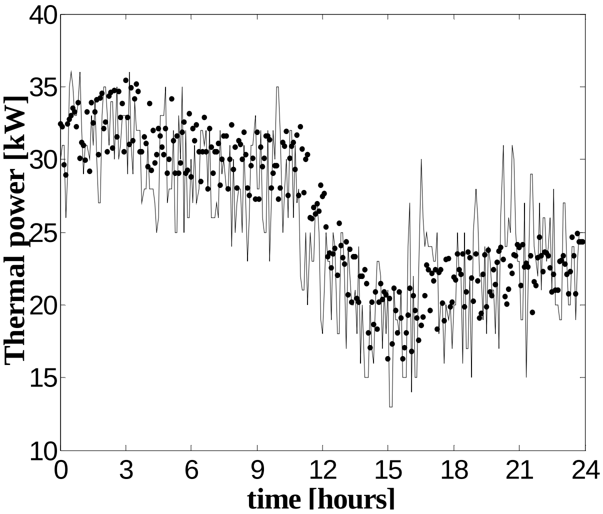

Fig. 4. The predicted values (♦) with SM and experimental data (—),14th Dec. 2006, building B2.

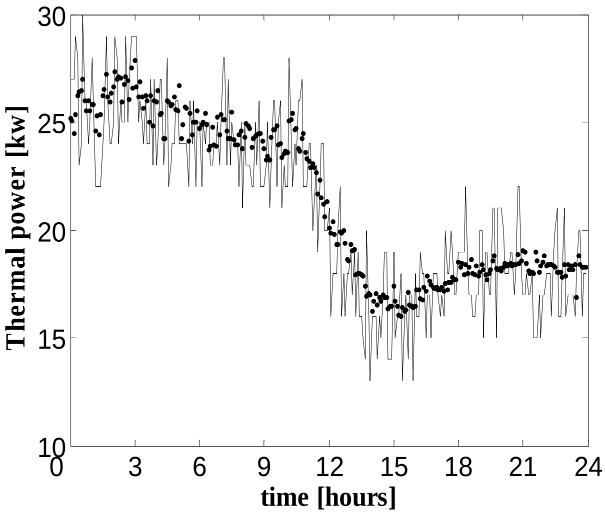

Fig. 5. The predicted values (♦) with ANNs model an experimental data (—), 14th Dec. 2006,building B1.

Table 2

	k_1	k_2	k_3	k_4	k_5	k_6	R	r
B1	-35.495	0.0655	-0.0226	-0.0013	0.1535	1.1194	0.9491	0.0036%
B2	-39.504	-0.0229	0.0331	0.0008	0.0204	1.4067	0.9232	0.0046%
B3	-28.760	-0.5250	-0.0474	-0.0024	0.1373	1.0220	0.9095	0.004%
B4	-24.661	-0.1146	-0.0663	-0.0019	0.2562	1.1038	0.9518	0.0026%

The purpose of this work is to develop accurate prediction models. The values of the relative error parameters, both for the statistical model and the artificial neural networks models are presented in table 3.

Table 3

r%	Statistical Model		Articifial Neural Networks Model	
	2nd December 2006	14th December 2006	2nd December 2006	14th December 2006
B1	2.34	3.04	3.76	2.14
B2	4.35	3.16	3.96	3.29
B3	0.60	1.18	0.44	0.05
B4	0.58	2.94	4.38	0.33

It can be noticed that the differences between the measured and the predicted energy during an entire day is less than 4.5%, an error considered acceptable [6]. In order to find if these models may be used to predict the heat load dynamic of buildings some graphs were drawn.

In figure 1 an artificial neural networks model is presented. The mathematical model pursues with accuracy the experimental data. Figs 2-5 show the experimental and the predicted values in the test process of the statistical and the artificial neural networks models. The predicted values were obtained by using experimental data for other days not considered initially as representative. Details regarding the difference between measured thermal energy marked with a thin line and predicted data marked with points indicate a remarkable match between the aspect of the graph for the measured data and the profile of the graph for the predicted data. This observation indicates that statistical and artificial neural networks methodologies presented in this paper are recommended not only to appreciate the necessary heat load for a future day, but also to predict the heat load dynamic of a building.

CONCLUSIONS

The paper studies soft computing methods created for simulation and prognosis of space heating consumption. Because of overcome problems connected with constructing a reliable analytical model for buildings heat load demand, a statistical model and an ANN one were studied. This approach is strongly based on the accuracy of a large collection of acquired data via a district heating monitoring system. The models were verified by testing different experimental data collected in a substation from the District Heating Company of the city of Iasi (Romania).

The statistical model is a nonlinear one, based on weighing of the input variables by means of six constants which may be calculate for each building by regression analysis. A detailed analysis that is referring to the inertial time spent for heat transfer through the inertial structural elements is presented.

An ANN with backpropagation topology, one hidden layer, the sigmoidian activation functions and generalized delta learning rule was chosen. For each building, the ANNs were trained for 3000 epochs and the obtained results were very good, this meaning correlation coefficients higher than 0.95.

The error parameter calculated for both models is less than 4.5%, denoting that errors introduced by data processing in models construction do not exceed the inherent errors resulted from data acquisition. The graphs drawn with measured and predicted data indicate that the models are appropriate also for the prediction of the heat load's dynamic.

ACKNOWLEDGEMENT

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Defining the governing parameters for reliable numerical simulation of smoke evacuation in underground parking garages

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SUMMARY

This paper discusses several issues concerning the CFD analysis or car parks during fire situations. A common modeling approach and given parameters are needed to obtain reliable results for the indication of the performance of the smoke evacuation systems. There are many modeling parameters which influence the results. CFD is a powerful tool that can be used for smoke evacuation simulation when properly used. There is a clear need for a simulation guideline that describes the following points:

- Demanded mesh sizes
- Detailed fire curve
- The fire material composition and particulate soot yield.
- Description of visibility calculation method.
- Turbulence model
- Fluid properties
- The time steps to be analyzed.
- Parameters to be analyzed (including scale).
- Clear description of demanded performance.

For the different steps in the modeling process parameters of importance are discussed. An advisable use of parameters is given either based on practical experience or on literature.

INTRODUCTION

Due to the increasing population density in the urban environment, the need for car parking spaces is growing. More and more large scale underground parking lots are being developed world wide. With this growth the concerns regarding safety within enclosed parking lots arose. Especially fire and smoke control became more difficult during the development of ventilation systems and legislation. Legislation mainly is based and developed upon dimensioning rules and simple calculation. Followed by implementing demands and wishes of the local authorities [1]. Nowadays mostly CFD (Computational Fluid Dynamisc) is used to verify the smoke and fire control by the proposed ventilation system.
The application of commercial CFD packages in verifying the efficiency of the safety ventilation systems has grown significantly in recent years. However, the available calculation procedures allow the user a very wide margin of freedom in the selection of the algorithms and related coefficients to be used in each case, which can lead to appreciable deviations in the end results [2, 3]. The inexistence of standards at European level, which could set clear criteria and guide the analyst through the simulation process, adds to this effect. The flexibility of the CFD tool is one of its biggest advantages but also its biggest threat, as the number of variables and parameters that it manipulates is significant and must be adapted to each type of calculation. It is therefore important that standard procedures are defined in order to ensure the reliability of the results.

The study presented in this article investigates and correlates several calculation techniques in order to define the complete calculation procedure. It includes several practical recommendations for preparing and running complete simulations, which were identified through extensive experience in this field.

RESULTS AND BEST PRACTICE

The simulation to predict the performance of a parking ventilation system is conducted in several steps. Per step several choices can be made which have different levels of consequences for the end results. The modeling steps are discussed briefly including the possibilities for the simulation tool user. Furthermore the choices for best practice are proposed.

Building and meshing the simulation domain

When translating the architectural and ventilation design drawings in a 3-dimensional model an adequate level of detail should be chosen in order to obtain reliable results. The level of detail affects the number of cells and will eventually affect the computation time. Small structures result in small cells. When defining the importance of details a simulation expert should consider and or research the effect of an object on the fluid flow. In case of a parking it is of importance to at least include beams and pillars. Furthermore the architectural design characteristics need to be evaluated. If the width of an object is smaller then 100 mm it is possible to neglect the object, if it is reasonably assumable that the flow will not be significantly influenced by the object. Beams are of particular importance because of the ceiling jet principle: due to the high temperature of the fire, a small layer of smoke is distributed at high velocity along the ceiling [4].

The fluid volume within the parking will be divided in a mesh grid in order to solve the Navier-Stokes partial differential equations. In many CFD packages the finite volume method is used for representing and evaluating these partial differential equations as algebraic equations. Values are calculated at discrete places on a meshed geometry. "Finite volume" refers to the small volume surrounding each node point on the mesh. In this method, volume integrals in a partial differential equation that contain a divergence term are converted to surface integrals, using the divergence theorem. These terms are then evaluated as fluxes at the surfaces of each finite volume.

The properties of the mesh are critical for the quality and reliability of the results but the available guidelines on this subject do not set clear statements regarding the mesh properties to be used for car fire simulation. There are different types of finite volume elements that can be used, to be known hexahedron, tetrahedron, prism and pyramids (figure 1).

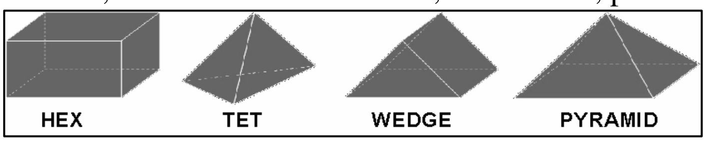


Figure 1- mesh types.

During the performed literature study there are no studies found regarding a comparison between the different mesh types. Based on practical experience it is advisable to mesh the fluid volume with tetrahedron and at least 3 prism layers along the ceiling. A tetrahedron mesh is unstructured, allowing the user to build complex geometries and locally dense mesh. The prism layers are used to simulate near wall flow accurate. In case the number of cells exceed the possible calculation number it is advisable to use a combined structured and unstructured mesh (hexahedron, prism and tetrahedron) mesh to reduce the number of volumes and nodes [5].

Special attention should be given to the mesh definition near the fire and the ventilation system components. For these locations it is advisable to use mesh sizes between 30 and 300 mm, depending on the complexity of the garage. The surface mesh should at least consist of 3 cells in each direction. But the mesh size has a critical influence on the processing calculation resources and calculation time. Care must be exercised not to exceed the number of cells that can be solved with nowadays computers.

Defining boundary conditions

The flow and dispersion of smoke within the fluid volume depends on the assumptions made for the boundary conditions. The boundary of the model can have an energy and or mass influence on the volume. Over the construction itself only a heat transfer can occur. Through openings, supply grills and exhaust grills mass transfer will be established. Furthermore in case of fire and smoke modeling an extra source within the model will be introduced. In the guidelines for car fire simulations in closed parkings there are no practical recommendations on how to model these components.

Supply grills can be modeled by representing the flow characteristics on either the grill surface or on a surrounding box. In case of fire modeling density differences will occur, which can influence the impulse of the supply air in case of the box method.

In the different guidelines there are no statements or recommendations made regarding the modeling approach. It is assumable that the slab is a heavy construction. Accumulation of heat in the concrete structure will influence the temperatures within the fluid volume. The temperature differences within the solid domain are large. It can be assumed that 3-dimensional conduction will occur. Most CFD packages include a simple wall heat transfer model. The software in that case calculates the 1 dimensional temperature difference dependent heat transfer according to the average heat transfer coefficient. In case of fire modeling this will over exaggerate the energy loss from the parking. Resulting in under estimating the temperatures in the fluid domain. In case a solid is taken into account in the model energy losses will be influenced by the accumulation and three dimensional heat transfer.

Fire modeling

For modeling the fire itself, two procedures are available: the combustion model and the inert fire model, each with its own advantages and disadvantages. When using a combustion model the different materials and their properties need to be added to the simulation. Once a predefined part of the car catches fire, the fire spread is calculated as a function of the spatial distribution of temperature, fuel, oxygen and their variation in time. Although this method is the most realistic, the calculation time and complexity of this type of simulation are very high and the results difficult to interpret. An added obstacle is that it requires a very large amount of input parameters that are normally not available.

In the inert model, a flame domain is created in which the energy is released. Smoke is added to the model in the ‘ground’ surface of the flame domain. Fire growth and decay can be controlled by the user as well as the smoke production, whose mass flow is in this case related to the rate of heat release from the car fire. This enables designers to compare results for different systems and or different parking layouts.

In both cases radiation is excluded from the model. It is not possible to correctly model the radiation from a smoke cloud, due to the lack of reliable data.

The inert model gives a good representation of the situation occurring during a fire resulting from a fixed burning process. Beside the fact that the results are comparable to experimental data [5, 6], it is also the best choice regarding time efficiency. The flame model can be presented on top of the car in which the energy is released. It is important to use fire curves per car, since the energy is released in that particular flame domain. The curve shown in figure 2 represents experimental data of the heat release rate for one car based. Depending on the local legislation the curve can be changed regarding the total energy and mass assumed to be burned during the fire. Nevertheless the growth and decay are equal. The criteria for defining flash-over between the first and second car depends on local legislation. It is advisable that the second car catches fire after 8 minutes and the third 4 minutes later. In a discussion with local authorities a maximum time for the fire department to extinguish the fire should be set, normally between 20 and 25 minutes. The period after extinguishing the fire should also be simulated in order to indicate the time needed to clear the garage.

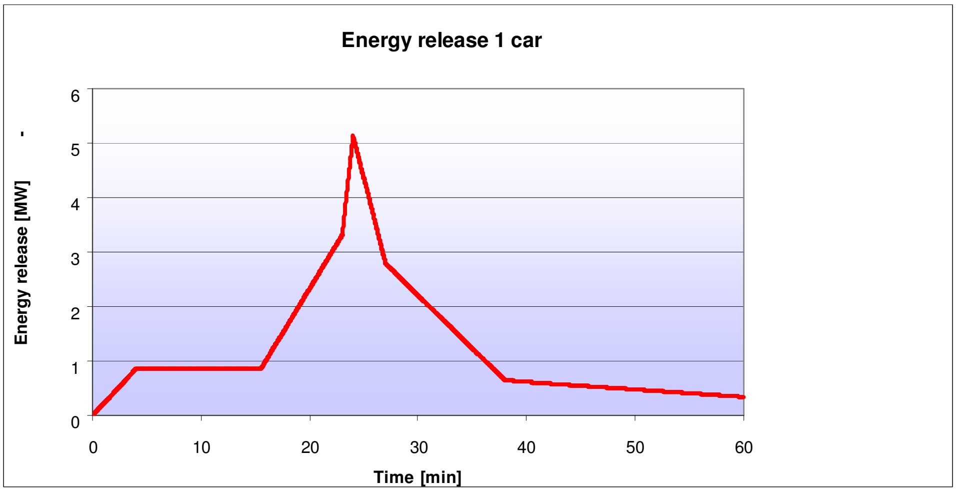


Figure 2- one car energy release curve

Smoke modeling

The most commonly used criterion to quantify the smoke spread and to judge the performance of the parking ventilation systems during a fire situation is the resulting visibility in the occupied spaces. The visibility in the smoke cloud is calculated as a function of the root mass per volume and the burned material. The calculation methods are based on empirical data resulting from several test performed to relate soot production to mass loss of specific material samples. The results of these tests are published in terms of particulate soot yield or the optical density of smoke [8, 9, 10]. The optical smoke density depends also on the type of combustion, this being fuel dependent or oxygen dependent, in other words, a well or poorly ventilated fire. It is known that the effect of smoke on the visibility is also depending on temperature; unfortunately, no empirical laws have been derived describing this effect. It is also not possible at this moment to calculate the coalescence of smoke particles when they collide.

The smoke mass flow is calculated as function of the total released energy (convective + radiation). The chemical reaction of the burning process is excluded and the smoke mass flow is equal to the mass loss of the fuel (car). A car fire in a parking is classified as a well ventilated fire, the fire is fuel controlled meaning that the oxygen level is not a relevant parameter.

The mass flow is calculated using equation 1:

$$\phi_{m\_smoke} = Q_{fire} / \Delta H_{eff}, \qquad (1)$$

ϕ_{m_smoke} = mass flow smoke production [g/s]

Q_{fire} = energy release car fire [kW]

ΔH_{eff} = Effective heat of combustion [kJ/g]

The smoke optical density corresponding to the assumed composition can be calculated as followed.

$$OD = \frac{y_{smoke} \cdot K_m}{\ln(10)}, \qquad (2)$$

OD = Optical density [m^2/g]

y_{smoke} = Particulate yield [g/g]

K_m = Extinction coefficient [m^2/g]

The particulate yield corresponds to the composition of materials. The Extinction coefficient for flaming combustion is 7600[m^2/g] [10].
Large scale car fire tests have been carried out by several institutes. The result of these tests is that the optical density of a car fire is estimated to be 400 [m^2/kg]. Results show that the value is in between 200 [m^2/kg] and 600 [m^2/kg] [11 - 17].
The smoke and energy are released in the flame domain. The local smoke temperature is calculated within the CFD program.
Because radiation is not simulated and adiabatic walls are used, a reduction factor is used to calculate the convective heat release. A value of 0.7 has proven to produce reliable results.
The formula for the energy in the sub domain now becomes:

$$Q_{conv} = 0.7 \cdot Q_{fire}, \qquad (3)$$

Q_{conv} = convective energy release [kW]

Sub models

Several additional choices must be made when conducting a simulation process, namely when selecting sub-models used. The most important of these is the turbulence model, of which the most commonly used is the Kappa Epsilon model. This model predicts the behavior of vortices smaller then the local mesh cells and while research has been carried out in order to investigate the bias created by this assumption, its results are commonly accepted.
More precise turbulence models are available, but these result in significantly more complex meshes and computing requirements, which with the computer technology currently available to engineering designers make their use impractical.

Fluid model

The fluid properties in the model should be based on the properties of an ideal gas. In this way the density difference due to temperature is included in the simulation. This is of importance for the exhaust fans, since they exhaust a volume flow. Of course the natural convection depends on density differences as well.

Turbulence model

Underground parking lots are mostly low structures with a large floor area. As discussed before the flow along the ceiling is of high importance. The near wall function is related to the used turbulence model. It should be considered if the used turbulence model gives a well prediction of the flow near the wall. Furthermore the flow around construction parts should be checked. More detailed turbulence model needs a more dense mesh, resulting in larger calculation times. With the current computer technology it is advisable to choice either for the Kappa-epsilon or the Shear Stress Transport turbulence model.

Simulation result analysis
The advanced sophisticated simulation software currently available allows for both steady state and time-dependent (transient) solutions. For the type of problems and phenomena modeled in the case of a car fire, transient simulations must be performed.

The performance indicators used to evaluate the parking ventilation system are visibility [9, 10, 12] and temperature. The effect of soot on visibility depends on the dilution and the burned material. In general the visibility is calculated as follows:

$$S = \frac{C}{K_m \cdot y_{smoke} \cdot \rho_{smoke}}, \qquad (4)$$

C	=	Empirical factor	[-]
S	=	Length of sight	[m]

The empirical factor depends on a large amount of factors, e.g. light-emitting or light-reflecting characteristics of signage elements, individual visual acuity, colour and others. In practice, the following values can be used:

C	=	3	light-reflecting signs
C	=	8	light-emitting signs

The particular yield and extinction factor are both very much influenced by the type of fire (ventilated, smoldering, material etc). Therefore it is more accurate to combine formulas 2 and 3 and use data gained by experience.

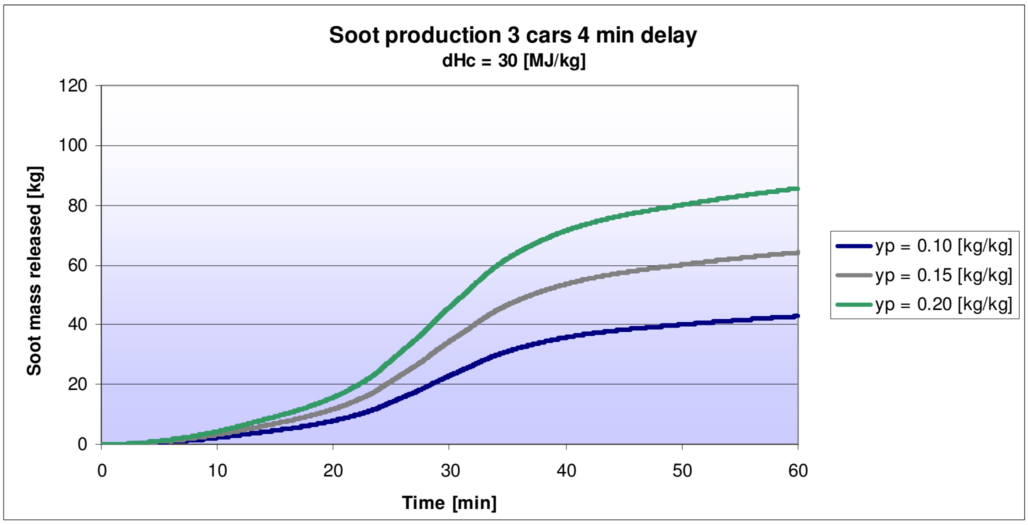
$$S = \frac{C}{\ln(10) \cdot Dm \cdot \rho_{smoke}}, \qquad (5)$$

The temperature is used to indicate the situation created by the fire and to evaluate the possible damage to the parking structure. Nevertheless, as the parking is modeled adiabatic the modeled temperature on the construction surface will be higher then what will actually occur.

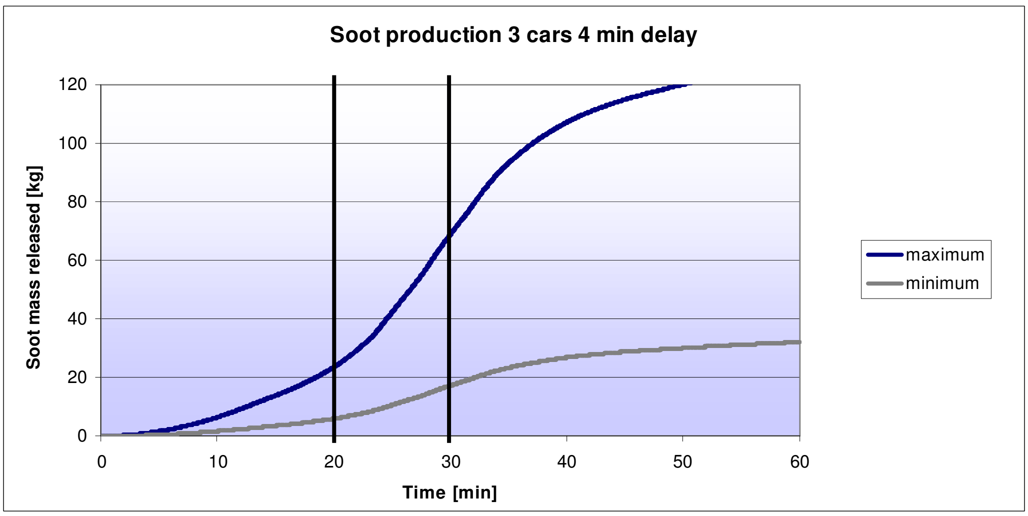
Parameter analyses
As for each simulation and calculation applies rubbish in is rubbish out, this as well counts for CFD simulations. In order to give an impression of the sensitivity of the smoke/soot production two parameters are analyzed for their maximum and minimum values [18]. As shown in equation 1 the smoke/soot production depends on soot yield, effective rate of heat release and energy release. The total energy release is kept stable for all cases. The most common soot yield is approximate 0.15 [kg/kg]. Material study shows that it can vary between 0.1 [kg/kg] and 0.2 [kg/kg]. The specific rate of heat release is common set at 30 [MJ/kg]. Depending on the type of car considered this can vary between 20 [MJ/kg] and 40 [MJ/kg] . The used fire for the analyses is a 4 minute delay of three cars according to figure 1.

Graphs 1 and 2 show the resulting smoke production in time by varying the parameters as described above. The total range by varying the parameters at the same time is shown in graph 3. The average assumed time for the fire brigade to arrive at the location is between the 20 and 30 minutes. This, as shown in graph 3, results in an absolute minimum smoke production of 5.8 [kg] and an absolute maximum of 68 [kg]. It can be concluded that the system is highly sensitive to these values. As appointed earlier there is a need for clear guidance to create parkings that are equally safe in terms of fire hazards.

-graph 1-
Fixed energy release and particulate yield



-graph 2-
Fixed energy release and effective heat of combustion



-graph 3-
Range of results for fixed heat release

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Exploiting the thermal mass in an energy efficient building – a comparison exercise between IES Apache and TRNSYS models

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SUMMARY

Cooling modern office buildings significantly contributes to their CO_2 emissions during summer. One possible approach to improve their summer energy performance in moderate climates would be to cool the building's floor slab by passing cold air through it at night. During the day, warm ambient air is passed through the cold slab and thus it enters the air-conditioning system of the building at a lower temperature. In order to evaluate the energy performance benefits of such a concept, a combination of full-scale experimental studies, Computational Fluid Dynamics modelling (using ANSYS CFX) and network modelling (using IES Apache and TRNSYS) has been performed. Both IES Apache and TRNSYS models agree well in their predictions and the energy benefit of the concept is clearly demonstrated.

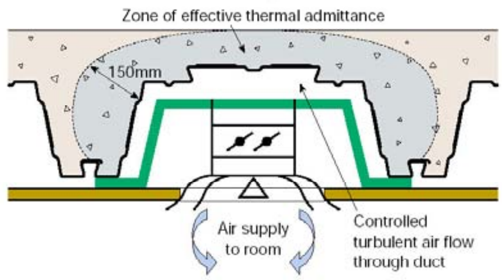
INTRODUCTION

Many modern office buildings need constant air-conditioning in the summer in order to remove heat gains and achieve acceptable levels of indoor thermal comfort. In moderate climates, one promising approach used to reduce the air conditioning energy demand of office buildings without reducing comfort is passive cooling by night ventilation in conjunction with utilising the thermal mass of the building. Passive cooling by night ventilation means passing cool night air through the office space to ensure absorption of the "coolth" by the building elements, such as the floor, walls, furniture etc. However, this system suffers from a lack of control of the amount of "coolth" absorbed and discharged by the office space. As a result the office space is often overcooled in the morning and may require additional cooling to bring temperature to a comfortable level. One way to minimize this effect is to introduce the cooling to the inner parts of the building elements e.g. by embedding air ducts within the floor slab.

The Air Cooled Slab (ACS) concept used in this work is based on a composite flooring system with air ducts, which would offer both an integrated service capability and a thermal storage system. Extensive experimental, Computational Fluid Dynamics and Whole Building Energy studies of the ACS were carried out under the European RFCS funded EEBIS project and details can be found in [1].

Throughout the nighttime during hot periods, cold outside air is passed through the ducts embedded in the slab thereby cooling it. Throughout the daytime, when cooling loads increase due to solar and internal gains and ambient temperature rise, external air is pre-cooled by passing it through the slab. The storage capacity of the slab depends on its thickness and its effective penetration depth. In a typical building with natural ventilation only a relatively thin depth of concrete (typically 50mm to 75mm) is effective for efficient heat transfer and storage

[2]. If the flow inside the slab is enhanced, the thermal capacity of the slab increases for a 24-hour temperature cycle to around 150mm [3], which is illustrated in Figure 1.
The climate in the UK has a large diurnal temperature swing, approximately 12K, with the night minimum temperature during summer being much lower than 22ºC. It means that there is a considerable potential for "coolth" storage during the night time and possibly enough to significantly reduce the need for an air-conditioning system in the building.
Figure 2 illustrates the three modes of operation that characterise the use of the ACS. In the night cooling mode, cold ambient air is forced through the channels using mechanical ventilation. During this mode the temperature of the slab is reduced to approximately 18-20ºC. In the morning, when the ambient air has a low temperature, it is used directly for ventilation (mechanical or natural, mode 2). The passive cooling mode 3 is activated during day when ambient temperature rises and solar and internal heat gains cause the office temperature to cause thermal discomfort. Warm ambient air is passed through the slab and could either be directly introduced (via floor or ceiling diffusers) in the office space or additionally cooled by the air-conditioning system of the building.



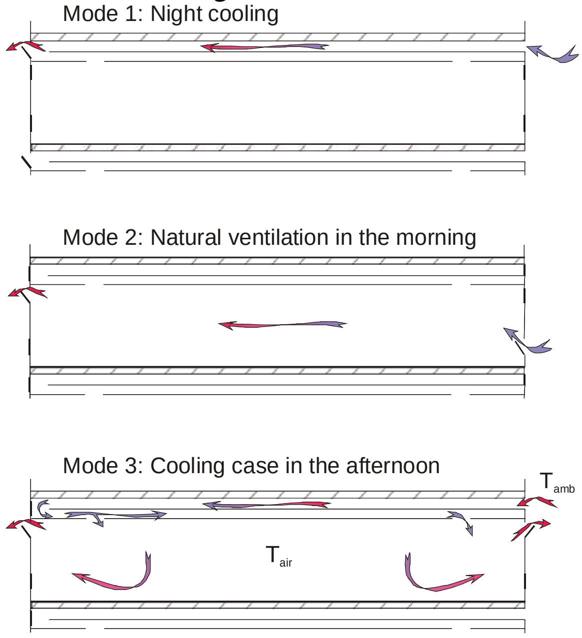


Figure 1 Zone of effective thermal admittance of the ACS (Variation with ceiling diffuser and distribution in the active trough)

Figure 2 Running modes of ACS

Figure 3 shows a cross section of the floor slab. In the inactive channels air does not have a direct contact with the slab and serve as distribution ducts only (Figure 3)

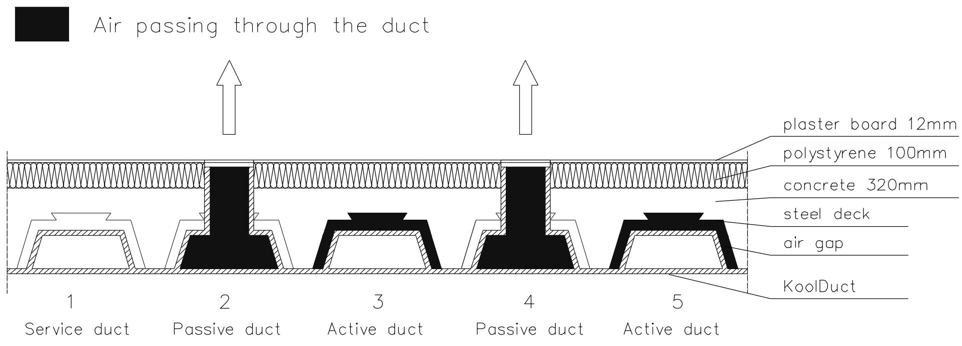


Figure 3 Cross section of the ACS. In the active ducts air has direct contact with the slab. The passive ducts are only used for distribution of the air into the office space and in these air does not have a direct contact with the slab. An additional duct is used for services.

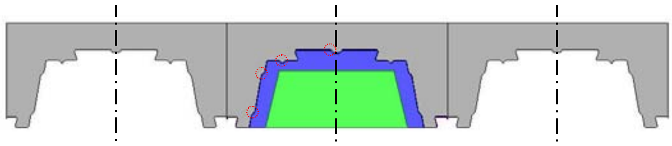
This article presents a numerical study of the performance of the ACS in a realistic energy efficient building. The main objective of this article is to compare the performance of the ACS using two different modelling approaches embedded into two different software packages for building energy analysis (TRNSYS and IES Apache). The rest of the article is organised as follows: the modelling section presents details of the models and their validation followed by a results section. The article concludes with a discussion section.

METHODS

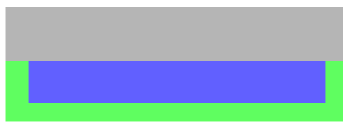
Development of a reduced model of the ACS for use with a whole building simulation models

As mentioned in the introduction of the article a series of experiments were performed on a full scale ACS prototype (12m long, 5 channels) and these results are presented elsewhere [1]. In parallel with the experiments a Computational Fluid Dynamics (CFD) model of the ACS was developed and validated against the experimental data [1]. The results from the CFD model have been used to create a submodel of the ACS for use with IES Apache and this is described in the rest of this section.

The Apache environment provides models for standard HVAC elements, such as rooms, boilers, chillers, heating and cooling coils, fans, chilled ceilings, but a user defined model is needed to account for the effects of the thermal storage of the slab. The ACS is modelled as a separate room, which behaves in a similar way as the ACS. Apache is a zone model in which flow variables are assumed to remain constant within a given zone, which normally represents a room. Each wall of this room is assumed to be at a uniform temperature and a heat transfer coefficient with respect to the average room temperature accounts for the convective heat transfer to the wall.



a)



b)

Figure 4 Reduction of the ACS model for use with Apache. a) sketch of a cross section of the ACS; b) simplified model of ACS for use with Apache. The grey, blue and green areas correspond to concrete, air and insulation, respectively.

A sketch of a cross section of the ACS model is presented in Figure 4a) and the reduced model for use with Apache in Figure 4b). The depth of the concrete corresponds to the depth of the zone of effective thermal admittance of the ACS, which is 150mm, as estimated in [3]. The thermal properties of the concrete in the reduced model are matched with the ones of the experimental prototype. An important parameter controlling the heat transfer to the concrete slab is the surface heat transfer coefficient between the air and the concrete slab and needs to be specified as a parameter in Apache. The heat transfer coefficient depends on the flow rate and a series of 6 CFD simulations for flow rates ranging from 10 l/s to 240 l/s were run with real weather data.

In Apache, the heat transferred to a wall (Q [W]) is given by

$$Q = h_c(T_b - T_s),$$

where h_c ([W/m^{2}K), T_b [C] and T_s [C] are the heat transfer coefficient, bulk room and surface temperature. An energy balance of the ACS gives the following relation:

$$Ah_c(T_s - T_b) = \dot{m}c_p(T_{in} - T_{out}),$$

where A [m^2], T_{in} [C], T_{out} [C], $\dot{m}$ [kg/s] and c_p [J/kgK] are the surface area, inlet temperature, outlet temperature, mass flow rate and heat capacity of the air. We can therefore compute h_c as:

$$h_c = \xi \frac{\dot{m} c_p}{A}, \text{ where } \xi = \frac{T_{in} - T_{out}}{T_s - T_b}.$$

T_{in} , T_{out} , T_b and T_s are found by appropriate averaging from CFD simulations of the ACS at selected $\dot{m}$. By minimizing the residual R:

$$R = \left(\sum_{i=1}^{n} \left(\left(T_{in,i} - T_{out,i}\right) - \xi \left(T_{s,i} - T_{b,i}\right) \right)^2 \right)^{\frac{1}{2}},$$

where the index i runs over the time series for T_{in} , T_{out} , T_b and T_s obtained from the CFD model a value for ξ is found that best satisfies $\xi = \frac{T_{in} - T_{out}}{T_s - T_b}$. The value of ξ minimizing R satisfies $\left(R^2(\xi)\right)' = 0$ and is:

$$\xi = \frac{\sum_{i=1}^{n} \left(T_{in,i} - T_{out,i}\right)\left(T_{s,i} - T_{b,i}\right)}{\sum_{i=1}^{n} \left(T_{s,i} - T_{b,i}\right)^2}$$

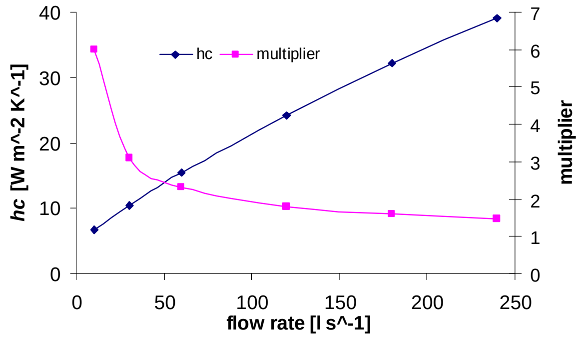


Figure 5 The multiplier ξ and heat transfer coefficient h_c for different flow rates.

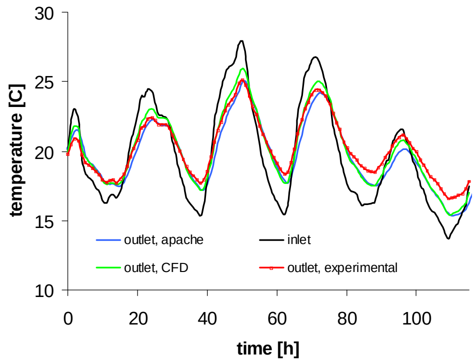


Figure 6 *Comparison between predictions for a single room model of ACS from Apache, experimental and CFD data. The flow rate is 120l/s.*

This procedure applied to flow rates ranging from 10 l/s to 240 l/s gives values for the multiplier ξ as shown in Figure 5. The heat transfer coefficient h_c is finally obtained by substituting $\dot{m}$, c_p and A into $h_c = \xi \frac{\dot{m} c_p}{A}$ and is presented in Figure 5.

IES Apache Model development and validation

The validity of the reduced model has been checked by comparison with results from the full scale physical and the CFD models. In the physical test, ambient air has been passed through the ACS at a fixed flow rate and the air temperature at the outlet has been monitored. Figure 6 shows the inlet (ambient) temperature and compares the temperature at the outlet of the ACS from the physical experiment, from the CFD and IES Apache models. It could be seen that the results from the reduced model compare very well with the ones from the experiment and CFD. Therefore the reduced model could reliably be used to further assess the performance of ACS under realistic conditions.

TRNSYS Model development and validation

TRNSYS is a zone model for building simulation, in which the building is represented by interconnected modules. TRNSYS offers a special type for the simulation of water or air flow through building components. Generally, TRNSYS is working with a transfer function for describing the thermal behaviour of walls and other planar components. For thermally activated building components a special type was developed using a limited two-dimensional, rectangular Finite Differences Model (maximum number of elements: 100). A representative section of the deck system has to be transformed into simplified mesh as shown in Figure 7 for the ACS.

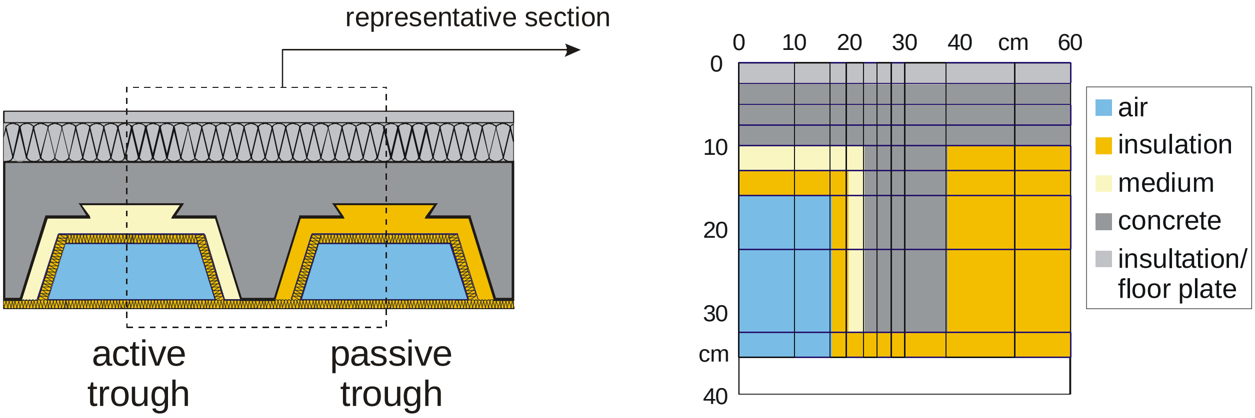


Figure 7 Development for reference section for TRNSYS Type 160 / 360

Based on this reference section the whole active deck element can be described (see Figure 8): In flow direction a partitioning into a number of equivalent sections is assumed (here: 6 partitions, each 2 m). Within a partition a logarithmic development of the medium temperature is considered. To get the full width of the deck element, a sufficient number of reference sections has to be assembled in parallel (here: 6 reference sections).

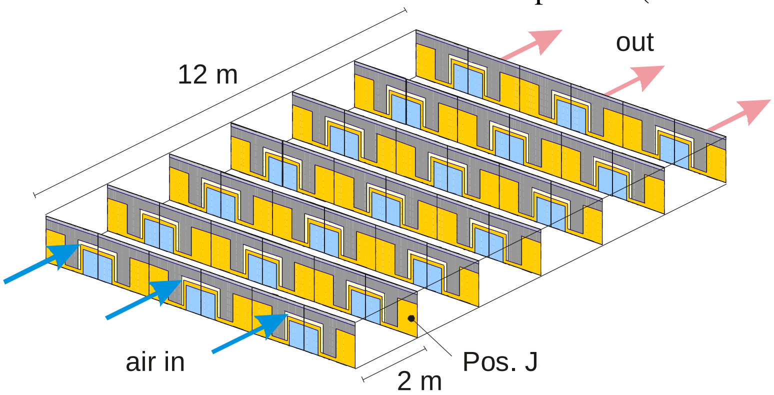


Figure 8 Expansion to a deck element

For the Validation of this numerical model measured data from [1] were taken. This measured data contain the temperature depending on the time (after switching the fans on) and over the length of the deck element. Figure 9 shows, that in the first minutes after forcing the ventilation there is a deviation between measured and calculated air temperatures, but after approximately 1 h the calculations fit very well with the measured values. Figure 10 shows the temperature over the length of the panel, the good correlation (after run time of two hours) is obvious.

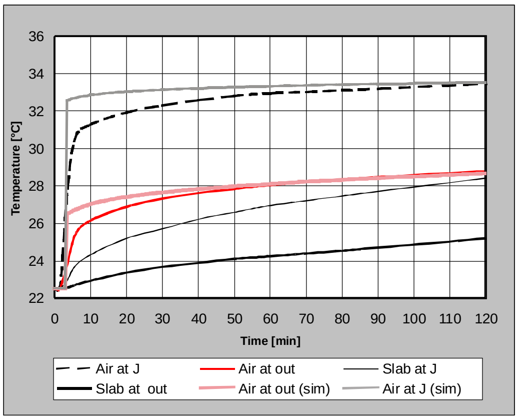


Figure 9 Switching on the panel, comparison of measured and calculated results

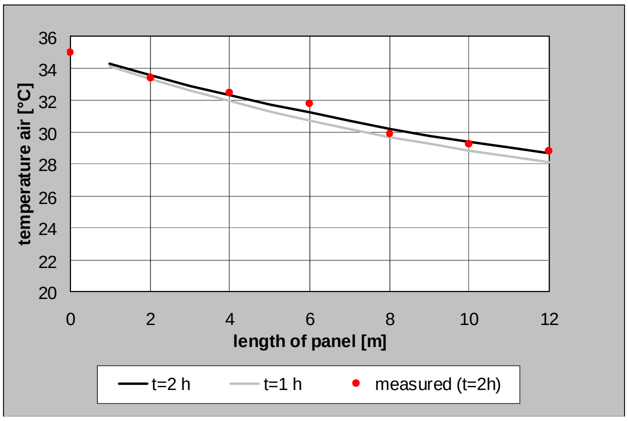


Figure 10 Development of temperature along the panel, comparison of measured and calculated results

RESULTS

The validated numerical models were used to simulate a whole office building, which is illustrated in Figure 11 and has been designed as a part of the EEBIS project [1]. The building has 4 storeys, the façades are oriented north and south, where the south façade is a transparent double façade. This building concept was investigated with the climatic data of Kew, UK, 1994.

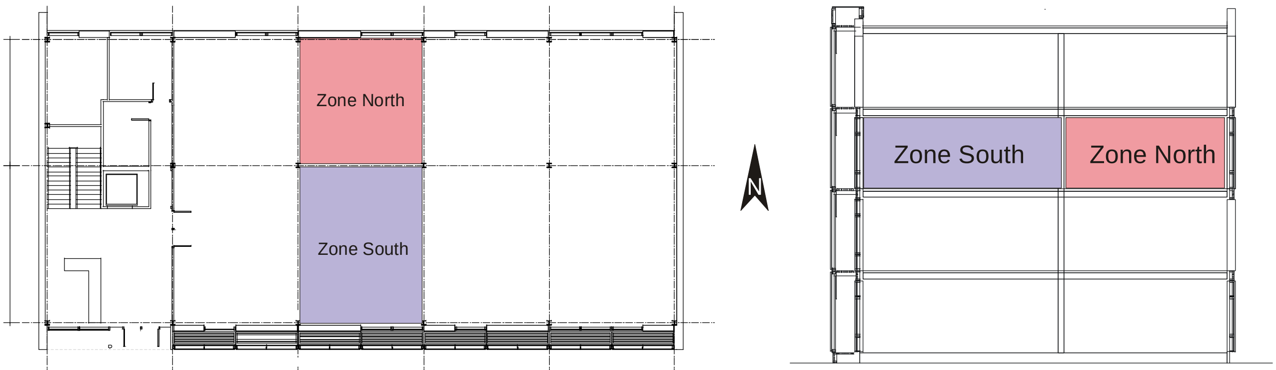


Figure 11 Office building with air cooled slab, ground plan and section plan

Figure 12 shows a comparison of the room temperatures (computed with the Apache and TRNSYS models) for the south oriented part of the third storey of the building. A hot period of the weather data was chosen in order to show the effect of the ACS. In the morning, the room temperature increases as in a conventional building. At noon the room temperature decreases due to the ACS being activated - pre-cooled air is entrained into the room. In the afternoon the air temperature rises again, but not as much as in a conventional building. Parallel calculations without ACS show, that the maximum temperature can be reduced about 4 K. At night, the cool ambient air is used to refresh the "coolth" storage of the ACS system. This work shows mainly two results:

a) relatively lightweight composite deck systems can be used as an efficient passive cooling system,
b) two different simulation tools, which were calibrated against measured values agree very closely.

The efficiency of the Air-cooled slab is shown in comparison to a conventional, non-air-conditioned building. Using a climate of south England, the maximum room temperature can be reduced by about 4 K. Also in combination with an air-conditioning, the results are interesting: the cooling demand (net energy) can be reduced from 10.1 to 2.8 kWh/m²a (set temperature cooling: 26 °C). The good correlation of Apache and TRNSYS results is evident, the differences are below 0.5 K for this investigated period. This shows that under similar assumptions both tools give similar results and could be reliably used to study similar problems.

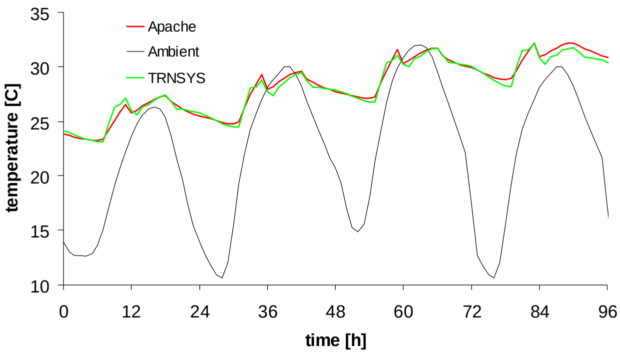


Figure 12 Comparison between predictions from TRNSYS and Apache.

DISCUSSION

Composite deep deck systems, which are the basis for the air cooled slab discussed in this paper, are generally not suitable for passive night cooling. Due to their shape they often come with a false ceiling and the thermal inertia has no contact to the room. The concept of forced air cooling of this deck system changes the situation: The mass of the slab can be efficiently used, the maximum room temperatures are similar to an exposed conventional concrete slab. Additionally the room temperature can be controlled, the surface temperature does not reach unwanted low values and an energy efficient combination with an air conditioning system is possible.

A disadvantage is the need for ducts, flaps and fans to run and control this system. In a further step the energy demand for conveying the air and the control system has to be taken into account. Regarding the investment costs it will be most interesting, if an airconditing system can be removed.

ACKNOWLEDGEMENT

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Modeling The Heat Transfer for a System of Pipes Embedded in a Wall

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SUMMARY

Compared to conventional heating and cooling systems, panel heating and cooling systems offer a number of advantages. Besides the lower air circulation, these advantages especially consist in the factor that the heat exchange between the heating system and the heated room takes place primarily by radiation and less by convection. It has been proved that this circumstance is physiologically advantageous for the occupant. Furthermore, such panel heating systems are very well suited to be used together with reversible heat pump systems, running on regenerative sources of energy.
In order to be able to dimension that kind of systems more effectively and to evaluate their economic efficiency, a software is implemented in MATLAB®. The software is to simulate the heat transfer occurring when panel heating (and cooling) systems are used, as close to reality as possible.
In a first step, the view factors between the surfaces that take part in the radiation exchange are specified. As the surfaces are assumed to be real grey emitters and the dependence of the intensity of emission as well as absorption on the angle is also taken into account, this can be carried out only numerically.
In a second step, an equation system which describes the radiation heat transfer between the surfaces, both heated and not heated, is to be solved. Therefore, either the surface temperatures or the provided thermal outputs need to be assumed. The equation system includes a balance (at steady state) between the heat received by a certain surface through radiation, and the heat lost to the environment through transmission.
In a further step, in addition to the radiation heat transfer, the convective heat transmission is to be taken into account. The corresponding heat transfer coefficients are being determined using Nusselt's laws. An overall heat transfer coefficient, consisting of a convective as well as a radiation part is to be accumulated.
The heat transfer inside the heated panel is also taken into account, and the solution of the corresponding heat conduction equation and the temperature distribution on the heated surface – at steady state regime - are provided by a MATLAB®-PDE-Toolbox. Thereby all the material characteristics of the surface as well as the overall heat transfer coefficient mentioned above are integrated into the solution of Fourier's differential equation as boundary conditions. Graphical presentation of the temperature distribution on the surface is used in order to illustrate the results.

INTRODUCTION

The purpose of this paper is to model the steady state heat transfer processes occurring when using panel heating/cooling systems.

The motivation for this paper consists of:

- product information given by manufacturers of panel heating/cooling systems is often not reliable;
- powerful tools to simulate such systems hardly exist.

On the other hand, the low temperature heating systems and the high temperature cooling systems are more and more used in connection with the re-generable sources of energy and the heat pumps. Thus, they become very interesting from the energy efficiency point of view.

METHODS

In order to be able to make a statement on the characteristics of a heating/cooling panel system, its behavior has to be modeled numerically.
A heating/cooling panel system is responsible to cover the heat loses/gains of the room where it is mounted, in order to provide the comfortable indoor climate.
The general equation for the heat balance of a room (for heating case only) states:

$$\dot{Q}_{nec} = \dot{Q}_{transmission} + \dot{Q}_{\inf iltration} \qquad (1)$$

where:

$\dot{Q}_{nec}$	is the necessary heat to be provided to the room;
$\dot{Q}_{transmission}$	is the transmission heat loss through inertial elements (walls) and non-inertial elements (windows);
$\dot{Q}_{\inf iltration}$	is the heat needed to heat the infiltrated outdoor air up to the indoor temperature.

When panel heating/cooling systems are used, the following types of indoor heat transfer occur:

- Radiation heat transfer between the active panel(s) and the indoor environment, but also between the non-active elements.
- Convective heat transfer between the active as well as the non-active elements and the surrounding indoor air.
- Heat conduction inside the active panels, especially from the heating pipes to the surface.

The whole heat transfer process is modeled with the aid of MATLAB®.
Several sets of data must be initialized – these are dealing with: the room geometry, the surface radiation properties and the ventilation data. This information will be processed further on by the program.
Briefly, the program consists of the following three crucial steps:

- Calculation of the view factors matrix (between all surfaces, assuming real-grey emitters);
- Calculation of the resulting temperatures of the non-active surfaces (temperatures of active panels are given, assumed to be the variable of the controlled process);
- Calculation of the resulting air temperature in the room under survey.

The logical diagram of the MATLAB® program is presented in Figure 1.

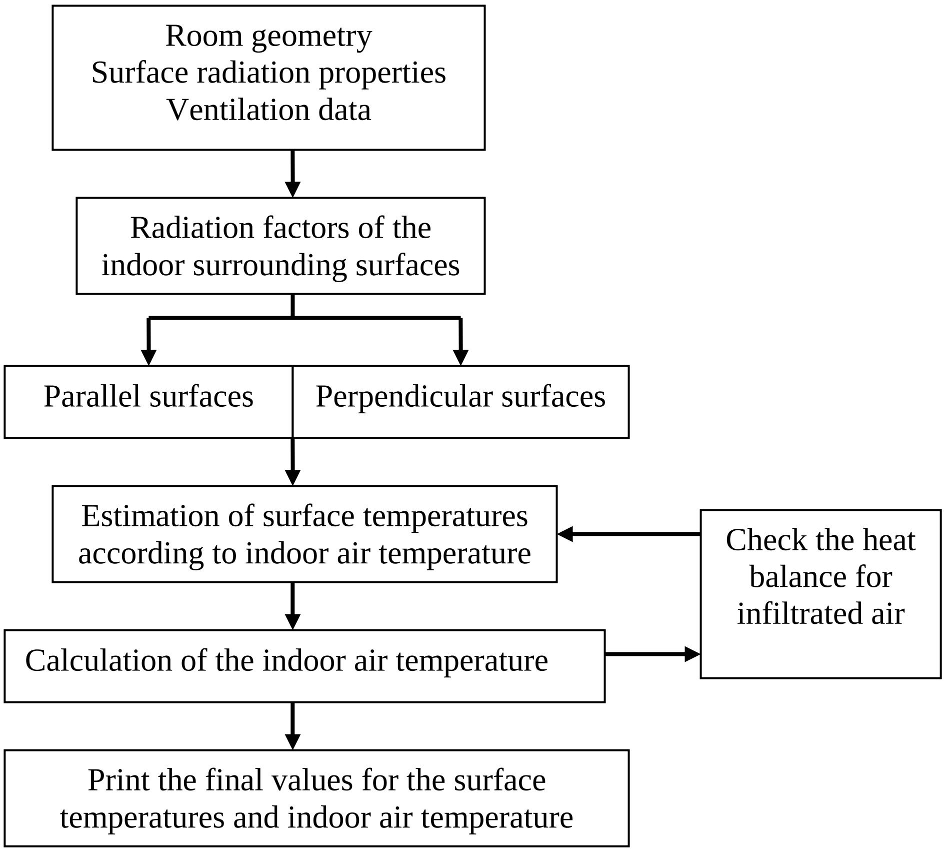

Figure 1. Logical diagram of the MATLAB® program

The geometry of the room modeled as an example in this paper is presented in figure 2.

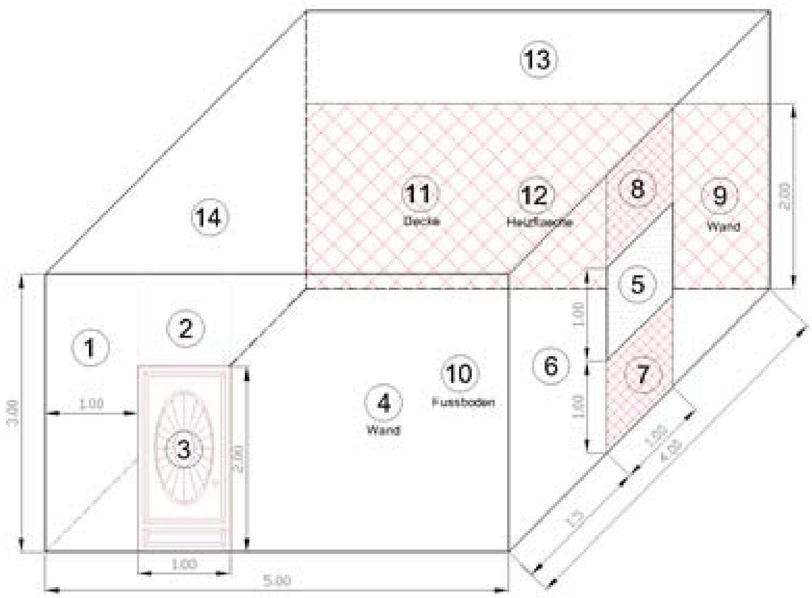

Figure 2. The geometry of the room under consideration

Possible properties for the 14 single surfaces of the example room geometry are given in Table 1.

Table 1. Possible properties for the 14 inner surfaces of the example geometry

Surface number	Emission factor for heat radiation normally on the surface ε_n	Heating surface yes/no	Temperature of the heating surface	Modified U value	Temperature of outdoor air	Observations
	-	-	°C	W/m²K	°C	-
1	0.93	no	-	1.20	20	Interior wall
2	0.93	no	-	1.20	20	Interior wall
3	0.93	no	-	2.40	20	Interior door
4	0.93	no	-	1.20	20	Interior wall
5	0.90	no	-	2.80	-15	Window
6	0.93	no	-	0.37	-15	Outer wall
7	0.93	yes	32	-	-	Heating surface
8	0.93	yes	32	-	-	Heating surface
9	0.93	no	-	0.37	-15	Outer wall
10	0.93	no	-	0.55	20	Floor
11	0.93	no	-	0.55	10	Ceiling
12	0.93	ja	30	-	-	Heating surface
13	0.93	nein	-	0.37	-15	Outer wall
14	0.93	nein	-	1.00	12	Interior wall

The indoor surfaces are considered to be real-grey emitters, meaning that their emission coefficients ε obey the conditions:

$$\varepsilon<1 \tag{2}$$

and

$$\varepsilon = f(\beta) \tag{3}$$

For most non-metal materials, the emission coefficient can be approached by the relation [1]:

$$\varepsilon(\beta) = \varepsilon(90°) \cdot \frac{1-(1-\cos\beta)^8}{0,978} \tag{4}$$

RESULTS

The modeling of radiation heat transfer carried out in MATLAB® takes place as follows:

- Numerical calculation of the view factors (for real-grey emitters) between <u>all</u> surfaces by a program implemented in MATLAB® (leading to a matrix) – presented as an example in Table 2, according to [2];
- Solution of an equation system leading to the intrinsic brightness for every single surface, depending on the surface temperatures;
- Calculation of temperatures of the non-active surfaces and of the indoor air, and the specific heat flux emitted by every surface;

Table 2. The resulted view factors for the example geometry

Receiving surfac	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Emission surface														
1	0.000	0.000	0.000	0.000	0.005	0.008	0.004	0.004	0.021	0.223	0.223	0.142	0.051	0.351
2	0.000	0.000	0.000	0.000	0.008	0.016	0.006	0.009	0.028	0.160	0.403	0.156	0.074	0.182
3	0.000	0.000	0.000	0.000	0.008	0.017	0.008	0.007	0.029	0.330	0.208	0.135	0.064	0.199
4	0.000	0.000	0.000	0.000	0.015	0.148	0.013	0.013	0.029	0.260	0.260	0.087	0.064	0.074
5	0.014	0.008	0.016	0.138	0.000	0.000	0.000	0.000	0.000	0.265	0.265	0.122	0.054	0.136
6	0.005	0.003	0.007	0.296	0.000	0.000	0.000	0.000	0.000	0.243	0.243	0.062	0.030	0.119
7	0.013	0.006	0.016	0.118	0.000	0.000	0.000	0.000	0.000	0.407	0.169	0.122	0.031	0.128
8	0.013	0.009	0.013	0.118	0.000	0.000	0.000	0.000	0.000	0.169	0.407	0.085	0.068	0.128
9	0.014	0.006	0.013	0.059	0.000	0.000	0.000	0.000	0.000	0.243	0.243	0.214	0.099	0.119
10	0.033	0.008	0.033	0.117	0.013	0.055	0.020	0.008	0.055	0.000	0.331	0.154	0.037	0.151
11	0.033	0.020	0.021	0.117	0.013	0.055	0.008	0.020	0.055	0.331	0.000	0.097	0.095	0.151
12	0.043	0.016	0.027	0.078	0.012	0.028	0.012	0.008	0.096	0.308	0.193	0.000	0.000	0.157
13	0.031	0.015	0.026	0.116	0.011	0.027	0.006	0.014	0.089	0.150	0.380	0.000	0.000	0.146
14	0.088	0.015	0.033	0.056	0.011	0.045	0.011	0.011	0.045	0.252	0.252	0.131	0.061	0.000

The convective heat transfer between the indoor surrounding surfaces and the indoor air takes place as follows:

- The information regarding the geometrical orientation of the surfaces (floor, ceiling, wall) must be provided;
- The convective heat transfer coefficients are calculated using Nusselt's law [3].

In order to calculate the surrounding surfaces temperatures, as well as the indoor air temperature, the radiation heat transfer must be combined with the convective heat transfer. The computation steps are as follows:

- The surrounding surfaces temperatures of the active panels (controlled process variable) need to be initialized;
- The temperatures of the non-active surfaces are iteratively calculated using the following balance of steady-state-condition for every single surface:

$$\dot{Q}_{res,i} = \dot{Q}_{rad,i} + \dot{Q}_{conv,i} + \dot{Q}_{cond,i} = 0 \tag{5}$$

- $\dot{Q}_{rad,i}$ is the heat flux a certain surface receives by radiation from the surrounding surfaces, expressed in [W];
- $\dot{Q}_{conv,i}$ is the heat flux a certain surface exchanges with the surrounding air by convection (depending on the air temperature and the convective heat transfer coefficient), expressed in [W];
- $\dot{Q}_{cond,i}$ is the heat flux a certain surface looses to the environment by heat conduction (depending on the material properties and the virtual outside temperature), expressed in [W] [4];

- The resulting mean air temperature in the room is iteratively calculated. Therefore, the number of air changes (caused by infiltration), as well as the outside air temperature need to be given.
- The following balance between the heat losses (due to the infiltrated air), $\dot{Q}_{air,loss}$, and the convective heat wins, $\dot{Q}_{conv,win}$, is used:

$$\dot{Q}_{air,loss} + \dot{Q}_{conv,win} = 0 \tag{6}$$

The heat losses $\dot{Q}_{air,loss}$ can be computed as follows:

$$\dot{Q}_{air,loss} = \frac{1}{3600} \cdot n_{air\ changes} \cdot V_{Room} \cdot \rho_{air} \cdot c_{p,air} \cdot \left(\vartheta_{air,indoor} - \vartheta_{air,outdoor}\right) \tag{7}$$

where:

$n_{air\ changes}$	is the number of air changes, expressed in [h^{-1}];
V_{Room}	is the volume of the room, expressed in [m^3];
ρ_{air}	is the density of the air, expressed in [kg/m^3];
$c_{p,air}$	is the air heat capacity, expressed in [J/kg K];
$\vartheta_{air,indoor}$	is the indoor air temperature, expressed in [°C];
$\vartheta_{air,outdoor}$	is the outdoor air temperature, expressed in [°C].

The heat wins $\dot{Q}_{conv,win}$ can be computed as follows:

$$\dot{Q}_{conv.win} = \sum_{i=1}^{n} A_i \cdot \alpha_{conv,i} \cdot \left(\vartheta_{air,indoor} - \vartheta_{surf,i}\right) \tag{8}$$

where

A_i	is the surface of element "i", expressed in [m^2];
$\alpha_{conv,i}$	is the convection coefficient calculated on the basis of the Nusselt law, expressed in [W/m^2K];
$\vartheta_{surf,i}$	is the temperature of the surface element "i", expressed in [ºC].

Finally, the indoor air temperature (which is the parameter that must be achieved) is compared to the medium radiation temperatures of the surface elements in the room, in order to be able to achieve the indoor thermal comfort [5].

After setting the temperature of the heating panel, the MATLAB®-PDE-Toolbox can be used to establish the temperature distribution and the local temperature gradient inside the panel as well as the heat flux provided. The model is based on Fourier's Partial Differential Equation. Therefore, the material properties of the panel as well as the boundary conditions (heat transfer coefficient, heat flux or boundary temperatures, calculated in the step before) need to be given.

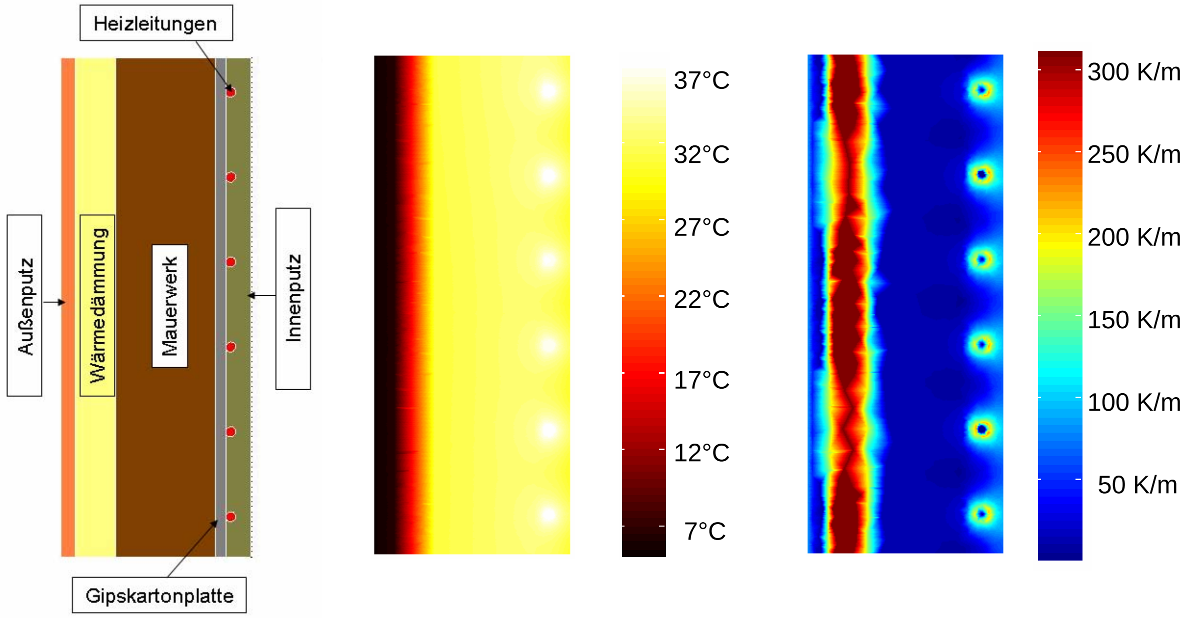

Figure 3. Example for a possible panel geometry, the possible temperature distribution inside and the local temperature gradient inside, 4000 s after switching on the heating panel

DISCUSSION

The work presented in this paper represents a starting model for the calculation of low-temperature-heating/high-temperature-cooling systems, and therefore has several limitations. First of all, the calculation of the surfaces temperatures inside the room, and of the indoor air temperature is made under steady state conditions and in the absence of any occupants (people, furniture pieces, etc.), whilst the calculation of the temperature distribution and heat flux inside the active wall can be performed dynamically. On the other hand, the theoretical model has not yet been verified in practice, such experiments are to be performed in the near future in the frame of a doctoral thesis.
A further development phase is previewed to connect the "room - active wall" system to the heating/cooling source (reversible heat pump), and to dynamically simulate them as a whole.
Nevertheless, the pragmatic approach and the thorough understanding of phenomena represent a strong advantage for future developments.

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12 June 2007 at 10:00 - 11:30

C03
Demand controlled systems

Personal Ventilation: from research to practical use

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SUMMARY

This paper describes an ongoing demonstration project on the use of personalized ventilation in practice. Several designs of air supply devices for PV were developed, tested and optimised based on the efficiency of clean air supply to occupants' breathing zone, control functionality, aesthetic, etc. Pilot installation of the PV system in a number of offices was used to gather better experience on the interaction between occupants and the PV system. The second stage of the project includes installation of the system and its performance in an office building. Computer based questionnaire survey on occupants' comfort and performance as well as measurement of the indoor environment in the offices will be performed before and after installation of the PV system. Surveys will be performed during the following winter and summer seasons. It is expected that the obtained results will provide knowledge important for future application of PV in practice.

INTRODUCTION

Numerous studies have associated the quality of indoor air with people's complaints and with their performance. It has been shown that poor air quality causes Sick Building Syndrome (SBS) symptoms such as increased prevalence of headache, decreased ability to think, increased dizziness, etc. [1]. Providing good air quality, on the other hand, increases occupant's productivity [2, 3, 4, 5]. Furthermore, it has been documented that the quality of inhaled air of low enthalpy (low temperature and humidity) is perceived to be better than when the enthalpy is high [6]. Thus an increased productivity due to improved indoor air quality and thermal environment will have an enormous economic impact which may cover all the capital and running costs of operating the building. The gain will often be greater than all other costs relating to the construction and operation of the building [7].

The first important step in improving indoor air quality is to remove or decrese the pollution sources. However it is not possible to remove all pollution sources. Occupants also generate pollution. Supply of more clean air to the room will dilute the polluted air and will improve air quality. This approach however leads to increase of initial costs due to larger ducts, air supply devices, etc., as well as increased energy consumption. Instead, it is better to keep the ventilation rate low but to distribute the air more efficiently in the room. With this respect the ventilation effectiveness, *VE*, defined as the ratio of the pollution concentration in exhaust air divided by the pollution concentration in air inhaled by occupants can be used to assess the performance of ventilation systems.

The total-volume ventilation systems (mixing and displacement) are capable of achieving only relatively low levels of ventilation effectiveness. The mixing ventilation aims for as good as possible mixing of the supplied clean air with the polluted room air and therefore typically has *VE* ≈ 1. Higher values of *VE* (*VE*≈6) can be achieved with displacement ventilation under laboratory conditions depending on the type and location of the pollution source. In practice *VE*≈1.4 is typically achieved in rooms with displacement ventilation [8]. However displacement ventilation may cause draught discomfort. Field study in 120 rooms with displacement ventilation identified that up to 50% of occupants were dissatisfied with indoor air quality [9].

Personalized ventilation (PV) aims to supply clean air directly to the breathing zone of room occupants and therefore it has potential to achieve *VE* substantially higher (*VE≈50* or more*)* than the *VE* of mixing and displacement ventilation [10]. The significant improvement in people' thermal comfort and perceived air quality, decrease in the reported Sick Building Syndrome (SBS) symptoms and tendency for improved work performance when PV is used, has been documented in human subject experiments performed in field laboratories [11, 12, 13]. Furthermore it has been shown that PV can efficiently protect occupants from airborne transmission of infectious agents, which are present in air exhaled by sick occupants [14].

Personalized ventilation has been applied for many years in theatres and vehicle cabins, in most cases in order to improve occupants' thermal comfort. Only few studies report on the performance of PV in office buildings [15, 16]. The effectiveness of the personalized ventilation depends greatly on the design of the PV Air-Terminal Device [17, 18, 19]. Design of PV with high *VE* easily applicable in practice is not an easy task. Apart of the *VE* other factors, such as background room air distribution, control strategies, occupant activities, appearance of air supply devices, etc., are also important [20, 21].

This paper describes an ongoing demonstration project on development, optimization and installation of PV in an office building. Computer based questionnaire survey on occupants' comfort and work performance as well as continuous measurement of the indoor environment in the offices is used for monitoring and evaluating the performance of the PV.

THE PERSONALIZED VENTILATION

Design

The first step of the project was focused on development and optimization of an air terminal device (ATD) for the personalized system. Some of the important criteria for the design of the ATD were: 1) to be able to supply maximum clean (personalized) air to inhalation; 2) to cause minimum draught discomfort for the occupants; 3) to improve occupants' thermal comfort and 4) to cause minimum transport of infectious agents between occupants. With this respect the knowledge obtained from previous research was carefully evaluated [10, 17, 19]. Other factors important for the practical use of the PV were considered as well. The most important of those were related to levels of control provided to users, installation at different workplaces, aesthetic and appearance. As a result several prototypes were developed. Figure 1 shows one of the designs. The ATD is attached to an arm which can be rotated around its vertical axis. This allows for changing the direction of the personalized flow in horizontal plan. The design of the ATD allows also for changing the direction of the supplied personalized flow in a vertical plan. In this way the occupant can direct the personalized flow to the face or chest or any other angle. The flow rate of the personalized air supplied through the arm to the ATD can be controlled, i.e. the occupant can adjust the preferred target velocity at his/her face or body. The size of the ATD was defined based on its appearance and on the aerodynamic of the generated personalized flow.

Figure 1. The personalized ventilation in use.

Performance of the ATD

The performance of the ATDs was tested in a full-scale air movement room. The room was ventilated by mixing ventilation (swirl diffuser placed in the middle of the ceiling). Breathing thermal manikin resembling human was used during the experiments. Experiments at different flow rate supplied from the PV, different distance between the ATD and the manikin and different angle of discharge of the personalized flow were performed. The velocity field in the personalized flow generated by the developed ATD was carefully identified. The most important part of the test was to identify the performance of the PV with regard to providing clean air in inhalation. For this purpose tracer gas was mixed with the supplied ventilation air (mixing ventilation). The supplied personalized air was free of tracer gas. The concentration of tracer gas in air inhaled by the manikin, the air exhaust from the room, at several points in the room and in the supplied personalized flow was measured. The data were use to calculate the personal exposure effectiveness (PEE) defined as:

$$PEE = \frac{C_{I,O} - C_I}{C_{I,O} - C_{PV}} \qquad (1)$$

where:
$C_{I,O}$ – is the pollution (tracer gas) concentration in the inhaled air without PV;
C_I – is the pollution (tracer gas) concentration in the inhaled air with PV;
C_{PV} – is the pollution (tracer gas) concentration in the personalized air;

The personal exposure efficiency, PEE, is an index representing the percentage of clean air contained in the inhaled air. It is developed for evaluation of the performance of personalized ventilation systems [17].

The results of the tests showed an improvement in inhaled air quality by 40% (*VE*≈1.7) in comparison with mixing ventilation alone (*VE*≈1). The ATDs were further improved in order to increase their performance.

Testing of the PV under real conditions

Pilot installations of the PV system were tested in order to gather better experience on the interaction between occupants and the PV system. The administration building of the manufacture of the PV systems was use. The PV was installed at three workplaces used by several of the employees for shorter (several hours) or longer (several days) periods of time. The response of the users was very positive. Apart of the achieved preferred thermal comfort and improved quality of the inhaled air (the inhaled air was felt fresh and clean) the users reported an improved self-estimated performance with the PV, especially during the afternoon hours. The response of the users with regard to the provided control of the local velocity and direction of the personalized flow as well as the appearance of the PV was positive as well. Further modifications and improvements of the developed PV systems are in progress.

APPLICATION OF THE PV IN PRACTICE

Installation of the PV system in an office building

A consulting company (partner in this project) was requested by a building owner to improve the indoor environment of a 200 years old building where severe indoor climate problems were reported. The building, located downtown Copenhagen, is occupied by a company dealing with preparation of regulatory documents (low, taxation, etc.). Thus the employees perform office work in front of computer screens. The building (the old building) does not have mechanical ventilation. The ventilation through the windows is insufficient to ventilate and cool down the building. The opening of the windows for getting fresh air creates a lot of thermal discomfort during the winter. Therefore among other improvements of the building (better roof isolation, better solar shading, floor renovation with low polluting materials, etc.) installation of mechanical ventilation was requested. Since the office work performed in the building is rather responsible and requires deep concentration a high quality environment for each occupant was requested.

The ceiling height in many of the rooms in the old building is below 2.4 m (2.2 m in several rooms). Furthermore there is not enough space (typically needed in practice) for installation of air terminal devices and ducting system. This made the task more difficult. The design of the total volume ventilation ended with mixing room air distribution with air supply devices located on the walls. In order to account for the occupants' preferences with regard to thermal comfort and to provide them with high quality of inhaled air personalized ventilation in conjunction with the mixing ventilation was designed. The mixing ventilation can be used alone or in conjunction with the personalized ventilation. The design allows for maximum 20 L/s per occupant outdoor air in total, supplied by the two systems. The minimum amount of outdoor air supplied to each room is 5 L/s per occupant. The outdoor air is distributed between the total volume system and the personalized ventilation systems depending on the occupants' preferences. The PV systems at each workplace supply maximum of 15 L/s and the personalized flow rate can be reduced to 0 L/s. The installation of the system will start by the end of April 2007.

Field survey

A survey on the impact of the PV system in regard to occupants' health, comfort and performance is designed. Occupants' cognitive stress level is in the focus of the survey as well.

The survey will be performed before and after renovation of the building. Two groups of occupants are included in the survey: group A includes occupants working in the building which will be renovated (the old building) and group B includes occupants working in the attached relatively new building (the new building) which will not be renovated (this group will be used as a reference). Much less complains due to poor indoor environment have been recorded in the new building. The two buildings are interconnected. They are rented by the same company. Three Phases are considered. Phase 1 includes a survey before renovation of the old building. This phase is in progress. After renovation of the old building two more phases are planned: Phase 2 when mixing ventilation is used alone and Phase 3 when personalized ventilation in conjunction with mixing ventilation is used. In this way the effect of the ventilation in general and of the personalized ventilation in particular will be identified. Future surveys during following winter and summer seasons are planned in order to identify the level of occupants' satisfaction after use of the personalized system for relatively long period of time.

The method of survey includes long term survey and short term survey. Both, the long term and the short term surveys will be performed before and after the renovation of the old building. The long term survey is planned to continue for 3-4 weeks during each of the phases of the project, i.e. before the renovation and after the renovation twice (only mixing ventilation and mixing ventilation in conjunction with personalized ventilation). The long term survey includes occupants' response to the indoor environment collected by questionnaires, work performance test and continues physical measurements. The questionnaire survey and the work performance tests are computer based and performed via internet [22]. At least three times during a week (in the afternoon of different days) an invitation for participation appears on the screen of each participant in the survey. The occupant votes on short questionnaire including questions on occupants' thermal comfort (7 point ASHRAE scale) and its acceptability (continues scale from clearly unacceptable to unacceptable and then from acceptable to clearly acceptable), acceptability of air quality, acceptability and preferences of air movement as well as where it is felt, SBS symptoms (eye irritation, headache, ability for concentration, tiredness, stress level, etc.) and self-assessed performance. After completion of the short questionnaire follows work performance test which includes proof reading of text with errors which have to be found. The time for this test is limited to 5 min. The work performance of each participant will be analyzed based on the number of identified errors and the length of the processed text. The long term survey includes also continuous measurement of air temperature, relative humidity and CO_2 concentration in the surveyed rooms.

The short term survey will be performed during each of the three phases. It will be performed only once during the 3-4 weeks long survey. It will include detail paper questionnaire on occupants' comfort, health conditions, work environment. Simultaneously spot measurements of velocity and temperature at each workplace at different heights will be performed as required in the present thermal comfort standards [23, 24]. This short term survey will last not more 15 min at each workplace.

CONCLUDING REMARKS

The field survey before installation of the ventilation system and renovation of the building (Phase 1) is in progress. Phases 2 and 3 are expected to take place in August- September 2007. It is expected that the obtained results will provide knowledge important for future application of PV in practice.

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Opportunities in the design of control-on-demand HVAC systems

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SUMMARY

Control-on-demand operation of HVAC-systems can reduce energy for heating, cooling, and the drive energy of fans and pumps resulting in better control at a lower operational cost. Analysis of typical Nordic CAV and VAV systems indicates a potential for substantial savings using new components and alternative system design. Component developments include more efficient motors, variable speed drives, pumps and fans as well as laminar flow heat exchangers and smart air-supply devices. Alternative system designs can reduce pressure drops by replacing valves and dampers with direct speed control of the pumps and fans and reduce mean flow rates by means of control-on-demand. Furthermore, separating the conditioning and distribution pressure drops, e.g. going from in-line to parallel conditioning, adds another dimension of design to further reduce the energy to fans and pumps.

INTRODUCTION

Demand patterns of buildings have changed considerably over the past 20 years, in particular for offices and other types of commercial building. As a result, space conditioning is largely accomplished by means of variable outdoor air flows instead of using heaters and coolers.

Design capacity and utilization

Increasing internal loads combined with improved building envelopes and heat recovery has drastically reduced heating and full capacity heat recovery operation. On the other hand, demand for cooling has increased. In Nordic countries, however, the use of free cooling by means of outside air still makes the duration of active cooling of supply-air quite short. Hence, the *design capacity and utilization* of new HVAC systems have to be studied carefully. The design capacities of the systems for heating, heat recovery ventilation and cooling are rarely used and must therefore be turned down most of the year to match demand with supply. The short operating periods for supply-air heating, cooling and heat recovery, together with large variations in occupancy, is an open invitation for *control-on-demand* (COD) systems with new possibilities of system design.

Control-on-demand (COD)

Control-on-demand implies a variation of supply in response to a variation of demand of some utility. In HVAC systems, the utilities involved are usually heating, cooling and volumes of clean air. Traditionally, heating and cooling systems operate with COD of room temperature based on feed-back control by means of a room sensor. Often a complementary feed-forward control, based on the outdoor temperature, is used for heating but rarely for cooling. The occupant determines the demand level by means of the room thermostat.

In ventilation, constant flow CAV systems have commonly been used. Even modern VAV systems often operate with fixed minimum outdoor-air supply rates and there is no feed-back regarding air-quality. On the other hand, the flow will normally vary in response to room temperature, i.e. thermal COD. In special applications, such as rooms with large variations in occupancy (lecture rooms, theatres, etc.), feed-back control by means of an indicator for the load or the actual air-quality is becoming increasingly popular. VAV systems with feed-back control are often known as DCV systems. This category should include all kinds of ventilation with feed-back control and not only those with respect to air-quality. With COD, not only the control strategy per se, but also the hydraulic *system design* is important for good results.

System design

Utilization of the design capacity and the wish for COD imposes conditions which must bear on the system design. With a focus on electric drive energy, new system designs can provide saving opportunities primarily by means of reduced pressure drops and reduced mean flow rates. In this respect, there are two basic system design factors: flow design and control.

In this study we have looked at two alternatives to common design practice. The first concerns reduction of pressure drop by replacing dampers and valves for flow control by direct speed control of fans and pumps [4]. The second underlines the importance in variable flow systems to distinguish between transport and conditioning pressure drops.

Drive energy to pumps and fans

A major objective of this study has been to find ways of reducing the drive energy to the fans and pumps of HVAC-systems. Second only to lighting, drive energy to fans and pumps dominate the use of electricity for building operation. SFP values help in reducing the specific use but do not address the rise in absolute terms from increasing demand. The principle ways to accomplish low energy use in this sector is by means of reduced design pressure drops (alternative system layout, new components and sizing), reduced mean flow rates and mean pressure drops (COD), and improved efficiency of pumps, fans, motors and motor drives.

METHODS

This study primarily looks at the opportunities of saving drive energy to fans and pumps in control-on-demand HVAC systems in general and some alternative system designs in particular. Since conditioning of the supply-air totally dominates the space demand for heating, cooling and ventilation, the case-studies will look at possible savings in this area (c.f. the section on system design). Important aspects are: *Design capacity and its utilization*, *control-on-demand* principles of ventilation, heat recovery, heating and cooling, *capacity control of air-coils*, *system design*, and savings on *drive energy to fans and pumps*.

Design capacity and utilization

Modern offices and other commercial buildings have very short demand periods for heat recovery and heating. The use of novel air-supply devices [6] also permits low supply-air temperatures with no comfort problems and thus provides a possibility to maximize the use of free cooling by means of outdoor air. Equations (1) to (3) are used to illustrate the difference in relative design capacities and operating times for heat recovery, supply-air heating and supply-air cooling (see Figure 3 for designations of heat exchanger outlet temperatures).

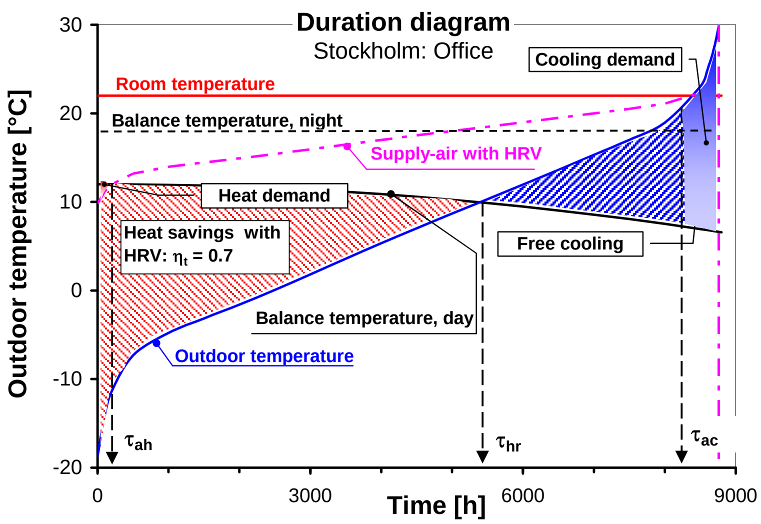
$$\dot{q}_{hr,d} = \frac{\dot{Q}_{hr}}{\dot{Q}_{sah,\max}} = \frac{t_{sa2} - t_{oa,\min}}{t_{ea} - t_{oa,\min}} \text{ at } \tau = 0\,[\text{h}] \text{ and } R_{\tau,hr} = \frac{\tau_{hr}\,[h]}{8760\,[h]} \tag{1}$$

$$\dot{q}_{ah,d} = \frac{\dot{Q}_{ah}}{\dot{Q}_{sah,\max}} = \frac{t_{sa3} - t_{sa2}}{t_{ea} - t_{oa,\min}} \text{ at } \tau = 0\,[\text{h}] \text{ and } R_{\tau,ah} = \frac{\tau_{ah}\,[h]}{8760\,[h]} \tag{2}$$

In Nordic climatic conditions, dehumidification of supply air is not a major problem and the specific enthalpies *h* below can be approximated by the temperature *t*.

$$\dot{q}_{ac,d} = \frac{\dot{Q}_{ac}}{\dot{Q}_{sac,\max}} = \frac{h_{sa4} - h_{sa3}}{h_{oa\max} - h_{sa6}} \text{ at } \tau = 8760\,[\text{h}] \text{ and } R_{\tau,ac} = 1 - \frac{\tau_{ac}\,[h]}{8760\,[h]} \tag{3}$$

Fahlén [2] describes how the design capacity of air-heaters and air-coolers can be modulated from design values by means of temperature control and/or flow control on the liquid side. Another possibility, which is pointed out, is the direct effect of controlling capacity with a variable air flow rate in coils with the minimum heat capacity flow rate on the air-side (the normal situation). This is not a viable option in traditional, in-line air-handling units, but an interesting possibility in alternative, parallel designs (c.f. Figure 4 a).



Input data:

$\bar{t}_{year}$ = 6.6 °C

$t_{oa,\min}$ = -19 °C

$t_{oa,\max}$ = 30 °C

t_{room} = 22 °C

Δt_{sun} = 5.4 °C

Δt_{int} = 10 °C

$t_{sa,\min}$ = 18 °C

$\eta_{t,\max}$ = 0.7

Figure 1. Duration diagram of the temperatures of outside air t_{oa}, supply-air t_{sa2} (after heat recovery) and the balance temperature t_{bt} for an office building.

Figure 1 provides an example of the temperature duration diagram for an office building. This illustrates the low daytime utilization factors of heat recovery, heating and cooling in such buildings, which opens up for *control-on-demand systems* and alternative *system designs*.

Control-on-demand (COD)

Although not the focus of this study, COD per se is obviously a major design factor [1, 3]. There are two main objectives for COD in air-conditioning HVAC systems: satisfaction of room thermal comfort and air quality. Feedback of the room status controls the three subsystems of heat recovery, heating and cooling to provide a thermal environment in relation to demand. The subsystem of air volume supply handles the air-quality aspect. Control can also be enhanced with feed-forward of load related disturbance factors.

The wide range of operation of COD systems affects the control difficulty. Fahlén [1, 2] advocates feed-forward control of the supply temperature to modify the capacity of air-coils in relation to demand. This effectuates a COD of the open loop gain, which greatly reduces the control difficulty. Otherwise the dynamic difficulty, D_{dyn} [1] according to (4), will increase as demand goes down. For instance, a reduction of the air flow will increase the delay time τ_d (transport time between heat exchanger and control sensor) and Δy_{max} (maximum change of the output variable) will increase if the thermal capacity of the coil remains constant while the air flow goes down. The control difficulty also goes up with a high required static accuracy, i.e. a low value of the permissible control deviation Δy_s.

$$D_{dyn} = \frac{\Delta y_{\max}}{\Delta y_s} \cdot \frac{\tau_d}{\tau_r} \tag{4}$$

Equation (5) below illustrates the need for COD of heat recovery. Below the balance temperature t_{bt}, the heat recovery system operates with constant, maximum efficiency $\eta_{t,max}$. Above the balance temperature, η_t is controlled in relation to demand. With a CAV system, the heat capacity flow rate $\dot{C}_{sa}$ and the maximum efficiency $\eta_{t,max}$ are constant and can be moved outside the integral. In the case of a VAV system, however, neither $\dot{C}_{sa}$, $\eta_{t,max}$ or η_t are constants since both $\dot{C}_{sa}$ and η_t are controlled on demand and $\eta_{t,max}$ is a function of $\dot{C}_{sa}$.

$$\dot{Q}_{hr} = \int_{\tau_0}^{\tau_{ah}} \dot{C}_{sa} \cdot \eta_{t,\max} \cdot (t_{ea} - t_{oa}) \cdot d\tau + \int_{\tau_{ah}}^{\tau_{hr}} \dot{C}_{sa} \cdot \eta_t \cdot (t_{ea} - t_{oa}) \cdot d\tau \tag{5}$$

The air heating coil only operates below the balance temperature. Eq. (6) is subject to the same considerations regarding CAV and VAV systems as indicated above. The capacity of the air heating coil has to vary in relation to variations of outside loads (outdoor temperature $\rightarrow t_{oa}$, insolation $\rightarrow t_{bt}$) and inside loads (occupancy $\rightarrow \dot{C}_{sa}$, t_{bt}, equipment $\rightarrow \dot{C}_{sa}$, t_{bt}).

$$\dot{Q}_{ah} = \int_{\tau_0}^{\tau_{ah}} \dot{C}_{sa} \cdot \left[t_{bt} - (1 - \eta_{t,\max}) \cdot (t_{ea} - t_{oa})\right] \cdot d\tau \tag{6}$$

Finally, equation (7) provides the necessary air-coil cooling capacity presuming that free cooling covers demand between τ_{hr} and τ_{ac} (see figure 1). At the time τ_{ac}, the COD system reduces air flow to the level decided by air-quality feed back or the estimated minimum.

$$\dot{Q}_{ac} = \int_{\tau_{ac}}^{\tau_{year}} \dot{C}_{sa} \cdot \left[t_{bt} - (1 - \eta_{t,\max}) \cdot (t_{ea} - t_{oa})\right] \cdot d\tau \ \text{ with } \ \tau_{year} = 8760\,[\text{h}] \tag{7}$$

Control-on-demand (COD) of air-coils

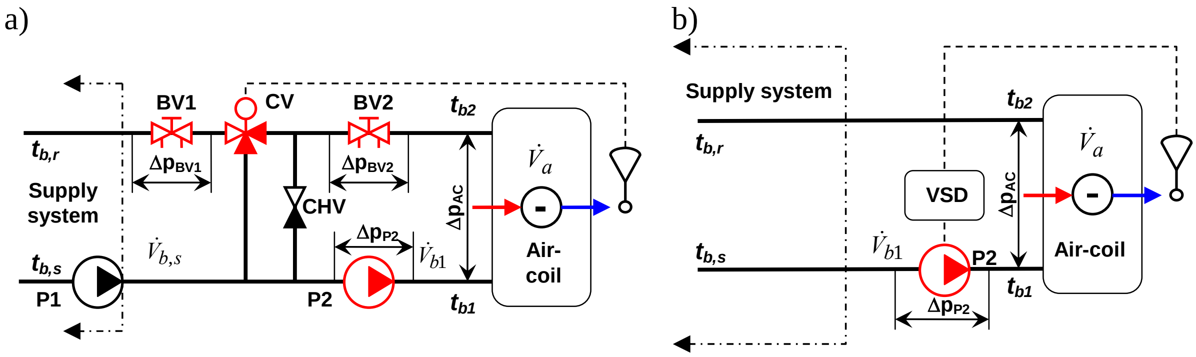
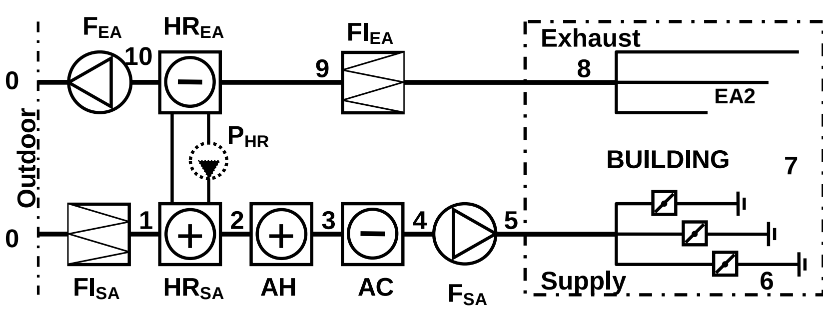


Figure 2. Control by means of a) variable inlet temperature and b) variable coil flow.

Trüschel [7] has categorized and studied current practice of capacity control of coils. Fahlén [2] also discusses the pros and cons of alternative control arrangements with the objective of maximizing the heat transfer capacity in relation to the drive energy of fans and pumps. There are three basic control methods: 1) Variable system supply temperature, 2) variable coil inlet temperature (with constant or variable coil liquid flow) and 3) variable coil liquid flow (c.f. Figure 2). Changing the coil control subsystem from traditional valve control (figure 2a) to direct, variable speed pump control (figure 2b) rids the system of all the control pressure drops of balance and control valves. This will not only drastically reduce the drive energy to pumps but also greatly simplify the system.

System design

Due to the wide range of operation of COD systems, small inputs and losses, which are considered insignificant in traditional systems, must be considered in COD [4, 6]. New designs should also maintain controllability and operability over a wide range of demand to achieve better control at a lower cost of energy. Figure 3 provides a schematic of a traditional dedicated outdoor air system with in-line air-conditioning units.



AH air-heater
AC air-cooler
EA exhaust-air
F fan
FI filter
HR heat recovery
P pump
SA supply-air

Figure 3. Typical supply and exhaust ventilation system with heat recovery.

In this arrangement, the pressure drops of heat exchangers (HR, AH, AC) require fan power all the time even though the utilization may be quite low. It would be desirable to have the same integration period for drive power, eq. (8), as for heat transfer capacity, eq. (5)-(7). Figure 4 presents two alternative designs based on a parallel arrangement of the conditioning units. In this type of design, the conditioning units can operate independently of the general ventilation system. Hence the coils and fans can be individually optimized for efficient heat transfer and controlled by demand. It is also feasible to add or subtract units without affecting the rest of the system.

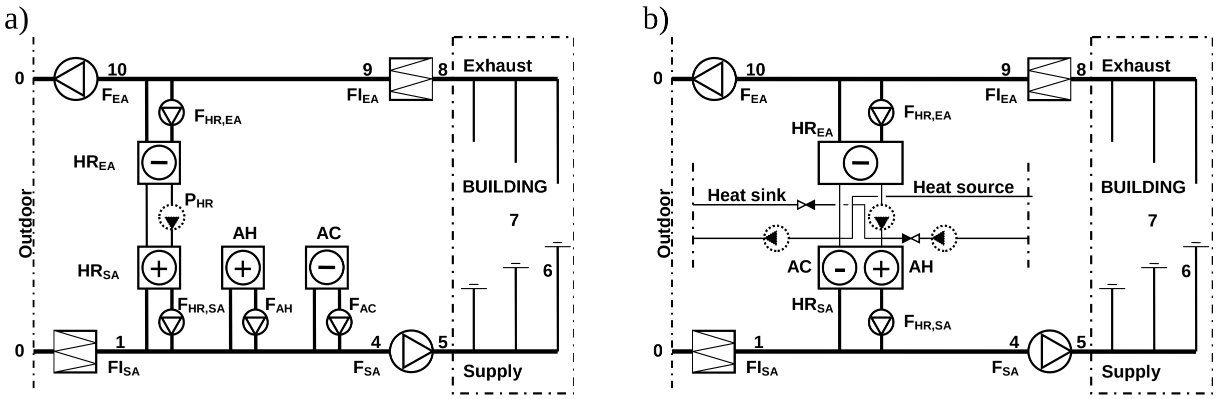


Figure 4. Alternative mechanical exhaust and supply air systems with heat recovery:
a) Parallel arrangement; b) integrated heat recovery, heating and cooling with ground storage.

Finally, there will of course always be opportunities for more efficient components such as new electric motors, drives, fans, pumps and smart air-supply devices. Furthermore, laminar flow heat exchangers can further reduce the pressure drop while simultaneously improving the efficiency of motors and motor drives (different torque characteristics in laminar flow).

Drive energy to fans and pumps

The drive energy to fans and pumps can be reduced through the following actions (individually or in combinations): 1) reduce the number of pumps, 2) the operating hours, 3) the flow rates and pressure drops and 4) increase the efficiencies. Alternative system layouts, control strategies, and the selection of coils will influence the first three items. New pump motor controls enhance the possibilities of more efficient control. Improved hydraulic design and more efficient motors will address the final savings opportunity. COD related variations of flow and operating time will determine the aggregated drive energy according to eq. (8):

$$W_e = \int_{\tau 1}^{\tau 2} \frac{\dot{V} \cdot \Delta p(\dot{V})}{\eta_p(\dot{V})} \cdot d\tau \quad [\mathrm{W}] \qquad (8)$$

In the case of an in-line system, $\tau_1 = 0$ and $\tau_2 = \tau_{year}$ whereas a parallel system will have the same integration time as the heat transfer duty, i.e. $\tau_2 = R \cdot \tau_{year}$. For a CAV system, the integrand will be a constant and $W_{ep} = \dot{W}_{ep} \cdot \tau_{year}$.

RESULTS

The results of simple calculations show the importance of considering the wide range of operation of modern COD air-conditioning systems.

Design capacity and utilization

In apartment buildings, low internal loads result in a balance temperature in the range 14 - 19 °C, depending on the time of year, and with average utilization factors for heating and heat recovery. There is little need for active cooling which is usually avoided. Regarding office buildings, however, internal loads result in much lower balance temperatures, 7-12 °C as a diurnal average and much lower on a daily basis. This results in low utilization factors for heating and heat recovery. Active cooling is limited to outdoor temperatures exceeding 20 °C.

Table 1 compares the design capacities of the air-coils for heat recovery, heating and cooling of an office and an apartment building, see eq. (1) to (3). The table also compares the relative operating times of the respective units. It is obvious from the table that the design conditions will be very different and that the low utilization of heat exchanger duty should reflect on the design of office systems.

Table 1. Relative design capacities $\dot{q}$ and relative operating times R_τ of heat recovery (hr), air-heaters (ah) and air-coolers (ac).

Building type	**Relative design capacities**			**Relative operating times**		
	$\dot{q}_{hr} \approx$	$\dot{q}_{ah} \approx$	$\dot{q}_{ac} \approx$	$R_{\tau,hr} \approx$	$R_{\tau,ah} \approx$	$R_{\tau,ac} \approx$
Apartment building	0.70	0.23	0.75	0.80	0.52	0.04
Office building	0.70	0.05	0.54	0.60	0.02	0.09

Pressure drop, drive energy to fans and pumps and thermal losses

The minimum pump power, $\dot{W}_{ep,\min}$, of a liquid-air coil is that required to overcome the heat transfer pressure drop of the coil. Fahlén [2] has shown that direct flow control with a VSD pump can reduce the required pumping power at the design capacity of a coil from 2.5 to $1.0\,\dot{W}_{ep,\min}$ and at half the design capacity the pumping powers are 1.5 and 0.13 respectively!

Using design pressure drops of an in-line system design based on data from Saetre [5] and comparing the total pressure drop with a parallel system based on the same components, the maximum difference is 25 %. A realistic savings potential would be around 10 % with the relative operating times of the presented case-study. In systems with careful design and sizing of ductwork, filters etc., the relative benefit of using a parallel design will increase.

Regarding thermal losses, Maripuu [6] has shown by practical and theoretical investigations that the temperature change can be substantial at low flow rates even if the ducts are apparently well insulated. For instance, going from an air velocity of 5 m/s to 0.5 m/s, the relative temperature loss can change from 0.2 to 0.9 over a distance of 100 m. Even from this respect, a distributed conditioning system may have advantages over a central layout.

DISCUSSION

Only a fraction of the design capacities for heating, cooling and ventilation are used on average. Hence, COD of HVAC systems can drastically reduce the energy use for space conditioning. Much of the conditioning of modern Nordic offices is in fact primarily handled by COD of the ventilation flow rate (DCV) with little use of heat recovery or heating and cooling coils. With little need for supplementary heating or cooling, the pressure drops of heat exchangers become important factors for energy efficiency. Also, when duct systems have been sized and shaped to minimize pressure drops, the dominating pressure drops will be those of filters, terminal devices, balance dampers and heat exchangers. The first two can be addressed on a component level by development of efficient, low pressure difference, air filters and terminal devices. The last two items, however, can also be mitigated on a system level.

Commonly, HVAC systems have a centralized structure, with a common supply unit and distribution via branches with balance dampers or valves. However, there are alternative solutions based on a combination of central and local distribution of fans or pumps with variable speed drives. Such designs can get rid of the balance and control pressure drops. Furthermore, it is possible to avoid the pressure drops of heat exchangers during all those hours of operation when they are out of the action by choosing a parallel arrangement instead of the common in-line system. Supply-air heaters, for instance, will often be active during less than 2 % of the year but provide a flow resistance 100 % of the time.

Separating the heat transfer pressure drop from the distribution pressure drop with a parallel arrangement also adds another dimension for optimizing the control. With in-line systems, the air flow rate is always determined by ventilation requirements and not by heat transfer conditions. Depending on the type of supply of heating and cooling, it may be more or less beneficial to use a different air flow rate than that required by ventilation. This permits better use of available temperature. For instance, large differences are desirable with district heating and district cooling whereas low differences may be of interest for heat pumps with ground storage. This will extend the period of free cooling and avoid supplementary heating during heat pump operation (low water supply temperature to air-heaters, high brine supply temperature to air-coolers). Parallel air-conditioning units and fans will also contribute to a system with high flexibility. Since adding or subtracting parallel units will not affect the operation of others, it is possible to adapt the capacity without costly changes in other parts of the system.

Capacity of air-coils can be controlled by means of the liquid supply temperature and/or flow rate and the air flow rate. The optimum combination will depend on the heating or cooling supply but moving from traditional valve control to direct flow control by means of local variable speed pumps saves on drive energy and, as a bonus, provides a simpler subsystem. The pumps may be small but they often operate constantly and the energy use is much larger than reflected by their drive power. Such parasitic powers become noticeable when the large inefficiencies of HVAC systems have been mitigated. Another example of such parasitic losses is the relative increase of heat leakage in supply ducts at reduced flow rates. At low flow, the temperature rise may render the supply temperature insufficient for the required cooling duty.

NOMENCLATURE

c	specific heat capacity [J/(kg·K)]			$\dot{V}$	volume flow rate [m^3/s]
$\dot{C}$	heat cap. flow rate [W$_{th}$/K]; $\dot{C} = c_p \cdot \dot{M}$			W	work, mechanical or electric [J]
$\dot{M}$	mass flow rate [kg/s]; $\dot{M} = \rho \cdot \dot{V}$			$\dot{W}$	power, mechanical or electric [W]
p	pressure [Pa]			Δ	difference, change (e.g. pressure, temp.)
$\dot{Q}$	power, thermal [W$_{th}$]			η	efficiency, temperature [K/K$_{max}$]
t	celsius temperature [°C]			ρ	density [kg/m^3]
	Subscripts			τ	time [s] or [h]
ac	air-coil				**Abbreviations**
ac	air-cooler	*hr*	heat recovery	BV	balance valve
ah	air-heater	*min*	minimum	CAV	constant air volume (flow rate)
bt	balance temp.	*max*	maximum	COD	control-on-demand
d	design condition	*oa*	outdoor-air	DCV	demand controlled ventilation
e	electric	*p*	const. press.	VAV	variable air volume (flow rate)
ea	exhaust	*sa*	supply-air	VSD	variable speed drive

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Simulation of the Effects of Occupant Behaviour on Indoor Climate and Energy Consumption

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SUMMARY

In this study the influence of occupant behaviour on energy consumption were investigated in simulations of a single room occupied by one person. The simulated occupant could manipulate six controls, such as turning on or off the heat and adjusting clothing. All control actions were carried out with the aim of keeping the PMV value within predefined limits in accordance with CR1752 [1]. An energy consuming and an energy efficient behavioural mode were simulated. A reference simulation was made during which the occupant had no control over the environment.

The occupant was able to keep the thermal indoor environment close to neutral when he/she had the possibility to manipulate the controls. The energy consumption was similar within each behavioural mode regardless of the PMV limits. However, the energy consumption in the energy consuming behavioural mode was up to 330 % higher than in the energy efficient behavioural mode.

INTRODUCTION

Buildings account for more than 40 % of the energy consumption in the EU member states and households are responsible for consuming more than 26 % [2]. Consequently, reductions of the energy consumption in buildings are instrumental to the efforts of alleviating the EU energy import dependency and comply with the Kyoto Protocol.

Indeed, occupant behaviour influences the amount of energy consumed to sustain a comfortable indoor environment. However, the extent to which occupant behaviour affects building energy consumption is largely unknown. The purpose of this study was to investigate the extent of this influence. This paper describes simulations of a naïve and a rational behaving occupant. The naïve occupant controlled the indoor climate using an energy expensive behaviour, while the rational occupant controlled the indoor climate in an energy efficient way.

METHODS

The simulations were carried out using a dynamic building simulation software [3]. The model consisted of a single room (4 m x 7 m) with a single occupant seated in the middle of the room. The room had one exterior wall (facing south) with a window and a heater underneath it. The building was placed in a suburban environment in Copenhagen, Denmark. All simulations were annual simulations using the Danish Design Reference Year for Copenhagen.

The simulated occupant could manipulate four different controls to adjust the environment (table fan, window opening, blinds, and heating) and two controls by which the occupant could adjust to the environment (clothing insulation and metabolic rate). All control actions were carried out with the aim of keeping the PMV value within predefined limits. Two behavioural modes were simulated. In the behavioural mode 1, the indoor environment was controlled in an energy expensive manner (naïve occupant), while the controls were operated in an energy efficient way in behavioural mode 2 (rational occupant). In both behavioural modes, three limits for the PMV index were set in accordance with the guidelines in CR1752 (+/-0.2, +/-0.5 and +/-0.7 for quality categories A, B, and C, respectively) [1], resulting in a total of six simulations. A seventh reference simulation was made during which the occupant had no control over the environment. In this simulation the occupant only controlled the clothing insulation and the metabolic rate.

Table 1: Setup of the simulations.

Criteria	Behavioural mode 1	Behavioural mode 2
A (-0.2<PMV<0.2)	Simulation 1A	Simulation 2A
B (-0.5<PMV<0.5)	Simulation 1B	Simulation 2B
C (-0.7<PMV<0.7)	Simulation 1C	Simulation 2C

In each simulation, all control actions were used to maintain PMV within the predefined limits. An example of the two behavioural modes for criterion A is given in Figure 1. Here it is seen that in behavioural mode 1, at increasing PMV, the table fan was turned on at PMV=0.03. If that did not stop the increase in PMV, the window was opened at PMV=0.06, blinds drawn at PMV=0.09, a clothing garment was removed at PMV= 0.11 and the metabolic rate was decreased to 1 met when PMV was higher that 0.14. Finally the heating was turned off when the PMV value increased beyond 0.17. When the PMV decreased below 0, the heating was turned on, the metabolic rate and clo value was increased, blinds opened, window closed and the fan was turned off, in that specific order.
In behavioural mode 2, the order of controls was inverted so the occupant turned off the heat as the first thing, when feeling warm – instead of turning on the fan.
In simulations B and C, the sequence of control actions at increasing or decreasing PMV were unchanged but the PMV value at which a control action was taken was increased according to the +/- 0.5 and +/- 0.7 criterion.

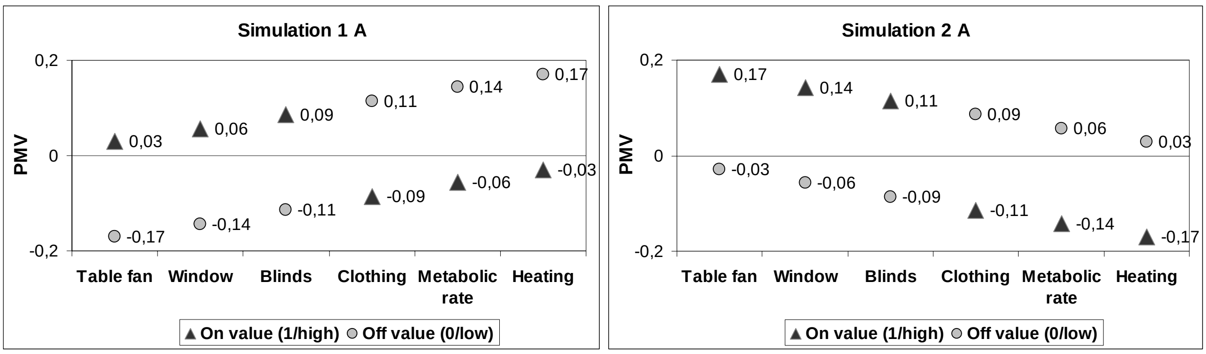

Figure 1: The control scheme for the energy consuming and the energy saving Behavioural modes for criterion A.

The occupant was constantly present in the room and was sleeping from 22:30 till 6:00 in the morning. During weekends the occupant slept from 24:00 till 8:00. During the sleeping periods the on and off values for the controls that adjusted the environment were multiplied by two to simulate that it takes a higher level of discomfort to act while sleeping than while being awake. The control of clothing and metabolic rate during the night is described below.

Table fan: When the fan was turned on the air speed at the occupant was increased by 1 m/s and the electric power consumption was increased by 50 W. These values were based on measurements using a normal table fan and a hot sphere manometer. During the measurements an average increase in air speed of approximately 1 m/s was detected when the fan was pointed directly at the occupant at a distance of 3 m. When the fan was pointed slightly to one of the sides of the occupant an airflow increase of 0.25 m/s was measured. Due to numerical problems in A simulations, the air speed was only increased by 0.25 m/s in simulation 1A and 2A. The power was kept at 50 W.

Window: The aerodynamic size of the window opening (corresponding to the size of a sliding window) was set each time the window was opened and remained unchanged until the window was closed. The opening size of the window depended linearly on the air change rate in the time step previous to the opening event. An air change rate of 0 h^{-1} in the time step prior to the opening event lead to an opening size of 0.6 m² while an air change rate of 2 h^{-1} in the time step prior to an opening event lead to an opening size of 0.12 m². When the air change rate with closed window exceeded 2.5 h^{-1}, the window was not opened even though PMV exceeded the window opening control value.
This was chosen because the air change rate depended on the wind speed outside the building. When there was a strong wind the air change rate was high and the window opening was small when the window was opened. When there was no wind the air change rate was small and when the window was opened it was opened completely
When the window was open the air speed at the location of the occupant was increased by:

$$V_{air} = 0.2 \cdot \frac{Q \cdot Vol}{A_{opening} \cdot 3600} \qquad (1)$$

Where
V_{air} is the air velocity at the occupant [m/s], Q is the air change rate [h^{-1}], Vol is the volume of the room, $A_{opening}$ is the aerodynamic area of the window opening.
The fraction in equation 1 is the air speed in m/s in the opening. The factor of 0.2 is multiplied because the airspeed decreased as a function of the distance to the window opening.
When the window was closed and the fan was off the air speed at the location of the occupant was 0.1 m/s.

Blinds: The binds were on/off controlled. They were external blinds that reduced the solar heat gain coefficient by a factor of 0.14 and reduced the direct energy transmission (short wave) by a factor of 0.09. The blinds were closed every night.

Clothing: The clothing insulation of the occupant could assume two values (Hi and Low), which were set each day at 6 o’clock in the morning on the basis of the outdoor temperature. This was done to model the action of taking on or off a piece of clothing. Both the time of day when the clothing insulation values were determined and the clothing insulation values were modelled according to [3] in the area of natural ventilation. The clothing insulation values were calculated using the following relation:

$$Clo_{Hi} = -0.024 \cdot T + 1.1 \quad (2)$$

$$Clo_{Low} = -0.015 \cdot T + 0.83 \quad (3)$$

Where *Clo* is the insulation of the occupants clothing [Clo], *T* is the outdoor temperature at 6:00 in the morning [°C].
During the night the clothing value was regulated continuously between 1.0 Clo and 2.5 Clo depending linearly on the PMV value. This was done to model a blanket or duvet that can be taken on or off in small increments while sleeping.

Metabolic Rate: The metabolic rate depended linearly on the PMV value assuming 1.0 Met at the PMV=off control value and 1.3 Met at the PMV=on control value. The metabolic rate of the occupant was 0.8 Met while sleeping.

Heating: The heating system comprised a water based radiator and a boiler with an efficiency of 66 %. The supply temperature to the radiator was 65 °C at outdoor temperatures below -12 °C and 20 °C at outdoor temperatures above 17 °C. Between -12 °C and 17 °C the supply temperature varied linearly with the outdoor temperature. The water flow through the heater was either on or off.

Infiltration rate: The simulated building had two cracks at different heights in each exterior wall. All cracks connected the interior of the building to the exterior environment. The local wind pressure coefficient of the faces of the building was determined according to [5]. The opening area of the cracks was determined by running a simulation with closed window and aiming for an average infiltration rate of 0.25 h^{-1}. In a study from 1985 [6] the average infiltration rate in 14 Danish dwellings ventilated by natural ventilation was measured. In this study, an average value of 0.19 h^{-1} was obtained, but it is stated that the 14 dwellings were among the most tightly sealed in the Danish housing mass.

Lighting: When the occupant was awake, the electrical lighting was turned on when if the daylight level dropped below 150 lux on a horizontal surface 0.6 m above the floor level at the location of the occupant (in the middle of the room). The light was only turned off when the occupant went to sleep, resulting in the light being on for the entire day if it was turned on in the morning.

RESULTS AND DISCUSSION

As seen in figure 2, the PMV index was close to neutral for a very large part of the year in all the simulations with active occupant behaviour. For the reference case with passive behaviour the PMV was far from neutral in a large part of the year (below -1 or above 1 during 72% of the year).
The PMV index was similar in the simulations with active occupant behaviour and attained values outside the control criteria for only small parts of the year. This means that the occupant was successful in controlling the environment within the comfort criteria.

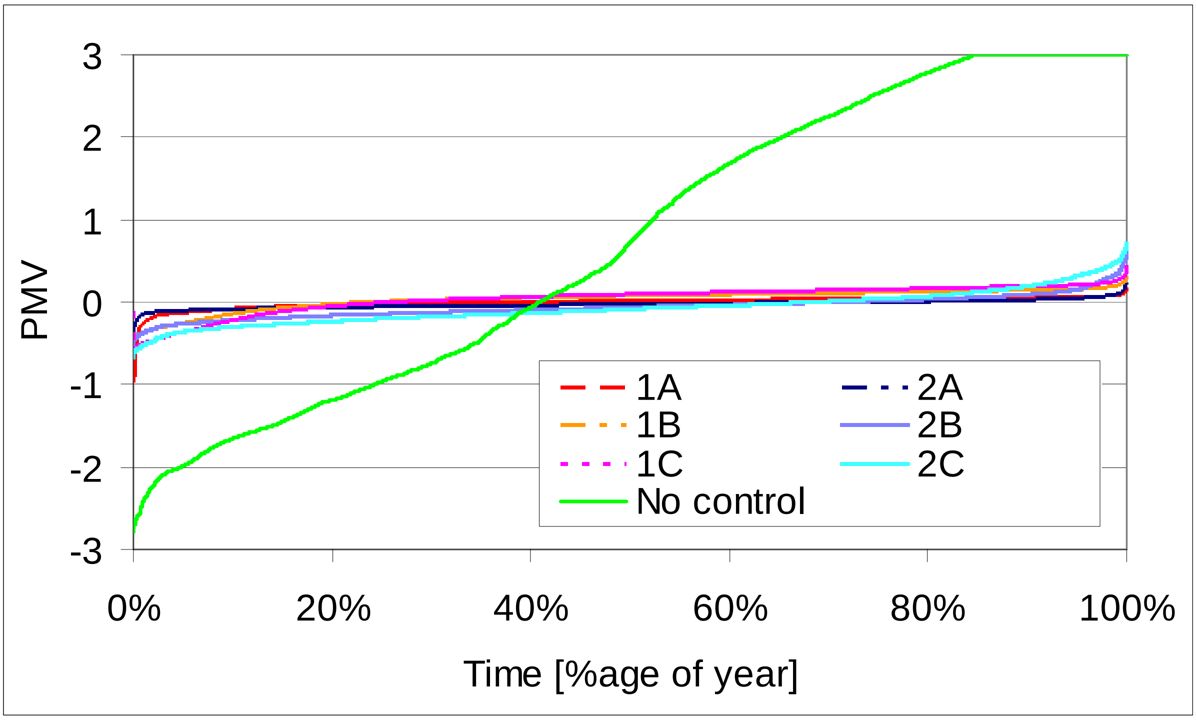

Figure 2: Duration curves for the PMV index in the 6 simulations and for the reference simulation. The figure shows how long time (in percentage of a year) the PMV index was below a certain value.

The total energy consumption refers to primary energy consumption. According to the Danish building code [7] this was calculated as the sum of all energy consumptions, where the electric consumptions were multiplied by 2.5. The highest primary energy consumption of 3948 kWh/year (simulation 1A) was 3.30 times higher than the lowest primary energy consumption of 1198 kWh/year (simulation 2C). Within each comfort control criteria the largest difference in primary energy consumption between the two behavioural modes was 324 % (3882 kWh/year simulation 1C was 3.24 times higher than 1198 kWh/year in simulation 2C). Within the two behavioural modes the largest difference in primary energy consumption was 117 % (1400 kWh/year in simulation 2A was 1.17 times higher than 1198 kWh/year in simulation 2C). This means that the comfort control criteria had much less impact on the primary energy consumption than the behavioural mode.

Table 1: Energy consumption in the simulations. The primary energy was calculated by multiplying electricity consumption by 2.5 according to the Danish building code. [7]

energy consumption pr. Year [kWh/year]	1A	1B	1C	2A	2B	2C	No control
Heating	2532	2372	2346	923	768	720	1812
Fan	380.1	423.6	431.0	1.4	0.3	0.1	0.0
Circulation Pump	13	13	13	3	2	2	13
Lighting	174	172	171	187	189	189	131
Primary energy for heating, ventilation and lighting	3948	3891	3882	1400	1246	1198	2171

Heating: The heating was turned on more often in Behavioural mode 1 than in Behavioural mode 2. Similarly, narrowing the control criteria resulted in higher energy consumption for heating.

Fan: A narrowing of the control criteria resulted in a decrease in the energy consumed by the fan in Behavioural mode 1. In Behavioural mode 2 this was opposite. In Behavioural mode 1 the fan was turned on as the first action when the occupant felt warm and turned off as the last control action when the occupant felt cold. In Behavioural mode 2 this was opposite meaning that the fan was turned on as the last control action when the occupant felt warm and was turned off as the first control action when the occupant felt cold. This meant that the fan was turned off more frequently as the control criteria narrowed in Behavioural mode 1. In Behavioural mode 2 the fan was turned on more frequently when the control criteria narrowed.

Circulation pump: In the behavioural mode 1 simulations, the heating was on for a longer period than in behavioural mode 2. This meant that the circulation pump was on for a longer period, which resulted in larger energy consumption in behavioural mode 1 than in behavioural mode 2.

Lighting: The differences in energy consumption for lighting are due to differences in daylight level. These differences were caused by differences in the use of blinds in the simulations.

CONCLUSIONS

Occupant behaviour affected the energy consumption in the building by up to 330 %. The behavioural mode affected the energy consumption in the room by up to 324 %, while the control criteria affected the energy consumption by up to 117 %.
All simulations with active occupant behaviour resulted in near neutral thermal sensation, compared to passive occupant behaviour.

The results of the study underline the importance of appropriate occupant behaviour for the consumption of energy to climatize buildings by quantifying the difference between a naïve and a rationally behaving occupant.

ACKNOWLEDGEMENT

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User control actions in buildings: Patterns and impact

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SUMMARY

In most buildings, occupants operate control devices such as windows, shades, luminaries, radiators, and fans to bring about desirable indoor environmental conditions. Knowledge of such user actions is crucial toward accurate prediction of building performance (energy use, indoor climate) and effective operation of building service systems. This paper describes an effort to observe control-oriented occupant behavior in two office buildings in Austria over a period of one year. Thereby, user control actions as related to one or more of the building systems for ambient lighting, shading, window ventilation, and heating were monitored together with indoor and outdoor environmental parameters. The collected data is being analyzed to explore relationships between the kinds and frequency of the control actions and the magnitude and dynamism of indoor and outdoor environmental changes. Moreover, implications of user actions for building performance (e.g. energy consumption) are studied.

INTRODUCTION

Multiple studies have been (and are being) conducted internationally to collect data on building users' interactions with building control systems and devices (see, for example, [1], [2], [3] and [4]). Such data can bring about a better understanding of the nature, type and frequency of control-oriented user behavior in buildings and thus support the development of corresponding behavioral models for integration in building performance simulation applications. Moreover, such data could support the effective (and proactive) operation of building service systems for indoor environmental control. The present contribution describes an effort to observe control-oriented occupant behavior in 42 offices in two office buildings over a period of one year. Specifically, states and events pertaining to occupancy, systems, indoor environment, and external environment were monitored. Weather stations, a number of indoor data loggers, and digital cameras were used to continuously monitor – and record every five minutes – such events and states (occupancy, indoor and outdoor temperature and relative humidity, internal illuminance, external air velocity and global irradiance, status of electrical light fixtures, position of shades). The results reveal distinct patterns in the collected data. Specifically, control behavior tendencies show dependencies both on indoor and outdoor environmental parameter. A summary of these tendencies are presented and their principal potential as the basis of empirically grounded user action models are explored.

METHODS

Object

Data collection was conducted in two office buildings in Vienna, Austria. One of these belongs to the Vienna University of Technology. We refer to this building henceforth as FH.

The second building is a large high-rise office complex (referred to, in this paper, as "VC"). An important feature of VC is its use as one of the major seats of international organizations, resulting in a very diverse occupancy profile in cultural terms. We selected 13 scientific staff offices in FH and 29 single-occupancy offices in VC. All selected offices in FH face east, situated on the 4th, 5th and 6th floors. Ten offices are single-occupancy, two are double-occupancy, and one is triple-occupancy. In case of VC, 15 offices face north (code: "VC_NO") and 14 face south-west (code: "VC_SW"). The offices are located on the 12th and 13th floor of the building. To exemplify the layout of these rooms, Figure 1 provides the schematic layout of two single-occupancy and one double-occupancy offices in the 5th floor of FH and three single occupancy offices in the 12th floor of VC_SW. The working stations are equipped with desktop computers and in some cases with task lights. Both VDT-based and paper-based tasks are performed.

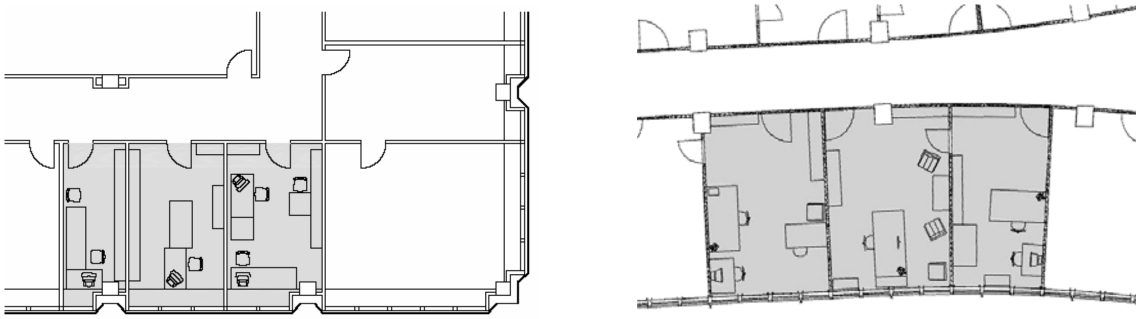

Figure 1. Schematic plan of sample offices: Left: FH; Right: VC_SW

The offices in FH are typically equipped with the followings environmental control systems: Three/four luminaries (58W each), divided into two circuits manually controlled via switches near the office door; External motorized screen shades operated by a switch mounted on a panel under the window; Fan coil under the window for fine adjustment of temperature.
The systems installed in the offices of VC_SW+NO are as follows: Three rows of luminaries with 9 or 12 fluorescent lamps (38 W) divided into two circuits and manually controlled by two switches near the entrance door; internal manually operated shading; Three to four fan coil units under each window for fine adjustment of temperature.

Monitored parameters

The intention was to observe user control actions pertaining to lighting and shading systems while considering the indoor and outdoor environmental conditions under which those actions occurred. Occupancy and the change in the status of ambient light fixtures were captured using a dedicated sensor. Shading was monitored via time-lapse digital photography: The degree of shade deployment for each office was derived based on regularly taken digital photographs of the façade. Shade deployment degree was expressed in percentage terms (0% denotes no shades deployed, whereas 100% denotes full shading). The external weather conditions were monitored using a weather station, mounted on the top of the building in case of VC_SW+NO and, in case of FH, on the rooftop of a close by university building. Internal climate conditions (temperature, relative humidity, illuminance) were measured with autarkic loggers distributed across the workstations. To obtain information regarding user presence and absence intervals, occupancy sensors were applied, which simultaneously monitored the state of the luminaries in the offices. All of the above parameters were logged regularly every 5 minutes. Monitored indoor parameters included room air temperature (in ^{o}C), room air relative humidity (in %), ambient illuminance level at the workstation (in lx), luminaries' status (on/off), and occupancy (present/absent). Monitored outdoor environmental parameters included air temperature, relative humidity, wind speed (in m.s^{-1}) and wind direction, as well as horizontal global illuminance and horizontal global irradiance (in W.m^{-2}). Vertical global irradiance incident on the façade was computationally derived based on measured horizontal global irradiance [5]. Collected data were stored and processed in a data base for further analysis. Thereby, the primary data structure follows a distinction between various types of "events" and "states" that occur at a certain point in time or persist over a certain time period [2]. For the purposes of the present analysis the range of data considered was limited to working days between the hours 8:00 to 20:00. The collected data was primarily analyzed to explore hypothesized relationships between the nature and frequency of the control actions on one side and the magnitude and dynamism of indoor and outdoor environmental changes on the other side.

RESULTS

Occupancy

Figure 2 shows the mean occupancy level in FH and VC_SW+NO over the course of a reference day (averaged over the entire observation period). Note that these values represent the presence in/at the users' offices/workstations, not merely the presence in the building. Moreover, as Figure 3 demonstrates, the occupancy patterns can vary considerably from office to office.

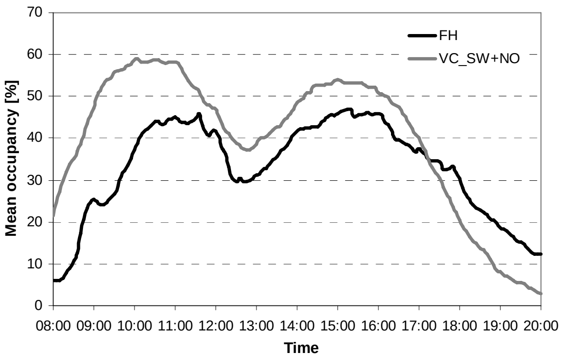

Figure 2. Mean occupancy level for a reference day in FH and VC_SW+NO.

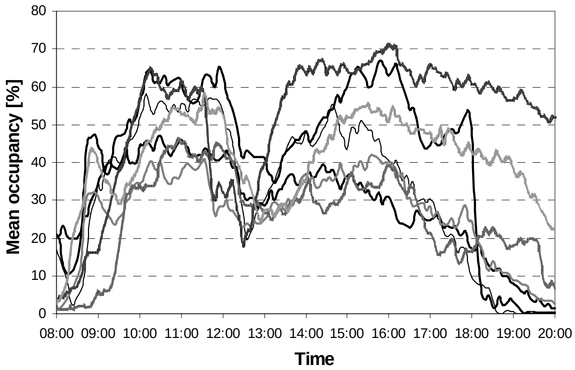

Figure 3. Observed occupancy levels in 7 different offices in FH for a reference day.

Lighting

Figure 4 shows the observed effective lighting operation in the course of a reference day expressed in terms of effective electrical power. The information in this Figure concerns the general light usage in all observed offices. Figure 5 shows the probability that an occupant would switch the lights on upon arrival in his/her office as a function of the prevailing task illuminance level immediately before arrival. Figure 6 shows the normalized relative frequency of (intermediate) actions "switching the lights on" (by occupants who have been in their office for about 15 minutes before and after the occurrence of the action) as a function of the prevailing task illuminance level immediately prior to the action's occurrence. Normalization denotes in this context that the actions are related to both occupancy and the duration of the time in which the relevant illuminance ranges (bins) applied. Figure 7 shows the normalized relative frequency of all "switching the lights on" actions (upon arrival and intermediate) as a function of the time of the day. In this case too, actions are normalized with regard to occupancy. Note that Figure 7 includes also the corresponding mean global horizontal irradiance levels.

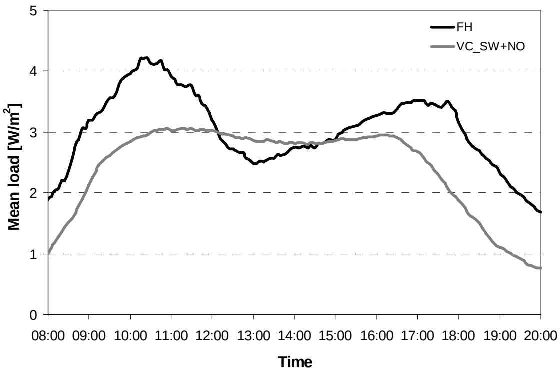

Figure 4. Lighting operation in FH and VC_SW+NO offices.

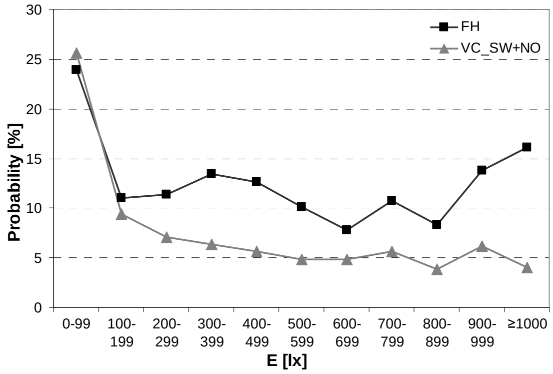

Figure 5. Probability of switching the lights on upon arrival in the office in FH and VC_SW+NO.

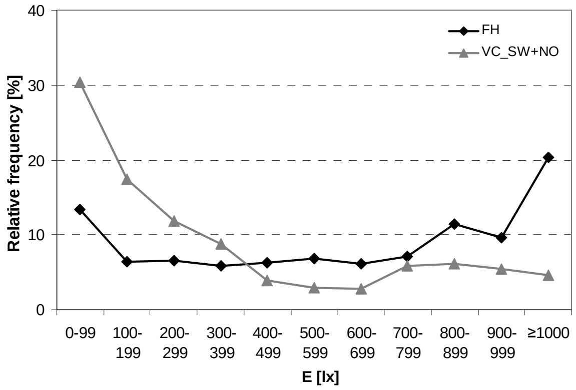

Figure 6. Normalized relative frequency of intermediate light switching on actions in FH and VC_SW+NO.

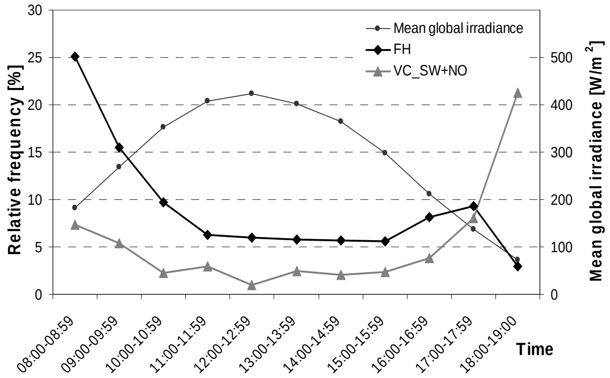

Figure 7. Normalized relative frequency of switching the lights on actions in FH and VC_SW+NO over the course of a reference day.

Figure 8 shows the probability that an occupant would switch off the lights upon leaving his/her office as a function of the time that passes before he/she returns to the office. Figure 9 shows the normalized frequency of the (intermediate) "switching the lights off" actions as a function of the prevailing illuminance level immediately prior to the action's occurrence. Normalization denotes in this case the consideration of occupancy and the applicable durations of the respective illuminance bins while deriving the actions' frequency.

Shades

Figure 10 shows the mean monthly shade deployment degree for FH, VC_NO and VC_SW. Figure 11 shows the mean shade deployment degree as a function of the incident irradiance on the façade. Figures 12 and 13 show the normalized relative frequency of the actions "opening shades" and "closing shades" as a function of global vertical irradiance incident on the facade. Normalization means that the frequency of actions (opening and closing shades) is related here to both occupancy and the duration of times in which the prevailing irradiance was within a certain range (bin).

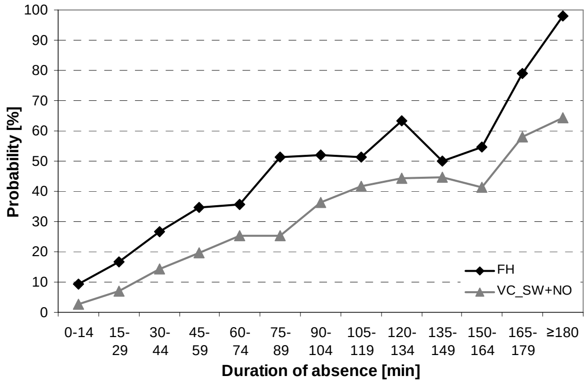

Figure 8. Probability of switching the lights off as a function of the duration of absence from the offices in FH and VC_SW+NO.

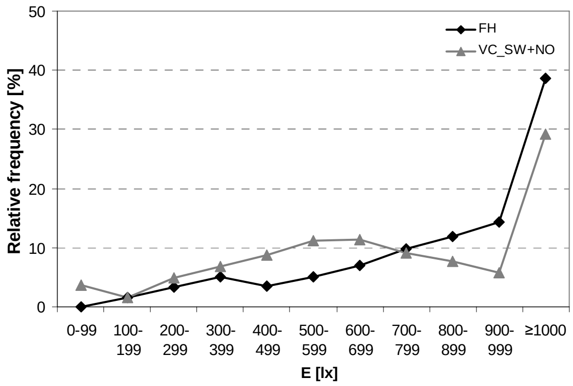

Figure 9. Normalized frequency of intermediate switching the lights off actions in FH and VC_SW+NO offices.

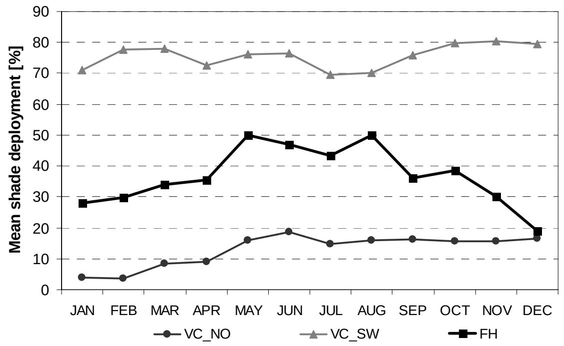

Figure 10. Mean monthly shade deployment degree in FH, VC_NO and VC_SW.

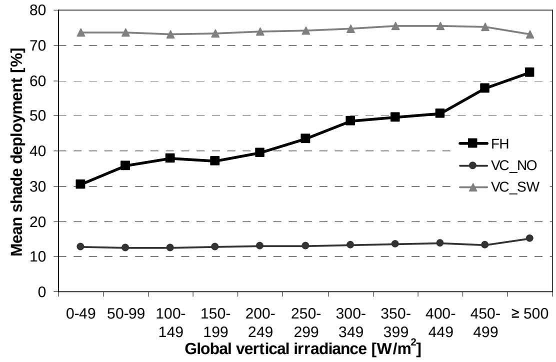

Figure 11. Mean shade deployment degree as function of global vertical irradiance in FH, VC_SW and VC_NO.

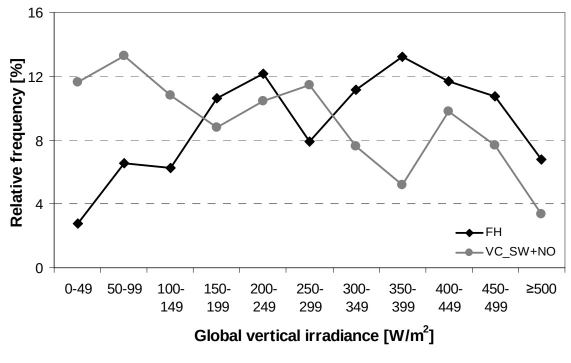

Figure 12. Normalized relative frequency of opening shades as a function of the global vertical irradiance in FH and VC_SW+NO.

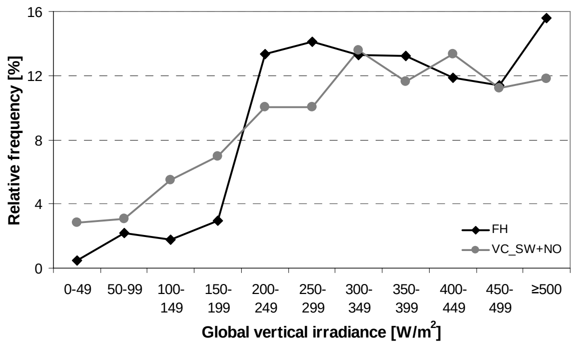

Figure 13. Normalized relative frequency of closing shades as a function of the global vertical irradiance in FH and VC_SW+NO.

Note that in the above figures opening/closing actions are not limited to actions resulting in fully opening/closing the shades. Rather, they denote a relative occupant-driven change in the position of the shades. This means that even an incremental change (e.g. changing from 80% to 40% or changing from 20% to 40%) is considered to be an opening/closing action.

DISCUSSION

The monitored occupancy in FH and VC (Figure 2) and the obviously related lighting loads (see Figure 4) reveal a similar pattern (also known from other office buildings). However, the maximum occupancy levels are different. This may be due to the circumstance that FH houses offices for teaching and research staff, who spend a considerable amount of time in classrooms and laboratories. This underscores the need for typologically differentiated occupancy models for different buildings. Patterns of this kind can be used for simulation runs in terms of corresponding hourly schedules (see Figures 14 to 16). Such simulations can be applied, for example, to explore the impact of thermal improvement measures on the building's energy use. On a more general level, our observations regarding these buildings suggest that the environmental systems in a considerable number of office buildings may in fact be "over-designed", in a sense that they are dimensioned for occupancy levels that seldom occur. The dependency of the action "switching on the lights" on prevailing illuminance levels for the two monitored buildings (see Figures 5 and 6) shows no clear pattern. The data merely suggests that only illuminance levels below 100 lx are likely to trigger actions at a non-random rate.

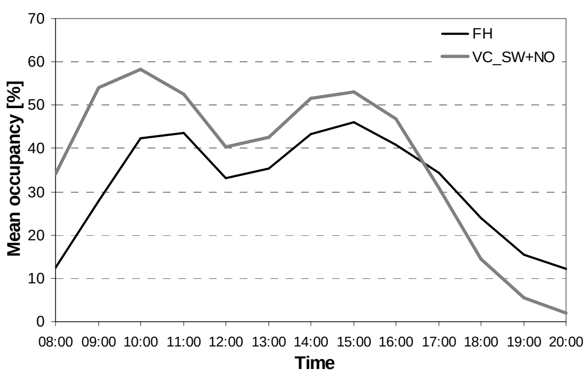

Figure 14. Illustrative simulation input data regarding mean hourly occupancy levels for FH and VC_SW+NO.

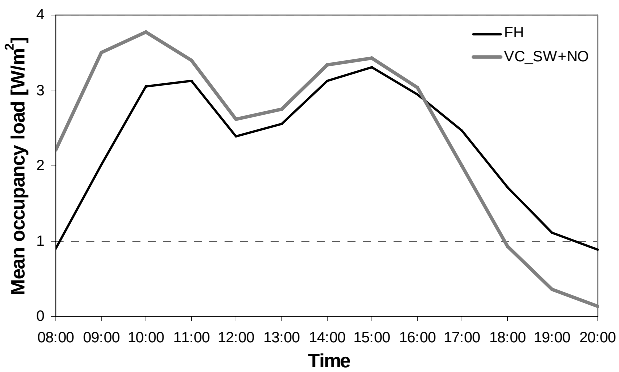

Figure 15. Illustrative simulation input data regarding mean hourly people (sensible) load for FH and VC_SW+NO.

As to the action "switching the lights off", a clear relationship to the subsequent duration of absence is evident for both FH and VC_SW+NO (see Figure 8). Occupants do switch off the lights more frequently if they are going to be away from the offices for longer periods. On the other hand, lights are not necessarily switched off by the occupants if the illuminance level in the office is already sufficient (or more than necessary) for performing typical tasks. In fact, such intermediate switching off actions appear to occur at a noticeably higher rate only once the illuminance level in the office rises above 1000 lx (see Figure 9).

The mean shade deployment levels differ from building to building and façade to façade (see Figures 10 and 11). In case of FH, where we studied the east-facing façade, a difference in the level of shade deployment can be seen between the high-radiation summer months and the low-radiation winter months (Figure 10). Moreover, an evident relationship between shade deployment and the magnitude of solar radiation is observable (Figure 11). The latter provides a very effective basis for modeling the state of shades for this building (see Figure 17). In case of VC_SW and VC_NO the shade deployment level does not vary much in the course of the year or in terms of vertical irradiance classes, but there is a significant difference in the overall shade deployment level between these two facades (approximately 75% in the case of south-west-facing façade, 10% in the case of the north-facing façade). The relative small variation range in the monthly shade deployment levels in VC_SW and VC_NO may be partly due to the fact that the manual shade operation mechanism is, in this case, much more difficult to handle than the mechanically supported shade operation system in FH.

Our observations did not reveal a clear relationship between "opening shades" actions and the incident radiation on the façade (see Figure 12). However, the corresponding analysis of the "closing shades" actions shows for both FH and VC_SW+NO a higher action frequency once the incident radiation rises above 200 W.m^{-2} (see Figure 13).

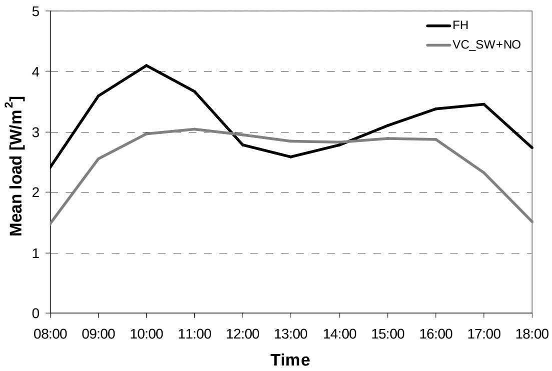

Figure 16. Illustrative simulation input data regarding mean hourly lighting load for FH and VC_SW+NO.

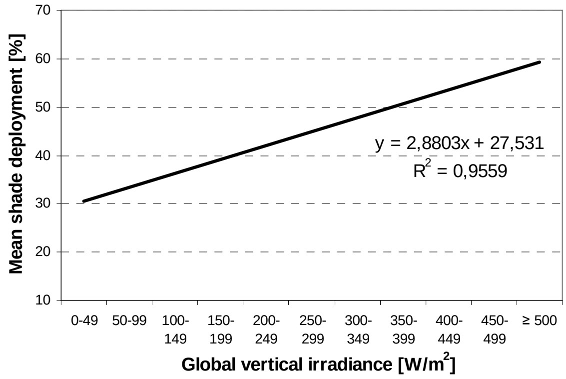

Figure 17. Illustrative model of shade deployment as a function of incident irradiance on FH's façade.

Figure 18 illustrates the potential for reduction of electrical energy use for lighting in the sampled offices. Thereby, three (cumulative) energy saving scenarios are calculated. The first scenario requires that the lights are automatically switched off after 10 minutes if the office is not occupied. The second scenario implies, in addition, that lights are switched off, if the daylight-based task illuminance level equals or exceeds 500 lx. The third scenario assumes furthermore an automated dimming regime, whereby luminaries are dimmed down so as to maintain an illuminance level of 500 lx while minimizing electrical energy use for lighting.

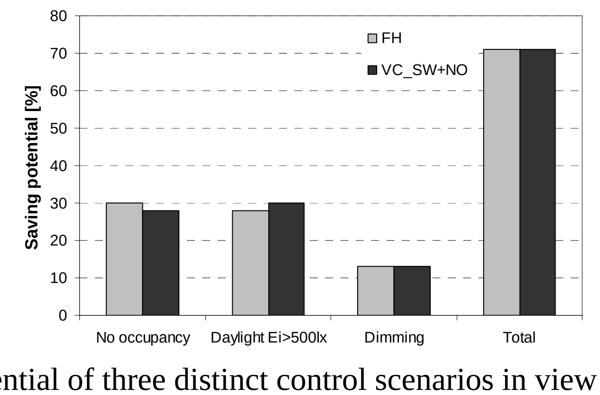

Figure 18. Saving potential of three distinct control scenarios in view of electrical energy use for lighting in FH and VC_SW+NO

The estimated saving potential in electrical energy use for lighting of the sampled offices is significant. The cumulative energy saving potential for all sampled offices in FH and VC_SW+NO is 71% (Table 1). This translates (for VC_SW+NO) into a cumulative annual energy saving potential of 17 kWh.m^{-2} or (given current energy prices) 1.3 €.m^{-2}. This would imply, that in the VC complex, annually roughly 130,000 € could be saved by a comprehensive retrofit of the office lighting system toward dynamic consideration of occupancy patterns and daylight availability. (Note that a lighting system retrofit and the resulting electrical energy use reduction would increase the heating loads and decrease the cooling loads. Given the magnitude of required cooling loads in office buildings, the overall thermal implications of a lighting retrofit are positive both in energetic and monetary terms.)

Table 1. Saving potential (electrical energy for lighting) for various scenarios and buildings

		Energy saving scenarios			
Building	Saving potential in	1	2	3	1+2+3
FH	%	30	28	13	71
VC_SW+NO	[%]	28	30	13	71
	$kWh.m^{-2}. a^{-1}$	6.8	7.2	3.0	17.0
	$€.m^{-2}. a^{-1}$	0.53	0.56	0.24	1.32

CONCLUSION

We presented a case study concerning user control actions in two office buildings in Vienna, Austria. The results imply the possibility of identifying general patterns of user control behavior as a function of indoor and outdoor environmental parameters such as illuminance and irradiance. The compound results of the ongoing case studies are expected to lead to the development of robust occupant behavior models that can improve the reliability of building performance simulation applications and enrich the control logic in building automation systems. Moreover, the obtained information will support the assessment of energy saving potential due to consideration of occupancy and behavioral patterns in office buildings.

ACKNOWLEDGEMENT

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Evaluation of Occupants' Behavior in Workplace

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SUMMARY

This paper reports on the occupants' behavior in various workplaces. The investigation was conducted by measurements and questionnaire to identify seat occupancy rate, occupancy rate and number of steps during working hours in real offices where occupants' behavioral pattern seemed different depending on work contents. Seat occupancy rate and occupancy rate of three different offices were ranging from about 20 to 50% during working hours. The seat occupancy rate and the occupancy rate of T Engineering Laboratory, in which the number of the occupants going in and out of the office seemed larger than that in other offices, were lower than those in the other offices where our previous studies were conducted. Although rather large difference among individuals in the number of steps was observed, the average number of steps an hour was about 550 steps. And the average walking distance an hour was estimated about 300 m.

INTRODUCTION

Comfort and energy conservation should be compatible in office buildings. Task and ambient air-conditioning system can be one of the solutions of that matter. In that system, chamber studies have identified that adjusting to the individual thermal preference contributed to the occupants' comfort [1] [2]. However, it has been pointed out that regular operation of task air-conditioning system may not accomplish energy conservation. General office workers are not always in their seats or around their seats. Usually in Japan, air-conditioning system is designed under the assumption that the occupied area per person is 5~10 m^2. In our previous studies, it was found that the number of occupants in office was smaller than that assumption. The capacity of existing air-conditioning systems in office may be excessive. The actual number of the occupants seated should be identified to operate task and ambient air-conditioning systems or other HVAC systems appropriately.
In this study, field investigation on the following items was conducted to identify the occupants' behavioral pattern during working hours.

1) Seat occupancy rate
- to understand the occupants' behavioral pattern in workplace for appropriate control of task air-conditioning systems in office.

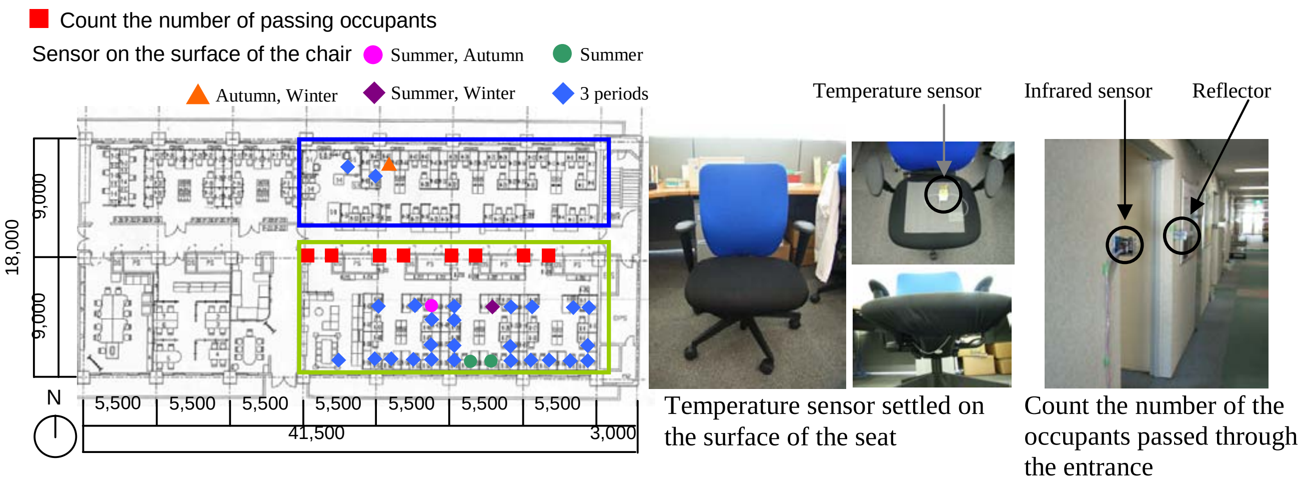
2) Occupancy rate
- to understand the proper capacity of ambient air-conditioning systems in office based on energy conservation.

3) Number of steps
- to understand individual variation in metabolic rate.

4) Place to rest
- to understand the actual conditions how and where occupants use office except when they work.

METHODS

The investigation was conducted for all 15 days in 3 seasons (summer : August 22~26, 2005, autumn : October 17~21, 2005, winter : January 16~20, 2006) on the third floor in T Engineering Laboratory.
Figure 1 a) shows the plan of T Engineering Laboratory and the measurement points. Figure 1 b) shows the picture of measuring instrument. Table 1 shows the measurement items. Questionnaire survey was conducted to the all occupants to identify their work contents and their place to rest. The temperature sensor was settled on the surface of the chair at 30 points shown in Figure 1 a). Whether he/she seated or not was judged by the change of the surface temperature of the seats in one minute [3]. The number of the occupants who passed through the entrance was counted by the measurement device that was equipped with a infrared sensor in the shoulder height (+1400mm) [4]. Occupancy rate was calculated from the number of the occupants going in and out of the office. Nineteen occupants were chosen as the subjects to measure the number of steps during working hours. They carried the memory type pedometer. The walking distance was calculated from the number of steps and his/her length of a step. It is quite usual in laboratories that the occupants often go back and forth between the office, the different room on the other floor and the other building to do experiment etc. So that it was supposed that the number of steps in laboratories was larger than that in other buildings. Similar investigations were conducted in other buildings shown in Table 2 in our previous studies.



a) Plan of T Engineering Laboratory and measurement points

b) Picture of measuring instrument

Figure 1. Outline of the investigation in T Engineering Laboratory.

Table 1. Measurement items and method

Measurement items	Measurement method
Work contents Place to rest	Questionnaire
Occupancy rate	Count the number of the passing occupants by measuring of the voltage generated by infrared sensor
Seat occupancy rate	Temperature change on the surface of the seat
Number of steps	Memory type pedometer

Table 2. Outline of the buildings in our investigations

	S Building	J Building	T Building	T Engineering Laboratory
Investigation period	September 2003	December 2004	September－November 2001	3 seasons (Aug, Oct, 2005, Jan, 2006)
Total Number of the target workers during the investigation	43	257	240	31
Work contents	Clerical work, engineering	Clerical work	Clerical work, engineering, sales and marketing	Research
Measurement items	Seat occupancy rate Occupancy rate	Occupancy rate	Seat occupancy rate	Seat occupancy rate, Occupancy rate, Walking distance

RESULTS

1) Seat occupancy rate (compared with our previous studies)

In this study, seat occupancy rate was defined as follows.

seat occupancy rate = (the number of the subjects seated) / (the total number of the subjects) (1)

The working hours in each building was defined as 9:00~18:00. Figure 2 shows the variation of seat occupancy rate and Table 3 shows the average seat occupancy rate during working hours of each building. Seat occupancy rate of T Engineering Laboratory is the average of 3 different investigation periods. The results of T building and S building are also shown.
The seat occupancy rate during working hours was ranging from about 20 to 50%. The seat occupancy rate became lower in each building at lunchtime. The average seat occupancy rate in T Bldg. was 28%, that in S Bldg. was 42% and that in T Lab. was 26%. Seat occupancy rate in laboratory was lower than those in other office buildings.

Figure 3 shows the frequency of the seating for each length of time. It could be seen that about half of the seating was within 5 minutes in each building.

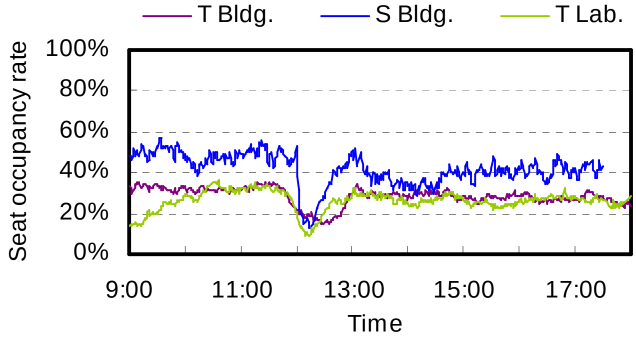


Figure 2. Seat occupancy rate during working hours of three different buildings.

Table 3. Average seat occupancy rate (9:00~18:00)

	T Bldg.	S Bldg.	T Lab.
Average seat occupancy rate	28%	42%	26%

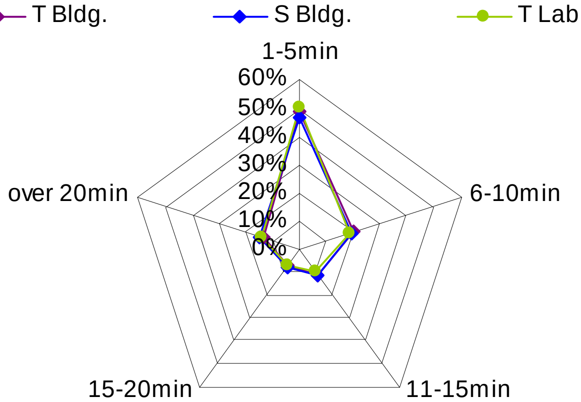


Figure 3. Frequency of the seating for each length of time.

2) Occupancy rate (compared with our previous studies)

In this study, occupancy rate was defined as a ratio of the number of the occupants in the office including the occupants who were standing or walking in the office to the number of the occupants on register.

occupancy rate = (the number of the occupants in the office) / (the number of the occupants on register) (2)

Figure 4 shows the variation of the occupancy rate in three offices. Table 4 shows the average occupancy rate of each office during working hours. Occupancy rate of T Engineering Laboratory is the average of 3 different investigation periods. The occupancy rate during working hours was ranging from about 20 to 50%.

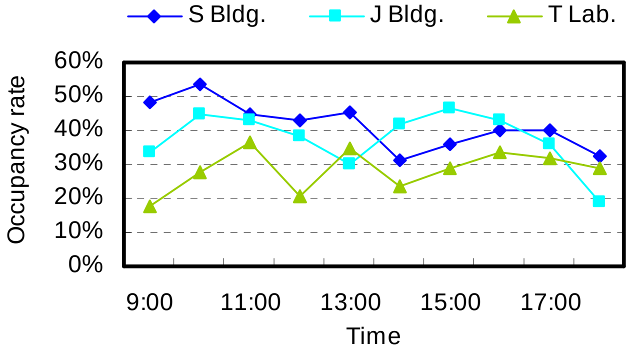


Figure 4. Occupancy rate during working hours.

Table 4. Average occupancy rate (9:00~18:00)

	S Bldg.	J Bldg.	T Lab.
Average occupancy rate	41%	38%	30%

3) Number of steps (T Engineering Laboratory)

Figure 5 shows the variation of the number of steps an hour during working hours of representative subjects in each investigation period. The characteristics of the number of steps of each subject are as follows.

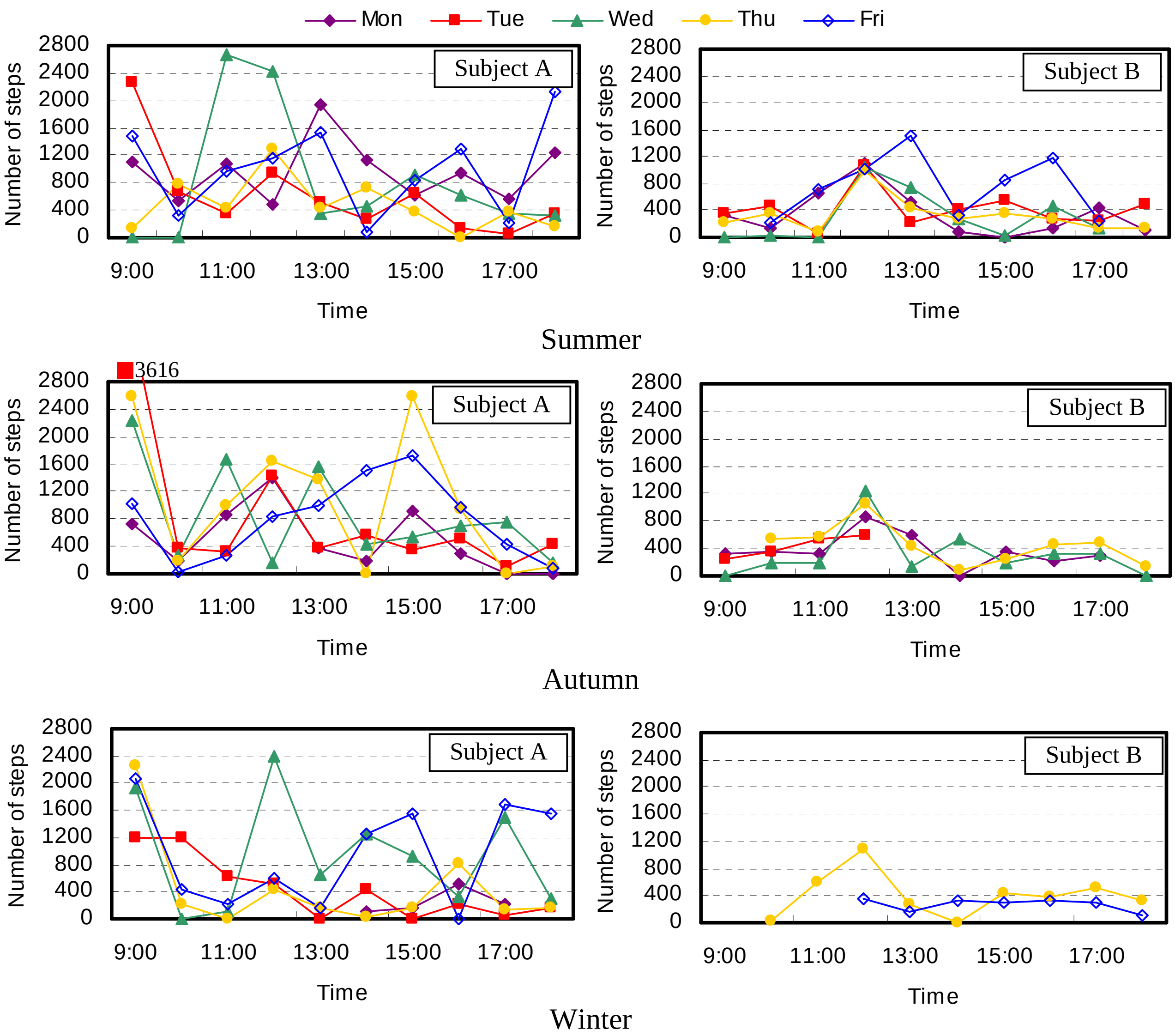


Figure 5. Number of steps of representative subjects.

Subject A

There was a large difference in the number of steps between each time. The maximum difference in the number of steps an hour was 2657 steps (see 10:00 and 11:00 on Wednesday in summer). It was supposed that there was a large difference in metabolic rate during these hours. The variation of the number of steps between each time of subject A was larger than subject B in each investigation period.

Subject B

Subject B was absent from work for 3 days in January, so the results of subject B in winter consisted of only 2 days' data.

In summer, the number of steps on Tuesday, Wednesday and Thursday showed a similar change from 11:00 to 12:00. Also the number of steps on each day except for Friday showed a similar change from 12:00 to 13:00. The number of steps of subject B was usually smaller than 800 steps excluding at 12:00 in each period. The variation of the number of steps between each time was similar in every period.

There was a large difference among individuals in the number of steps as can be seen between subject A and subject B.
It was supposed that there was a large difference among individuals in metabolic rate due to walking. Walking distance of each occupant was calculated from the following equation under the assumption that the length of each step was fixed individually. The length of a step was measured for each subject.

$$\textit{walking distance} = (\textit{the number of steps per hour}) \times (\textit{length of a step}) \qquad (3)$$

The average walking distance an hour of nineteen subjects was about 300 m. In fact, the length of a step was not always fixed. However walking distance in this study was roughly estimated, it could be one of the indices to grasp the difference in metabolic rate.

4) Place to rest (T Engineering Laboratory)
Figure 6 shows the frequency of each place to rest that was daily used by the occupants in T Engineering Laboratory. The percentage of the smokers in that laboratory was 30% and 24% of the occupants chose “smoking area” as a place to rest. That is to say, almost all the smokers in that laboratory took rest in the smoking area. The total percentage of “his/her desk” and “around his/her desk” was 40%. That is, nearly half of the occupants took rest around their desks. It can be said that the occupants did not go far from their desks to take rest.

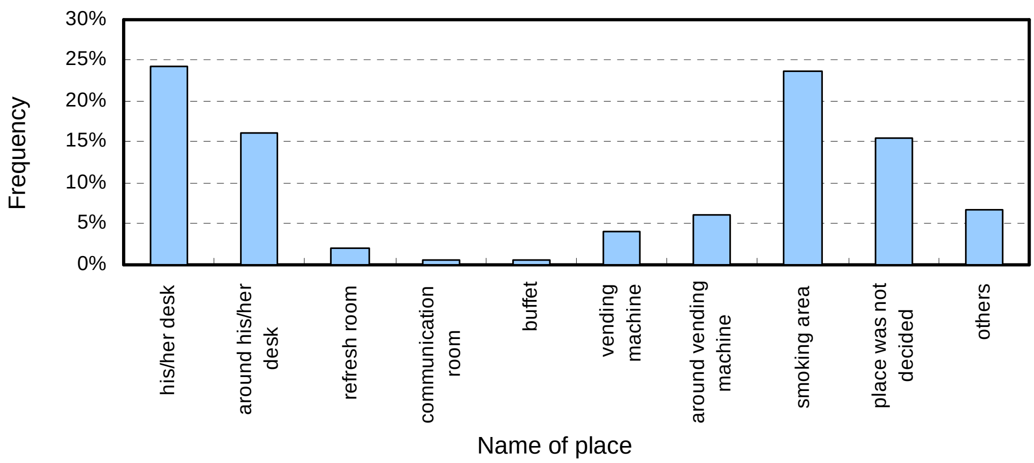


Figure 6. Place to rest.

DISCUSSION

The number and the length of the occupants stay in their task region and in ambient region are expected to differ due to work contents. Although it was supposed that the seating time in laboratories was shorter than that in general offices where clerical works, engineering and sales have done together, there could be seen a similar tendency that most of the occupants stayed in their seats for a short time (within 5 minutes) in any kinds of workplaces.
HVAC systems were so far designed under the assumption that occupants usually stay seated. However the results of the investigations in this study have identified that the time of the

occupants seated was much shorter. Also about half of the occupants on register worked out of the office. Who is the target of air-conditioning systems? Occupants' behavioral pattern should be applied to air-conditioning system design in workplaces to manage both comfort and energy conservation. The results of this study can contribute to design appropriate air-conditioning systems or HVAC systems based on the occupants' behavioral pattern. Also the results of the number of steps have indicated that there was a large difference among individuals in metabolic rate. The individual difference in metabolic rate should be considered for much meticulous control of task air-conditioning system. It is supposed that personal air-conditioning system is effective to resolve discomfort due to the individual difference in metabolic rate. Moreover about half of the occupants took rest around their desk. The environmental design based on appropriate evaluations of the occupants' behavioral pattern may achieve occupants' comfort and energy conservation in office building simultaneously.

CONCLUSIONS

In this study, occupants' behavior in workplaces was evaluated by seat occupancy rate, occupancy rate, number of steps and place to rest. The following results were obtained.

1) Seat occupancy rate during working hours was ranging from about 20 to 50%. Seat occupancy rate at lunchtime became lower in each building. About half of the seating was within 5 minutes in each building.

2) Occupancy rate during working hours was ranging from about 20 to 50%.

3) There was a great difference among individuals in the number of steps.

4) Almost all the smokers in the office took rest in the smoking area. About half of the occupants did not go far from their desks to take rest and took rest around their desks.

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Using Semiotics to Understand the Interplay Between People and Buildings

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SUMMARY

The purpose of this study is to examine the usefulness of semiotics in deepening our understanding of the interplay between people and buildings, and for providing the basis for personalizing intelligent buildings using intelligent software agents. The context is the increasing demand for improved working conditions at the same time as conserving energy. Semiotics, the study of signs, has a wide and varied scope. The study first decides relevant approaches; the two approaches used here are i) considering design as communication and ii) particular techniques from a branch dealing with organizations, Organizational Semiotics - *semantic* and *norm* analysis, including *ontology charting*. These methods are applied to the overall building scenario as well as the personalization system. Recommendations for designers are to use this approach to focus on designing to influence behaviour, and to use semantic and norm analysis to capture and embed norms explicitly.

INTRODUCTION

The overall aims of our work are increased well-being and energy conservation in indoor environments. In this context there are the following specific motivations:

- To encourage the saving of energy. The recent Stern Review [1] proposes three policies to address the threat of climate change: carbon pricing, innovation, increasing awareness and encouragement of energy saving. Each of these policies provides motivation for understanding better how people interact with building systems.
- To apply and investigate new technology impact, e.g. ambient intelligence, sensors.
- To understand more about individual responses to the environment, to be able to personalize the environment to improve health, general well-being and productivity.
- To protect and enhance people's well-being indoors. As a global average, we now spend 80% of our time indoors [2] and people are living longer, increasing their interest in what habits and conditions are important short and long term.
- To create better models and predictions. Predictions from physical models often only account for a portion of actual performance, sometimes due to human factors.

Intelligent Buildings

A variety of definitions have been proposed for the term intelligent building, but for our purposes we consider two main approaches: one view based on the use of technology [3], and the other view as a broader concept based on a mirroring of human intelligence, where the building behaves intelligently in a holistic sense, within its environment [4]. Both these definitions are useful for this discussion: technology gives us new kinds of environments to investigate (how will people react to and use these new environments?); energy conservation and well-being are both part of sustainability (how can we encourage people to interact with intelligent buildings to improve sustainability?).

The specific context of our work is a project to personalize intelligent buildings, extending work of a number of research teams [3] [5] [6] [7] [8]. These systems use artificial intelligence and sensor networks to control environmental variables such as temperature, light, sound and air quality. The design of such systems provides requirements for the study of building/people interplay, though our perspective is not limited to these, because of the more general motivations described above. Figure 1 gives an overview of our proposed system which involves sensor networks sensing the environment and the people's actions, a multi-agent system representing the people, and controlling actuators. The horizontal part shows the sensors giving environmental and occupancy data to the agent system, which controls the actuators. The diagonal part shows the involvement of people: the occupants are represented by their individual preferences learnt from data collected by the personal agents and by direct input; the other stakeholders, such as the facilities manager and the owner, supply policy in the form of rules, or norms, relating to the allowed and desired behaviour of the personal agents. A key aspect of the design is the provision for occupants to be actively involved in the system operation [9].

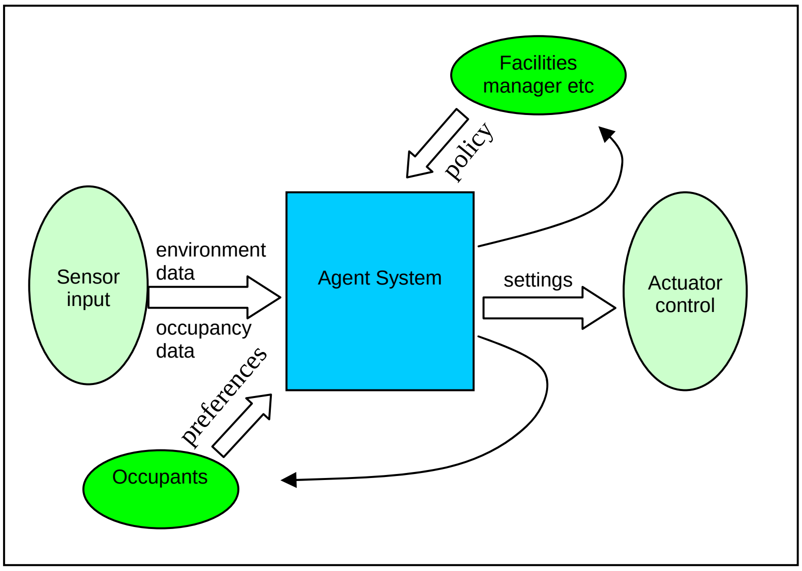

Figure 1. Overview of multi-agent system for control of the building environment

Semiotics

We propose using semiotics to deepen our understanding of the interplay between people and buildings, because semiotics, the study of signs, concerns how things are signified to us. What messages is the building giving us, visually, and through our other senses, directly (experientially) and indirectly (via information)? How do we know what we can do in a building? How do we understand how to do those things? How can buildings elicit responses from us? Semiotics, the study of signs, is a wide and varied field used across disciplines (art, advertising, linguistics and others) (for an introduction see e.g. [10]). A broad definition of semiotics is given by Eco: 'semiotics is concerned with everything that can be taken as a sign' ([11], p7, quoted in [10], p2). Thus, a sign is anything that can stand for something else – encompassing words, images, sounds, gestures and objects. Semiotics became a major approach to cultural studies in the late 1960s, but has not played such a role in computer science, though it is considered by some as providing a much needed part of knowledge representation [12], and of particular relevance for human computer interaction [13]. A development specific to the examination of communication and information in organizations, Organizational Semiotics (OS) [14] gives specific concepts and methods for examining organizations. As semiotics has been applied so widely, there are a variety of methods available. Within the context given above, and to obtain specific as well as general results, our objectives are to identify semiotic techniques that are appropriate for analyzing people/building interplay, and then apply the identified techniques.

METHODS

This section describes the semiotic approaches chosen for our context: *design as communication,* and specific techniques from OS.

Design as communication

A particular application of semiotics considers how the design communicates to the user. We can consider a large object, such as a building, or a small object such as a soap dish [15]. The message that an object gives us has intrinsic and extrinsic aspects. Intrinsic aspects concern the object itself, for instance whether it has a handle of some sort, inviting us to pick it up – 'by their look they invite certain actions' ([16] quoted in [17], p234). Extrinsic aspects concern such things as the context (e.g. where the soap dish is placed), and our own knowledge, deductive powers and experience. A consideration of the communication aspects of an object constitute good design, which may or may not include a conscious evaluation of the product as message. [17] discusses this issue in terms of 'stimulus-response capability', giving examples of door latches and numeric keypads, but also stressing the difficulty of making explicit the user's implicit knowledge and the need to watch what people do, not just listen to what they say. From a different perspective, [18] examines shopping malls, exposing their meaning of 'consume' on different levels. [19] examines how an organization's values, ethos, beliefs and behavioural codes are 'made visible' in the workplace.

Signs can be classified in different ways, which assists breaking down the message into its components. Pierce (1839-1914), one of the founders of semiotics, suggests three main forms of signs: i) symbol/symbolic: fundamentally arbitrary; ii) icon/iconic: resembles the signified; iii) index/indexical: indicates the sign in some way, so that the sign can be inferred (natural examples are smoke, thunder and footprints).

Organizational Semiotics

OS [14] provides specific methods for examining communication and information in organizations. A key concept in OS is that of *affordance*. An affordance refers to functionality (action that is *afforded*) that is provided by an environment to a responsible agent. By analyzing descriptions, problem statements, and other material, the agents, environments and affordances can be established; then a diagram can be drawn showing these relationships. Such a diagram is referred to as an ontology chart. Ontology charts can be used to gain insight into a problem and as a way of eliciting and representing user requirements. *Norms* can then be associated with affordances; these norms are the rules governing the normal use of the affordance.

RESULTS

In this section we examine the interplay from the point of view of design as communication on the macro (the overall building, building system) and micro (individual products such as radiators) levels, then apply semantic and norm analysis to the personalization system design.

Design as Communication

In recent years much effort has been expended upon creating more efficient products, including applying whole-life energy analysis, but less emphasis has been given to designing products that *encourage* energy-efficient *use*. In other words, the product as a *communication of how to reduce energy* is not considered. Designers are beginning to address this, for instance the Toyota 'EcoDrive Indicator' gives feedback to the driver – showing when they are driving in a fuel-efficient manner [21]. Since human use of products has so much impact on energy consumption, attention is beginning to focus on buildings (e.g. the 'Wattson', http://www.diykyoto.com/). Three ways that the product can encourage energy conservation have been identified by Lilley et al. [22]:

1. Intelligence. By observing the user's interaction and automatically adjusting to be more energy efficient. This can include decreasing irresponsible or anti-social behaviour.
2. Behaviour steering (scripts, affordances and constraints). By encouraging the user to use the product in a more energy efficient way through design.
3. Eco-feedback. Providing information to users to encourage pro-environmental behaviour.

For a wider view, specific to building control systems, consider these three together with the four manual and automatic control features given in [23], which also stresses the importance of designers making their 'intended operating strategies obvious, convenient and effective'.

If we consider buildings and building systems from the point of view of the messages they are giving us, we are asking not, do the spaces have the facility for X?, but, how is the facility for X indicated to the user? and how (assuming we want to encourage X) is the use of X (or a particular use of X) encouraged? When we enter a room, we see windows that can (maybe) be opened, a radiator that can be turned on etc. However, nothing tells us that we need to save energy, or how to save energy. We simply perceive (and expect) the freedom to use the objects as we want. So the system conveys to us the message that energy is free and available - the same message that has been driving society to use more and more energy.

Now consider the signs involved for an individual actuator, such as the thermostatic radiator valve in Figure 2, relating to existence, functionality, and how to operate.

Figure 2. Thermostatic Radiator Valve

- Firstly we can see that there is a control, the shiny silver top attracts us.
- It *invites* us to twist it by its shape. By handling it we can see that it twists clockwise and anti-clockwise. Anti-clockwise is iconic with opening.
- We can see numbers on the side and these go up when we turn the valve anti-clockwise, though they are on the side and so we can't see them very well. These suggest that the temperature will go up.
- There is a red band that gets wider as the valve is opened. This is iconic, red ≡ hot, so we think that opening ≡ hotter.

However, there is an ambiguity here. Does this mean that the radiator is hotter, or that the temperature of the air around the radiator is hotter? It seems that it should refer to the radiator as the valve is attached to the radiator. We are misled by the 'opening' of the valve - perhaps if it is partly opened, the radiator will be partly on? Of course, this is not the case, as this is a thermostatically controlled valve, so is sensing the temperature around the valve. This example shows that we need prior experience to read the conventions (e.g. twisting clockwise), but further understanding to fully control the device. Here the designers have included signs of functionality, but ambiguity remains without further knowledge.

Organizational Semiotics

Now we apply semantic and norm analysis techniques from OS, so that we can provide concrete specifications and, particularly, so that we: i) avoid loss of signification in moving toward automated systems (by analyzing the manual operation); ii) elicit and embed functionality and norms in automatic systems, so that they can be updated. Semantic analysis has four phases: problem definition, candidate affordance generation, candidate grouping and ontology charting [14]. A partial listing of semantic units extracted from the problem statement for the personalization system is shown in Table 1.

actuator action (of occupant) agents air quality (ventilation and CO2 levels) allowed alters amount (of clothing) ….	feedback goals group (of occupants) health and safety (agent) heart rate how good how well identification ….	personal agent personal profile personalizes policy preferences presence record repositories ….

Table 1. Partial listing of semantic units for the personalization system

From the semantic units, agents and affordances are identified, and then arranged as possible, or *candidate*, groupings. By combining these candidate groupings, a complete ontology chart can be drawn. Note that by following the procedure, the chart is derived exclusively from the requirements documents, so that it adopts the language and logic of the problem owners. The ontology chart for the proposed personalization system is too large to show here, so the manual situation is shown in Figure 3. A database structure can be generated from the ontology chart, for example in the form of a semantic temporal database (STDB) [14].

Norms govern the way individuals behave, think, judge and approach the world within a community [14]. Norms can be categorized in different ways; here we consider the category of behavioural norms, as most rules and regulations in organizations belong to this category. These concern what people must, may, and must not do, equivalent to the deontic operators ‘is obliged’, ‘is permitted’ and ‘is prohibited’. The general form of such a norm is

whenever <context>
if <condition>
then <agent>
is <deontic operator>
to <action>

Applying this to the building system example:

whenever a person occupies a room
if the person is staying in the room for a while AND (is an employee of the organization responsible for the room OR has been invited by the organization)
then the person
is permitted
to alter the room controls (window, actuators)

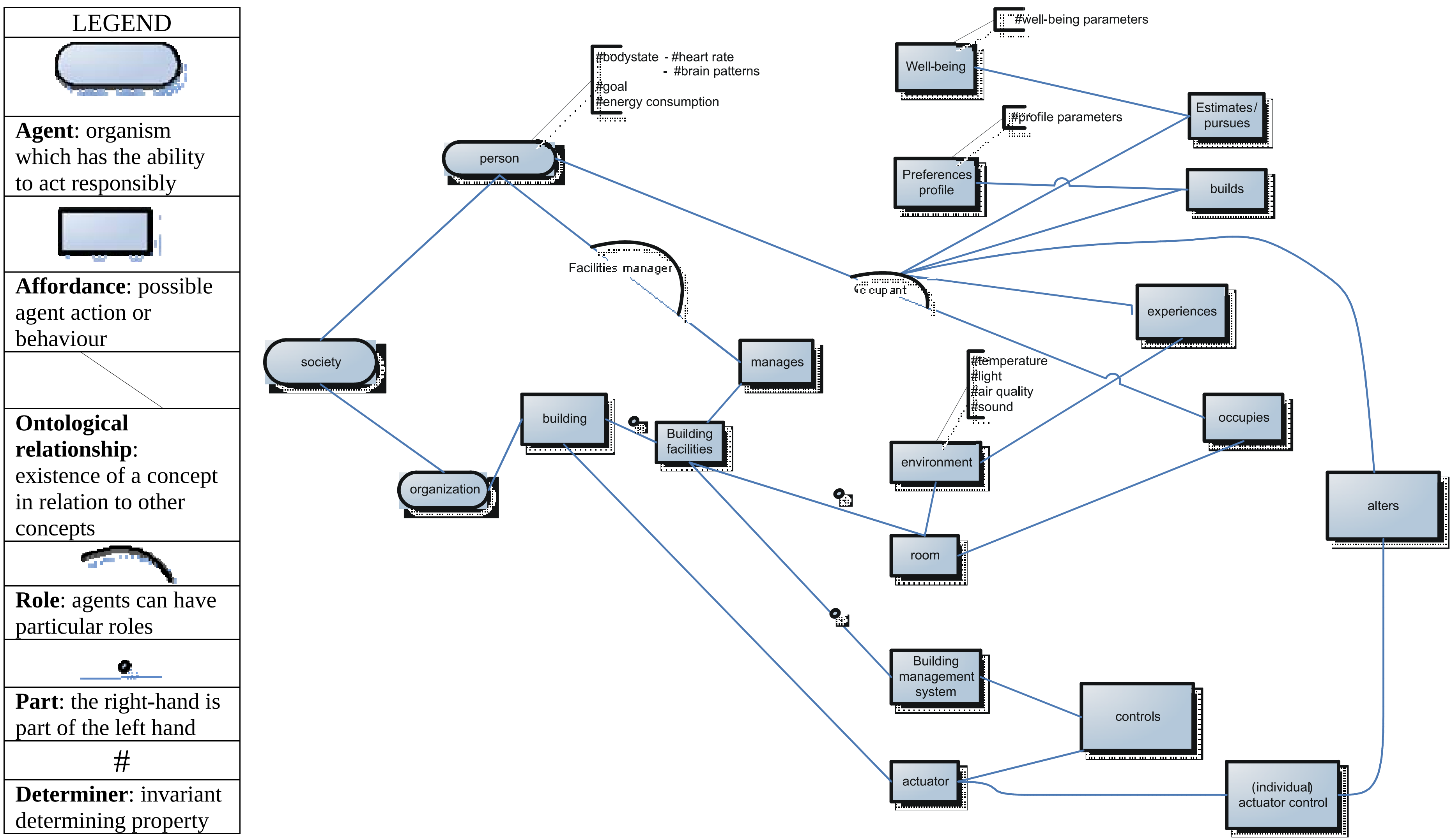

Figure 3. Ontology chart for manual control of environment

These norms can be kept computationally in a norm store, providing conditions and constraints for the system operations and being associated with the relevant affordances in the STDB. In this way, appropriate norms can be invoked whenever an action is requested. For the building example given above, the context is a person entering a room. If that person meets certain criteria, they are given permission to alter the room controls. The example can be translated into a norm specification language as follows:

```
whenever staying(person,room) AND (employs(organization,person) OR
invites(organization,person))
permitted to start alters(person,controls)
```

This shows that this norm is connected with the affordances ‘staying’, ‘employs’, ‘invites’ and coupled with the affordance ‘alters’ in the STDB, and the inner brackets show each context and agent associated with each affordance. Every time there is an operation *`to start alters(person,controls)`*, this norm will be invoked. Thus, each norm is associated with patterns of activity described in the ontological chart, so that the requirements of the system can be defined in detail. The condition specification given here is only one of many possible, for instance this one could be changed and only specify the owner of the room to alter the room controls. Capturing this level of detail, having a way of storing it in the system, and the ability to access and change it later (as well as the possibility of discovering norms by observing behaviour) is of importance for autonomous systems which replace manual operations and need to be able to change and evolve at the same time as involving a variety of stakeholders.

DISCUSSION

The ‘design as message’ approach enables us to focus on the overall message of the design, as well as affordances and constraints. Currently little is being done to convey energy conservation messages within the product itself. We are relying on general education, but this is inadequate, especially since there is actually a constant rise in electricity consumption due to electrical goods. The energy demand in the home from acquisition of more electrical equipment has increased by 70% between 1970 and 2001 [24]. The radiator thermostat example suggests a means of decomposing the signage of products, and can be applied to other levels, such as the overall system and the software application. Semantic analysis provides a different approach for the elicitation, verification and recording of requirements, which can be used for the design phase, as well as the embedding of behaviour norms. This will help to see whether the product conveys the message the designer wanted, as well as measuring usability and functionality [13]. Norm analysis gives a form for capturing behaviour as rules and thus provides an explicit representation for computational use.

This initial study shows that semiotic techniques are promising and indicates the value of furthering the work. Specific recommendations are to:

- Apply ‘design as communication’ perspective on different system levels.
- Use ontology charts to capture the signification from the start.
- Use norm embedding to enter rules explicitly and enable storage and updating.

ACKNOWLEDGEMENT

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Variable ventilation airflow rate in dwellings – costs and benefits

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SUMMARY

Ventilation systems with variable airflow rates (VAV) can be used to decrease the amount of energy used to heat and cool the supply air and move the supply and exhaust air. Additionally, the occupancy detection system can work together with the heating and cooling system and decrease other energy uses by changing the indoor climate demands when building is not occupied. However, a VAV system has higher installation and maintenance costs than a system with constant airflow rate (CAV). This paper gives examples of energy use and life cycle costs (LCC) for different ventilation systems in dwellings. The LCC was calculated with a computer program for LCC calculations of indoor climate systems, ProLive. The occupancy level of a building is one of the most important parameters regarding the energy use calculations of VAV systems. The results show that it can be beneficial with variable airflow rate in dwellings from an LCC perspective.

INTRODUCTION

People spend up to 90% of their time indoors [1]. To ensure people’s health and comfort when they are indoors, the indoor air quality and thermal comfort must be appropriate. An indoor climate system, consisting of heating, cooling and ventilation, serves this purpose [2,3]. The aim of the indoor climate system is to provide the building with a good thermal comfort and indoor air quality. The indoor climate system must also use as little energy as possible.

This paper focuses on the ventilation systems and the airflow rates of the ventilation systems in detached houses and multifamily dwellings. Each ventilation system can either have a constant airflow rate (CAV) or a variable airflow rate (VAV). The idea behind having a variable airflow rate is to cool with air, to heat with air, or to decrease the airflow rate at lower occupancy levels to ventilate when there is a need for it only. Variable airflow rate ventilation based on the demand for air only is sometimes called Demand Controlled Ventilation (DCV). Ventilation systems with variable airflow rate based on the demand for air can be used to decrease the used amount of energy to heat and cool the supply air and move the supply and exhaust air. However, a variable airflow rate ventilation system has higher installation and maintenance costs than a system with constant airflow rate.

This paper presents the life cycle costs (LCC) for different ventilation systems in dwellings both with and without variable airflow rate based on the occupancy. In dwellings in Sweden, variable airflow rate is very rare except for the fact that the stove hood often can be set to different airflow rates. This is a smaller part of the total airflow rate and the idea is not to reduce the airflow rate at low occupancy levels.

Objectives and limitations

The life cycle costs of the heating and ventilation systems were simulated using a computer program for life cycle costs of indoor climate systems, ProLive. The life cycle costs were done both with and without variable airflow rate for multifamily apartments and detached houses.

The occupancy level of a building is one of the most important parameters regarding the application of variable airflow rate ventilation systems. The occupancy level has not been measured. However it has been a parameter in the calculations. Only theoretical buildings have been simulated, one typical multi family dwelling and one typical detached house. Field measurements were not an alternative, since it would be difficult to measure the life cycle cost, particularly for variable airflow rate ventilation, that rarely exists in dwellings.

METHODS

One problem with the design of an indoor climate system is that there has been a predominant focus on initial costs. A life cycle approach could improve both the energy and economic performance of the indoor climate system. The life cycle cost of a product is the sum of all costs related to that product over its entire life span. Future costs are discounted to the value of today, the net present value, by the use of a discount rate of interest.

The computer program ProLive was developed to calculate life cycle costs for indoor climate systems in offices, schools and dwellings [4]. It handles thousands of combinations of heating, cooling and ventilation systems typical for Sweden and takes into account the initial costs for buying and mounting components through a power demand calculation, energy costs, maintenance costs, repair costs and costs for space loss due to system components. Costs are based on Swedish prices from Sektionsfakta [5], which is a known cost database for the building sector. Outdoor climate is obtained from the computer program Meteonorm [6], which simulates outdoor climate data for the entire world. ProLive has features to model productivity and health costs based on indoor temperature and airflow rate even if the user has to provide the correlation data between the parameters and the costs. ProLive uses Swedish costs in SEK excluding VAT (25% value added tax). 1 SEK ≈ 0.14 US$ ≈ 0.11 € as of 2007-02-15.

In this study, the price of heat was set to 0.6 SEK/kWh and the price of electricity was set to 0.8 SEK/kWh excluding VAT. The discount interest rate was assumed to be 1% for electricity, 2% for heat and 3% for other costs representing a real price increase for heat and even more for electricity. An annual value of 800 SEK/m² was assumed for space loss. It was assumed that there is a 20% deduction from the initial costs for both air and hydronic components compared to Wikells [5]. A 40 year calculation period was used. The scrap value was assumed to be zero. The occupancy detection cost was assumed to be 766 SEK, which is supposed to be equal to an IR occupancy sensor and the mounting of it. This could consist of a manually operated switch inside each apartment and the detached house respectively.

Buildings

None of the buildings were set up with cooling. Hydronic radiators were used for heating. Assumed data for the buildings are given by Table 1. The outdoor climate data are from

Stockholm, Sweden. Table 2 gives the apartment areas, the window areas, the heat transmission areas and the number of people per apartment. The rooms with supply devices are the rooms that are not kitchens or bathrooms. Exhaust devices are located in the kitchen and in the bathrooms.

The multifamily dwelling was assumed to have four storeys with two one-room, two two-room, two three-room and two four-room apartments on each storey. It was assumed that there were two exhaust devices for each apartment, except for the four-room apartments where it was assumed that there were three exhaust devices. The building was assumed to be a medium weight construction.

The detached house was set up with two storeys. There were three exhaust devices on the bottom floor and two on the second floor while there were two supply diffusers or air inlets on the bottom floor and three on the second floor. It was assumed that four persons were living in the house.

Table 1. Data for the simulated buildings. Internal load refers to the internal heat gain excluding people. Heat transmittance is the average value for the building.

	Detached	Multi family		Detached	Multi family
Storey height	3 m	3 m	Chiller COP	2	2
Building length	12 m	43.3 m	Room temperature	22°C	22°C
Building width	8 m	12 m	Internal load presence	3 W/m²	3 W/m²
Total floor area	192 m²	2080 m²	Internal load abscence	1 W/m²	2 W/m²
Heat transmission area	288 m²	Table 2	Heat recovery eff.	0.8	0.8
Window area	15 m²	Table 2	Heat plant eff.	90%	0.9
Heat transmittance	0.25 W/(m²·K)	0.365 W/(m²·K)	Solar rad. trans.	0.5	0.4
Supply air temperature	18°C	18°C	Leakage at 50 Pa	0.8 l/(s·m²)	0.8 l/(s·m²)

Table 2. Data for the multi family dwelling. The building was set up to consist of two of each of the listed apartments per storey and four storeys.

Nbr of rooms/apartment	1 room	2 rooms	3 rooms	4 rooms
Nbr of persons/apartment	1	2	3	4
Window area	3 m²	6 m²	8 m²	9 m²
Apartment area	30 m²	50 m²	80 m²	100 m²
Heat transmission area	34.2 m²	57.0 m²	91.2 m²	114 m²

Ventilation systems

Specifications of the indoor climate regarding thermal comfort and air quality are determined by requirements, recommendations, national regulations, or by the building's user. This helps to simplify the design process of an indoor climate system. Usually, the minimum and maximum temperatures and the supply airflow rate are set according to the amount of activity in the building. In Sweden, the requirement is currently 0.35 l/(s·m²), where the area refers to the floor area [3]. If nobody is in the building, there is no need for airflow at all if no situation may arise with health risks indoors. This can be solved by a certain minimum airflow rate or by starting the ventilation a certain time before the house or apartment is occupied. Since it requires planning from the user, the most realistic way is to set a minimum airflow rate.

The systems included in this study were, exhaust ventilation system (E) with air inlets at the windows in the rooms that were not kitchens or bathrooms, and supply and exhaust ventilations system with heat recovery (SEH) with ceiling diffusers for supply air. Constant airflow rate and variable airflow rate were simulated. In the case of the variable airflow rate system, an airflow control damper (two for the supply and exhaust system) and an occupancy detector for 766 SEK were assumed to be included for each apartment in the multifamily dwelling. That means that the airflow rate was varied for the entire apartment and not for each room in each apartment. For the detached house, it was assumed that the airflow rate for the entire house was varied in the same manner. There is need for an occupancy detector but not for dampers since it is enough to control the air handling unit, and that option is included in air handling units on the market.

Occupancy

The occupancy rate is noted O_P. It is the ratio between the actual number of people in the building at a certain time and the number of people the building is designed for. In this study, it was assumed that O_P was constant over the year, which reduces the number of parameters. Johansson [5] analyzed this simplification. The fan electricity was influenced slightly by the use of a time distributed occupancy rate since the fan power is not linear to the airflow rate. The energy for heating and cooling is influenced even less.

$O_p = 0$ means that the building is unoccupied. $O_P = 1$ means that the building has full occupancy. It is assumed that the designed airflow rate of 0.35 l/(s·m²) is needed when $O_p = 1$ and that it is reduced linearly according to Equation 1 when O_P decreases. The coefficient α sets the airflow rate ratio compared to 0.35 l/(s·m²) at $O_P = 0$ and q is the airflow rate in l/(s·m²) where the area refers to the floor. In the study, $\alpha = 0$ and $\alpha = 0.33$ were tested. At $O_P = 0$, that means an airflow rate of 0 for $\alpha = 0$ and 0.1155 l/(s·m²) for $\alpha = 0.33$. The occupancy rate also influences both the internal load from people and the internal load, excluding load from people according to Table 1.

$$q = 0.35 \cdot \left(O_P + \alpha\left(1 - O_P\right)\right) = 0.35 \cdot \left(O_P\left(1 - \alpha\right) + \alpha\right), \qquad (1)$$

RESULTS

Table 3 gives the resulting life cycle costs for the different simulated buildings and ventilation systems at $O_P = 60\%$ and $\alpha = 0.33$. From here, it is shown that it is best to live in a detached house with a supply and exhaust ventilation system with heat recovery and variable airflow rate. In the costs hydronic heating, connection to district heating is included.

Figure 1 shows the life cycle cost and the energy cost for the different systems in the multi family dwelling as a function of O_P. Lower O_P means reduced average internal heat load which increases the need for heating but if the airflow rate is reduced at the same time, this increase is dampened or turned to a decrease. Figure 2 gives the life cycle and energy costs for the detached house. Figure 3 shows the accumulated life cycle cost over time, which is the life cycle cost at a certain calculation period for both simulated buildings. It takes a little less than ten years to benefit from heat recovery and around 15 years to benefit from variable airflow rate.

Table 3. Output data from ProLive for the two simulated buildings. The costs and the LCC refers to the entire calculation period, 40 years. The areas in the denominators of the units are the floor area. O_P = 60% and α = 0.33.

Building	Multi family dwelling				Detached house			
Ventilation system	E		SEH		E		SEH	
Airflow rate	CAV	VAV	CAV	VAV	CAV	VAV	CAV	VAV
Air heating energy / (kWh/(year·m²))	0	0	3.8	3	0	0	2.8	2
Hydronic heating energy / (kWh/(year·m²))	66.8	52.2	29.7	27	69.2	57.3	42.4	40.1
Electrical fan energy / (kWh/(year·m²))	1.8	1.5	6.9	4.2	3.2	2.8	5.7	4.5
Total energy / (kWh/(year·m²))	69	54	41	34	72	60	51	47
Changed volume / (m³/(m²·year))	11038	8080	11038	8080	8623	6312	8623	6312
Initial cost, ventilation / SEK	160	190	281	311	74	84	148	159
Initial cost, hydronic heating / SEK	271	271	243	243	371	371	352	352
Initial cost, total / SEK	431	461	524	554	445	455	500	510
Cost, electricity / SEK	47	40	182	111	84	73	149	118
Cost, heat / SEK	1096	856	550	492	1136	941	743	690
Cost, maintenance / SEK	96	102	113	119	115	117	123	125
Cost, repair / SEK	27	76	79	128	23	40	86	103
Cost, space loss / SEK	108	108	278	278	25	25	84	84
Life cycle cost / SEK	1804	1643	1726	1682	1828	1650	1685	1630

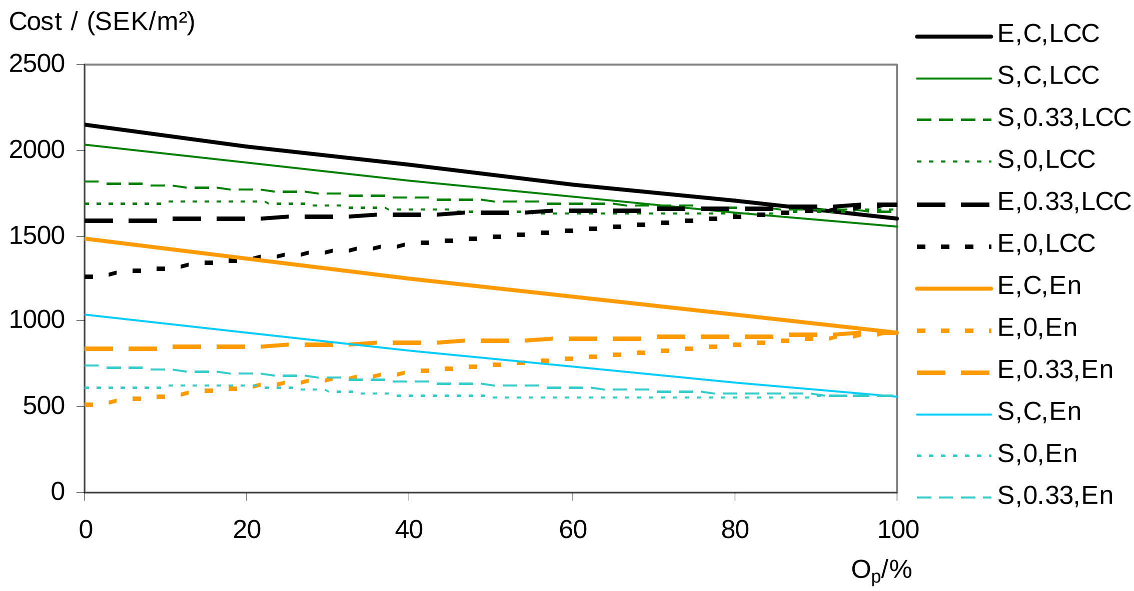

Figure 1. Life cycle cost (LCC) and energy cost (En) per floor area for the multi family dwelling. E means exhaust ventilation, S means supply and exhaust ventilation, C means constant airflow rate, 0 means α = 0 and variable airflow rate and 0.33 means α = 0.33.

The average U-value, which describes the heat transmittance through the building per area and temperature difference, and the building length, which affects the floor area, was varied for the detached house with the results shown in Figure 4. Bought energy would also be needed for household electricity and tap water which is not included in this simulation.

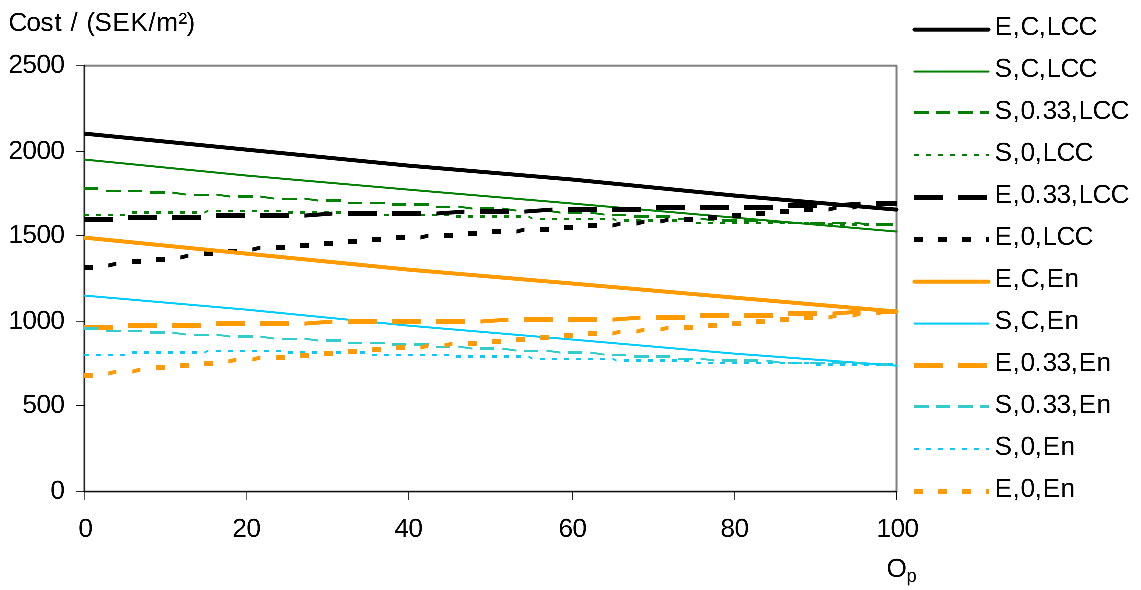

Figure 2. Life cycle cost (LCC) and energy cost (En) per floor area for the detached house. E means exhaust ventilation, S means supply and exhaust ventilation, C means constant airflow rate, 0 means $\alpha = 0$ and variable airflow rate and 0.33 means $\alpha = 0.33$.

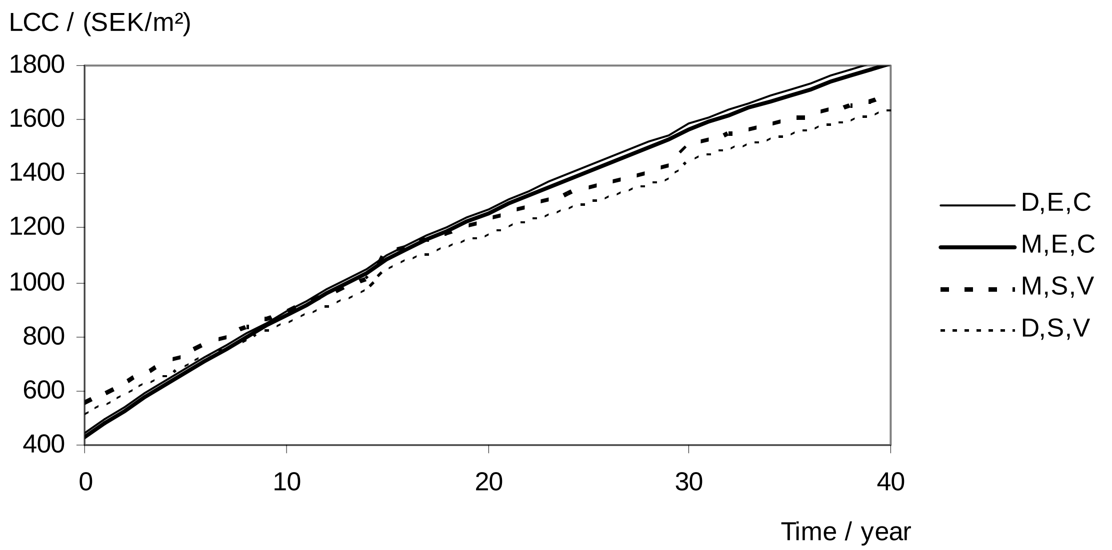

Figure 3. Accumulated life cycle cost (LCC) per floor area. E means exhaust ventilation, S means supply and exhaust ventilation, C means constant airflow rate, V means variable airflow rate with $\alpha = 0.33$, D means detached house and M means multi family dwelling. $O_P = 60\%$.

If it is assumed that the occupancy is either 100% or 0 in a certain apartment or house, a certain occupancy rate must correspond to a specific part of the day when the apartment or house is occupied. In Figure 5, it was assumed that the building was occupied during nights and empty during the days to test the assumption of using an average constant occupancy rate. This set up also gives the option to reduce the requirements on the indoor temperature when unoccupied. This possible temperature span decreases the energy use for this light weight detached house.

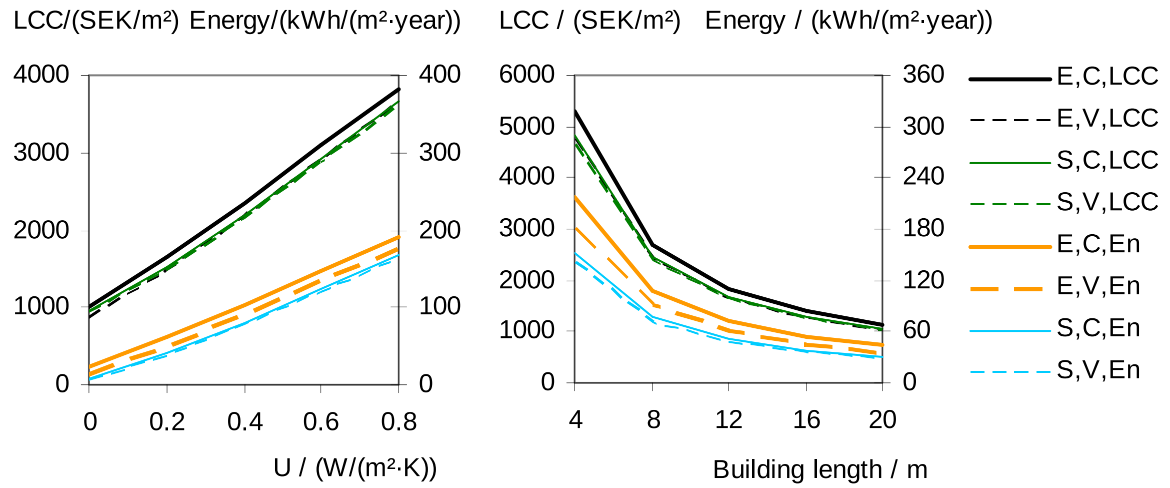

Figure 4. Life cycle cost (LCC) and annual energy use (En) for heating and fans for the simulated detached house when the U-value and building length were varied respectively. E means exhaust ventilation, S means supply and exhaust ventilation, C means constant airflow rate, V means variable airflow rate at α = 0.33.

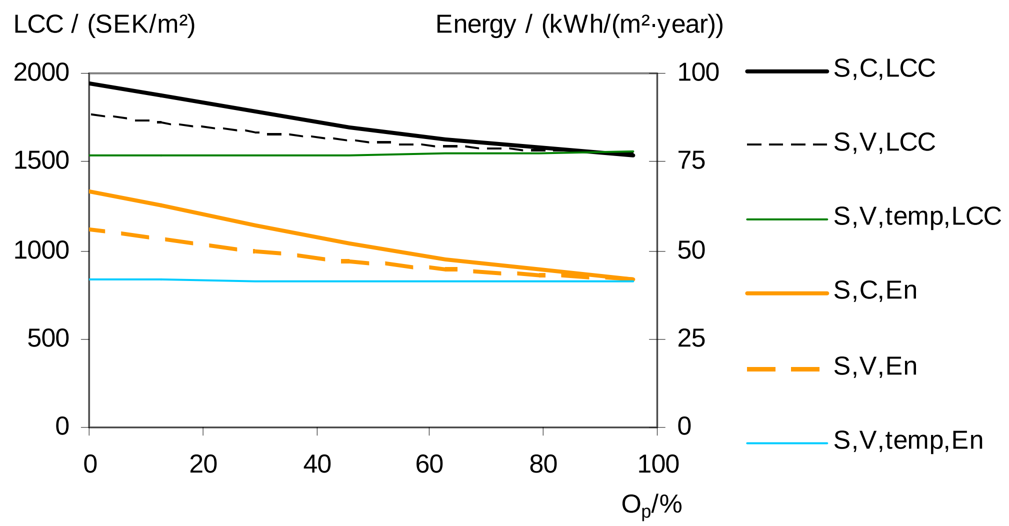

Figure 5. Life cycle cost (LCC) and annual energy use (En) for heating and fans for the simulated detached house as a function of O_P when the house was assumed to be occupied during nights. S means supply and exhaust ventilation, C means constant airflow rate and V means variable airflow rate at α = 0.33. temp means that the demands on temperature in that set up was changed during absence from 22°C to a span from 18°C to 26°C.

DISCUSSION

It has been shown that occupancy controlled variable airflow rate ventilation systems can be beneficial from a life cycle cost perspective in both multifamily dwellings and detached houses. The simulated occupancy detection cost was 766 SEK per apartment or house. This cost is reasonable for a manual system where the user chooses the number of people, for example at the main door. Right now, a lot of development is going on regarding cheaper sensors for detecting occupancy. There are indications that industry will come up with carbon dioxide sensors in this price range, which would be an alternative that is not dependant on the user to work properly. It would be possible to control each single room in each apartment or in the detached house instead of controlling the entire apartment or detached house but that would be more complicated and expensive. By doing that, air could be supplied where people are in the building which initiate a discussion of the form of the requirements on airflow rate. ProLive does not simulate that at this point of time but it could be a feature in the future.

It has also been shown that supply and exhaust ventilation with heat recovery can have a lower life cycle cost than exhaust ventilation systems, which has a lot of other disadvantages including draughts and poor control of the air movements as example [7].

Simplifications have been made. An example is the constant temperature efficiency of the heat recovery unit, which in reality would increase the benefit from lower airflow rate because of higher efficiencies at lower airflow rates. Another example is the constant occupancy rate over time. In fact, it varies both over the day, which does not seem to have much influence, and over the year which probably has more of an influence but such measured data is lacking.

With an occupancy detection system, it would be possible to control, for example, lighting or other use of house hold electricity. It is perhaps rather common that people switch off the lighting when they are not indoors but if it is assumed that 200 W per house could be controlled by the occupancy and O_P = 60%, 3.7 kWh/(m²·year) can be saved from the household electricity for the simulated detached house. On the other hand, the need for heating energy will increase since the internal heat load provides a certain amount of heat.

In this study, it has been assumed that the nominal airflow rate was set according to the Swedish building code. This airflow rate is based on the floor area and not on the occupancy rate, which should be the parameter controlling the airflow rate. It can be discussed whether or not the requirements in the building code are appropriate since the airflow rate per person will depend on the nominal floor area per person. From a moisture perspective, it can be argued that the indoor vapour content will increase if the airflow rate is reduced parts of the day. On the other hand, during winter time, the risk of having dry air will decrease with variable airflow rate. The occupancy detection could also be based on vapour production or a combination of occupancy parameters. More research is needed regarding the most apparent parameter in this study, the occupancy rate. Measurements will hopefully be performed in the future.

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Demand Controlled Ventilation (DCV) and Energy Savings : application on sites

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SUMMARY

DCV is one of the main issues in the near future due to the large potential of energy savings, the importance of maintaining a good IAQ with an adequate ventilation rate and the optimisation of system sizing.
Nevertheless the technology or sensor used must adequately follow the actual demand. The study shows on site results on two different technologies.
The first site is a collective dwelling installed with natural ventilation system situated in Nangis, near Paris (France) which has been renovated with hybrid ventilation and humidity controlled outlets and inlets. HR-VENT is an in-situ monitoring which exceptional character lies in the dimension as well as in the used measurement means. With more than 700 million data saved during two years out of 55 occupied dwellings, the experimentation made it possible to measure the effectiveness of a new ventilation system and contributed to enrich the knowledge on the operation of natural and hybrid ventilation in collective dwellings. From January 2004 to December 2005, the data of relative humidity, temperature, pressure and extracted airflow were saved every minute in each technical room, using specifically developed sensors. Directly related to the meteorological data, these measurements allowed to evaluate the performances of the humidity sensitive ventilation and the contribution of the mechanical assistance; they also determined the capacity of the hybrid system to improve indoor air quality and to control the thermal losses.
The second site is a meeting room of an office building which has been installed with a mechanical DCV system using optical (infra-red) sensors to determine the number of occupants. The results on site show the performance of the system and adequate determination of the real number of occupants, assessed by webcam, and the induced results in IAQ and energy.
Although DCV is common in residential ventilation, it is not yet frequently used in non residential area. We expect an increase of DCV systems in the future to fulfil the targets of energy regulations. The potential of savings is tremendous on the market and products and solution are more and more common

INTRODUCTION

DCV (Demand Controlled Ventilation) is one of the main issues in the near future due to the large potential of energy savings, the importance of maintaining a good IAQ with an adequate ventilation rate and the optimisation of system sizing.
In France, residential ventilation systems controlled with humidity are commonly used in new buildings and this DCV is the reference solution proposed by the French energy regulation in buildings updated in 2006. As humidity is an adequate indication of occupancy (and activities) in dwellings, humidity controlled systems have also been developed for natural ventilation and can be used in retrofitting. This will be the subject of the first site instrumented we present in this paper (figure 1). The object of HR-VENT project was to carry out a validation and an actualization of the knowledge on the operation of ventilation in general, and more particularly on the role of the humidity sensitive ventilation and its mechanical assistance (hybrid ventilation). The selected site is a gather of residential

buildings located at Nangis (France). The experimentation applied on 5 buildings (Figure 1) representing a total of 55 occupied dwellings, which were instrumented during two years, from January 2004 to December 2005.

Figure 1 : View of buildings of HR-VENT project.

In non residential building like offices, meeting rooms…, humidity is less adequate to represent the occupation of rooms. CO2 sensors are commonly used but an alternative is to use infra-red optical sensors, to determine not only the presence but the number of occupants. Optical DCV systems are commonly used in small and medium size rooms with average height as an alternative cheaper and equally efficient to CO2 sensors. Energy savings are given by Technical Agreements and are based on several on site studies of the systems. We present in this paper one of them.

Figure 2 : view of various occupations studied in the meeting room

FIRST SITE: HUMIDITY CONTROLLED VENTILATION IN RESIDENTIAL BUILDING

METHODS

In this ventilation system, the fresh air is provided by the humidity sensitive air intakes located above the windows in the main rooms (bedrooms and living room). The stale air is then evacuated in the sanitary rooms (toilets and bathroom) by humidity sensitive extract grilles, and in kitchen through the exhaust duct of the connected gas appliance. The ducts are served by a low pressure assistance fan whose operation is managed by the external temperature.In each building, each stack of dwellings was instrumented in order to save every minute the parameters for pressure, opening section of extract grilles, temperature and relative humidity in the technical rooms. In the kitchen, the operation of the connected gas appliances was measured by the temperature of the extracted burned components. Information was then sent and stored in a specific acquisition device.

RESULTS

Measurements carried out made it possible to assess the behavior of the humidity sensitive extract grilles (opening proportional to the relative humidity) located in the sanitary rooms.

Adaptation to the needs and statistical airflow

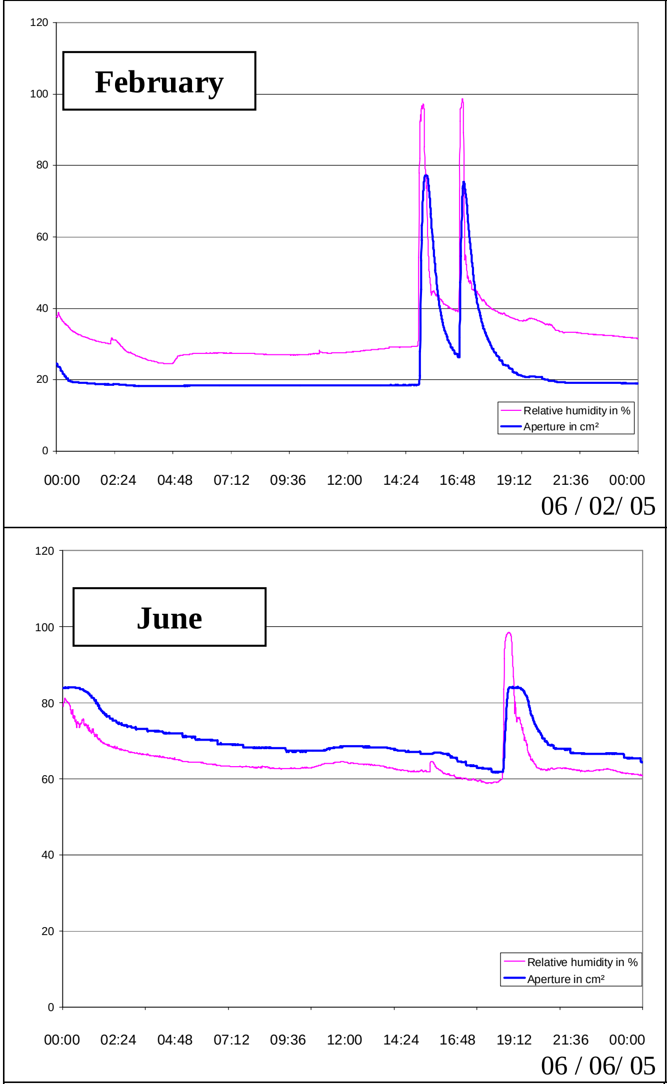


Figure 3 : Instantaneous view of the variation of opening (in cm², pink curve) and of the indoor relative humidity (in %, blue curve) of a humidity sensitive extract grille in bathroom, over 2 different climatic periods, on one day.

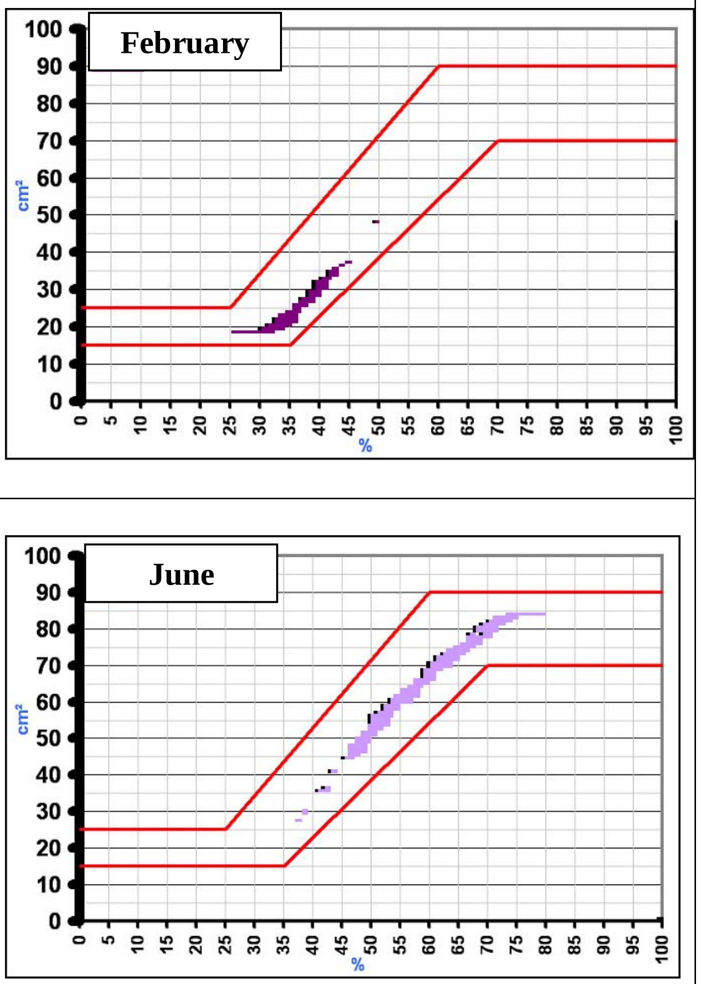


Figure 4 : Statistical view of opening (in cm²) according to the indoor relative humidity (in %) of a humidity sensitive extract grille in bathroom, over 2 different climatic periods, on one month*.

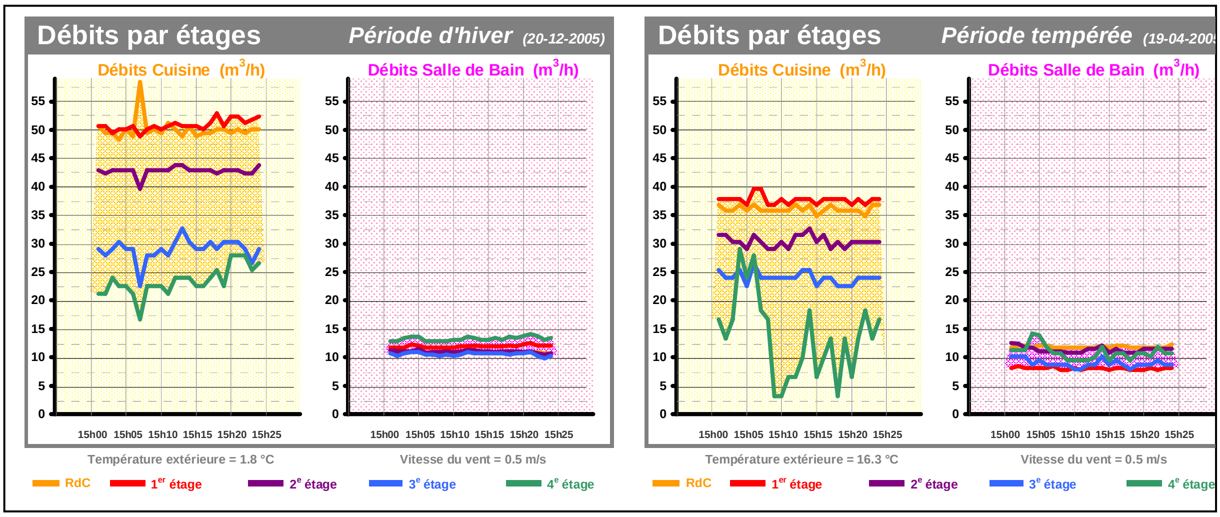
** This statistical representation corresponds to the cases without productions of humidity inside the room. Note: the red curves represent the tolerances of the product.*

As one can observe it on the graphs Figure 4, we can note that **<u>the average</u> opening of the humidity sensitive grilles depends on the season:** it is low in cold season and shifts towards a wider opening as the season becomes warmer, reflecting the seasonal evolution of the external absolute humidity along the year (low in winter, high in summer in Europe).

If the average airflow depends on the season, the **<u>instantaneous airflow</u>, for itself, follows the specific variations of relative humidity** in the room, as the graphs Figure 3 show it, and this occurs whatever the season. A shower, in summer as well as in winter, causes a fast rise in the interior relative humidity which will increase the opening of the grille during a few minutes, the time needed to evacuate the excess of humidity.

Balancing the airflow between the floors

Graphics Figure 5 present, for two different periods of the year, instantaneous airflow (each minute) in <u>natural ventilation</u> (without assistance nor humidity production) on nearly half an hour, in kitchens (fixed grilles) and bathrooms (humidity sensitive grilles), for five floors located on the same stack. The graphs from right-hand side show the **winter** measurements, those of left represent "**mid-season**".



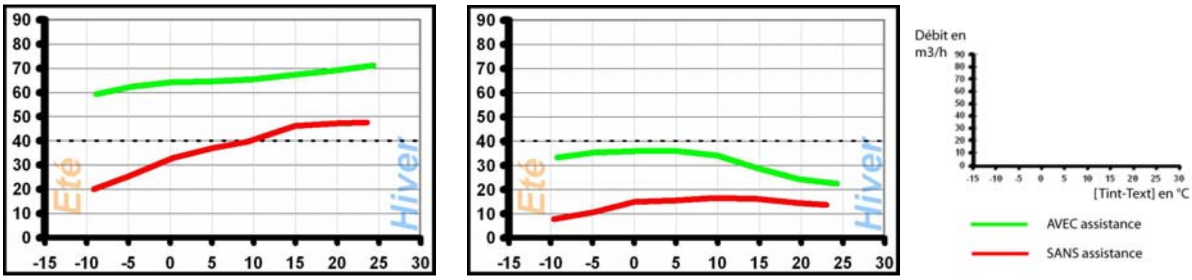
Fixed grille Humidity sensitive grille Fixed grille Humidity sensitive grille

Figure 5 : Airflow per floor - comparison between fixed grilles / humidity sensitive grilles

By balancing the airflows of the various floors at a stable and low average level without humidity production, humidity sensitive ventilation not only limits thermal losses; it also offers at each floor a great potential to answer to an important specific need for ventilation, as the common duct is not overloaded unnecessarily by the airflow of the other floors.

Contribution of the mechanical assistance

The results show **that the low pressure mechanical assistance succeeds in erasing any risk of reverse airflow, which is particularly visible in summer when the risk is the most important**. The mechanical assistance also showed a real capacity **to control the effect of thermal draught**, as one can note it on Figure 6. On the green curve (operation of the assistance fan), we can note an increase in the average flow of 30 m3/h for the fixed grille, 20 m3/h for the humidity sensitive grille which confirms the potential of the mechanical assistance for reducing the variation all the year long. The contribution of the mechanical assistance is actually stronger in summer than in winter, when its action is more needed.



Fixe grille **Humidity sensitive grille**

Figure 6 : Statistical evolution of the average airflow measured according to the temperature difference [Tint-Text] in all the dwellings.

Maximum and peak airflow

Figure 7 represents the airflow ranges (minimum-maximum) as well as the average airflow (red) measured for each day of the year on a humidity sensitive extract grille.

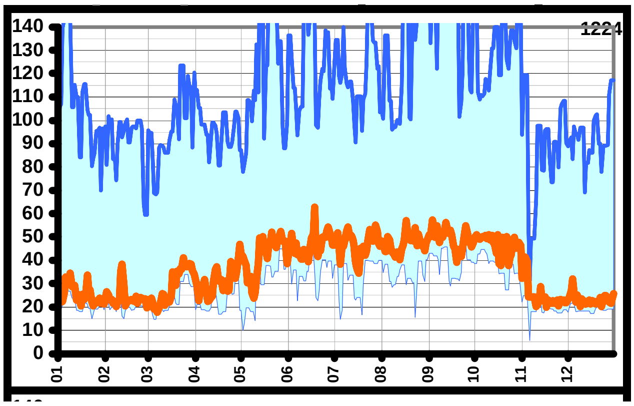
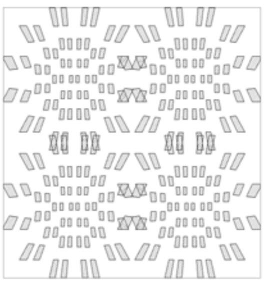


Figure 7: Daily airflow (range and average) measured in a bathroom over the year 2005. X-axis: time in days, Y - axis: airflow in m3/h.

We can note the great capacity of the humidity sensitive extract grille to modulate the airflow since a typical range of 90 m3/h is observed every day. But because of the punctuality of maximum airflow periods (corresponding mainly to the periods of shower or bath), we can observe that the average airflow is very close to the daily minimum airflow. That way, the punctual great airflow periods ensure the IAQ without having negative impact on the energy consumption.

SECOND SITE: OPTICAL DCV IN MEETING ROOMS



METHODS

Several meeting rooms have been installed with webcams to control occupancy and a monitoring of CO_2 measurements, airflows and sensor detection have been made during more than 200 days. Synthesis of measurements on one of these rooms is presented there. This room was 8,7m x 4,7m. The sensors due to the number of faces on the lenses have a projection zone and detects any movement crossing these zones

Figure 8 : projection zones on a 8mx8m zone

RESULTS

Maximum CO2 levels measured in the rooms where between 500 ppm and 1200 ppm while outdoor level was 500 ppm. These measurements are consistent with the regulation airflows used in France which has been proposed to maintain 1000 ppm ± 300 ppm for 300 ppm outdoors. Figure 9 shows an example of measurements in the room for a large occupancy.

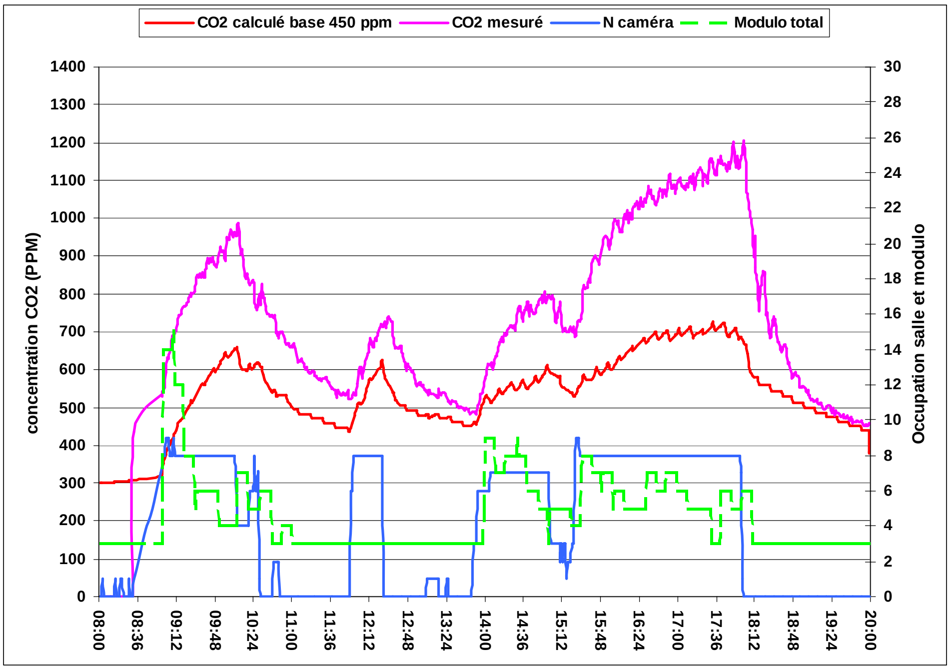


Figure 9 : Example of CO2 concentrations vs occupation
(red – calculated CO2, pink – measured CO2, blue – number of occupants (webcam))

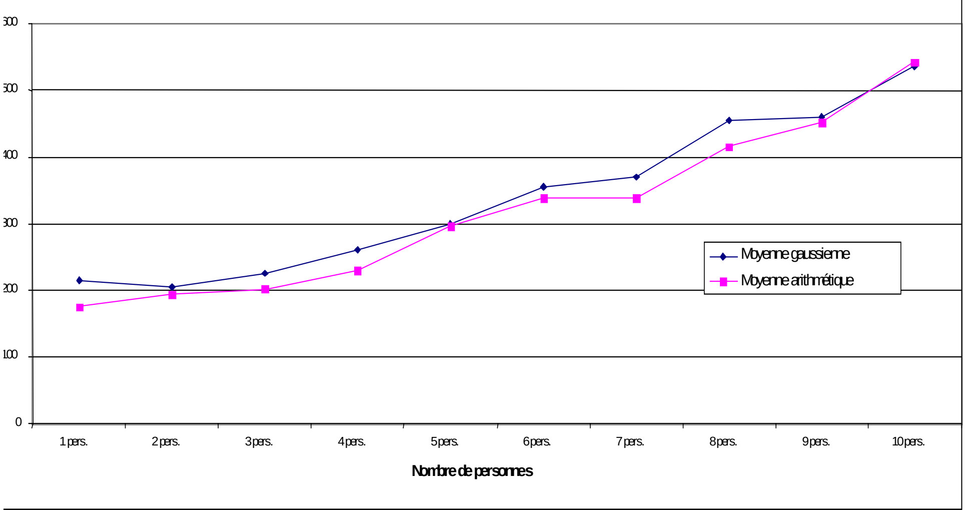


Figure 10 : number of detection vs number of occupants

In conclusion of various types of occupation, the number of occupants is correctly detected as shown in figure 10. Although it was requested to use the 3 rooms as much as possible, it was only used during 30 to 50% of time by in average 50% of the maximum occupation. This leads to an overall occupation rate (time x number) from 10 to 25%. This rate represents the potential of energy savings on the room, only based on heat losses during winter. The average payback time is less than 2 years and could be decreased to 1 year if the building were also air conditioned during summer. These values are coherent with a large study on 13 meeting rooms and 27 offices, realized by CETIAT, which led to 8% occupation rate in meeting rooms and 40% in offices. This study highlighted the huge amount of potential savings by implementing DCV.

CONCLUSION AND DISCUSSION

The measurements carried out within the framework of the HR-VENT project confirmed the performances of the humidity sensitive ventilation system in its capacity to improve indoor air quality, to erase the condensation risks and to limit the thermal losses. Its stabilizing role has been highlighted: it attenuates airflow imbalances between the floors, and limits the airflow variations over the year by offering a real control of the "natural motors" (wind and stack effect). Associated with humidity sensitive ventilation, the low pressure assistance fan optimizes the exploitation of the natural motors: the average airflow is limited in winter, contributing to the energy control, and positive airflow are assured all the year long, even in hot season. A complementary study made it possible to check that the system succeeds in removing the risks of condensation in all the main rooms, confirming the essential role of hybrid humidity sensitive ventilation system to ensure a good quality of air renewal. Even in refurbishment, the hybrid humidity sensitive ventilation system provides comparable airflow to those required by the regulation in force in the new building in France. A back quality control after 2 years of operation allowed also to check the performances of all tested humidity sensitive grilles, which confirms the reliability of this technology in time. The observations show the non-simultaneity and the time-distribution of the needs for ventilation, which confirms the relevance and the need for integrating this fact in calculations for ventilation networks dimensioning.

Measurements made on the meeting rooms showed that optical sensors can be used for detecting the number of occupants and offer a good detection for average height room. They have to been equally distributed in the room, each sensor covering a 4mx4m zone. These systems present a very good energy performance and payback time (<2 years) for small and average size rooms. For very large rooms, using only one CO2 sensor may become more profitable though. DCV systems had shown in previous study very good energy performance with a potential of savings of 92% for meeting rooms and 60% for offices.

Although DCV is common in residential ventilation, it is not yet frequently used in non residential area. We expect an increase of DCV systems in the future to fulfil the targets of energy regulations. The potential of savings is tremendous on the market and products and solution are more and more common

ACKNOWLEDGEMENTS
We thank CSTB and GAZ DE FRANCE for their collaboration in HR-VENT project; CETIAT and ALDES for their collaboration in optical DCV projects. We acknowledge and thank ADEME for the financial support of these studies

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Humidistat-controlled Heating and Ventilation Systems to Create Preservation Conditions in Historic Buildings in the Dutch Climate

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SUMMARY

In a marine temperate climate historic buildings that are equipped with thermostat-controlled heating systems show very low relative humidities (RH) during the heating season. This may lead to mechanical damage due to drying of hygroscopic materials like e.g. wood. Humidistat-controlled systems are investigated using two different cases. Two top monuments of the Netherlands serve for the case study: Hunting Lodge St. Hubertus and Muiden Castle.
The effectiveness of a humidistat-controlled heating system is investigated, both by simulation and by experiment. When using humidistat-controlled heating, the heating system is mainly used to maintain basic temperature levels during the heating season. During the humid seasons the heating system may be used to lower RH. Simulations are performed using Matlab Simulink. In the experimental setup a humidistat-controlled room is compared with a thermostat-controlled room. Simulation results show that humidistat-controlled heating is a good method to provide preservation conditions in historic buildings. Simulation results are validated by measurements. Though energy expenditure is significant lower, thermal comfort may decline.
A humidistat-controlled ventilation system is investigated by making use of simulation. In historic buildings with high visitor numbers, high moisture and CO_2-levels may occur. By selective ventilating with outdoor air, excessive moisture can be removed and low CO_2-levels maintained. Preservation effects mainly occur by eliminating high RH by ventilating with air that has a lower specific humidity or by slightly heating the ventilation air. To maintain healthy environment conditions, control of the system on CO_2-concentration is a priority.

INTRODUCTION

Originally, historic buildings did not have any other heating system than an open fire or some kind of local heating system. Sometimes a central heating system was installed afterwards. Measurements in one of the most valuable historic buildings in the Netherlands prove again that heating during the cold period leads to low indoor RH, causing damage to interior and objects [1]. Outside the heating season high RH often occurs, also causing risk for damage to interior and objects by e.g. mould growth [2]. In most cases the possibilities to fully control RH in a historic building, by e.g. installing a full air-conditioning system, is limited. Installing mechanical systems and ducts always will cause damage to the building and its historic authenticity. The high installation, maintenance and running costs are not even mentioned. Furthermore humidifying devices may lead to dramatic indoor air conditions with high surface humidities and condensation effects on cold indoor surfaces of the exterior walls, single glazing and roofs, or even condensation in the inner parts of the construction [3].

Theory

The principle of humidistat-controlled heating is controlling the heating system using a humidistat device [4]. High RH is prevented by starting heating. Reaching low RH during the cold season is prevented by limiting heating to maintain a certain lower temperature setpoint. The flowchart of the system is given in Figure 1a. The use of this control however is restricted. In summer it may be necessary to start heating and during wintertime it may be necessary to limit heating, causing thermal discomfort of occupants.

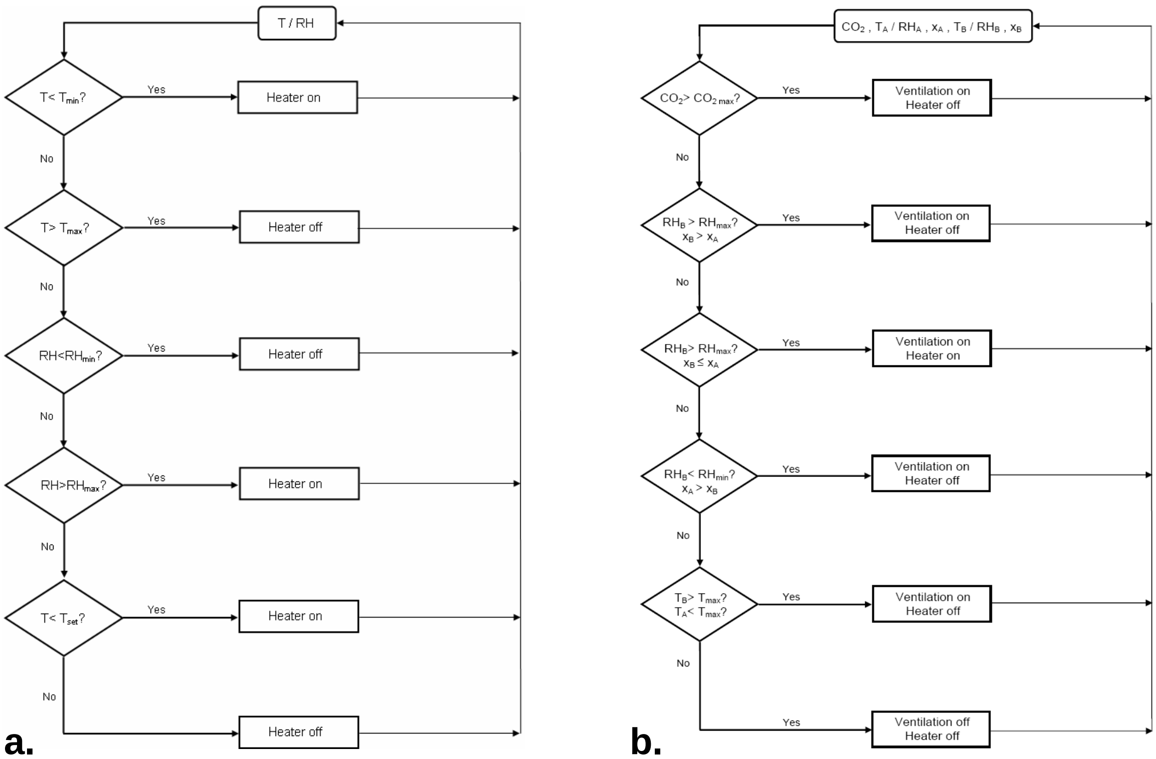


Figure 1. Flowcharts of the humidistat-controlled heating system for the case St. Hubertus (a) and the humidistat-controlled ventilation system for the case Muiden Castle (b).

The humidistat-controlled ventilation system is also being controlled by humidistat devices. Controlling RH is possible by ventilating with air that has a lower specific humidity or by slightly heating the inlet air. Humidistats and temperature sensors are installed in the location where ventilation air is extracted from, e.g. outside or an attic and in the location the where air is being supplied to (visitor zone). The specific humidity of both zones is calculated. Values are constantly monitored. To maintain a healthy and comfortable visitor zone, the CO_2 level is measured in the room the air is being supplied to. The objective of the ventilation system is to avoid high CO_2-levels and to control RH and temperature in the rooms. Ventilation only takes place if correction on CO_2 level, RH or temperature is needed. The flowchart of the system is given in Figure 1b.

Objectives

In the Netherlands there is little experience with humidistat-controlled heating and ventilation systems. The main objective of this research was to determine the suitability of humidistat-controlled systems in a maritime temperate climate like the Netherlands and to optimize the control strategy. Countries with a marine temperate climate have a cool winter, warm summer

and an uniform precipitation distribution, *Cfb* according to the Köppen climate classification system [5].
Second objective in the research was to construct heat and moisture simulation models in order to predict the suitability of humidistat-controlled systems in historic buildings.
Third objective is to gain insight on the (dis)advantages of these systems compared to basic heating systems or the use of full air-conditioning systems.

Cases

The effectiveness of humidistat-controlled systems is investigated using two cases. The first case describes the research of a humidistat-controlled heating system in Hunting Lodge St. Hubertus. The second case describes the research of the implementation of a humidistat-controlled ventilation system in Muiden Castle.

Case 1: humidistat-controlled heating
Hunting Lodge St. Hubertus was built from 1916 to 1922. Building and its interior were designed by the famous architect Berlage according to the legend of Saint Hubert. The building has the shape of the antlers of a deer's head. A large part at the ground floor level is now part of a guided tour. A part of the west wing is housed by the property manager. The building also serves as a location for weddings, receptions and occasionally, as housing for high placed guests. The annual visitor number in 2005 was 27.000.

Figure 2. An aerial view of Hunting Lodge St. Hubertus (left) and the dining room with its historic interior (right).

The exterior walls, floors and roof of the building are not thermally insulated. Exterior walls at ground floor level consist of masonry cavity walls. Floors are made of concrete, carried on metal beams. The windows consist of single glazing in steel frames. The roof is constructed out of wooden beams with slate covered sheathing.

Case 2: humidistat-controlled ventilation
Muiden Castle is a century old building that has played an important role in Dutch history. The castle dates back to the year 1283 and is one of the best conserved medieval castles in the Netherlands. In 1900 and 1960 the castle was renovated and since 1960 it is in use as a museum. Except from the museum function, the castle is also being used for activities like diners, parties and weddings. The annual visitor number in 2005 was 115.000.
Exterior walls are constructed out of masonry with a thickness ranging from 200 to 1600 mm. Indoor walls also consist of masonry and vary in thickness from 300 to 1300 mm. In most rooms indoor walls are decorated with plaster. The windows are all single glazed, mostly stained glass windows. The glazing is set in wooden or stone frames. Different windows of the castle can be blinded with wooden shutters. The floors consist of wooden planking on

wooden beams. Floors are mainly covered with tiles or marble. The roof is constructed out of a wooden sheathing covered with slates.

Figure 3. Muiden Castle as seen from the south side (left). The collection consists partly of medieval weaponry (right).

Apart from the historic interior, the castle houses an important collection consisting of paintings, furniture and artifacts like medieval weaponry. To optimize the visitor experience, an extensive renovation started in 2005. During times with high visitor numbers, the indoor air quality was poor [8]. In different locations throughout the castle mould growth was found on objects [9]. The planned renovation was a good opportunity to optimize indoor climate conditions by installing a humidistat-controlled ventilation system. Optimizing indoor conditions by making use of a humidistat-controlled ventilation system is not commonly used in the Netherlands. In this case, the effectiveness of the system will be investigated by making use of computer modeling.

METHODS

Case 1: humidistat-controlled heating

First, air temperature and RH measurements took place in different locations throughout the building for a period of one year. Next a simulation model for humidistat-controlled heating was developed to gain insight on the effect of the system on indoor climate, control strategies, needed heating capacities and optimal setpoints for the test set-up. The model was validated with measurements from the earlier monitoring. After the modeling an experimental set-up was installed in the monument. The experiment consisted of a humidistat-controlled room and a thermostat-controlled room. Testing started during the cold winter months and the tests were continued for a full annual cycle.

Modeling humidistat-controlled heating

Simulations on the indoor climate were performed using the heat and moisture model HAMBASE [5] coupled to Matlab Simulink [6]. The HAMBASE model contains the building model. The control strategy in the humidistat-controlled room is based on the flowchart as given by Figure 1a and modeled using Matlab Simulink [7]. First is checked if the room temperature is higher than the set minimum temperature T_{min}. If not so, the heater is switched on. Next is checked if the temperature is below the set maximum temperature. If not so the heater stays off regardless of RH conditions. If temperature is between the setpoints of minimum and maximum temperature, the controller continues to check if correction of RH is acquired by checking if the current RH is higher than the set maximum RH. If so, the controller switches the heater on until the RH is below RH_{max} or the temperature T_{max} is reached. In historic buildings where thermal comfort is needed, the possible provision of

limited thermal comfort by slightly expanding the controller is investigated. If RH is between RH_{min} and RH_{max}, heating is possible to raise indoor air temperature and increase thermal comfort. The heater will stay on until RH_{min} or the desired temperature T_{set} is reached.
The inputs of the HAMBASE-Simulink model are heat flow and moisture flow and the outputs are air temperature and relative humidity. Dependent on the input values is checked if heating is required according to the conditions as given in Figure 1. The output of the controller is zero or, if heating is required, the set heating capacity for this zone.
Setpoints of both types of controllers are given in Table 1. Settings of the thermostat-controlled room in the model and the experimental set-up are set to a constant temperature of 17°C to avoid fluctuations. Air exchange rate in the rooms was not measured and is set to an estimated value of 0.8 times per hour.

Table 1. Setpoints for the controller devices in the model.

Thermostat-controlled room		Humidistat-controlled room	
Start daytime	8 a.m.	T_{min}	10°C
Start nighttime	10 p.m.	T_{max}	25°C
T_{day}	17°C	T_{set}	17°C
T_{night}	17°C	RH_{min}	45%
		RH_{max}	55%

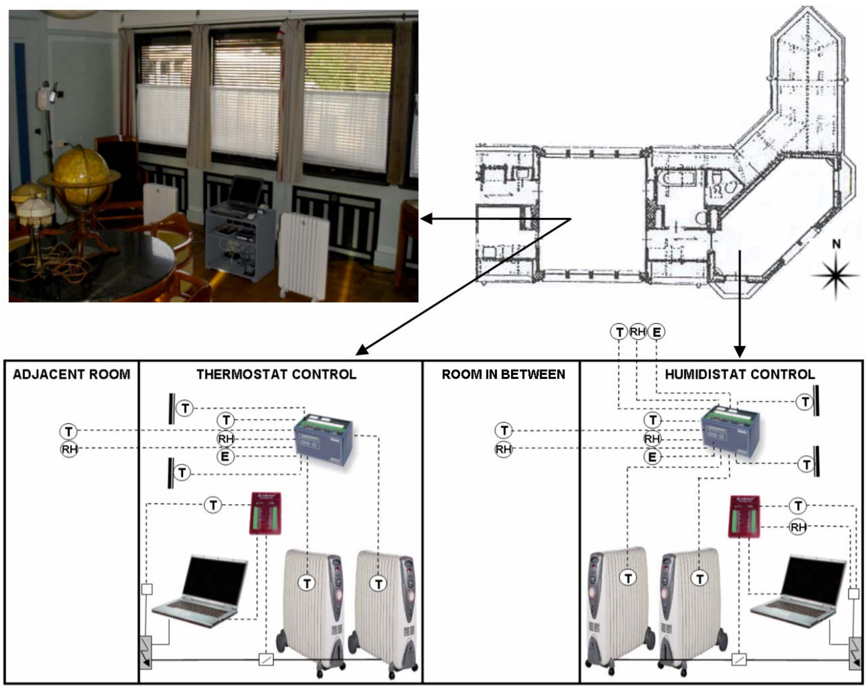


Figure 4. The upper left image shows the experimental set-up in the thermostat-controlled room. A floor plan of the two rooms where the set-up was installed in is given by the upper right image. A schematic representation of the configuration of the test set-up is given below.

Experimental set-up humidistat-controlled heating

For an experimental set-up two similar rooms in the historic building were selected on the first floor. During testing this part of the building was unused and doors and windows remained closed. There were no known moisture sources in this part of the building. Sun blinds were closed for about 60% of the window area during testing (Figure 4). The configuration used for the experimental set-up consisted in each room of a laptop computer for control, three oil filled electrical heaters of 3 kW each and a combined T/RH-sensor. The existing central heating system was switched off for these rooms. In one room the set-up was

installed to heat the room humidistat-controlled. The software was programmed according to the flowchart as shown in Figure 1. Every 10 seconds the software ran a loop with current air temperature and RH as input.
In another similar room the set-up was installed to heat the room thermostatically. Setpoints of both controllers were likewise as shown in Table 1. In the thermostat-controlled room day temperature is set to the same value as the night temperature. This is done to avoid deliberate fluctuations of RH in the historic interior and thereby limiting the risk of any damage done to the interior during the experiment. The electrical radiators were controlled by a simple on/off switch. Additional heat production was limited by using only a laptop computer per room to control the heaters. In the rooms under investigation indoor air temperature, surface temperature of north facing window and wall, RH and incoming solar radiation were monitored. In adjacent rooms air temperature and RH was measured. Outdoor climate properties like air temperature, RH and solar radiation were also monitored.

Case 2: humidistat-controlled ventilation

First, the indoor climate of the castle is modeled prior to the installation of the ventilation system. Indoor climate data unfortunately only are available for a short period of time, in the autumn of 2005. To obtain an indication of the indoor climate through the whole year, a model of the castle is created using HAMBASE. In this model only the basic heating system is integrated, like the situation prior to the installation of the ventilation system. Results are compared with the desired indoor climate conditions. The desired conditions are based on a healthy indoor air quality and conservation conditions for interior and objects (Table 2). RH boundaries are chosen to avoid e.g. mould growth and drying of organic objects. The maximum CO_2-level is based on the lowest class in the Dutch practice guidelines as given in NPR-CR 1752 [10].
Next, a model is set up to investigate the effectiveness of the ventilation system. This is done by making use of HAMBASE coupled to Matlab Simulink. In this case HAMBASE contains the building model and Matlab Simulink the model of the ventilation system and control. Obtained data are again compared with the desired indoor climate conditions. CO_2 levels are not modeled.

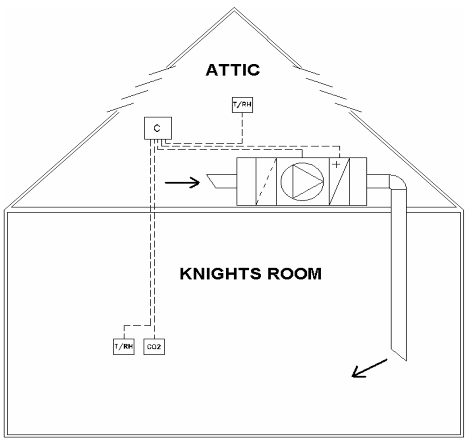


Figure 5. Principle of the humidistat-controlled ventilation system. The ventilation unit is controlled by the controller (C) which is fed with data from the sensors.

Table 2. Desired conditions for a healthy indoor climate and conservation conditions.

Relative Humidity	Maximal variation	Temperature [˚C]	Maximum CO_2-level [ppm]
40 – 65%	10	5 - 25	1520

The building is equipped with a central heating system with radiators and floor heating. During the renovation a ventilation system is installed in five important rooms, next to the existing heating system. Most of these rooms have a chimney. To minimize the impact on the construction, the existing chimneys are used to install the ducting. Each room has its own ventilation unit, which are installed on the attic. The ventilation units are composed of a ventilator, heater and filter section with EU7 quality. The ventilation system does not contain a humidifier or dehumidifier. Air is taken directly from the relatively air leak attic. Air is supplied in the underlying rooms through ducting in the existing chimneys and an inlet grille in the chimney. Inlet air leaves the room through exfiltration and opened doors. In this case the ventilation system of one of the five rooms, the Knights room, is lighted out. The ventilation unit for this room has a maximum ventilation capacity of 2200 m^3/h. This capacity is based on the amount of 35 m^3/h per person and is based on 62 persons. The ventilation capacity is controllable in different steps.

Modeling the indoor climate before the installation of the ventilation system

First a model of the situation prior to installing a ventilation system is created. In the simulation only the north part of the castle with five rooms (zones) are modeled. The Knights room is on the ground floor, with one underlying zone (cellar), one adjacent zone and three up lying zones. Non exterior or interzonal walls are modeled as adiabatic walls. The Knights room contains floor heating with a maximum capacity of 60 W/m^2 and a setpoint of 19˚C. The model is tuned using the available data as obtained in the autumn of 2005. Discrepancies between the results of model and measurements occur because the floor heating does not always have a constant temperature, which is assumed in the model. The air exchange rate of the rooms in the model is assumed to be constant 0.5 h^{-1} (Table 3).

Table 3. Settings for zone 2 (Knights room) of the models.

Before installation of ventilation system		After the installation of ventilation system	
Air exchange rate	0.5	Air exchange rate	At least 0.5
Start daytime	10 a.m.	T_{max}	25°C
Start nighttime	17 p.m.	RH_{min}	45%
T_{day}	19°C	RH_{max}	60%
T_{night}	19°C	T_{day}	14.5°C
		T_{night}	14.5°C

Modeling humidistat-controlled ventilation

Next, a model with the added ventilation system is developed. This is done by coupling the HAMBASE building model to Simulink. This model is constructed according to the flowchart as given in Figure 1b. First is checked if CO_2 levels are below the maximum desired level. Next is checked if correction on RH is needed to ventilate with possible drier air from the location air is being extracted from. Therefore both specific humidities are calculated and compared. If RH of the visitor zone is too high and the specific humidity of the extraction location is higher or equal to that of the visitor space, the ventilation air will be heated. If the measured RH is less than 5% out of the desired RH boundaries, the ventilation system will not work on full load. In this case the ventilator will work on 25% of its capacity to avoid high fluctuations in temperature and RH. If the CO_2 level and RH are within the desired boundaries, the temperature is checked. If the temperature in the visitor space is too high and the air in the extraction location has a lower temperature, ventilation starts.

Settings of the HAMBASE model are likewise as given in Table 3. In addition to the added ventilation system the setpoint of the floor heating system is lowered from 19.5˚C to 14.5˚C.

The air heater in the ventilation unit for the Knights room has a maximum heating capacity of 24 kW.

RESULTS

Case 1: humidistat-controlled heating

Figure 6 shows simulation results of temperature and RH zoomed in on January 14th to February 14th 2006 of the humidistat-controlled room in Hunting Lodge St. Hubertus. Simulation results are validated with measurements. Minor discrepancies occur possibly due to the estimated air exchange rate of 0.8. Visible is that with a T_{min} set to 10˚C it is not possible to maintain a minimum of 45% RH due to the low specific humidity of the outdoor air, which mostly occurs during wintertime (Figure 6: 22/01–04/02). Over the simulated period T_{min} has to be lowered to about 4˚C to maintain 45% RH in the Dutch climate.

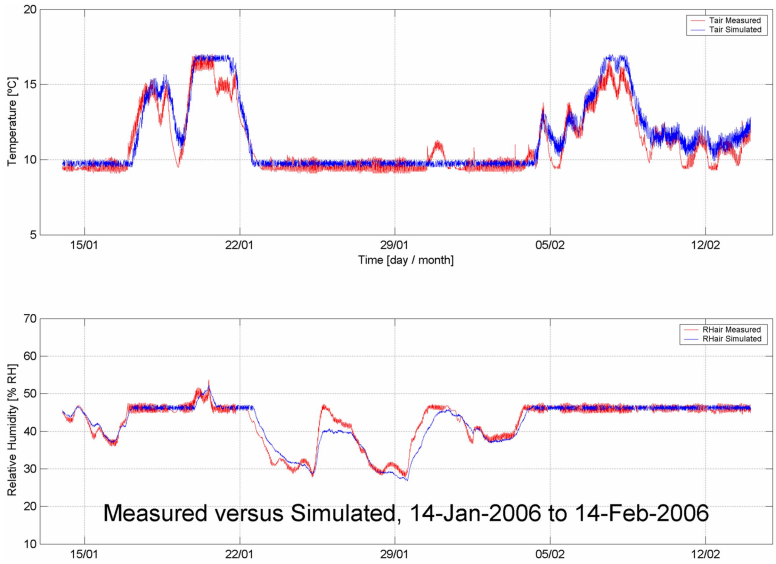


Figure 6. Simulation results of temperature and RH in the humidistatically heated room over the period from January 14th to February 14th 2006.

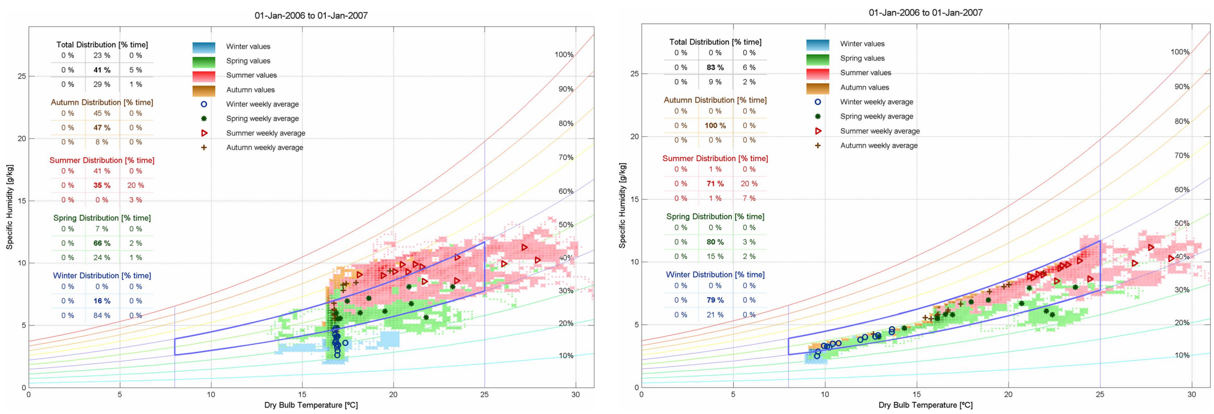


Figure 7. Psychrometric charts showing measurement results of the thermostat-controlled (left) and the humidistat-controlled (right) room from January 1st 2006 to January 1st 2007. In the thermostat-controlled room RH is for 30% of the data under 40% RH and for 23% of the data over 60% RH. RH in the humidistat-controlled room is for 11% of the data under 40% RH and for less than 1% over 60% RH.

In Figure 7 temperature and RH of the test set-up are plotted in a psychrometric chart both for a thermostat-controlled room as for the humidistat-controlled room.
In the thermostat-controlled room low RH occurred during periods of low specific humidity (winter time). In the same periods RH in the humidistat-controlled room is higher due to a lower indoor temperature.

In Table 4 annual energy expenditure of three different heating strategies is compared for one identical room. Values are obtained by HAMBASE simulation using the outdoor climate data of the year 2005. Results show that humidistat-controlled heating without limited comfort function uses about 30% less energy in comparison to a thermostat control.

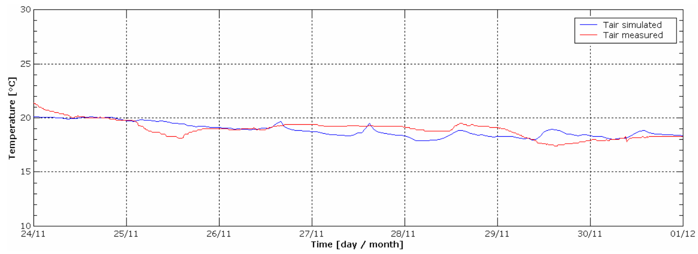
Table 4. Annual energy use of the identical zone using different heating strategies.

Heating strategy	Settings T/RH	Annual energy expenditure [kWh]
Thermostat-controlled[1]	15-20˚C / -	6133
Humidistat-controlled with limited comfort function[2]	10-25˚C / 45-55%	5431
Humidistat-controlled without limited comfort function	10-25˚C / 45-55%	4329

[1] day temperature 20˚C with a 5 K drop between 10 p.m. and 8 a.m.
[2] T_{set}= 17˚C

Case 2: humidistat-controlled ventilation

Indoor climate conditions are monitored from November 24th to December 1st 2005. In Figure 8 the measured results of the Knights room are compared to the simulation results over this period.



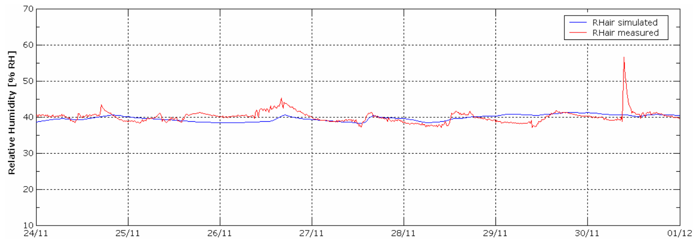


Figure 8. Results of the HAMBASE model compared with results obtained by measurement. The measured RH peak on the 30th of November is possibly caused by cleaning activities.

For the whole year, simulation results of the model prior to installation of the ventilation system show low RH during the heating season and high RH outside the heating season. In Figure 9 simulation results of the Knights room of the situation prior to installation and after the installation of the ventilation system are both plotted in the psychrometric chart. Results of the situation after installing the ventilation system show less low RH during the heating season. The main reason for this is a decrease in setpoint of the heating system from 19.5 to 14.5˚C. In the situation prior to installation 31% of the RH is below the desired minimum level of 40% RH. After the installation of the ventilation system this is 17%. Outside the heating season less high RH occurs thanks to the humidistat-controlled ventilation. In the situation prior to installation 28% of the RH is above the desired maximum level of 65% RH. After the installation of the ventilation system this is only 2%.

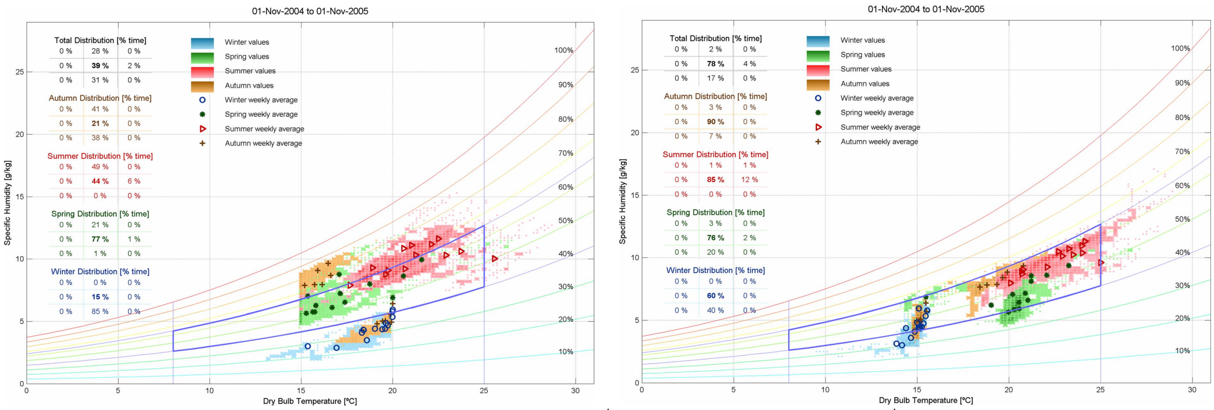


Figure 9. Simulation results from November 1st 2004 to November 1st 2005 before and after the installation of the ventilation system. Results show less low RH due to a decrease of the temperature setpoint and less high RH outside the heating season due to humidistat-controlled ventilation.

DISCUSSION

Humidistat-controlled heating is an efficient technique to create preservation conditions in historic buildings in the Dutch climate. The largest benefit is elimination of extremes in indoor RH. Temperature and RH fluctuations also are lower compared to thermostat-control with a night setback. Apart from providing improved conservation conditions energy expenditure is far lower compared to conventional heating to provide thermal comfort. Improved comfort can be provided by limited heating when RH is between desired boundaries, but this strongly depends on the specific humidity (g/kg) of the outdoor air. Historic buildings that are closed for winter season are ideal to implement a humidistat-controlled heating system. For humidistat-controlled heating T_{min} can be set to a lower value of about 4˚C to obtain a lower RH limit of around 45% in the Dutch climate. When comfort is required during specific times, the control strategy can be switched to a thermostat control in order to provide thermal comfort. In this case it is important to use a limiter to prevent too quick heating of the room. The use of a limiter in the control is also useful for situations that the installation restarts after e.g. a malfunction. The experimental set-up showed a side effect when having a room with humidistat control next to a room with thermostat control. This resulted in a wooden door that bend due to the difference in temperature and related RH. It is recommended to reduce these differences. Also literature shows heating may run out of control in rooms with a small air exchange rate and a high amount of hygroscopic materials, due to the release of moisture [11].

Historic buildings with high visitor numbers can be equipped with a ventilation system controlled by a humidistat device and a CO_2-sensor to improve indoor air quality and preservation conditions. Simulation results show that the implementation of a humidistat-controlled ventilation system can create more suitable conservation conditions in historic buildings. Indoor climate improvements consist of less low RH during the heating season due to a lower temperature setpoint. Outside the heating season high RH is corrected by slightly heating the ventilation air. Additional advantages of this system are the possibilities to filter the air and maintain healthy indoor air conditions by controlling CO_2-levels. Disadvantages

are the necessary ducting and a lower thermal comfort during the heating season due to a decrease of the temperature setpoint.
It is possible to maintain preservation conditions in historic buildings in the Netherlands without the use of (de)humidification. Values of the controller have to be selected for minimum use of the system, to be energy efficient and promote longevity of the components. Setting can differ per project and are dependent on both building physics and collection. If no comfort is needed the lower temperature setting has to be determined by assessing temperature sensitivity of collection or the presence of water filled pipework. Modeling is a useful tool to predict the impact of humidistat-controlled systems on the indoor climate, to determine optimal controller settings and to gain insight on energy expenditure.

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Home automation rather for safe or ease living than for control

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SUMMARY

The feedback of the smart home control from both young families and senior citizens studied in the two EU 5th FP projects: the Demulog and the Elderathome gave proof of the possibilities to improve the environmental quality by smart house technology. Respondents from the English, the Finnish and the French families applying for social housing as well as the Finnish and the Spanish elderly, a hundred from each group in each country have been personally interviewed.

Home automation guarantees amenities for easy living and comfortable indoor environment. At the same time, it should be energy efficient and facilitate the prevention of energy supply peaks. In order to be accepted, the new control systems should improve control over the indoor environment − not reduce control opportunities − and bring significant benefits for example in improved comfort, safety or saved energy. The comparison of the result to two previous Finnish end-user studies and some literature studies confirmed the results.

THE END-USER EXPECTATIONS ON SMART TECHNOLOGIES

Smart housing is expected to easy every day routines at all homes. It is aimed to assist people with any limitations, in independent living. The social policies of the European governments coincide as the societies face growing numbers of old people whom smart housing could help to stay in their own homes as long as they wish. Many solutions for smart housing, independent living or gerontechnology[1] have been tailored to fulfill all these expectations.

Information and communication technology, ICT, born in the world of academic freedom has been adopted by business for an even more effective handling and transfer of information. Not only time, but also travelling, energy and natural resources can be saved by the use of ICT. In addition, construction industry takes its share of the information society by applying ICT for housing in the first place by integrated technologies aiming to:

- *Comfort and functionality* by amenities satisfying the residents with high indoor air quality by efficient energy use thanks to building automation, and with entertainment or edutainment[2] via multimedia, with smart appliances for easy every day domestic tasks by home control[3] and inter-house communications systems as well as automated facilities
- *Safety and security* by enabling smart home functions preventing unfavourable events and unauthorised use of property and other resources

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[1] Gerontechnology is used for the technology targeted to senior citizens. Gerontology is the science of ageing.

[2] Edutainment is a term formed from words education and entertainment meaning information and communication services produced by end-user friendly interactive technologies

[3] Home control includes control and regulation of building service equipment. Such other control technology at home are automated doors, windows or curtains (called also as active structures), control of sophisticated home electronics and home appliance technology, gerontechnology, e-welfare, etc.

- *Controllability enabling sustainability and reliability* of housing technology in such a way that residents are aware of the controlling and regulation functions of the house − automatic and manual − and to interfere if needed
- *Equality* by mitigating difficulties to promote transparent information services and open up possibilities to learn ICT, and by removing obstacles from the physical environment to promote independent living by means of Design-for-All[4].

Latest housing production already incorporates many products with electrical engineering or digital information technology. Active operating systems are becoming as essential as other building services. New ideas and pioneering experiments concern intelligent construction materials and products. In old building stock, special expertise on the building methods and costs make the planning of new routes, refurbishment of networks and new installations possible. The information society in terms of the use of ICT is here to stay and so are media homes[5] relaying on implementation of ICT technology. Sophisticated applications of smart housing, such as active structures, flexible spaces and adaptable technology, develop slowly within conservative housing industry.

Modern smart house research finds solutions for everyday activities such as automated food preparing [1]. Problems of the integrated home, the sensor technology and integration of network protocols and automated systems together have been in focus. One idea is to find technology capable of identifying the presence of the occupant in order to help her or him in activities when she or he is in need of help. The integrated house features cover home automation, intelligent building services and active structures and integrate them as a functional combination, which makes everyday living easy for all, including the elderly.

Futures studies has identified that modern families in the industrialised world are in need of spare time and services. They lack easy everyday living conditions. The working life is demanding and people keep on working when shopping in self-service markets, cooking and laundering at home despite such conveniences as modern home appliances and prefabricated food products. They take care of their homes and garden, or have their car washed, escort their children to hobbies, keep up with information society and stay awake late for getting entertained or for having time of their own. Home has approached from two apart angles, focusing either on domestication of daily activities described above or on construction business of quality requirements of housing projects − based on provision of management, engineering, planning, estimating, tendering, ordering, etc. carried out during the whole building life-cycle. The same object, home is approached from the basis of both the technology push and the market pull concepts [2], [3], [4], [5].

Technology and our way of life are closely tied together in contemporary society and they mutually influence each other. How well-founded are the expectations on the technology and its ability to facilitate the everyday life of people in general? To understand the unsaturated smart housing markets the construction industry needs evidence-based end-user feedback of how to satisfy housing needs and wishes of their customers groups. Feedback on the use of smart home applications enables the satisfaction of the housing needs of the knowledge society where work and spare time are ever more fragmented and a holistic living

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[4] Design-for-All principle means design strategies and methods, which promote usability, accessibility and attainability of environments, products and services.

[5] Home control, building automation and the use of ICT are leading applications for this type of smart house. They are applications related to automated buildings, where building logic, one of the forms of building intelligence, is the key attribute of the design. Such terms as responsive or media home describe these houses better than smart house (about the definition of the building intelligence ple. see [2]).

accumulates short intervals of work and other activities, enjoyment or rest. The latest trends in mastering ones own life and having individual life styles add the variations to the customer needs. Constant changes in working and society add the need to know of activities at home now and in the future. Service level of the community and attractiveness of the housing promote the business life while skilful people are willing to move and to stay in such a municipality and work for local companies [6]. More sophisticated smart homes applications − applying several forms of building intelligence other than building logic[6] have developed quite slowly. They make the house smart in a wider sense with ubiquitous and wearable products or adaptive and learning[7] ambient technology, with intelligent construction materials[8] or flexible spaces and active structures such as automated doors and windows. Ambient intelligence challenges integration of more sectors to smart housing such as smart textiles and clothing, intelligent transport, intelligent work place, smart fire fighting facilities, etc.

METHODS

Firstly, a summary of the HVAC technology related results of the end-user requirements of the smart home control was made by an international comparison between the various customer groups based on feedback from two previous studies (Table 3, Table 4) and a thorough literature review. The results have been completed by the responses from both young families and senior citizens studied in the two EU 5th FP projects. Within the Demulog project (Living Conditions and Preferences of Families in Housing Need, Table 1) the numerical analyses were carried out (using the SPPS program). The qualitative responses to the open ended questions from the Elderathome project (The Prerequisites of the Elderly for living at home: Criteria for Dwellings, Surroundings and Facilities, QLK6-CT-200-00405, Table 2) have been grouped, listed and summed up. Results from two corresponding Finnish studies were used as reference values (Table 3). All samples in each country were dominated by residents who dwell in blocks of flats and in most cases in cities as well as their demography represented quite well the national averages cf. [10].

Table 1. Summary of the cases of the Demulog project (n=294) [7].[1]

Year	*Location*	*Target groups*	*Sample*
1. Autumn 2000 2. Spring 2000 3. Winter 2000-2001	1. Finland, Helsinki 2. France, Vienna 3. the United Kingdom, Peterborough	1. The subjects from 500 applicants for social housing provided by VVO[2]. 2. The subjects from some 1,500 households applying for social housing from OPAC Vienna[3]. 3. The subjects from the applicant lists of the Nene Housing Society Ltd in the Peterborough City[3] and the Peterborough City Council Housing Department.	1. 103 2. 101 3. 90

[1] Personal face-to-face interviews, the same questionnaire for the interviews used in each country; [2]VVO Group was a private social housing provider and an owner of 35,000 properties in 72 municipalities; [3]a private social housing provider.

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6 The five forms of Building Intelligence (BI) have been derived from the seven forms of human intelligence by Howard Gardner (1983). They are connectivity (speech synthesising and speech recognition, pattern recognition and applications, artificial intelligence, user-interface, use of background music and other stimulators of senses, etc.); self-recognition (building knows the state of it and the occupants), spatiality or ambient (a more conscious understanding of the spatial expression of the architecture, structures, interior design), building kinaesthetic or dynamics (a sense of change, active and moveable structures, furniture and equipment, adjustable and learning technology), and building logic (systems integration, embedded sensors to monitor the occupants' daily activities, combinativity).

7 The product learns from the user behaviour. The control algorithm of the product is based on neural networking combined which is touch by the user behaviour during the use of the product. Often the algorithm benefits also fussy logic.

8 Such materials for covering or windows, which are easy to clean, soil repellent or self-cleaning (without any end-user involment) or energy efficient preventing heat go out or over heat come in or making the natural lighting favourable without any disturbing reflectections, etc.; materials which transform according to the circumstances making possible (regulation) mechanisms operated without any external power sources, etc.

Secondly, to put the in detail reported HVAC technology related knowledge into the wider context of the end-user responses of the smart housing technology and use of the ICT at home a summary of the involving studies is given firstly.

Table 2. Summary of the Elderathome study from Finland and Spain (n=206) [8].[1]

Year	*Location*	*Selected case groups*	*Sample*
Spring and summer 2003	Finland Helsinki	Subjects aged over 55 years from three cases: - 32 members of an association of active seniors[2], - 28 tenants of two housing real estate companies in Puotila housing area[3] and - 46 randomly accessed elderly living in Vuosaari housing area[4] from the data bases of the Population Register Centre.	106
Autumn 2002	Spain Barcelona and neighbouring towns	Subjects in the age of over 55 years from 15 cases living in regular housing, in senior housing, in private and municipal service homes for elderly.	100

[1] Personal face-to-face interviews, the same questionnaire for the interviews used in each country; [2] A group of seniors, who plan a senior house of their own ; [3] The elderly were the majority and their houses need repairs.; [4] The largest residential area in Helsinki, where population was growing rapidly, because the recent new development since 1960's.

Table 3. Summary of the Smart Home survey carried out in Finland (n>700[2])[9].[1]

Year	*Location*	*Target groups*
1990	the city of Espoo in the Helsinki metropolitan area	1. Randomly accessed among those with early earnings of over € 40,000 by the Population Register Centre, 2. For comparison all members of the CAD/CAM Association of Finland

[1] Posted questionnaire; [2] the answering percentage of 70 per cent

Table 4. Summary of the Elderly housing survey carried out in Finland [10].[1]

Year	*Data collection location*	*Target groups*	*Sample*
1995	The city of Tampere, a few municipalities in the Pirkanmaa region	A randomly accessed subjects aged 55 to 70 years by the Population Register Centre.	over 300[2]

[1] Posted questionnaire; [2] the answering percentage of 32 per cent

THE GAB BETWEEN INFORMATION SOCIETIES CLOSING UP

Feedback evidencing the occupants' own experiences and attitudes should be considered as an important measure of the feasibility of the technology as the technological progressiveness. As to a large extent information is lacking of post-occupancy experiences from residential living in smart homes, the following conclusions were principally drawn from the different pre-occupancy studies by summarising the references from [7] to [31]:

- Yet it cannot be taken as a given that all households in Europe have a telephone either as a fixed line or a mobile phone. The shift from the majority of fixed phone line owners to the majority of mobile phone owners has taken place in many countries and it is predicted to take place in the whole of Europe. Almost every studied young family wished to have a computer at home in Finland. It has been predicted too that domestic use dominates business users in computer markets in Europe, now using the value of the markets as a measure. Signals towards increased use of information services all over close up the gap between leading information societies and followers.
- In an advanced information society, it is possible to use ICT tools not only to a greater extent but also for more multiple purposes.
- There is a need to raise housing awareness in general and smart homes awareness, in particular. Firstly, the potential customers do not know what a smart home is and experts have different opinions of the definition. Secondly, expert lack knowledge how to satisfy the wide

variety of customer needs. Most importantly, occupants do not know how the qualitative built environment could help them to manage easier in their daily lives, or how they could manage to live independently or longer at their present homes when they face some kind of uneasiness or after they have grown old.

- Technology is supposed to help in everyday family routines of busy life styles. Despite the strong focus in RTD on smart homes applications for independent living, it is not necessarily the age or the disabilities that create the need for help and applicable technology. Smart homes technology is needed as well in carrier families due to busy lives of the two with children, as well as because of the activities and hobbies requiring special living conditions in any age. Furthermore, independent living relies on abilities that smart home facilitates.
- Safety facilities such as safety stoves, movement monitoring, water leakage detectors or fire alarms, automatic on/off system of home appliances, and smart home security system are important for all end-user groups. For example, a well-equipped kitchen with easily moveable kitchen fitments facilitate safety while difficulties in reaching objects such as dishes and china do not cause any more accidents at home and also people with limitations in mobility can work easier in the kitchen than earlier.
- The automatic doors are most probably the most common active structure. For example, the elderly thought they might need moveable walls less frequently than the respondents representing all age groups, of whom a half thought that there is a need to move walls from every 5^{th} to 10^{th} year. Other active structures will be readily accepted when needed such as kitchen fitments which can easily be moved either manually thanks to a disengage mechanism or automatically by pushing a button. The same goes also with flexible thresholds, which will fall down automatically when one steps on them.
- Contradictorily to the willingness to use home automation, there is always a portion who does not want to have it and when asked, how much they would be willing to pay for a home control system, they may say they are not ready to take if they would get it even for free.
- National circumstances and cultural differences in lifestyles influence how ICT will be used in homes or how smart homes technology is welcomed as well as how willing the dwellers are to learn how to use this new technology.
- For example, the cultural norms of the personal social contacts can affect on the use of emails and phoning. The Spanish respondents kept in touch with children, relatives and friends by phoning and meeting them, while the Finnish did this also, but in addition, they used mail and email for the purpose. No one from Spain sent letters, however (a cultural difference between the more literate Finns and the more outspoken Spanish).
- For example the Finnish elderly unlike the Spanish elderly either already know how to use ICT or they want to learn it especially if there will be the need.
- For example studies show that the social services are connected quite rarely. The possibility of using the Internet for looking for medical advice or making an appointment with a doctor is promising if asked how many have ever tried it. When asked to consider a more dependent use of health information most elderly respondents did not accept the eservice.
- For example, videophones are not popular although experiments on using videophones in a service home in the city of Tampere gave proof of its usefulness in preventing loneliness, because contact by a videophone is more personal than by a voice phone. The few who would like to have videophones found the most important purpose for it to get help in emergence situations − women more often than men felt so. The possibility to know, how relatives are doing was the second best reason for having a videophone − men were some more interested than women in it.
- Customer profiles have turned out to be problematic because surprisingly customers have started to buy products which are not designed for their profiles. Older ladies buy products designed for young men. Such demographic parameters as age, incomes and education alone

do not influence on the attitudes towards ICT or smart housing features. Contrary, gender or living style matter. A more careful look at the total life situation of customers counts more.

- The utilisation of new technology differs between genders. Women are interested in making everyday living easy. They wish to support social contacts, like educative or otherwise advantageous eservices. Men favour entertainment, sports and games, young men in particular. Men like to advance their knowledge and skills in ICT via the ICT.
- Because pensions cannot compete with salaries, the elderly cannot be expected to be the best buyers of the ICT to home, but they are by no means against having it and are not onlookers when the use of ICT is in concern. Those in a very high age may not at all make acquaintance with ICT or smart housing technology.
- The research on the intelligent office buildings has proved that neither one intelligent feature makes the building intelligent nor can any good quality feature substitute and even mitigate bad quality of building installations and properties. The reality of co-effecting factors tells to take care of the good quality of all of the features for gaining the satisfactory results.

CONTROLLING POWER OVER ENERGY EFFICIENCY AND INDOOR AIR

The Demulog interview proposed energy-saving measures based on the time limitations of the electricity supply to prevent electricity consumption peaks. The acceptance of energy saving technology was low if the basic needs are not satisfied comfortable enough or the control set disturbs the routines of daily life, which in the case concerned the restriction of room temperatures and the energy efficiency of home appliances [13]. Nevertheless, the respondents in general did not reject all energy saving measures, such as lower temperatures when nobody was at home or during the night and in bedrooms.

During the Elderathome interviews seniors in Finland and in Spain accepted a temperature delimiter for the faucet, aiming at saving hot water and contributing to energy conservation (Table 5). It also prevents people from scalding their hands. Many of them had this device already installed because of safety and energy saving. Likewise, they appreciated home control facilitating easy living and energy efficiency, e.g. remote control possibilities and lamps that turned on automatically when someone entered a room and switched off when the room was empty. Ease of living was a welcomed benefit of independent living technology. It concerns also the previously mentioned adjustable kitchen cabinets, which are safe and easy to use by all fulfilling the principles of Design-for-All at the same time.

Just as the younger families who answered the Smart house survey [9], also the Finnish elderly interviewed for the Elderly housing survey liked such safety facilities as a safety stove, movement monitoring, water leakage or fire alarm, etc. [10]. Also, another end-user study carried out in the Pirkanmaa region in 2004 proved the importance of the safety technology when out of 39 smart home facility options, the top five were about safety [31].

On the basis of the two case studies: the Demulog interview and the Elderathome interview, facilitation of living or safety rather than energy saving alone was an acceptable argument for control technology. Parallel to this finding, Karjalainen [20] found with a nationally representative sample that user interest in the user adaptive and other new control systems was quite low in Finland. Users are afraid of loosing control over the systems. To be accepted, the new control systems should improve control over the indoor environment − not reduce control opportunities − and bring significant benefits e.g. in improved comfort or saved energy.

Table 5. Testing popularity of effective regulations, standards and recommendations (the percentages of responses[3]) [8].

I have already		I'd like to have now			I'd like to have if I need		I don't want to have	
FIN[1]	E[1]	FIN[1]	E[1]	**PERSONAL HYGIENE**	FIN[1]	E[1]	FIN[1]	E[1]
71	65	9	25	A temperature limiter (thermo-regulator) in the faucet[2]	11	3	9	7
	12	29	54	A hydro-massage system in a bath or a shower	29	2	43	32
84	85	7	3	An intimate hygiene shower next to the toilet seat	8	1	1	11
5	64	2	10	The heights of the bathroom furniture are easy to change.	75	6	19	20
100	81		17	Only one handle instead of two in the faucet to make it easy to use.		1		1
61		5		A sauna in your home[4]	2		33	
80		4		A common sauna in the building[4]	10		6	
13		3		Handrails in the bathroom[5]	81		3	
				SAFETY				
7	87	2	12	The slipping preventing floor material with normal appearance	82	0	9	1
6	82	2	15	Beautiful colour choices to help in seeing easily	85	0	7	3
9	41	2	12	An alarm system which can be easily installed in the future	78	28	12	19
	57	67	26	Instructions for the safe behaviour and the use of installed facilities	33	6		11
				HOME CONTROL				
	4	22	61	Remote control possibility (wireless) of home appliances and home electronics, e.g. turning lightning on and off, or opening the door	78	15		20
2	36	5	8	Alarm buttons, safety bracelet, motion censor and other such equipment to facilitate connections to other people in an emergency[2]	88	53	4	3
1	10	5	46	Lamps that turn on automatically when entering a room and which turn off when no one is present	68	8	26	36

[1] FIN ≡ Finland, E ≡ Spain, [2] the question concerns both safety and comfort, [3] pick of several choices of multiple-choice questions possible, [4] relevant only in Finland, [5] asked only in Finland

CONTROL SWITCHES

The Elderly housing survey revealed that most of the Finnish respondents in the age of 55 to 70 (60 percent) stick to the traditional control devices [10]. The office workers also tend to trust the traditional or familiar control devices [15]. The popularity of other forms of home controls was as follows: remote control units (21 percent), a separate control panel (13 percent), door side panels (8 percent), speech control (7 percent) and a computer (3 percent). The result from the Smart homes survey was quite similar. Out of new lighting controlling possibilities, a natural light control and the timing of the switching were more often acceptable among the elderly than the other age groups (Table 6).

Despite the results to the Elderathome interview also showed that remote lighting controllers were not in use among the Finnish elderly, however the elderly did favour them, when they were needed (Table 5). Nevertheless, in Spain the remote control possibility was either in use more than in Finland or it was accepted more unconditionally, while 60 per cent wanted to have it right away. When during the Elderathome interview the Finnish sample with a majority of women and the Spanish sample with a majority of men were compared, it was found that the sample with the male dominance favoured the remote controller more unconditionally than the one with the female dominance. This result was parallel to the findings of Karjalainen [19], who found that men rather than women use the control devices although good indoor air was more important for women.

Again the comparison between the female and male dominant samples proved that the women did not use alarm buttons, safety bracelet, motion censor and other such equipment to facilitate connections to other people in emergency as often as the men did. However, more often the women than the men were interested in having it if needed (Table 5). Different reasons separately or together might cause this gender difference. First, the mode of control counts; men like remote controllers [19]. Further on, men are more interested in home controls in general. Second, the wish to have a contact to other people which Fox (2004) claimed to be important to women might make women interested in controlling. In addition, women feel more vulnarable than men do [8], and that is why they might have considered that they will become interested in control possibilities when an emergency was involved.

Of the importance of the vulnerability to the need of a safety facilitation gave proof the fact that a good share of the interviewed Spanish respondents had the emergency call facility most probably because of need while one third of the Spanish respondents dwelled in the special housing designed for elderly. Many of them wanted to have it if needed as did the majority of the Finnish respondents. The Elderly housing survey concluded that the control made was not acceptable until the need for existed. Then even the video monitoring could be accepted in some cases. Many elderly did not bother about new technology when they felt fit and function and wanted to enjoy life and the services technology could offer.

Table 6. The grounds for control of lighting after the Elder housing survey and the Smart homes survey, the percentages of the respondents

Control operation	Elderly	All ages
Control according to daylight	48	12
Dimming	45	32
Timing of the switching on and off	37	11
Presetting for various situations, e.g. reading, watching tv, dining, having guests	19	14
Remote control	-	12
Control panel for multiple purposes	-	11

The pick of several choices of multiple-choice question possible

DISCUSSION

It seems as if a certain point has been reached in the evolving markets of smart homes. The attitudes of the end-users were rather accepting than rejecting the smart home technology. A new direction for the R&D work concerning smart homes technology needs to be taken. In order to win the markets the producers will most probably be dependent on new ideas to apply smart homes in a broad sense. A lively debate between them is wanted in order to embed this radical product in to the market and empower the lives of various future end-user groups.

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Estimating the relationship among occupant behaviours and indoor environmental parameters using Bayesian networks

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This paper investigates the relationship among occupant behaviors and indoor environmental parameters. We collect data from three aspects: indoor environmental variables measured by a wireless sensor network, body sensors carried by occupants, and questionnaires completed by occupiers. The wireless sensor network measures four parameters: temperature, humidity, light and barometric pressure. The body sensors measure collect data including burned calories, physical activity, body position, and galvanic skin response from their wearers. The questionnaire collects data such as occupier's behaviour and mood. The experiment is conducted in an air-conditioned office where temperature can be controlled by the occupants. Based on such information, we build up a Bayesian network model revealing how indoor environmental parameters and occupant's behaviour impact physical parameters of human bodies. We also compare our models with Fanger's thermal comfort model.

1. INTRODUCTION

The increasing miniaturisation of RF devices and microelectro-mechanical systems (MEMS), as well as the advances in wireless technologies, has generated a great deal of research interest in the area of wireless sensor networks (WSNs), which provide a promising infrastructure for gathering information about parameters of the physical world.

A WSN system is a group of sensors linked by wireless medium to perform distributed sensing tasks. WSN systems have attracted a wide interest from academia and industry alike due to their diversity of applications. Recent advances in wireless sensor network technology have enabled the development of low-cost, low-power, multi-functional sensor nodes that are small in size and allow untethered communication over short distances. Such systems allow close monitoring and adaptive control of building equipment, materials performance (such as those in the façade) and environmental conditions including temperature, air flow, indoor air quality, lighting , sound and the stated well-being of the occupant in relation to their surroundings.

Systems will also enable a proper cohesive data management system to be organised for buildings which will help to understand the strategies required for operating the building so that energy and water savings are achieved; environments are healthier; and there is closer relationship between the occupant and the building. It will also help to achieve a leaner and more effective design, construction, operation and facilities management regime. For example we should be able to understand very much more deeply the patterns of use and also enable prediction of failure conditions thus informing maintenance regimes.

A Bayesian network is a graphical model that encodes probabilistic relationships among variables of interest. When used in conjunction with statistical techniques, the graphical model has several advantages for data analysis. Firstly, a Bayesian network can be used to learn causal relationships, and hence can be used to gain understanding about a problem domain and to predict the consequences of intervention. Secondly, because the model has both a causal and probabilistic semantics, it is an ideal representation for combining prior knowledge (which often comes in causal form) and data. Therefore, it is worthwhile to use Bayesian networks to explore the relationship among various variables that might influence indoor air quality, occupant's productivity, and other variables of interest. Readers are referred to Jensen (2001), and Neapolitan(2003) for better understanding in the Bayesian networks.

There is now much evidence to show that the environment of buildings can affect the work performance of the occupants and can alter their state of well-being (Clements-Croome, 2005). This means that environmental design can enhance the value of the built asset for an organisation. This paper investigates the relationship among occupant behaviors and indoor environmental parameters. We collect data from three aspects: indoor environmental variables measured by a wireless sensor network, body sensors carried by occupants, and questionnaires completed by occupiers. The wireless sensor network measures four parameters: temperature, humidity, light and barometric pressure. The body sensors measure collect data including burned calories, physical activity, body position, and galvanic skin response from their wearers. The questionnaire collects data such as occupier's behaviour and mood. The experiment is conducted in an air-conditioned office where temperature can be controlled by the occupants. Based on such information, we build up a Bayesian network model revealing how indoor environmental parameters and occupant's behaviour impact physical parameters of human bodies.

2. EXPERIMENT SETTINGS

We are interested to build a sense diary that can serve two-fold purposes: to minimise energy consumption, and to maximise occupant productivity. In order to achieve these two objectives, a research plan is set as shown in Figure 1. We planed to use wireless sensors to measure

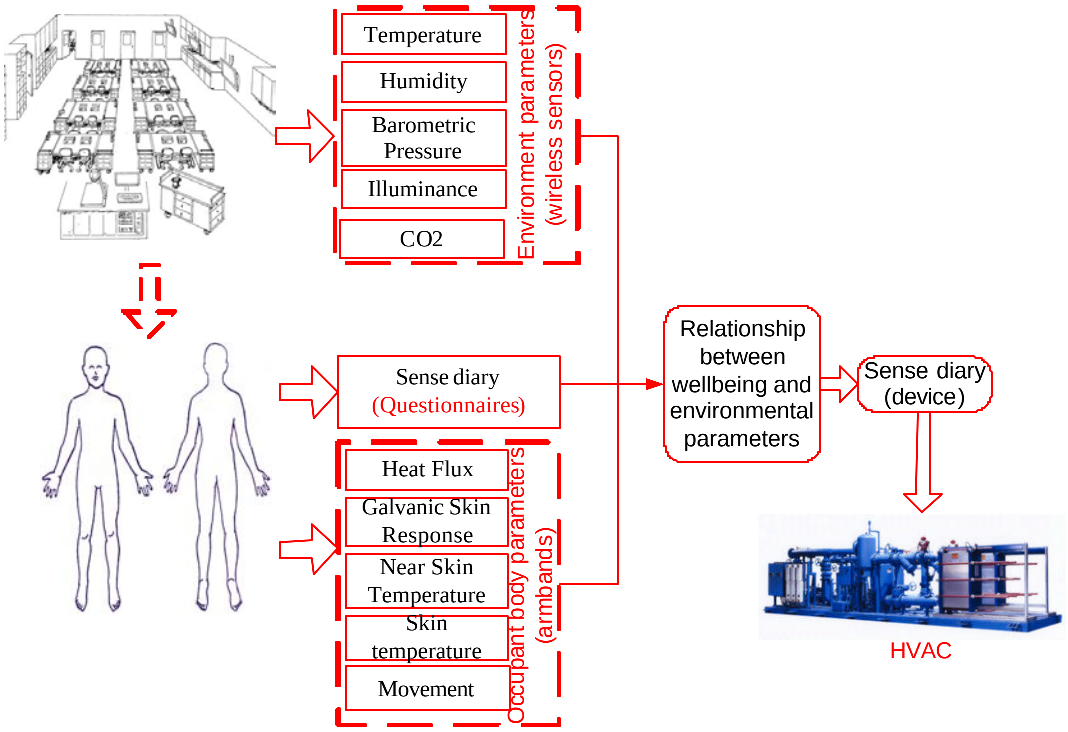

Figure 1. Research Plan

environmental parameters, and apply armbands to measure occupant's body parameters. The armbands can also be used to measure occupant behaviours through their embedded sensors. With such parameters, people are able to explore how the change of environmental physical parameters can impact on occupant body parameters such as skin temperature. As some characteristics can not be measured with sensors, we designed a questionnaire, also called ***soft sense diary*** (see Appendix). The collected data are then analysed, and relationship among variables are explored. The wireless sensor kit was bought from Crossbow (www.xbow.com), and three identical armbands from BodyMedia (www.bodymedia.com/).

The wireless sensor kit has four smaller sized sensors, each of which measures temperature and relative humidity (see Figure 2a); and two larger sized, each of which measures temperature, relative humidity, and light (see Figure 2b).

The armband measures several parameters, below lists some

- **Accelerometer** measures vertical and horizontal motion of its wearer;
- **Heat flux** measures how much heat the wearer's body is giving off.
- **Galvanic skin response** measures skin conductivity affected by physical exertion and emotional stimuli such as psychological stress
- **Skin temperature** reveals the body's core temperature trends affected by the level of a person's physical exertion or lack thereof

Figure 2a

Figure 2b

Figure 2c

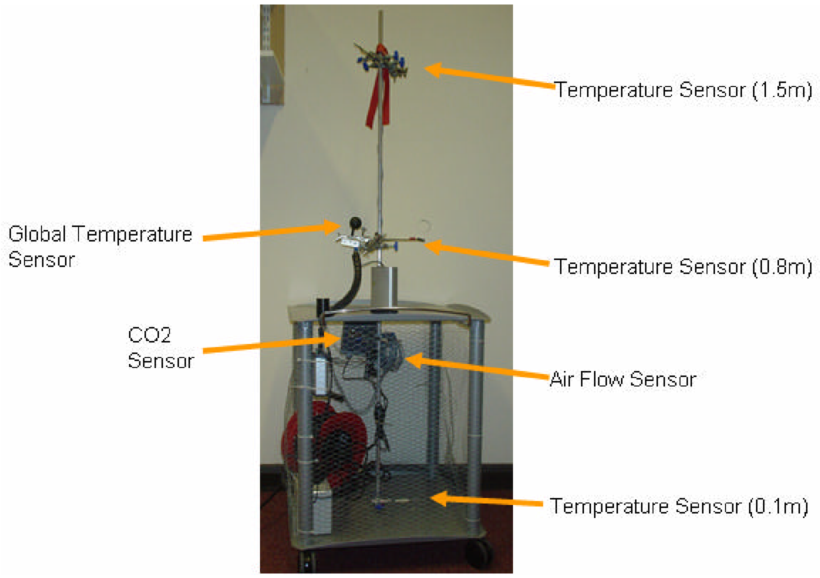

Figure 3 A sensor set

In order to collect Carbon dioxide and mean radiant temperature, a sensor set was used to measure these parameters (see Figure 3). As shown in Figure 3, apart from three temperature sensors and an air flow sensor, a carbon dioxide sensor and a global temperature sensor are installed in the sensor set.
Figure 4 shows an example of parameter distributions collected from an armband wearer. For the figure, for example, around 10:00 o'clock, there is a dramatic change in heat flux and skin temperature, this is because the armband wearer went outside in a lower temperature environment.

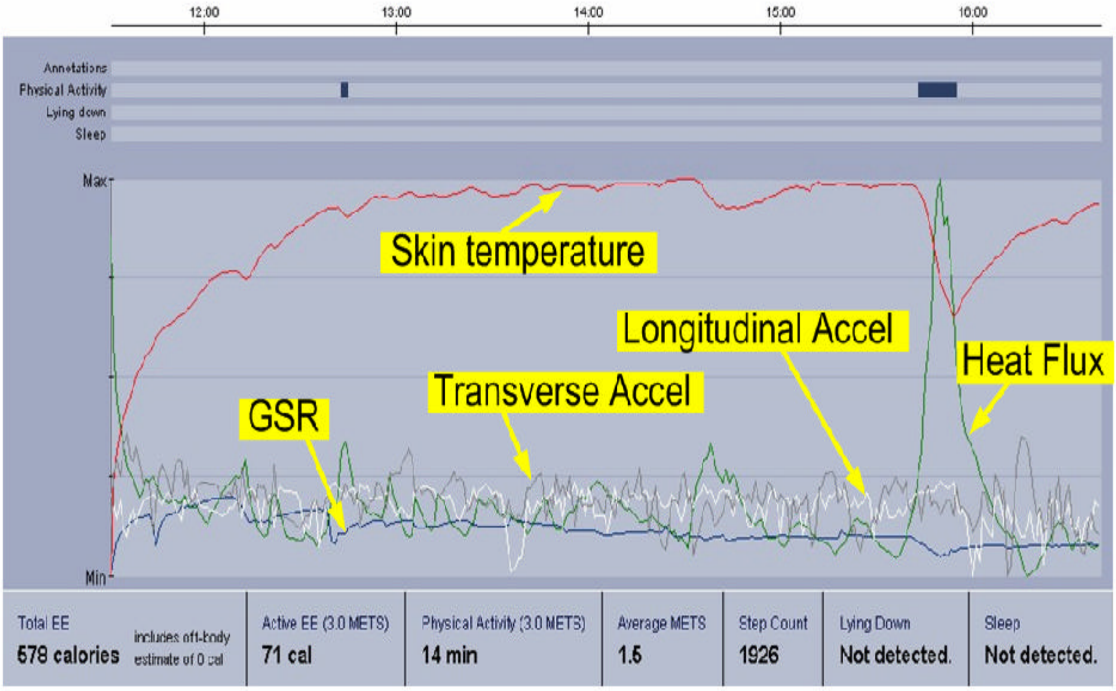

Figure 3: an example of the parameter distributions of an object

We set three day experiment in a 3m×5m office. In the experiments, objects wearing armbands were working in the office, and the sensor set and the wireless sensor kit were placed in the office. By adjusting the air-conditioner temperature one degree Celsius every 40 minutes, the temperature was changed from 16 to 28 degree Celsius.

The object profiles are listed in Tables 1, 2 , and 3.

The first experiment (8th November, 2006)

Subjects	Gender	Age range	Height (cm)	Weight	Clothing
A	Female	20-25	160-165	55	0.8
B	Male	35-40	180-185	75	0.8
C	Male	45-50	180-185	80	0.8

Table 1. Object profiles in the first day experiment

The second experiment (9th November, 2006)

Subjects	Gender	Age range	Height (cm)	Weight	Clothing
B	Male	35-40	180-185	75	0.8
C	Male	45-50	180-185	80	0.8
D	Male	45-50	170-175	70	1.0

Table 2. Object profiles in the second day experiment

The third experiment (8th December, 2006)

Subjects	Gender	Age range	Height (cm)	Weight	Clothing
E	Female	45-50	170-175	75	0.8
F	Male	40-45	170-175	80	0.8

Table 3. Object profiles in the third day experiment

In a certain degree, Galvanic skin response (GSR) can reflect wearer's mood. Hence, we compare the GSR of the objects in Figure 4, from which one can see the different patterns.

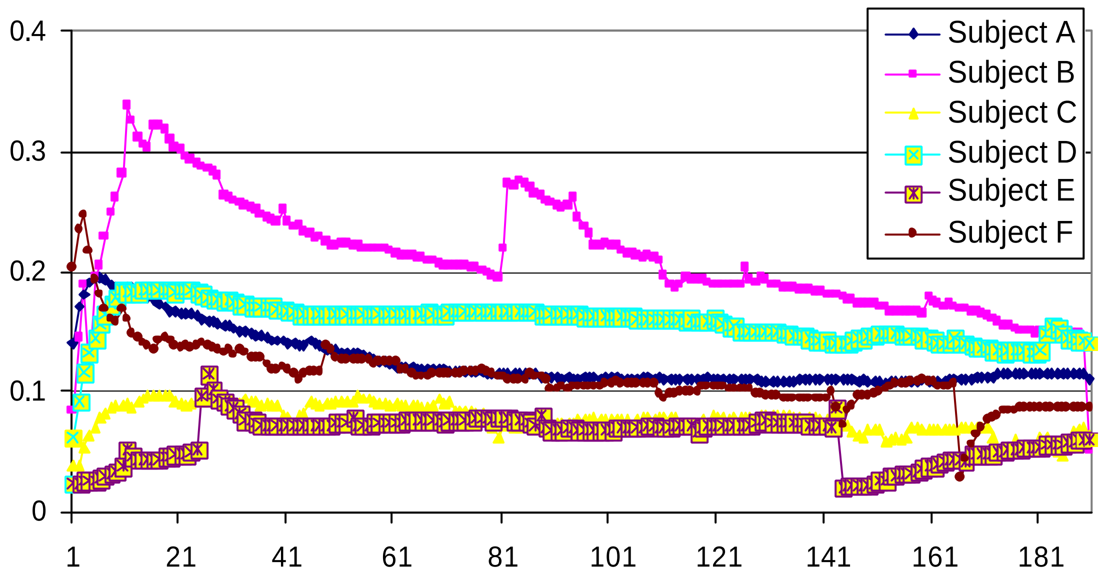

Figure 4. A comparison of GSR of the armband wearers

3. BAYESIAN NETWORK DEVELOPMENT

Through the above experiment, data were collected, and then analysed. A Bayesian network model is then built based on the data. The algorithm of building a Bayesian network model can be found from to Jensen (2001), and Neapolitan (2003).

For convenience, we borrowed a Bayesian software package developed by Cheng (2001). The Bayesian network model built on our collected data is shown in Figure 5. It shows, for example, that the Near Body Temperature (shown in *Near_body_tem* box) is a factor resulting in the

change of GSR (shown in *GSR_averag* box), while GSR is one of dependent factors to Energy Expenditure (shown in *Energy_expend* box).
The strength of the causal relationships can also be estimated from the Bayesian network models. For example, from the model, it shows that the causal relationship between Near Body Temperature and GSR is stronger than the causal relationship between GSR and Energy Expenditure.

CONCLUSIONS

Improving indoor environment to achieve a high quality living and work area can improve people's well-being and productivity.
This paper applies Bayesian networks to model the causal relationships between indoor environmental parameters and occupant's body parameters, on the basis of the data collected by a wireless sensor network, and body sensor sets.
Essentially, occupant behaviour, building services systems, indoor and outdoor environments are factors relevant to energy consumption, and occupant productivity. By revealing their relationships, it would be helpful for practitioners to design and maintain a better indoor environment for occupants.

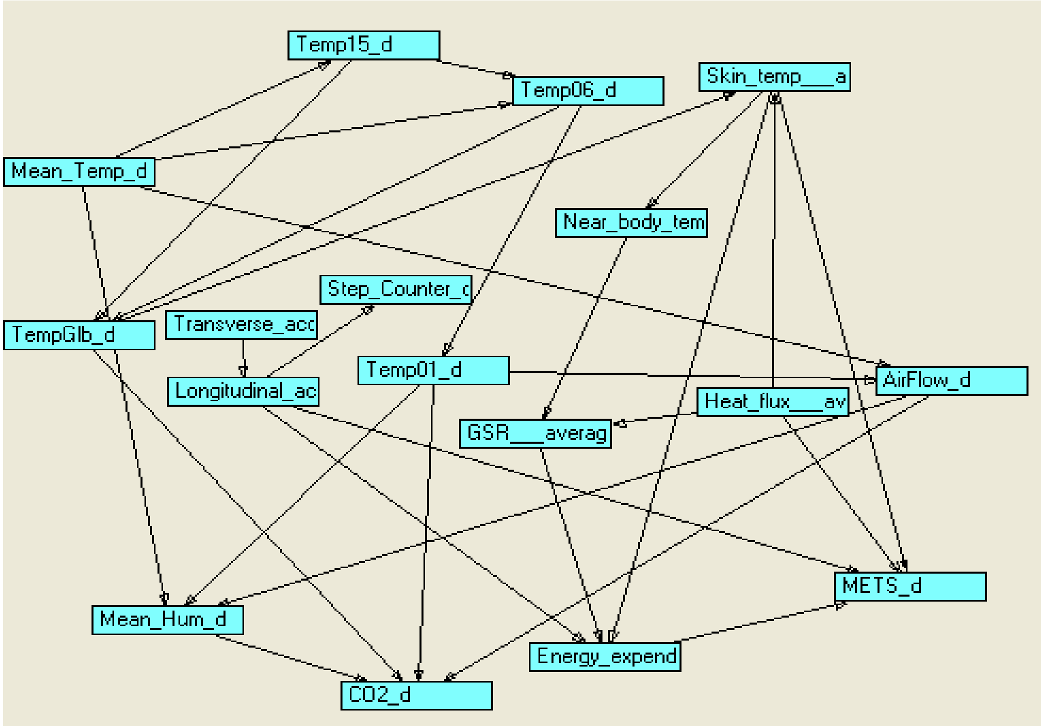

Figure 5. The Bayesian network model

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Appendix

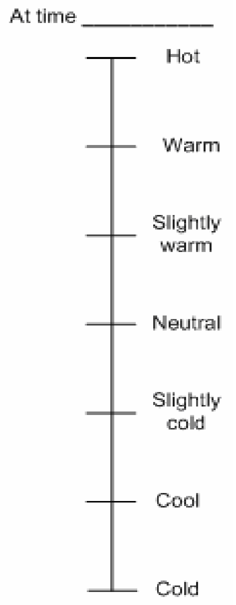
Sensors Recording Parameters of Indoor Environment and Occupiers

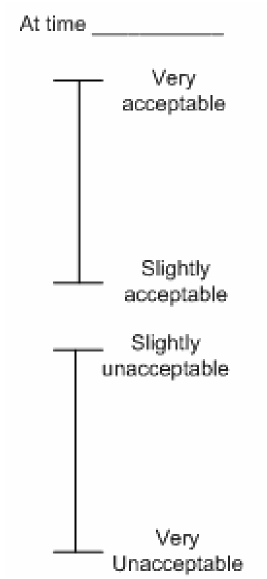
A *Soft Sense* Diary for Recording Occupant Response to Indoor Environment and Occupier Behavior

Date:__________________ Time:__________________

Comfort Data: Recording response to environmental conditions

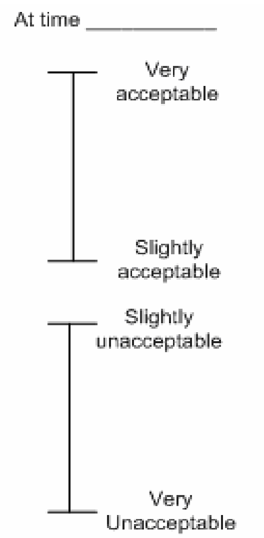
A. About the indoor temperature, I feel B. Do you feel air movement around you?





B. Do you feel air movement around you?

Yes☐ No☐



C. What is your general feeling about the office environment?

It is bad |———————————————————————| It is good.

Breakfast:
No ☐ Yes ☐

Lunch:
No ☐ Yes ☐

D. About the indoor illumination, I feel
At time _________
It is too bright |———————————————| It is too dim.

E. About air quality
At time _________
Air stuffy |———————————————| air fresh

F. About air quality
At time _________
Air humid |———————————————| Air dry

Occupant behavior: Recording Actions and Activities

Entering and Leaving the Room

I leave the room, at time….. To do the following activity
___________________ ________________________

I open the door, at time
________; ________; ________; ________;

I shut the door, at time
________; ________; ________; ________;

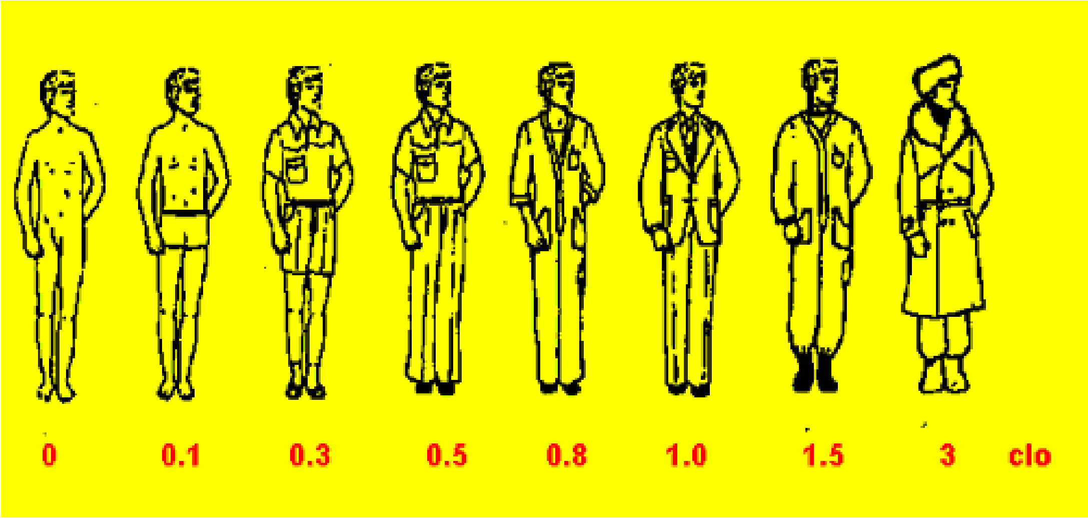
I switch on the air conditioner, at time
________, set temperature to be ______; ________, set temperature to be ______;

I switch off the air conditioner, at time
________; ________; ________; ________;
Referring to Figure 1, your clothing unit is ____.

Occupant profiles

Age range: ______ | Gender:_________

Height range _______ | Weight range:_________



Achieving Effective Power Demand Control on A University Campus by Applying Quality Projection Methods

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SUMMARY

Current power demand control techniques are achieved by power measurement, demand projection, and utility unloading capabilities. According to the metering readings, the actions of unloading utilities will be taken if the projection demands exceed the contract capacity level (desired demand power). As the chosen projection method can affect the performance of the demand control strategy, the aim of bill cutting highly relies on the precision of the projection methods. In this project, the power usage of Tsing Hua University in Taiwan was metered and base on the metered data, different power demands were projected due to projection methods. These projection demands were used to compare with the real power demand, and the comparison results will be used to examine the accuracy of each projection methods. Moreover, utility unloading strategies will be applied with the projection demands to prevent the power demand to exceed the desired level.

INTRODUCTION

The power demand is increasing continuously due to the fast economic growth and the change of life style. The demand required for air conditioning systems during the summer in Taiwan can cause serious problems to the power supply system as the peak load is approaching the maximum capacity. As a result, the power supply company has to invest more money to build up the reserve margin capacity for demand needs. In addition, the cost of energy is increasing obviously as the rising of Brazil, Russia, India and China. Therefore, it is crucial to reduce the problems of short of reserve margin capacity by controlling the growth of power demand. On the other hand, the strategy to achieve good energy usage efficiency is often adopted by power demand side around the world. In addition, the methods to reduce the peak load such as load management, demand control and off-peak power usage are often applied in developed countries. Later on in this paper, the roles of demand control techniques, demand projection methods and the simulation of projection methods in energy management will be shown.

The power demand increasing due to air conditioning systems in summer brings big problems to the power supply system in Taiwan. Even though the Tai-power company is applying reason rate to reduce the peak load, the result is very limited. Hence, the only solution for Tai-power is to build up more reserve margin capacity. However, building up the power plants to match the future needs is a difficult task in Taiwan as the compact to the environment and the short of land. Hence, it is necessary to find out the methods to reduce the growth of power demand by applying efficient energy saving under the pressure of energy shortage.

The definition of contract capacity is the average power consumption in a fixed period of time which is called the demand interval. The demand interval is regulated by each power supply company, and it is 15 minutes with Tai-power in Taiwan. The contract capacity is the agreement between the power supply company and its end users. The volume of power demand indicates the requirements of the end users and it means that the least capacity that the power supply company has to offer to meet users' needs. For power supply companies to offer quality power for the users, it is necessary to invest more money to build up the reserve margin capacity while is needed, and the required investment is then be used to set up the rules for over demand surcharge (penalty charge).

The power demand control technologies were merely found in heavy industrial cases as the costs were high. By the advance of microcontroller technologies with power measurement and software, the prices of power demand control equipments are getting cheap and the performances are also getting better. As a result, the implementation of power demand control is getting popular these years in medium and small enterprises and residential and commercial users were targeted as the next focusing market. This is because that power demand control can bring benefits to both the power suppliers and the end users.

DEMAND MANAGEMENT

Analyzing the power consumption model in Taiwan, the heaviest demand during the peak load in summer comes from the use of air conditioners. Hence, minimizing the peak load by implementating the real-time load management over the air conditioners and related devices is considered as the best solution for effective results. According to the records, the peak load of the summer time occurs between 10 AM to 12 AM and 2 PM to 4 PM, and the second period consumes more than first period does. To a power supply company, enough reserve margin capacity is required to meet user's demand, and on the other hand, the users have to pay the bill because of the power consumption. For contract users, the power consumption required by many of the users is not always higher than the contract capacity levels, and money can be saved if good load control is implemented. Later on, the relationship between contract capacity and demand control will be explained.

The agreement between the end users and the power supply companies is often called the contract capacity and this means the highest power requirement from the users. The volume of contract capacity means the plan of generating capacity and reserved margin capacity to the power supply company. In the other hand, the contract capacity shows the relations between the power bill, fixed bill, and the over demand surcharge to the end users. Although a higher value of contract capacity can reduce the over demand surcharge (penalty charge), the fixed bill can take more money from the users and vise versa.

In Taiwan, the Tai-power meters several different types of data such as accumulated kW/h, maximum demand power and so on. The maximum demand level will be cleaned as zero after being recorded, and this piece of data will be compared with the contract capacity level every 15 minutes. If the newly recorded data is bigger than the old recorded data, the old data will be replaced, and the old data will be saved if the new data is smaller than the old one. As a result, the maximum demand power will be remained, and this data will be used to count the power bill and the penalty charge. To understand the power need of a system, the power demand is often a better solution than real-time power consumption as it is the average power consumptions during time intervals. Later on, four graphs will be used to show the meaning of metered data between the real-time power consumption and the demand power. First one is a real-time power consumption recorded in a convenience store, and power demand is used to show the trend of the power consumption in the second one. Figure 3 is the power consumption record of a university, and the demand power can show more details such as the time, the value and times when exceeding the desired demand level happens.

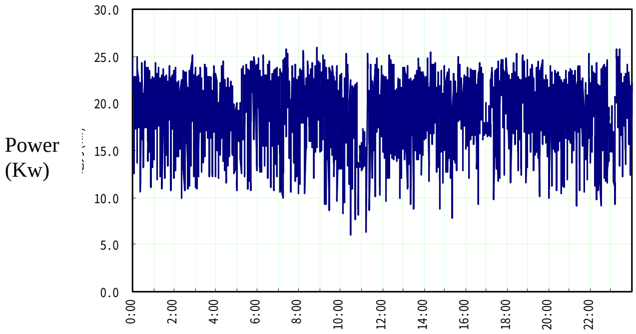

Figure 1: Real-time power consumption from a convenience store

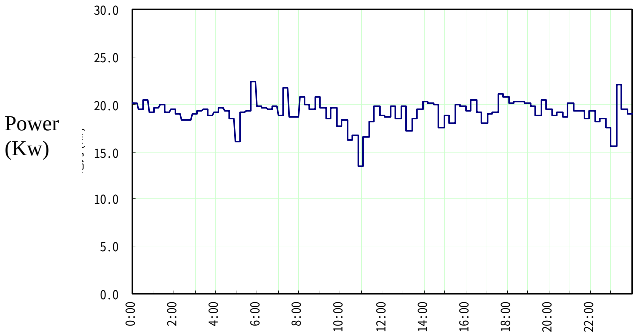

Figure 2: Power demand acquired by the convenience store

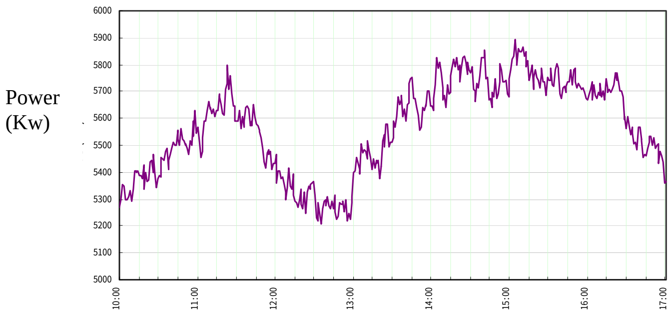

Figure 3: Power consumption recorded in a university

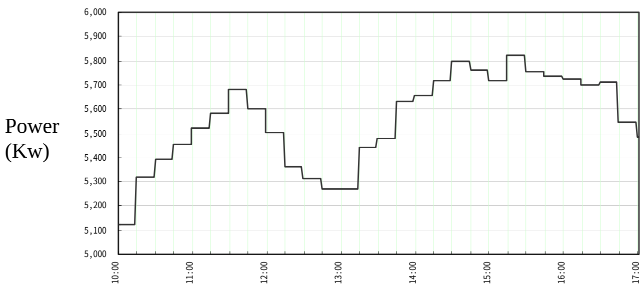

Figure 4: power demand recorded in a university

RESULTS

Demand control method

To achieve summer demand control in Taiwan highly depends on the load control plan and unloading strategy which can be used to indicate the unloading sequence of some secondary devices temporarily to achieve the power demand control to prevent the power usage not to exceed the contract capacity level. As the air conditioning devices including the fan systems, condensers and the pumps are the major targets, to apply real time control over these devices is considered as the best approach to reduce the power demand, and three key issues are: setting the best deal of desired power demand, the unloading strategy and the power demand control.

The desired power demand setting

The desired power demand which indicates the maximum volume of power demand is also the agreement between the Tai-power and the end users. Hence, power bill can be cut by setting an appropriate desired power demand level, and the power demand can be used as the target for load control. Very often, the desired power demand is calculated and analyzed by reviewing the power bill in the pervious year and the used devices.

Unloading strategy

The time interval defined by Tai-power is a 15-minute period, and the recorded maximum demand in a month will be used to charge the power bill including penalty charges. Therefore, users have to think the relation between their own systems composed by devices and the contract capacities. Every single device has it own priority while changing the operation mode to reduce the power demand to meet the desired demand level is needed. According to the priorities, the unloading plan of a system can be built and here use a fan system as an example, step1, to turn off the fan and step 2 to low the fan speed and so on. In addition, the plan of unloading devices will affect the performance of the demand control, e.g. to prepare 4 devices which need 100 KW individually to reduce 100 KW from exceeding the desired power demand is an idea plan.

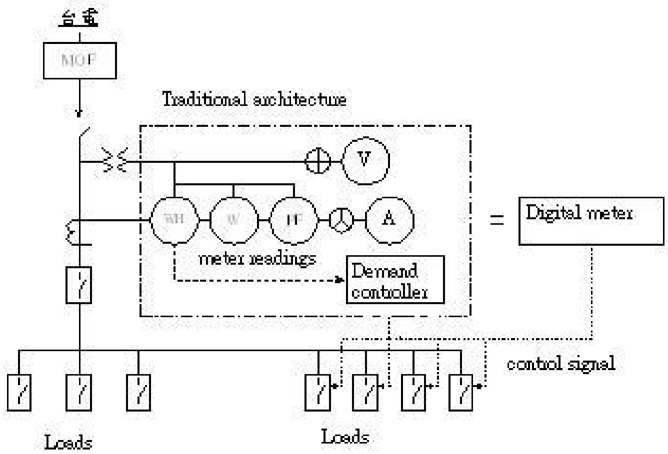

Figure 5: the architecture of a power system and a demand control system

Demand control strategy

Figure 5 shows the architecture of a power system and a demand control system which includes power meter, protecting relay and so on. In a traditional demand control system, the demand is projected by counting the pulses from the meter. If the projected value is higher than the desired demand level, the system will activate the relay to turn off the defined devices. Nowadays, as the progressing of mciroprocessing technologies, the demand control can be implemented within the power meter. As a result, the cost to implement power demand is much cheaper than it was and it will be a huge advantage while promoting this system.

rules of demand control

There are two parts included in the early stage of demand control system, the power metering and control. The front end, power meter, takes voltage transformer (VT) & current transformer (CT) to measure the power consumption. The measured data will be sent to the demand control part to unload devices while needed. The demand control includes parts such as signal input, function setting, display, record, calculate, unloading function and warming. The performance of device unloading comes from the accuracy of demand predicting. Figure 6 shows the rule of demand projection and unloading sequence. In the figure, Q indicates the projection demand, Ps is the desired demand and Pr is the targeted demand (Pr≦Ps). The cycle of demand projection, warning, device unloading, and device loading will take place every 15-minutes.

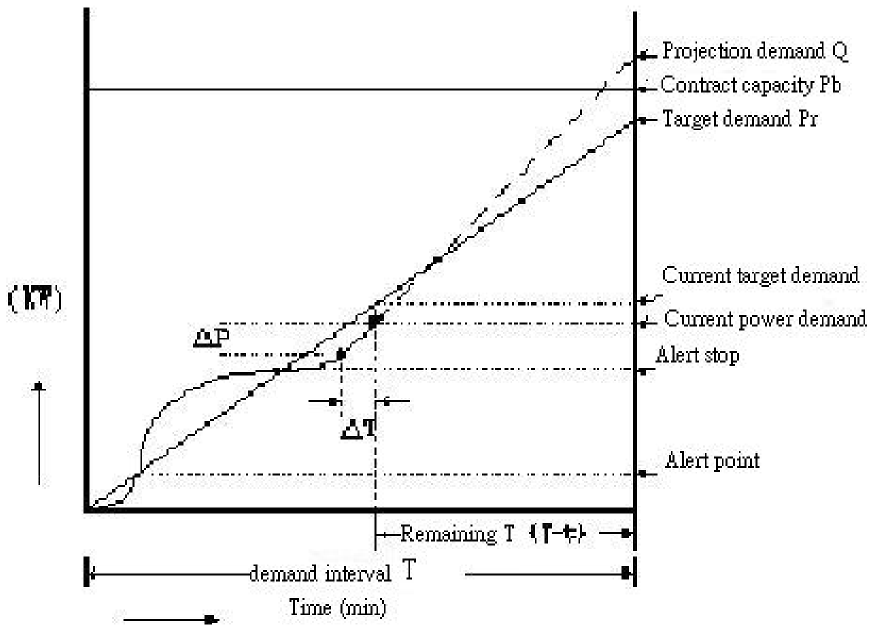

Figure 6: the rule of demand projection

Demand projection

Demand projection is an important function of demand control, as it directly affects the performance of demand control. After microcontroller and DSP were used in power metering, the function of demand projection is often implemented within a digital power meter. Despite of the rules mentioned in the previous section, there are few different rules for different power systems with different characteristics. Here uses a power meter as an example and the rule is:

$$\text{Demand} = \frac{1}{T} \bullet \int_{t}^{t+T} P(t)dt \quad \text{---------} \quad (1)$$

T：demand interval

P(t)：instantaneous power

Actually three different rules are offered to users, and they are instantaneous projection, first-order projection and second-order projection. Instantaneous projection shows the average demand in the past time interval. The first order presents the real-time power demand with the remaining time interval. The second order is used to indicate the changing rate of the first order result and the remaining time interval. Here are these rules:

$$\text{Demand} = \frac{1}{t2 - t1} \bullet \int_{t1}^{t2} P(t)dt \text{ ------- } (2)$$

In order to avoid a huge difference between the projection demand and the measure result, a synchronous mechanism to receive external signal is included within the demand control. While no external signal is available for synchronous, a sliding window method can be used. The main difference between the sliding window method with the previous method is that the time interval will be cut into several sub time interval, and this solution can delivery quality results while comparing with the measured meter reading.

Here use a university dormitory as an example, and several demand projection methods were applied to find out the hiding characteristics of these methods. These methods include instantaneous, first order, second order and sliding window method. From the simulation results, the results of demand projection are not quite the same due to the method, see Figure 7.

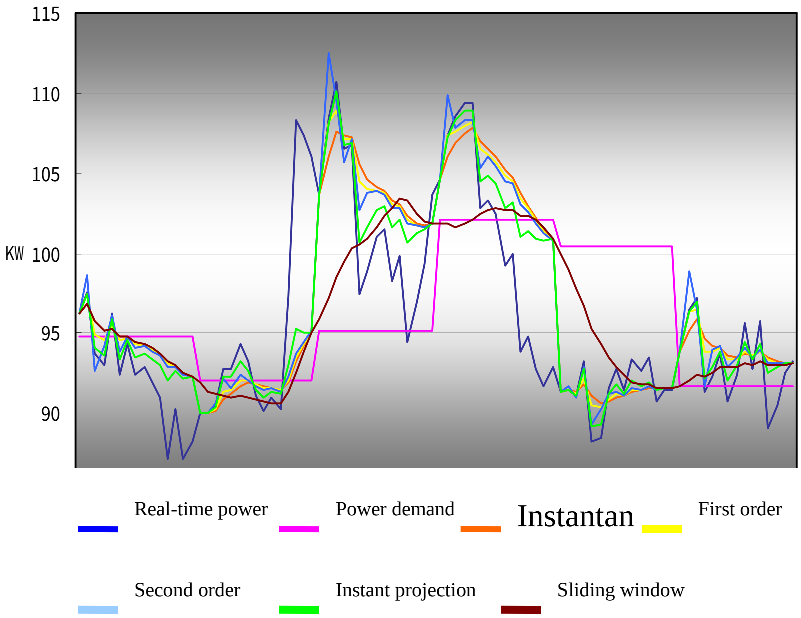

Figure 7: Demand projection results from different methods

DISCUSSION

Quality demand control can reduce the peak power requirement and also reduce the penalty charge because of exceeding the contact capacity. However, it is essential to set a desire demand level and unloading plan for achieving quality demand control. Through this research, it can be found that users has to apply different demand projection methods while the loading is constructed by different devices, e.g. to choose a fast reacting demand projection method while the ratio between unloading quantity and loading quantity is low. The finding of this research will be applied to the supermarket energy management project.

ACKNOWLEDGEMENT

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Long-term performance of VAV laboratory ventilation

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SUMMARY

The objective of the paper is to evaluate the long term performance of the HVAC system in a laboratory building equipped with an advanced fume cupboard technology and a building automation system. The exhaust airflow of the fume cupboard was controlled in terms of sash position and a control unit. In addition, selected fume cupboards were equipped with presence sensors. The air supply and exhaust flow rates of the laboratory rooms were monitored during two years in normal operation conditions. The data series were presented as cumulative distributions, which enabled comparisons of the air flows between various fume cupboards or laboratory rooms. The performance of the VAV fume cupboard system was compared with a hypothetic fume cupboard with constant exhaust airflow providing the same face velocity as the VAV system. It turned out that saving potential when using the VAV system was about 60%.

INTRODUCTION

Control of indoor climate in chemical, biochemical and toxicological laboratories is a challenging task from the point of view of occupational health, energy economy and maintenance. Airborne contaminants occurring in the laboratory environment may have irritating, allergic and carcinogenic effects or may affect specific organs, such as the central nervous system. Therefore, effective control of hazardous substances handled in these laboratories is absolutely essential.

The strict requirements for protection against airborne hazardous contaminants have led to the use of large air flow rates in laboratory buildings. Typically air flow rates in laboratory buildings are about 5 times higher than in office buildings [1]. Although high air flow rates in laboratory rooms are necessary for safety reasons, another reason for high energy consumption is the fact that standard design and control technology have been applied for decades. The technology for laboratory fume hoods, the first barrier in the prevention of exposure to air contaminants, and demand control ventilation have made notable advances during recent years. These advancements together with building automation systems (BAS) offer notable benefits for improving working environment and energy economy in laboratory buildings. The modern BAS produces vast amounts of data for remote monitoring, control, fault identification and diagnostics purposes.

In practice, it seems that service personnel use the BAS data quite narrowly to detect alarms and to react immediately in order to correct the faulty function and cancel the alarm signal. More systematic and automated fault detection and diagnostic procedures based on the on-line BAS data have been developed and validated in real buildings during recent years [2]. Both the alarm-oriented practice and automated on-line diagnostics are based on short term changes in the HVAC system. Although these procedures have proved very useful, long-term monitoring of the BAS data can offer further benefits. Long-term monitoring data reveals how the systems are used in real buildings. Such information is valuable for planning refurbishments in existing buildings and designing new laboratory buildings.

The general objective of the paper was to evaluate the long term performance of the HVAC system in a laboratory building equipped with advanced laboratory fume hood technology and building automation system. The specific objectives were as follows:

- to monitor the of use of the laboratory fume hoods in terms of sash position
- to compare the VAV hood system with the hood system involving constant air flow
- to describe dynamic behaviour of the hood system due to change of sash position
- to produce simple reporting methods to interpret the BAS data relevant to the laboratory hood technology from the standpoints of building owner, end-user and equipment manufacturer.

METHODS

Building

The long-term performance evaluations were carried out the recently built section of the Finnish Institute of Occupational Health located in the centre of Helsinki. The seven-floor building, built in 2000, consists of two parts: a laboratory wing and an office wing. The building was equipped with a modern building automation and monitoring system based on the Lon bus network. The general HVAC system consisted of mechanical supply air and exhaust systems and a cooling mode. Some one hundred laboratory fume cupboards were mounted in the laboratory wing of the building. During the design phase, the chemists and other users of laboratory facilities were asked to estimate the degree of use of the laboratory hoods. This information served as a rough estimate for designing the airflows in the laboratory wing.

The exhaust airflow of the fume cupboard hood (1.2 m wide, nominal exhaust flow 160 l/s) was controlled through sensor detecting sash position and a control unit with an alarm function. The control system (Fanison) maintained the face velocity at least at 0.4 m/s in the sash opening in the normal use of the fume cupboard. In normal working conditions the sash position higher than 33 cm was seldom used. At the lowest sash position, a slot of 2 cm remained between sash edge and cupboard table. The supply air flow was adjusted accordingly based on the exhaust air flow rate. In addition, selected fume cupboards were equipped with presence sensors. When the operator had left the sash open leading to too low face velocity, flashing light and sound alarms were launched.

The performance of the laboratory ventilation was monitored during two years in normal operation conditions including nights and holidays. The monitoring included supply and exhaust flow rates, sash position and presence of an operator in front of a laboratory hood. The sampling frequency of these data was once per minute. In addition, the set points of other operation parameters of the HVAC system were collected.

In order to analyse the huge amount of data collected, a database and a Java application for calculation purposes were created. This enabled versatile selections and filtering of data based on time periods of interest and other relevant parameters.

In order to prevent contaminant dispersion from the fume cupboard, it is necessary that the flow control system responses rapidly to changes in the sash position [3]. The response time of the sash sensing control system was tested in a laboratory room with three laboratory fume hoods by abruptly changing the sash position of one laboratory fume hood and recording the change of the joint exhaust air flow of fume hoods and supply air. The sampling frequency in recording in this test was 0.5 Hz.

RESULTS

The data series gathered by the building automation system was presented as cumulative distributions. The distributions enabled the consideration of parameters of interests including comparisons of the exhaust air flows of a fume hood at different time windows (Figure 1). The distributions in Figure 1 show differences in the exhaust flows of the fume hood analysed with different time windows over one year. Curve A represents the whole year including nights, weekends and holidays, Curve B working hours and C the time when a person was standing in front of the laboratory hood. From curves A and B we can see that about 80% of time the exhaust air was less than 50 l/s. In case of a person working with the fume hood, the corresponding value is 60 l/s. Lower part of curves A and B show that about 20% of time the exhaust flow was less than 40 l/s. This means that the operator had worked with a narrow sash opening or left the cupboard unintended with the sash slightly open. Generally, the curves show that the sash of the laboratory fume hood was left down or slightly open for most of the time. Curves A and B are close together but curve C representing presence by the hood showed higher air flows.

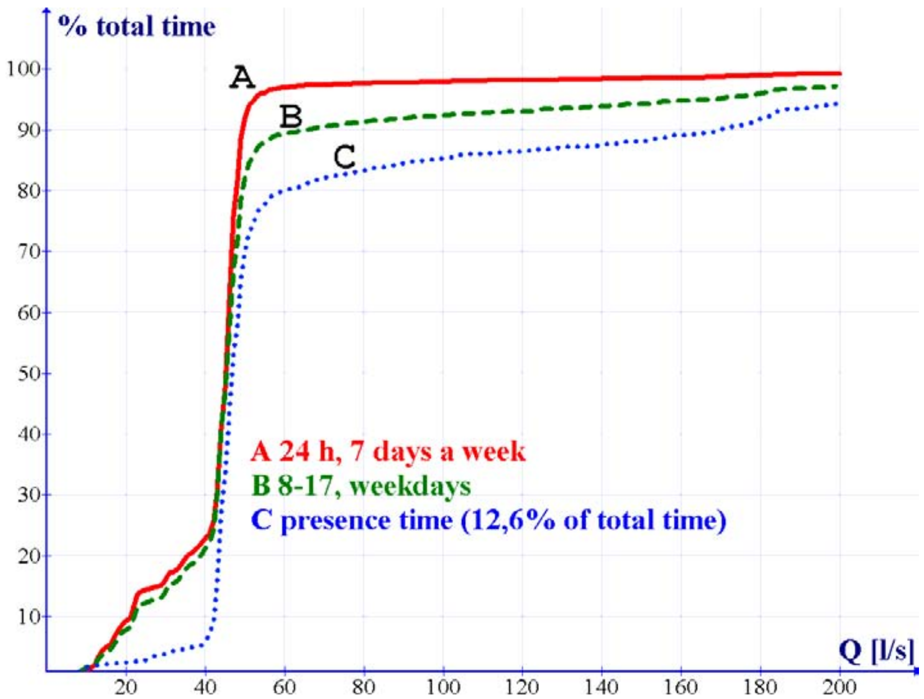

Figure 1. An example of the exhaust air flows of a laboratory fume hood in terms of cumulative distributions based on a monitoring period of one year.

The performance of the VAV fume cupboard system was compared to a hypothetic fume cupboard with constant air volume (CAV) providing the same face velocity as the VAV system (Figure 2). The saving potential of the VAV system was simply determined from the relation 1 - *A1/A2,* where A1 is the area restricted the cumulative air flow curve, y-axis and line of 100% (the VAV system) and A2 is a rectangular area (160 l/s x 100% i.e. the CAV system). Generally, it turned out that the saving potential with the VAV system ranged from 50% to 70%.

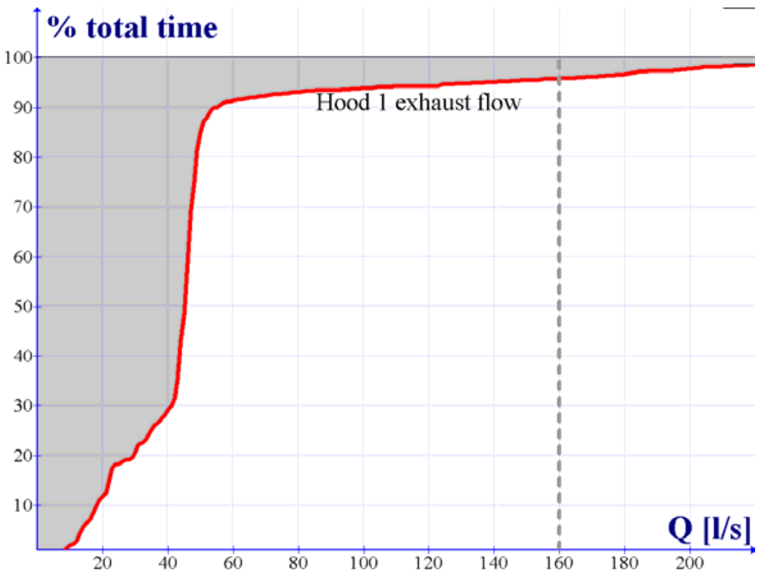

Figure 2. The idea for determination of the saving potential of the VAV system compared with the CAV system at the air flow of 160 l/s. In the case presented in Figure 2, the potential is 70%.

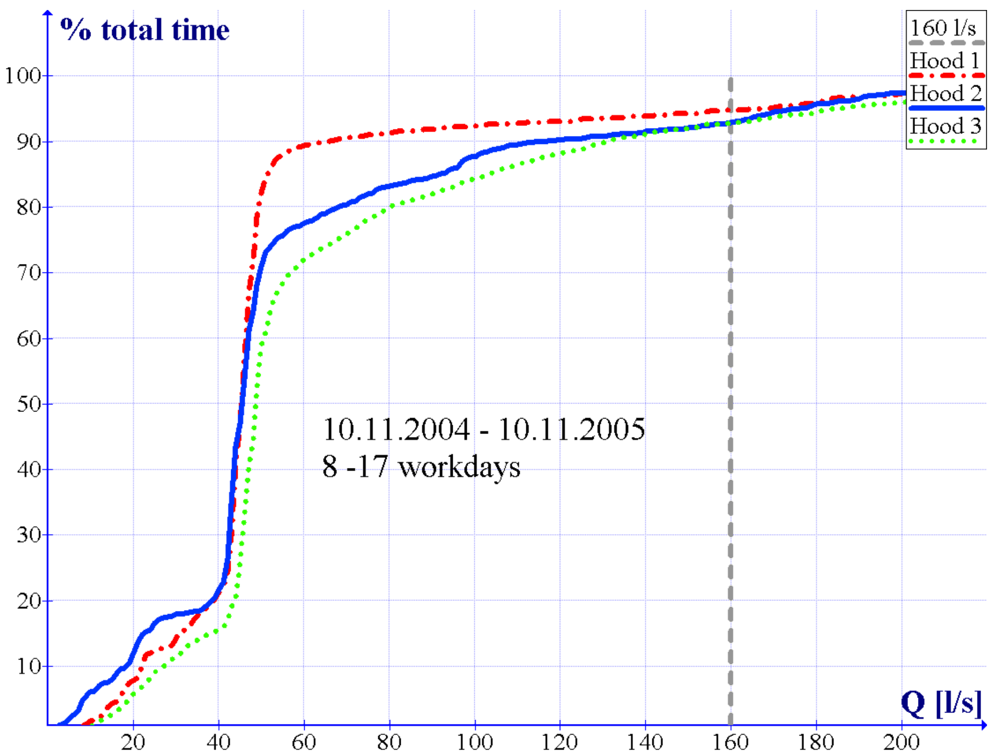

Figure 3. An example of cumulative exhaust air flows of three laboratory fume cupboards in a laboratory room monitored during working hours over one year. The saving potentials were as follows: Hood 1 67%; Hood 2 63%; and Hood 3 58%.

The curves in Figure 3 show that laboratory fume Hoods 2 and 3 were used more often than Hood 1. The saving potentials ranged from 58% to 67%.

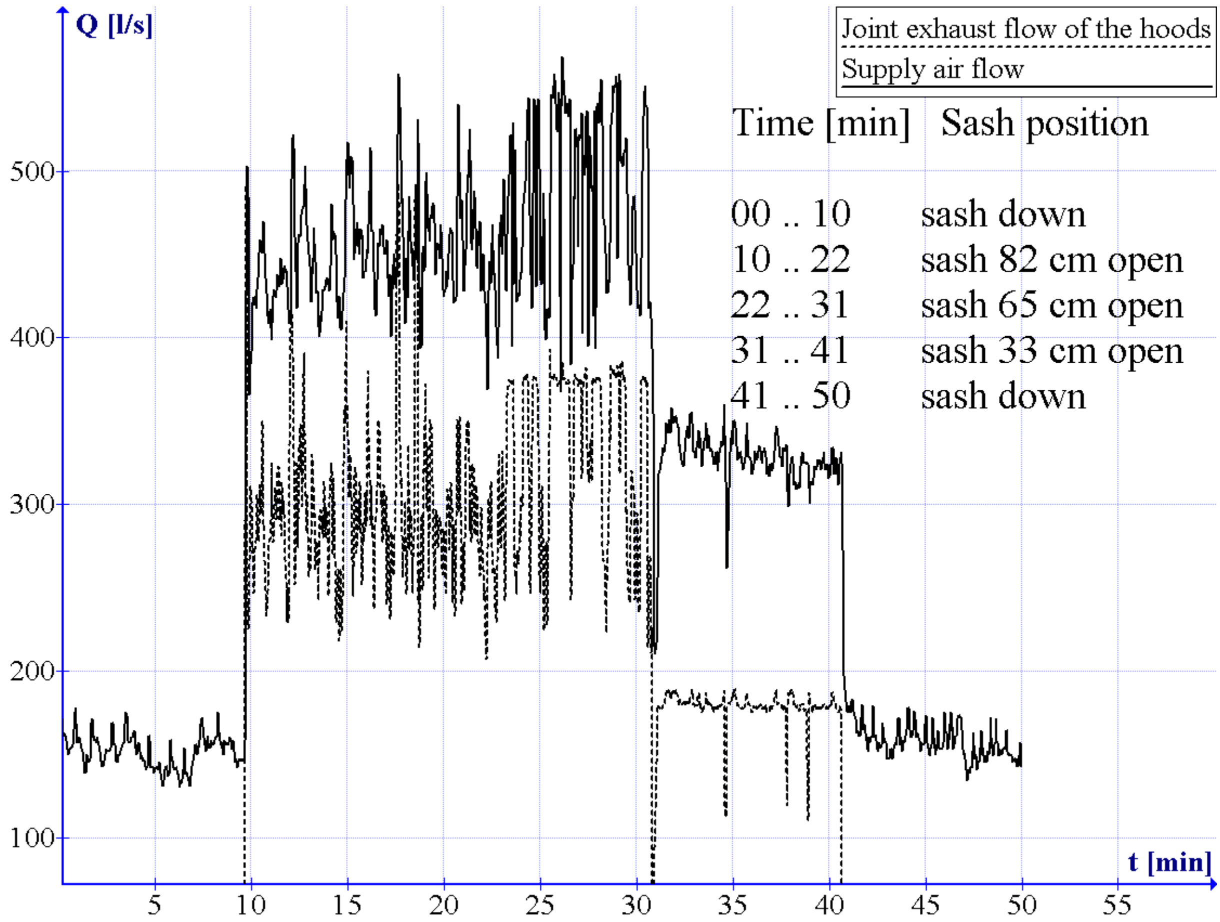

Figure 4. The dynamic behaviour of the VAV control system in a laboratory room.

The dynamic behaviour of the control system is described in Figure 4. It indicates that the exhaust flow of the hoods and the supply air flow into the room responded rapidly when the position of the sash was changed. In normal use the most common sash position is around 33 cm. The fume cupboard is fully open at sash position of 65 cm. The extreme position of 82 cm is not used. The test shows that the response time was less than three seconds, which can be regarded as short enough to prevent dispersion of air contaminants from the laboratory fume cupboard.

DISCUSSION

The paper demonstrates the benefits of long term monitoring of the BAS data of the laboratory building equipped with advanced laboratory fume hood technology and VAV ventilation. The two year monitoring produced information on how the laboratory HVAC systems were used in actual work. In order to analyze and present the gathered large data series a data base was created. The data base enabled consideration of air flows and other parameters of interest in terms of selected time windows. Cumulative distributions proved to be a simple and informative way to compress and present the data series.

Two 'knees' appeared in the typical cumulative distribution curves, one in the lower part and another in the upper part of the curve (see for instance Figure 3). The upper 'knee' revealed that higher airflows than 50 l/s were not used very often, only 20% to 30% of the working time. The lower 'knee' indicates that the exhaust air flow of the fume hoods was less than 40 l/s for some 20% of the working hours. This means that the sash position was low for most of the time. Consequently, one can conclude that the degree of use of the laboratory fume hoods was low in the demonstration building during the monitoring period. Yet, the conclusion is not so straightforward, as a laboratory fume cupboard may fulfill an important safety task without an operator if there is a contaminant releasing operation within the cupboard. It is also good to remember that it is typical of research laboratories that the number of chemical and biological analyses may vary heavily from one year to another depending on the research projects active at any given time.

The advanced hood technology with VAV ventilation proved to be very energy efficient compared to the CAV system. The saving potential introduced in this paper ranged from 50% to 70% measured against the energy consumption of the CAV system. These figures are in line with the results obtained in a study concerning the retrofitting of demand-controlled ventilation in a biochemical laboratory building [4].These figures strongly emphasize the benefits of advanced fume cupboard technology combined with the modern BAS system. Implementation of these technologies promotes energy economy without compromising the achievement of safety requirements.

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Personalized ventilation: impact of airflow direction at the breathing zone on inhaled air quality

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SUMMARY

Personalized ventilation aims for supplying clean air to the breathing zone of each occupant. The impact of the direction of the personalized flow on the inhaled air quality was studied. Experiments were performed in a full scale room. Personalized ventilation in conjunction with mixing ventilation was used. Breathing thermal manikin resembled person seated at a desk. Tracer gas, mixed with room air was used to simulate pollution. No significant difference in inhaled air quality was identified when the flow was supplied from front, left, right and above at distance between the personalized air supply device and the face of the manikin, 0.3, 0.4 and 0.5 m. This result was obtained at different levels of dilution of the personalized air with the polluted room air.

INTRODUCTION

Personalized ventilation aims for supplying clean air close to the breathing zone of each occupant [1]. Therefore when it is well designed it has potential for improving significantly the inhaled air quality [2, 3]. In this respect the interaction of the personalized airflow with the free convection flow which exists around human body is of major importance. The interaction of the flows and thus the quality of the inhaled air is affected by several parameters, such as the characteristics of the personalized flow, e.g. mean velocity, turbulence intensity and direction, body posture, room air temperature, etc. [1].

The impact of personalized flow rate, e.g. the target velocity of the personalized flow, has been studied [2, 3]. The impact of the direction of the personalized airflow has been studied only partly [4]. The purpose of this study was to identify the impact of direction of personalized airflow on inhaled air quality.

METHODS

Experimental facilities and conditions

Experiments were performed in a full-scale test room (5.4 m x 4.7 m x 2.6 m (H)) located in a large hall where temperature was kept the same as in the test room. Test room was ventilated by mixing ventilation and was equipped with desk with PV system. The ventilation air was supplied trough a swirl diffuser located in the middle of the ceiling. The room air was exhausted through air terminal device located at one of ceiling corners.

The PV system had a circular air terminal device named round movable panel. This air terminal device is designed to supply airflow at low turbulence intensity. The initial diameter of the supplied jet is 0.185 m. 100% personalized air in inhalation can be achieved in points on the jet axis at distance up to 0.5 m from the air terminal device when the supplied flow rate is above 10 L/s. The air terminal device is attached to a flexible arm which allows for changes of direction of the supplied personalized flow against occupant sitting at the desk. Detail description of the device is given in [3].

A breathing thermal manikin simulated a person sitting on a chair behind the desk. The manikin is shaped as an average-size female. The body surface temperature is kept the same as the skin temperature of average person in state of thermal neutrality. The manikin was dressed in a long-sleeve blouse, panties, sports trousers, short socks and light sports shoes. The thermal manikin was simulating human breathing by the use of artificial lungs. The breathing pattern simulated was the normal for a seated person: 2.5 seconds of inhalation, 2.5 seconds of exhalation and one second of pause, giving a breathing cycle length of 6 sec. The respiration rate was 6 l/min. Exhalation was through the nose of the manikin, and inhalation – through the mouth. Figure 1 shows the manikin position at the desk with PV terminal device.

Figure 1.Experimental set-up.

The total volume ventilation system supplied 60 L/s air well mixed with tracer gas (Freon 134A). The supply air temperature was controlled in order to keep constant room temperature of 23 °C.

The PV ATD was always positioned in a way that the mouth of the manikin was in the centre of the supply air jet. The personalized air was supplied at flow rate of 12.5 L/s and at room temperature. Experiments at different dilution rate of the personalized air with the polluted room air were performed. This is shown in Table 1.

Table 1. Dilution rate of clean personalized air and polluted room air.

Experiment Number	1	2	3	4	5	6
clean air / re-circulated air (L/s / L/s)	12.5/0	10/2.5	7.5/5	5/7.5	2.5/10	0/12.5

The impact of the direction of the personalized flow against the face of the manikin and the distance between the PV air terminal device and manikin's face on the inhaled air quality was tested. The tested combinations of distance and direction are listed in Table 2. Figure 2 helps to understand better the experimental conditions. In total 36 experiments at different conditions were performed.

Table 2. Set of tracer gas measurements performed on each of the 6 workstations

Measurement Number	Position of the RMP in relation to the face of the manikin	Distance between the RMP and the manikin's face
1	Perpendicular	0.3 m
2	Perpendicular	0.4 m
3	Perpendicular	0.5 m
4	45° to the left	0.4 m
5	45° to the right	0.4 m
6	45° above	0.4 m

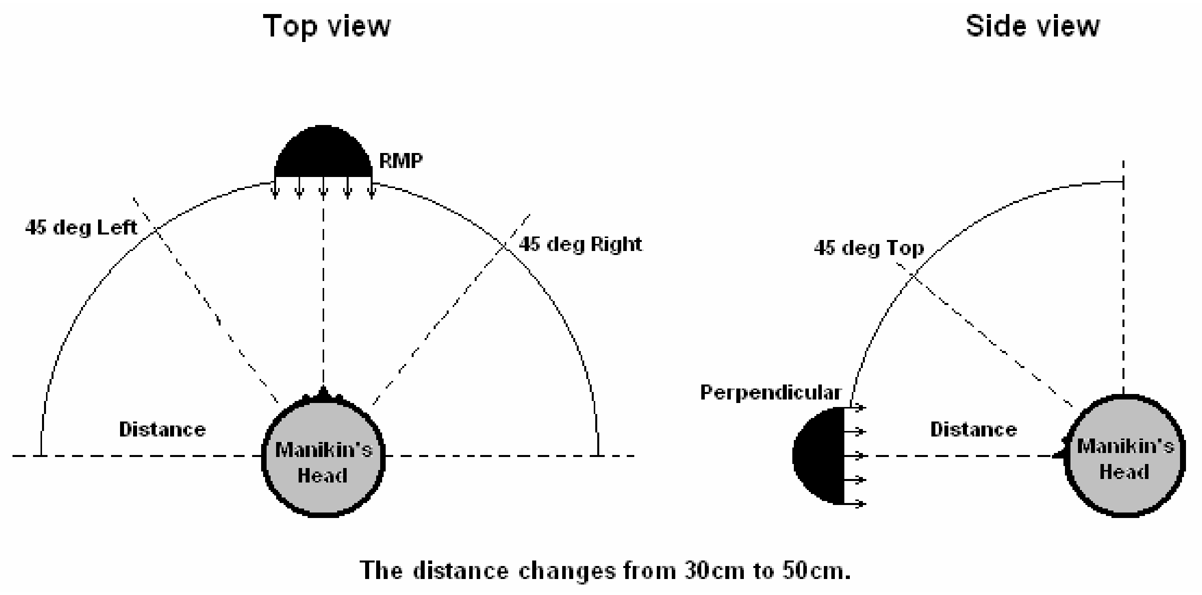

Figure 2. Positions of the RMP according to Manikin's face

Measurements and data analyses

During the experiments air temperature and humidity in the Tall Hall, air temperature and humidity in the test room, temperature of the air provided with PV was measured and controlled.

The tracer gas concentration in the outdoor air (referred as clean air), the exhaust air, in the middle of the room, at the PV supply, air inhaled by the manikin and outside the test room was measured. At each location concentration values from at least 8 consecutive measurements were obtained, and the averaged value was calculated. Photoacoustic Gas Analyser Innova 1302 was used to dose the tracer gas and to perform the concentration measurements. The measurements were performed after steady state conditions were reached.

The measured results were used to define the personal exposure effectiveness, PEE, by the following equations:

$$PEE = \frac{C_{I,O} - C_I}{C_{I,O} - C_{PV}} \qquad (2)$$

where:
$C_{I,O}$ – is the pollution (tracer gas) concentration in the inhaled air without PV;
C_I – is the pollution (tracer gas) concentration in the inhaled air with PV;
C_{PV} – is the pollution (tracer gas) concentration in the personalized air;

The personal exposure efficiency, PEE, is an index representing the percentage of clean air contained in the inhaled air. It is developed for evaluation of the performance of personalized ventilation systems [2].

In calculating the personal exposure effectiveness the tracer gas concentration in the exhaust air was used instead of the concentration in the inhaled air without personalized ventilation. The concentration in the inhaled air without PV is assumed to be the same as the concentration in the exhaust because mixing ventilation is used in the climate chamber.

RESULTS AND DISCUSSION

Figure 3 shows the PEE as a function of the distance between the supply air terminal device and the face of the manikin. The PEE obtained at different direction of the personalized flow against the face is shown in Figure 4. In the two figures the comparison is made at different dilution level. The results in Figure 3 show that the PEE decreased only slightly when the distance between the RMP and manikin's face was increased. This result was expected since the used air terminal device provides jet with relatively long potential core. The analyses of the results did not reveal significant impact of the personalised flow direction on the PEE. The results shown in Figure 4 are somehow scattered but without any tendency for an impact of the flow direction. The results in Figures 3 and 4 reveal that the PEE decreases with the increase of the dilution ratio. This result was expected.

The obtained results are important for analyses of perceived air quality reported by people provided with control of direction of personalized flow.

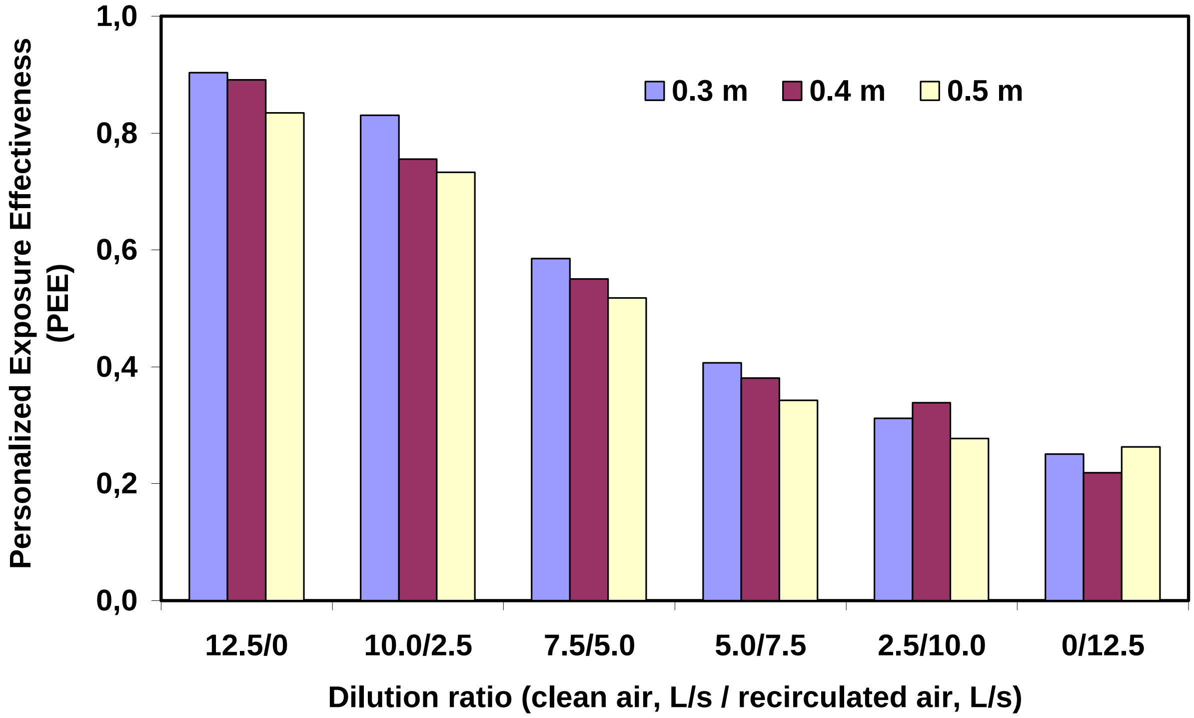

Figure 3. *PEE* as a function of the distance between the thermal manikin's face and the RMP positioned in front of the manikin's face.

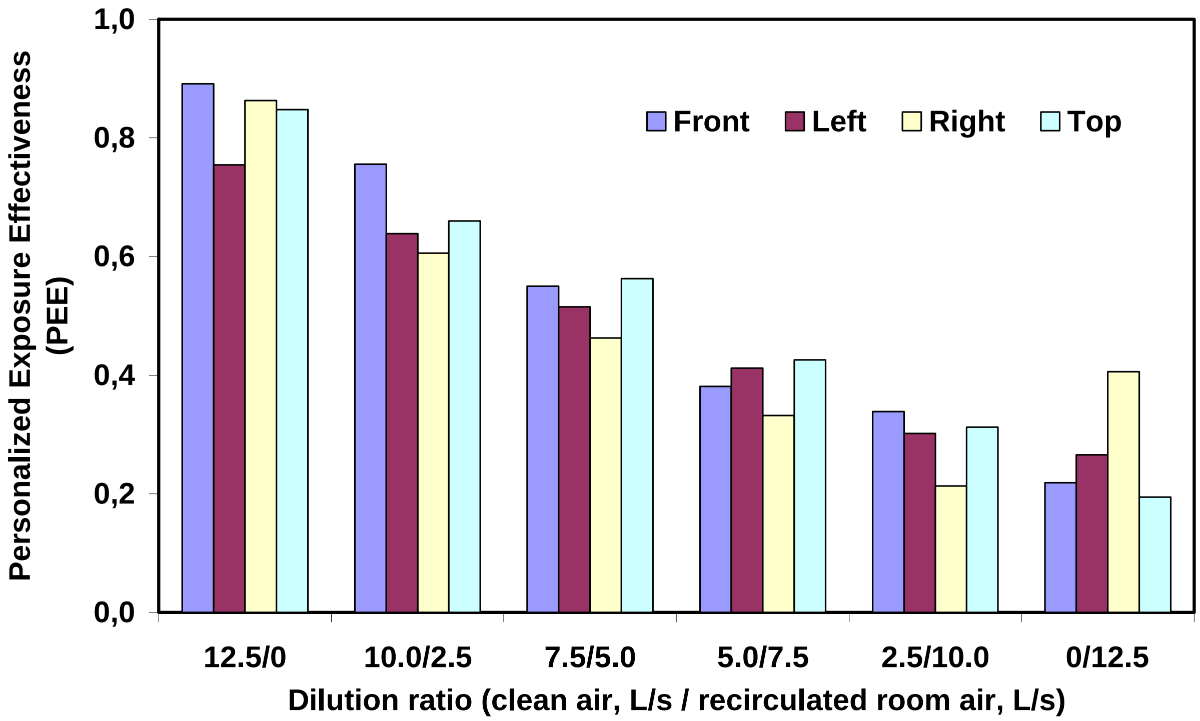

Figure 4. *PEE* as a function of the direction of the RMP according to the manikin's face. Distance between the RMP and the manikin's face is 0.4 m.

CONCLUSION

The results of this study reveal that the inhaled air quality measured by thermal manikin did not change when the personalized flow from a circular air terminal device (RMP) was supplied from front, left, right and above.

ACKNOWLEDGEMENT

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Cool the Office With Moving Air

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SUMMARY

Personal control over moving air will eliminate the number one complaint in the office environment – thermal discomfort. The use of moving air to cool provides the ability to accommodate different needs among people for comfort due to varying metabolism, efficiency of heat rejection, and clothing. Cooling with moving air saves substantial energy through increased ventilation effectiveness and higher operating temperatures. Increased comfort means increased productivity. Workstation design can be adapted to provide both personal air control and displacement ventilation at reduced first cost with faster construction, lower operating costs, cleaner air and 100% personal satisfaction.

DISCUSSION

The number one complaint of office occupants is an uncomfortable environment. People can, however, adjust conditions to meet their personal need for thermal comfort by simply adjusting air flow from a local air supply at their workstations. A study by Khedari and others [1] showed the dramatic effect of moving air on people. Also a survey by the University of California at Berkeley showed that most people in an office setting preferred more moving air.

The best use of the cooling effect of moving air is to get away from the mind set of air conditioning buildings with a “one environment fits all” approach that fits no one. The aircraft and automotive industries have taken advantage of moving air controlled by occupants for many, many years. Unfortunately, because of lack of space, the velocity of the air in both cases is high and tends to feel drafty.

Tests by Fred Bauman and his group at UC Berkeley have demonstrated that changing the air flow can change the cooling temperature perceived by occupants from 0° to 15°F (0° to 9°C). In the Khedari study of 288 college students, air at 80°F (27°C) moving at 400 FPM (2 m/s) feels like 65°F (18°C), a difference of 15°F (9°C).

ASHRAE Standard 55 through 2004 (p. 6) suggests the use of moving air, provided it’s under personal control. Khedari found that the effect shown in Standard 55 and reflected here (Fig. 1) as ASHRAE Thermal Comfort Tool, or TCT, is understated and probably is based on sensible cooling only. So, by varying the air flow you can control the perceived temperature from 80°F to 65°F (27° to 18°C). That range should satisfy 100% of occupants at all times. Providing personal control will eliminate the number one complaint – thermal discomfort – and improve productivity in every office. The system does this by accommodating the vastly different needs among people for comfort. Why is there so much difference in people's needs? Because basic metabolism varies from person to person (Fig. 2).

On top of that, metabolism can vary at any time due to varying conditions, such as controversial phone calls; or being rushed; or not feeling well due to health problems, etc.
People are comfortable when the heat removed is equal to the heat they generate and absorb from external sources, such as the sun, lights, office machines, etc.

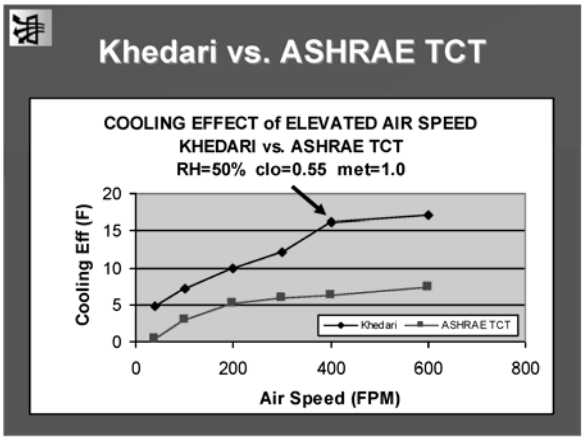

Figure 1.

According to information from the Mayo Clinic, the basal metabolic rate for people is 10-12 kilocalories per pound (22-26 kilocalories per kilogram) of body weight per day. Fig. 3 compares the caloric output of 1360 kilocalories for a typical woman weighing 125 pounds (54 kg) to that of 1960 kilocalories for a typical man weighing 180 pounds (86 kg): a difference of 600 kilocalories, or <u>44%</u>!

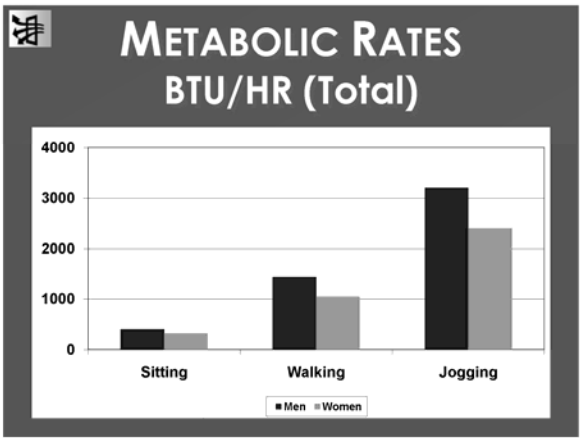

Figure 2.

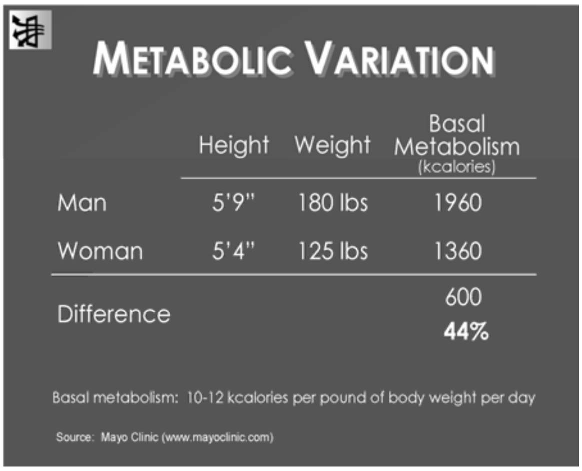

Figure 3.

If more heat than is generated through basic metabolism is removed, the person feels cold. Any less and he feels hot.

<u>Is it any wonder that both people cannot be comfortable under the same temperature conditions?</u>

In addition, differences in efficiency of heat rejection and clothing affect the cooling needs of men vs. women. A smaller body is a more efficient radiator, further increasing the need for individual environmental control. The simple solution is the system shown in Fig. 4.

By introducing air at the desk level, the occupant can change the cooling effect with a damper - just like in his car. Because of better space availability compared to a car, velocities are lower, and, therefore, air flow is quieter and more comfortable. This arrangement is familiar to everyone from his or her car without special training!

The system also saves substantial amounts of energy.

Tests at UC Berkeley have shown that discharging supply air flow near the occupant increases ventilation effectiveness - a savings both in energy and the cost of filtration (Fig. 5).

Figure 4.

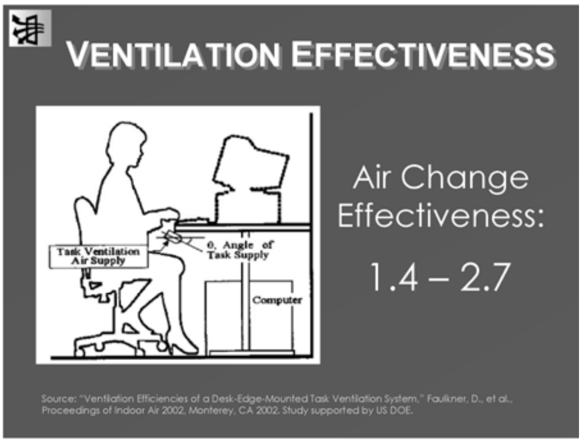

Figure 5.

Furthermore, a very great savings in energy is achieved by using moving air instead of cold air. Room temperatures can be set substantially higher because of the cooling effect of the moving air. As we have seen, moving air lets people set "perceived temperatures" from 80° down to 65°F (27° to 18°C) – a range of 15°F (9°C). The thermograph in Fig. 6 shows 60% of heat is given off by the upper body, and, thus, a personal air terminal positioned at the worksurface is most effective in making people comfortable with the least expenditure of energy. Many studies show that increased comfort means increased productivity [2]. Now let's put productivity in perspective.

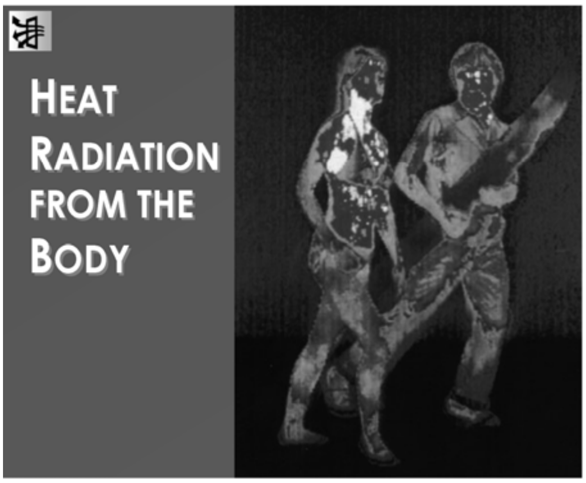

Figure 6.

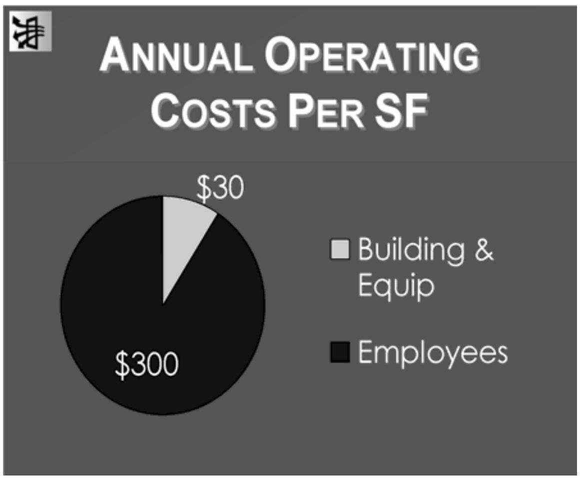

Figure 7.

Annual payroll costs in office buildings are approximately $200 to $300 per square foot ($2,150 to $3,225 per square meter), or 90% of the total costs of owning and operating an office facility. Even a small increase in productivity can offset total annual building costs of $20 to $30 per square foot ($215 to $323 per square meter).

Fig. 8 illustrates a desk or workstation with a personal air terminal. Note that displacement ventilation can easily be added for better IEQ, cleaner air and reduced cross contamination.

Displacement ventilation has been proven to increase comfort, save energy and reduce absenteeism.[3]

Fig. 9 shows a typical office with integrated task/ambient personal control and displacement ventilation. Note the near absence of floor grilles.

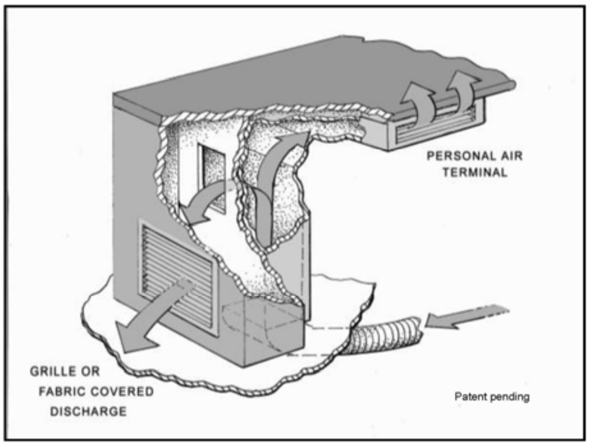

Figure 8.

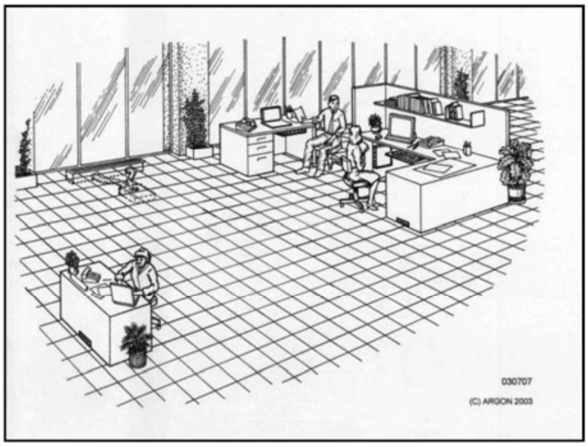

Figure 9.

Moving air with task/ambient personal control and displacement ventilation can eliminate the #1 complaint of office workers, save energy, have cleaner air and increase productivity. LEED credits can be earned with a sustainable green indoor environment, control by occupants, better IEQ, and lower energy cost.

All this can be done at reduced first cost.

Faster construction, lower operating costs, cleaner air and 100% personal satisfaction are the rewards.

Figure 10.

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Comfort Zone Or Acceptable Comfort Zone? : Comparison of Resident' Behavior of Operating Air Conditioner According to Charge for Energy

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SUMMARY

This research compared two groups' operation behavior of cooler, one is the resident of studio apartment who pays their energy charge per month, and the other is the resident of university dormitory who doesn't pay energy charge beside their rental fee. The amount of time to use a cooler and room temperatures were measured when they turn on/off air conditioner. Residents were interviewed about their cooling needs and decision factor on their operation behavior. Main results are as follows.

1. Dormitory group used the air-conditioner twice as much as studio apartment group.
2. Dormitory group turned on/off air-conditioner at lower temperature compare to the studio apartment group.
3. Studio apartment group's acceptable comfort zone was in or higher than ASHRAE summer comfort zone, but dormitory group's that is in or lower than ASHARAE comfort zone.

INTRODUCTION

Building systems have been developed and complicated, and it follows the energy which is consumed for the building is extending too. In a life cycle of the building, most using much energy is a dwelling phase. Big effects to these energy consumptions in a dwelling phase are the behavior and attitude of residents, and energy charge affects to these residents' behavior and attitude. Consequently, we can assume that there would be a big difference between the residents who pay for the energy or not; house and office. Meanwhile, the electric consumption in Korea was increased 4 times much as 20 years before. Big parts of these residential energy consumptions are consent, lighting and cooling. Especially, cooling demand is expected to increase fast. On this background, this research compared two groups' operation behavior of cooler according to charge for energy.

METHODS

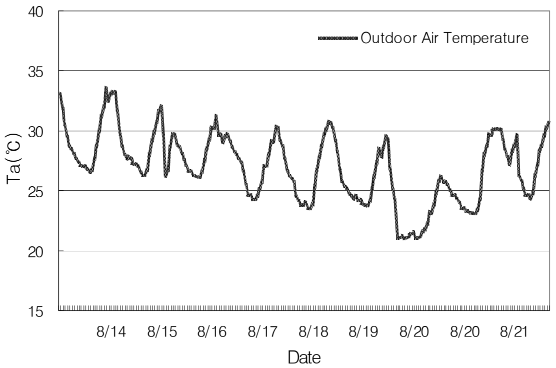
Thermal environment and using time of air conditioner are measured in seven studio-apartments and eight rooms in university dormitory where air conditioners were located. Measurement was conducted during 9 days from August 13th to 22nd in 2006. Subjects were selected to twenties who can operate air conditioner well and were limited to independent resident for restricting effects from other generations. We do an experiment in two groups' operation behavior of cooler; one is the residents who pay their energy charge per month, and the other is the residents of university dormitory who doesn't pay energy charge beside their rental fee. This is for finding out whether charging for energy has effect on using time of air conditioner. Air temperature and relative humidity were measured at 110cm high from the floor and air temperature blowing from air conditioner was also measured to detect the condition of on/off of air conditioner. Air temperature, relative humidity, and air temperature blowing from the air conditioner are measured with RH-temperature data logger.

Interviews were also conducted to investigate characteristics of residents' behaviors of operating air conditioner in addition to physical environment; time of staying at home, a way they operate air conditioner, a purpose that they use air conditioner, their Clo value, sex, age, kinds of AC, and spending electric power.

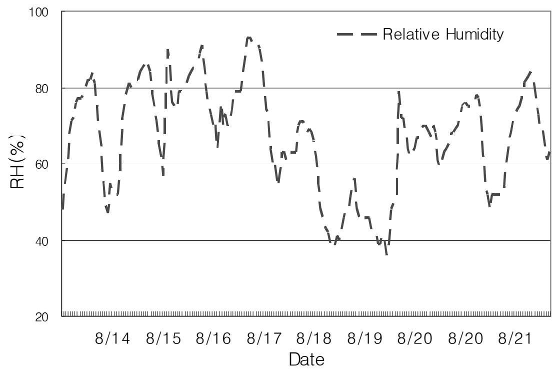
RESULTS

Outdoor temperature and relative humidity

Outdoor weather data was obtained form Korea Meteorological Administration. The mean outdoor temperature was 26.93℃ and relative humidity was 67% during experiment period.



a) Outdoor air temperature



b) Relative humidity

Figure 1. Climate during experiment periods

Comparison on the amount of time to use an air conditioner between two groups

Table1 shows that the results of the time used with air conditioner for two groups. The dormitory group used air conditioner twice as much as the studio-apartment group: Dormitory group spend 43hours 55minutes and studio-apartment group spend 23hours 21minutes in the experiment period. All of the residents in two groups were university students or graduate students. Although considering their life style was similar, we calculated index of amount of time to use an air conditioner to more accurate analysis. The method of calculating of index was that amount of time to use an air conditioner divided by mean time for which residents stayed at home. As the result, index of dormitory (0.4) was higher than that of studio apartment (0.17). Even though, we excluded A-6 and A-7 that never used air conditioner during experiment period from a calculating, index of studio apartment became 0.24 to be still lower than dormitory. It was one of the reasons that residents even maintained running air conditioner when they went out for thermal comfort to come back. While, when going out, residents of studio apartment always turned off their air conditioner.

A-3 of studio apartment group was the highest both time of use and index of time, because this one stayed at home by far than others. Most of studio apartment except A-3 was lower than dormitory in index of time. Also, despite of high temperature of outdoor, A-6 and A-7 never turned on air conditioner because of charge for energy.

Table 1. Operating time of air conditioner.

Studio-apartment			Dormitory		
Resident	Operating time (hour, minute)	Index of operating time	Resident	Operating time (hour, minute)	Index of operating time
A-1	29h 8m	0.29	B-1	49h 24m	0.50
A-2	10h 36m	0.10	B-2	45h 58m	0.46
A-3	64h 4m	0.44	B-3	73h	0.50
A-4	40h 40m	0.28	B-4	54h 52m	0.38
A-5	18h 8m	0.08	B-5	37h	0.37
A-6	0m	0	B-6	31h 32m	0.31
A-7	0m	0	B-7	33h 20m	0.37
			B-8	26h 16m	0.27
Avg.	23h 21m	0.17	Avg.	43h 55m	0.40

* Index of operating time = Operating time of air conditioner / inhabitant time

Indoor Thermal Environment when residents operate air conditioner

There was a gap between two groups about mean temperature at the point time that air conditioner turned on or off. Table 2 shows that indoor air temperature fight at the point time that residents turn on air conditioner and turn them off. It appeared that residents of the studio apartment start air conditioner at 29.9℃ of air temperature, and switch off at 27.0℃. While, residents of the dormitory turn on air conditioner at 27.0℃, and turn off at 23.7℃. It means that residents of studio apartment turn on and off at higher temperature than residents of dormitory.

Like above these results, we know that the dormitory group not to charge for energy used air conditioner in lower temperature than the studio apartment group to charge for energy.

Table 2. Mean indoor air temperature when AC was turned on or off.

Studio-apartment			Dormitory		
Resident	On(℃)	Off(℃)	Resident	On(℃)	Off(℃)
A-1	31.0	28.8	B-1	29.2	24.3
A-2	31.3	28.9	B-2	26.2	23.2
A-3	29.1	26.3	B-3	28.4	24.5
A-4	29.1	26.5	B-4	26.1	23.4
A-5	29.9	27.5	B-5	26.7	24.3
A-6	Never used		B-6	26.7	23.5
A-7	Never used		B-7	26.3	23.5
			B-8	26.5	21.1
Avg.	29.9	27.0	Avg.	26.9	23.7

Interview of resident' cooling behavior

Interview was conducted to help understanding about residents' behavior of operating air conditioner. When asked whether residents still remain air conditioner turn on for going out, more than 50 percent of the dormitory residents blamed 'yes', but all of the studio apartment residents answered 'no'. Setting temperature of air conditioner in both groups was similar, 20℃. They controlled indoor temperature by switch on and off, not the setting temperature. The main reason why they turn on air conditioner is high temperature, the other is to exsiccate. Also, it seems that their habituation of operating air conditioner is only to turn on or off cooler than to regulate. When they ho to sleeping, all in both groups stopped cooler and opened windows, but only one in the dormitory remained cooler running beside opening windows. Indoor mean Clo value of the studio apartment group was higher rather than the dormitory group: the studio apartment group was 0.15 clo and the dormitory group was 0.24 clo. When asked "How much could you pay for using air conditioner?", they answered each 34 dollars on the dormitory and 70 dollars on the studio apartment. It means that residents of the studio apartment who pay for energy have more recognition than them of the dormitory.

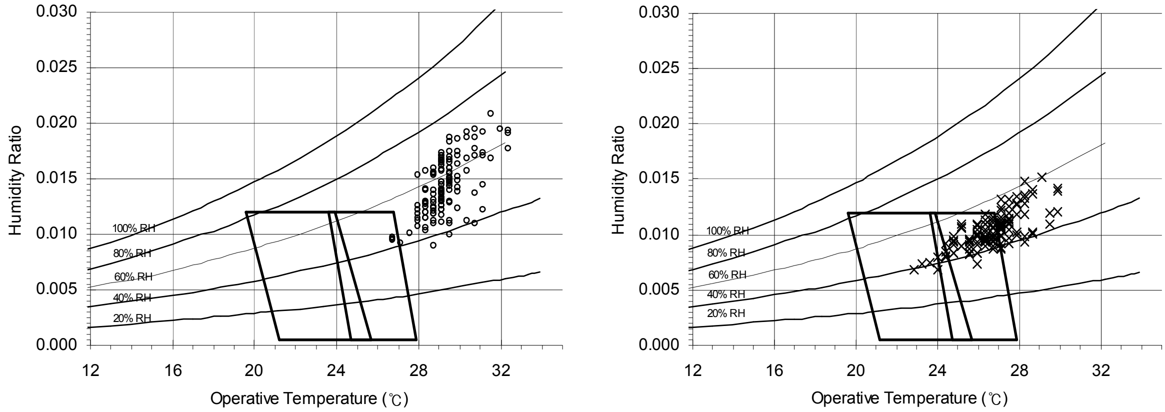
Comparison with a previous research

We compared this result with previous research, which was studied from Nu-Ri Bae for finding out resident' operating behavior of air conditioner according family members as well as charge for energy.[1] The previous research was conducted during 60days from July 3rd to August 31st in 2004 in six family apartments' living rooms there air conditioner were located. Indoor thermal environment was measured and air temperature blowing from air conditioner was also measured to find out the condition of on/off of air conditioner. Mean outdoor temperature was 25.2℃ and relative humidity was 76.5% in experiment period.

1. Thermal environment of three groups

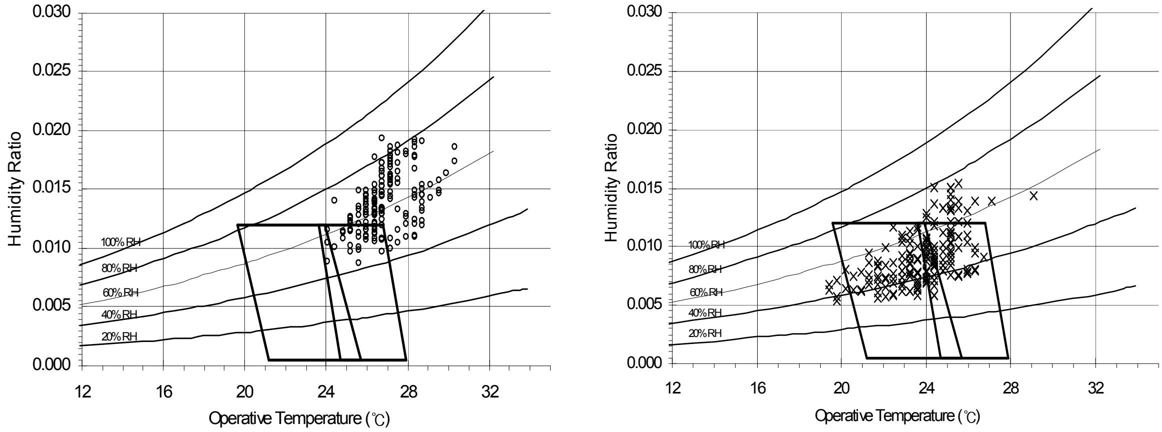
In Figure 2~ 4 thermal environments at the point time air conditioner was started and was stopped in three groups compared with thermal comfort range of ASHRAE Standard 55. [2]
First of all, compared with thermal environment according charge for energy, three groups have a clear distinction. Figure 2 shows that the result of studio apartment group. When residents started air conditioner, most cases of thermal environment were on very high temperature that is not contained ASHRAE comfort zone and when air conditioner was stopped, some cases was covered or accessed ASHRAE comfort zone of summer. Also, Figure 4 indicated that family apartment was as same as studio apartment but was rather high range. An air conditioner was started and switched off at thermal range that was not included ASHRAE comfort zone.

On the other hand, Figure 3 displayed that residents' behavior of the dormitory was different from other two groups. They turned on air conditioner at air temperature whose one third already was included on summer comfort zone of ASHRAE and turned off it at air temperature whose most was contained on lower air temperature, winter comfort zone, as well as summer comfort zone of ASHRAE. It means that range of thermal environment which residents controlled was significantly different according charge for energy. Judging from air temperature, residents can use air conditioner on very low air temperature in case of no charging for energy. From interview, there are some responses that residents still keep cooler in running until feeling cold and wear more clothes instead of stopping cooler.



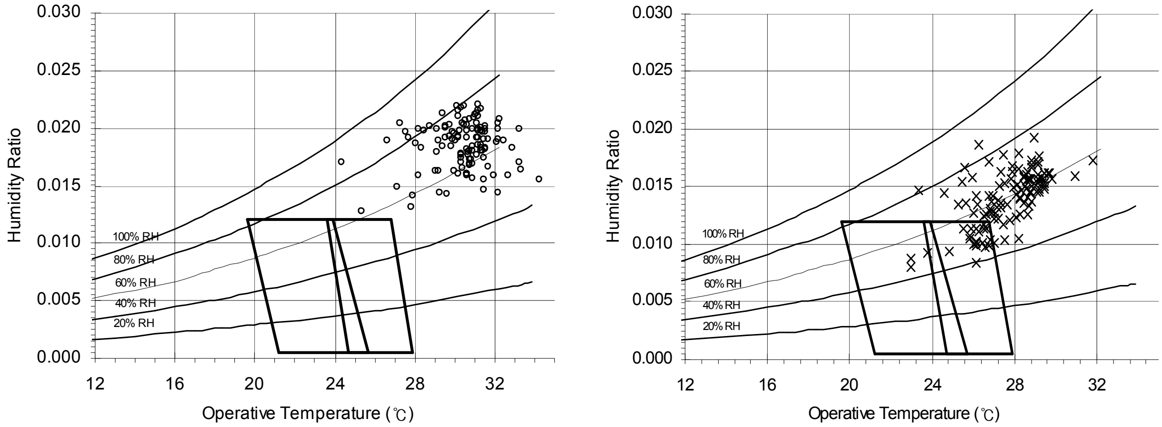
a) Thermal range when AC was turned on b) Thermal range when AC was turned off

Figure 2. Indoor thermal range on ASHRAE comfort zone when AC operated in studio apartment



a) Thermal range when AC was turned on b) Thermal range when AC was turned off

Figure 3. Indoor thermal range on ASHRAE comfort zone when AC operated in dormitory



a) Thermal range when AC was turned on b) Thermal range when AC was turned off

Figure 4. Indoor thermal range on ASHRAE comfort zone when AC operated in family apartment

2. Prediction of time to use air conditioner

What effect will outdoor weather factors including mean temperature, highest temperature, lowest temperature, mean relative humidity and highest wind velocity have on time to use air conditioner? The relationship between outdoor weather factor and using time was calculated by linear regression model. As a result, the relationship with both mean temperature and highest temperature is the strongest in the family apartment, the relationship with lowest temperature is the strongest in the dormitory and the relationship with both highest temperature and lowest temperature is the strongest in the studio apartment. It is appeared that residents of the family apartment start air conditioner when outdoor temperatures getting hotter but residents of the dormitory turn off it when weather become colder. In other words, residents in family apartment don't use air conditioner usually and then start it when air temperature reach the maximum, but the residents in dormitory use air conditioner always and then they switch off it when air temperature go down.

Table 3. Correlation coefficient between outdoor weather and time of using AC

Weather factor	Studio apartment	Dormitory	Family apartment
	R	R	R
Mean outdoor air temperature	0.33	0.34	0.36
Highest outdoor air temperature	0.32	0.34	0.35
Lowest outdoor air temperature	0.33	0.40	0.26
Mean relative humidity	0.19	0.27	0.34
Highest wind velocity	1E-06	0.02	0.11

Using time of air conditioner of each group was predicted through regression model with mean outdoor temperature that is related in all groups comparatively high. 26℃ and 30℃ of outdoor air temperature were applied to regression model, and mean using time of each group was calculated. Showing fig. 6, the dormitory has each value of 319 minutes and 540minutes, the studio apartment has each value of 222minutes and 364minutes and the family apartment has value of 119mimutes and 202minutes. From this graph, we can see the order of using time of air conditioner in same outdoor temperature is Dormitory, Studio apartment, and Family apartment. And an angle of inclination by temperature increasing shows same order.
The reason why residents of the family apartment use less time compared with studio apartment could be from generation gap about energy saving and charging for energy, different thermal sensation between ages, and housewife' economical conception, etc.

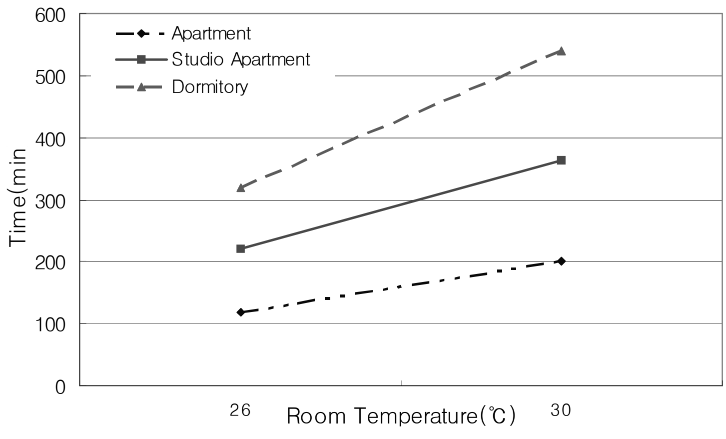


Figure 5. Predicted AC using time by outdoor air temperature

DISCUSSION

When people turn off air conditioner, which means room air temperature became cool enough to them. This enough cool temperature was different between three groups. Of course, it is because of economical factor. Thermal comfort zone like ASHRAE Std.55 is based on sensory comfort in climate chamber. It was calculated from experiment results in climate chamber. Subjects didn't get the influence from economical factor, and tried to find out "just comfortable" temperature.
If we call this "just comfortable" temperature as "sensory comfort zone", we can call acceptable comfort zone in this study as "cognitive comfort zone". Sensory comfort zone is related on human sensory system – physiological & physical characteristics. Cognitive comfort gets the influence from many factors – economical situation, thermal history, adaptation state, etc. These differences made the difference between many chamber study results and field study results. In ordinary, field study results show wider acceptable range than chamber study results. This tendency respected to the revise of ASHRAE Std. 55 (2004).
Both of them – Sensory & Cognitive Comfort – are necessary and worth. However, broad acceptable range may be desirable in point of human thermal adaptation and environment of the earth.

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C04
Calculation of energy performance and implementation of EPBD

HVAC System Efficiencies for EPBD Calculations

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SUMMARY

Whole-building energy performance calculations are required by the EPBD as part of compliance with minimum standards for new buildings and of the energy performance certification process for existing buildings. These are integrated calculations that include the effects of the building design and of the installed or proposed HVAC and lighting systems. The performance of HVAC – and especially air-conditioning - systems can vary very widely and it is therefore important to estimate system efficiencies realistically. On the other hand, the calculation process itself should not be unduly onerous. This paper explains how these contradictory pressures have been dealt with in the development process of the HVAC component UK compliance tool for non-domestic buildings, SBEM (Simplified Building Energy Model) and summarises its main features..

INTRODUCTION

Energy-related building codes for new construction have traditionally focussed on building envelope issues and, in many countries, only on heating energy use. In recent years, however, there have been substantial increases in demand for electricity for lighting and air-conditioning, especially in commercial buildings. At the same time climate change and carbon emissions from all fuels have become a focus of energy and environmental policy. As a result this traditional emphasis on heating and insulation has come to look incomplete.

This is reflected in the requirements of the Energy Performance of Buildings Directive (EPBD), which calls for the use of integrated energy calculations that take account of all energy sources and all building-related end-uses of energy. These integrated calculation methods are to be used as the basis of building codes for new construction throughout Europe, though Member States have some flexibility to define the calculation process. In many countries the same calculation method will be used for the Energy Performance Certification of buildings when they are constructed, sold or let.

Crucially, these integrated calculations have to reflect system efficiencies as well as envelope performance. There are many alternative types of HVAC system of very different efficiency and so the impact of system selection and specification on carbon emissions can be as large as that from building envelope decisions, especially for new buildings. This is especially so for systems that provide cooling or ventilation in addition to heating. Increasing insulation requirements and moves to impose minimum efficiencies for boilers and other heat generators have already reduced heating demands, more of which are met by internal heat gains. Since the cost-effectiveness of increasing insulation thicknesses is subject to the law of diminishing returns, the economic benefits of further improvements are now relatively small.

Thus the realistic representation of HVAC systems in these calculation tools is of substantial importance. It is far from clear how best to do this as there are somewhat conflicting requirements for simplicity and consistency of use on the one hand, and technical robustness on the other. This paper describes the approach that has been adopted in the UK Simplified Building Energy Model (SBEM) and places this in the context of developing practices in other Member States.

OBJECTIVES

Calculation procedures for EPBD applications should ideally satisfy a number of somewhat contradictory requirements:

- Technical Credibility
    - Technically sound calculation process
    - Produce realistic results
- Discrimination
    - More efficient systems should give better figures
- Repeatability
    - Different users should get the same results
- Transparency
    - Both the data and the process should be auditable
- Ease of use and manageable data requirements
    - To reduce errors and cost

In practice it is difficult to satisfy these simultaneously and it is necessary to balance the competing requirements against each other. This balance is likely to be different for different applications - for example, the relative importance for design purposes may be different from that for Energy Performance Certification.

PRINCIPLES, OPTIONS AND EUROPEAN STANDARDS

The calculation of HVAC system efficiency can be addressed at several levels of complexity. At one extreme, the simplest methods may miss out or over-simplify important factors. On the other hand, complex methods such as detailed system modelling can be very data-intensive and time-consuming to apply. Poor data quality may introduce greater uncertainty than is associated with the modelling itself. [1] In existing buildings obtaining precise information is time consuming, costly and sometimes impossible. Absolute precision is not the most important consideration, especially if it compromises the other attributes such as repeatability. Given the uncertainties of data, of calculation and of the way that buildings are operated, any decision that is driven solely by small apparent differences in performance is fragile.

The important outcomes are the actions that result. For new buildings these are the design decisions that designers are encouraged to consider. For existing buildings they are: the selection (or at least consideration) of possible improvements and decisions on the selection from a range of options of buildings available to purchase or rent.

The calculation of heating and cooling *demands* of spaces and the associated energy *consumption* of HVAC systems that satisfy these demands are obviously closely linked processes. However, the two calculation processes have somewhat different constraints.

Heating and cooling demands can be adequately calculated at monthly intervals, for example by using the monthly heat balance and utilisation factor approach of EN13790[2]. Hourly demand calculations are used as design tools but regulatory applications are unusual in Europe. The EN13790 monthly procedure (now adapted to include sensible cooling as well as heating) is becoming the most widespread approach. [•]

For most HVAC systems, calculations using hourly (or shorter) time-steps are more natural, since this interval is better aligned to the inherent time constants of the systems. Hourly calculations can

[•] EN13790 also permits the use of hourly calculations and includes a simplified hourly procedure.

either be full system simulations or simpler “bin methods”. For some types of system – especially those providing only heating – an annual seasonal efficiency can be adequate. However, even here, the determination of annual seasonal efficiencies can become cumbersome for complex systems, such as those with bivalent heating. While the results from hourly demand calculations could be easily aggregated into monthly figures for a monthly system calculation the reverse procedure is much more difficult.

Definitions

The definition of “system efficiency” for HVAC systems is less straightforward than appears at first sight, because of the difficulty of attributing energy for fans, pumps and controls to the different end-uses (heating, cooling, ventilation). The EPBD standards resolve this by separating the energy associated with these, mainly transport, components from the losses associated with the generation of heating or cooling from fuels or electricity. The energy associated with fans and pumps (and controls) is treated as a separate item denoted as “*auxiliary energy*”. The consequent definitions for system heating and cooling efficiency then become more straightforward - but are now different from the more familiar meanings that include the auxiliary energy.

“Auxiliary Energy”: is the energy used by the fans pumps and controls of a system, irrespective of whether this supports heating, cooling or ventilation.

For heating, the “System Coefficient of Performance”, SCoP, is the ratio of the total heating demand in spaces served by an HVAC system divided by the energy input into the heat generator(s) - typically boilers. It takes account, for example, the efficiency of the heat generator, thermal losses from pipework and ductwork, and duct leakage. It does not include energy used by fans and pumps

For cooling the “System Energy Efficiency Ratio” SEER: is the ratio of the total cooling demand in spaces served by a system divided by the energy input into the cold generator(s) - typically chillers. It takes account of, for example, the efficiency of the cold generator, thermal gains to pipework and ductwork, and duct leakage. It does not include energy used by fans and pumps. Since many cooling demand calculations only estimate sensible cooling, the definition may be extended to include allowances for deliberate or inadvertent latent loads.

prEN15243

prEN15243[3] is the draft standard that deals with the calculation of HVAC system efficiencies. It contains a number of informative annexes that illustrate different approaches, but it does not prescribe specific calculation procedures. It permits HVAC system performance to be calculated either monthly or hourly. The draft standard provides a framework that can be used to assess whether particular methods are appropriate to particular circumstances.

The standard identifies nearly 40 mechanisms that can affect the relationship between the cooling demand of a building and the energy used by an HVAC system in meeting that demand. These are mapped against 20 or so types of HVAC system to show which mechanisms may apply to which system types. It then requires any calculation procedure to declare which system types it claims to cover, and how it addresses each of the applicable mechanisms. It does not prescribe how each mechanism should be handled (although there are “informative” suggestions). So, for example, the effect of heat gains to cold ducts could be calculated in detail or included as a fixed allowance (or ignored). Different degrees of sophistication may be appropriate to different situations. The draft standard simply demands that the user should know and thus be in a better position to judge whether a particular procedure is appropriate to his needs.

Table 1. Overview of mechanisms identified in prEN15243

10 within-room mechanisms
9 mechanisms relating to terminals
1 mechanism covering auxiliary energy
7 mechanisms relating to other distribution features
9 mechanisms relating to heat and cold generation

Table 2. System types identified in prEN15243

Code	System name	Does not always include heating provision and may be used with separate heating system (including room heaters which may be within terminals)	Does not include integral provision of ventilation and may be used with separate (cooled) ventilation system
A	**All-air systems**		
A1	Single duct system (including multi-zone)		
A2	Dual duct system		
A3	Single duct, Terminal reheat		
A4	Constant Volume (with separate heating)	✔	
A5	Variable Air Volume (with separate heating)	✔	
B	**Water-based systems**		
B1	Fan coil system, 2-pipe	✔	✔
B2	Fan coil system, 3-pipe		✔
B3	Fan coil system, 4-pipe		✔
B4	Induction system, 2-pipe non change over	✔	
B5	Induction system, 2-pipe change over		
B6	Induction system, 3-pipe		
B7	Induction system, 4-pipe		
B8	Two-pipe radiant cooling panels (including chilled ceilings and passive chilled beams)	✔	✔
B9	Four-pipe radiant cooling panels (including chilled ceilings and passive chilled beams)		✔
B10	Embedded cooling system (floors, walls or ceilings)		✔
B11	Active beam ceiling system	✔	✔
B12	Heat pump loop system	□	✔□
	Packaged Air Conditioning Units		
C1	Room units (including single duct units)	✔	✔
C2	Direct expansion single split system	✔	✔
C3	Direct expansion multi split system (including variable refrigerant flow systems)	✔	✔

THE UK APPROACH

At the time of writing, few EPBD calculation methods for non-domestic buildings have been produced. This section deals with the methodology that has been in use in the UK since April 2006. Similar procedures are used in the Netherlands and Germany – some of the British procedures mimic those already in use in the Netherlands. The current Dutch procedures for HVAC systems appear to be less detailed (in some respects) - for example, in not explicitly including duct leakage. The German standard [4] includes more detailed calculation procedures in some areas, such as the calculation of pump energy use (including for heat rejection devices).

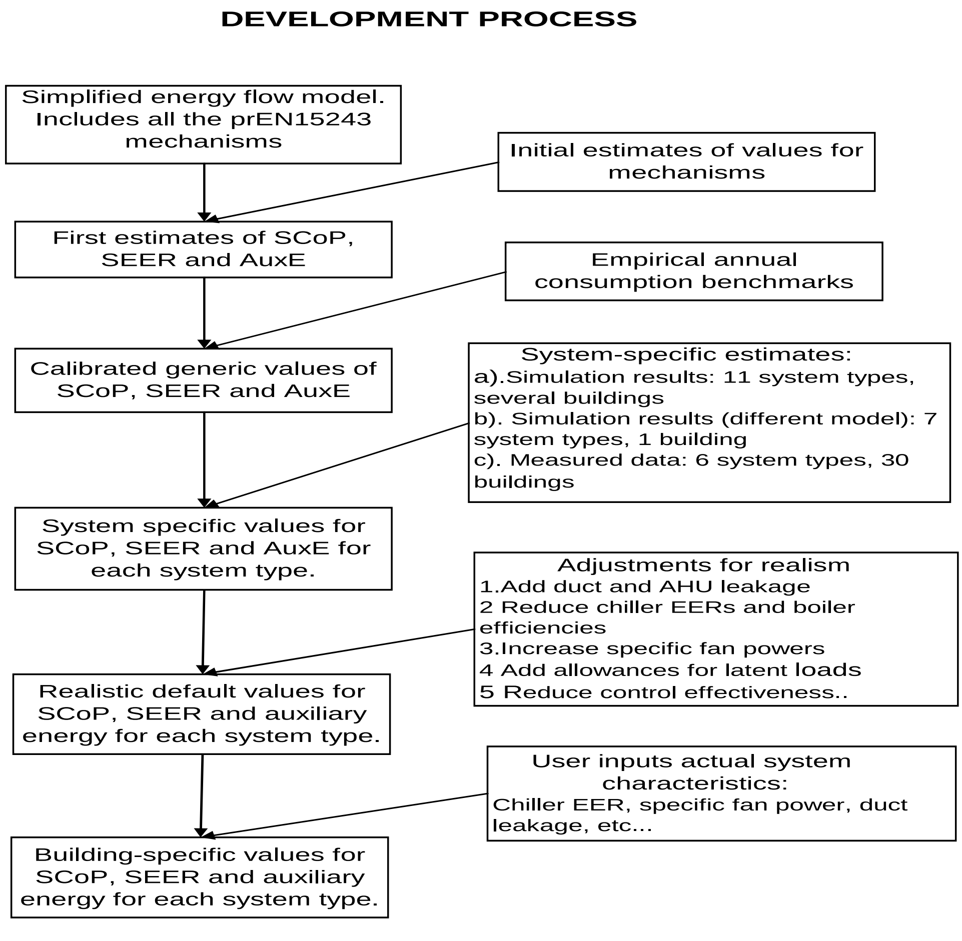
The standard calculation procedure for non-domestic buildings in the UK is the Simplified Building Energy Model (SBEM) [5] [6], although accredited hourly simulation procedures are also acceptable. The following section describes the development and basis of the HVAC system modelling in SBEM. The basic heating and cooling demand model is the monthly procedure from EN13790. [7] This imposes constraints on the way that the system performance is characterised. For consistency with the demand calculation, monthly values of the performance parameters (auxiliary energy, SCoP, SEER) are needed. •

The basis of the calculations in the UK is a comparison between the calculated carbon emissions of the proposed or actual building with those of a “notional” or reference building. The notional building has the same dimensions, geometry and use pattern as the actual building and is exposed to the same weather. However, the levels of insulation and the system efficiencies have fixed reference values. The notional building does not therefore have a specific HVAC system, but defined auxiliary energy, SCOP and SEER values.

Each serviced space in a building is assigned to an HVAC system. Each month the heating and (sensible) cooling demands and auxiliary energy are calculated and converted to energy demands by applying the appropriate SCoP and SEER. Each system type is associated with a vertical temperature gradient which is used to adjust the demands according to the height of the space and whether or not it is provided with de-stratification fans. There is also an adjustment to reflect the degree of direct radiant heat (or cooling) that falls on the occupied area. For most spaces and system these adjustments are small, but they can be significant in high, well-ventilated spaces.

The development procedure we adopted is summarised in Figure 1 and is described in more detail below. Because of the short timescale for implementation of the EPBD, there was limited time (or resources) for supporting research.

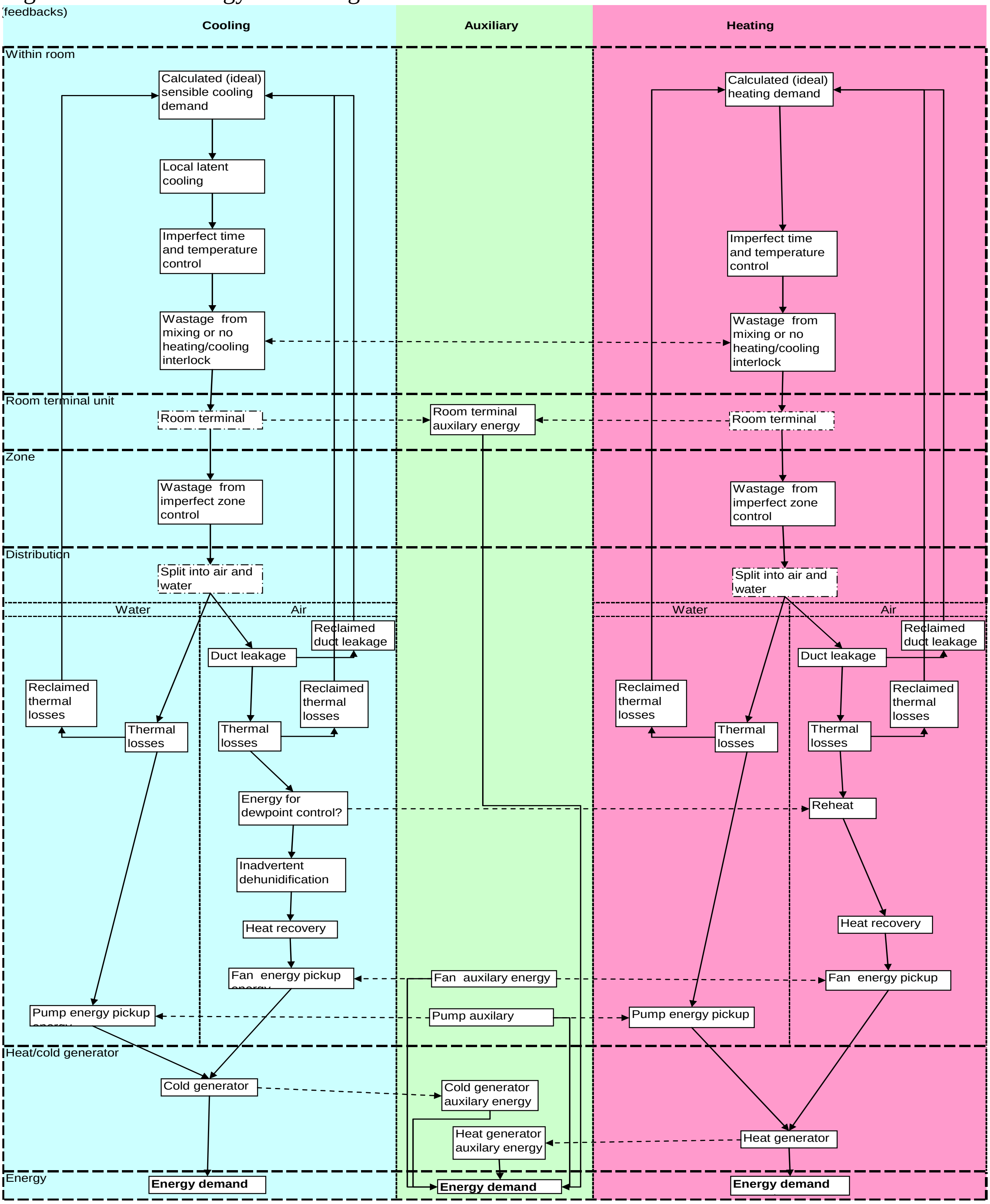
Figure 1 HVAC Model Development Process



• At present the parameters in SBEM are the same for each month. Values that varied monthly would be more realistic, and will be developed – especially for auxiliary energy – but the practical difference in terms of annual consumption estimates is actually rather small.

An energy flow diagram was constructed that included all the prEN15243 mechanisms. This flow diagram was implemented as a spreadsheet. The logic diagram is shown in Figure 2

Figure 2. HVAC Energy Flow Diagram



The basic philosophy is to provide a consistent set of parameters that address all the mechanisms in prEN15243. The energy flow diagram is simplified in that some of the parameters are relatively aggregated – for example, heat pickup in chilled water distribution pipework is expressed as a percentage of the cooling energy flow handled. There are many parameters whose values are difficult to determine reliably, many of which have only a small impact on the final result. System-specific default values are used for these. The user is be able to influence the system performance in two ways: firstly by his choice of system types (which sets the values of system-specific parameters) and secondly by adjusting the value of selected parameters. [•]

We next made first estimates of typical values of the flow sheet parameters and calculated initial figures of the three performance parameters (auxiliary energy, SCoP, SEER) . With some relatively small adjustments to the initial assumptions, the consumption figures that these implied were brought into in general alignment with empirical benchmarks [8] . This provided us with calibrated generic estimates of the parameter values.

In parallel with this, we brought together several sets of existing comparisons between the energy consumptions of different types of systems in offices. These included two sets of simulation results using different models to compare different systems in identical buildings. One of the studies examined 11 different system types in a number of buildings, while the other examined 7 system types in a single building – but modelled the system components in more detail. We combined these results with measured data from 30 buildings covering 6 system types [9], to develop a set of system-specific values for SCoP, SEER and auxiliary energy. For each system type, we then adjusted the spreadsheet parameters until the spreadsheet generated the same figures.

Since the simulations assumed idealised control and other conditions, we then degraded some parameters to provide less optimistic default assumptions. In particular we added duct and AHU leakage, reduced chiller EERs and boiler efficiencies, increased specific fan powers, added allowances for latent loads, and reduced control effectiveness.

The resulting "default" consumption levels straddle the "typical" consumption benchmarks (some systems being better than the benchmark, others worse). The idealised figures straddle the equivalent "good practice" benchmark.

As mentioned above, the user has the possibility of introducing specific values for some important parameters – currently specific fan power, duct leakage, chiller and boiler seasonal efficiency - in order to claim reasonable benefit from energy-efficient system design. In addition, of course, the choice of an inherently more efficient system is rewarded by the use of default values for that system type.

The calculation of the seasonal efficiency of boilers and (especially) chillers is not entirely straightforward, especially when there are multiple chillers and a degree of oversizing. Methods of handling this have been reported elsewhere. [10] [11]

DISCUSSION

This paper summarises the development of a practical approach to representing HVAC system in energy calculation software. Clearly there are other possibilities, notably the use of explicit hourly system modelling. In principle explicit modelling is more versatile, but it requires more data (for which default assumptions could be provided) and detailed assumptions about system configurations and sizing. The latter could probably be provided via standard system templates subject to later detailed specification. Such complexity is arguably more technically complete but this has to be balanced against the greater difficulties for those charged with checking calculations and verifying installations. The model described here allows the user to influence calculated system performance by choice of system and by defining the performance of the major system components,

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[•] The provision of access to over-write more of the defaults is a likely future development.

but contains many fixed default values. These may be opened up to user modification if experience justifies this.

An alternative approach would be to define explicit engineering calculation rules for the energy use by different elements of the system. This is effectively the approach taken in Germany [12]. In principle this is more transparent and more flexible than the UK approach (at least in the latter's current implementation), but this transparency is somewhat offset by the need for large numbers of look-up tables and very extensive documentation. Nevertheless, when implemented as software, the result has a rather similar appearance to that of SBEM. It is intended that SBEM will continue to develop in the light of user experience – and taking note of parallel developments in other Member States. Although a single Europe-wide methodology seems unlikely, it does seem likely that, over time, there will be increasing convergence of approach.

The current version of SBEM has been developed to a tight timescale and time and resources have meant that a number of desirable features are not yet incorporated. The software has been in use since April 2006 It can be downloaded, together with a Userguide and associated (UK-focussed) databases from www.ncm.bre.co.uk .

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Comparison of two calculation methods used to estimate cooling energy demand and indoor summer temperatures

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SUMMARY

As a part of the Finnish implementation of the Energy Performance of Buildings Directive, a computational exercise was carried out to choose a simple method for cooling energy estimation. Three different variants of the hourly calculation method and the monthly method were implemented. A model office building was used as the object for the cooling energy predictions. The IDA-ICE building energy software was used to produce reference results. The predicted space cooling energy numbers for a three-month summer-period were compared with the reference results. The comparison shows that the simplest one-capacity variant of the hourly method is underestimating the reference results up to 25% and overestimating up to 30%. A more detailed four-capacity variant is underestimating up to 21% and overestimating up to 19%. The monthly calculation method is not underestimating but overestimating the reference up to 75%. The hourly method is able to estimate the indoor summer maximum temperatures with a deviation less than two degrees from the reference.

INTRODUCTION

The 2003 published Energy Performance of Buildings Directive (EPBD) [1] is calling for a methodology for calculation of the integrated energy performance of buildings. The methodology shall be set at national or regional level. The Finnish approach was to develop a simple and transparent calculation method, which could be easily implemented. The heating energy calculation was principally based on an existing, simple and standardised method [2] working with a monthly calculation period. For cooling energy demand calculation a choice between two methods had to be done. The first method, running with a time-step of one hour, is based on a simplified thermal resistance-capacitance model of the building zone. It is in principle capable to predict the cooling energy demand and to follow the daily temperature changes inside the building. The second alternative is a variant of the monthly heating energy calculation method [2] now applied to estimate cooling energy demand. Proposals of both methods for EPBD use have been published [3, 4] but with a very restricted evidence of performance. Especially the local weather conditions and the culture for implementing buildings and their heating, cooling and ventilation systems influence the calculated results. For this reason it was important to have some further evidence and comparison results.

The objective of the study was to compare the two cooling energy demand calculation methods to have some facts and arguments for the Finnish EPBD methodology implementation work.

METHODOLOGY

The principle of the comparison study was to calculate the cooling energy demand of a well-defined building zone using the different methods and to compare these results with the results obtained with a reference. As a reference *IDA Indoor Climate and Energy 3.0* (IDA-ICE) building simulation software was used [5]. It is a detailed whole-building simulator allowing all issues fundamental to building energy calculation: shape, envelop, glazing, HVAC systems, controls, light, etc. The mathematical models are described in terms of equations in neutral-model-format (NMF), which makes it easy to replace and upgrade program modules. IDA ICE has been shown to perform well in several computational comparisons [6, 7].

The object of the computation exercise was one floor of a model office building, Fig 1. The dimensions, component properties, and load profiles of this building are described in detail in a model building report [8]. For the purpose of the implementation of the cooling energy calculation methods, four zones in the floor were selected: two office rooms (R1 and R2) with a floor area of 10.5 m^2 each and two open plan offices (O1 and O2) with a floor area of 295 m2 each, as shown by Fig. 1.

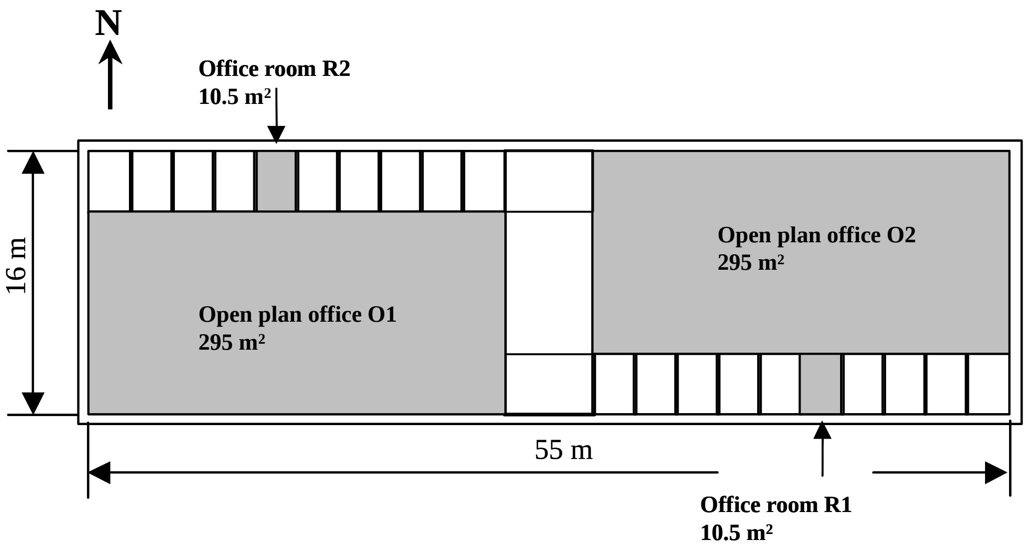

Figure 1. Model building floor plan and studied zones.

Two different constructions were specified for the model building. Construction 1 was a standard construction according to the model building [8] with light walls and a heavy concrete floor/ceiling. A second construction, Construction 2 was defined to find out the influence of the building mass on the calculation results. In Construction 2 walls are identical with Construction 1, but the floor and ceiling are considerably lighter. Table 1 shows the ceiling/floor for Construction 2.

Table 1. Ceiling/floor thermo physical properties for Construction 2.

		Thickness (mm)	Thermal conductivity (W/m K)	Density (kg/m^3)	Heat capacity (J/kg K)
1	floor covering	3	0.18	1390	1000
2	chipboard	22	0.14	400	1500
3	steelwork + air gap	200			
4	mineral wool sound insulation	50	0.045	110	840
5	plasterboard	13	0.21	800	840

The effective thermal thickness of the construction components was defined according to the simplified calculation of heat capacity Annex A of ISO 13786 [9].

The masses and thermal capacities for the office room and the open plan office for the two constructions are shown in Table 2. C_m is the effective heat capacity for the room and A_{floor} is the floor area.

Table 2. Masses and thermal capacities of construction components.

	Construction 1		**Construction 2**	
	Office room	Open office	Office room	Open office
Mass of external wall (kg)	96	1620	96	1620
Mass of internal walls (kg)	530	1825	530	1825
Mass of floor and ceiling (kg)	9116	256160	610	17166
C_m / A_{floor} (kJ / Km^2)	322	294	60.0	32.5

Components for heating and cooling exist in each room of the building. Ventilation air through a mechanical ventilation system is supplied to each room. The controlled ventilation air supply temperature is a function of the exhaust air temperature, where a linear relation exists for supply between 21˚C and 17˚C and exhaust between 22˚C and 24˚C, respectively. The ventilation supply has a daily and a weekly schedule. The heating and cooling systems and the ventilation maintain the indoor air temperature between specified set limits of 21˚C and 24˚C around the year.

A part of the input data, like internal gains, is defined in the model building description [8]. As weather data Helsinki 1979 weather was used. Other input data like solar heat gain through the window structure and infiltration are taken according to the reference software to focus on the calculation of the building zone thermal behaviour. The input data is arranged as hourly or monthly data, according to the method.

HOURLY CALCULATION METHOD

The hourly calculation method for cooling is based on the conservation of energy at the temperature node points of a simple resistance-capacitance model of the building zone. The base for this model was adopted from [3]. Because the base model did not include the description of mechanical ventilation, which is the usual way of system implementation in Finland, this feature was added to the model Variant_1, Fig 2.

The model runs on an hourly basis where the calculation time step is one hour and the input/output data are arranged as hourly data. The thermal mass of the calculation zone is lumped into one thermal capacitance. The heat transfer connections between the thermal nodes are described with four resistances. The mathematical solution is based on the Crank-Nicolson difference scheme. Both heating and cooling modes can be calculated, as well as variable room air temperature and variable cooling system operation. A node point and a heat capacity flow for mechanical ventilation supply are also included.

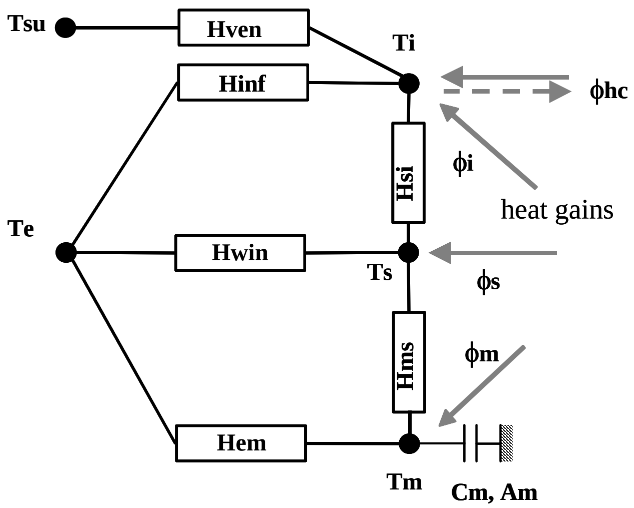

Figure 2. Hourly method, model Variant_1.

In Fig. 2, T_e is outdoor air temperature, T_i indoor air temperature, T_m mass temperature, T_s is a star (operative) temperature, H_{ven} ventilation heat flow, H_{inf} the infiltration heat flow, H_{win} the window thermal conductance, Hem the external structure thermal conductance, and H_{ms} and H_{is} thermal conductances between respective node points. The heating power added or cooling power removed from the room is ϕ_{hc}, heat gains to the internal nodes are $\phi_{i,}$ ϕ_s and $\phi_{m,}$ the effective heat capacity for the zone C_m and the equivalent area for the zone capacity A_m.

The computation procedure [3] begins with calculation of the internal gains, including the solar heat gain. The gains are allocated to the internal node points according to certain weights. The heating/cooling power is kept equal to zero at the first stage. The mass-node temperature is determined from the Crank-Nicolson energy conservation equation for the mass-node. The star temperature and the air temperature are determined from the node energy conservation equations. If the resulting air temperature T_i is in the temperature control set limits, the need for heating or cooling is zero and the calculation can proceed to the next hour. If the resulting air temperature is above the higher set limit, the air temperature is set equal to the upper limit (24 ^{o}C) and the cooling energy needed to keep this temperature level is calculated. In case of heating the procedure is analogous.

To find out whether increasing the number of the node points gives any advantage in the cooling energy demand calculation, the model Variant_1 was further modified into a four-capacity model Variant_2, Fig 3. The star node was rejected because the model was used for energy calculations and there was no need for an operative temperature. Two capacity nodes representing the building mass (walls, ceiling, floor) in more detail were added and the air node was also given a heat capacity.

The calculation procedure is in principle similar than for the previous variant. There are four differential conservation equations instead of one and no equation for star node. The solution is based on a difference scheme. To keep the mathematical manipulation as simple as possible, the usual matrix inversion is avoided by a sequential solution of the equations. The gains are identical but allocated in a slightly different way to the node points.

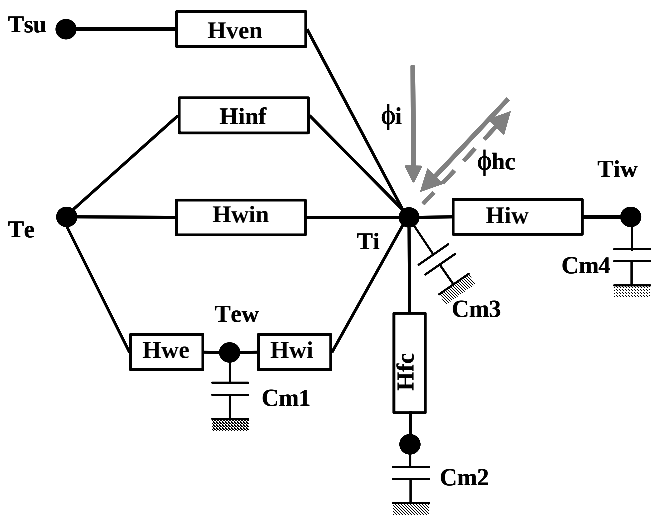

Fig. 3. Hourly method, model Variant_2.

Finally a third model Variant_3 was made out of Variant_2 by disengaging the external structure from the outdoor air point Te and engaging it to a new sol-air temperature node T_{sa} to account for the absorbed solar radiation on the external wall structures. The determination of the sol-air temperature was based on the energy balance of the building external surface.

MONTHLY CALCULATION METHOD

The monthly method to calculate cooling energy demand [4] is based on earlier methods used for heating energy calculation [2]. The time period for the calculation is one month. All input data is one-month average or cumulative values. The main principle of the method is the long-term energy balance for the building zone. Without cooling the cumulative energy from the heat gains (solar + internal) equal the heat losses. The indoor temperature is free floating and exceeds the set temperature from time to time. Introducing cooling to the zone keeps the internal temperature below or equal to the set temperature. The cooling energy is less than the gains because a part of the losses is compensating the gains:

$$Q_c = Q_g - \eta_l Q_l \tag{1}$$

where Q_c is cooling energy demand, Q_g is thermal energy from gains, Q_l is loss term and η_l is the utilization factor for heat losses. It has been shown [4] that equation (1) can be converted into

$$Q_c = (1 - \eta_g) Q_g \tag{2}$$

where η_g is now the utilization factor for heat gains initially used for the heating demand calculations. The utilization factor η_g depends on the gain-loss ratio, the time constant of the zone and two parameters a_0 and τ_0. Different values for these parameters have been used [4,10,11] depending on the application. Here the values a_0 = 1.83 and τ_0 =83.3 for cooling of non-residential buildings according to NEN 2916 standard were adopted [4].

RESULTS

With the hourly method and the reference software a period of four months May – August was calculated using the hourly weather data of Helsinki 1979. The first month was a pre-period to let the structure temperatures find their natural temperature levels. The three following months were for energy calculation. The monthly method was utilising the monthly average temperatures and cumulative energy values derived from the same hourly weather data.

Results for the space cooling energy demand during the period June, July and August for the different calculation methods, zones and building constructions are shown in Table 3.

In case of the single office rooms Table 3 shows that all three implementations of the hourly method mainly underestimate the reference values. Variant_3 seems to perform best except in case of the North facing lighter Construction 2, where it overestimates the reference value by 17 %. This is a consequence of the sol-air temperature, which is the only difference between Variant_2 and Variant_3. The monthly method predicts the cooling energy demand very close to the reference values for the south facing office rooms but overestimates up to 76% for the north facing rooms where the gains are much smaller.

Both methods overestimate the reference cooling energy demand numbers for the open space office case. The percentage difference is larger for the North facing office than for the South facing office. The construction type seems not to have any significant effect on the hourly method results. The monthly method overestimates less for the light structure Construction 2.

Table 3 Predicted space cooling energy demand for the period June – August.

Calculation method		Office room R1 facing South		Office room R2 facing North		Open office O1 facing South		Open office O2 facing North	
		cooling energy kWh	cooling/ reference	cooling energy kWh	cooling/ reference	cooling energy kWh	cooling/ reference	cooling energy kWh	cooling/ reference
constr. 1	Hourly V_1	131	0.79	38	0.75	2478	1.12	1531	1.24
	Hourly V_2	132	0.79	40	0.80	2385	1.08	1471	1.19
	Hourly V_3	149	0.90	49	0.98	2613	1.18	1649	1.33
	Monthly	179	1.08	88	1.76	3006	1.36	2158	1.75
	IDA	166	1	50	1	2211	1	1237	1
constr. 2	Hourly V_1	175	0.90	67	1.03	3279	1.12	2244	1.30
	Hourly V_2	155	0.80	57	0.87	3104	1.07	2050	1.18
	Hourly V_3	188	0.97	76	1.17	3511	1.20	2455	1.42
	Monthly	194	1.00	104	1.59	3464	1.19	2614	1.51
	IDA	194	1	65	1	2916	1	1732	1

In general, the rate of overestimation of the reference values is increasing with decreasing structure capacity and decreasing gains for the hourly calculation method, and vice versa. The monthly calculation method behaves in a different way. The rate of overestimation is increasing with increasing structure capacity and decreasing gains.

One important factor in assessing the need for active cooling is the behaviour of the indoor air temperature during the summer period. If this temperature is exceeding the specified limits without cooling, there obviously is a need for cooling. The hourly calculation method is able to provide this information. The temperature limits, which are used for energy calculation, are inactivated and the indoor air temperature is let to “float” during the computation. The results for the period June – August show that the hourly Variant_1 is overestimating the maximum indoor temperature by 2.2 ^{o}C compared with the reference result. The hourly Variant_2 is overestimating the maximum temperature only by 1.8 ^{o}C. The monthly method is not able to provide useful information related to temperature maximum or minimum values.

CONCLUSIONS

A computational exercise for the assessment of two simplified calculation methods, the hourly method and the monthly method has been carried out. The space cooling energy demand for four different zones of an office building was predicted with three variants of the hourly method and the monthly method. The IDA-ICE building energy software was used as a reference.

The predicted cooling energies for a three-month period vary from 75% to 175% compared to the reference results. The hourly method implementations underestimate the reference values for a heavy structure and high gain and overestimate for a light structure and small gain values. The monthly method mainly overestimates, the more the smaller the internal gains.

As a part of the Finnish implementation for the cooling energy calculation method of the Energy Performance of Buildings Directive, a one-capacity variant of the hourly calculation method was further developed, tested and proposed for the Finnish authorities. For the Finnish implementation of the EPBD the monthly calculation method was chosen.

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Simplified Model of Seasonal Energy Consumption by Air Conditioning System in Non Residential Buildings

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SUMMARY

When analyzing the current standards issued to be complementary to the EPBD Directive, one can have an impression that the energy consumption by AC system depends on the specific heat gain and the air flow only. The author's main idea was to prove that there are some other but also very important parameters of the AC system design and operation which have to be also taken into account seriously. The most important of them are: type of air treatment and control, chillers' efficiency and their operating parameters, type of climatic conditions data. To prove this, a specific calculating procedure for cooling season energy consumption by HVAC system in office building placed in Kraków, Poland was defined. As a result of simulations and analysis provided for CAV type AC system, it can be clearly seen that in central European climatic conditions particular concern in each HVAC project must be paid on "cooling source" type, efficiency and control as well as air flow treated optimization.

INTRODUCTION

Officially starting from 1st of January 2006 all of the EC countries are obliged to follow the rules included in the EPBD Directive. Different countries are on different stages of preparation to this target. Different countries have also different structures of energy consumption in buildings. Northern countries pay more attention to heating systems and southern ones the opposite, to the A/C systems. Poland is somewhere in the middle of both. During last months a lot of documents and standards related to this subject were published in Brussels. For instance, when studying the prEN 13790 standard [1] one can come to the following conclusions:

- The calculation methods for energy consumption in building due to the space heating season only are already well developed and most of them are sufficiently precise
- For cooling and air conditioning the situation is not so much clear however the energy consumption of these systems makes substantial percentage of total energy consumption, particularly in mid and southern Europe
- Most of the models presented for cooling period are based on the "sensible heat" balance of the building envelope. This may lead to some "inaccuracies" because in most solutions cooling is coupled with dehumidification which produce an "additional energy consumption" in building.

So, the main target of the author's research was to prove how strongly the energy consumption during summer season is affected by the following aspects:

- Precisely defined class of the indoor thermal comfort
- Type of climatic data
- System components' characteristics, particularly chillers and fans

- Operating parameters and control

To achieve this target a simplified model for energy consumption in summer season was built and the basic assumptions and calculation results are discussed in this paper based on an example of typical “open space” type office located in Kraków (Poland).

METHODS

Air properties – Mollier diagram

All the calculations of the air thermodynamic properties have been provided using an “air properties model” based on formulas described in [2] and also on “graphic coordinates” of Mollier diagram built. In some cases it was necessary to find the thermodynamic parameters of a “crossing point” of two particular air treatment processes. It is important to mention that all the air treatment processes were built as linear in the diagram. The accuracies of determining the particular air parameters in this model were assessed higher then:

- temperature: ± 0,1 ^{o}C
- specific enthalpy: ± 0,1 kJ/kg
- relative humidity: ± 0,5 %
- humidity: ± 0,1 g/kg

Climatic data

The most difficult problem for summer season energy use calculations for buildings in Poland is to get representative and averaged climatic data including hourly defined outdoor conditions like: air temperature, air humidity, solar radiation heat flux. Those three are the minimum necessary to be used by the model presented in this paper. Because, so far these data are not officially available, the calculations had to be provided in two, following ways:

1) For quasi dynamic hourly calculations the following data and assumptions were used:

- temperature and humidity in “hourly mode” were applied as the recoded data in the year 2001, by one of the meteorological station in Kraków,
- for the solar heat gain calculations the data published in the polish standard [8] were used. It gives solar heat flux on particularly oriented outside walls hour by hour but only for one “representative” (“calculating”) day for each month of summer season. So it has to be assumed when determining the room or building “thermal characteristics” that all the days are the same in each month and they are similar to the “calculating day”.
- Because of lack of clear reationship between the outdoor air temperature and humidity and the solar radiation intensity, the model had used them separately (it means, the influence of outdoor temperature onto heat gain of the building was assumed to be almost negligible in comparison with solar- based ones)

2) For the “comparative” calculations to those described above it was assumed that:

- temperature and humidity of the outdoor air were taken from the same polish standard [8] as the solar flux data and all the days of the same month were concerned “the same” as the “representative” one.

Hopefully, in near future, polish government will publish full data according to the European standard [7] and in that case, the model will be amended to be fully hourly one.

Building heat gain calculations

There is no doubt that energy consumption of the building during the “summer season” strongly depends on both, the internal and external sources heat gains. The basic problem to cope with in calculations of energy use by HVAC system in summer is to have a simple but reliable model for calculations of heat gains to the building. Creating such a model let us in particular consider the following requirements and conditions:

- calculations of heat and humidity gains in "hourly mode"
- heat gain from solar radiation through windows and "transparent walls" should be done with highest attention. The accumulation in construction should be taken into account
- the heat gain through "non-transparent" walls may be treated as "less important" and it can be calculated assuming that heat flux transferred through the defined wall is quasi constant and it is a function of the difference between the "daily averaged solar outdoor temperature" and indoor air temperature.
- the gain from the inhabitants and users is achieved based on "average square area per person" [1]

Cooling & heating demand determination

CAV systems one can call as "supply air quality control" system because the transient heat and humidity balance for the building envelope is matched by changing the air supply parameters (represented by the point "N" at the Mollier's diagram). It means that supplied air have to balance both, the sensible and humidity gains of the building (represented respectively by equations 1 and 2.

$$V_S \cdot \rho \cdot c_p \cdot (T_p - T_N) = \sum \Phi_s \qquad (1)$$

$$V_S \cdot \rho \cdot (X_p - X_N) = \sum W \qquad (2)$$

This point can be called like a "target of the air treatment processes in the AHU". The base of the "a priori" optimization of the A/C system is *"to achieve this target when using as less energy as possible in current climatic conditions of the outside air"* .

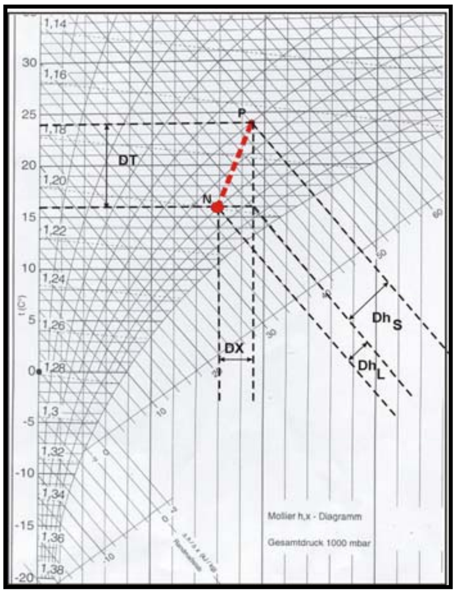


Figure 1. The supply air parameters determination in "h-X" diagram

No doubts that there are several strong links and limits which play major parts in this game:

- when extending DT we are loosing the thermal comfort lever in the users' zone
- when decreasing the DT we grow up with the air flow causing both the energy consumption (by fans) and investment costs (AHU and duct dimensions) increases.

Thus we can assume that "position" of the point "N" in the Molllier's diagram responds the current (transient) state of the building. The energy demand calculations were provided in hourly mode basing on that during each separate hour a determined supply air as well as the outdoor air parameters remain constant (points "N" and "Z" in the diagram). Then, according to the particular assumptions, the air treatment processes are automatically designed (see figure 2).

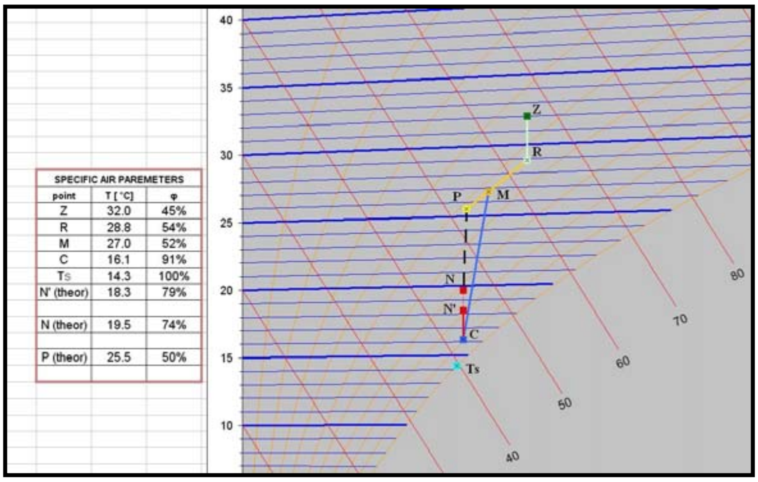


Figure 2. The air treatment processes designed with the help of the model converted into a software package

A crucial element of these calculations is the cooling coil and its capacity control which determines the chiller water outlet temperature which strongly influence the chiller efficiency and subsequently the electric power consumption. So the cooling coil model was identified taking into account that:

- the cooling coil supply water temperatures are in correlation with “average, weighted external surface temperature - Ts” of the cooling coil. (It was assumed, based on “good refrigeration practice” that Ts is approx. 1,0 ^{o}C lower then the water outlet temperature)
- for calculated system, contact factor “Cf” of a cooling coil remains constant.
- in all calculations the air treatment process in the cooling coil is represented by a “straight line” determined by two points, one representing the cooling coil air inlet parameters and the second representing the parameters “Ts” and relative humidity φ = 100 %
- the nominal operating parameters for the chiller were optimized to get the maximum available EER efficiency (maximum possible chilled water outlet temperature). Afterwards, the chiller is assumed to operate with the same set point through all over the season.
- to match the required cooling coil capacity (in transient conditions) 3-way mixing valve and continuous controller was used.

One of the author’s main concern was to determine as precisely as possible the EER efficiency of a chiller as a function of current operating conditions. After analysis it was developed a regression of EER as a function of both, outside air temperature and chilled water outlet temperature (see equation 3).

$$EER(T_{w2}, T_{az}) = (a \cdot T_{az} + b) \cdot T_{w2} + c \cdot T_{az} + d \qquad (3)$$

where:

T_{az} - outside air temperature

T_{w2} - water outlet temperature

a, b, c, d - linear regression factors

This function was currently done based on data of one selected manufacturer for the scroll chillers with R407C. It was validated for the ranges: T_{az} between +20 and 35 ^{o}C and T_{w2} between +5 and + 12 ^{o}C. In the future it is intended to develop this function as more general one (dependant also on the type of refrigerant and EUROVENT data).

Algorithm of the energy consumption for cooling season

First step of the calculations was to determine the **averaged, mean seasonal heat and humidity gains** into the building which was based on averaging of the solar heat gain for the daily operation period through allover the season. To get this the following formulas were used:

- The averaged seasonal sensible heat gain [W]:

$$\Phi_{s,mean}^{season} = \Phi_{g,mean}^{season} + \Phi_{w,mean}^{season} + \Phi_{s,u} \cdot \varphi_u + \Phi_{comp} \cdot \varphi_{comp} + \Phi_{add} \qquad [W] \qquad (4)$$

where:

$\Phi_{g,mean}^{season}$ - heat gain through the glass (windows or transparent walls”) [W]

$\Phi_{w,mean}^{season}$ - heat gain through the “non transparent” walls calculated on the base of “mean seasonal solar temperature” [W]

$\Phi_{s,u}$ - sensible heat gain from the users [W]

Φ_{comp} - sensible heat gain from computers [W]

Φ_{add} - extra, additional sensible heat gain in the building [W]

φ - “averaging factors”

- The average solar gain for the entire building was calculated using weighted factors for all of the windows' (glass walls) major orientations as shown in the equation 5.

$$\Phi_{g,mean}^{season} = \Phi_{g,mean,E}^{season} \cdot n_{g,E} + \Phi_{g,mean,SE}^{season} + ...+ \Phi_{g,mean,NE}^{season} + \Phi_{g,mean,H}^{season} \qquad [W] \qquad (5)$$

where:

$\Phi_{g,mean,E}^{season}$ - heat gain through the glass in the wall orientation "E" [W]

$\Phi_{g,mean,H}^{season}$ - heat gain through the glass in the horizontal "roof" [W]

$n_{g,E}^{s}$ - windows, glass walls surface in the wall orientation "E" divided by the total glass walls' area of the building

b) The seasonally averaged value of the humidity gain were calculated according to the following formula (6):

$$G_{w,mean}^{season} = w_{1,u} \cdot n_u \cdot \varphi_u^{season} \cdot 0{,}277 \cdot 10^{-6} + G_w^{ex} \qquad [kg/s] \qquad (6)$$

where:

$w_{1,u}$ - humidity gain from 1 person (user), related to the activity [g/hour]

n_u - total amount of the persons in the building

φ_u^{season} - seasonal averaging factor for the amount person in the building

G_w^{ex} - additional, "extra" humidity gain in the building [kg/s]

Finally it was possible to define the **mean seasonal air inlet parameters** (point $\mathbf{N}_{\boldsymbol{mean}}$ in the Mollier's diagram, Figure 3). These were calculated assuming that air supply flow volume $\mathbf{V_S}$ was already determined (for nominal conditions) and it rests constant as for the CAV system. Using the above parameters as a "target" for air treatment processes through allover the season, hour by hour it was calculated the cooling and heating (if required) demands, chiller efficiency and subsequently electrical power consumption for the predefined relevant outside air temperature and humidity.

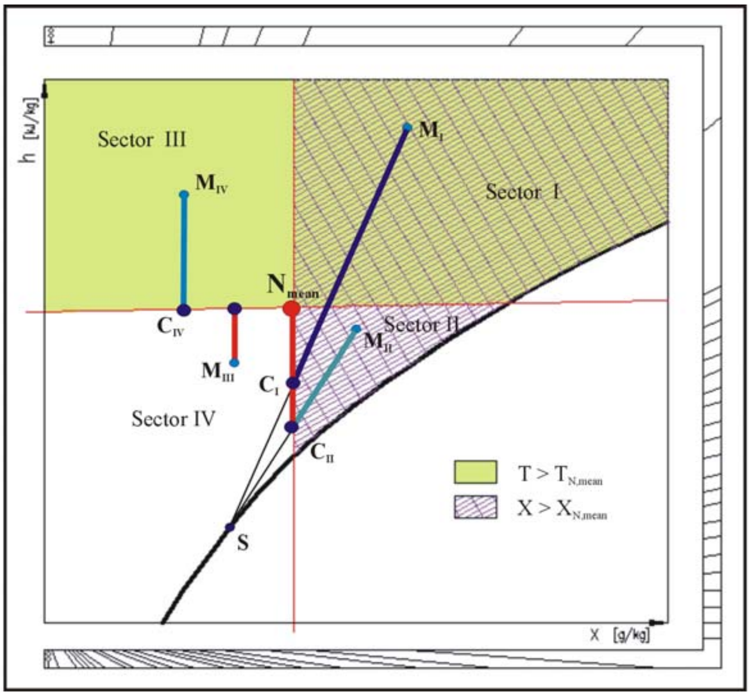


Figure 3. The air treatment processes dependent on cooling coil inlet parameters for cooling/heating demand calculations

Dependant upon different air treatment strategies applied including: mixing, heat recovery, free cooling as well as current outdoor air parameters, the cooling coil air inlet parameters were defined. (point "M", see figures 2 & 3). Relating to the particular position of point **M** the following air treatment control was realized:

i) If $X_M > X_{N,mean}$ (sectors I and II) then the cooling coil treats the air until $X_C = X_{N,mean}$ (with the constant "mean HE surface temperature" - point **S**). After this the secondary heater is activated to achieve the $\mathbf{N_{mean}}$ parameters.
ii) if $X_M \le X_{N,mean}$ and $T_M > T_{N,mean}$ (sector III) then the cooling coil treats the air until $T_C = T_{N,mean}$. No heating is required in that case. The indoor air humidity is subsequently lower then required. No more air treatment is done.
iii) if $X_M \le X_{N,mean}$ and $T_M \le T_{N,mean}$ (sector IV) then no cooling is required. Secondary heating coil heats the air until $T_C = T_{N,mean}$. The indoor air humidity is subsequently lower then required. No more air treatment is done.

In all cases above the **cooling capacity** was calculated for each separate operating hour as:

$$\Phi_i^C = V_S \cdot \rho \cdot (h_M - h_C) \quad \text{[kW]} \tag{7}$$

where:

ρ - air density [kg/m^3]

$h_M ; h_C$ - specific enthalpy of the cooling coil inlet and outlet air [kJ/kg]

At the same time the **heating capacity** required by secondary heater was derived from:

$$\Phi_i^H = V_S \cdot \rho \cdot c_p \cdot (T_{N,mean} - T_C) \quad \text{[kW]} \tag{8}$$

where:

C_p - air specific heat [kJ/{kg ^{o}C)]

$T_{N,mean} ; T_C$ - temperature of the heating coil inlet and outlet air [kJ/kg]

The electricity consumption of fans as well as pumps driving the water through the heat exchange coils were calculated according to the formula (9). There was assumed that both fans and pumps power consumption remains constant through allover the season.

$$P_{EF}^C = \sum \frac{V_F \cdot DP_F}{\eta_T} \quad \text{[W]} \tag{9}$$

where:

V_F - fluid volume flow [m^3/s]

DP_F - fan/pump available pressure required [Pa]

η_T - total efficiency of the fan/pump motor set [-]

So finally, the **total electricity consumption** by an HVAC during defined cooling season was provided according to the following formula (10):

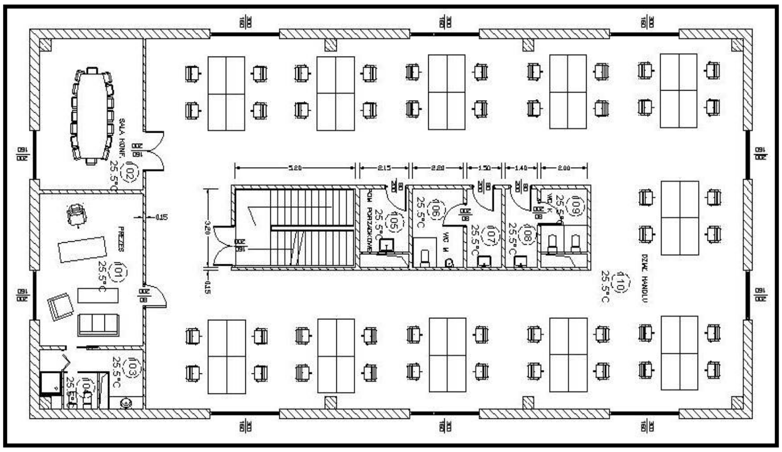
$$EC_E^{season} = \left(\sum_1^{\tau_{tot}} \frac{\Phi_i^C}{EER(T_{w2}, T_{az})_i} \right) + \frac{P_{EF}^C \cdot \tau_{tot}}{1000} \quad \text{[kWh]} \tag{10}$$

and **total heat consumed** by secondary heater (when necessary):

$$EC_H^{season} = \left(\sum_1^{\tau_{tot}} \Phi_i^H \right) \cdot 0{,}0036 \quad \text{[MJ]} \tag{11}$$

RESULTS

For testing this model one example of the new, designed building was calculated and entirely checked out using standard, traditional methods. This was “single floor, open space type office, air conditioned with CAV system” placed in Kraków. The basic assumptions and input data are shown in the figure 4



Input data							
Daily operating period time (5:00 - 19:00):		h	od		do		
			8		18		
working days (from 1 to 7):		days	5				
Beginning and end of the cooling season		d, m	start		end		
			d	m	d	m	
			1	4	30	9	
Building characteristics:							
Useful surface of the floor to be air conditioned	A_B	[m^2]	390,74				
Total surface of the external walls	A_{wo}	[m^2]	319,55				
Volume of the building to be conditioned	V_B	[m^3]	1367,59				
Mass of the building construction per 1 m2 of the floor air conditioned	m_{1A}	[kg/m^2]	300				
Heat transfer factors of the external walls	k	[W/m^2 K]	0,35				
Radiation absorption factor of the external walls	A		0,7				
Percentage of the "transparent walls" surface to the air conditioned floor	n_g		10,65%				
Percentage of the particular orientation "transparent wall" surface to the total "transparent walls" surface	n_g^x		N	E	S	W	H
			15%	35%	15%	35%	0%
Ratio of the transparent to entire window area	φ_{r1}		0,8				
Solar radiation protection device factor *(for nominal conditions)*	φ_{r2}		0,75				
Solar radiation protection device factor *(for mean seasonal conditions)*	φ_{r2}		0,9				
Transparency factor of the glass	φ_{rg}		0,8				
Floor area per person factor	A_{1u}	[m^2]	8,0				
Averaging factor for people (users) quantity	φ_u		0,9				
Floor area per computer factor	A_{1comp}	[m^2]	12,0				
Unitatry heat gain from a computer	q_{1comp}	[W]	200				
Extra, additional sensilbe heat gain in the building	Φ_{add}	[W]					
Extra, additional humidity gain in the building	$G_{w,add}$	[g/s]					
Indoor parameters:							
Thermal comfort category according to prEN 15251 ***(I, II or III)***			I				
Indoor air temperature according to prEN 15251	T_{in}^C	[°C]	25,5				
Indoor air realive humidity according to prEN 15251	RH_{in}^C	[%]	50%				
The nominal temperature difference between the supply and extract air	DT_{in}^C	[°C]	8				
The minimum outside outdoor air flow per person	V_{1f}	[m^3/h·os]	30				
Total heat gain per person (activity and clothing dependant, EN7730)	$q_{T,u}$	[W]	145				
Humidity gain per person (activity and clothing dependant, EN7730)	w_u	[g/h]	75				
Outdoor air nominal parameters:							
Nominal (calculating) outdoor air temperature for cooling (climatic zone)	T_{out}^C	[°C]	32				
Nominal outdoor air humidity for cooling (climatic zone)	RH_{out}^C	[%]	45%				

Figure 4 The most important input data for the calculations example

Using the presented model several different tests and comparisons were done. Because of limited volume of this article only one of them is presented in which two type of climatic data and following consequent calculations procedures were compared: the hourly defined outside air temperature and humidity (2001y) and averaged climatic data based on polish standard [6]. The results are presented in the table (Figure 5)

Simulations results comparison		PN 03420 [1]	Year 2001 [2]	Increase [2] above [1] in %
Mean seasonal gain of total heat	[kW]	**17,52**	**16,94**	-3%
Maximum cooling capacity required	[kW]	**35,80**	**44,12**	23%
Maximum heating capacity required	[kW]	**3,75**	**9,54**	154%
Seasonally averaged chiller efficiency		**4,03**	**4,29**	7%
Seasonal electricity consumption by chiller	[kWh]	**7 258,33**	**6 099,10**	-16%
Seasonal electricity consumption by cooling water pumps	[kWh]	**277,80**	**249,80**	-10%
Seasonal electricity consumption by heating water pumps	[kWh]	**1,84**	**4,72**	157%
Total electricity consumption by ACS during cooling season	[kWh]	**14 428**	**13 243**	-8%
Heat consumption by secondary heater during cooling season	[GJ]	**2,55**	**3,56**	40%

Figure 5. Most important figures comparisons for the example presented in the paper

DISCUSSION & CONCLUSIONS

When analyzing the tests results provided according to the presented model the following conclusions can be written:

- The source and type climatic data have rather poor influence on heat gains of a building and subsequently the size of AHU (air handling unit)
- Selection of the chiller and its efficiency is greatly significant in the total electricity consumption during cooling season
- Hourly model of energy used by chiller (based on the EER function of outdoor air and chilled water outlet temperatures) makes the result much more reliable and realistic. For this test example the averaged EER for the season were approx 40% higher then EUROVENT data !
- These presented results are only an example to show how important is such a model for air conditioning systems design process where on-line system energy consumption may be provided according to the EPBD Directive
- The government of each country where the EPBD Directive is obligatory should pay a great attention and make efforts to validate and publish as soon as possible climatic reference data in hourly mode coherent with the standard [5] which would allow to built and validate such tools as those presented in the text above

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New correlations for the standard EN 1264

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SUMMARY

The standard EN 1264 is born as technical standard for designing and installing radiant floors for heating purposes. In the last years a revision of the standard took place in order to extend the existing calculation method for determining heat flow output also to other radiant systems (walls and ceilings) and operating conditions (heating and cooling).

Nevertheless there are some aspects which have not so far been taken into account, like backward heat flows or how to determine, in cooling conditions, minimum surface temperature and related cooling capacity of the radiant system. In this paper the concepts for correctly consider those aspects are shown and the introduction of correlations and equations which can be integrated in the standard are presented.

INTRODUCTION

The Standard EN 1264 [1, 2, 3, 4] includes a simplified method for sizing radiant floors, by defining pipe spacing, pipe diameter and material type of insulation etc. Although it is a simplified method, its application has some limits, therefore it cannot always be applicable. The existing method has one limit, i.e. the determination of the downward heat flow, which is not directly specified and it is roughly estimated when defining the mass flow rate of the system.

The new draft of standard makes possible to size not only radiant floors, but also radiant ceiling and walls, by defining different heat exchange coefficients on the surfaces (Table 1). It is also defined how to calculate the cooling heating capacity in steady state conditions, but it is not clear how to fix by calculations the limits of the cooling capacity, since a minimum surface temperature in cooling conditions has to be set.

Table 1. Total heat exchange coefficients [W m^{-2} K^{-1}] proposed in the draft of the standard

	heating	cooling
Wall	8	8
Floor	10.8	7
Ceiling	6	10.8

BACKWARDS HEAT FLOW EVALUATION

When sizing a radiant system, reference to the operative temperature has to be made. (i.e. both the radiant and convective heat exchanges with the surroundings are considered). Once calculated the heat loss due to external surfaces, thermal bridges, and the ventilation load, the heating specific capacity of the radiant system is the ratio between the calculated load and the area of the radiant system. The radiant system can be thought as a heat exchanger, having on one side the water and on the other side a medium with infinite heat capacity (the room). The heating capacity is proportional to $(\Delta\vartheta_H)^n$, where n can be approximated to 1, since $1{,}00 < n < 1{,}05$ and where $\Delta\vartheta_H$ is the mean logarithmic temperature:

$$\Delta\vartheta_{\mathrm{H}} = \left(\vartheta_{\mathrm{VH}} - \vartheta_{\mathrm{1H}}\right) / \left(\ln \frac{\vartheta_{\mathrm{VH}} - \vartheta_{\mathrm{RH}}}{\vartheta_{\mathrm{RH}} - \vartheta_{\mathrm{1H}}} \right), \qquad (1)$$

For calculating the heat exchange coefficient in EN 1264 a simplified method, based on parameters reported in tables, is proposed, leading to the equation:

$$K_H = \mathrm{B} \prod_{\mathrm{i}} (\mathrm{a}_{\mathrm{i}}^{\mathrm{m_i}}), \qquad (2)$$

The resulting specific heating capacity of radiant systems is defined by the equation:

$$q_{1H} = \mathrm{B} \prod_{\mathrm{i}} (\mathrm{a}_{\mathrm{i}}^{\mathrm{m_i}}) \cdot \Delta\vartheta_{\mathrm{H}}, \qquad (3)$$

The backwards heat flow can be roughly approximated considering the heat flowing in the adjacent space calculated through the following equation:

$$q_{2H} = U_2 \cdot \left(\vartheta_w - \vartheta_{2H}\right), \qquad (4)$$

The backwards thermal transmittance U_2 may be calculated as inverse of the one dimension resistance of the whole structure from the conditioned room to the adjacent space R_{tot} (or the inverse U_{tot}-value), once known the upward resistance R_1 (or the inverse U_1), which can be calculated via the following equation (q_{1H} is calculated by equation 3):

$$q_{1H} = U_1 \cdot \left(\vartheta_w - \vartheta_{1H}\right), \qquad (5)$$

The thermal transmittance from pipes level to the adjacent room can be calculated as follows:

$$U_2 = \frac{1}{R_2} = \frac{1}{R_{tot} - R_1} = \left(\frac{1}{U_{tot}} - \frac{1}{U_1} \right)^{-1}, \qquad (6)$$

CASE STUDIES

The models used in the analysis are EN1264 method, Finite Element Method (FEM) and Finite Difference Method (FDM) for types A and B systems [2]. FEM can be considered as a reference method, since it allows to model the geometry of a circular pipe; FDM is also a correct method. Usually such models are based on a rectangular mesh and therefore there is the need to correctly model systems containing circular pipes. In heating case the pipe can be approximated by a square having length [6]:

$$\ell = \sqrt{\pi} r, \qquad (7)$$

and internal resistance

$$R_C = 2 \cdot D \cdot \ln[r/(r-d)]/(\pi \cdot k_p), \qquad (8)$$

As detailed methods two commercial software have been used: STRAUSS for FEM and HEAT2 for FDM.

The water inside pipes has been considered in turbulent flow (inner convection resistance negligible) and the same water temperature has been used in the different cases. Pipe material is PE-X (k = 0.35 W m^{-1} K^{-1}). Thermal conductivity of the concrete layer where pipes are embedded has been set to 1.2 W m^{-1} K^{-1}. The structural slab below the systems is 200 mm thick and two values for the thermal conductivity have been fixed: 0.67 W m^{-1} K^{-1} and 1.2 W m^{-1} K^{-1}.

Type A

Simulations have been carried out by considering the data reported in Table 1. For both the conditioned and the adjacent rooms the same temperature (20°C) has been used. Pipes have 20 mm external diameter and 2 mm thick. Pipe distances have been varied from 50 mm to 375 mm. As for the concrete layer and the position of the pipe within it, data can be seen in Table 2. Two values of floor covering materials have been simulated (0.015 m^2 K W^{-1} and 0.20 m^2 K W^{-1} thermal resistance respectively). The insulation layer is 30 mm thick and thermal conductivity is 0.04 W m^{-1} K^{-1}.

Table 2. Thickness of the concrete layer and position of the pipe within it

Thickness of the concrete layer [mm]	Position of the pipe above the insulation [mm]		
60	0	10	30
80	0	20	40

Type B

Simulations have been carried out by using the same boundary conditions of Type A. Two circular pipes have been considered: 20 mm external diameter (2 mm thick) and 10 mm external diameter (1.7 mm thick). Pipes are directly in contact with thermal insulation and distances between pipes have been varied from 50 mm to 450 mm. Three floor covering materials have been simulated (0.015 m^2 K W^{-1}, 0.15 m^2 K W^{-1} and 0.20 m^2 K W^{-1} thermal resistances). The concrete layer above pipes is 60 mm. The insulation layer is 30 mm or 40 mm thick and thermal conductivity is 0.04 W m^{-1} K^{-1}. The additional conductive element above pipes is 0.5 mm thick with 52 W m^{-1} K^{-1} thermal conductivity. Two dimensions have been considered: continuous covering (L = T) or 10 mm distance between each device (L = T – 0.01 m).

RESULTS AND DISCUSSION

Since simulations have been performed under the same conditions and with fixed temperature, in order to generalize the results reference has to be made in each simulation to the results of the FEM method q_{ref}, thus defining the heat flow ratio Ξ:

$$\Xi = q \,/\, q_{ref}, \qquad (9)$$

Type A

Results for the conditioned room heat flow ratio Ξ_1 and the adjacent room heat flow ratio Ξ_2 are reported in Table 3. As it can be seen, FDM gives the same results as FEM. Existing method is also accurate regarding heat flux from pipes to the conditioned room. The adjacent room heat flow calculated via equation (4) is not accurate and precision decreases as the pipes spacing enlarges (Figure 1). The heat flowing in the adjacent room Ξ_2 is dependent most of all on the covering material $R_{\lambda,B}$. Tile covering (0.015 m^2 K W^{-1}) and 20 mm wood covering (0.15 m^2 K W^{-1}) are the two extreme cases, i.e. for each pipes distance and resistance of covering layer, the value is always between the curves represented in Figure 1. It is therefore possible to consider the pipes spacing and the floor covering resistance as parameters for correcting the values of Ξ_2 through the equation (coefficients are reported in Table 4):

$$q_{2H} = U_2 \cdot (\vartheta_w - \vartheta_{2H}) \left[\left(a \cdot R_{\lambda,B}{}^{b} \cdot T\right)^2 + \left(c \cdot R_{\lambda,B}{}^{d} \cdot T\right) + \left(e \cdot R_{\lambda,B}{}^{f}\right) \right], \qquad (10)$$

At the end the proposed equation gives the results reported in the last column of Table 3. The error is slightly higher when pipe distance is great, but the result is on the safety side.

Table 3. Conditioned room heat flux ratio Ξ_1 and adjacent room heat flux ratio Ξ_2 for Type A

	Ξ_1		Ξ_2		
	FDM	EN 1264	FDM	equation (4)	equation (10)
Mean value	0.99	0.99	1.00	1.56	1.00
St. dev.	0.01	0.04	0.01	0.54	0.05

Table 4. Coefficients for evaluating the heat flux in the adjacent room to be used in eq. (10)

a	b	c	d	e	f
$4.2686 \cdot 10^{-2}$	$-7.6348 \cdot 10^{-1}$	-1.1165	$-1.9091 \cdot 10^{-1}$	1.0490	$-1.3619 \cdot 10^{-2}$

Type B

Results for the conditioned room heat flow ratio Ξ_1 and for the adjacent room heat flow ratio Ξ_2 are reported in Table 5. Also in this case FDM gives the same results as FEM. Existing method is accurate regarding heat flux from pipes to the upper room. The heat flow in the adjacent room calculated via equation (4) is not accurate and precision decreases as the pipes spacing enlarges (Figure 2). The heat flow ratio Ξ_2 is depending most of all by the pipes spacing *T* and external diameter *D* of the pipe, but it does not depend on insulation layer thickness. It is possible to write the following equation for determining the heat flux towards the adjacent room (coefficients are reported in Table 6):

$$q_{2H} = U_2 \cdot (\vartheta_w - \vartheta_{2H}) \left[\left(a \cdot D^{b} \cdot T\right)^3 + \left(c \cdot D^{d} \cdot T\right)^2 + \left(e \cdot D^{f} \cdot T\right) + \left(g \cdot D^{h}\right) \right], \qquad (11)$$

In this way for each pipes spacing the curve is always between the curves reported in the diagram of Figure 2. The proposed equation gives the results reported in the last column of Table 5. The error is slightly higher when pipe distance is great, but also heat flow in the conditioned room is not very accurate in these cases.

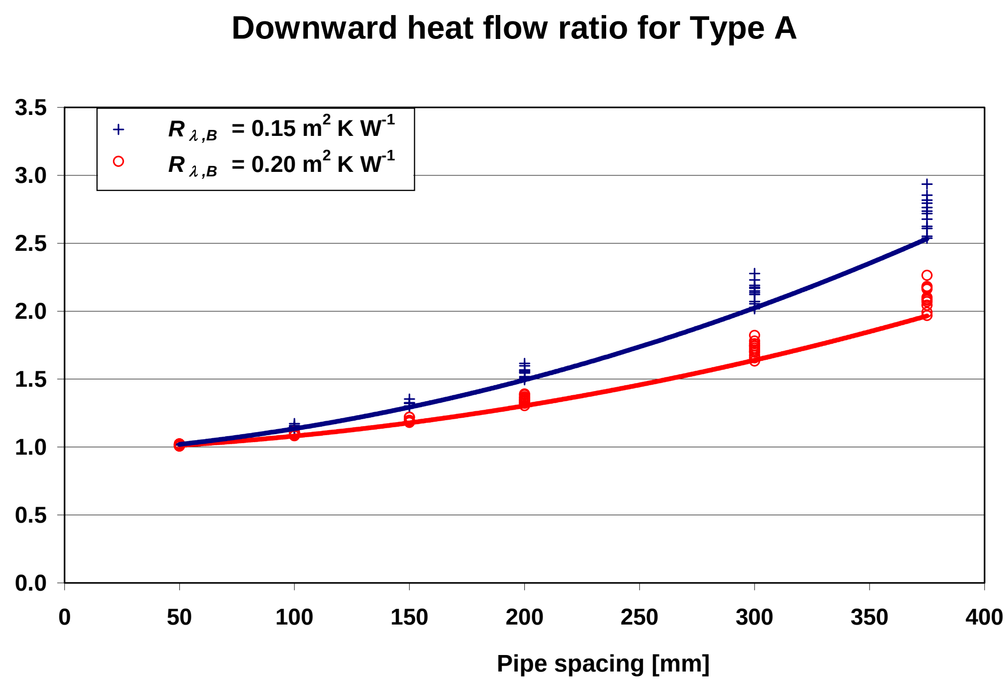

Figure 1. Values of Ξ_2 in the case of existing method based on equation (4) for Type A.

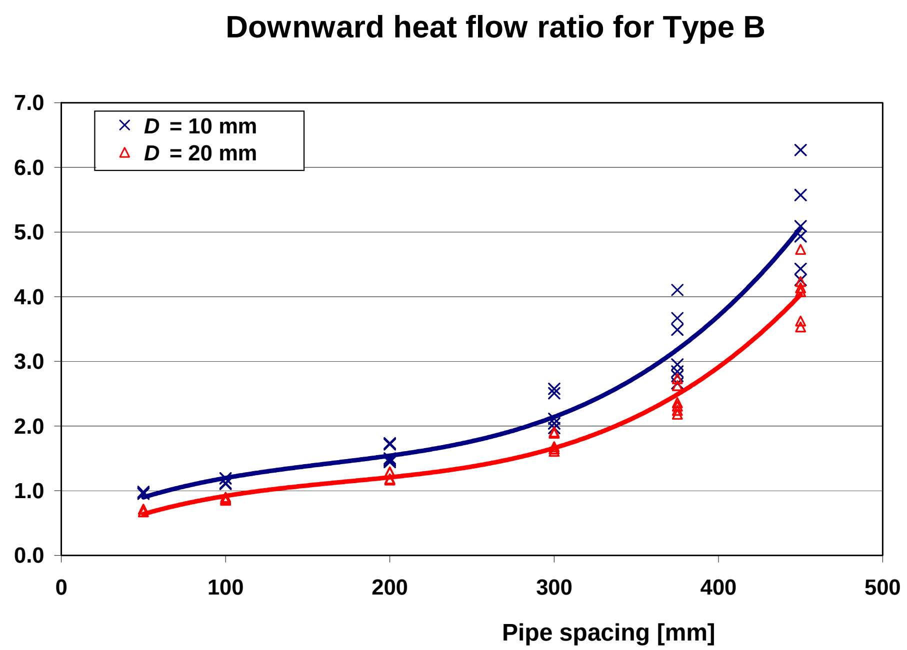

Figure 2. Values of Ξ_2 in the case of existing method based on equation (4) for Type B

Table 5. Conditioned room heat flux ratio Ξ_1 and adjacent room heat flux ratio Ξ_2 for Type B

	Ξ_1		Ξ_2		
	FDM	EN 1264	FDM	equation (4)	equation (10)
Mean value	1.01	0.92	0.99	2.21	0.98
St. dev.	0.00	0.13	0.00	1.29	0.10

Table 6. Coefficients for evaluating the adjacent room heat flux to be used in eq. (11)

a	b	c	d	e	f	g	h
-39.146	0.68386	149.60	0.90524	-612.77	1.1118	17.617	0.58472

MINIMUM TEMPERATURE IN COOLING CONDITION ANALYSIS

The existing EN 1264 takes into account the heat output from the radiant floor and it fixes a surface temperature which cannot be exceeded. For radiant floor 29°C for the occupancy space, 35°C for outside area. For radiant ceiling the surface temperature is related to the radiant asymmetry, but it is usually better not to exceed 35°C. For radiant walls the maximum allowed temperature is related to the risk of burning, i.e. about 40°C. Maximum surface temperature is the average surface temperature, i.e. the maximum surface temperature above or below the pipe in the section where water temperature is equal to:

$$\vartheta_{\mathrm{w}} = \vartheta_{1\mathrm{H}} + \Delta\vartheta_{\mathrm{H}}, \qquad (12)$$

When calculating the thermal output of a radiant floor in cooling conditions, the minimum allowable surface temperature must be taken into account. Such temperature for a radiant floor is 19°C, while for the other radiant systems the dew point temperature is the reference temperature. It must be underlined that such surface temperature is above the section where the supply water temperature occurs. The question is how to consider in a proper way the minimum temperature, once it is known the maximum output of the radiant system, since the cooling capacity has to be determined in the section where water temperature is equal to:

$$\vartheta_{\mathrm{w}} = \vartheta_{1\mathrm{C}} - \Delta\vartheta_{\mathrm{C}}, \qquad (13)$$

The idea is to evaluate, for the maximum heating capacity, the average temperature of the surface and determine the difference between such temperature and the maximum surface temperature. Therefore the following equation can be written:

$$\overline{\Delta\vartheta_{\mathrm{H}}} = \vartheta_{\mathrm{surf,max}} - (\vartheta_{1\mathrm{H}} + \mathrm{q}_{\mathrm{H}}/\alpha_{H}), \qquad (14)$$

At the same time for the cooling capacity a similar equation can be calculated:

$$\overline{\Delta\vartheta_{\mathrm{C}}} = (\vartheta_{\mathrm{iC}} - \mathrm{q}_{\mathrm{C}}/\alpha_{C}) - \vartheta_{\mathrm{surf,min}}, \qquad (15)$$

Case studies

Detailed simulations have been carried out for different systems with FDM, since the accuracy of the method has been previously demonstrated. The analysis has been carried out with the reference limit temperatures reported in Table 7. Simulations have been carried out by considering for the heat transfer coefficients the data reported in Table 1. The water inside pipes has been considered in turbulent flow (i.e. convective inner resistance negligible). Pipes have 20 mm external diameter and 2 mm thick. Material is PE-X (k = 0.35 W m^{-1} K^{-1}).

Table 7. Surface temperature limits [°C] considered in this work

	heating	**Cooling**
Wall	40	19
Floor	35	19
Ceiling	29	19

Type A

Internal resistance due to water convection in the pipe has been considered negligible Pipe distances have been varied from 50 mm to 300 mm. The concrete layer is 60 mm thick and the position of the pipe within it has been set above insulation layer or in the centre of the concrete layer. Three values of floor covering materials have been simulated: 0 m^2 K W^{-1}, 0.015 m^2 K W^{-1} and 0.20 m^2 K W^{-1} thermal resistance respectively. Thermal conductivity of the concrete layer where pipes are embedded has been set to 1.2 W m^{-1} K^{-1}. The insulation layer is 30 mm thick and thermal conductivity is 0.04 W m^{-1} K^{-1}. The structural slab is 200 mm thick with 0.9 W m^{-1} K^{-1} thermal conductivity.

Type B

In this case pipes are directly in contact with thermal insulation and distances between pipes have been varied from 50 mm to 300 mm. Four floor covering materials have been simulated, having thermal resistances 0 m^2 K W^{-1}, 0.015 m^2 K W^{-1}, 0.15 m^2 K W^{-1} and 0.20 m^2 K W^{-1}. The concrete layer above pipes is 70 mm thick and thermal conductivity has been set to 1.2 W m^{-1} K^{-1}. The insulation layer is 30 mm thick and thermal conductivity is 0.04 W m^{-1} K^{-1}. The structural slab is 200 mm thick with 0.67 W m^{-1} K^{-1} thermal conductivity. The additional conductive element above pipes is 0.5 mm thick and its thermal conductivity is 52 W m^{-1} K^{-1}. The dimension of the conductive covering equal to the pipes spacing ($L = T$).

Results and discussion

The two temperature differences given by equations (12) and (13) have the same value only when the overall heat transfer coefficient is the same for heating and cooling and if the following condition is valid:

$$\vartheta_{1C} - \vartheta_{\mathrm{surf,min}} = \vartheta_{\mathrm{surf,max}} - \vartheta_{1H}\,, \qquad (16)$$

Since equation (16) cannot be achieved and the heat transfer coefficients are not always the same for heating and cooling conditions, the error Λ can be calculated as follows:

$$\Lambda = \overline{\Delta\vartheta_{C}} - \overline{\Delta\vartheta_{H}}\,, \qquad (17)$$

Results of Λ for a set of simulations are reported in Figure 3 for type A and in Figure 4 for Type B. The overall thermal resistance from pipes level to conditioned room R_{tot} as well as the pipes spacing are the factors that mostly influence the results. The hypothesis of a linear relationship between Λ and pipes space T leads to the following equation:

$$\Lambda = -a \cdot R_{tot}^{-b} \cdot T + c \cdot R_{tot}^{-b}\,, \qquad (18)$$

where the coefficients are reported in Table 8 for the different cases.
In this case the minimum temperature can be directly calculated, once known the maximum allowable heating output for all configurations of radiant systems:

$$q_{CV} = \alpha_C \cdot \left(q_H / \alpha_H - \Lambda - \vartheta_{surf,\max} + \vartheta_{1H} - \vartheta_{surf,\min} + \vartheta_{1C}\right), \qquad (19)$$

where q_{VC} means the cooling output in the section of the supply temperature (i.e. in the section where the water enters in the room). Equation (19) can be solved for the different systems, leading to:

$$q_{CV} = \alpha_C \cdot \left(q_H / \alpha_H - \Lambda - p\right), \qquad (20)$$

where p = 2°C for a radiant floor, p = 13°C for a radiant wall and p = 8°C for a radiant ceiling.

Once known the heat flow q_{CV}, the calculation of the supply water temperature is possible, since K_c is known:

$$\vartheta_{VC} = \vartheta_{1C} - q_{CV} / K_C, \qquad (21)$$

From the supply water temperature, once fixed the maximum temperature difference between supply and return, it is possible to evaluate the maximum cooling output of the system q_C.

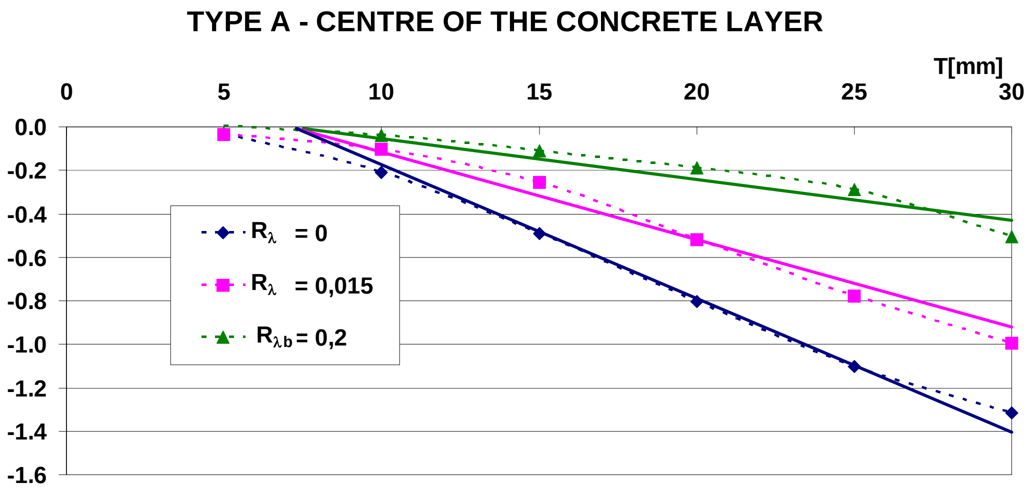

Figure 3. Values of Λ [°C] for Type A (floor)

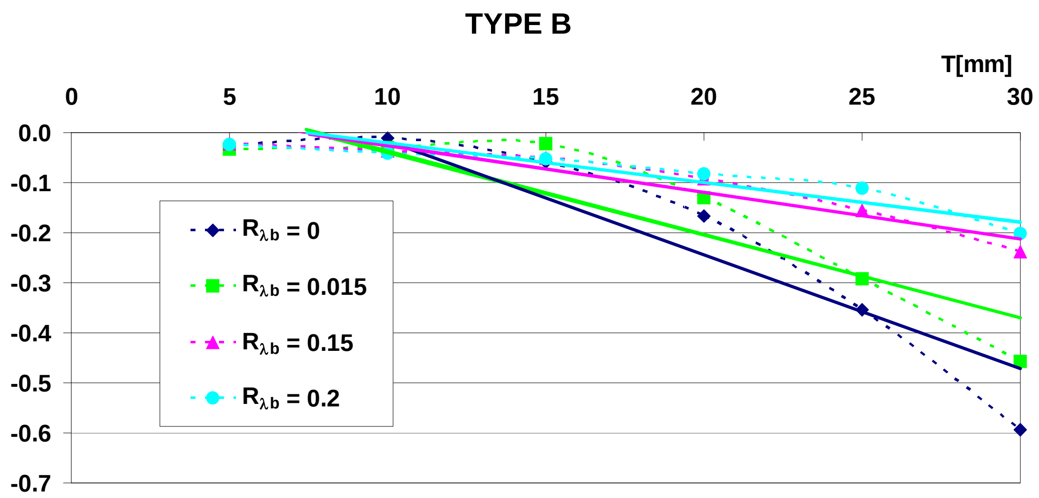

Figure 4. Values of Λ [°C] for Type B (floor)

Table 8. Coefficients to be used in equation (18)

		a	B	c
Floor	A – 30 mm above insulation	-0.88	-0.501	0.066
	A – 0 mm above insulation	-0.61	-0.597	0.458
	B	-0.34	-0.628	0.026
Wall	A – 30 mm above insulation	-3.56	-0.412	0.267
	A – 0 mm above insulation	-2.61	-0.502	0.235
	B	-1.18	-0.590	0.112
Ceiling	A – 30 mm above insulation	-1.87	-0.332	0.140
	A – 0 mm above insulation	-1.52	-0.373	0.144
	B	-0.78	-0.398	0.086

CONCLUSIONS

When pipes are approximated thorough equations (2) and (3) the use of FDM leads to no error and the accuracy can be assumed to be the same as FEM. As for hating flow behind the pipes, both for type A and B, correlation for determining the correct values are proposed and seem to be accurate (error less than 2% in the worst case for type B). Also modification for taking into account minimum temperature of supply water temperature in cooling conditions, starting by the value (calculated or measured) of the maximum allowable heating output, has been proposed and it seems to be accurate enough (on cooling heat flux less than 2%).

SYMBOLS

α	overall heat flow coefficient from surface to space
a_i	multiplication factors which depend on thermal and geometrical characteristics of the radiant system
B	coefficient characteristic of the radiant system
D	external diameter of the pipe
d	thickness of the pipe
k	thermal conductivity
k_p	thermal conductivity of the pipe
ℓ	equivalent length of the square approximating the pipe for FDM simulations
L	length of conductive layer above pipes for type B radiant systems
m_i	exponential factors which depend on thermal and geometrical characteristics of the radiant system
q_{ref}	reference heat flow from FEM in the considered simulations
r	external radius of the pipe
R_C	equivalent internal resistance of the pipes for FDM simulations
$R_{\lambda, B}$	resistance of the covering layer
T	distance between pipes
U_1	thermal transmittance from pipes level to conditioned space
U_2	thermal transmittance from pipes level to backwards
$\Delta\vartheta$	mean logarithmic temperature
ϑ_V	supply temperature of the water
ϑ_R	return temperature of the water
ϑ_W	mean temperature of the water
ϑ_1	temperature of conditioned room
ϑ_2	temperature of backwards room

$\vartheta_{\mathrm{suf,max}}$ maximum average surface temperature
$\vartheta_{\mathrm{suf,min}}$ minimum surface temperature at supply point
$\overline{\Delta\vartheta}$ difference between surface average and maximum/minimum surface temperature
Ξ heat flow ratio
Λ error when considering $\overline{\Delta\vartheta}$ in heating and cooling case
$_C$ cooling case
$_H$ heating case

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Estimation Method of Energy Consumption of Hot Water Radiant Heating System

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SUMMARY

The purpose of this paper is to develop the calculation method of energy consumption for hot water radiant heating system. The method was developed based on 40 experiments which were carried out in a wooden house built in an environmental chamber in the Building Research Institute (BRI). The method was steady state calculation for simplicity, of which necessary input information was limited to the one we can get easily in design stage such as catalogue value and design plan, besides the heating load calculated by simulation. Therefore, by using this method, we can design the heating system or use for rating of the system for energy saving. The calculation results of the energy consumption were compared with the measured results and then it was shown that these results agreed well with various condition.

INTRODUCTION

Up to now, in Japan, energy performance of residential buildings for heating and cooling has been evaluated by the heat loss coefficient and air tightness of the envelope. On the other hand, as residential heating/cooling equipments such as room air conditioner and radiant heating system have become more efficient, design of these equipments as well as the envelope has become more important for energy savings. Therefore, energy calculation methods for these various equipments have been required in order to compare and rating the energy performance of various heating systems based on primary energy consumption. In this paper, a calculation method of hot water radiant heating system is presented.

OUTLINE OF THE CALCULATION METHOD

Annual energy consumption of hot water radiant heating system can be calculated by using the method presented here. Before this energy calculation, heating load needs to be calculated by simulations. Necessary input information besides the heating load was limited to the one we can get easily in design stage such as catalogue value and design plan (Figure 1).

DEVELOPMENT OF THE CALCULATION METHOD

Outline

The method was developed on the basis of experiments. Various hot water radiant heating systems, of which components such as radiator, boiler, and hot water circulating piping were different, were constructed in a wooden house built in an environmental chamber in the Building Research Institute (BRI), and then heat loss and heat supply from the components were measured under various heat loads by changing the thermal insulation performance of the envelope, ventilation rate and external temperature. The method was based on the steady-state data for simplicity although heat flows and temperatures are usually changing. Therefore, experimental room was controlled to be steady state and then the data was used when all temperatures were constant.

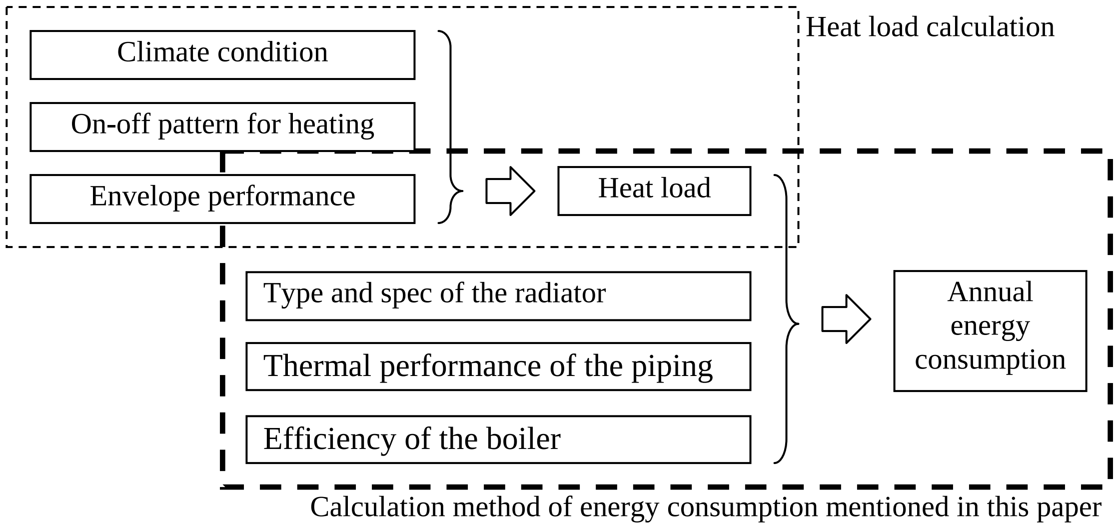


Figure 1. Necessary Input Information for Calculation Method

Experiment

1) Experimental House

A radiant floor panel was constructed at the 1st floor of a two-story wooden house (Fig. 2 and 3) in a chamber which temperature and humidity could be controlled. This wooden house is equipped with central ventilation system by which air supply and exhaust rate can be controlled. Thickness of the insulation at the floor and the wall can be changed.

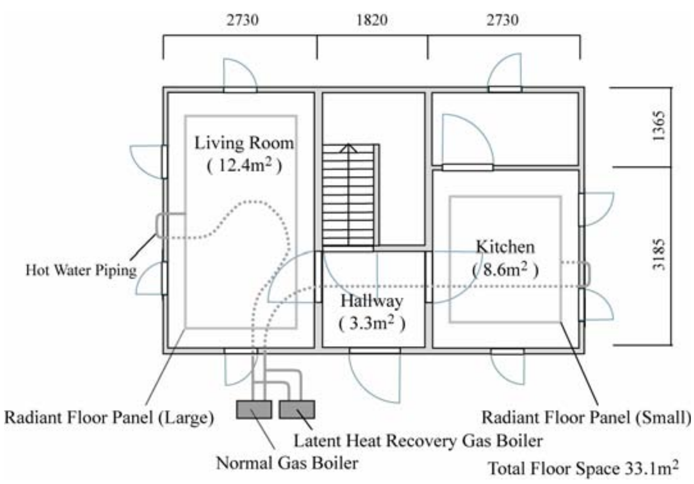


Figure 2. Floor Plan

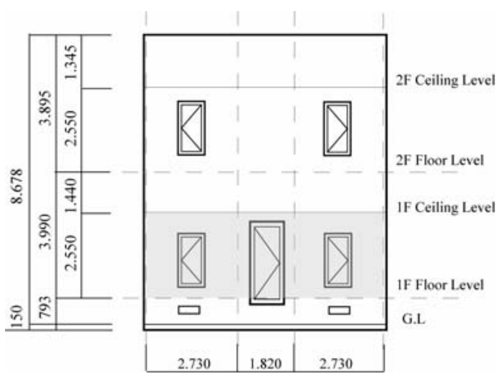


Figure 3. Elevation View

2) Floor Heating System

A normal gas boiler and a latent heat recovery gas boiler were tested, which were used for both floor heating and hot water supply. Supply water temperature of the normal gas boiler

for heating are 60 ^oC and that of the latent heat recovery gas boiler are 40 or 60 ^oC which can be selected with the controller. Even if 40 ^oC were selected, supply water temperature is controlled to be 60 ^oC when room temperature does not reach set temperature. Output of both boilers for heating is 2.56 ~ 14.0 (kW).
Two radiant floor panels were installed in the living room and the kitchen (Fig.2). Figure 4 shows the cross section of the panel.
Hot water piping consisted of two tubes for supply and return water which were bound up. Two types of the piping were used, which were insulated type and non-insulated type. The heat loss coefficient of non-insulated type was 0.21 (W/mK) and one of insulated type was 0.16 (W/mK). The length of the piping between the boiler and the panels was 10m.

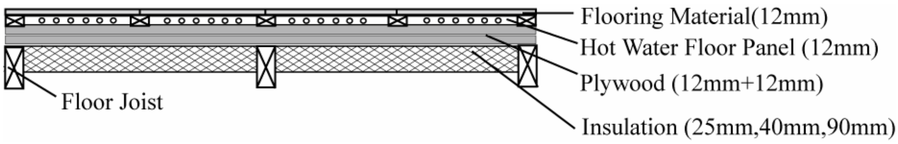


Figure 4. Cross Section of Radiant Floor Panel

3) Parameters of the experiments
The experimental parameters shown in Figure 5 were 1) type of the boiler and hot water supply temperature, 2) type of the piping, 3) ambient (chamber) temperature and 4) thermal performance of wooden house which was established with changing thickness of the floor insulation and ventilation rate of the rooms. With combining these parameters, 40 experiments were carried out.

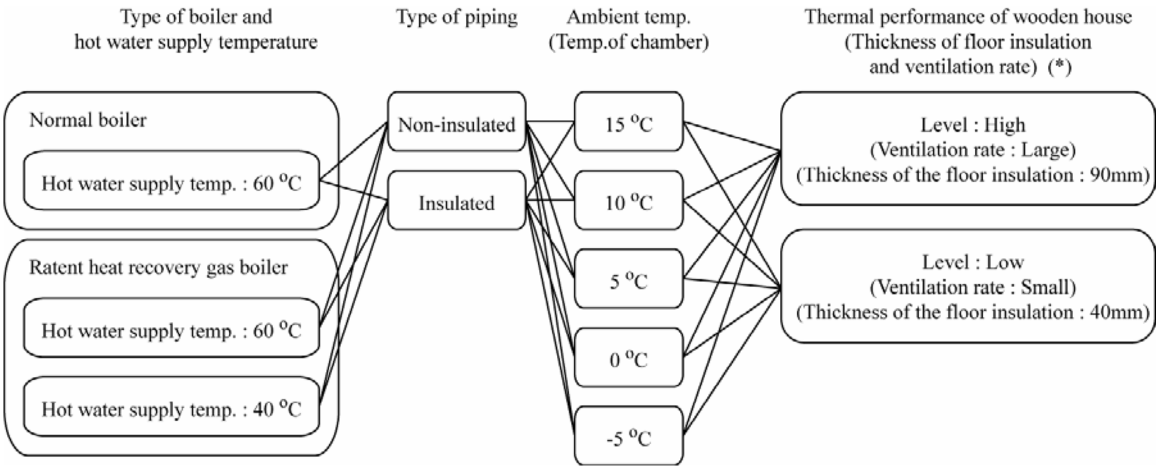


Figure 5. Combination of Experimental Patterns

* Heat loss coefficient per floor area of “level high” was 4.2 (W/m^2K) with ventilation rate 17.5 (m^3/h) and that of “level low” was 2.7 (W/m^2K) with ventilation rate 63(m^3/h).

4) Time of the operation and the measurement
The floor heating was operated for 6~11 hours until the temperatures of the floor and the panel became constant. The measurement was started an hour before the start of the operation and stopped 3 hours after the operation was stopped in order to observe the decrease of the temperatures.

5) Experimental results
Among 40 experiments, one result is shown in Figure 6, which was carried out under the situation that the boiler was normal gas boiler, the piping was non-insulated and the external temperature was 5 ^oC. Supply water temperature was kept about 75 ^oC for 60 minutes after

starting the operation to raise the floor surface temperature rapidly. The water from the boiler was intermittently supplied at about 60 ^oC and the cycle of on-off operation was 20 minutes. For 120-180 minutes, water was supplied for 15 minutes and stopped for five minutes as well as combustion of the gas boiler. After 170 minutes when the room temperature reached the set temperature 20 ^oC, the operation time shortened gradually. The floor surface temperature was constant at about 31 ^oC and moved somewhat up and down by on-off of the water supply. The supply water temperature at the input of the panel was approximately 3 ^oC lower than that at the output of the boiler and the decrease of the supply water temperature was shown. On the other hand, the decrease of the return water temperature was not found because the return water seemed to receive the heat from the supply water, which temperature was higher than that of the return water, in the pair tubes.

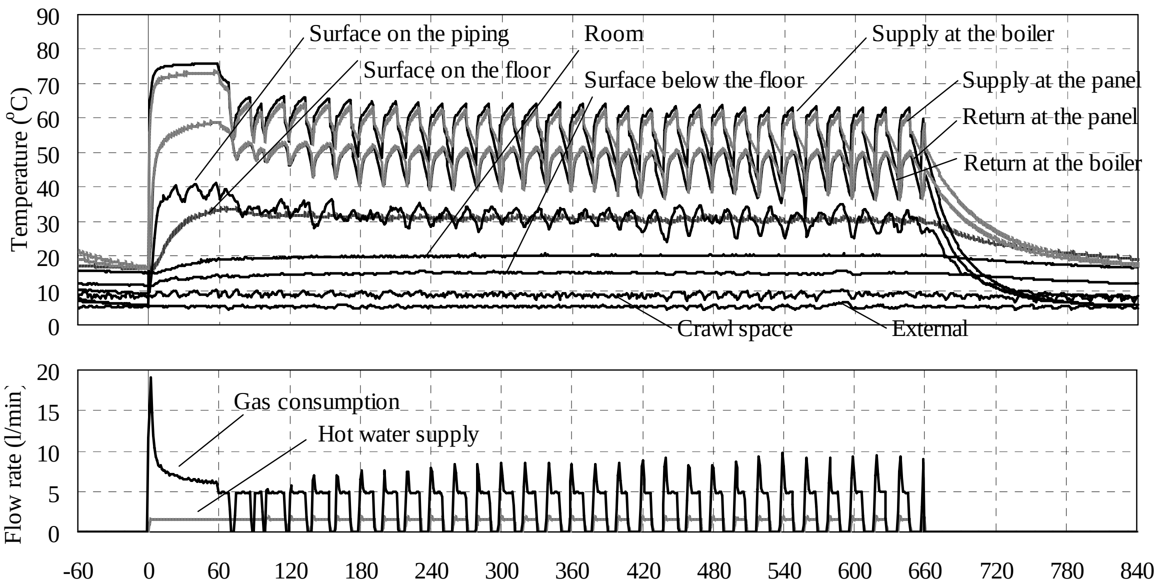


Figure 6. Experimental Result

Heat supply to the room, heat loss from the panel, heat loss from the piping and heat loss from the boiler which were defined in Figure 7, were calculated. The results were shown in Figure 8. The heat supply to the room as well as the heat loss from the piping increased as the external temperature lowered. The heat supply was influenced by the external temperature, e.g. 300~700 (W) with the external temperature 15 ^oC and 1500~2000 (W) with the external temperature -5 ^oC. On the other hand, heat loss from the boiler was not influenced well by the external temperature and heat supply to the room, and it was constant at 500~700 (W) at the normal gas boiler and at 250~400 (W) at the latent heat recovery gas boiler.

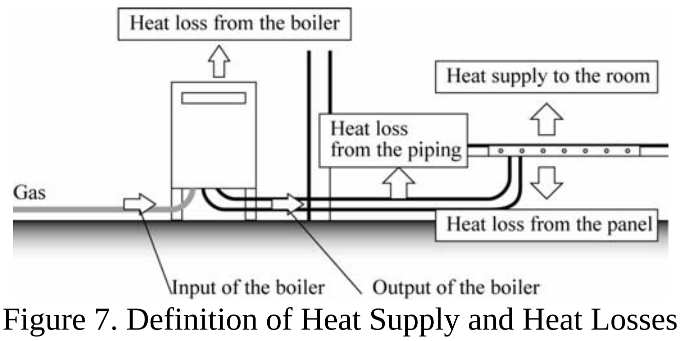
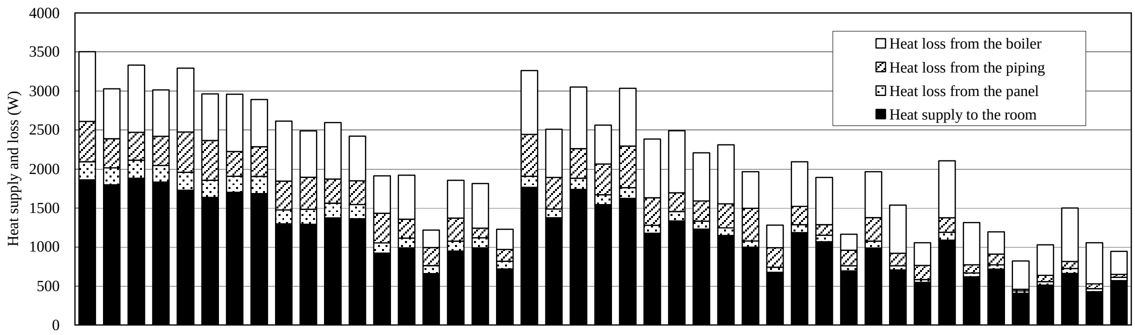


Figure 7. Definition of Heat Supply and Heat Losses



Type of the boiler *1	N 60	L 60	N 60	L 60	N 60	L 60	N 60	L 60	N 60	L 60	N 60	L 60	N 60	L 60	L 40	N 60	L 60	L 40
Type of the piping *2	N		I		N		I		N		I		N			I		
External temp.	-5°C				0°C				5°C				10°C					
Level of thermal performance of the room	Low (Thickness of the insulation = 40mm)																	

Type of the boiler *1	N 60	L 60	N 60	L 60	N 60	L 60	N 60	L 60	N 60	L 60	L 40	N 60	L 60	L 40	N 60	L 60	L 40	N 60	L 60	L 40	N 60	L 60	N 60	L 60	L 40
Type of the piping *2	N		I		N		I		N			I			N			I			N		I		
External temp.	-5°C				0°C				5°C						10°C						15°C				
Level of thermal performance of the room	High (Thickness of the insulation = 90mm)																								

*1 N60:Normal gas boiler
L60:Latent heat recovery gas boiler of which supply temperature was 60 °C
L40:Latent heat recovery gas boiler of which supply temperature was 40 °C

*2 N:Non-insulated piping
I:Insulated piping

Figure 8. Heat Supply and Heat Loss

Development of the estimation method of the energy consumption

Energy consumption (E) of the boiler is given by,

$$E = \frac{1}{e_{hs}}\left(q_{pnl} + q_{pipe}\right), \tag{1}$$

where:

e_{hs} efficiency of the boiler;
q_{pnl} heat supply to the panel in W;
q_{pipe} heat loss from the piping in W.

Heat supply to the panel is given by,

$$q_{pnl} = q_{pnl,spy} + q_{pnl,loss} = \left(\frac{T_{pnl} - T_{room}}{R_u} + \frac{T_{pnl} - T_{crawl}}{R_d}\right) A_{fp} \tag{2}$$

where:

$q_{pnl,spy}$ heat supply from the panel to the room in W;
$q_{pnl,loss}$ heat loss from the panel in W;
T_{room} room temperature in ^{o}C;
Tcrawl crawl space temperature in ^{o}C;
R_u thermal resistance between the panel and room in m^{2}K/W;
R_d thermal resistance between the panel and crawl space in m^{2}K/W;
A_{fp} area of radiant floor panel in m^2.

Equation (2) can be modified without using T_{pnl} as follows;

$$\begin{aligned} q_{pnl} &= q_{pnl,spy} + \frac{1}{R_d}\left(q_{pnl,spy} \times R_u + T_{room} - T_{crawl}\right) \\ &= \frac{R_u + R_d}{R_d}\left(q_{pnl,spy} + \frac{T_{room} - T_{crawl}}{R_u + R_d}\right) \end{aligned} \tag{3}$$

In the parentheses in Equation 3, the first term represents heat supply from the panel to the room and this equals the loss from the room to the outside through the envelope except the floor (L_{env}). The second term represents heat loss from the room to the crawl space through the floor (L_{pnl}) when floor heating is not operated. Therefore, sum of the first term and the second term equals to the heat load (L) calculated heat load calculation and Equation 3 can be modified as follows;

$$q_{pnl} = \frac{R_u + R_d}{R_d}\left(L_{env} + L_{pnl}\right) = \frac{R_u + R_d}{R_d} L \qquad (4)$$

The calculated results of the heat supply to the panel as mentioned above were compared with the measured results in Figure 9. The calculated results agree well with the measured results.

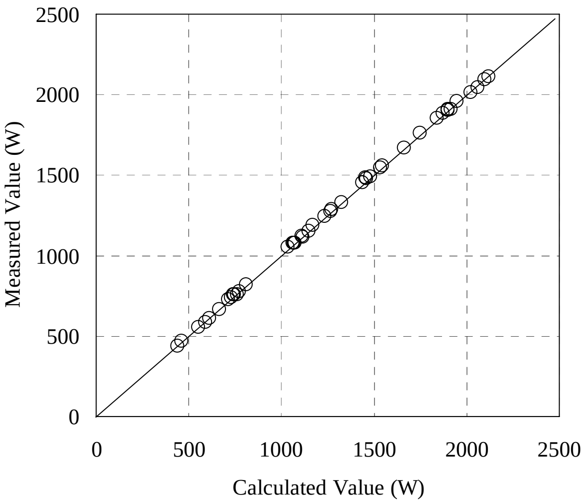


Figure 9. Comparison of q_{pnl}

Heat loss from the piping is given by,

$$q_{pipe} = K_{pipe} \times \left(T_{water,ctrl} - T_{crawl}\right) \times d \times l_{pipe} \qquad (5)$$

where:

K_{pipe}	thermal coefficient of the piping in W/mK,
$T_{water,ctrl}$	set temperature of supply water in ^{o}C,
T_{crawl}	crawl space temperature in ^{o}C,
d	decreasing rate of average water temperature,
l_{pipe}	length of the piping in m.

Decreasing rate of the average water temperature (d) is the coefficient which represents that average heat loss decreases when on-off operation because water temperature decreases when water supply stops and which is given by,

$$d = r + \frac{1 - e^{-\left(K_{pipe} h (1-r)/c\right)}}{K_{pipe} h / c} \qquad (6)$$

where:

r	operating rate which is the ratio of the water supply period,
c	thermal capacity of the water and pipes in J/mK
h	on-off operation cycle in s.

It is difficult to define operating rate (r) qualitatively because it depends on how to control the boiler. In this paper, r was calculated from the heat supply to the panel q_{pnl} conveniently assuming that r is relative to q_{pnl} which shows in Figure 10. Figure 11 shows comparison between the calculated heat loss from the piping and measured one. Calculated results do not agree well with the measured one. The difference between the average water temperature and the crawl space temperature could be explained well, so the assumption that Kpipe was constant did not seem to be relevant.

Figure 12 shows the measured efficiency of the boiler. When the output was higher than the lower limit of the boiler 2.56kW, the boiler ran continuously and the efficiency was constant. On the other hand, when the output was lower than 2.56kW, the boiler ran intermittently and the efficiency decreases. In this report, in case the output is lower than the lower limit, approximately curve of the efficiency was made on the assumption that heat loss was constant as below.

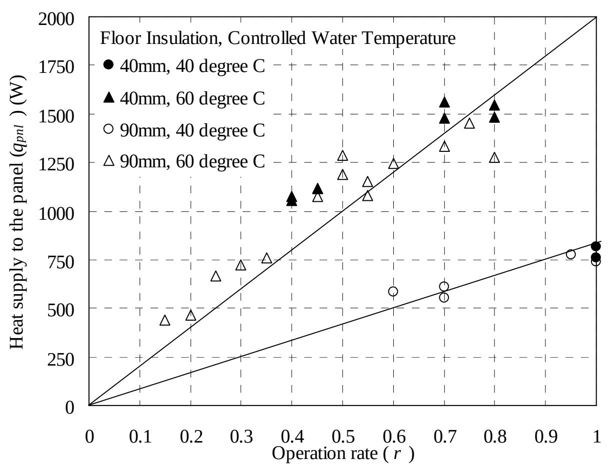


Figure 10. Relation between q_{pnl} and r

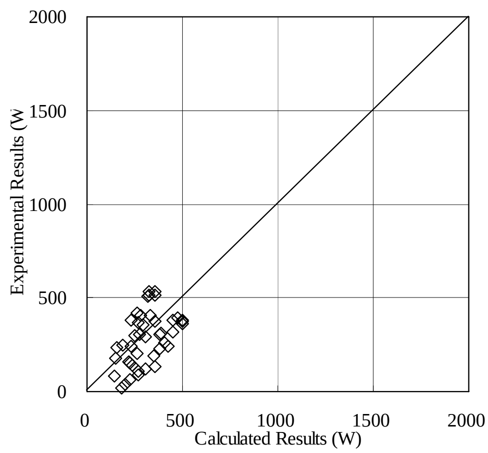


Figure 11. Comparison of Heat Loss from Piping

Table 1. Heat Loss from Boiler (which assumed in the calculation)

Normal gas boiler with output water temperature 40 ^{o}C	853 (W)
Latent heat recovery gas boiler with output water temperature 60 ^{o}C	640 (W)
Latent heat recovery gas boiler with output water temperature 40 ^{o}C	320 (W)

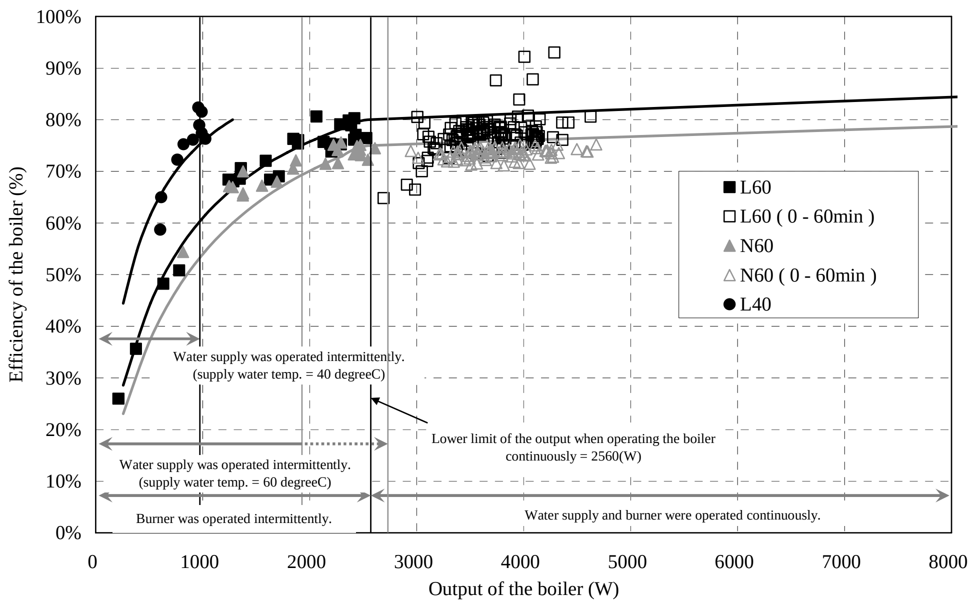


Figure 12. Efficiency of Boiler

Procedure of the calculation

Figure 13 shows the procedure to calculate the primary energy consumption from the heat load calculated by simulation and design value of the floor heating.

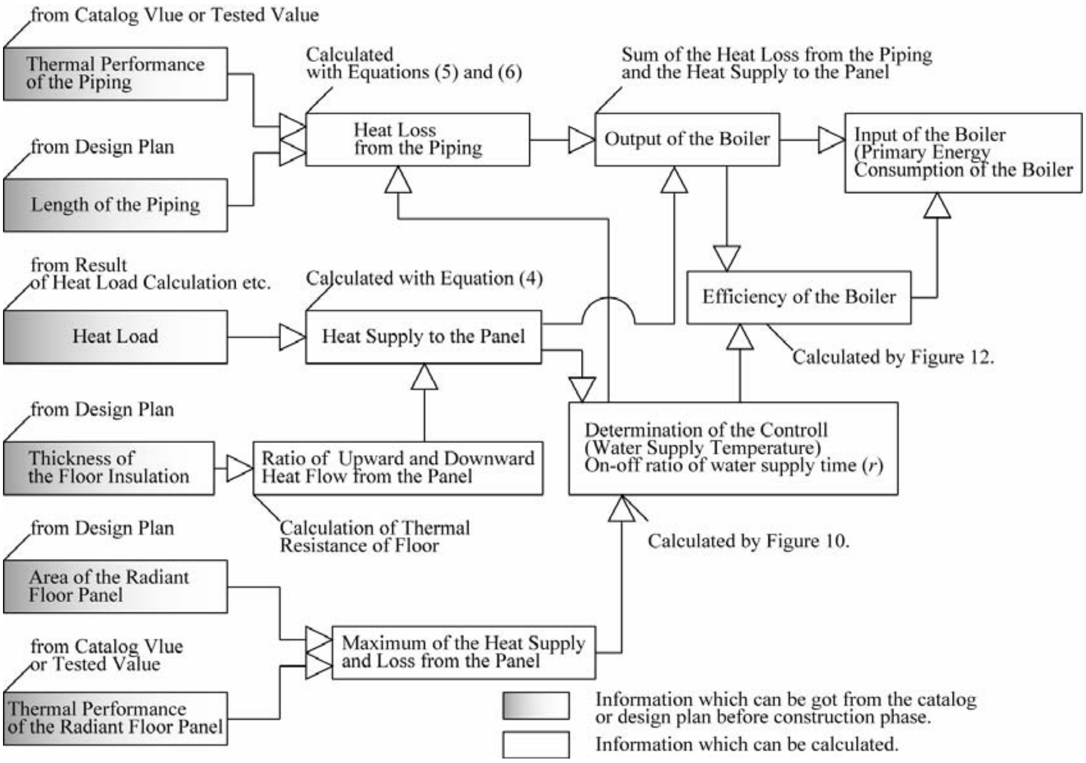


Figure 13. Procedure to Calculate Primary Energy Consumption

Comparison of the calculated energy consumption with measured one

Energy consumption which was calculated as mentioned above was compared measure one (Fig. 14). The calculated results agree well with the measured results.

CONCLUSION

The estimation method was developed based on the 40 experiments. The calculation results of the energy consumption agreed well with the measured one. This calculation needs input we can get by the heat load calculation and catalog value. Therefore, by using this method, we can decide which heating system should be used such as radiant heating, room air conditioner and so on, and in case of floor heating, we can decide the capacity of the boiler, type of the piping, thickness of the insulation of the floor and so on. This calculation can also be used for the rating of the heating system for energy saving.

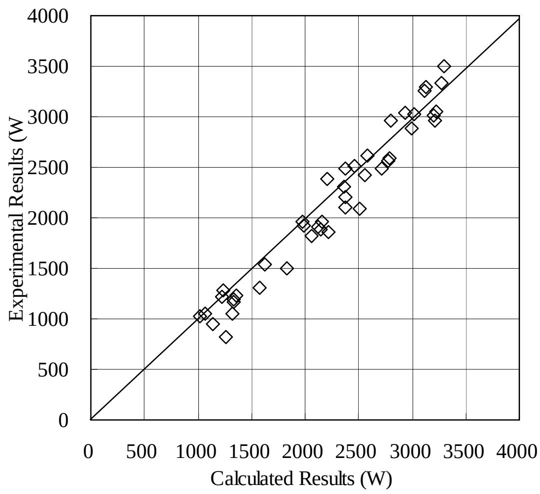


Figure 14. Comparison of Primary Energy Consumption

Energy consumption can not be calculated by this method a few minutes after the system on because the developed method was steady state calculation. In Japan, intermittent air-conditioning is more popular than continuous air-conditioning. Therefore, it is necessary to develop the non steady state calculation of the energy consumption.

Annual Variability of Energy Usage in a Small Office Building Due to Weather

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SUMMARY

This paper gives an indication of the range of natural variability in energy usage of a particular building due solely to the variability in the outside weather conditions. The annual total, heating, and cooling energy usage of a small light frame office building was determined from 30 years of hourly weather data for 15 locations representing a cross section of the eight climate zones in the USA and the Typical Meteorological Year (TMY2) for the same 15 locations and for seven locations with Weather Year for Energy Calculations (WYEC). It was found that for the 15 cities, the 30-year range of the total annual energy usage averaged 18% of the 30-year average annual energy usage. The 15-city average coefficient of variation (standard deviation/average) for heating was 31.7% and 8.1% for cooling. The range over average was 132% for heating and 33% for cooling.

Substantial differences were found when the energy usages calculated from WYEC and TMY data were compared to the 30-year statistics for each location for the annual total, cooling and heating energy usages. Comparing the TMY or WYEC to the 30-year average by locations, there were 10 of 22, 21 of 22, and 12 of 21 cases which were outside the average plus or minus one standard deviation for total, cooling and heating annual energy usage respectively.

INTRODUCTION

Computer modeling of buildings is now a common technique to size heating and cooling equipment and to evaluate the potential energy savings of design alternatives. In order to simplify the process and speed the computations, a “typical” weather year is usually used in this modeling. Even though there has been considerable research in developing these “typical weather years”, unfortunately using a single synthesized weather year does not give an indication of the range or variability in the year-to-year differences in weather. This naturally occurring between-year variation is extremely important when trying to make comparisons between pre- and post- retrofit energy data. These variations can lead to disappointments when actual alternatives have been put into buildings and expected “savings” are not realized due to variation in the actual data from the weather data used in the estimation. The purpose of this paper is determine the range of natural variability in energy usage of a small light-frame office building due to the variability in the outside weather conditions and to evaluate how well the synthesized weather years compare to actual weather data for this particular type and size of building.

METHODOLOGY

The annual energy usage and extreme monthly electricity and gas demand of a small light-framed office building was determined for 15 locations representing the eight climate zones in the US using 30 years of hourly weather data. This information was also determined using the TMY2 weather data for each of the cities and for seven of the locations which had WYEC weather data sets available.

Description of Test Building

The building simulated was a single story, frame construction 455m^2 (4,900 ft^2) office building. It was arranged in a square plan form to avoid any orientation issues. The floor was a concrete slab with R-0.9, 0.6m (R-5, 2-ft) wide edge insulation. The walls were 2x4 0.4m (16”) on-center light frame construction. The window area was a 20% window-to-wall area ratio and the windows were equally distributed around the four walls. There were no overhangs or external shading devices. Insulation, window and door specifications, and equipment efficiencies were all as specified by ANSI/ASHRAE/IESNA Standard 90.1-1999. Wall and roof R-values were R-2.29 and R-5.28 (R-13 and R-30) respectively for all locations except for Fairbanks where the walls were R-2.29+R-1.32 (R-13+R-7.5) continuous insulation and a roof of R-6.7 (R-38) was used in Fairbanks and Helena. Occupancy sensors and outdoor air damper controls were not implemented. The occupancy, equipment, and lighting schedules are given in Colliver and Jarnagin [1]. Heating and cooling were supplied with a single zone packaged DX rooftop unit with gas heating. The efficiency of the equipment was the minimum specified in Std 90.1-1999 (AFUE-0.74 and EER-8.7). The heating setpoints were 21.1C (70F) during occupied periods, 18.3C (65F) unoccupied and the cooling setpoints were 23.9C (75F) occupied, 26.7C (80F) unoccupied.

Description of Locations Simulated

The US Department of Energy has identified eight climate zones for the United States. Weather data from the 15 cities presented in Figure 1 were used to simulate the energy usage. These fifteen cities were chosen to represent a wide variety of climate zones including a variation in temperature and humidity. Both dry and humid conditions were chosen for several of the climate zones.

Weather Data Sources

Thirty years of hourly weather data for each of the locations were obtained from the Solar and Meteorological Observation Network (SAMSON) [2]. An exception was Baltimore which had only had 28 years of data which could be converted into an acceptable format. The time period covered by these data was 1961-1990. More recent data are currently being used to generate a newer version of SAMSON, however these updated datasets are not yet available. The data were converted from the SAMSON format to the .bin format used by the simulation with a utility provided with the energy simulation program. An investigation of any long-term trend of the annual data to investigate any impacts of climate change is not included in this paper.

Rather than simulating multiple years of weather data, annual energy usage is typically estimated with some generated “typical” year of weather data which is supposed to represent the long-term weather conditions of the location. There are several types of these “typical” weather years such as: a) Test Reference Year (TRY), b) Typical Meteorological Year (TMY or TMY2 [3]), and c) Weather Year for Energy Calculations (WYEC2 [4] or IWEC [5]). All of these data sets have been generated by combining various months of actual weather data into a set of 8760 hours of weather data. The intended use of these data sets is for computer simulations of buildings and space conditioning systems to facilitate performance comparisons of different system types and building and equipment configurations. Because these data sets represent typical rather than extreme conditions, they are not suited for designing systems to meet the worst case conditions occurring at a location. It is valuable however to see the differences in peak usage between these "typical years" and actual weather years.

Figure 1. Locations Analyzed

The energy usage for all 15 of the locations was also simulated using the TMY2 data set and was determined for seven of the locations for which WYEC datasets were available. The seven WYEC2 locations were: Miami, Phoenix, El Paso, Albuquerque, Seattle, Boise and Chicago.

Energy Model Used

The simulations were run using a freeware 8760 hourly building energy use analysis tool based upon DOE 2.2 [6]. The model has been approved by the California Energy Commission as a 2001 Title 24 non-residential alternative calculation method.

RESULTS

Total Annual Energy Usage

The 30-year average, standard deviation, maximum value and minimum value of the total annual simulated energy usage for the building located in each of the 15 cities is presented in Table 1. It was found that standard deviation of the total yearly energy usage varies from 2.5 GJ/yr for San Francisco (a very mild climate) to 34.9 GJ/yr for Fairbanks (an extremely cold climate).

Table 1. Annual Energy Usage, GJ/yr

Climate Zone	City	Average	Std Dev	Max	Min	Coeff of Variation %	range/avg %	max/avg %	min/avg %
		30 years of weather data							
1	Miami	266.0	6.2	277.1	253.6	2.3	8.9	104.2	95.3
2	Houston	243.4	5.3	253.0	228.6	2.2	10.0	103.9	93.9
2	Phoenix	274.9	9.3	293.0	255.0	3.4	13.8	106.6	92.8
3	El Paso	230.4	3.3	235.9	224.6	1.4	4.9	102.4	97.5
3	San Francisco	203.5	2.5	209.1	198.7	1.2	5.1	102.8	97.6
3	Memphis	242.7	8.6	259.1	227.7	3.5	12.9	106.8	93.8
4	Albuquerque	234.0	6.6	247.3	218.9	2.8	12.1	105.7	93.5
4	Baltimore	250.1	10.0	267.7	223.3	4.0	17.7	107.0	89.3
4	Seattle	220.1	7.4	244.6	206.9	3.4	17.1	111.2	94.0
5	Boise	270.1	19.5	345.5	247.0	7.2	36.5	127.9	91.5
5	Chicago	305.0	18.5	343.5	259.0	6.1	27.7	112.6	84.9
6	Burlington	332.9	17.3	365.2	284.6	5.2	24.2	109.7	85.5
6	Helena	323.3	22.5	382.2	285.3	6.9	30.0	118.2	88.2
7	Duluth	384.7	19.9	430.4	327.4	5.2	26.8	111.9	85.1
8	Fairbanks	553.8	34.9	621.5	453.5	6.3	30.3	112.2	81.9
		Average of 15 locations				4.1	18.5	109.5	91.0

The standard deviation of the total energy used over a number of years is not as informative as an indication of the relative annual fluctuations to the total average annual energy usage. The maximum annual and minimum annual usage values, and the coefficient of variation (defined as the standard deviation as a percentage of the mean) and the range as a percentage of the mean are also presented in Table 1.

It was found that the minimum fluctuation occurred in San Francisco where the standard deviation of the annual energy usage was only 1.2% of the annual usage and the maximum variation (range/average) was only 5.1% over all the years. This very mild location is representative of a maritime climate which has little fluctuations.

Greater between-year variations occur in most of the other locations and tend to increase in the colder climates. The coefficient of variation of annual energy usage varied from 1.2% in San Francisco to 7.2% in Boise with an average of 4.1% for all 15 locations. The ranges (maximum annual energy use minus minimum annual energy use) as a percentage of the average annual energy use varied from 4.9% in El Paso to 36.5% in Boise with an average of 18.5% for all 15 locations.

The distribution of the annual maximum and minimum uses were approximately evenly distributed around the average and did not appear to be skewed either high or low. The average of the 15 locations maximum annual usage for a location divided by the location average was 109.5% and 91.0% for the minimum/average. Stated alternatively, the 15-city average minimal and maximal annual energy usage was 91.0% and 109.5% respectively of the city's long-term average annual energy usage.

Annual Heating and Cooling Energy Use

The statistics for the variation of the annual heating and cooling energy uses are presented in Tables 2 and 3. Since there is negligible heating in any of the years for Miami, that location is not included in the heating statistics. Also since this was an office building with internal heat gains, there was cooling required even in the colder climates.

Table 2. Annual Heating Energy Usage, GJ/yr

	30 years of weather data							
City	Average	Std Dev	Max	Min	Coeff of Variation %	range/avg %	max/avg %	min/avg %
Miami	0							
Houston	3.7	2.8	12.2	0.3	75.0	318.8	327.6	35.1
Phoenix	0.5	0.4	1.3	0.0	84.3	267.7	268.8	7.1
El Paso	3.8	1.6	7.1	1.5	40.9	144.8	184.7	67.5
San Francisco	3.0	1.1	6.0	0.9	38.5	172.9	203.4	49.8
Memphis	15.5	6.2	29.3	4.0	39.9	162.9	188.7	43.0
Albuquerque	21.3	5.7	34.4	9.4	26.7	117.6	162.0	60.6
Baltimore	39.1	8.6	57.4	13.5	22.1	112.3	146.7	44.1
Seattle	34.6	7.2	57.5	22.1	20.7	102.3	166.3	80.7
Boise	65.0	18.3	134.5	42.8	28.1	141.0	206.7	91.4
Chicago	99.2	17.2	134.6	57.9	17.3	77.4	135.7	70.6
Burlington	134.4	16.6	166.4	88.5	12.3	57.9	123.8	75.1
Helena	127.9	21.7	181.8	91.8	16.9	70.4	142.2	86.4
Duluth	181.5	19.2	227.1	124.0	10.6	56.8	125.1	76.4
Fairbanks	326.2	33.4	393.2	234.2	10.2	48.7	120.5	80.0
	Average of 14 locations				31.7	132.3	178.7	62.0

The standard deviation of the heating values range from a minimum of 0.4 GJ/yr in Phoenix to a maximum of 33.4 GJ/yr in Fairbanks. The standard deviation of the cooling values range from a minimum of 1.5 GJ/yr in Burlington to a maximum of 7.3 GJ/yr Phoenix.

Relative annual fluctuations for the heating and cooling are also presented in Tables 2 and 3. The fluctuations for heating were much greater in proportion than for total energy use with the larger fluctuations occurring in the warmer climates. The range/average percentage varied from 48.7% in Fairbanks to 319% in Houston with a fourteen-city average of 132.3%. The coefficient of variation varied from 10% in Fairbanks to 84% in Phoenix. The fourteen-city average of the annual-maximum-heating-energy/30-year-average-heating-energy percentage was 179% and the average of the annual-minimum-energy/30-year-average percentage was 62%.

Table 3. Annual Cooling Energy Usage, GJ/yr

City	30 years of weather data Average	Std Dev	Max	Min	Coeff of Variation %	range/avg %	max/avg %	min/avg %
Miami	89.9	4.7	98.5	80.5	5.2	20.0	109.5	89.5
Houston	67.1	4.2	75.7	56.4	6.3	28.8	112.9	84.1
Phoenix	87.8	7.3	102.1	72.9	8.3	33.3	116.3	83.0
El Paso	49.8	2.5	54.5	44.3	5.0	20.5	109.4	88.9
San Francisco	30.4	2.3	35.0	25.8	7.5	30.4	115.2	84.7
Memphis	54.6	4.2	63.9	46.2	7.8	32.5	117.1	84.6
Albuquerque	33.9	1.9	38.4	31.5	5.5	20.4	113.1	92.7
Baltimore	37.1	2.3	41.7	32.5	6.3	24.8	112.5	87.8
Seattle	17.8	1.7	21.6	14.2	9.4	41.8	121.1	79.3
Boise	27.2	2.4	31.8	23.3	8.8	31.2	116.6	85.4
Chicago	28.7	2.4	33.9	24.2	8.4	33.6	117.9	84.3
Burlington	20.6	1.5	25.0	17.7	7.5	35.1	121.1	86.0
Helena	16.2	2.0	21.5	12.6	12.1	54.6	132.5	77.9
Duluth	17.4	2.2	22.1	13.3	12.4	50.5	127.0	76.4
Fairbanks	17.3	1.8	20.6	13.3	10.3	42.0	118.9	76.8
	Average of 15 locations				8.1	33.3	117.4	84.1

The fluctuations for cooling were also much greater in proportion than for total energy use with the larger fluctuations occurring in the cooler climates with less cooling required. The range/average percentage varied from 20% in Miami to 54.6% in Helena with a fifteen-city average of 33.3%. The coefficient of variation ranged from 5.0% in El Paso to 12.4% in Duluth. The fifteen-city average of the annual-maximum-cooling-energy/30-year-average-cooling-energy was 117% and the average of the annual-minimum-energy/30-year-average was 84%.

Alternatively this can be interpreted as - on the average for all the locations considered the maximum annual energy used was 179% and 117% of the 30-year average annual energy usage for heating and cooling respectively; and the minimum annual energy used was 62% and 84% of the 30-year average annual energy usage for heating and cooling respectively.

Comparison to Annual “Typical” Weather Data

The total average annual energy use for the 15 locations using the 30 years of actual weather data and the annual energy use estimated from the seven WYEC and 15 TMY weather data sets are presented in Table 4.

For the fifteen locations, the annual energy usage with WYEC and TMY data were on the average 0.5 and 1.4 percent higher respectively than the 30-year average annual usage. However, it was found that the annual energy calculated from the WYEC weather data was less than the average minus one standard deviation in three of the seven WYEC locations available and was greater than the average plus standard deviation in one other WYEC location. The annual energy calculated from the TMY weather data was less than the average minus one standard deviation in two cases and higher than the average plus one standard deviation in four of the fifteen cases. In summary, the annual energy estimated using TMY and WYEC data was outside the +/- one standard deviation from the average in 10 of the 22 cases.

Table 4. Annual Total Energy Usage - 30-yr, WYEC and TMY Data, GJ/yr

City	Total Annual Energy Usage, GJ/yr				
	30-year Data Average	WYEC	TMY	WYEC/ 30-yr avg %	TMY/ 30-yr avg %
Miami	266.0	251.4*	253.9	94.5	95.5
Houston	243.4		237.1		97.4
Phoenix	274.9	261.9*	266.0	95.3	96.8
El Paso	230.4	223.9*	222.1*	97.2	96.4
San Francisco	203.5		191.4*		94.0
Memphis	242.7		236.6		97.5
Albuquerque	234.0	238.7	234.5	102.0	100.2
Baltimore	250.1		268.0**		107.1
Seattle	220.1	229.1**	219.9	104.1	99.9
Boise	270.1	279.6	278.2	103.5	103.0
Chicago	305.0	324.8	322.8**	106.5	106.1
Burlington	332.9		358.9**		107.8
Helena	323.3		336.5		104.1
Duluth	384.7		430.8**		112.0
Fairbanks	524.9		572.5		103.4
	Average of 15 locations			100.5	101.4

Notes: * Less than the 30-year average minus 1 standard deviation
** Greater than the 30-yr average plus 1 standard deviation

A greater disparity is pointed out when the annual heating and cooling are investigated independently. The annual heating and annual cooling energy predicted for the locations using the WYEC and TMY weather data sets are presented in Table 6. The average annual heating energy using WYEC and TMY was 154% and 136% respectively of the 30-year average annual heating energy. In four of the six WYEC locations and in eight of the 14 TMY locations the estimated heating energy was greater than the average plus one standard deviation of the 30-year average.

Table 5. Annual Heating and Cooling - 30-yr, WYEC and TMY Data, GJ/yr

City	Annual Heating Energy, GJ/yr					Annual Cooling Energy, GJ/yr				
	30-year Data Average	WYEC	TMY	WYEC/ 30-yr avg %	TMY/ 30-yr avg %	30-year Data Average	WYEC	TMY	WYEC/ 30-yr avg %	TMY/ 30-yr avg %
Miami	0.0					89.9	79.3*	81.3*	88.2	90.4
Houston	3.7		6.6**		176.8	67.1		59.9*		89.4
Phoenix	0.5	0.8	1.0**	173.3	228.8	87.8	77.5*	80.7	88.3	91.9
El Paso	3.8	8.3**	4.3	216.3	110.9	49.8	41.0*	43.7*	82.5	87.8
San Francisco	3.0		4.3**		145.5	30.4		20.6*		67.5
Memphis	15.5		18.8		120.9	54.6		46.9*		85.9
Albuquerque	21.3	31.4**	30.3**	148.1	142.2	33.9	29.6*	28.0*	87.3	82.4
Baltimore	39.1		61.0**		155.8	37.1		31.7*		85.3
Seattle	34.6	49.0**	39.5	141.7	114.2	17.8	12.4*	13.7*	69.6	76.9
Boise	65.0	79.8	79.6	122.7	122.3	27.2	22.4*	21.2*	82.0	77.7
Chicago	99.2	121.3**	120.3**	122.3	122.3	28.7	25.2*	23.6*	87.6	82.3
Burlington	134.4		162.6**		121.0	20.6		16.9*		82.1
Helena	127.9		144.1		112.7	16.2		13.3*		81.7
Duluth	181.5		230.8**		127.2	17.4		12.1*		69.8
Fairbanks	326.2		346.5		106.2	17.3		14.6*		83.9
	Average of 14 locations			154.1	136.2	Average of 15 locations			83.6	82.3

Notes: * Less than the 30-year average minus 1 standard deviation
** Greater than the 30-yr average plus 1 standard deviation

The average annual cooling energy using WYEC and TMY was 83.6% and 82.3% respectively of the 30-year average annual cooling energy. In all seven of the WYEC locations and in 14 of the 15 TMY locations, the estimated cooling energy was less than the average minus one standard deviation of the 30-year average.

Monthly Peak Demand

The statistics for the monthly peak electric and gas demand for each of the locations using the 30 years of weather data and the WYEC and TMY data were also determined. For the monthly peak electrical demand, the 15-city average of the range as a percentage of the 30-year average was 4.7% and the average coefficient of variation was 1.1%. Therefore it can be concluded that there was very little between-year variation in the peak monthly electrical demand. The 15-city average of the maximum/average and minimum/average was 102.5% and 97.8% respectively for monthly electrical demand. The monthly peak electric demand for one city was above the average plus one standard deviation and one city was below the average minus one standard deviation for both the WYEC and the TMY weather data sets.

For the natural gas demand, the 15-city average of the range as a percentage of the 30-year average was 48.4% and the average coefficient of variation was 11.8%. The 15-city average monthly maximum/average and minimum/average was 127.3% and 78.9% respectively. The monthly peak gas demand was above the average plus one standard deviation for three cities for the WYEC data sets and for one city using the TMY data sets. There were no WYEC nor TMY data sets which under predicted the peak gas demand by more than minus one standard deviation.

SUMMARY AND CONCLUSIONS

The annual energy usage of a small light frame office building was determined from 30 years of hourly weather data for 15 locations representing a cross section of the eight climate zones in the USA. The energy usage was also determined using the Typical Meteorological Year (TMY2) for the same 15 locations and for seven of the locations which had Weather Year for Energy Calculations (WYEC) data available. It was found that for the 15 cities, the 30 year range of the total annual energy usage averaged 18% of the 30-year average annual energy usage varying from 4.9% in El Paso TX to 36.5% in Boise, ID. There was greater variation when the annual heating and cooling usages were compared. The 14-city average coefficient of variation (standard deviation/ average) for heating was 31.7% and 8.1% for cooling. The range over average was 132% for heating and 33% for cooling.

Substantial differences were found when the WYEC and TMY energy usages were compared to the 30-year average of the annual energy usage for each location. Four of seven WYEC and six of 15 TMY annual energy usage values were outside +/- one standard deviation from the 30-year average of the corresponding location. However when cooling and heating were separated it was found that in all seven of the WYEC data and in 14 of the 15 TMY cities the electricity usage for cooling using the WYEC or TMY data was less than the average minus one standard deviation. For heating, in four of the seven WYEC and in 8 of 15 TMY cases the gas usage for heating was greater than the average plus one standard deviation of the 30-yr average heating energy. There was very little between-year variation in the peak monthly electrical demand and the average coefficient of variation for peak monthly gas demand was 11.8%. There were two locations in which the monthly peak electrical demand for either TMY or WYEC was outside +/- one standard deviation from the average and four locations where the peak gas demand was above the average plus one standard deviation.

This information gives an indication of the range of natural variability in energy usage of a particular building due solely to the variability in the outside weather conditions. This naturally occurring between-year variation is extremely important when trying to make comparisons between pre- and post- retrofit energy data. It also indicates that there needs to be additional work to investigate why there are such large differences between the long term data and the data customarily used to represent "typical" weather. It is suspected that variations in size and type of building, schedules and ventilation loads impact the applicability of these typical weather data to the specific case. It also points out that as the computational speeds have increased it may be time to start using actual long-term weather data rather than a synthesized weather data set. An investigation of any long-term trend of the annual data to investigate any impacts of climate change is not included in this paper.

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General Requirements for the Energy Performance of Buildings in Latvia

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ABSTRACT

The paper reflects the development of building energy performance legislation in Latvia and shows the current general requirements of Latvian building codes relating to building energy performance. Paper explores the possibility of implementation of Directive 2002/91/EC in Latvia, taking as an example trial buildings energy certification and labeling projects held in Ogre town in 2002-2006. These projects provided the opportunity to develop and examine in Latvia two schemes of building energy certification and labeling. The results of these projects are shown in the paper as well.

INTRODUCTION

Housing resources in Latvia are significantly growing up now. Since 1990 till 2005 the increase of housing area had reached 3.5 millions m^2. Until the end of 2002 it was possible to construct buildings in accordance with USSR building codes, which allowed poor insulation quality. Due to low buildings' energy performance the heat consumption of residential sector in 2003 took 74% of all produced heat. In order to improve that situation Latvian Building Code LBN 002-01 *"Thermal performance of building envelope"* [1] was prepared and came into force on January 1, 2003.

Since Latvia in May 2004 became a member state of the European Union it undertook the obligations to implement all EU legislation, including the requirements in the area of the energy efficiency of the buildings.

Currently Latvia is under the obligation to implement the requirements of the Directive 2002/91/EC on the energy performance of buildings [2]. According to that directive all member states of the European Union have to introduce buildings' energy performance certification. The directive does not forbid member states to implement their own buildings' energy certification schemes as no common methodology can be developed in so short time.

The implementation of this directive had to be done until 4th January 2006. Practically full transposition of the requirements of the directive in Latvia could be achieved not earlier than by the end of 2007. Until 2008 Latvia has to improve the existing legislation and to develop the new building codes in order to improve energy efficiency of buildings and to meet the requirements of the directive.

CURRENT LEGISLATION ON ENERGY PERFORMANCE OF BUILDINGS

Until the end of 2002 it was still possible to construct buildings in Latvia following the requirements of the outdated soviet building code – SniP II-3-79** "Building heat engineering" [3]. According to this code the average heat transfer coefficient of external walls was 1.25 (m^2·K)/W. At the beginning of 2006 the part of buildings which were constructed pursuant to SniP II-3-79** took 75% of all constructed buildings in Latvia or 3.5 millions m^2. Due to the poor quality of buildings' insulation properties the average total heat consumption of Latvian dwelling buildings at the end of 1999 was 250 kWh/m^2 [4] and hot water part took 56% of total heat consumption.

The municipality of Riga city adopted and on January 1, 2000 implemented the rules that provide for the installation of individual heat substations with heat meters in all the buildings. That measure was aimed at the reduction of heat consumption of buildings. The replacement of heat substations has resulted in reduction of buildings' annual specific heat consumption by 10, 52 % [4]. Nevertheless, it should be noted that such measure was successfully implemented only in one town of Latvia and the installation of individual heat substations by itself does not improve the thermal quality of the existing buildings.

The first step in order to improve situation in the field of energy efficiency of buildings at the scale of whole Latvia was developing of Latvian building code LBN 002-01 "Thermal performance of building envelope". This building code came into force on January 1, 2003 and it is mainly based on EN and ISO standards adapted to Latvian conditions. Since then the requirements to heat transfer coefficients for separate parts of buildings' envelope became at least three times better comparing to the previous building code SniP II-3-79**. The Latvian building code LBN 002-01 has radically changed the approach to the limiting of building heat losses. Instead of limitation of only thermal transmittance for the particular building elements it uses also the concept of reference building. The building heat losses should not exceed building heat losses of normative reference building:

$$H_T \leq H_{TR}\,, \qquad (1)$$

where: H_T - building heat loss coefficient, W/K and H_{TR} - building heat loss coefficient of reference building, W/K.

According to LBN 002-01 the reference building is built using the normative values of heat transfer coefficients of building's elements and external glazing of building takes ≤ 20% of the floor area. The heat transfer coefficients of real building's elements can be smaller or higher than the normative values but not higher than maximal level of heat transfer coefficients. The normative and maximal values of heat transfer coefficients according to requirements of LBN 002 - 01 "Thermal performance of building envelope" are given in Table 1. The external glazing of building can also be more or less than 20% of the floor area. The main requirement is that the heat loss coefficient of designed building must be in conformity with Equation 1.

Table 1. Normative and maximal values of heat transfer coefficients in LBN 002-01, $W/(m^2 \cdot K)$

Building element	Dwelling houses		Public buildings		Industrial buildings	
	U_{RN}	U_{RM}	U_{RN}	U_{RM}	U_{RN}	U_{RM}
Roofs and slabs that are in contact with outside air	0,2κ	0,25κ	0,25κ	0,35κ	0,35κ	0,5κ
Slab on ground	0,25κ	0,35κ	0,35κ	0,5κ	0,45κ	0,7κ
Walls with ρ<100kg/m^3	0,25κ	0,30κ	0,35κ	0,4κ	0,45κ	0,5κ
Walls with $\rho \geq$100kg/m^3	0,3κ	0,40κ	0,4κ	0,5κ	0,5κ	0,6κ
Windows, doors and glassed walls	1,8κ	2,7κ	2,2κ	2,9κ	2,4κ	2,9κ

Although new Latvian building code will have great effect on building energy performance it cannot be called an energy performance regulation as it deals only with one of the factors – building transmission heat loss. It is only the first step in the process of determining the mandatory value of Building Energy Performance. Taking into account that fact, it was decided to improve this Building Code. The last changes in LBN 002–01 were done in September 2006. According to the new requirements of LBN 002–01 the energy performance of each new designed building should be estimated.

For the evaluation of energy performance of new designed and reconstructed buildings the modified LBN 002–01 uses the specific heat losses coefficient which is calculated for standard climatic conditions according to LBN 003–01 "Building climatology". The specific heat losses coefficient is calculated according to the following equation:

$$e_G = E_{\Sigma G} / L, \quad kWh/m^2, \qquad (2)$$

where: A- heated built-in area, m^2 and $E_{\Sigma G}$ – building total estimated annual heat consumption in standard climatic conditions, kWh.

Building total estimated annual heat consumption is calculated as following:

$$E_{\Sigma G} = H_T \times T_{gd} \times 24 \times 10^{-3}, \text{ kWh}, \qquad (3)$$

where: T_{gd} - number of the degree-days of standard year.

The implementation of specific heat losses coefficient in Latvian normative base was the first step in creation of energy effective legislation in Latvia. Although that was not enough in order to implement the requirements of Directive 2002/91/EC. For that purpose the energy certification scheme for new and existing buildings should still be developed. Currently Latvian government works out the drafts for two new building codes, which will provide the requirements for the evaluation of energy performance of buildings.

FUTURE DEVELOPMENT OF LATVIAN BUILDING CODES

In order to create the normative basis for the evaluation of energy performance of new and existing buildings Latvian government prepared the draft of Law on energy efficiency. The main aim of this law is to create energy awareness of buildings' inhabitants and to reduce CO^2 emission. The draft legislation determines only general requirements for buildings' energy efficiency. According to these requirements buildings' energy efficiency is calculated on the basis of the following criteria: heat transfer coefficient of buildings' elements, parameters of heating, hot water and air conditioning systems, data on lighting, location of building, climatic conditions and indoor air parameters.

For existing buildings proof of energy efficiency (energy certificate) is calculated by district energy inspectors and for new and reconstructed buildings this document is prepared by buildings' designer. The energy certification for all buildings in Latvia should be done in three years period after the adoption of Law on energy efficiency. In order to implement this law it is necessary to choose the scheme of energy certification for existing and new buildings. At the moment there exist two buildings' energy certification schemes for existing buildings in Latvia.

The first scheme for energy certification of existing buildings on the basis of really measured energy consumption was created in 2000 by Institute of Heat, Gas and Water Technology of Riga Technical University [5]. This scheme is based on unique criteria – *standardized specific heat consumption* which takes into account only energy consumption for space heating and hot water supply. In order to improve this scheme in accordance with requirements of Directive 2002/91/EC the existing criteria was extended by electricity and gas consumptions [6]. The new energy certification criteria - standardized specific energy consumption is calculated using the following equation:

$$q_{st} = \frac{q_{heat} G_{st}}{G} + \frac{\left(q_{hw} + q_{el.} + q_g\right) A}{30n} + q_{el.c} \text{ , kWh/m}^2, \quad (4)$$

where: q_{st} – standardized total heat consumption, kWh/m^2 per year; q_{heat} – measured actual annual specific heat consumption for space heating, kWh/m^2 per year; q_{hw} – measured actual annual specific heat consumption for hot water needs, kWh/m^2 per year; G_{st} - number of degree-days in a standard year ; G - number of degree-days in a rating year; A – total heated space, m^2; *30* – standardized occupancy level, m^2/person; n – actual number of occupants, persons; $q_{el.}$ – electricity consumption of for household needs, kWh/m^2 per year; $q_{el.c}$ - electricity consumption for common utilities, kWh/m^2 per year; q_g – gas consumption, kWh/m^2 per year.

The developed energy certification scheme is based on results of the research on Latvian building energy consumption and it fits to Latvian climatic conditions. For development of structure of the energy certification scale the energy consumption of 488 buildings was analysed during the six year time period. The structure of the energy certification scale was harmonized with GBC2000 concept. The numerical values of the scale are based on two main consumptions: common energy consumption and best present consumption for which the level of LBN 002-01 was used. Energy consumption of all other levels can be found in Table 2 [6].

The second scheme for energy certification of existing buildings is proposed by State Dwelling Agency. This scheme is based on the principles of energy certification introduced in Denmark [7]. The rating scale for energy certification proposed by Dwelling Agency is shown in Table 3.

Table 2. The rating scale of buildings' energy certification by total energy consumption

Nr.	Description	Energy rating		Values, kWh/m^2·year
5	25% better than 3 level		Gold certificate	=125
4	10% better than 3 level		Silver certificate	=149
3	Best present consumption	A	Excellent	149.01-166
2	3 level +1/3(0 level - 3 level)	B	Very good	166.01-201
1	3 level +2/3(0 level - 3 level)	C	Good	201.01-235
0	Common consumption	D	Fair	235.01-270
-1	0 level +1/2(-2 level - 0 level)	E	Bad	270.01-313
-2	Worst consumption	F	Very bad	>313.01

Table 3. The rating scale of buildings' energy certification proposed by Dwelling Agency

Energy rating	Heat consumption, kWh/m^2	Water consumption, m^3/m^2	Electricity, kWh/m^2	CO^2, kg/m^2
A	0 – 95	0 – 0.6	0.0 – 12.6	0.0 – 17.0
B	95 – 111	0.6 – 0.7	12.6 – 15.8	17.0 – 25.6
C	111 – 126	0.7 – 0.8	15.8 – 19.0	25.6 – 34.1
D	126 – 143	0.8 – 1.0	19.0 – 22.1	34.1 – 42.6
E	143 - 159	1.0 – 1.1	22.1 – 25.3	42.6 – 51.1
F	159 – 175	1.1 – 1.3	25.3 – 28.4	51.1 – 59.7
G	175 – 190	1.3 – 1.4	28.4 – 31.6	59.7 – 68.2
H	190 – 206	1.4 – 1.5	31.6 – 34.8	68.2 – 76.7
I	206 – 222	1.5 – 1.7	34.8 – 37.9	76.7 – 85.2
J	222 – 238	1.7 – 1.8	37.9 – 41.1	85.2 – 93.8
K	238 – 254	1.8 – 2.0	41.1 – 44.2	93.8 – 102.3
L	254 – 270	2.0 – 2.1	44.2 – 47.4	102.3 – 110.8
M	≥270	≥2.1	≥47.4	≥110.8

According to this scheme of energy certification all Latvian existing buildings belong to rating less that "F". The reconstruction works of Latvian buildings in order to improve energy rating till "A" or "B" will be economically inexpedient. According to Latvian building code LBN002-01 the heat consumption of new buildings should be not higher that 145 kWh/m^2 – that is equal only to rating "E" pursuant to the scheme of Dwelling Agency. It should also be taken into account that prEN 15217 "*Energy performance of buildings - Methods for expressing energy performance and for energy certification of buildings*" provides for only seven energy classes from "A" to "G".

The general standards for energy certification are still not fully developed at the European level. The adoption of EN standards will be useful for all European member states, creating united approach to buildings' energy certification.

RESEARCH ON ENERGY PERFORMANCE OF EXISTING BUILDINGS IN LATVIA

In order to study the energy consumption of existing buildings and to test energy certification scheme proposed by Riga Technical University the trial energy certification project "ENCERB" was held in Latvia in 2004-2006. In the framework of this project the total energy consumption of 139 multi-storey apartment buildings with total area of 354 265 m^2 was analysed. This project was implemented with the support of EU LIFE programme in Ogre town.

The main energy certification indicators of the buildings examined during the project are given in Table 4.

Table 4 Main energy certification indicators of buildings

Indicator	2004/2005	2005/2006	Absolute differences	Difference, %
Degree days	3816.3	3978.7	162.4	4.3
Total energy performance of buildings, kWh/m^2	211.74	213.08	1.34	0.63
Standardized annual specific energy consumption, kWh/m^2	225.02	224.03	-0.99	0.44

In order to promote building energy certification and create energy awareness among inhabitants, it was decided to use buildings' energy labels. These labels were placed on buildings visible for all inhabitants. Anyone interested could easily study building's energy rating in comparison with other buildings.

The numeric values of changes in the categories of the energy consumption of the buildings during energy certification are given in Table 5.

Table 5.Comparison of the categories of the energy performance of the buildings

Category	A	B	C	D	E	F	Total:
Number of buildings in 2004/2005	0	18	80	37	4	0	139
Proportion, % in 2004/2005	0	12.9	57.6	26.6	2.9	0	100.00
Number of buildings in 2005/2006	0	20	81	36	1	1	139
Proportion, % in 2005/2006	0	14.36	58.3	25.9	0.72	0.72	100.00

The comparison of the categories of the energy performance of the buildings has revealed that in 2005/2006 the energy consumption of the buildings has been slightly reduced, since the number of the buildings rated "A" and "B" has been increased. When compare the energy consumption of the buildings in 2004/2005 and 2005/2006 one can see that the energy consumption has been increased only in one building.

The implementation of energy certification scheme proposed by Riga Technical University in Ogre town has let to create correct image of buildings' energy consumption and it gave the possibility to draw attention of inhabitants to buildings energy consumption, stimulating them to improve energy efficiency of buildings.

CONCLUSIONS

1. Until the end of 2002 it was still possible to construct buildings in Latvia following the requirements of the outdated USSR building code – SniP II-3-79** "Building heat engineering".
2. By the beginning of 2006 the part of buildings which were constructed pursuant to SniP II-3-79** took 75% of all constructed buildings in Latvia or 3.5 millions m^2. Due to the poor quality of buildings' insulation properties the average total heat consumption of Latvian dwelling buildings at the end of 1999 was 250 kWh/m^2.
3. The introduction of specific heat losses coefficient in Latvian building code LBN 002-01 "Thermal performance of building envelope" was the first step in creation of energy effective legislation in Latvia. Although that was not enough in order to implement the requirements of Directive 2002/91/EC.
4. In order to create the normative basis for the evaluation of energy performance of new and existing buildings Latvian government prepared the draft Law on energy efficiency. At the moment this law could not be implemented due to the lack of adopted energy certification scheme in Latvia. But it should be admitted that standards for energy certification are still not fully developed also in Europe. The adoption of EN standards will be useful for all European member states, creating united approach to buildings' energy certification.
5. The trial implementation of energy certification scheme introduced by Riga Technical University had shown that great majority of Latvian buildings belong to rating "C". The positive effect of implementation of energy certification scheme can be achieved using also energy labels.

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Energy performance indicator and energy performance requirements: a Polish approach to implementation of EPBD

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SUMMARY

The EC Directive on Energy Performance of Buildings created a challenge for all Member States, especially in sense of adoption of new requirements and calculation methodology. This paper describes the Polish experts' solution. It has been suggested that the distinction between energy performance requirements and determination of building energy class should be introduced to Polish regulation. Two levels of requirements are considered: first regulation of gains and losses and second regulation on efficiency of hot water preparation and lighting efficiency. Assignment of energy class for particular building is proposed in relation to reference building and expressed by dimensionless integrated energy performance indicator WZE. The WZE is normalized in a sense that its value equal to 1 represents the building energy performance identical to the reference building. The methodology takes into account different impact of energy sources on environment by utilization of weightings representing specific energyware.

INTRODUCTION

All the Member States have faced a challenge related to implementation of the Directive EC/91/2002 on energy performance of buildings (EPBD) [1]. The implementation deadline has been met by some of the countries, whereas the others were delayed due to the different reasons. Poland is among the countries which asked for an extension time.

Implementation of requirements settled by European Directives is obligatory for all the member states but the way chosen is entirely left for the national decisions. To support harmonization of implementation towards Europe the Commission mandated CEN to prepare set of new standards supporting objectives of Directive.

In Poland, works on implementation of EPBD have been voluntarily undertaken by Association of Energy Auditors (non-profit organization of public benefit) in 2004. Five working groups have a structure reflecting typology of energy performance problems identical as in the CEN Umbrella document [2]. The idea was to follow the new CEN standards (also on a stage of project) as close as it was possible in order to facilitate harmonization with EU standardization in a future. The final proposal of the scope and form of energy certificate was a compromise between requirements of the drafts of standards and ability of its adaptation (lack of energy inventory of buildings and related metrological data). Once, after few years of assessment system operation, such a data will become available this fact should be easily taken into account in a process of updating the regulations. The method proposed [3] - has an additional advantage – it can be used along with changing energy performance requirements. It uses concept of reference buildings and introduces dimensionless indicator for the assessment which is not entirely in line with recommendation of the indicators proposed by prEN 15217 Energy performance of buildings - Methods for expressing energy performance and for energy certification of buildings [4]. Further, there was a clear agreement of the stakeholders that the assessment should follow asset rating instead the operational rating.

METHODS

Requirements and Energy Performance

Among the others the basic Directive aims are:

1. Establishment of new (more demanding) energy performance requirements, taking into account their updating according to changes in technological and economical circumstances;
2. Adoption of calculation methodology comprising elements of energy balance described in the Annex to the Directive.

These aims are obligatory for all member states both old and new ones. In spite of acceptance of the Directive before the extension of EU there was no any additional transition period foreseen for new members. However there are many projects financed by EIE program focused on different issues of EPBD. Concerted Action, the project dedicated to information exchange among the national experts working on Directive implementation is the example of the EC help to MS. All these projects showed a variety of approaches taken by the countries, but they also are indicating European efforts towards harmonization of setting requirements procedure and methodology of calculation. For illustration of the span of approaches, the energy performance as the term is differently understood across the Europe. Some of the countries are assigning to this term energy delivered or equivalent emission whereas the others are taking heat losses or gains. Detailed description of the approaches is provided in report of ENPER [5], the Proceedings of International Conference on the Energy Performance of Buildings, Brussels, September, 2005 [6].

Determination of Energy Performance Indicator

The distinction between energy performance requirements and determination of building energy class was introduced in Polish regulation. In regards to requirements, two levels are considered: first regulation of gains and losses and second regulation on efficiency of hot water preparation and lighting efficiency. Assignment of energy class for particular building is proposed in relation to so called reference building and expressed by dimensionless indicator called integrated energy performance indicator *WZE*. The *WZE* is not a subject of regulation - requirements; it is a result of application of partial lower level requirements. Integrated energy performance indicator (*WZE*) is described by Eqn.1.

$$WZE = N_g \cdot f_g + N_w \cdot f_w + N_o \cdot f_o + N_{ch} \cdot f_{ch}, \quad (1)$$

where coefficients N represents a level of requirements fulfilment (ratio of energy use in analyzed building to reference building) according to heating and ventilation, hot water, lighting and cooling that is denoted respectively by subscripts “*g*”, “*w*”, “*o*” and “*ch*”. The coefficients *f* describe share of energy use for specific purposes versus an overall energy use *Q*, taking into account whole building or analyzed operational zone (e.g. apartment) (eqn 3).

$$f_g = \frac{\sum_i \left(w_i \cdot \sum_j E_{gi,j} \right)}{Q} \qquad (2a)$$

$$f_w = \frac{\sum_i \left(w_i \cdot \sum_j E_{wi,j} \right)}{Q}, \qquad (2b)$$

$$f_o = \frac{\sum_i \left(w_i \cdot \sum_j E_{oi,j} \right)}{Q}, \qquad (2c)$$

$$f_{ch} = \frac{\sum_i \left(w_i \cdot \sum_j E_{chi,j} \right)}{Q}, \qquad (2d)$$

$$Q = \sum_i \left(w_i \cdot \sum_j E_{gi,j} \right) + \sum_i \left(w_i \cdot \sum_j E_{wi,j} \right) + \sum_i \left(w_i \cdot \sum_j E_{oi,j} \right) + \sum_i \left(w_i \cdot \sum_j E_{chi,j} \right), \qquad (3)$$

Summation by "*i*" is related to different sources of energy supplied to analyzed building. This summation takes into account different impact of energy sources on environment, weightings representing specific energyware w_i , are presented in table 1. Summation by "*j*" is related to a number of zones of specific operation (e.g. in case of heating "*j*" can indicate different temperature zones, in general number of zones within specific energy use can be different e.g. number of temperature zones can be different then the lighting zones). Symbols $E_{gi,j}$, $E_{wi,j}$ $E_{oi,j}$ $E_{chi,j}$ are denoting energy use for specific purposes in specific zone that is related to specific energy source.

Table 1. Weights assigned to energyware

Energyware	Weights (*w*)
Electricity (in Poland basically from coal burning)	2,5
Biomass	0,5
Solar, wind and geothermal energy	0
Other energywares	1

Table 2. Values of WZE indicator and related energy classes

	Values of *WZE* indicator	Energy class of building
	from 0 to 0,25	A
	from 0,26 to 0,50	B
	from 0,51 to 0,75	C
	from 0,76 to 1,00	D
	from 1,01 to 1,25	E
	from 1,26 to 1,50	F
	over 1,51	G

The *WZE* is normalized in a sense that its value equal to 1 represents the building energy performance identical to the reference building. The reference building is the building similar to assessed one in terms of shape, operational areas and, profiles of use but with components fulfilling minimum requirements for newly constructed building (e.g. *U* values), referenced system efficiencies, and energy weights equal to 1, except those used for media transport (eg. energy used for circulation pumps in heating system). To some extent *WZE* value indicates cost of energy in relation to the reference building. The *WZE* is divided by intervals representing the energy performance class of the building, table 2. Number of classes depends on marketing assumptions of energy assessment scheme. The values presented in table 2 are the proposals for case of Poland.

Determination of heating and ventilation component

Construction of heating and ventilation component in Eqn.1 can be done in the following steps:

1. Calculation of delivered energy demand E_g for each operational area of assessed buildings with respect to energyware, installation efficiencies, under standard use and weather. Calculation of net energy is performed according to Polish standard, but the intention is to use CEN EN 13790 [7] after its adaptation to Polish building stock features
2. Same as in p.1 but done for reference building E_{gr} with defined installation efficiencies in regards to production, distribution and regulation, as well as reference weights (basically equal 1).
3. Finally, the component N_g of Eqn.1. can be calculated according the formula:

$$N_g = \frac{\sum_i \left(w_i \cdot \sum_j E_{gi,j} \right)}{\sum_i \left(w_{r,i} \cdot \sum_j E_{gri,j} \right)} \qquad (4)$$

Energy demand for hot water preparation

Use of hot water depends on particular needs and therefore is not regulated in requirements. The only regulated requirements are efficiencies of 1 m^3 of hot water preparation and distribution, [8]. Calculation of hot water component of Eqn. 1:

$$N_w = \frac{\sum_i \left(w_i \cdot \sum_j E_{w1\,i,j} \right)}{\sum_i \left(w_{r,i} \cdot \sum_j E_{w1r\,i,j} \right)} \qquad (5)$$

where: *w* is energy weights, E_{w1} and E_{w1r} energy for m^3 of hot water preparation in assessing and in reference buildings. Equation 5 shows the level of fulfilment of energy performance requirements expressed in energy needed for 1 m^3 of hot water.

Moreover, for the purpose of energy class assignment some averages of hot water demands are listed in regulations as but just for use in calculation procedure of f_w coefficient. At the same time, for information purposes it is possible to estimate energy used for hot water in reference building.

Lighting component

Basic mean of energy reduction for lighting is application of energy efficient hardware. Thus, as the measure of lighting system efficiency for regulation purposes the specific corrected capacity of lighting unit W/(m^2·100 lx) is proposed, [9]. Again, the same approach as for hot water is applied, the regulation refers to technical solutions not to the use of lighting.

The coefficient of fulfilment of lighting energy requirements N_o for Eqn.1 is the ratio of weighted averages of the specific capacity of assessed and referenced lighting systems in the whole building. The averages are weighted by areas of different lighting demands. The referenced specific capacities vs. light fluxes for different lighting purposes are listed in the regulation.

Determination of coefficient f_o requires estimation of energy use for lighting. This can be done if a capacity of the unit and the operating time are known. The basic formula applied for this purpose is taken from the PrEN 15193: Energy performance of buildings — Energy requirements for lighting, [10] in GJ/year:

$$E_O = 3{,}6 \cdot 10^{-6} \left(\sum_j \left(P_{p,j} \cdot t_{p,j} \right) + \sum_j \left(P_j \cdot t_{u,j} \right) \right), \quad (6)$$

where $P_{p,j}$ is total installed capacity of emergency lighting units, W; $t_{p,j}$ is time of operation of emergency lighting units in a year, h/r; P_j is total installed capacity of built in lighting units W; $t_{u,j}$ is time of operation of built in lighting units in a year, h/r ; Summation on „j” goes over lighting zones.

RESULTS

Some analysis of buildings using energy performance indicator *WZE* has already been done. Very interesting study analysed school buildings in one of the towns in southern Poland [11]. The figure 1 presents the comparison of estimations of *WZE* with integrated energy characteristics EA according to Display campaign [12]. The figure presents pretty good correlation between these two methods (R^2= 0.6935). *WZE* indicator turned to be more generous for buildings that used renewable energy. Figure 2 presents changes of energy performance indicator *WZE* due to thermomodernization of schools. The modernization of schools reduced initial values of *WZE* by 10 to 70 % and in vast majority of cases reduced values of *WZE* below 1.

The proposal of Polish certificate counts four pages and covers information on energy use for all purposes for the assessed and reference buildings along with recommendation about potential improvements as well as graphical presentation of energy performance indicator WZE. The example for thermomodernized apartment building (no air conditioning) is presented below on Figure 3.

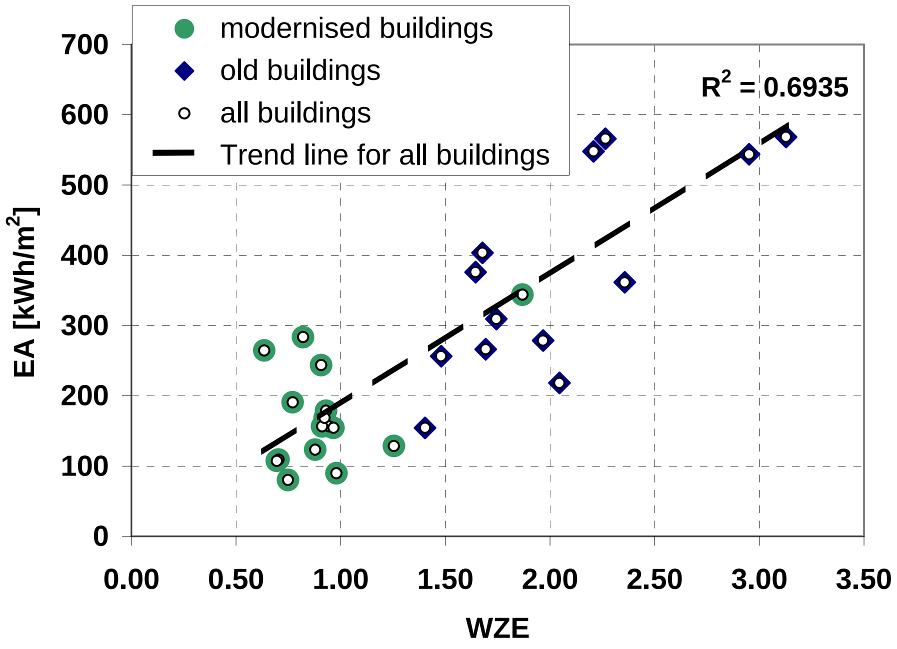

Figure 1. The comparison of energy performance of building evaluated using energy performance indicator *WZE* and integrated energy characteristics EA according to Display campaign [11].

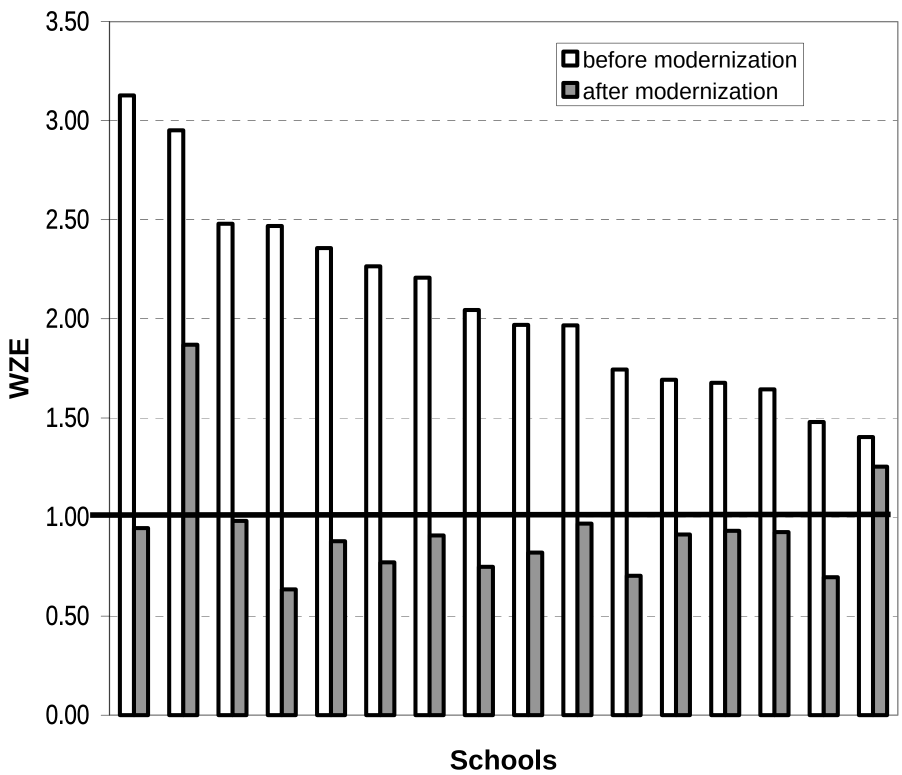

Figure 2. The change of energy performance indicator *WZE* for thermodedernized schools [11].

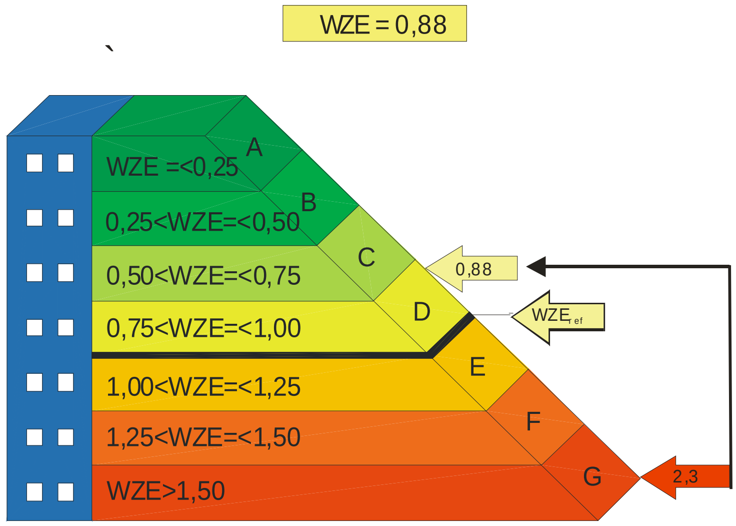

Figure 3. Label from energy certificate showing improvement of energy standard

The label on Figure 3 indicates strong improvement of building standard due to thermomodernization. *WZE* value has been changed from 2,3 to 0,88 that means the energy cost in building after investment is reduced approximately 2,6 times and finally is 12% lower then maximum normalized cost in similar newly constructed building (the reference building). Concept of reference buildings requires that reference values of all components influencing energy performance of buildings are specified.

DISSCUSSION

Presented method of determining integrated energy performance indicator and related to it energy class assignments are the basis for preparation of energy certificate according to the directive on energy performance of buildings. It is important that energy certificate can be elaborated for every type of building for which delivered energy and hot water requirements are estimated. For others types of building than apartment one the components of Eqn.1 related to lighting and cooling can be easily taken into account. Reference building concept made this approach general and easy to interpret (*WZE*=1 denotes the building fulfilling maximum/minimum permissible values with energy supplied from reference source).
Low values of *WZE* (< 1) indicates that buildings components have better parameters than reference ones or/and efficiencies of the systems are better than reference ones or/and energy source is more ecological than reference one. The *WZE* concept has moved the regulation requirements towards partial regulations (building components oriented). Energy class of a building is a resultant of partial requirements and technological solutions of installations.
Of course some Polish energy engineers are afraid that WZE calculations are more time consuming (analysed building is evaluated twice), however special software with data reference values base could solve this problem. Other doubts are associated with the fact that energy performance indicator *WZE* is not sensitive to building shape and allow architects to design buildings with low area to volume ratio.

The contradictory approach where the energy performance requirements are expressed in terms of energy or emissions normalised over the building area is undertaken is some countries. This gives even more freedom in architectural, constructional and material choices in extreme it would be acceptable to erect poor buildings entirely supplied by renewable.

Therefore, energy performance requirements are usually accompanied by additional regulations related to building components. Moreover, the requirements using energy terms needs detailed database of energy use along with coincident weather and profile of building use to decide about the energy class. In the absolute scale school without pool will be better ranked that one with it. Thus, very often, the typology of buildings is introduced, which means necessity of preparing number of scales related to number of identified building types and their equipments. In case of multi purpose buildings (eg. multifamily residential building, with shops on the first floor and restaurant on the roof level) the situation is extremely difficult to solve. Moreover as new CEN standards introduce the concept of different indoor environment categories the number of scales should by multiplied by number of indoor environment categories e.g. 3. If not buildings that use additional energy for creation of higher level of indoor environment would be punished by low energy performance category.

ACKOWLEGEMENT

The concept of energy performance indicator *WZE* has been developed by group of volunteers form Association of Energy Auditors. The work was cosponsored by the Ministry of Construction (formerly Ministry of Transportation and Construction).

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Application of the European Directive for the Energy Performance of Buildings in Cyprus

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SUMMARY

It is a fact that today in Cyprus there are no building-regulations concerning thermal insulation of the building envelope and consequently no restriction exist regarding energy performance. After joining the EU in May 2004, Cyprus has to comply with various directives of the Union. One of them is the Directive on the Energy Performance of Buildings 2002/91/EC. In order to comply with this requirement, a special committee was formed for the development of the required methodology and software to be used for this purpose. The objective of this paper is to present the methodology and program developed and its characteristics together with the type of output the user is expected to get. Considering that Cyprus is fully dependant on imported energy, it is believed that this new directive will lead to energy efficient buildings, and in the long run it will benefit the economy of the island.

INTRODUCTION

Today there are no building-regulations concerning thermal insulation of the building envelope in Cyprus and consequently no restriction exist regarding energy performance. After joining the EU on 1st May 2004, Cyprus has to comply with various directives of the Union. One of them is the Directive on the energy performance of buildings 2002/91/EC [1]. In order to comply with this requirement, a special committee was formed under the Ministry of Commerce, Industry and Tourism for the development of the required methodology and software to be used for this purpose. These are mostly based on the CEN/TC 89 'Energy performance of buildings – Calculation of energy use for space heating and cooling' [2] and the Cyprus standard CYS 98: part 1 published in 1999 on thermal insulation and rational use of energy in domestic buildings [3]. After a short description of the methodology the program developed for this purpose is described.

METHODOLOGY

The field of application of the present methodology covers both new residential buildings and new buildings that are not used as residences with floor area less than 1000 m^2, without central system for air treatment. The objective of the methodology is to establish the way and the standards that should be followed for the calculation of the energy performance of buildings as it is determined in the field of application, aiming to:

- Achieve indoor comfort conditions and ventilation requirements that will ensure the quality of the air inside the building and the hot water production requirements with the minimum possible consumption of energy.

- Avoid or minimize any pathological situations in respect of structural elements, arising from internal condensation, having negative effect on the durability of the structural elements and the quality of the air inside the building.

Despite its small size Cyprus has variable and distinctive weather patterns. For the purpose of this methodology, four climatic zones are set in the island; seaside, inland, semi-mountainous and mountainous as shown in Fig. 1. These zones were set according to the climatic data available from the Cyprus Meteorological Service.

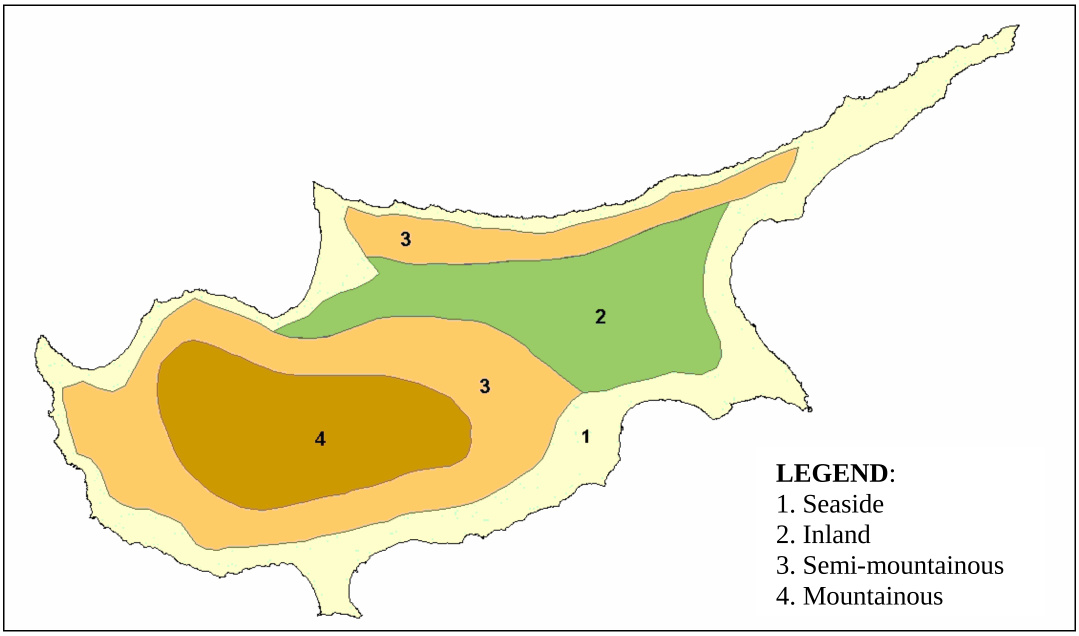

Fig. 1 Map of Cyprus showing the four climatic zones

Restrictions in the annual nominal useful energy needs and total annual consumption.

The annual nominal useful energy needs for heating a building, N_{ic}, examined with the present methodology, considering its form, thermal characteristics of the structural elements of its shell and the useful thermal gains from solar radiation entering the building, **should not exceed** the annual nominal useful energy needs for heating of the reference building, $N_{i.}$

Similarly, the annual nominal useful energy needs for cooling a building, N_{vc}, examined with the present methodology, considering its form, thermal characteristics of the structural elements of its shell and the thermal load from solar radiation entering the building, **should not exceed** the annual nominal useful energy needs for cooling of the reference building, N_v.

No restrictions are applied for the annual nominal useful energy needs for hot water production, N_{ac}.

The total annual consumption in primary energy of the building, N_{tc} in toe/m^2 per year, should not exceed the maximum allowed value of the total annual consumption in primary energy, N_t, as this is determined in the decree for the requirements of minimum energy efficiency, published by the Minister of Commerce, Industry and Tourism according to the law. This is calculated by adding the annual load for heating, cooling and hot water by multiplying each by its annual utilization factor (F_{pu}) and dividing by the equipment efficiency (η):

$$N_t = \frac{N_i F_{pui}}{\eta_i} + \frac{N_v F_{puv}}{\eta_v} + \frac{N_a F_{pua}}{\eta_a}, \qquad (1)$$

The annual nominal useful energy needs for heating and cooling the reference building should be achieved by using the following parameters:

1. Reference values of the thermal transmittance (U_{ref}) of the structural elements of the building that separate the building from the external environment or from areas not being air-conditioned or from other buildings attached to them.
2. Reference value for the solar factor ($g\perp_{ref}$).
3. 20% of the ratio of the area of the openings to the useful floor area.

The values presently applied are shown in Table 1.

Table 1 Reference values

Parameters	**Value**
Thermal Conductance of the roof, $U_{ref\text{-}roof}$	0.9 W/m^2 °C
Thermal Conductance of the walls, columns and beams, $U_{ref\text{-}walls}$	0.9 W/m^2 °C
Thermal Conductance of the exposed floor, $U_{ref\text{-}floor}$	0.9 W/m^2 °C
Thermal conductance of the openings, $U_{ref\text{-}openings}$	4.5 W/m^2 °C
Solar factor of the glazing, $g\perp_{ref}$	0.655
Note: The above values are subjected to periodical revision according to the decree for the requirements of minimum energy efficiency, published by the Minister of Commerce, Industry and Tourism according to the law.	

Each building should satisfy and not exceed the minimum requirements of the thermal characteristics of the building's shell as these are determined in the decree for the requirements of minimum energy efficiency, published by the Minister of Commerce, Industry and Tourism according to the law. The present values are shown in Table 2.

Table 2 Minimum requirements

Parameters	**Value**
Thermal Conductance of the roof, $U_{min\text{-}roof}$	1 W/m^2 °C
Thermal Conductance of the walls, columns and beams, $U_{min\text{-}walls}$	1 W/m^2 °C
Thermal Conductance of the roof, $U_{min\text{-}floor}$	1 W/m^2 °C
Thermal conductance of the openings, $U_{min\text{-}openings}$	5 W/m^2 °C
Note: The above values are subjected to periodical revision according to the decree for the requirements of minimum energy efficiency, published by the Minister of Commerce, Industry and Tourism according to the law.	

As this is the first implementation of this procedure, these factors are high enough compared with other Member States in order not to disturb the building industry. They will however be updated (possibly lowered) at least every 5 years.

Indoor comfort conditions are determined as follows:

1. The buildings should retain constant indoor temperature at 20°C for the heating period and 24°C for the cooling period.
2. The number of indoor air renewal, that will ensure its quality and comfort conditions, should not be less than 0.6 renewals per hour, which should be achieved naturally under usual conditions of operation (without mechanical ventilation).
3. For the consumption of hot water, 40 liters per person per day at the temperature of 37°C is considered (make-up water supply temperature of 15°C).

Calculation of annual nominal useful energy needs for heating, N_{ic}

The energy needs for heating of a building is the quantity of energy that is required in order to maintain the internal temperature constant during the period of heating. These needs do not represent the real consumption of energy because the tenants of the buildings do not maintain the same conditions and do not use the building the same way. In practice, the real consumption would be different depending on the habits of the owners.

Nevertheless, the energy needs for heating of a building is an objective criterion for the comparison of its thermal characteristics; the bigger they are the colder will be the building during the winter period or the more will be the consumption of energy required for maintaining internal comfort conditions.

The energy needs for heating are calculated by the method of calculation that is determined by European Standard EN ISO 13790. An important simplification is that the building is considered as one zone with the same temperature in all the spaces.

The annual nominal useful energy needs for heating, N_{ic} in kWh/m^2, is the algebraic sum of the heat losses by the shell of the building, Q_t; the heat losses due to infiltration and ventilation, Q_v; and the useful heat gains, Q_{gu}, as a result of the lighting, the tenants, the equipment and the rate of solar radiation that enters the building through the openings:

$$N_{ic} = \frac{\left(Q_t + Q_v - Q_{gu}\right)}{A_p}, \qquad (2)$$

The heat losses by the shell of the building, Q_t concern the heat losses through the walls, columns, beams, openings, the roof and the floor, because of the temperature difference between internal and external temperature. These are given by the following equation in kWh:

$$Q_t = Q_{ext} + Q_{ina} + Q_{pe} + Q_{pt}, \qquad (3)$$

where Q_{ext} is the heat losses through the walls, columns, beams, openings, the roof and the floor in direct contact with the external environment; Q_{ina} is the heat losses through the walls, columns, beams, openings, the roof and the floor in contact with spaces that are not being heated; Q_{pe} is the heat losses through the structural elements in contact with the ground; and Q_{pt} is the heat losses through thermal bridges.

Heat losses through the walls, columns, beams, openings, the roof and the floor in direct contact with the external environment are given by the following equation in kWh:

$$Q_{ext} = \frac{U * A * DD * 24}{1000}, \qquad (4)$$

where U is the thermal transmittance (W/m^2 °C); A is the area of element, measured internally (m^2); and DD is the Degree Days.

Heat losses through the walls, columns, beams, openings, the roof and the floor in contact with spaces that are not being heated, are given by the following equation in kWh:

$$Q_{ina} = \frac{U * A * DD * 24 * \tau}{1000}, \qquad (5)$$

where $\tau = \frac{1-(\Theta_\alpha - \Theta_{atm})}{(\Theta_i - \Theta_{atm})}$; Θ_α is the temperature of the unheated space (°C); Θ_{atm} is the temperature of the external environment (°C); and Θ_i is the indoor temperature (°C).

The value of τ may be calculated with precision, using the method that is determined in the standard EN ISO 13789 or simply be considered as a conventional value taken by a table given.

Heat losses through the structural elements in direct contact with the ground, are given by the following equation in kWh:

$$Q_{pe} = \frac{(\psi_j B_j) * DD * 24}{1000}, \qquad (6)$$

where ψ_j is the linear factor of thermal conductivity of the structural element, j (W/m°C); and B_j is the internal perimeter of the floor and/or of the horizontal length of wall in contact with the ground j, internally (m).

Heat losses through thermal bridges, are given by the following equation in kWh:

$$Q_{pt} = \sum(\psi_j B_j) * \frac{DD * 24}{1000}, \qquad (7)$$

where ψ_j is the linear factor of thermal conductivity of the thermal bridge j (W/m°C); and B_j is the length of the thermal bridge j (m).

Heat losses due to infiltration and ventilation, are given by the following equation in kWh:

$$Q_v = \frac{\rho * Cp * R_{ph} * A_p * P_d * DD * 24 * (1 - \eta_v)}{1000}, \qquad (8)$$

where ρ is the density of air (kg/m^3); Cp is the specific heat of air (J/kg°C); R_{ph} is the number of air changes per hour (h^{-1}) depending on the position and the height of the building; A_p is the useful floor area (m^2); P_d is the mean internal height of the building (m); and η_v is the efficiency of heat recovery system.

Useful heat gains, Q_{gu} in kWh, as a result of lighting, occupants, equipment and the rate of solar radiation that enters the building through the openings are given by:

$$Q_{gu} = \eta * (Q_i + Q_s), \qquad (9)$$

where η is the utilisation factor of the heat gains (not all the heat gains are useful since a percentage can lead to internal overheat); Q_i is the internal heat gains; and Q_s is the heat gains due to solar radiation.

The utilization factor depends on the thermal inertia of the building and on the ratio between total heat gains and the total thermal losses of the building.

Internal heat gains, Q_i in kWh are given by:

$$Q_i = \frac{q_i * A_p * M * days * hours}{1000}, \qquad (10)$$

where q_i is the internal loads (W/m^2); A_p is the floor area (m^2); and M is the duration of the heating period in months.

Heat gains due to solar radiation, Q_s in kWh are given by:

$$Q_s = G_{sul} * \sum_j \left[X_j * \sum_n A_{snj} \right] * M, \qquad (11)$$

where G_{sul} is the mean monthly solar radiation on a 1 m^2 vertical surface of south orientation for the heating period (=108 kWh/m^2); X_j is the orientation factor of openings j; A_{snj} is the area of surface n, in orientation j, (m^2); j is the orientation; n is the number of surfaces in orientation j; and M is the duration of heating period in months.

It should be noted that angles of inclination of 60^o are considered vertical, otherwise they are considered horizontal.

The area A_s is calculated for each opening or group of similar openings, as far as glazing factor, solar protection and shading, are of the same orientation:

$$A_s = A * F_s * F_g * F_w * g_{\perp}, \qquad (12)$$

where A is the total area of openings (m^2); F_s is the solar obstruction factor due to shading caused by obstacles as other buildings, hills, trees, etc., other parts of the building, overhangs, verandas, and vertical obstacles and fins; F_g is the ratio of glazing area to the area of the opening; F_w is the correction factor for the glazing inclination; and $g_{\perp}$ is the solar factor of the glass at the vertical direction and indicates the percentage of solar radiation that passes through it.

The factor F_s has a range of values between 0 and 1, but in practice it cannot ever be less than 0.27 (characteristic percentage of diffuse radiation).

Calculation of the annual nominal useful energy needs for cooling, N_{vc}

As in the heating case, for cooling the real consumption would be different than the one estimated from this methodology, depending on the habits of the owners. Nevertheless, the cooling needs of a building is an objective criterion for the comparison of its thermal characteristics; the higher the cooling needs the hotter will be the building during the cooling period or the more will be the consumption of energy needed in order to maintain comfortable conditions. The cooling needs are calculated by the method of calculation determined by European standard EN ISO 13790. An important simplification is that the building is considered as one zone with the same temperature, like it happened for heating.

This methodology is the same with the methodology for the calculation of the heating needs. While for the heating period the non useful heat gains are responsible for overheating, for the cooling period are the ones causing cooling load and need to be removed.

The cooling needs of a building, N_{vc} in kWh/m^2, are calculated by the equation:

$$N_{vc} = \frac{Q_g(1-\eta)}{A_p}, \qquad (13)$$

where Q_g is the total cooling loads of the building, (kWh), given by Eq. (14); η is the utilisation factor of cooling loads; and A_p is the useful floor area (m^2).

$$Q_g = Q_1 + Q_2 + Q_3 + Q_4, \qquad (14)$$

where Q_1 is the cooling loads because of the exposure of each individual building element to the external environment (effects of temperature difference and incident solar radiation); Q_2 is the cooling loads due to the solar radiation entering the building through the openings; Q_3 is the loads due to infiltration and ventilation; and Q_4 is the internal loads due to the equipment, occupants and artificial lighting.

Cooling loads because of the exposure of each individual building element to the external environment (effects of temperature difference and incident solar radiation) in kWh are:

$$Q_1 = M * U * A * (\Theta_m - \Theta_i) + U * A\left(\frac{\alpha * I_r}{h_e}\right), \qquad (15)$$

where U is the thermal transmittance of each building element of the shell, (W/m^2 °C); A is the area of element, (m^2); Θ_m is the mean external temperature of the cooling period, (°C); Θ_i is the indoor temperature, (°C); α is the solar absorption factor of the external surface; I_r is the mean monthly solar radiation on each orientation, (kWh/m^2); h_e is the exterior factor of transport of heat (e.g. 25 W/m^2 K); and M is the duration of the cooling period in hours.

Cooling loads due to solar radiation entering the building through the openings in kWh are:

$$Q_2 = \sum_j \left[I_{rj} \sum A_{snj} \right], \qquad (16)$$

For the calculation of solar factor $g_\perp$ of the openings, the solar protection factor, $g_\perp$’, is taken into consideration as well. During the heating period this factor was ignored, which means that during winter, no solar protection is considered, e.g. shutters are assumed to be open allowing the building to benefit from the solar radiation entering through the openings (solar gain). On the contrary, because during the cooling period this is considered as load, solar protection is applied.

Loads due to infiltration and ventilation in kWh are:

$$Q_3 = Q_v, \qquad (17)$$

Internal loads due to the equipment, the occupants and artificial lighting in kWh are:

$$Q_4 = q_i * M * A_p, \qquad (18)$$

where q_i is the internal load in W/m^2.

Annual nominal useful energy needs for hot water production, N_{ac}

The annual nominal useful energy needs for hot water production in kWh are calculated by:

$$N_{ac} = \frac{M_{aqs} * Cp_{water} * \Delta T * \eta_d}{A_p * 3600 * 1000}, \qquad (19)$$

where M_{aqs} is the mean daily hot water consumption (lt); Cp_{water} is the specific heat of water (J/kg°C); ΔT is the temperature difference between cold and hot water, (°C); n_d is the number of days the building is occupied; and A_p is the useful floor area, (m^2).

Contribution of Renewable Energy Sources (RES).

The contribution of RES should be calculated and justified. For this purpose the contribution of RES systems can only be used if the systems or the equipment are certified according to the rules and existing legislation.

SOFTWARE

The software developed, which is currently in version 1.0, can be used for the estimation of the energy requirements of residential buildings and buildings up to 1000m^2 without centralized air conditioning system, for heating, cooling and domestic hot water. From the results obtained by using the software it can be decided whether a building fulfils the minimum energy requirements specified. According to the directive no building should be built without fulfilling the minimum energy requirements. Additionally, the results can be used for the issue of the energy performance certificate of the building.

The software estimates the energy requirements based on the envelope and orientation of the building. The software cannot be used for design purposes or system sizing. The building is considered as one zone. The software is user friendly and it is provided free to all engineers. The software has libraries of readymade structures and materials (walls, openings, slabs, roofs, floors, etc.) that can be found in buildings erected in Cyprus for the last 20 years, whereas the user has the capability to add a new structure not included in the libraries, under some circumstances. By choosing a city, a municipality or a village from a database the climatic conditions are automatically obtained based on the altitude of the area. Additionally the user can use U-value for walls, slabs, roof and openings up to the values specified in Table 2 (the program does not allow higher values).

The output of the program for a case investigated is shown in Fig. 2. As can be seen the building satisfies the criterion for the heating but not the cooling one. In this case the engineer should take various measures in order to reduce the cooling load such as, use better type of fenestration or improved wall insulation and if these are not enough, in cooperation with the architect, should try to reduce the glazing area, add overhangs, etc.

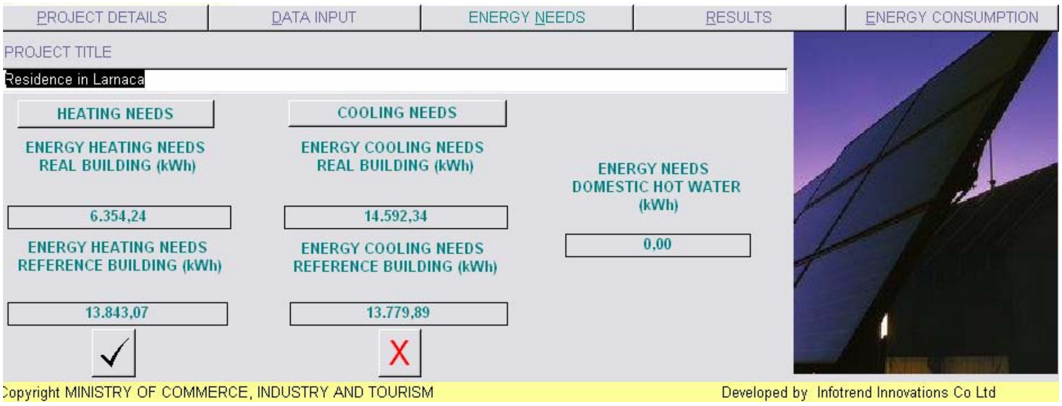

Fig. 2 Output of the software developed for the energy performance analysis of buildings

CONCLUSIONS

The methodology and program developed in Cyprus to satisfy the Directive on the energy performance of buildings is presented in this paper. It is believed by the authors that this methodology will improve the thermal performance of buildings in Cyprus as at the moment there are no building-regulations concerning thermal insulation of the building envelope. Considering that Cyprus is fully dependant on imported energy, it is believed that this new directive will lead to energy efficient buildings, and in the long run it will benefit the economy of the island. This will be more significant in a few years time as more strict minimum requirement will be imposed, not considered now in order not to upset the industry and the methodology will expand in new areas not covered at present (e.g. ventilation).

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The simple hourly method of prEN 13790: a dynamic method for the future

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SUMMARY

The heat and cooling needs of a building considering climate, building envelop, energetic and ventilation system and indoor climate is at the heart of the set of CEN standards devoted to the application of the European directive

This new standard described two methods: a monthly one, improved from the previous version of 13790 standard, and a new one, based on an hourly calculation considering the thermal balance of the room at each step

Why is it a simple method? Basically, one must consider the final user position. Then, and whatever the calculation method is, a simple method must rely on simple input (easy to obtain, easy to understand), and outputs easy to check. The simple hourly methods aims to fulfil these points as its inputs are the same as for the monthly method.
One application is the new French RT2005 regulation applicable to building built after 1st September 2005.

INTRODUCTION

The European Energy Performance in Building Directive (EPBD, 2002) demands that before the year 2006 all Member States of the European Union implement energy performance (EP) regulations, including minimum EP requirements for all new buildings and EP certificates for all existing buildings when built, sold or rent. In a short period, a whole set of European (CEN) standards have been developed to facilitate the Member States with suitable calculation methods and/or performance criteria for calculation methods. These methods cover building transmission and ventilation heat loss, internal and solar gains, daylighting, heating and cooling demand, heating and cooling system losses, energy consumption for hot water production, ventilation systems and lighting. Evidently they also include renewable energy systems.

The directive explicitly states that the European Commission intends further to develop standards such as EN ISO 13790, also including consideration of air-conditioning systems and lighting. EN ISO 13790 ‘Calculation of Energy Use for Space Heating', EN ISO 13790, 2003) is the international standard for the calculation of the energy use for space heating for residential and non-residential buildings. It is the successor of the possibly still better known residential-only EN 832 'Calculation of energy use for heating – Residential buildings’.

THE PREN 3790 STANDARD

STRUCTURE OF THE STANDARD

The structure of the draft of the revised EN ISO 13790 (2) has been adapted to maximize the common use of procedures, conditions and input data, disregarding the type of method:

- Full description of a monthly (and seasonal) method for cooling, very similar to the method in the current EN ISO 13790:2003 for heating.
- Full description of a simple hourly method for heating and cooling, to facilitate easier introduction of hourly and weekly patterns (e.g. controls, user behaviour)
- For dynamic simulation methods: procedures concerning boundary conditions and on input data, that are consistent with the boundary conditions and input data for more simplified types of methods.

The latter is to ensure compatibility and consistency between the different types of methods.

The standard provides for instance common rules for the boundary conditions and physical input data irrespective of the chosen calculation approach.

BASIS OF THE HOURLY METHOD

The model is a simplification of a dynamic simulation, with the following intention:

- same level of transparency, reproducibility and robustness as the monthly method:
- clearly specified, limited set of equations, enabling traceability of the calculation process,
- reduction of the input data as much as possible,
- unambiguous calculation procedures;
- with main advantage over the monthly method that the hourly time-intervals enable direct input of hourly patterns.

In addition, the model provides the following:

making new development easy by using directly the physical behaviour to be implemented;

keeping an adequate level of accuracy, especially for room-conditioned buildings where the thermal dynamic of the room behaviour is of high impact.

The model used is based on an equivalent resistance — capacitance (R-C) model. It uses an hourly time step and all building and system input data can be modified each hour using schedule tables (in general, on a weekly basis).

The model makes a distinction between the internal air temperature and mean temperature of the internal (building zone facing) surfaces (mean radiant temperature). This enables its use in principle for thermal comfort checks and increases the accuracy for taking into account the radiative and convective parts of solar, lighting, and internal heat gains, although one should be aware that the results of the simple method at hourly level are not validated.

The calculation method is based on simplifications of the heat transfer between the internal and external environment, as shown in Figure 1.

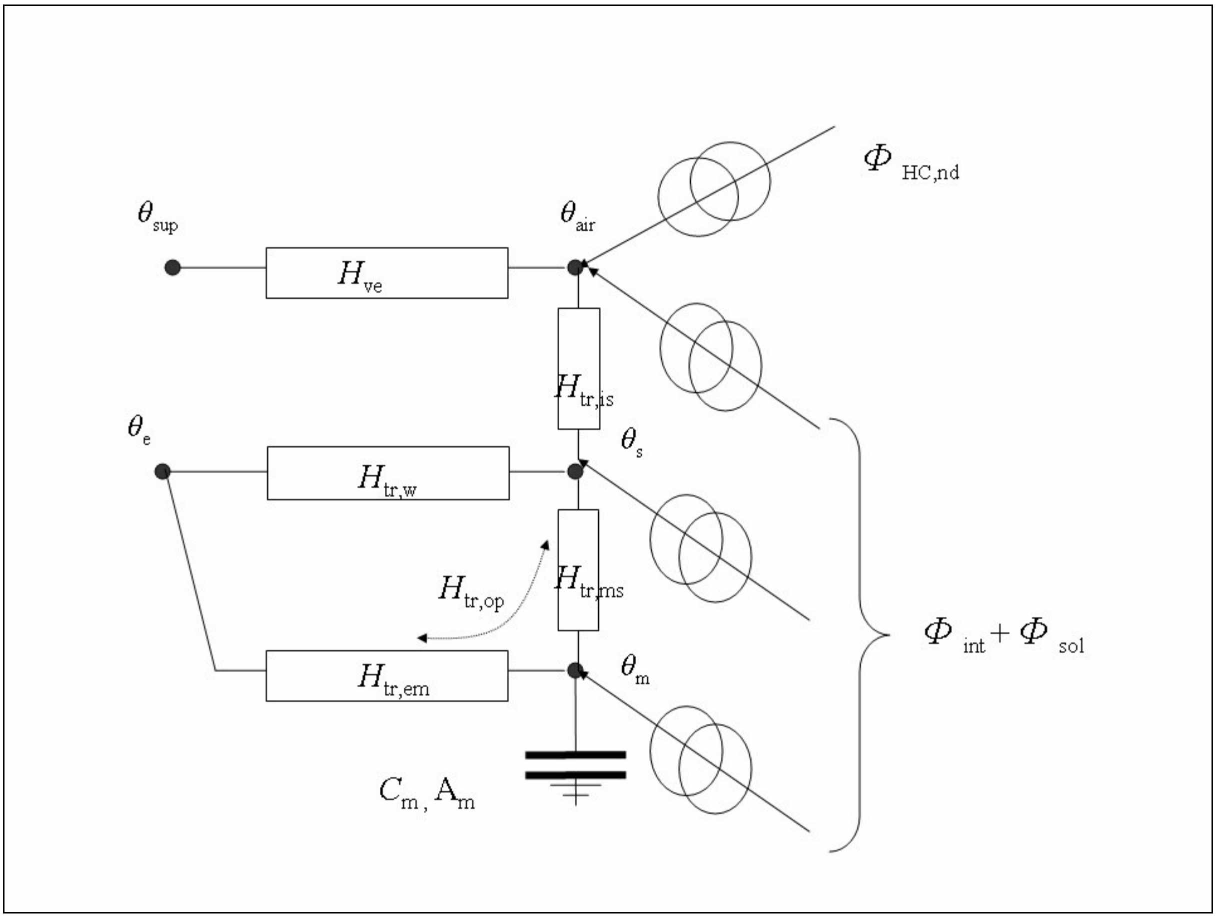


Figure 1 – 5R1C model

The heating and/or cooling need is found by calculating for each hour the need for heating or cooling power ΦHC,nd (expressed in watts and counted positive for heating and negative for cooling), that needs to be supplied to or extracted from the internal air node (□air) to maintain a certain minimum or maximum set-point temperature. The set-point temperature is a weigthed mean of air and mean radiant temperature. The default weighting factor is 0,5 for each.

Heat transfer by ventilation, Hve, is connected directly to the air temperature node θint, expressed in degrees Celsius, and to the node representing the supply air temperature θsup.

Heat transfer by transmission is split into the window part, Htr,w, expressed in watts per kelvin, taken as having zero thermal mass, and the remainder, Htr,op, expressed in watts per kelvin, containing the thermal mass which in turn is split into two parts: Htr,em and Htr,ms. Solar and internal heat gains are distributed over the air node, θair, the central node, θs (a mix of θair and mean radiant temperature θr,mn) and the node representing the mass of the building zone θm. The thermal mass is represented by a single thermal capacity, Cm, expressed in megajoules, located between Htr,ms and Htr,em. A coupling conductance is defined between the internal air node and the central node. The heat-flow rate due to internal heat sources, θint, expressed in watts, and the heat-flow rate due to solar heat sources, □sol, expressed in watts, are split between the three nodes.

The hourly energy needs for heating and/or cooling, QHC,nd, expressed in megajoules, are obtained by multiplying ΦHC,nd, expressed in watts, by 0,036. Similarly, the internal and solar heat gains, Qint and Qsol, expressed in megajoules, are obtained by multiplying Φint and Φsol respectively, expressed in watts, by 0,036.

VALIDATION TESTS

PRESENTATION OF THE PREN 15265 STANDARD (1)

This standard was prepared by the TC89 WG6 group. It specifies a set of assumptions, requirements and validation tests for procedures used for the calculation of the annual energy needs for space heating and cooling of a room in a building where the calculations are done on an hourly basis or less.

The standard does not impose any specific numerical technique for the calculation of the room heating or cooling need and the internal temperatures of a room.

The purpose of this standard is to validate calculation methods used to:

- describe the energy performance of each room of a building;
- provide energy data to be used as interface with system performance analysis) HVAC, lighting, domestic hot water, etc).

The validation procedure is used to check the energy need for space heating and cooling based on a transient sensible heat balance model, taking into account:

the external surface heat balance
the conduction through the building envelope
the thermal capacities of external and internal structures
the internal surface heat balance
the air heat balance
the heat balance solution method

All other aspects are given either by prescribed boundary conditions or by input data and are not part of the model validation. It is assumed that prescriptive calculation models have to be used according to existing European standards.
The system performance analysis and moisture balance are not within the scope of this standard.

VALIDATION TESTS

This standard does not impose any specific numerical technique for the calculation of the room heating or cooling load and the internal temperatures of a room.
For the validation of any existing or new numerical solution to the assumptions and the procedures defined in this standard, the procedures included in this clause shall be satisfied.
For the validation tests specified in this clause, the results provided by any numerical solution model shall be within the range indicated for each test.

The internal dimensions of the room are: length = 3,6 m; depth = 5,5 m; height = 2,8 m. The external wall including window glazing is facing West. The areas of the components of the reference room are given in Table 2.

Table 1 — Areas of reference room components

	External wall	Window Glazing	Internal wall left	Internal wall right	Internal wall back	Floor	Ceiling
Area (m²)	3,08	7,0	15,4	15,4	10,08	19,8	19,8

The solar parameters (considered here as independent of the solar angle) of the glazing component are given in Table 2:

Table 2 — Solar parameters of the window glazing components

Component	Transmittance	Reflectance	Absorptance
Shading	0,20	0,50	0,30
Pane	0,84	0,08	0,08

For opaque components the following values are taken:

solar absorptance of all wall surfaces □sr = 0,6

solar absorptance of the roof

αsr = 0,9

The thermophysical characteristics of the window components are:

a) Shaded DP - Double pane glass with external shading device (Figure 2):

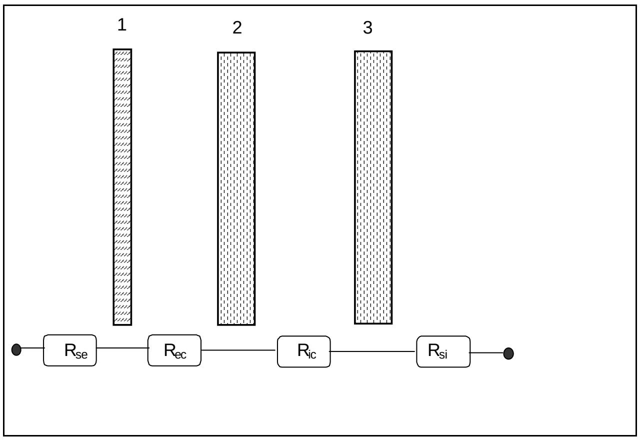


Figure 2 — Double pane window glazing with external shading

Thermal resistances including convection and long wave radiation:

external surface Rse = 0,0435 m2.K/W

cavity between external blind and external pane Rcav = 0,080 m2.K/W

cavity between external pane and internal pane Rie = 0,173 m2.K/W

internal surface Rsi = 0,125 m2.K/W

Thermal transmittance of the glazing system Ug = 2,37 W/(m2.K)

Total solar energy transmittance of the glazing systemg = 0,20

b) DP - Double pane glass without external shading device (Figure 3):

Thermal resistances:

external surface Rse = 0,0435 m2 K/W

cavity between external pane and internal pane Rie = 0,173 m2 K/W

internal surface Rsi = 0,125 m2 K/W

Thermal transmittance of the glazing system Ug = 2,93 W/(m2K)
Total solar energy transmittance of the glazing systemg = 0,77

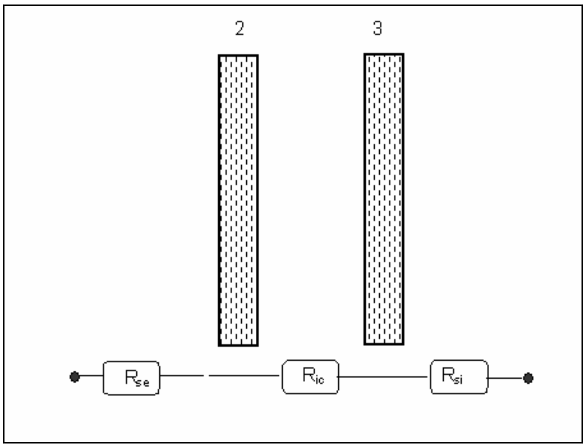


Figure 3 — Double pane window glazing without shading device

The thermophysical characteristics of the walls, ceiling and floor are given in Table 3.

Table 3 — Thermophysical properties of the opaque components
(order of layers from external to internal)

	d m	λ W/(m·K)	ρ kg/m³	cp kJ/(kg·K)
Type 1 (external wall)				
outer layer	0,115	0,99	1800	0,85
insulating layer	0,06	0,04	30	0,85
masonry	0,175	0,79	1600	0,85
internal plastering	0,015	0,70	1400	0,85
Type 2 (internal wall)				
gypsum plaster	0,012	0,21	900	0,85
mineral wool	0,10	0,04	30	0,85
gypsum plaster	0,012	0,21	900	0,85
Type 3c (ceiling)				
plastic covering	0,004	0,23	1500	1,5
cement floor	0,06	1,40	2000	0,85
mineral wool	0,04	0,04	50	0,85
concrete	0,18	2,10	2400	0,85
Type 3f (floor)				
concrete	0,18	2,10	2400	0,85
mineral wool	0,04	0,04	50	0,85
cement floor	0,06	1,40	2000	0,85
plastic covering	0,004	0,23	1500	1,5
Type 4c (ceiling/roof)				
plastic covering	0,004	0,23	1500	1,5
cement floor	0,06	1,40	2000	0,85
mineral wool	0,04	0,04	50	0,85
concrete	0,18	2,10	2400	0,85
mineral wool	0,10	0,04	50	0,85
acoustic board	0,02	0,06	400	0,84
Type 4f (floor)				

acoustic board	0,02	0,06	400	0,84
mineral wool	0,10	0,04	50	0,85
concrete	0,18	2,10	2400	0,85
mineral wool	0,04	0,04	50	0,85
cement floor	0,06	1,40	2000	0,85
plastic covering	0,004	0,23	1500	1,5
Type 5 (roof)				
rain protection	0,004	0,23	1500	1,3
insulating	0,08	0,04	50	0,85
concrete	0,20	2,1	2400	0,85

The hourly mean values of climatic data are given in an annex of the satndard, starting on 1st January and ending on 31st December.
The latitude is 49° and the values are given in solar time.

RESULTS AND DISCUSSION

The results for heating, QH, and cooling, QC, (expressed in kWh) are provided for the complete year (the hourly pattern of the calculated internal temperatures is not part of the validation) and compared to reference values by calculating: Different software were used to produce the reference results which are given in Tables 5 and 6.

rQH = abs(QH - QH,ref) / Qtot,ref
rQC = abs(QC – QC,ref) / Qtot,ref

Three levels of accuracy can be checked with levels: A,B,C. The validation tests are complied if for each of the cases 5 to 12 (8 test cases) :

Level A : rQH ≤ 0,05 and rQC ≤0,05
Level B : rQH ≤ 0,10 and rQC ≤0,10
Level C : rQH ≤ 0,15 and rQC ≤0,15

Table 4 — Reference results for tests 1 to 4 (informative)

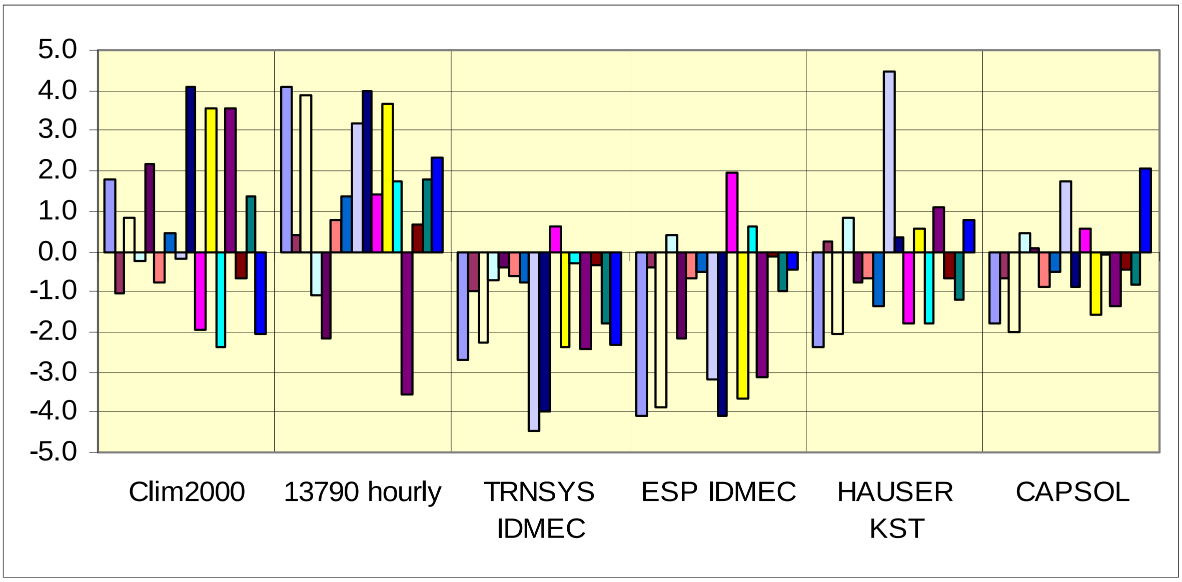
Test	QH,ref kWh	QC,ref kWh	Qtot,ref kWh
1	748,0	233,8	981,8
2	722,7	200,5	923,2
3	1368.5	43,0	1411,6
4	567,4	1530,9	2098,3

Table 5 — Reference results for tests 5 to 12 (normative)

test n°	QH,ref kWh	QC,ref kWh	Qtot,ref kWh
5	463,1	201,7	664,8
6	509,8	185,1	694,9
7	1067,4	19,5	1086,9
8	313,2	1133,2	1446,4
9	747,1	158,3	905,4
10	574,2	192,4	766,6
11	1395,1	14,1	1409,3
12	533,5	928,3	1461,8

The results obtained by different computer codes are shown in figure 3

Figure3 : results of test cases for different methods



The 13790 hourly method, though more simplified than the majority of the other ones, provides results with, as the other ones, the best level of accuracy (A).

CONCLUSION

The European Energy Performance in Building Directive (EPBD) requires methods for the calculation of the energy performance for use in the context of building regulations. A pair of simplified methods have been developed for the calculation of the energy needs for heating and cooling of buildings. A revised version of EN ISO 13790 has been prepared in which a level playing field is created for both simple and detailed methods, by introducing a coherent set of procedures with respect of boundary conditions and assumptions that applies to different types of methods. Validation tests show that the hourly simplified method is well suited for calculation both heating and cooling needs and therefore to be used in both warm, moderate and cold European climates while keeping input data as the same level of simplicity (and the robustness) than monthly ones..

REFERENCES

1 - prEN 15265 " Thermal performance of buildings — Calculation of energy needs for space heating and cooling — General criteria and validation procedures " CEN TC89 WG6

2 - ISO/FDIS 13790 "Energy performance of buildings — Calculation of energy use for space heating and cooling" CEN TC 89 WG4

Calculation of the energy efficiency of ventilated and air-conditioned buildings

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SUMMARY

The paper presents a part of the methodology for energy efficiency calculation in buildings, elaborated in Romania, regarding especially the ventilated and air-conditioned buildings. After the adoption in our country of the EU regulation no. 91/2002 as internal law no. 325/2005, the Faculty of Building Services from the Technical University of Civil Engineering Bucharest was designed by the Public Ministry of Constructions to elaborate the calculation methodology for the energy efficiency of buildings. Knowing the extent of the EU regulations in this field, we decided that Romanian methodology could correspond to the greatest part of the European similar methodology. For the air-conditioned buildings, we selected the monthly simplified method as appropriate. For more particular cases, we recommend the simplified method with a one-hour time step and, for very special situations we recommend the use of specialized software, which should agree to the conformity tests proposed by the EU legislation. We made out observations regarding the difficulties faced out during our work, as well as comments regarding the capability of the simplified method to consider the energy consumed at the air-conditioning system level for water vapor condensation, a special situation appearing in our climate, even if the room internal humidity is not controlled.

INTRODUCTION

Consistent actions at national level, regarding building energy consumption performance and reduction, have been started by applying the law 10/1995 concerning the “Quality of Constructions”. Consequently, this law has established rules and objectives related to comfort, indoor air quality and energy consumption reduction for new buildings [1].

Two years later, several normatives (C107/1-5) has imposed design demands for thermal and humidity transfer through building envelope for new projects or rehabilitation actions. These normatives have been introduced limits concerning the minimal thermal resistance for external construction elements and global heat losses coefficient.

In addition, since 2000, there have been 3 national normatives (NP 047, 048 and 049) implemented for expertise, certification and audit energetic actions concerning existing buildings. The field of application for these normatives is dealing only with building envelope, heating and hot water production systems.

Moreover, the law 325/2002 has established a Rehabilitation National Programme for block of flats (collective dwellings) that represents the main type of dwellings within Romania’s cities. The specialist qualification in the field of energetic expertise, certification and audit projects for civil buildings has been conducted since 2003 by following the OG 550 (government legislative act emitted by Ministry of Transports, Constructions and Tourism). In the same time, the equivalent procedure for industrial buildings has been coordinated by Ministry of Energy.

METHODS

Implementation of European Building Performance Directive (EPBD) in Romania
More recently, in December 2005, the law 372/2005 dealing with energetic performance of buildings has been approved [2]. It is worthwhile to mention that this law transposes UE Directive 91/2002 in Romania. As a consequence, the Calculation Methodology of Buildings Energy Efficiency has been fulfilled in 2006 and adopted on January 1st 2007 [3]. This method comprises three distinct parts: the first one is dealing with envelope building issues, the second one concerns computations for heating, production of hot water, ventilation, air conditioning, and lighting energy consumption while the last one refers to energetic certification and audit. The field of application for this methodology refers *to all buildings* within Romania.

On another hand, there are specific actions for using specialized software in the field of energetic certification and audit:

- TRNSYS tests: this code has been extensively tested and validated following prEN 15265 procedures by the research team of Building Services Laboratory of Technical University of Civil Engineering Bucharest; taken into account that this code is very popular among the specialized research community, it is planned to develop special executable versions adopted for different kinds of buildings using more friendly interfaces (the computation method will be based on detailed hourly methods)
- cooperation between AIIR (Romanian Association of Building Services Engineers) and VABI (Society for Automation in Buildings and Building Services, Netherlands): the aim is to adapt and use VABI's software to specific Romanian conditions for energetic certification and audit projects completed by Romanian recognized experts
- use of software: any Romanian or foreign software that fulfills the testing demands recommended by prEN 15625 will be homologated for use by energetic auditors

We add here that another normative is to be prepared (by Ministry of Transports, Constructions and Tourism and Ministry of Industry and Commerce), concerning the method and the analysis of its results concerning the assessment of HVAC systems, as well as the procedure for specialist's training according to the proposed methodology. The objective of this action is to answer to the UE Directive 91/2002 specific demand related to inspection of air conditioning systems.

As actions carried out, we quote here the programs accomplished according to OG 211/2003 and OG 187/2005 (government legislative act). The way the population can be assisted to finance the expenses required for thermal rehabilitation works of the buildings within these programs is the following: 1/3 allowances from the state budget; 1/3 yearly approved funds to this aim from the local budgets and 1/3 repair funds of the owner associations. The main projects carried out or to be carry out are the following:

- pilot actions (2 blocks of flats – collective dwellings), 2003-2004
- Romania/Swiss Program (12 blocks of flats – collective dwellings, 2003-2004
- 31 locations within a new program assisted by Swiss Embassy, 2005-2007

- government action for 391 buildings in 17 localities (approximately 14,4 MEuro), 2006-2007

We present in the next section details regarding the application of Romanian computation method for buildings energy efficiency, with focus on issues dealing with ventilation and air conditioning systems energy consumption.

Particularities of Romanian Buildings Energy Efficiency Methodology

Romanian calculation methodology of buildings energy efficiency was basically developed using European standards and European proposed standards but also taken into account issues from Romanian normatives (referenced below).

As already stated, the method consists of 3 main parts:

a) thermal characteristics of building's envelope (building overall characteristics – HT, HV – SR EN ISO 13789 [4]; design thermal values – EN ISO 10456; thermal resistance and thermal transmittance – EN ISO 6946; thermal bridges and surface temperatures – EN ISO 10211, EN ISO 14683, EN ISO 13788; thermal performance of windows, doors and shutters – EN ISO 10077-1, EN ISO 10077-2, EN 410, EN 673, EN 13363; heat transfer via the ground – EN ISO 13370/national method as alternative)

b) building's services energetic consumptions (heating – SR EN ISO 13790 [5], prEN 15316), prEN15315 (primary energy, CO2), NP 048 (revised) as alternative calculation method; hot water - prEN 15316, NP 048 (revised) as alternative calculation method; ventilation/ air conditioning – SR EN ISO 13790, pr EN 15241, pr EN 15242, pr EN 15243, pr EN 15265, pr EN 13779 [6], pr EN 13792, alternative calculation method for energy consumption for cooling and free internal temperature in summer; lighting – pr EN 15193/1)

c) certification and energy audit

Concerning the method for assessing ventilation and air conditioning systems energy consumption, there are the following computation procedures:

- energy consumption for cooling (monthly method according to European standards)
- energy consumption for cooling (detailed hourly method according to European standards)
- evaluation of internal air temperature during summer (according to French normative– RT 2005)
- evaluation of real ventilation flow rates and ventilation systems energy consumption (according to European standards)
- evaluation of air conditioning systems energy consumption (according to European standards)

The procedures listed before follow generally the methods presented in European standards or proposal of European standards, adding local data and references. Concerning the estimation of energy consumption for cooling, another energy balance scheme was defined as explained further below.

The energy balance diagram (see figure 1) is extremely important for building – system coupling. It allows rapid assessment of energetic consequences in the case of system revision or adaptation of a part of the system. As a result, the auditor can correctly and quickly operate changes to improve the system energy consumption.

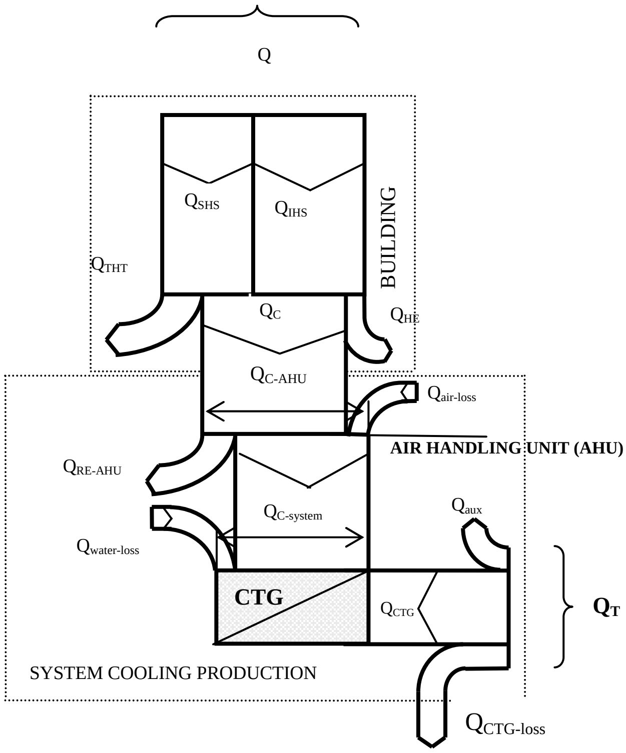

Figure 1: Flows and energy balances for building and systems
Q – overall building thermal balance ; Q_{SHS} – solar heat sources; Q_{IHS} – internal heat sources; Q_{THT} – transmission heat transfer; Q_{HE} – heat extracted by free cooling; Q_C – energy need for space cooling; $Q_{C\text{-}AHU}$ – energy need for cooling at AHU; $Q_{C\text{-}system}$ – energy need for cooling at system cooling production; $Q_{RE\text{-}AHU}$ – renewable energy for cooling used at AHU (energy gain); $Q_{air\text{-}loss}$ – energy loss due to air leakage and heating of the air in ducts; $Q_{water\text{-}loss}$ – energy loss in system distribution; Q_{CTG} – system cooling production energy consumption; $Q_{CTG\text{-}loss}$ – energy loss within system cooling production; Q_{aux} – energy consumption for auxiliary equipments within system cooling production (pumps, fans, etc.); Q_T - total delivered energy for cooling (total energy consumption for building cooling)

The diagram takes into account the system components and their task within the system, introducing 3 levels of energy balance: building, air handling unit and cooling production. An interface, labeled CTG – cooling thermodynamic generator (Figure 1), was placed between thermal energy extracted from the system and primary energy delivered in order to indirectly estimate the performance coefficient of thermodynamic cooling system. It is worthwhile to mention that in the Figure 1 other heat flows can be added depending on system characteristics. These can straightforwardly attached by the auditor (for example renewable energy sources or energy recovery that takes place frequently in the air handling unit). According to Figure 1, the building energy balance can be written in the following manner in order to evaluate the energy need for space cooling:

$$Q_C = Q_{SHS} + Q_{IHS} - Q_{HE} - Q_{THT} \qquad (1)$$

In addition, we can straightforwardly calculate the energy need for cooling at air handling unit, taken into account all the fluxes that take place at this level on Figure 1 (energy need for space cooling, adding energy loss due to air leakage and heating of the air in ducts and deducting renewable energy for cooling used at AHU that reduces energy demand for cooling at AHU):

$$Q_{C\text{-}AHU} = Q_C + Q_{air\text{-}loss} - Q_{RE\text{-}AHU} \qquad (2)$$

Taken into account energy loss within system distribution from cooling production to AHU (for example due to heat gain of chilled water and on occasion water leakage in system distribution), we can write the energy balance for system cooling production:

$$Q_{C\text{-}system} = Q_{C\text{-}AHU} + Q_{water\text{-}loss} \qquad (3)$$

Further, we can assess the total delivered energy (electricity, gas, coal, oil, etc.) for space cooling, given mainly by system cooling production energy consumption along with energy loss within this system and energy consumption for its auxiliary equipments (pumps, fans):

$$Q_T = Q_{CTG} + Q_{CTG\text{-}loss} + Q_{aux} \qquad (4)$$

We consider that such diagrams as that indicated in Figure 1 (completed with specific values for flux energy) could be extremely useful. This can be performed for different types of buildings (diverse structures and occupations) and climates. We believe that a European Project on this topic could lead to valuable data. For example, we can compare the diagram of a building and its cooling system with the better diagram achieved for a “reference” building equipped with a “reference” cooling system. This allows directly identifying the solutions for reducing energy consumptions. This kind of results could help enormously the auditors as well as the architects and engineers involved in building conception and its HVAC systems.

It is obvious that the latest form of the Methodology for Building Performance takes into account some simplifications. We decided to point out one of them, by showing some simulation results of latent and sensible cooling loads resulting from a cooling coil operation during a summer period, for a given location. The aim of these simulations was to judge the importance of the latent cooling power within the cooling process, knowing that the latest Methodology underestimates it, taking into account only the sensible cooling. The study is presented in the next paragraph.

Assessment of cooling demand for vapor condensation

The climate in Southern part of Romania is rather hot and humid during summer. For instance, external temperature and air moisture content taken into account for the design of HVAC systems in Bucharest are 34,5°C and 11,95 g/kg, respectively. Therefore the energy loss due to vapor condensation on cooling coils surface may be high and thus not be ignored without the risk to underestimate energy consumption for cooling. In order to verify this supposition, we carried out a TRNSYS simulation [7], in which we modeled a cooling coil operation under similar weather conditions taken from a typical weather file for Bucharest.

The TRNSYS model had focused on the cooling coil thermal behavior when this coil is part of an air handling unit traversed by an outdoor airflow equal to 15 000 m^3/h. The set point temperature of the air exiting the cooling coil is 22 ºC, temperature at which it will be supplied in the ventilated zone as primary air, for dilution purposes. The zone thermal load was supposed to be removed by an air-water system placed within the zone (fan coil units or similar).

The general overview of the TRNSYS model for the cooling coil is depicted in figure 2, where we can notice the following TRNSYS subroutines as component parts of the model:

- ”Weather” – corresponding to the weather data file processing, supplying information for the outdoor air entering the cooling coil; the weather file was taken from the Meteonorm database and corresponded to the town of Bucharest, Romania;

- ”Cooling coil” - models the cooling coil, describing the heat and mass transfer between the airflow passing through the coil and the chilled water circulating inside the coil pipes; the OUTPUTS of interest for this model which will be represented are: the outlet air temperature ($t_{out,CC}$), the total cooling power ($P_{tot, cool}$), the sensible cooling power ($P_{sens,cool}$) and the latent cooling power ($P_{lat,cool}$) resulted from the transfers within this heat exchanger;
- ”Water flow calculation” – which calculates the water flow entering the cooling coil, depending on the signal received from the ”Temperature controller” routine (a PID controller)-see figure 1; this signal results from the comparison between the air temperature setpoint imposed at the coil exit (22 ºC) and the effective temperature calculated by the model in the same place; the PID controller compares this temperature difference and conveys a signal to the vane actuator which will change the vane position to adjust the chilled water flow through the pipes, in order to reach the air temperature setpoint at the coil exit. If the air temperature entering the coil falls below the setpoint value (during the night), the model is able to assume that is no heat and mass transfer in the coil, so the exit air temperature will equal the entering one.
- ”By pass flow” – calculates the chilled water flow by-passing the cooling coil as a result of the PID regulation;
- ”Conversion routine”- transforms the TRNSYS default units for cooling power (kJ/h), in SI units (kW), for the three calculated cooling powers: total, sensible and latent.
- ”Psychrometrics” - calculates the remaining air parameters when two of them are supplied (e.g. dry air temperature and relative humidity);
- and finally, the subroutine ”Output to file” enables TRNSYS post processor to write the output variables of the model in a text file.

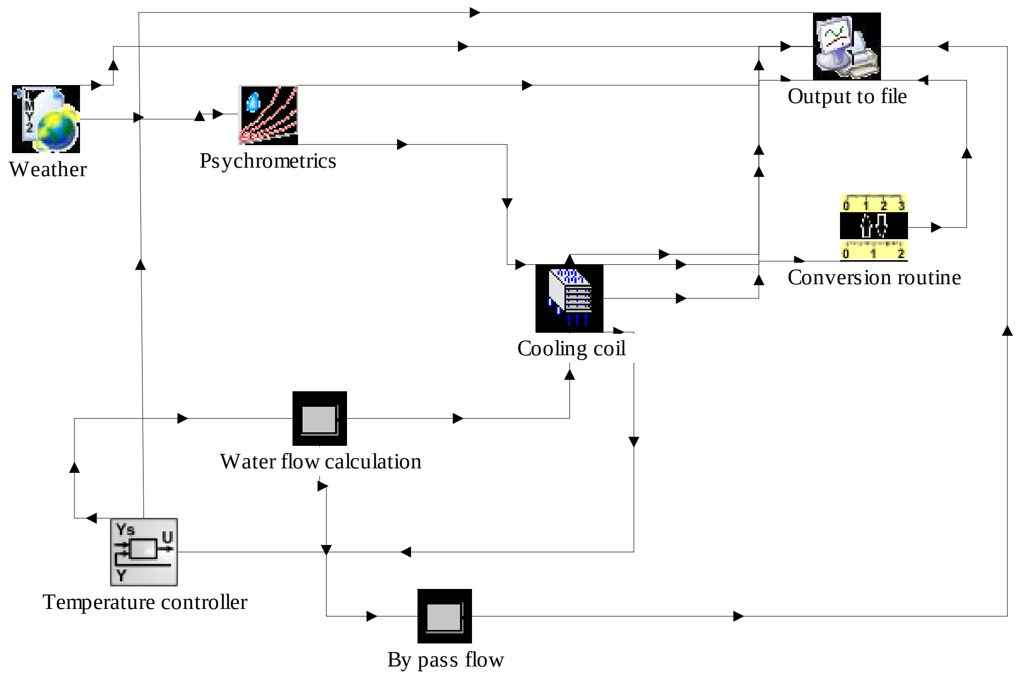

Figure 2: The general overview of the TRNSYS model of the cooling coil

The simulations were performed for one month summer period, more precisely: from 1st July to 31st July, considering the Bucharest weather file loaded in the TRNSYS model.
The results issued from these simulations are represented in the figures 3 and 4, for the first and the third week of the simulation period.

RESULTS

In the figure 3 are represented on the same graph the three component of the cooling power: total (Ptot_cool), sensible (Psens_cool) and latent (Plat_cool), as well as the outside air temperature entering the cooling coil (t_outside) and the exit air temperature from the cooling coil (t_out_CC). The results correspond to the second week of the simulation period (from 8 to 15th July). We can notice on this figure two important behaviors:

- the good regulation of the temperature t_out_CC at its setpoint value (22 ºC) during the periods when t_outside overtakes this value, and
- the small values of the latent power, appearing only when the dew point of the temperature entering the cooling coil (t_outside) is above the mean temperature of the coil surface (equivalent to the average chilled water temperature across the coil); this is equivalent to the difference between the moisture content of the air entering the cooling coil and the moisture content of the air exiting the cooling coil, producing condensation on the external coil surface.

If we integrate these cooling powers over the whole week, it results a total cooling load $Q_{tot,cool}$ = 4987 Kj and a latent cooling load $Q_{lat,cool}$= 260 kJ, or a 5,2% from $Q_{tot,cool}$.

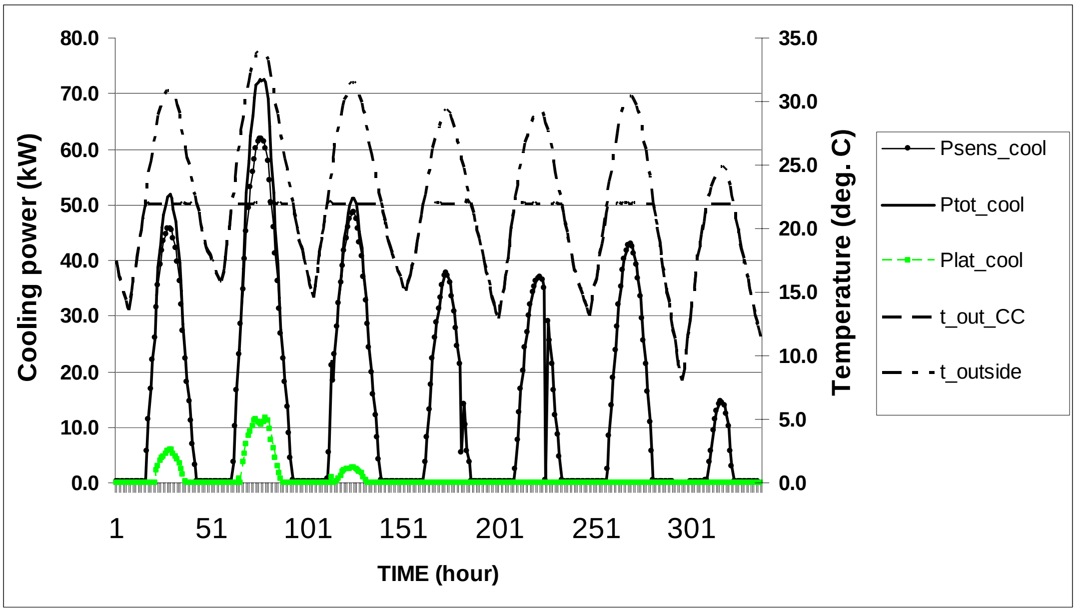

Figure 3: Temperatures and cooling powers calculated for the second week of simulation

A similar representation is depicted in figure 5, but for the third week of the simulation period (16th – 23rd July). We could notice than the latent power profile and magnitude is quite different than in the first case. This is due to the moisture content difference between the air entering and the air exiting the cooling coil, which is bigger for this period of time, according to the meteorological data picked from the Bucharest weather file.

If we integrate again these cooling powers over the third week, it results a total cooling load $Q_{tot,cool}$ = 4717 Kj and a latent cooling load $Q_{lat,cool}$= 1084 kJ, equivalent to 23% from $Q_{tot,cool}$. This result shows clearly that the moisture content of the air entering the cooling coil (in this case, outside air) is of great importance, and for warmer and more humid climates the percentage of the latent cooling load from the total cooling load could be even bigger than 23% and the neglecting of this part could mean to big underestimations.

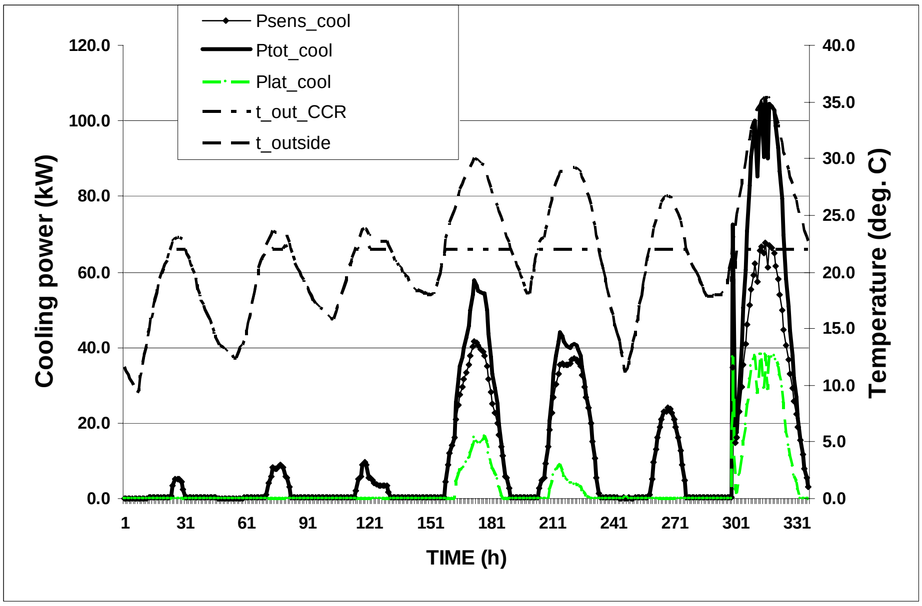

Figure 4: Temperatures and cooling powers calculated for the third week of simulation

DISCUSSION

Concerning the implementation of UE Directive 91/2002 in Romania, we notice that important steps forward have been made, especially by establishing Calculation Methodology of Buildings Energy Efficiency (based mainly on European documents within EPBD). On the other hand, there still is a lack of national data (as climatic data that need to be urgently updated). Moreover, it is necessary to elaborate a kind of “Application handbook” of national Buildings Energy Efficiency Method in order to add specific data for different parameters that can be used as “default” values.

There is a need also to improve the computation precision without making the methods more complicated. It was shown for instance that if we do not take into consideration energy demand for latent heat, the differences can reach 23% of energy underestimated from the total energy consumed for cooling.

ACKNOWLEDGEMENT

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Proposal of a statistical model for the estimation of energy demand for space heating in residential buildings

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SUMMARY

This study deals with the statistical analysis of a set of data related to 50 multi-family residential buildings, located in the district of Torino (Italy), in order to find simplified correlations for the assessment of the energy demand for space heating.
Starting from the standard deterministic equations, this study revises them maintaining their physical basis, and simplifies the most involved terms. This simplification is carried out making a statistical analysis of the energy quantities that affect the performance of a building. Results are compared with those calculated in a detailed way using the Standard procedures, for the available sample of buildings.
The resulting formulas define the relations between the energy performance of a building, and its geometrical and thermal properties.
Therefore, this model is particularly useful to estimate energy demand for space heating for existing residential buildings in a simple and prompt way, and it may represent a useful tool for their energy certification.

INTRODUCTION

The study of the building energy demand has become a topic of great importance not only among technicians, because of the significant increasing of interest around energy sustainability which has grown up after the emanation of the EPB European Directive [1].
The topic of the evaluation of the energy requirements through a theoretical calculation, following the asset rating approach [2], can be aimed at characterizing the theoretical energy performance [3] and at foreseeing the actual consumption of a building [4], [5].
Moreover, the energy performance benchmarking of a large number of buildings needs tools reliable and trustworthy, but also simple and fast, to reduce costs and data processing time.
The number of actors, technicians and not, interested in holding energy consumption provisional tools has stimulated the search of new, and possibly simplified, models based on statistical analysis [6].
In Italy, an interesting example is CONSIP model [7]. CONSIP society, controlled by the Italian Finance and Economy Ministry, was born with the goal of rationalize expenses for the Public Administration assets and services. It has defined a methodology for the determination of the heating primary energy demand and the quality indicators of the offered service, regarding the tenders for the integrated management service supply (Global Service) of all the Public Administration buildings. In Consip model, the equation of the transmission loss coefficient has a statistical origin, because its proportionality coefficients for geometrical and thermo-physical quantities are obtained with a linear regression on a sample of buildings.
The present study considers only space heating of residential buildings, which are responsible of an important part of the total primary energy demand: in EU, space heating accounts for

more than 50% of primary energy demand of residential and service buildings, which account for 40% of total primary energy demand [8]; similar percentages are for the other developed countries.
Starting from Standard deterministic equations, we revises them maintaining their physical meaning, and simplify the most involved terms. This simplification is carried out making a statistical analysis of the same terms calculated in a rigorous way (using Standard EN 832) on a sample of 50 multi-family residential buildings placed in the district of Torino (Italy). They were built in different periods between the end of the Nineteenth Century and the end of the last Century. Relationships are searched between the age of the building and its main geometric and thermo-physical properties: the shape ratio, the volume, the opaque and glazed surfaces, their transmittances, the external and internal temperatures. All the available data are filtered with a statistical approach and are suitably combined to find out general relations. The analysis pays particular attention to model the free heat gains (e.g. internal and solar gains). As a result, the number of requested input data needed to evaluate the energy performance of a building is very limited: the thermal transmittances of the most important envelope components and geometric data.
To validate the model, results are compared with those from the application of the European Standard EN 832 (included in the available data set).
The correlations that we have found determine in a direct, simplified and fast way the energy demand parameter, which otherwise would require an involved procedure, if calculated according to Standards approach. Therefore, this model is particularly useful to estimate energy demand for space heating for existing residential buildings, and it may represent a useful tool for their energy certification.

METHODS

The proposed model estimates the energy demand for space heating in standard conditions of the entire building, whose control volume is defined by the external envelope. It is a single zone model.
The two most important performance indexes are the transmission loss coefficient (C_d, in W/(m^3°C)) and the normalised energy demand for space heating (*FEN*, in kJ/(m^{3}DD)). The available data set of 50 residential buildings contains these indexes and all the related quantities, calculated according to Standards [9], [10], [11], [12], [13], [14].
C_d evaluates the heat lost by transmission, and depends on the geometrical and thermo-physical characteristics of the building. It is defined as follows:

$$C_d = \frac{\dot{Q}_t}{V \cdot (T_a - T_e)}, \qquad (1)$$

$$\dot{Q}_t = m_1 \cdot \left(\sum_{i=1}^{n_{op}} \left(U_{op,i} \cdot f_{op} \cdot S_{op,i} \cdot (T_a - T_e)\right) + \sum_{i=1}^{n_f} \left(U_{f,i} \cdot S_{f,i} \cdot (T_a - T_f)\right) + \sum_{i=1}^{n_c} \left(U_{c,i} \cdot S_{c,i} \cdot (T_a - T_c)\right) \right) + $$
$$+ m_2 \cdot \sum_{i=1}^{n_w} \left(U_{w,i} \cdot f_w \cdot S_{w,i} \cdot (T_a - T_e)\right), \qquad (2)$$

where *V* is the gross volume, *U* are transmittances (W/(m^2°C)), *S* are the surfaces, T_a is the internal design temperature (20°C), T_e is the external temperature, *f* are the orientation coefficients, *m* are the thermal bridge coefficients; subscripts are: *op* for vertical opaque

elements, f for basement floor or concrete slab on pilotis, c for attic floor or plane roof, w for glazed surfaces .
The terms in equations (1) and (2) have been elaborated with a statistical approach, focalizing the following aspects:

1. determination of the most important heat transfer surfaces (S_{op}, S_f, S_c, S_w)
2. orientation coefficient for vertical elements (f_{op}, f_w)
3. thermal bridge coefficients (m_1, m_2)
4. basement and attic temperatures (T_f, T_c)

The envelope components that contribute to dispersions are: external walls, windows and floors toward ground and roof. Generally, external walls are characterised by a single value of transmittance K_{op}, and S_{op} is their total surface. Vice versa, the building may have different types of windows, e.g. double glazing in the apartments and single glazing in common parts (typically stairs). For this reason and for a purpose of detail without giving up simplicity, the model suggests to specify the properties of two types of windows per building (S_{f1}, S_{f2}). The terms related to the basement and the concrete slab on pilotis are distinct, like those related to the roof and to the attic floor, because the corresponding temperatures are different.
From the sample of buildings, a mean statistical orientation coefficient has been calculated both for the vertical walls, f_{op}, and for the windows (without distinction of glass type), f_w: it is the mean value of the ratio between the increased element surface and the actual surface, for the whole sample of buildings, and weighted on the respective areas.
The thermal bridge coefficient for vertical opaque surfaces m_1 follows the prescriptions of the CTI SC1 recommendation [9]. Coefficient m_2, related to windows, is set m_2 = 1.2.
In the available data set, the design temperatures of the basement and of the attic are obtained according to the Italian Standard [10]; the present model uses their corresponding mean values calculated on the analysed sample of buildings.
U transmittances are determined according to CTI Italian recommendation [9] for opaque components, and to the European Standard [14] for windows.
The ventilation loss coefficient C_v accounts for a hourly air change n = 0.5 h^{-1}; this is a standard value, typical of residential buildings; the conversion coefficients to calculate the net heated volume from the building gross volume are taken from [9], and they depend on the age of the building.
FEN estimates the primary energy demand for space heating. It is defined as follows [12]:

$$FEN = \frac{Q}{\left(T_a - \overline{T}_e\right) \cdot V \cdot N} \qquad (3)$$

Q is the conventional primary energy demand (kJ), N are the days of the heating season and $\overline{T}_e$ is the mean temperature of external air for the heating season according to [14]; $\overline{T}_e$ = 5.56°C for Torino. In particular, Q is the heat lost by transmission and ventilation, reduced by the free gains (solar, internal), the result increased to count for the global efficiency (production, distribution, control and emission mean seasonal efficiencies).
The proposed *FEN* is based on the assumption that the thermal equipment works continuously, without intermittence or attenuation. Equation (3) has been elaborated in order to maintain its physical meaning and to simplify its calculation, with a statistical approach. The following formula is an elaboration of (3), and it underlies the most significant contributions:

$$FEN = \left(1 - \eta_u \cdot \frac{E_g}{E_l}\right) \cdot \left(C_d + C_v\right) \cdot \frac{86.4}{\eta_g} \;, \quad (4)$$

where η_u is the energy gains utilization factor, η_g is the mean global seasonal efficiency of the system, E_g are the energy gains (solar and internal), E_l is the energy lost by transmission and ventilation. 86.4 is a time conversion factor from days to seconds.
The energy lost by transmission through the envelope (kJ) can be expressed as:

$$E_{d,stat} = C_{d,stat} \cdot V \cdot (T_a - \overline{T}_e) \cdot 86.4 \cdot N \quad , \qquad (5)$$

$C_{d,stat}$ is calculated with the statistical correlation reported in the Result section, formula (13); this is an approximation, because $C_{d,stat}$ requires the design temperatures and not the mean seasonal temperatures. Nevertheless the only terms involved in the approximation are those related to the attic and basement, that contribute (where present) very slightly to the energy calculation.
Similarly, the energy lost by ventilation through the envelope (kJ) is:

$$E_{v,stat} = C_{v,stat} \cdot V \cdot (T_a - \overline{T}_e) \cdot 86.4 \cdot N \quad , \qquad (6)$$

$C_{v,stat}$ is the ventilation loss coefficient; as explained, it is a constant value for residential buildings.
The energy gains are mainly solar and internal.
Internal energy gains are calculated using the following (in kJ):

$$E_i = Q_{i,spec} \cdot n_{apt} \cdot N \cdot 86.4 \quad , \qquad (7)$$

where n_{apt} is the number of apartments in the building and $Q_{i,spec}$ is the endogenous heat produced per apartment; in the available data set, $Q_{i,spec}$ is calculated with the formula reported in [9] for residential buildings:

$$\begin{aligned} Q_{i,spec} &= \left(6.25 - 0.02 \cdot S_{apt}\right) \cdot S_{apt} \qquad && S_{apt} \le 200\ m^2 \\ Q_{i,spec} &= 450 && S_{apt} > 200\ m^2 \end{aligned} \qquad (8)$$

The analysis of the results obtained with (8) led to express $E_{i,stat}$ with the following equation:

$$E_{i,stat} = \left(Q_{i,stat} \cdot C_{st} \cdot \frac{S_{rf}}{n_{apt,rf}} \right) \cdot n_{apt} \cdot N \cdot 86.4 \, , \qquad (9)$$

$Q_{i,stat}$ represents the endogenous heat per unit of area obtained applying (8) to the reference apartment of the examined sample of buildings, whose floor surface is the mean value of the standard apartment of these buildings ($S_{stat,ref}$ = 83 m^2, std.dev = 16 m^2). C_{st} is the reduction factor of the reference floor and it considers the common parts of the building included into the control volume (e.g. the stairs area); it is calculated as the mean value for the available data set. S_{rf} is the surface of the reference floor while $n_{apt,rf}$ is the number of apartments in the reference floor of the building. S_{rf}, $n_{apt,rf}$ and n_{apt} are input data of the model.
The starting point to calculate solar gains are formulas reported in [13]. The proposed approach substitutes suitable mean statistical values to the parameters included in these formulas. First of all, a unique value of the daily solar irradiation for the heating season $\overline{I}_s$ has been calculated: we computed the mean irradiation weighted on the days of the heating season for the eight main orientations for Torino, from data reported in [14]. Then we found their

linear correlation versus the orientation coefficients proposed by the Italian standard [10]. $\bar{I}_s$ corresponds to the calculated f_w coefficient: $\bar{I}_s$ = 6370 kJ/m^2.

Standard [13] prescribes a complex procedure to determine the properties of windows that affect solar gains. These properties are typical of the specific window of the specific building. On the contrary, the present model do not require all these information, though very difficult to determine, because it provides a constant coefficient C_w, that accounts for them. It is the mean value calculated on the available sample of buildings, of the product of the solar transmittance with the coefficients of curtains and frame, weighted on the corresponding glazed surfaces. The shading coefficient F_s is not included in C_w because the analysis demonstrated that it strongly depends on the single analysed case; however, the contribution of shading is accounted for in a simplified way, with the coefficient C_{sh}. It depends on the building and is another input data; it is defined as follows:

$$\begin{cases} F_s < 0.7 \rightarrow C_{sh} = 0.5 \\ 0.7 \le F_s < 0.85 \rightarrow C_{sh} = 0.75 \\ F_s \ge 0.85 \rightarrow C_{sh} = 0.95 \end{cases} \qquad (10)$$

The three categories are: buildings very shaded (C_{sh} = 0.5), with an average shading (C_{sh} = 0.75) and with poor or no shading (C_{sh} = 0.95). Finally:

$$E_{s,stat} = C_{opt} \cdot C_w \cdot C_{sh} \cdot \bar{I}_s \cdot S_{w,tot} \cdot N \qquad (11)$$

C_{opt} optimise the correlation between (11) and the corresponding Standard formulas in [13]. Now, we have the elements to express the ratio between the energy gains and losses:

$$\left.\frac{E_g}{E_l}\right|_{stat} = \left.\frac{E_s + E_i}{E_d + E_v}\right|_{stat} \qquad (12)$$

The energy gains utilization factor (η_u) and the mean global seasonal efficiency of the system (η_g) are calculated as the mean value over the available set of buildings.

RESULTS AND DISCUSSION

The goal of our study is to provide a useful tool for the assessment of the energy performance of new and, above all, existing buildings, easy and fast to use, which produces results in agreement with the Standards. The manipulation of classical analytical models by means of a statistical analysis has two advantages: it maintains the physical meaning of the classical models (increasing accuracy) and, being the new formulas simpler, it leads to rapid results. Table 1 reports the mean statistical orientation coefficients and the thermal bridge coefficients for vertical opaque and glazed surfaces, obtained from the analyzed sample of buildings. These coefficients are included in the proposed formula for C_d:

$$\begin{aligned} C_{d,stat} = {} & m_1 \cdot \left(1.117 \cdot U_{op} \cdot \frac{S_{op}}{V} + 0.65 \cdot U_{bs} \cdot \frac{S_{bs}}{V} + U_{pl} \cdot \frac{S_{pl}}{V} + 0.84 \cdot U_{at} \cdot \frac{S_{at}}{V} + U_{pr} \cdot \frac{S_{pr}}{V} \right) + \\ & + 1.2 \cdot \sum_{i=1}^{2} \left(1.115 \cdot U_{w,i} \cdot \frac{S_{w,i}}{V} \right) \end{aligned} \quad , \qquad (13)$$

where the subscripts are: *bs* for the basement floor, *pl* for the concrete slab on pilotis, *at* for the attic and *pr* for the plane roof.

Table 1. Coefficients used in the formula of C_d (13)

Coefficient	Units	Statistical mean value	Standard deviation
f_{op}	-	1.117	8.15e-3
f_w	-	1.115	18.5e-3
m_1	-	See table 1	-
m_2	-	1.2	-

Formula (13) is based on the following hypotheses. T_a = 20°C by default, while T_e = -8°C is specific for the district of Torino. The statistical values of the temperatures obtained from the available sample of buildings are: T_{bs} = 1.8°C (std.dev. 3°C) for the basement and T_{at} = -3.5°C (std.dev. 1.5°C) for the attic.
In (13), coefficients 0.65 and 0.84, that multiply the terms related to heat losses through basement and through the attic respectively, are affected by these assumptions about temperatures. However, in our sample of buildings the mean value of heat losses through the basement and the attic are, respectively, 8% (standard deviation 3.9%) and 10% (standard deviation 4.5%). Therefore, it is possible to use (13) for regions with climate different to Torino, with a modest error.
Compared with (1) and (2), equation (13) calculates the heat loss coefficient with less input data: only the transmittances of the most important envelope components and the geometric dimensions (surfaces and gross volume) of the building. The last data can be easily measured. Formula (13) is compared on the available sample of buildings with the methodology set by the Italian Standard, and the correlation is very good: $R^2 > 0.99$ and the proportionality coefficient is close to 1. The good correlation is because (13) is very similar to the Standard procedure; our purpose was to calculate C_d in a simpler way without giving up accuracy, because it is a very important quantity for *FEN*, as showed in (4).
Another important parameter is the ratio between energy gains and losses (12). The comparison with the same quantity obtained from the procedure prescribed in Standards [9], [13] shows a correlation index $R^2 > 0.85$ and a proportionality nearly unitary.
The proposed model strongly simplifies the solar gains; nevertheless, the comparison between their calculation according to the Standard procedure [13] and the proposed formula (11) shows a good correlation $R^2 = 0.92$, with a coefficient of proportionality of 0.976 (C_{opt}).
Efficiency η_g has been calculated as the mean value on the available data set. Unfortunately, equipment data were available only for a little number of buildings; for them, calculation according to Standard has given the mean value η_g = 0.804. Lacking specific data for every dwelling, this value has been extended to the whole sample, in coherence with their destination (all residential buildings).
All the parameters determined with the statistical approach and related to *FEN*, are collected in table 2. It also includes their standard deviation from the mean value.
The *FEN* proposed by our statistical study (FEN_{stat}), expressed in kJ/(m^{3}DD), results by introducing the foregoing terms in equation (4):

$$FEN_{stat} = \left(C_{d,stat} + C_{v,stat} - 0.968 \cdot \left(2.329 \cdot C_{sh} \frac{S_{w,tot}}{V} + 0.291 \cdot \frac{S_{rf}}{V} \frac{n_{apt}}{n_{apt,rf}} \right) \right) \cdot \frac{86.4}{0.804} \quad , \qquad (14)$$

$C_{d,stat}$ and C_{sh} are calculated according to (13) and (10) respectively. All the other quantities in (14) are only geometric, easy to determine.

Table 2. Coefficients determined from data of the available sample of buildings; they are used to calculate the proposed statistical *FEN*

Coefficient	Units	Statistical mean value	Standard deviation
$Q_{i,stat}$	W/m^2	4.6	-
C_{st}	-	0.914	45e-3
$\bar{I}_s$	kJ/m^2	6370	-
C_w	-	0.467	37e-3
C_{opt}	-	0.976	-
η_u	-	0.968	11e-3
η_g	-	0.804	43e-3

One limitation of the present model is that some of its coefficients have been calculated, with the statistical approach, on climatic data of the district of Torino. These data are: the temperatures related to the basement and to the attic in (13), the mean seasonal external temperature $\bar{T}_e$ and the mean global solar irradiance $\bar{I}_s$. However, the more the climate is similar to Torino, the smaller is the error that affects the correlations.
The results of equation (14) are reproduced in figure 1, having statistical *FEN* in abscissa and *FEN* according to Standard [13] in ordinate.

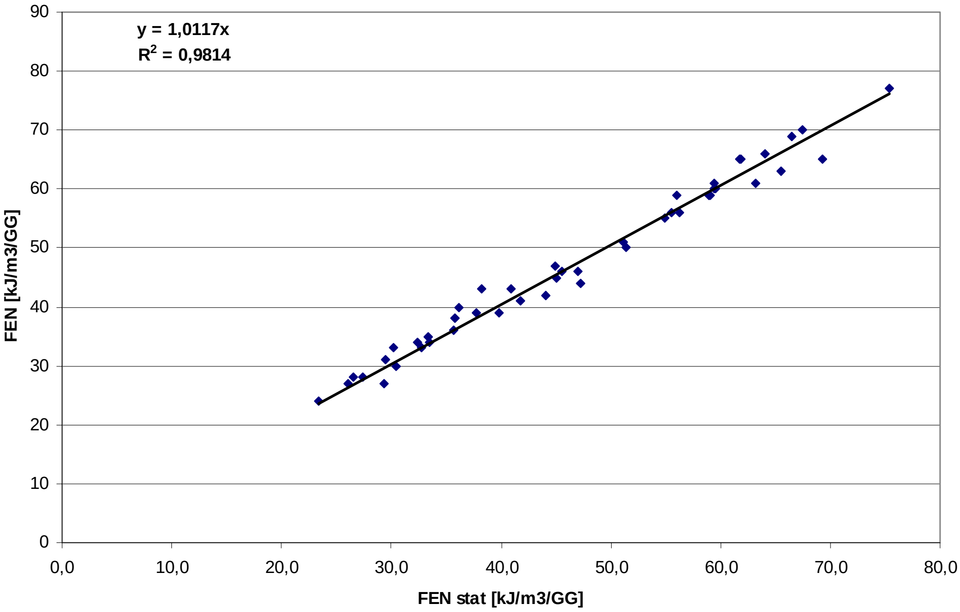

Figure 1. *FEN* calculated with Standard EN 832 versus *FEN* calculated with the proposed statistical formula (14); points refer to the available sample of buildings

The purpose of this graph is primarily to validate the proposed model. The results are in very good agreement, showing a proportionality coefficient close to 1 (1.012) and a correlation index $R^2 > 0.98$.
The most important result of our model is that it estimates in a direct, easy and quickly way the conventional energy demand parameter, which otherwise would require an involved procedure, if calculated according to the Standards approach.

The only required input data are the thermal transmittances of the most important components of the envelope, and some geometric dimension of the building. The last ones are easy to determine.
The model has been calibrated for the typical climate of the district of Torino; for buildings built in locations with analogous climatic conditions (external temperature, solar irradiance and DD), correlations (13) and (14) are in very good agreement with Standard procedure [9], [10], [12], [13].
Therefore, this model is particularly useful to estimate energy demand for space heating for existing residential buildings and it may represent a useful tool for their energy certification.

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All Year heating and cooling load analysis for small hotel buildings in Guiyang City China

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SUMMARY

Guiyang is the capital city of Guizhou province in southwest China, located in the Mild Climatic zone in China. With the economy booming in recent years, the energy consumption in heating and cooling buildings in Guiyang continues to increase. This paper presents a computer simulation studies using DeST software package, which is broadly used to simulate building energy consumption in China. The all-year round simulation for cooling and heating load for a small hotel building has been carried out with various parameters, and the results will be useful for the guidance of energy efficient building design in Guiyang.

INTRODUCTION

Guiyang is the capital city of Guizhou province in southwest China, located in the Mild Climatic zone in China. It's weather is featured comfort in summer and cool and humidity in winter. With the economy booming in recent years, the energy consumption in heating and cooling buildings in Guiyang continues to increase. As a result, the pollution from the energy consumption has been becoming more and more serious. In fact, Guizhou province is part of the districts where sour rain and SO_2 is limited by Chinese central government. The demands of coal in Guizhou province in 1990 is 27.14 million tonnes, and it is 45.91 million tonnes in 2000, According to the traditional forecast method, it will be 82.33 million tonnes in 2010. Furthermore, the SO_2 emission in 2000 is 2.18 million tonnes, it will be 39.1 million tonnes in 2010.Therefore, the analysis of the all year cooling and heating load in building will be useful for the guidance of energy efficient building design in Guiyang.

RESEARCH METHODS

In order to research the impact of thermodynamic parameters of envelop structure on all-year cooling and heating load of building, We present a computer simulation studies using DeST software package, which is broadly used to simulate building energy consumption in China. The all-year round simulation for cooling and heating load for a small hotel building has been carried out with various parameters. The model building consists of three floors, the ground floor consists of hall room and meeting room as well as porter room, it is 3.6 meters height. The first floor is for guests , it is 3.0 meters height. The second floor is for games such as playing card and chess, it is 3.0 meters height. The indoor temperature designed is 27℃, relative humidity designed is 50%, the actual thermodynamic parameters of the envelop structure in the building is as followings: the U-value of exterior wall is 1.49 w/(m^2.k), the U-value of roof is 0.812 w/(m^2.k), the U-value of exterior windows is 4.45 w/(m^2.k). And the heating sources in the building consists of lights and people. the planform of model building can be see in Figure 1.

Figure 1.Planform of model building

RESULTS

The impact of U-value of exterior wall on the all-year cooling and heating load

In DeST, we set up different U-value for the exterior wall but keep others stable. The result for the all-year cooling and heating load of the building is in Table 1

Table 1. All-year cooling and heating load on declining U-value of exterior wall

U-value of wall ($w/(m^2.k)$)	All-year heating load (kw.h)	All-year cooling load (kw.h)
2.095	95857	43566
1.519	85897	42567
1.275	81589	41999
1.079	79245	42545
0.976	77265	41819
0.902	76315	42363
0.824	74970	42212
0.706	72421	41543

0.591	70994	41863
0.532	70344	42202

The result from the simulation study shows: with the coefficient of heat transfer of exterior wall (U-value) decreasing each 0.1 w/(m^2.k), the all-year heating load in the building drops 1.98% in average, but the all-year cooling load almost holds the line.

The impact of U-value of window on the all-year cooling and heating load

In DeST， we set up the U-value of exterior wall as 1.0 w/(m^2.k) and keep other parameters stable, but change U-value for exterior windows, the result for the all-year cooling and heating load of the building is as Table 2.

Table 2. All-year cooling and heating load on declining U-value of glaze

U-value of wall (w/(m^2.k))	All-year heating load (kw.h)	All-year cooling load (kw.h)
5.8	79245	42545
4.5	77405	39967
3.2	72048	40068
2.5	69496	40137

The result from the Table2 shows: with U-value of exterior window decreasing from 5.8 w/(m^2.k) to 2.5 w/(m^2.k), the all-year heating load in the building drops 12%, but all-year cooling load doesn't change.

The impact of glazing ratio on the all-year cooling and heating load

In DeST， the U-value of exterior wall is set up as 1.0 w/(m^2.k),and the U-value of windows keep 3.2 w/(m^2.k),as well as the 0.812 w/(m^2.k) for roof， but we change the glazing ratio (window area to wall area), the result of the all-year cooling and heating load of the building is as following Table 3 and Table 4 .

Table 3. All-year heating load on rising glazing ratio（kw.h）

Glazing ratio	South	North	East	West
0.05	71037	71721	71667	73884
0.10	71454	74824	71496	73059
0.15	74957	80166	71352	72387
0.20	78920	85703	71234	71846
0.25	83134	91361	71139	71415
0.30	87514	97111	71065	71074
0.35	92013	102933	71008	70829
0.40	96597	108810	70969	70694
0.45	101253	114733	70946	70610
0.50	105970	120692	70937	70570
0.55	110740	126683	70952	70671
0.60	115556	132698	71014	70813
0.65	120408	138738	71087	71123

0.70	125294	144801	71170	71596

Table 4. All-year cooling load on rising glazing ratio（kw.h）

Glazing ratio	South	North	East	West
0.05	45220	48399	45627	36843
0.10	58175	59225	46385	39361
0.15	71359	69698	47185	42071
0.20	84478	79981	47984	45004
0.25	97547	90171	48799	48138
0.30	110575	100334	49641	51368
0.35	123499	110446	50512	54668
0.40	136363	120504	51432	58046
0.45	149140	130516	52389	61456
0.50	161808	140503	53358	64901
0.55	174412	150470	54346	68312
0.60	186988	160407	55372	71769
0.65	199497	170336	56336	75196
0.70	211976	180274	57330	78574

The result from the simulation shows: with the glazing ratio (window area to wall area) in south wall increasing each 5%, the all-year heating load in the building grows 4.47% and the all-year cooling load consumption grows 12.8% in average for a building implemented standard U-value in Guiyang; With the glazing ratio in north window increasing each 5%, the all-year heating load in the building grows 5.56% and the all-year cooling load grows 10.74% in average; it is interesting that with the changes of the glazing ratios in the east and west walls, the all-year heating load have no significant changes; with the glazing ratio in the east wall area increasing each 5%, the all-year cooling load grows 1.17% in average; the glazing ratio in the west wall increasing each 5%, the all-year cooling load 6.00 % in average.

CONCLUSION

From above case study, we draw the conclusions as the followings:

The coefficients of heat transfer (U-value) of exterior wall and window are the important factors to the all-year building heating load in Guiyang city, but it almost has no significant impact on the all-year cooling load;

The glazing ratios in south and north walls have a significant impact on all-year heating load, but those in east and west walls have less impact;

The glazing ratios in all orientations have significant impact on cooling load with importance degree order of north、south、east and west.

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Empirical Correlations of Solar and Other Weather Parameters for the Capital Zone "Damascus" In Syria

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ABSTRACT

This work presents a mathematical representation of a few chosen weather parameters of the capital zone "Damascus" in Syria.

Seasonal models, as an alternative to the use of hourly historical weather data, were suggested and used to generate synthetic weather data for the following parameters:

- relative humidity;
- atmospheric pressure;
- global solar radiation intensity.

Such mathematical models were derived for the heating season (November to April) and for the air-conditioning season (June to September) in the zone involved. Moreover, daily sunshine-hours rate was also represented by single model throughout a year. These models were developed based on long time records of data.

The choice of the best model was based on three statistical parameters: the coefficient of determination R^2, the mean relative error e_m and the t-test.

The predicted values along with the recorded data were plotted on the same scale in order to consider a visual comparison between them. It is visually noticeable that good agreement between the predicted and observed data values was obtained.

Keywords: Building design; Thermal analysis; Mathematical models; Weather parameters

INTRODUCTION

Since the greatest share of the energy consumption is used in building [1], it was worth to carry out research in this field. Therefore, the research was mainly directed towards the development of mathematical models for thermal system calculations and simulations.

The proliferation of high power personal computers, have helped in the advancement of the science of system calculation and simulation. Currently, there exist many computer programs for the energy analysis of buildings, [2]. A list of these programs is beyond the scope of this article. Most of these programs require hourly meteorological data of the site as part of the input. Therefore, they are provided with an hourly Typical Meteorological Year "TMY" database for the considered locality. Such TMY database contains a sequence of 8760 hourly values of some chosen weather parameters. Since this TMY database occupy a large area on the PC's hard disc, effort to represent it by using a mathematical representation of the weather parameters were made ([4] to [9]). Therefore, a decision was made to identify seasonal models, as an alternative to the use of hourly historical meteorological data and to generate synthetic meteorological data. Consequently, within the framework of this paper, seasonal models were suggested for the following weather parameters:

- relative humidity;
- atmospheric pressure;
- global solar radiation intensity.

Such mathematical models were derived for the heating season (November to April) and for the air-conditioning season (June to September) in Damascus zone. Moreover, a single model throughout a year was derived also to generate daily sunshine-hours rate.

DATA USED

In order to identify mathematical seasonal models for the weather parameters mentioned in the introduction, the hourly data for each of those parameters, are required. Such hourly weather data used in this analysis are extracted from the hourly TMY database for the Damascus zone. The procedure used to organize and develop this TMY database is given in [3]. Development of such a TMY database was based on the available hourly weather data for the Damascus zone, which were measured by the Department of Meteorology.

SUGGESTED SEASONAL MODELS

New mathematical models representing the data of the sunshine-hours rate, relative humidity, atmospheric pressure and global solar radiation intensity are suggested. The procedure used to identify such mathematical model is described in detail in a previous article [10]. Initially, several models were attempted. Finally, one important type of curvilinear response model, namely a particular case of the multi-polynomial regression model, was considered [11 to 16]. Modern software was used to minimize the sum of the squares of the residuals and to perform the above mentioned calculations [17].

Daily sunshine-hours rate N was represented by single model throughout a year. The adequate model for predicting sunshine-hours rate, for an optional date during the year was obtained. This model is expressed in the following relation:

$$N(d) = a_0 + a_1 \text{Sin} [a_2 (d - a_3)] \qquad (1)$$

The explanatory variable for this parameter is d. d denotes the day under consideration and takes the values 1 (1st January) through 365 (31st December) representing the year's days. This model is mainly applicable within the range from 2-14h.

Weather parameters such as relative humidity, atmospheric pressure and global solar radiation intensity, show considerable changes both daily and annually. In the estimation studies, these parameters can be considered to consist of two components.

The relative humidity and atmospheric pressure at any instant during a day of the heating and air-conditioning seasons can be estimated from a single expression:

$$\mathbf{y}(\mathbf{t}_1,\mathbf{t}_2)=\mathbf{a}_0+\mathbf{a}_1\mathbf{t}_1+\mathbf{a}_2\mathbf{t}_1^2+\mathbf{a}_3\mathbf{Sin}\left[\frac{\pi}{12}(\mathbf{t}_2-10)\right] \qquad (2)$$

The suggested models for relative humidity are mainly applicable within the range from 50-100% and from 5-80% for the heating season and air-conditioning season, respectively. Moreover, the suggested models for atmospheric pressure are mainly applicable within the range from 900-1000mbar and from 930-1100mbar for the heating season and air-conditioning season, respectively.

The adequate models for predicting global solar radiation intensity for an optional time during the heating and air-conditioning seasons are obtained. These models are expressed in the following general relation:

$$I_t(t_1,t_2)=I_1(t_1)\,I_2(t_2) \qquad (3)$$

$I_1(t_1)$ and $I_2(t_2)$ are given in the following relations:

$$\mathbf{I}_1(\mathbf{t}_1)=\mathbf{a}_0+\mathbf{a}_1\mathbf{t}_1+\mathbf{a}_2\mathbf{t}_1^2 \qquad (4)$$

$$\mathbf{I}_2(\mathbf{t}_2)=\frac{1}{\pi}+\frac{1}{2}\mathbf{Sin}\left[\frac{\pi}{12}(\mathbf{t}_2-6)\right]-\frac{2}{\pi}\left\{\frac{\mathbf{Cos}\left[\frac{\pi}{6}(\mathbf{t}_2-6)\right]}{3}+\frac{\mathbf{Cos}\left[\frac{\pi}{3}(\mathbf{t}_2-6)\right]}{15}+\frac{\mathbf{Cos}\left[\frac{\pi}{2}(\mathbf{t}_2-6)\right]}{35}\right\} \qquad (5)$$

These models are mainly applicable within the range up to 600 W/m^2 and 1150 W/m^2 for the heating season and air-conditioning season, respectively.

The various estimators above, a_k, were determined by using the least square curve fitting technique. The estimators a_k and the statistical parameters R^2, $\sigma(\phi)$ and e_m of these models are listed in Tables 1 to 4.

Table 1: Estimators and statistical parameters for sunshine-hours rate model

Weather parameter	a_0	a_1	a_2	a_3	R^2	$\sigma(N)$ [h]	e_m
Sunshine-hour rate	8.6411	3.3957	172.142 10^{-4}	98.6487	0.92	0.71	0.073

Table 2: Estimators and statistical parameters for the seasonal models of air relative humidity

Season	a_0	a_1	a_2	a_3	R^2	$\sigma(\phi)$ [%]	e_m
Heating	59.9700	17.1947 10^{-3}	-4.9325 10^{-6}	-18.2980	0.90	5.33	0.071
Air-conditioning	38.9788	07.4552 10^{-3}	-1.5553 10^{-6}	-29.1878	0.95	4.56	0.086

Table 3: Estimators and statistical parameters for the seasonal models of atmospheric pressure

Season	a_0	a_1	a_2	a_3	R^2	$\sigma(p)$ [mbar]	e_m
Heating	0984.641	-27.9543 10^{-3}	7.5039 10^{-6}	19.4222	0.92	5.60	0.005
Air-conditioning	1014.300	21.2620 10^{-3}	-7.7477 10^{-6}	30.0247	0.96	4.48	0.003

Table 4: Estimators and statistical parameters for the seasonal models of global solar radiation intensity

Season	a_0	a_1	a_2	R^2	$\sigma(I_t)$ [W/m^2]	e_m
Heating	495.3268	-9.8596 10^{-2}	4.9532 10^{-5}	0.90	6.93	0.070
Air-conditioning	920.6284	0.0	0.0	0.96	6.16	0.024

The t-test (two sides) criterion was used to check whether the assumed models are adequate to represent the long-term set of weather data and correspond to the TMY. The test was achieved using a confidence level of 95% with the confidence limit $1.96(\sigma/\sqrt{\mathbf{n}})$. It was found that the P-values for sunshine hours rate, relative humidity, atmospheric pressure and global solar radiation intensity are 0.94, 0.96, 0.93 and 0.98 respectively. One can see that, for each of the four meteorological parameters used P-value>0.05.

CONCLUSION

Mathematical seasonal models were suggested for the following weather parameters:

- - Relative humidity,
- - Atmospheric pressure,
- - Global solar radiation intensity.

Such mathematical models were derived for the heating season and for the air-conditioning season in the Damascus zone.

It is proposed to estimate the above mentioned weather parameters at any instant during a day of the heating and air-conditioning seasons in terms of a single model consisting of seasonal and daily harmonics. Moreover, daily sunshine-hours rate was presented by single model throughout a year in the form of a sinusoidal periodic function. They were developed based on long time records of data. The technique used in determining the best-fitting curve is the method of 'least squares'

The choice of the best model was based on three statistical parameters: The coefficient of determination R^2, the mean relative error e_m and the t-test.

Since we deal with large sets of data (4344 or 2928 data points for the heating or air-conditioning season) the suggested models can estimate the above-mentioned parameters with acceptable R^2 and e_m.

The suggested mathematical models can be used in the CLIMA computer program for the dynamic analysis of heat transfer in building [2], as an alternative to the using hourly historical weather data.

Such approach needs to be followed for the remaining zones of the country under consideration.

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On Development of Design Day for Cooling Energy Need Calculations

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1 Introduction

The paper implies that the methods most frequently used for cooling load calculation (simplified dynamic modelling like e.g. admittance procedure) can be also applied when estimating the annual cooling energy use of buildings. Such approach makes good use of well-established and recently improved tools in another area – in energy performance rating of buildings with regard to their sustainability.
The idea is to replace the design day for cooling load calculation by design day for cooling energy need (DEDCEN) while keeping the same calculation procedure. This would enable to introduce the dynamic impact of weather to the calculation of cooling energy use.

2 Methodology

Three different methods were developed and employed in searching for appropriate 12 DEDCENs representing individual months of a year. Two methods were based on statistical analysis of design reference year data. The statistical analyses were developed from TMY2 (M1) and EN ISO 15927-4 (M2) procedure [1] [2].
In the third method (M3) we used known values of the cooling energy need for well-defined building types to select the DEDCENs from all-year data.
Cooling energy need calculated with DEDCENs and admittance procedure should be as similar to known values of the cooling energy need for all of the building types as possible.
The results of the three tested methods were compared to each other and evaluated by the IEA building energy simulation test (BESTEST). Four realistic building configurations (600, 620, 900 and 920) were selected.

3 Results

The results show that the predictions of cooling energy use based on 12 DEDCENs varied from 96 to 108 % of the values obtained from the complete reference year data (Fig.1). The best results were obtained by the method M3, results were within 96 to 103 % of the all-year values. In other words, it was proved that instead of the all-year database representing 365 days (8760 hourly steps) we can use 12 DEDCENs (one for each month of the year) requiring only 288 hourly steps while keeping acceptable accuracy.

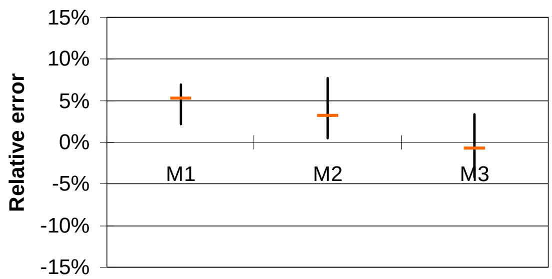

Figure1: Relative errors of cooling energy need calculation using DEDCENs compared to all-year data (set to 100%)

4 Conclusions

The developed methodology has a potential to improve standard procedures for the prediction of cooling energy use of buildings in terms of accuracy and efficiency.

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Indicators for Study of Micro-Climate Impacts on Urban Sustainability

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SUMMARY

Key factors of urban sustainability development are energy efficiency measures that reduce carbon emissions, exhaust heat as well as strategies to mitigate climate change and Micro-Climate (MC). The paper details part of study in identifying a suite of environmental indicators that fulfill this role.

Reviews and comparison of current assessment tools conclude that economic, social and comfort parameters are not emphasized within assessment criteria of most tools for urban sustainable development (USD)(Ian Cooper 1999). Focus of tools seems to be on issues regarding energy, landscape, resources and built environment. The selection criteria for additional environmental assessment indicators are to decrease building energy consumption and alleviate heat island (HI) effects. Influence of climatic variables upon neighborhoods, urban activity and energy load of building envelope performance have an impact on mitigating MC and should be reflected in Urban Environmental Indicators (UEI) for building assessment criteria. Effectiveness of mitigation strategies for MC would lead to further understanding of contribution to energy demands on building envelope performance.

Coupling USD model to building envelope performance would relate consequences resulting from building thermal load to mitigate MC in terms of urban comfort. An investigation into developing methodology towards linking building energy demands and MC modifications is described. UEI for buildings are identified for further research to mitigate MC on USD and the energy demand of building envelope performance.

INTRODUCTION

Improving energy efficiency in buildings and infrastructure is often considered to be one of the most cost-effective measures for cutting down carbon emissions and considerable energy saving potential has been demonstrated in different countries. The purpose of energy conservation, in a narrow sense, is to efficiently maintain a comfortable indoor climate as an environmental unit. However, beyond the limits of this unit are larger external environmental units such as regional communities and urban cities which is an integral part of the global environment. Urban environment provides a good intervention point for building efficiency improvements as they can be coupled with urban renewal measures and provides synergy when linked together. Key factors of USD are energy efficiency measures that reduce environmental impacts such as greenhouse gas emissions (GHG), exhaust heat as well as strategies to mitigate climate change and MC impacts. Much of current research focus is on energy usage in urbanscape because energy consumption is related to production of GHG such as methane, carbon dioxide, ozone, nitrous oxide. Reports by (Intergovernmental Panel on Climate Change 1996) and OECD note that approximately 25 - 30% of total energy-related carbon dioxide (CO2) emissions come from buildings and infrastructure. (Kenneth Button 2002) provides comprehensive discussion of concepts regarding measurement of urban sustainability in terms of energy consumption and environmental impact.

Enhanced urbanization has an impact on MC conditions as a result of energy demands of buildings. Furthermore, atmospheric pollution, combined with pollutants emitted from building materials and anthropogenic activities, influence energy behavior of ventilation and air conditioning in a way that tend to enhance HI. Further to higher sophisticated factors of comfort levels such as thermal load, spatial design, social interaction concerns, there are additional expectations for the energy efficiency of a building for not only reducing environmental impacts but also providing more advanced quality of urban built environment.

Energy efficiency of building is dependant on performance of total building system and requirements for building performance evaluation gave rise to various types of assessment systems for designing energy efficient buildings. Current building assessment systems are predominantly focused on improvement in areas of urban ecological restructuring (UER) by improving urban spatial heterogeneity and energy efficiency of building envelope but lack depth of research in other dimensions such as socio-economic concerns and values of most investors, designers and urban planners. This study reviews current perspectives on market barriers to efficiency investments. Without pragmatic and realist economic assessment together with multiple disciplinary approaches tailored to particular circumstances, efficiency investments would not succeed in market transformation/acceptance. Value case for energy efficient buildings is compelling due to increasing number of case studies (Greg Kats 2003), employing conventional and advanced property valuation methods show that efficiency investment returns, cost and benefit analysis of good quality office buildings varies from 15% less to 11.5% with energy efficient features costing average of additional 2-6%. To further strengthen economic case for energy efficient property by investigations into key selection criteria for financial, environmental and social indicators. The next logical step is to research into design and development of integrated valuation methodology to investigate effectiveness of UER due to MC impacts and further understanding of energy demands on building envelope performance.

AIM

This paper is investigating whether the purpose and focus of attention of indicators and performance criteria for assessing the environmental performance of buildings and urban environment need to be reviewed. It is considering whether building assessment systems should move from addressing the already complex task of assessing environmental protection and resource efficiency to addressing a much more problematic one – the sustainability of the urban built environment. As argued by (Raymond J Cole 1999), the current building assessment framework is deficient since it only assess performance against relative rather than absolute, criteria. As a result, there is insufficient guarantee that buildings that scored highly in the assessment are making a substantive contribution to increased ‘environmental’ sustainability at urban or global scale. This is because current building assessment systems relate to sustainability of buildings and urban built environment as a simple matter of energy and resource flows without due regard to socioeconomic and political dimensions of sustainability(Ian Cooper 1999). It is unlikely the current building assessment systems, while capable of assessing individual aspects of building performance, could cope with the varying interests of wide range of stakeholders present in the built environment. In order to address sustainability in a wholesome, integrated and iterative approach, a range of co-coordinated methods and techniques for undertaking assessments of different components of sustainability should be adopted as the productive way forward(Raymond J Cole 2000).

Currently, this study is attempting to investigate whether it is feasible to reform existing, unified building assessment methods, such as GBC 2000, BREEAM, LEED and CASBEE, in an attempt to make them accommodate assessment of broader, non-environmental components of sustainability(Singh Intachooto and Vimolsiddhi Horayangkura 2007). It is important to note that there are many current different interpretations/methodologies to balance three macroeconomic concerns from industry and academic(Terry Boyd and Phillip Kimmet 2005). Nonetheless, this study represents one such approach deemed feasible for integrating environmental-socio-economic aspects into building assessment method so that the result of proposed integrated methodology would maximize the benefits of energy efficiency in building envelope, property value and the impacts on urban environment by analysis of total building system.

BUILDING PERFORMANCE INDICATORS

Improvement of building performance depends on various elements of technology from the level of material selection to building components and finally to the entire building (Singh Intachooto and Vimolsiddhi Horayangkura 2007). The development of energy-efficient building service systems and the methodology of technology integration into buildings by means of whole building approach need to be studied. This study focuses on commercial office building stock and research into methodology of linking building energy demands with UER from property valuation's point of view to balance conventional economic interests with environmental and social objectives as a framework for encouraging institutional concerns about sustainability by:

1 Assessment and formulation of an integrated valuation methodology for evaluating social, economic and environmental performance in commercial office buildings in urban context.
2 Verification of the valuation methodology in ensuring that social, economic and environmental performance standards are met.
3 Discussion of investigation into developing assessment indicators for property valuation and building envelope efficiency as well as microclimate mitigation strategies in urban context.

Based on the above, the research would include an assessment into the prediction of future short-and long-term scenarios, and a description of Business As Usual (BAU) scenario to be analyzed in relation to cutting GHG emissions from perspective of property investment(Marilyn A. Brown 1998). In addition to addressed concerns, implications of selection criteria for social, economic and environmental indicators to integrated valuation methodology are to be investigated. This is to be conducted by means of data mining and multiple regression analysis with microclimate and energy efficiency simulations. Case studies of conventional and green office buildings from property investment point of view are to be included as part of study. Results would undergo process of verification and review with respect to aim of investigation.

ENERGY EFFICIENCY INDICATORS

Energy efficiency of a building is dependant on the performance of the total building system, and the energy performance of the building system. Buildings consume more than 40% of final energy consumption within the EU and contribute to carbon dioxide emissions that cause global warming. In order to reduce GHG emissions, energy efficiency issues are addressed by reduction in energy consumption of buildings and enhancement of energy performance. Therefore, initiatives that reduce total energy consumption of buildings can be identified as one of the many important leverage points in the effort to reduce additional emissions of carbon dioxide emissions into the atmosphere.

The three main features listed by (Alan Meier ;Thomas Olofsson and Roberto Lamberts 2002) covers an adequate definition of an energy efficient building:

1 Containing energy efficient technologies, operating as designed, in reducing energy use
2 Supplying required amenities and features expected for standard use of commercial office building for at least 60 hours per week
3 Operated in a manner to be efficient, which gives evidence of low energy use when compared to other similar commercial office buildings

In addressing the objective and sub-objectives of research scope and focus, this study endeavors to address the following:

1 How to assess building energy efficiency of commercial office buildings in a transparent manner and which aspects can be identified for extraction of property valuation study?
2 How to verify proposed assessment procedure and to apply in study of existing green and ordinary commercial office building stock?
3 What implications of this proposed assessment procedure have for study of property valuation of green office buildings?

Studies from the U.S.A and Australia have shown that energy improvements have the potential to enhance these countries' economic prosperity and at the same time, lower GHG emissions(Australian Government Productivity Commission 2005). In this context, energy efficiency refers to maintaining or increasing the level of useful output or outcome while reducing energy consumption, encompassing both supply and demand side efficiency. Achieving greater energy efficiency is attractive to governments and environmental organizations concerned with climate change as it offers prospect of significant reductions in emissions – at low or negligible costs to energy users(Sam C.M.Hui 2003).

URBAN ENVIRONMENTAL INDICATORS

The enhanced urbanization has an impact on MC conditions as a result of energy demands of buildings. Furthermore, atmospheric pollution, combined with pollutants emitted from building materials and anthropogenic activities, influence energy behavior of ventilation and air conditioning in a way that tend to enhance HI.

Modification of earth's surface through urbanization has an impact on local climate. Urban areas, especially those of horizontal growth, are subject to bizarre weather patterns, collectively known as MC. These MC differ from prevailing weather patterns in neighboring rural areas. The particular MC phenomenon associated with urban areas is known as HI. HI phenomenon can occur at a range of scales; it can manifest around a single building, small canopy or a large portion of the urban environment. The magnitude of climate difference can be quite large depending on weather conditions, urban thermo physical and geometrical characteristics, and anthropogenic moisture and heat sources in the city(Haider Taha 1997). Depending on geographic location and prevailing weather conditions, HI may be beneficial or detrimental to urban dwellers and energy consumers. Generally speaking, HI are unwanted because they contribute to cooling loads, thermal discomfort, and air pollution or reduce heating energy requirements(Yoshika Yamamoto 2006).

In an urbanized environment, there are significant modifiers of natural cycle of heating and cooling(Linda A. George and William G Becker 2003). These include:

1 Human-made structures such as streets and buildings.
2 Urban surfaces tend to heat up faster than natural surfaces, such as vegetation, that retain water.
3 Introduction of anthropogenic heat sources from heating and ventilation systems, industrial processes and internal combustion engines.

To define and to make effective use of urban economic and environmental indicators requires an appreciation of the context in which they are developed. Urban areas, because of their concentration of population and productive capacity, are inevitably major generators of GHG emissions (Kenneth Button 2002). The need for UEI is to provide guidelines as to the damage being done to the urban environment and the links to economic activities. There are a number of key criteria for consideration in the selection of UEI:

1 Indicators should be relatively small in numbers and reflect important urban environmental effects and trends
2 Additional indicators to relate one another in key sectors with multi-dimensional effects
3 Important for indicators to be sensitive to prevailing underlying conditions and offer forecasts of possible occurrence
4 Indicators to be quantifiable
5 Indicators be consistent across urban areas

Urban planning is increasing confronted with climatic questions connected to projected land-use transformations. The growing awareness that HI and global warming are both the result of mass consumption of energy and resources led to adaptation of similar mitigation measures: energy and resource savings in buildings, energy-saving traffic systems and improvements to urban air circulation(Yoshika Yamamoto 2006). Current assessment system is designed only to assess buildings on an individual basis. There is therefore, a need for comprehensive techniques to assess the overall effects of mitigation strategies for entire cities in order to mitigate UHI effects. Although social and economic demands are currently dominating environmental issues, study have shown that planners are aware of the need for optimizing urban planning processes with respect to MC in order to achieve better public acceptance of their projects(D. Scherer 1999).

FINANCIAL INDICATORS

Studies(Green Building Council (Australia) 2006) have shown that households, businesses, investors, governmental agencies have numerous opportunities to invest in cost-effective, energy efficient buildings despite awareness of the environmental concerns and energy independence. The pursuit of energy efficiency would seem to be in the interests of any rational producer or consumer, firm or household yet the market is not adopting such measures even when it seems to be cost effective for them to do so(Jonathan Garo Koomey 1990).

Introducing environmentally responsible and innovative solutions in buildings is difficult. Major groups of barriers in construction projects include financial, technical, and psychological barriers in addition to complacent clients and incompetent professionals and conservative building codes. Other products and services face obstacles that hinder their adaptation, even when consumer economics appear favorable. Conditions hindering cost-effective investments in energy efficiency and clean energy resources received considerable attention because of widespread environmental, national security and macroeconomic repercussions. Understanding the barriers to this technological adaptation is essential in defining potentially effective investment strategies and governmental incentives(Marilyn A. Brown 2001).

Based on insights gained from investigating building professionals who pursue energy-efficient buildings in Austria, Canada, England, Germany, Scotland, Sweden, and Thailand (Singh Intachooto and Vimolsiddhi Horayangkura 2007), five circumstance which financial barriers would emerged have been identified:

1 Innovative solution affects multiple components requiring higher overall investment
2 Innovative solution may not be worthwhile within a reasonable period of time
3 Innovative technology commands higher investment than commonly used standard solutions
4 Budget lacks financial compensation for devising and developing innovative solutions
5 Mismatched design and construction budget – overspending in certain areas while hoarding expenditure on basics that are necessary for implementing innovative solution

The perception of property as a commodity is changing into emphasis of building characteristics and performance as major determinants of a property's worth and market value, thereby requiring new ways of assessing worth and value (Green Building Council (Australia) 2006). Commercial property valuation represents a major mechanism that could allow environmental and social considerations to be more aligned with economic return. Initial considerations are currently being explored for incorporation of environmental and social issues into valuation theory and practice(Thomas Lutzkendorf and David Lorenz 2005). Possible sustainability key performance indicators are identified and principles for assessing performance along the life cycle of buildings are being investigated(Terry Boyd and Phillip Kimmet 2005). The simultaneous consideration of economic, environmental and social issues can provide a more robust assessment approach and lead to greater reliability of assessment results(Rachel Castillo & Ngai Chi Chung 2004). This would create a more robust assessment approach and lead to greater reliability of valuation results. In addition, opportunities and implications afforded by synergies between sustainable design and risk management are identified for property risk assessment in investing and insurance functions are included as part of this study(Philip Kimmet and Per-Arne Wikstrom 2005).

DISCUSSION

The study aims for improvements in air quality, reduction in health risks and benefits for occupant productivity whilst reducing MC impacts through urban ecological restructuring and environmental efficiency(A.M. Papadopoulos and N. Moussiopoulos 2004).

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C05
Building automation and information systems

Web Service based integration of building information systems

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SUMMARY

The development of new information technologies and standards like Internet and WWW, building control networks, data communication protocols, wireless sensors, mesh networks, product data technologies etc. enables the development of the intelligent, real-time buildings and related new business models in the marketplace of real estate information systems. On of the most promising integration technology in this context is Web Services technologies. The paper first introduces the starting points as well as needs and future challenges for the development. Basic generic ideas of Web Services technologies are then introduced. Main consideration is thereafter focused to the introduction of standard based building automation Web Services approaches: BACnet Web Services and Open Building Information Exchange (oBIX). The basic ideas and concepts of these two methods will be introduced and discussed.

INTRODUCTION

The operation and maintenance of buildings have become dependent on various building automation and information systems (HVAC control, electricity and lighting control, burglar alarm, access control, CCTV), technical building management systems (energy management, FM, BIM) and financial - administrative systems (accounting, personnel administration, renting, space management). In addition, integration of building systems with enterprise systems is becoming more and more important.

These existing building information systems are typically fragmented and isolated from each other forming “automation islands”, which do not communicate together. A consequence of this can be the different systems contain overlapping information, which creates a risk for data integrity. Also because the lacking communication between different systems, it is common nowadays that the information needed by the integrative applications must be collected up manually from different systems and reports.

One of the most important future challenges is the interfacing and integration of the existing applications and to enable the sharing of the information already existing in different systems. New information technologies and standards like Internet and Web, building control networks, data communication protocols, wireless sensors and mesh networks, product data technologies etc. enable the development of the integrated, intelligent real-time buildings and new ICT based building services. Regarding the systems integration, Web Services technologies have become the most promising integration technology.

Standard based open system architectures including object models, interface and service definitions are needed to enable the subsystem and component development. The most important standards official and industry standards will be discussed.

WEB SERVICES AS A GENERIC INTEGRATION TECHNOLOGY

In recent years Internet has become a basic element of data communication network infrastructure. A big challenge still is the integration the systems to enable automatic data transfer between various applications without the need of manual intervention. One solution is the programmable Internet, which fundamental building block is Web Services.

The idea of Web Services (http://www.w3.org/2002/ws/) is to connect applications (and machines) by the same way as e-mail connects people and Web connects people to information. Web Services are platform independent, specifications are based on existing standards and the easy programming of Web Services may affect for application-to-application (machine-to-machine, B2B) communication in the same way as the easy programming of html affects into the breakthrough of the www.

If the objective is performance or reliability, then there are better technologies than Web Services. If the objective is to integrate different loosely coupled components and services, which are decentralized over Internet, then Web Services is the right technology.

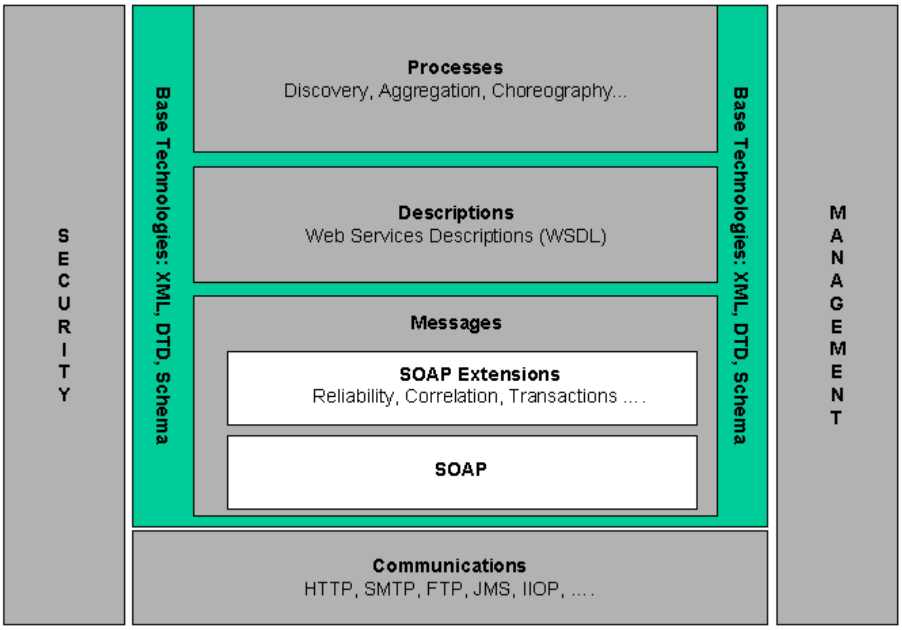

Figure 1 Web services architecture stack

The Web Services architecture is built on the Internet and the XML family of technologies. Main technologies are SOAP (communications protocol), WSDL (service describtion), UDDI (service discovery) and XML technologies (representation of documents) [5]. These Web Services technologies are language, platform, operating system and hardware independent technologies and they provide a standard means of interoperating between different software applications and decentralizing loosely coupled components and services into the Internet. The main Web Services architecture stacks are illustrated in Figure 1.

Unlike many other approaches to integration, Web Services connect devices simply by agreeing on message format, rather than forcing disparate devices to run identical software. Additional functionality for building secure, reliable, transacted Web Services—are possible through a set of recently published specifications (SOAP messaging formats) known as the Advanced Web Services architecture. The ubiquity of HTTP and the simplicity of SOAP make them an ideal basis for implementing Web Services that can be called from almost any environment. Web Services runtime communication is shown in Figure 2.

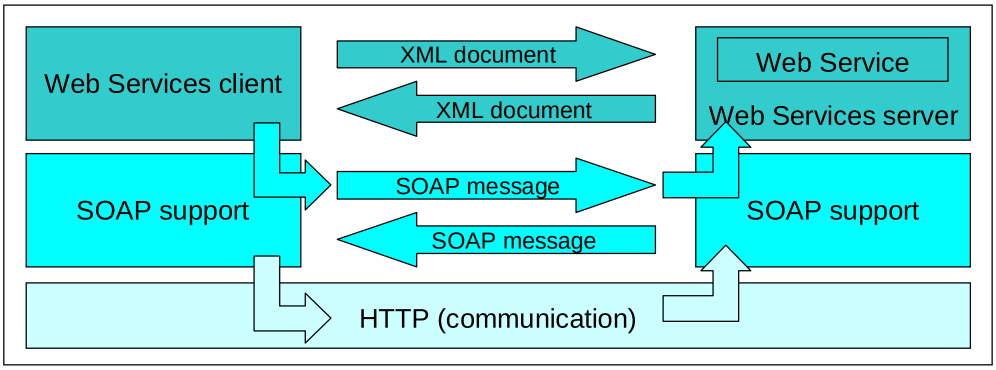

Figure 2. Web Services runtime communication

WEB SERVICES AS A INTEGRATION TECHNOLOGY OF BUILDING AUTOMATION AND INFORMATION SYSTEMS

Creation of the new Web Services Interfaces into building automation systems has been motivated by recognition that Web Services will be emerging as the predominant technology for the integration of a wide variety of enterprise information systems. The objective has been therefore to define standard means of using Web Services for integration and exchange of facility data from disparate data sources, including building control networks, with a variety of business enterprise applications.

The standardization efforts for the application of XML/Web Services to Building Automation started under the umbrella of CABA (http://www.caba.org) in April 2003, when the first meeting of the CABA XML Guideline Committee was held. In that time the different interest groups like BACnet, LON, Modbus, etc. societies worked together. At the end of 2003 it became clear that rather than a lightweight guideline an official or industrial standard was needed. Also it became clear CABA was not the right umbrella organization to bring the standard into the practice.

At that time the name oBIX (Open Building Information Xchange) was given to CABA’s XML/Web Services Guideline committee and discussion started about which standardization organization oBIX should be joined. There were finally two main alternatives: OASIS and ASHRAE. LonWorks and more business oriented members were targeting to IT like standard and were supporting OASIS. BACnet and HVAC oriented members wanted that the new standard should be HVAC and building domain originated and they were promoting ASHRAE, hosting also BACnet standardization.

The discussions took about six months without reaching consensus. The opinions were uncompromising and the consequence was that the CABA’s oBIX committee joined to OASIS but without BACnet society. The BACnet opposition started their own XML/WS standardization process with a new work item in ASHRAE SSPC 135 (Standing Standard Project Committee), in its XML working group (XML-WG).

BACnet/WS standardization progressed in the beginning quite quickly and the draft BACnet/WS standard was published for the first public review in October 2004. The second draft of the standard was published for the public review in March 2006. The final version of BACnet/WS standard [1] was approved by the ASHRAE Standards Committee and by the

ASHRAE Board of Directors in September 2006; and by the American National Standards Institute (ANSI) in October 2006.

The standardization work within OASIS oBIX committee did not start as fast as within BACnet/WS committee. However, several draft versions of the standard were published in Web. The first public review version was published in July 2006. After resolving the review comments OBIX TC approved the first version of the standard in December 2006 [4]. The objective is to get the standard approved by OASIS in early 2007.

BUILDING AUTOMATION WEB SERVICE TECHNOLOGIES

BACnet Web Services

The BACnet/WS standard *135-2004c-1 Adding BACnet/WS Web Services Interface* [1], addendum c to the BACnet standard is logically in two parts

- *Annex N - BACnet/WS Web Services interface (normative)* to the BACnet standard
- New *Clause H.6 - Using BACnet with the BACnet/WS Web Services Interface* to *Annex H* of the BACnet standard (*Combining BACnet with non-BACnet networks)*

This interface proposed in Annex N is intended to be "protocol neutral" so that the defined Web Services can be used with any underlying protocol including BACnet, LonWorks, Konnex, MODBUS or legacy proprietary protocols.

This has been accomplished by defining an application program interface (API) to read and write the common elements of all building automation and control systems such as values, schedules, trend logs, and alarm information using services such as 'getValue' and 'setValue' that use a simple "path" to define the intended data source. An example of such a path would be: "/ABC HQ/Conference Room A/Space Temperature".

The standard provides powerful mechanisms for "localization" where certain types of data such as time, date and numbers can be formatted according to local custom and language. Text names and descriptions may also be accessed using the local language.

The second part of the addendum, Clause H.6, prescribes the gateway mapping specifically to and from BACnet messages.

The combined effect of the BACnet/WS annexes is to provide a set of generic Web Services that can interfaced to any building automation protocol as well as to describe exactly how this interface would work with underlying BACnet systems.

Implementations of the services shall conform to the Web Services Interoperability Organization (WS-I) Basic Profile, which specifies the use of Simple Object Access Protocol (SOAP), over Hypertext Transfer Protocol (HTTP), and encodes the data for transport using Extensible Markup Language (XML), using the datatypes and canonical encodings defined by the World Wide Web Consortium XML Schema.

<u>Data Model</u>

BACnet/WS interface specification establishes a minimal set of requirements for the structuring and association of data exchanged with a BACnet/WS server.

Any node may have a value. The possible types for a node's value are limited to the primitive datatypes "String", "Integer", "Multistate", "Boolean", "Real", "Date", "Time", "DateTime", and "Duration". Nodes that have a value may also have other attributes related to that value, such as minimum, writable, etc.

An attribute is a single aspect or quality of a node, such as its value or its writability. Every node exposes a collection of attributes. Some attributes are required for all nodes, and some are conditionally required based on the value of other attributes. Some are localizable and may return different values based on an option in a service request.

A path is a character string that is used to identify a node or an attribute of a node. The hierarchy of nodes is reflected in a path as a hierarchy of identifiers arranged as a delimited series, similar to the arrangement of identifiers in URL for the WWW. A path like "/East Wing/AHU #5/Discharge Temp" identifies a node, and a path like "/East Wing/ AHU #5/Discharge Temp:InAlarm" identifies the InAlarm attribute of that node.

To allow for an arbitrary number of logical arrangements of nodes, a single node may logically appear to be in more than one place in the hierarchy through the use of a reference node. Reference nodes may be used to build alternate logical arrangements of nodes since the children of a reference node may differ from that of its referent node.

One of the basic concepts of the data model is Node representing the object of the model. The types of the Node are (the allowable values for the corresponding attribute): Unknown, System, Network, Device, Functional, Organizational, Area, Equipment, Point, Collection, Property and Other.

BACnet/WS includes the following Services: getValue, getValues, getArray, getArraySize, setValue, setValues Service, getHistoryPeriodic, getDefaultLocale, and getSupportedLocales.

Mapping BACnet to BACnet/WS

BACnet/WS interface is intended to be "protocol neutral" and the defined Web Services can be used with any underlying protocol. When applying BACnet/WS standard the underlying protocol has to be mapped to BACnet/WS concept.

Table 1 shows an example of the mapping when the underlying protocol is BACnet. In the table BACnet/WS values and value related attributes have been mapped to BACnet Object prosperities.

Table 1. Value and Value Related Attribute Mappings to BACnet Object Properties

BACnet Object Type	Value	ValueType	Units	Maximum	Minimum	Resolution
Accumulator	Present_Value	"Integer"	Units	Max_Pres_Value		
Analog Input	Present_Value	"Real"	Units	Max_Pres_Value	Min_Pres_Value	Resolution
Analog Output	Present_Value	"Real"	Units	Max_Pres_Value	Min_Pres_Value	Resolution
Analog Value	Present_Value	"Real"	Units			
Averaging	(varies)	"Real"				
Binary {Input	Present_Value	"Boolean"				
Binary Output	Present_Value	"Boolean"				
Binary Value	Present_Value	"Boolean"				
Calendar	Present_Value	"Boolean"				
Command	Present_Value	"Integer"				
Device	System_Status	"Multistate"				
Event Enrollment	Event_State	"Multistate"				
Life Safety Point	Present_Value	"Multistate"				

BACnet Object Type	Value	ValueType	Units	Maximum	Minimum	Resolution
Life Safety Zone	Present_Value	"Multistate"				
Loop	Present_Value	"Real"	Output_Units	Maximum_Output	Minimum_Output	
Multistate Input	Present_Value	"Multistate"				
Multistate Output	Present_Value	"Multistate"				
Multistate Value	Present_Value	"Multistate"				
Pulse Converter	Present_Value	"Real"	Units			
Schedule	Present_Value	(varies)				

Open Building Information Exchange – oBIX
The basic references for oBIX are the oBIX specification[4] and the official oBIX committee Website [5].

The core of the oBIX standard includes discovery, data access, historical trends and alarming services for HVAC, security, energy management and lighting. The services are autonomous and have a loosely coupled relationship to a central discovery service.

The data service allows clients to read and write oBIX set of primitive data types. This data could be real-time control points such as actuators and sensors, or it could be vendor properties. In addition to explicit client request for data values, the data service supports data subscription in two ways client initiated polling and server initiated callbacks.

Historical data is needed for reporting and analysis within a system for optimizing performance, efficiency and user experience. The oBIX historical data service will be used to query trend data that has been collected by the server as time-stamped value samples.

The alarm service will provide a normalized representation of an alarm event record. It will have the ability to query alarms that have occurred on the server, the ability to subscribe for new alarms that are generated and the ability to acknowledge existing alarms.

<u>oBIX architecture</u>

The oBIX architecture is presented in Figure 3. It is based on the following principles

- Object Model: a concise object model used to define all oBIX information.
- XML Syntax: a simple XML syntax for expressing the object model.
- URIs: URIs are used to identify information within the object model.
- REST: a small set of verbs is used to access objects via their URIs and transfer their state via XML.
- Contracts: a template model for expressing new oBIX “types”.
- Extendibility: providing for consistent extendibility using only these concepts.

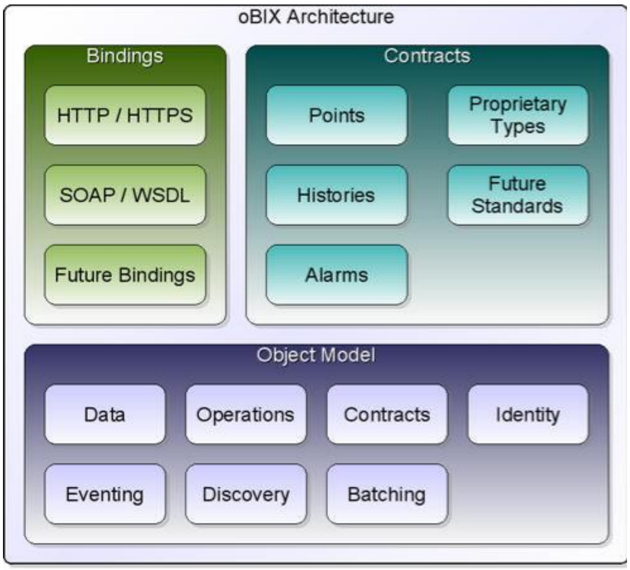

Figure 3. oBIX architecture

oBIX object model

The oBIX specification is based on a small, fixed set of object types. These object types map one to one to an XML element type. The oBIX object model is summarized in the illustration of Figure 4.

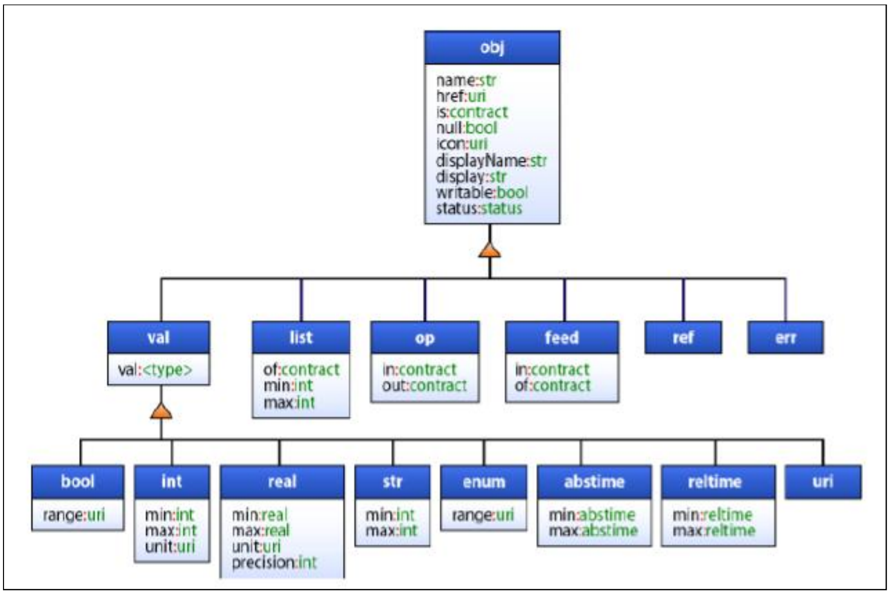

Figure 4. oBIX object mode

The root abstraction in oBIX data model is the *obj* element object. Every XML element in oBIX is a derivative of the *obj* element. Any *obj* element or its derivatives can contain other *obj* elements.

oBIX roadmap

A common view within oBIX community is that oBIX 1.0 is just the starting point towards more comprehensive applications and integration with enterprise systems. There are already preliminary ideas how oBIX 1.0 should be developed further. As an illustration a roadmap towards oBIX 2.0 is presented in Figure 5 [6].

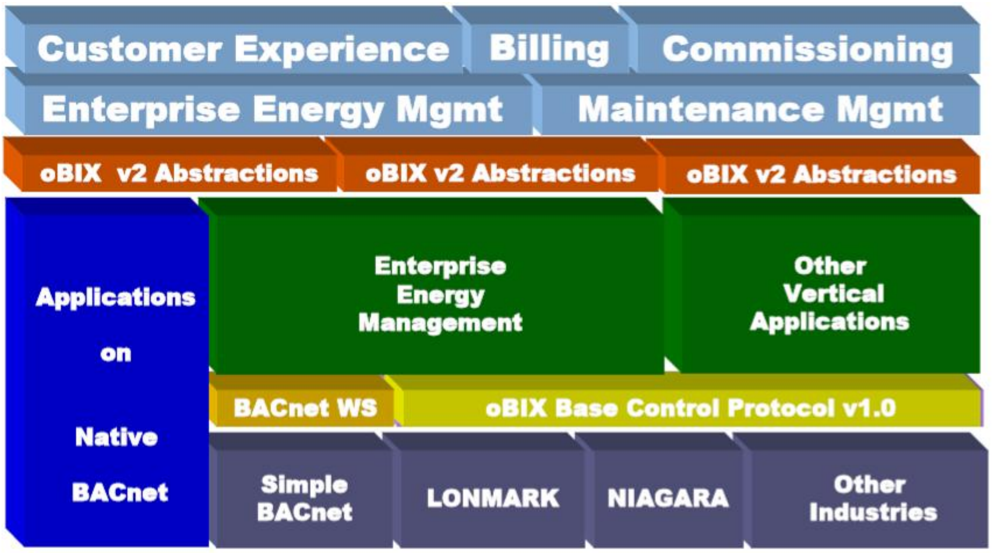

Figure 5. Roadmap to oBIX 2.0

Other Web Services approaches – OPC-UA

In addition to BACnet/WS and oBIX maybe the most important other Web Service approach in building industry is OPC Web Services. Its main application domain is industrial automation, but it is applicable to the building automation as well. However, because it is not building domain specific it won't be handled in more detail here. OPC Web Services are based on the new OPC Universal Architecture, OPC-UA [7], [8].

A big change compared to the earlier versions of OPC that users will notice right away is that OPC-UA is not based on Microsoft COM/DCOM. The whole world, including Microsoft, is embracing open, internet-based communication standards. OPC has done the same by basing UA on TCP/IP, HTTP, SOAP, and XML. Not being tied solely to Microsoft platforms makes OPC-UA highly scalable and applicable to different HW platforms.

DISCUSSION

A major problem in today's building information systems is that the building information systems are fragmented and isolated forming "automation islands", which don't communicate together. The systems can contain overlapping information for their applications because of lack of communication. Integration of the data of different sources to produce reliable real-time management information is very difficult. Systems are typically proprietary without compatibility in core information. These create also dependence from the system vendors.

Web Service technologies described in the paper enable the standard based solution to all these problems. From the viewpoint of technological development having competing approaches has positive influence. Negative is that there will still remains automation islands. In addition, Web Services technology without application area service specification (standards) is not very valuable. It means that it is important to develop new Web Services interfaces into building automation systems.

ACKNOWLEDGEMENT

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Open Web Services-based Indoor Climate Control System

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SUMMARY

The paper describes an open, extensible software and hardware framework supporting all aspects of control and monitoring of a Smart Building. We designed an 'intelligent', real-time control system capable of both autonomous process control and interaction with human operators. The system can easily accommodate both functional extensions and a broad array of sensors and control devices.

The system described below represents a clear bias towards pervasive, open-source, Internet and Web technologies and away from the proprietary or vertically specialized networks, protocols, and application frameworks typical for the current industrial automation systems. The basic goal of this research was to find out if and at what level of effort one can build a functional prototype of a Building Automation System (BAS) assembled exclusively from open-source elements. The answer we found is affirmative and the results of the experiment have the potential to affect the building-automation industry in a way similar to the impact of Linux in the domain of operating systems and of the Asterix server on the VoIP industry.

INTRODUCTION

When first analyzed by a technologist raised in the Internet cocoon, the current BAS's seem to have originated in a different technological civilization. The only commonality of the two technology stacks is the notion of a packet network; from this point on, there does not seem to be one concept or idea that is shared. This technological abyss only gets wider as we move up the protocol stack. This seems to translate into very different business models for software companies involved in both fields. We present a more detailed discussion of this "digital divergence" phenomenon in another paper submitted to this conference [1].

This paper focuses on a practical aspect of subverting the current technological status quo in building automation by creating a nucleus of a disruptive-technology framework, a time-honored mechanism of change and progress in the Internet-centric universe. The goal of this paper is to stimulate the change or, at least, to contribute to the metamorphosis that seems to be germinating in more than one research location. We have no doubt that the current BAS architectures will disappear within a decade, and that building control will completely converge with IP networks. The really interesting question is, "How do we get there?" To gain such an insight we have designed, implemented, integrated, and deployed a complete control system using only open-source or free standard software technologies and protocols, and off-the-shelf electronic components, assembly of which requires only basic technical skills. We hope that the findings described here will enable faster, cheaper, and more efficient implementation of automation systems. The application we particularly cherish is to provide research facilities with a complete, open software testbed for building automation, so that the researchers can focus on solving real problems in fields such as indoor-climate control, rather than having to struggle with cobbling together instrumentation necessary to perform experiments and measurements. We also hope to contribute to the discussion related to emerging standards, security issues, and "open vs. proprietary" software and business models.

METHODS

The overall system design is Internet-centric, following the critical requirement of providing an efficient mechanism to ensure an easier and faster process of bringing building automation systems into the fold of the global digitally converged communications infrastructure. The system uses only pervasive Internet network and open-source subsystems, applications, and application-development tools. These tools rely in turn on a number of standard protocols and/or software development methodologies.

All system functions are implemented as Web Services and all real-time signaling uses pervasive messaging technologies proven effective in mainstream communication systems, such as JMS. Sensors and control devices used with the system are required to be Internet Protocol-based, but the XML-coded device data structures are based on the BACnet networking standard [2].

Although our goal was to build a system with a broad functionality, potentially covering all important aspects of building automation, we have deliberately approached the project as a focused exercise in system design for rather specific functionality. The issue we address is climate control in a space where users can express their preferences for several factors and realize them using personal environmental modules, while the system is responsible for global optimization and resolution of potential conflicts.

Further, the research reported here differs from the work of several industrial consortia on unification of BAS protocols using XML [3-5] and standardize BAS related web services. Our focus is on the system architecture. We believe that the system described in this paper could be readily adapted to provide a reference implementation of protocols being designed in the arduous industrial standard setting process practiced by the ASHRAE BACnet/XML Working Group [3], oBIX [4], and CABA [5].

RESULTS

The requirements for the system can logically be grouped into three categories:

1. *Broad system goals:* a) The system should support interaction of the building automation elements with the human occupants of the space, and b) it should have the ability to make 'smart'[1] decisions in response to ever changing conditions in the building.
2. *Specific system functionality, including*: a) Adjusting environment variables to increase occupants' productivity; b) Tracking space occupants; c) Maintaining knowledge of the physical space including information about zones; d) Monitoring environmental factors in zones; e) Allowing individuals to communicate personal environmental preferences

[1] The notion of 'smart' or 'intelligent' system should be interpreted conservatively. The vocabulary merely implies that the system integrates an inference engine.

(temperature, noise, humidity, etc); f) Providing a visual interface (similar to a weather map) to monitor the space/zones and their parameters; and g) Possessing the ability to self-adjust based on user preferences and output from the inference engine.

3. *Additional considerations*: We aimed at building a solution that is vendor-independent, and uses only open-source tools and components, unless none were available, in which case we resorted to public standards.

BASIC CONCEPTS

We have used the following basic concepts in designing the system:

- *Controlled space* – a physical space where the system controls environmental factors such as temperature and humidity.
- *User presence* – Identity of the user (or users) present in the environmental zone.
- *Environmental zone* – a unit of space controlled by the system, which assumes that all environmental factors have uniform values within the environmental zone.
- *Environmental profile* – a set of key/value pairs that describe environmental parameters of the environmental zone. The system strives to maintain environmental parameters of the zone within a given range from the values specified by the environmental profile.*Environmental preference* – a set of key/value pairs that describe environmental parameters selected by the space occupant as the most desirable. Environmental preference of the user present in the environmental zone affects the environmental profile associated with this zone.*Device* – any hardware or software entity that can generate or accept data. The system receives information about environmental parameters of the environmental zone, and influences these parameters through devices installed in the zone.

TECHNOLOGIES AND STANDARDS

The following technologies have been used to assemble and integrate the system:

- Web Services [6] – communication among system components
- BACNet [2] – base for internal device description
- CLIPS [7] – open-source inference-engine platform
- Java 2 Enterprise Edition [8] – set of APIs for building enterprise applications
- Open Source technology stack:
  - JBoss [9] – J2EE Application Server
  - Struts [10] – Web Application framework
  - Axis [11] – Web Services framework
  - Hibernate [12] – Persistence framework
  - Hypersonic [13] – Database Management System
  - JMS [14] – Java Messaging System
  - AJAX [15] – Web GUI implementation methodology

SYSTEM ARCHITECTURE AND COMPONENTS

Figure 1 below shows all major components of the system. For each component, the communication interfaces and dependencies are depicted. The description of each interface contains information about the technology used for its implementation.

The communication pattern between system modules is based entirely on the Web Services model. This applies to both "read" and "write" operations, as described below. In addition, since devices and other modules can generate events that require handling, we implement two mechanisms to propagate event information: for time-critical events, we use JMS[2] [14]; for event supporting services, such as archiving, we use a pooling mechanism under Web Services.

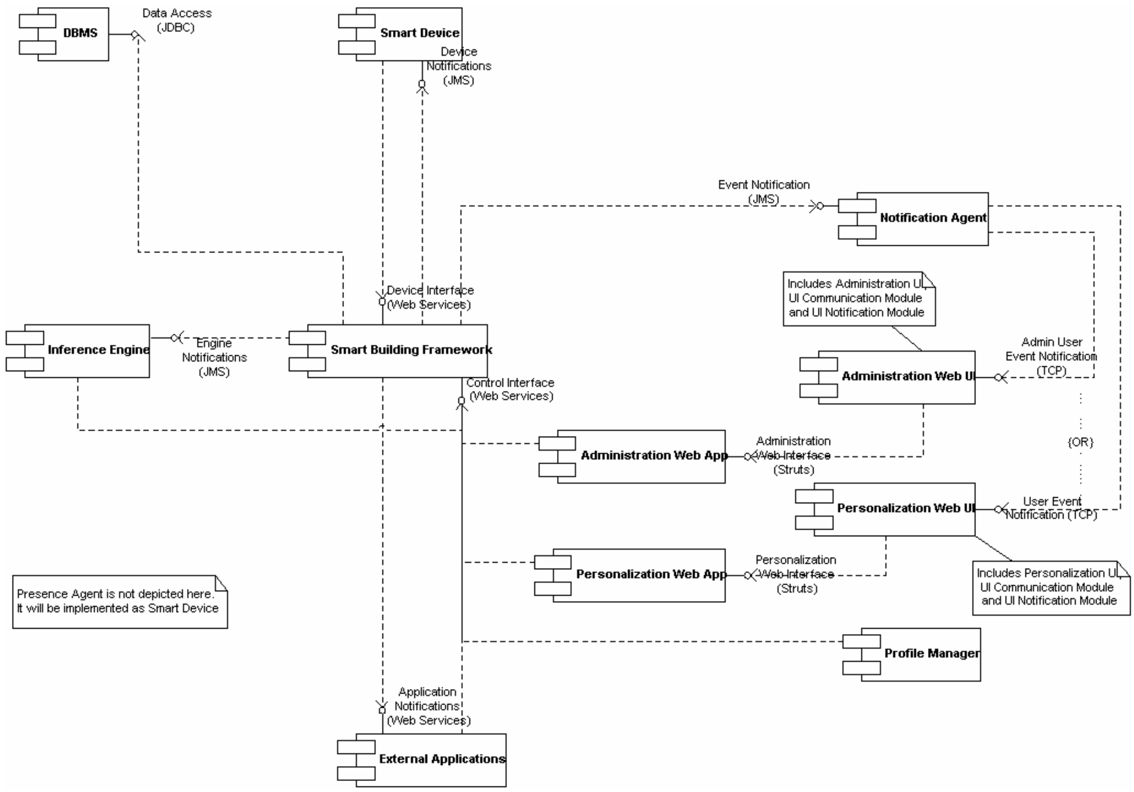

Figure 1. UML Diagram of System Components

Most of the modules on the diagram are self-explanatory. Software modules provide services roughly corresponding to the system requirements listed earlier in this section. Three modules of special interest are a) the Smart Building framework, b) the Inference Engine and its integration with the system, and c) the Smart Device. These three elements will be described below in more detail. The system is extensible, as it can accommodate External Applications of any kind. A good example might be an energy optimization engine, a computational fluid dynamics application, or a set of web services reaching out to external data sources, such as a weather forecast.

Smart Building Framework

The Smart Building Framework is the core of the Smart Building system to which all other system components connect. It implements the following services:

- *Device status monitoring and data collection*
- *Device data archiving and retrieval*
- *Access to and manipulation of an object-oriented representation of the system state*
- *Event-based communication among system entities*
- *System configuration*

[2] JMS engine is an integral part of the JBoss Application Server [9]

Web Service APIs

Most of the services described above are exposed through a set of ***web service APIs***:

1. *Device Web Service:* allows external devices to connect to the system. External devices use this service to register their presence and to report changes in their state.
2. *Object Access Web Service:* provides access to object-oriented representation of the system state maintained by the framework. The Object Access interface can be used to both read and write properties of the objects maintained by the system. Write operations on the system state may translate into commands issued to the components of the system controlled by the Smart Building Framework. The service is also used by system components that need to access the information about the system entities (for example, Inference Engine or Administration Interface).
3. *Event Access Web Service:* implements a messaging system for sending prioritized, point-to-point or system-wide alerts related to the state of the system. For example, the Inference Engine might send an alert to the Administration Interface component, if the readings from the external devices exceed a certain threshold. Both sending and retrieving operations are supported. The service is used by all system components that need to send and/or retrieve alerts.
4. *Archive Web Service:* allows starting and stopping archiving of the data submitted by the external devices connected to the system. It also allows retrieval of the archived information. The Archive Web Service is used by the system entities that rely on the archived information about devices.

The web services implemented by the Smart Building Framework can be accessed not only by the components of the Smart Building system but also by other external (remote) entities that need to access the information about the status of the building infrastructure.

Communication between modules

All components of the Smart Building system can learn about the system-state changes through polling functionality embedded in all implemented web services. This approach is simple and works well with standard web infrastructure. However, it also has significant drawbacks such as embedded latency in the system responses due to a set polling interval. To handle propagation of time-critical events we designed a server-side component implementing JMS-based notifications. Each component interested in receiving notifications from the Smart Building framework can embed JMS client functionality. This also applies to our wrapper for devices and to the Inference Engine, as well as to the Administration and Personalization Interfaces. The JMS messages may carry either actual module commands, or a request for a module to poll the message sender for information.

Smart Building Inference Engine

The inference engine provides the ‘intelligence’ driving the Smart Building Framework. While the devices connected to the framework might be autonomous enough to make their own limited decisions, there exists a range of decisions that are beyond the capabilities of any single device. The inference engine fills that gap by gathering information from the entire framework, making decisions based on the gathered data and then controlling the framework to execute the decisions.
Communication: The inference engine communicates with the rest of the Smart Building Framework via a set of web-service interfaces. These allow it to extract information necessary

for making a decision, and then to control the framework to execute a decision taken. In particular, the framework interfaces allow:

- reading device input
- controlling (writing to) devices)
- reading and modifying current environmental data
- sending system alerts

Since the communication mechanism is documented, it is possible to replace the engine with another agent that would drive the framework.

Engine: The inference engine is implemented as a stand-alone Java application. Its main functionality, the decision making, is split into *decision beans*. The application schedules a bean to execute at a specified time after the application start. From then on, a bean is in charge of its own scheduling. In addition to making decisions, the beans can also be used for simpler tasks, for example to translate the information retrieved from the devices into current condition data.

It remains unspecified whether the entire decision logic should be implemented as one complex bean, or as a several (possibly many) simpler beans. The inference engine supports any approach. Typically, when a bean is executed, it will gather required data, analyze them and send the results (a decision) back to the framework. The bean can then ask the inference engine to be scheduled again at some later time.

Simple decision beans can be implemented as Java objects. In the case of large systems and complex decision beans, the capabilities of the Java language might prove to be insufficient. For these scenarios, we have integrated a rule engine to allow us to use a more sophisticated decision making mechanism. The rule engine we used, CLIPS [7], can improve over Java in the areas of a) processing more advanced condition statements, b) resolving the conflicts among multiple conditions and c) general performance of complex-condition processing. Both Java and CLIPS decision beans can be used simultaneously.

Example: Below we provide a very simple example of a CLIPS script. The reader should not infer that this nearly trivial example represents the norm. There is no limit to the complexity of the rules that can be incorporated into the entire framework, and some of the rules we have implemented are rather complex.

The data retrieved from the framework are supplied to the script as CLIPS facts. The output of the script is also facts that are asserted or retracted based on the input.

```
;;;=========================================================================================
;;; This script establishes user presence in a zone based on a reading from a presence sensor
;;;  Input facts - presence-sensor; Output facts - zone-user
;;;=========================================================================================
;;;  DEFTEMPLATES
(deftemplate presence-sensor
  (slot status)
  (slot user-id)
  (slot activity))
(deftemplate zone-user
  (slot user-id))
;;;  RULES
(defrule current-user ""
  (presence-sensor (status active) (user-id ?u) (activity ?a))
  =>
  (assert (zone-user (user-id ?u)))
  (printout t "user present is: " ?u crlf))
```

Smart Device

The Smart Device component of the system is a Java wrapper implementing a generalized API used to communicate with real devices. This module provides a two-way translation between device proprietary protocols (if any) and Web Services used by the Smart Building Framework and it may implement a JMS client (see *Communication between Modules* section above). A driver for each real device in the system is a specialization of the Smart Device. As mentioned in Methods, we decided to use an XML-coded BACnet [2,3] data model to describe device data structures. This decision is a compromise we accepted for several reasons, some of them discussed in [1]. We recognized the facts that a) BACnet is a public standard under a committee control, b) its device data structures represent decades of domain-specific engineering experience and, as such, it is unlikely we will be able to come up with a better data model, and c) the particular choice of the device data model is not really relevant to the goals of this research[3].

We have implemented several devices to interact with the system, including a Universal Device (as a template for future development of device drivers), a Presence Agent identifying personnel entering a controlled space, and a simulated A/C heater-thermometer device simulating temperature changes according to a simple physical model driven by input from the Inference Engine. The most interesting device though is the so-called PEM controller. PEM (Personal Environment System) is a commercial product designed to provide an individually controlled work environment. The module, installed on the user's desk (Fig. 2) regulates air temperature and air supply to the desktop; controls the radiant heat panel output; and provides background noise masking and task lighting level. In its basic configuration, the module is locally controlled and not networked. It can be connected to a data bus, but the cost of the controller and proprietary software is substantial. For the purpose of the project we implemented an IP-based system controller for the PEM device. The controller is inserted between the manual slider board and the PEM proper. When active, the IP controller overrides manual user preferences, and instead adjusts the environmental parameters to the user preferences stored by the system, subject to optimizations necessary to take into account preferences of users in other cubicles and energy usage constraints in the controlled space. The system's Inference Engine implements the optimization procedures. Manual controls are replaced by a small application running on the user's workstation, which however can be deactivated by a system operator.

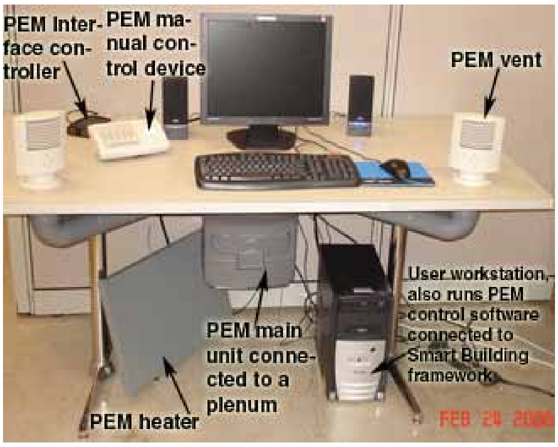

Figure 2: System controlled Personal Environment Module

The PEM controller is activated by user login to the workstation, although it can also be started by a signal when the building security system reports that the user is entering the building security gate. This *per se* is not particularly innovative, but the system offers certain niceties, such as biometrics identifying the person and transferring its environmental

[3] Use of such data structures does not imply that the system is compatible with BACnet network protocol layers, although, since the device data structures are rigorously derived from the BACnet paradigm [2], the system could be relatively easily adapted to access and control most of the existing BACnet BAS infrastructures.

preferences to the proper cubicle, together with the person's VoIP phone number. Further, the global optimization functionality mentioned provides a relatively new and interesting element. The PEM controller implements a complete HTTP server on top of the IP stack, and is simply connected to the office LAN. The application running on the server communicates with controller's A/D and D/A converters, providing full control of PEM control elements, as well as providing additional temperature and humidity sensors plus several auxiliary industrial standard 0–10V inputs and outputs.

DISCUSSION

While the existing implementation and deployment focus on environmental controls and human efficiency issues, we believe that the system described here offers a nontrivial technological advantage over current proprietary industrial solutions, and provides a sustainable foundation for future collaborative development of Smart Building software by academic and industrial consortia and alliances. To facilitate such a process, the system has also been carefully designed to avoid intellectual property infringement of existing patented or otherwise protected frameworks while replacing many aspects of their functionality [1].

The modular design of the system permits replacement of any of is components by a proprietary or simply different implementation. Our focus on open-source software provides an affordable starting configuration. Device drivers, which are a very significant obstacle to interoperability, can be implemented in a matter of days or even hours by extending or replacing software modules provided in templates. The selection of Web Services implemented in the framework covers all typical uses we have identified in BAS systems, and implementation of an operational system of medium complexity provides a strong proof of the concept.

The research presented here is complementary to the work of ASHRAE and OASIS/oBIX; those groups focus on protocols and we have focused on systems research. We believe that our system can provide an almost instant reference implementation of the results of the work of these industrial consortia. This represents an important step in validation of ideas leading to future Building Automation Systems.

ACKNOWLEDGEMENT

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Supporting disaster management by means of ICT

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SUMMARY

In modern big buildings there are many different technical systems typically without any automated interoperation with building Facilities Management systems (FM). In the case of a disaster (fire) the problem is that the rescue personnel (fireman) can't get relevant up-to-date information of these buildings. A new advanced ICT support system has been developed, which reports focused real time information into the moving rescue vehicle in the case of fire using wireless data transfer. The support system combines the location based real time information about fire and building technical systems with static information of FM and reports it to different kind of end users. The system is in test use in several buildings in Finland. Generally the integration of building information systems enables the development of totally new kind of applications. The disaster management support system developed is a good example of the potential of this kind of systems integration.

INTRODUCTION

In shopping centers, hospitals, hotels, big office buildings, industrial areas and public underground spaces there are many different technical systems like building automation system, HVAC, fire alarm, fire protection, smoke control, access control, CCTV, lifts, etc. typically without any automated interoperation with building Facilities Management systems (FM). It's very common that the facility data is updated regularly (during the life cycle) in these kinds of buildings. In the case of a disaster (fire) however, the rescue personnel (fireman) can't get relevant up-to-date information even about these buildings (spreading of the fire, status of building technical systems, locations of dangerous materials, critical spaces for company's businesses etc.).

The aims of the work were to study which information about building and it's technical systems is needed in the case of disaster (fire), how to combine the absolutely correct and up-to-date location based real time information about fire and building technical systems with static information of FM and to report it to different kind of end users, and especially to develop a new prototype to report that information for different purposes like fire master in moving rescue vehicle going to lead the fire fighting and personnel inside the burning building.

METHODS

The first problem was to find out what information from building and it's technical system is needed most in the case of fire incident. The information needed about burning building was studied interviewing eight fire masters in different parts of Finland. Interviews consisted fixed and open questions and one interview took as an average two hours. The interviews were performed in 2003. The fixed questions were related to the information needed from spaces, structures, technical systems, dangerous materials, application usability, manner of representation (graphics etc.) and information needed in different situations (approach, first 10 minutes, after first 10 minutes, for archive). Open question was for example "what information the leader of fire fighters does really need in the case of fire".

The main problem was to find out the modern ICT solution (integration, data transfer and implementation technologies) to combine data from building's real time systems (automatic fire alarm, building automation, HVAC, automatic fire extinction, smoke abatement, motorized fire damper and other fire protection systems, safety and secure, access control, lifts, CCTV etc.), always up-to-date building facility management database information and maintained CAD pictures (layout, floor plans) and send it using wireless data transfer to the moving rescue car. The main approach was to use as much as possible existing building automation standards and standard based open technologies and integrate those existing "lego blocks" using modern Internet software integration technologies called web services (http://www.w3.org/2002/ws/).

The results of the interviews described above were utilised later on when directing the development work of the new prototype for disaster (fire) management support.

RESULTS

The results of the interview survey are presented in Table 1 [1]. The Table contains also information which building systems have to be integrated to get that information.

Table 1. Information need in the case of fire and systems containing a part of that information.

Information	FM	Fire alarming system	BAS	Other technical system
Fire initiation place	x	x		
Spaces under fire	x	x		
Layout picture (always up-to-date) • rescue ways • attach routes • locations of main valves, switches and building technical system centres	x			
Floor plans (always up-to-date) • dangerous spaces and materials • valuable spaces	x			
Localization diagrams	x			
Fire sections, fire doors, fire dampers (open or closed)	x		x	x
Supporting structures, intermediate floor structures	x			
Lifts, shafts, tewels	x			x
Smoke abatement zone	x			
Status information of sprinkler, ventilation and other technical systems (and effect areas)	x		x	(x)
Temperature levels	x		x	
Contact information (security company, service company, energy company, water supply plan, property owner etc.)	x			
Underground spaces	x			

At this stage of the study the architecture of ICT based disaster management system was developed. One of the most important decisions was the selection of web service technology as the system integration technology of the ICT system platform.

The main components and connections of the building disaster managements system are presented in Fig. 1. The main idea is to combine all necessary information of building and it's technical systems for different interest groups like authorities, service providers (security company) and people inside the building (personnel, security manager, control room persons etc.) in the case of disaster.

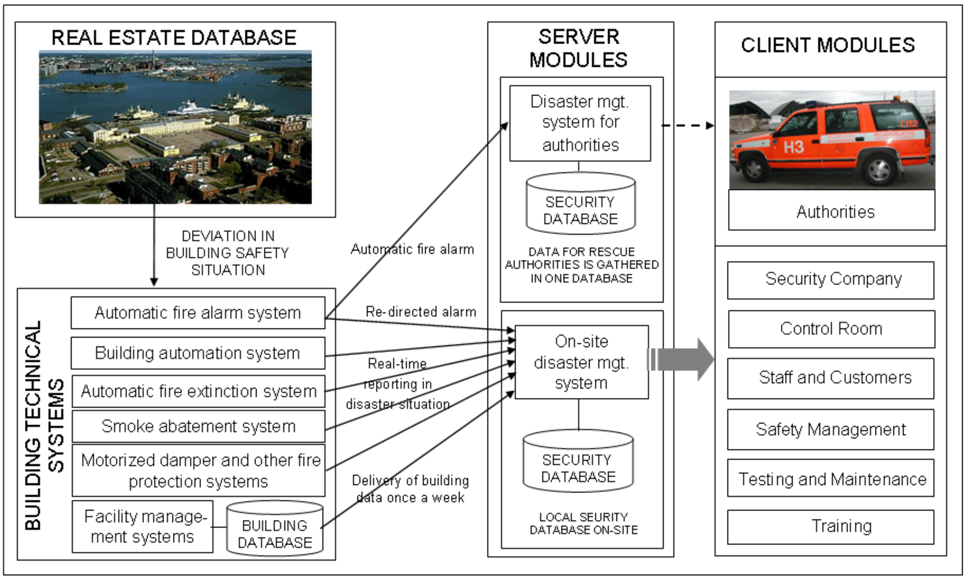

Figure 1. Main components and connection of ICT based building disaster managements system.

The work covered the study of two different versions of the system: authorities (emergency center and rescue center) version and version for people inside the building. The main components and connections of disaster management system for authorities (case fire incident) are presented in Fig. 2 and the data communication for that in Fig. 3.

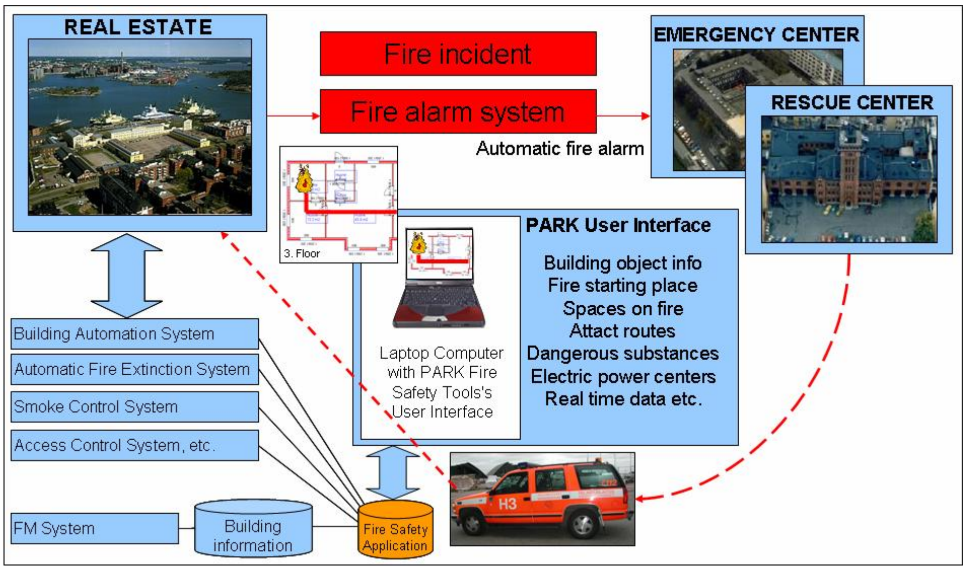

Figure 2. Main components and connection of disaster management system for authorities – case fire incident.

The main communication model for local users is based on the broadband data communication supported by short message service (SMS) based text messages. For the authority version wireless data communication like 3G, EDGE, GPRS and WLAN data transfer was selected.

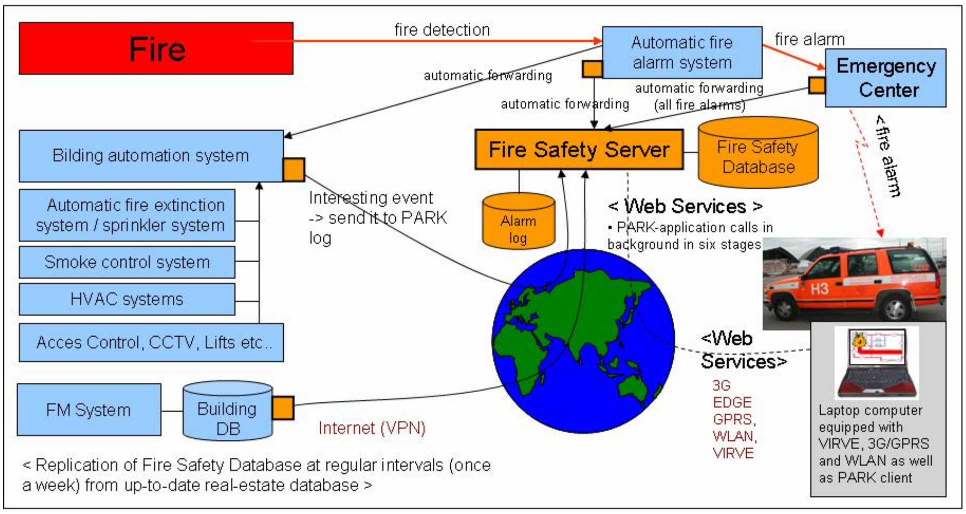

Figure 3. Data communication of disaster management system for authorities – case fire incident.

The main component of disaster management system for people inside the building is presented in Fig. 4.

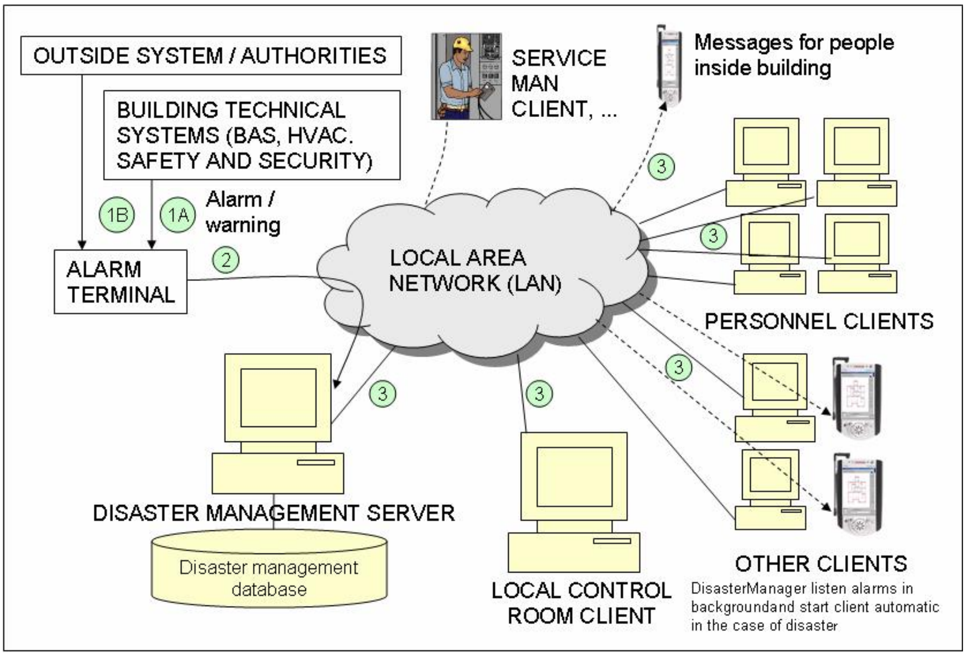

Figure 4. Main component of disaster management system for people inside the building.

The main result of this study was a new prototype for disaster (fire) management.

PROTOTYPE

A new advanced ICT support system has been developed, which reports (displays) focused real time information into the moving rescue vehicle in the case of fire using wireless data transfer. The support system combines the location based real time information about fire and building technical systems like fiery spaces, ventilation system's status, activated loops of the fire-extinguishing system and smoke control system etc. with static information of FM like real estate's layout, floor plans, location and effect areas of systems/equipments, spaces of dangerous substances, valuable spaces, fire sections etc. and reports it to different kind of end users. The context aware system supports different end user groups like fireman, local control room operator and the personnel occupying the building etc.

"Layout" function of the disaster management prototype is presented in Fig. 5. The main idea was to integrate real time information of fire and the facility management information of the location of fire detectors and show it by means of CAD technologies. In this example the part of the building which is on fire is blinking red in the layout picture. Other information included in the prototype are attach routes for fire fighters, rescue ways and the locations of main valves, switches and building technical system centres.

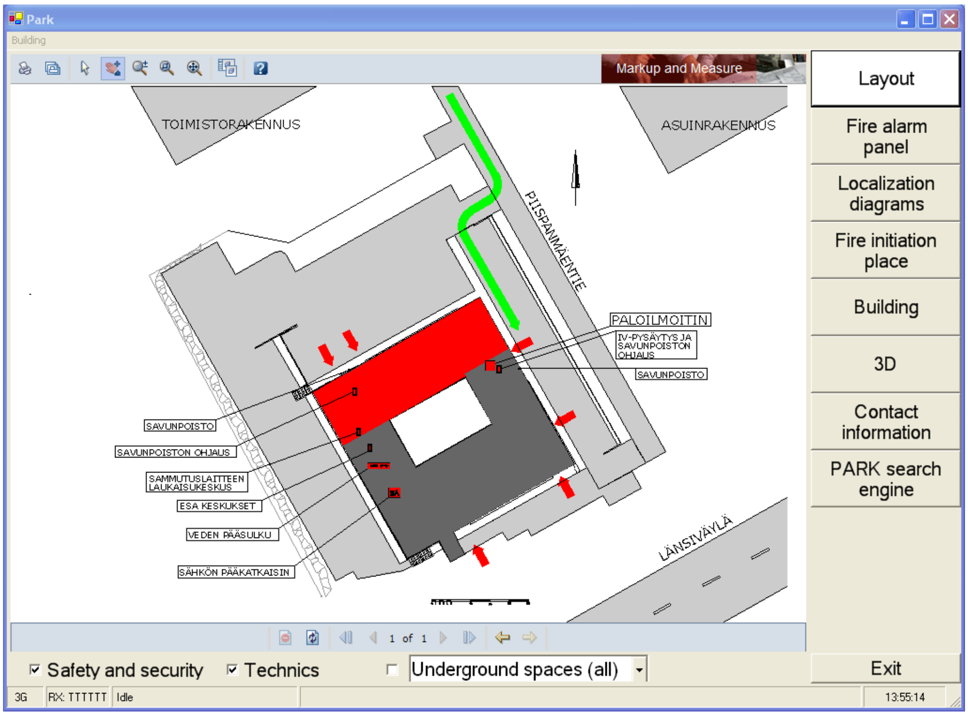

Figure 5. "Layout" function of the disaster management prototype application.

"Fire initiation place" function of the disaster management prototype is presented in Fig. 6. The main idea is to combine the exact place where the fire has started with space information from facility management system and show the result in floor planes. Information included in the prototype are fire sections and how near the fire are to dangerous materials (risks for fire fighters) and valuable spaces for business of the company working in the building. Fire initiation place is coloured red and other selected information is green.

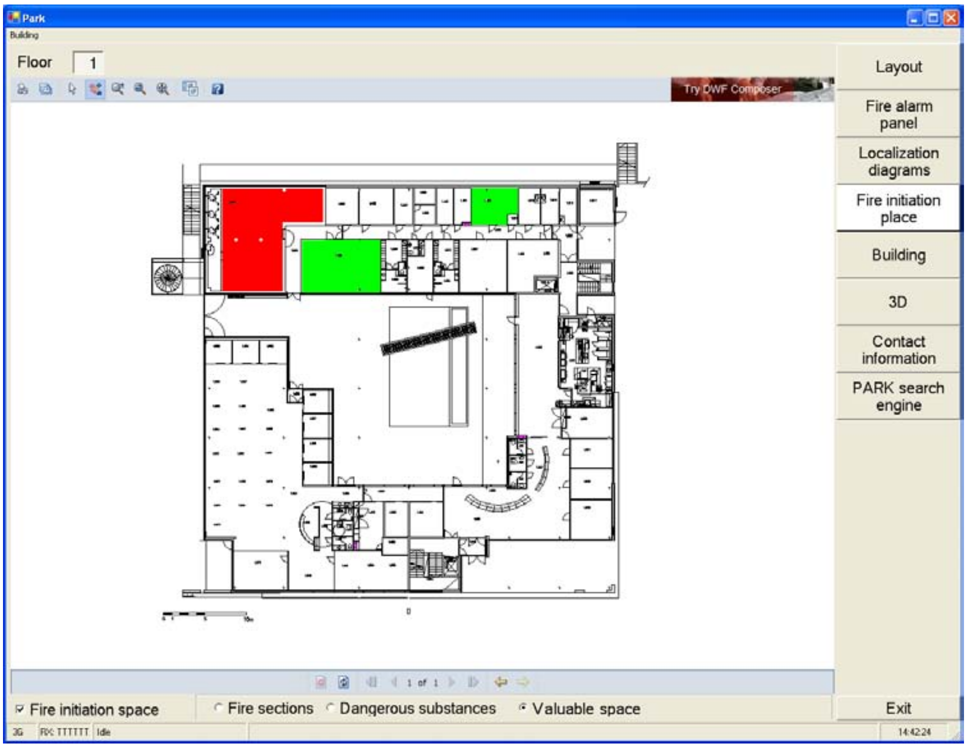

Figure 6. "Fire initiation place" function of the disaster management prototype application.

"Building" function of the disaster management prototype is presented in Fig. 7. This function combines real time information of fire with exact location information and real time status information of building technical system (status information of ventilation, sprinkler, smoke abatement, motorized fire damper etc.) into always up-to-date building floor plans.

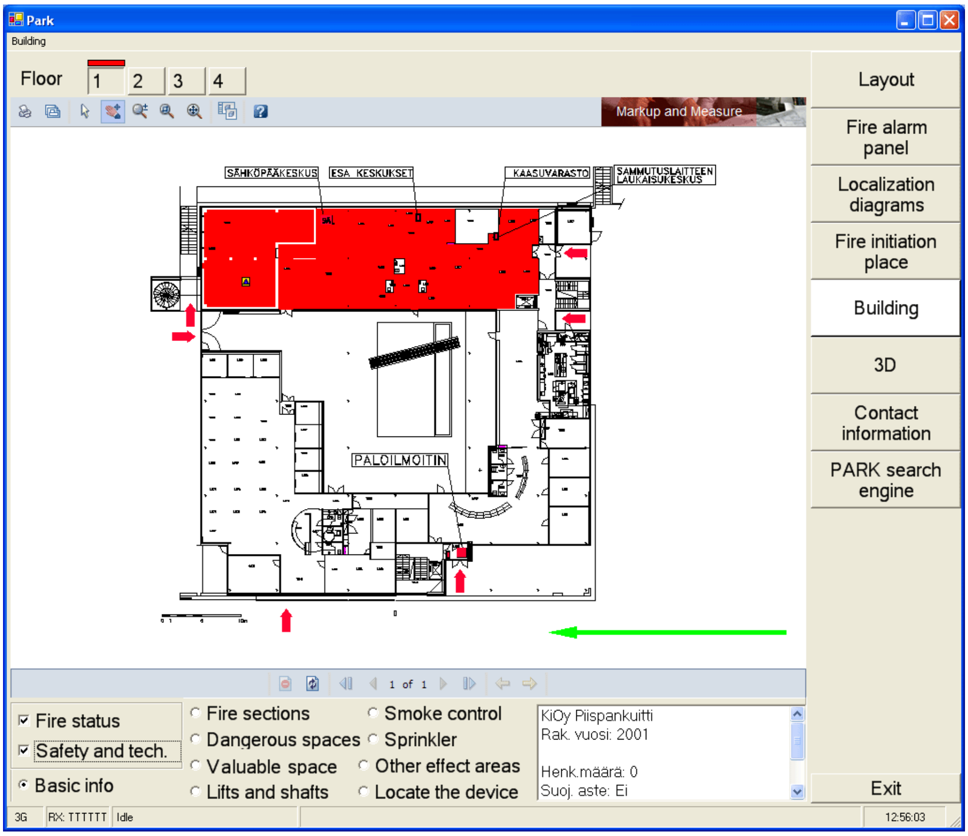

Figure 7. "Building" function of the disaster management prototype application.

“3D” function of the disaster management prototype shows wireframe model of building and combine building real time (fire) and static information (dangerous materials, critical spaces for business etc.) into that 3D model.

The system is presently in test use in several big buildings in Finland.

DISCUSSION

The integration of building information systems enables the development of new kind of applications which utilise any information existing in information space. The disaster management support system developed is a good example of the potential of the systems integration.

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Fire Safety, Building and Construction, Telecom and Other Companies: Siemens Oy, Esmi Oy, Fläkt Woods Oy, Falck Security Oy, Marioff Corporation Oy, HB Sisäilmatutkimus Oy, TeliaSonera Finland Oyj, Buildercom Oy, NCC Building / Property Development, Invisian Oy, Inpecta Oy, Axel Group Logistic Systems Oy, WM-Data, Securitas Systems Oy, State Security Networks (Finland’s Public Authority Network VIRVE).

Insurance Companies: Federation of Finnish Insurance Companies, IF Insurance.

Public Authorities: Helsinki Rescue Department, West-Uusimaa Rescue Department, Emergency Response Centre Administration.

Financing: Tekes.

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Space heating control of an individual dwelling by a fuzzy controller acting on the flow rate of a heating floor

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SUMMARY

A lot of complementary solutions to save energy in buildings already exist. If it is well insulated, a building can produce itself a part of its thermal energy consumption, tanks to renewable energy devices such as solar collector. This clean energy mustn't be spoiled; therefore a good control must be applied to its use. The fuzzy control is one of these solutions. We applied it to the control of the heating floor of an individual dwelling. The design and the validation of the controller were carried out in simulation with the software TRNSYS 16.

INTRODUCTION

The integration of renewable energies in a dwelling is a complex process. Many devices are able to produce clean and cheap energy. It concerns electrical energy, domestic hot water and heating. However, this clean energy must be optimized by the consequent control of the actuators using this energy.

Heating floors are one of the most interesting heating devices. They produce a high comfort level to the user and are supplied by a low temperature heating fluid, which facilitate the integration of renewable energies. Energy savings could be made if we could lower their operation level when it is possible during the inoccupation periods in the dwelling. This would optimize the performances of the heating floor and in the same time lower the use of boilers using fossil combustible or electrical energy from the grid.

Since 2005 a partnership between AMPERE and CETHIL and supported by the Region Rhone Alpes carried out a study on the design of a global controller which could optimize the contribution of renewable energies and control the thermal and electrical comfort. Concerning the thermal comfort, the testing of our temperature control procedures requires a reference control to be compared to and appropriate testing conditions. For this reason, we use simulation to test our controller in different buildings. One of these building is the model of a dwelling designed in the process of the Task 26 lead by the International Energy Agency [1]. This model was designed to test the performances of existing solar combisystems, and was chosen for its quality and its simplicity. Our reference controller is a fuzzy controller, one of the most accurate controllers sold to the public. Fuzzy controllers are usually used for office buildings which are occupied during the day and unoccupied during nights and week end. For our dwelling, occupation and inoccupation periods are inverted, which is more unusual. This study will show the interest of this adaptation.

We tested its performances in this house, which we located in Lyon, France.

DESCRIPTION OF THE MODEL

Description of the building

The building we chose is a single family home. Its surface is 140 m² large and it is located on two levels. It was designed to consume 60 kW.hr/m²/year for space heating in Zurich, Switzerland, which represents good thermal performance [1]. It is modelled in TRNSYS 16 with type 56 and the interface TRNBUILD.

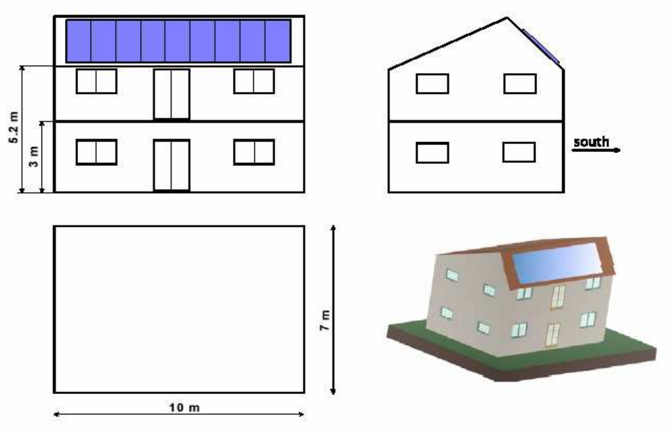

Figure 1: The task 26 dwelling

Windows are double glazing, modelled with type 2001 from the TRNBUILD library.

Table 1: Windows surface

Wall's orientation	North	South	East	West
Windows surface (m²)	3	12	4	4

Table 2: Walls description

Wall	Orientation	Surface [m²]	Conductance [W/m²K]
External Wall	North	50	0.342
External Wall	South	50	0.342
External Wall	West	40.5	0.342
External Wall	East	40.5	0.342
Roof	North	61.4	0.227
Roof	South	25	0.227
Ground	Horizontal	70	0.196
Internal Wall		200	2.686

Description of the thermal energy production systems

For the heating space, we integrated a solar collector and a wood boiler. The surface of the collector is 20 m². Its efficiency is modelled by second order quadratic function.
The boiler was in the first design of the IEA a gas boiler but it will be a wood boiler in our case. Its power is 15 kW, its efficiency is 90%.
The solar collector and the boiler are connected to two loops which supply with hot heating fluid two heat exchangers in a storage tank. This tank is a cylinder tank, it is 1,7 m high with a volume of 0,5 m^3. The tank is heated in priority by the solar collector. Its pump is activated as

soon as the temperature of the fluid exiting the collector is 5°C higher than the tank's bottom's temperature. If the temperature of the tank diminishes lower than 50 °C, the pump of the wood boiler is activated until the temperature of the tank reaches 70°C. The tank also heats domestic heat water.

Table 3: TRNSYS types used for the simulation [2].

System	Type	Details
Solar Collector	Type 1	TRNSYS library
Boiler	Type 700	TESS library
Storage Tank	Type 60	TRNSYS library

Description of the heating floor

The house is heated with a heating floor similar to the floor modelled in the Task 26 [3]. Its surface is 140 m² large, the two stages are supplied with two different loops. It is modelled directly in the definition of the floors of the building. The pipe grid is defined as an active layer of the floor. The pipe spacing from centre to centre is 10 cm, the outside diameter of the pipes is 2 cm. The heating fluid has a 30% glycol fraction, is mass heat is 3.74 kJ/kg. A diverting valve maintains the incoming fluid temperature around 35 °C. The heating fluid is heated in a heat exchanger in the storage tank.

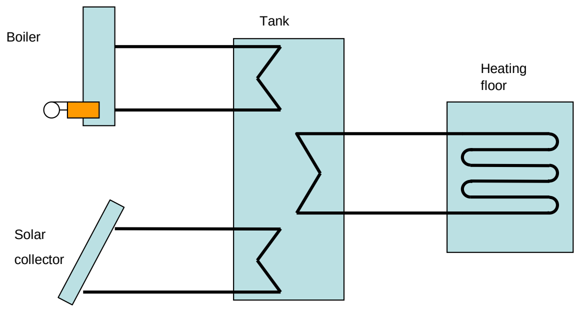

Figure 2: Hydraulics scheme of the dwelling

THE FUZZY CONTROLLER

Principle of fuzzy control

Among the temperature controller available nowadays, we can found the fuzzy controller. Based on fuzzy logic [4], it controls the temperature when the building is occupied and also estimates the necessary time to restart space heating before the end of the inoccupation periods to make sure that the temperature respects the high set point when the next occupation period begins.

Fuzzy logic is a command law which is calculated according the variation of some chosen variable. Considering the case of a single variable e, a grid is realized on the interval $[e_{\min}, e_{\max}]$ defined by the user, which is supposed to be representative of the possible variation of e. We get an ensemble of nodes $\{e_{\min},...,e_i,...,e_{\max}\}$ and sub intervals. To each node e_i corresponds a command value $\mu(e)$, set by the user. For an unspecified e, the command $\mu(e)$ is calculated as following :

$$\text{if } e < e_{\min}, \ \mu(e) = \mu(e_{\min})$$
$$\text{if } e > e_{\max}, \ \mu(e) = \mu(e_{\max})$$

if $e \in [e_i, e_{i+1}]$, then

$$\mu(e) = \mu(e_i)\frac{\mu(e_{i+1}) - \mu(e)}{\mu(e_{i+1}) - \mu(e_i)} + \mu(e_{i+1})\frac{\mu(e) - \mu(e_i)}{\mu(e_{i+1}) - \mu(e_i)}, \qquad (1)$$

In the case of n variables, a *n*-dimension grid is realized and a value $\mu(N_i)$ is set for each node N_i. The command value of an unspecified vector V is the average of each $\mu(N_i)$, taking only the nodes which define the net to which *V* belongs. This average is balanced by the distance of each node N_i to V.

Use of the fuzzy controller in the building

The controller controls the temperature of the building in varying the flow rate of the heating fluid in the heating floor.
The user must first define the periods of occupation and inoccupation in the building. Then he sets the high temperature level for occupation and the low temperature level for inoccupation. The management of inoccupation consists in to steps. The first step consists in turning off the pump of the heating floor, which leads to the descent of the temperature until it reaches the low temperature level. The controller maintains then the temperature of the building around this level. The second step is the restart step. The controller calculates the time when it must produce the maximal flow rate in the heating floor, in order that the temperature reaches the high temperature level at the moment when the occupation period begins. On each time step, the controller calculates the time needed for the temperature to reach the high temperature level. If it is longer than the time before the next occupation period, then the restart step begins and the command is set to its maximum. The restart time is calculated in function of the external temperature and the internal temperature of the building.
The variations intervals are respectively determined between the minimal external temperature which depends of the climate and the high temperature level and between the low and high temperature levels. The restart duration *t* is a linear function of the external temperature T_e, that's why only two nodes are used for the grid on this variable. We use seven no des for the variation of the internal temperature T_i, to guarantee the best precision. The fuzzy function which defines the restart time for each node $(T_{e,j}, T_{i,k})$ with $(j,k) \in \{[1;2],[1;7]\}$ is:

$$t_{(T_{e,j},T_{i,k})} = \alpha \frac{T_{\max} - T_{e,j}}{T_{\max} - T_{\min}}\left(\frac{T_{i,k} - T_{\max}}{T_{\max} - T_{\min}}\right)^3, \qquad (2)$$

We have $T_{\max} = 20$ °C, $T_{\min} = -10$ °C, and $\alpha = 10000$ hr. The two first values are defined by the weather conditions. The last one has been empirically chosen.
For the occupation periods, we also use fuzzy control to calculate in function of the internal and external temperature the best flow rate command for the heating floor which will maintain the temperature of the building around the high temperature level. Seven nodes are used on both variables. The fuzzy function which defines the flow rate for each node $(T_{e,j}, T_{i,k})$ with $(j,k) \in \{[1;7],[1;7]\}$ is :

$$Q_{(T_{e,j},T_{i,k})} = Q_{\max} + \frac{Q_{\min} - Q_{\max}}{6}(j-1) - \beta\frac{(6-k)}{6}Q_{\max}, \qquad (3)$$

For the simulation, we have $Q_{\max} = 700$ kg/hr, $Q_{\min} = 0$ kg/hr and $\beta = 0{,}3$.

STUDY OF THE CONTROLLER'S PERFORMANCES

Description of the simulation

The controller was set with inoccupation periods between 6 A.M. and 7 P.M., from Monday to Friday. The low temperature level is set on 10°C and the high temperature level to 20°C. The real inoccupation periods are between 8 A.M. and 7 P.M., but the inertia of the building is such that the descent step can be anticipated. The flow rate of the heating floor can be set between 0 and 700 kg/hr, the fluid's temperature is set to 35 °C. However, our modelling choice imposes minimal flow rate value in the floor : the floor is divided in a limited number of nets. The lower is the minimal limit, the greater is the number of nets. 250 kg/hr is the lower limit our simulation program can bear.
Furthermore, 200 litres of domestic hot water, temperature 45°C, is produced each day. The simulation takes place in Lyon Bron, France, during the two first week of January. The time step is 6 minutes. We monitored three phenomena: the first one is the evolution of the temperature in the building. To satisfy our performance criterion, the temperature must remain around the high temperature level with a limit of more or less 0,5°C during the occupation periods. The second one is the evolution of the heating power supplied by the heating floor to the building. We will estimate the energy spent to heat the dwelling. The last one is the evolution of the tank temperature and the evaluation of the energy supplied by the solar collector and the boiler.

Results

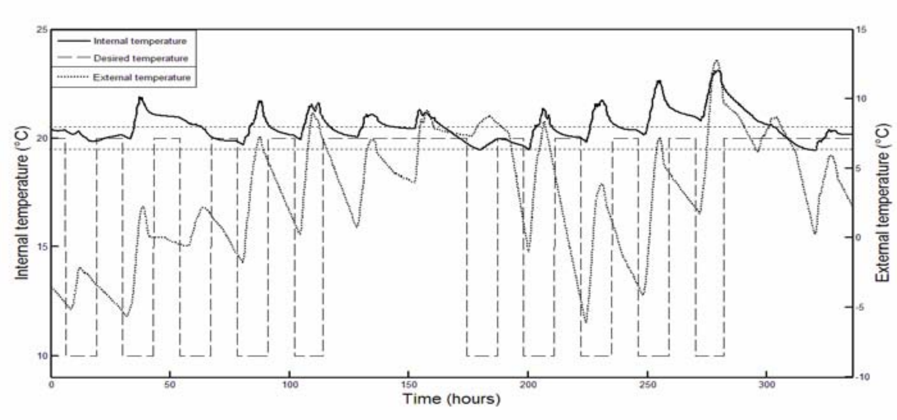

Figure 3: Evolution of the temperature in the dwelling

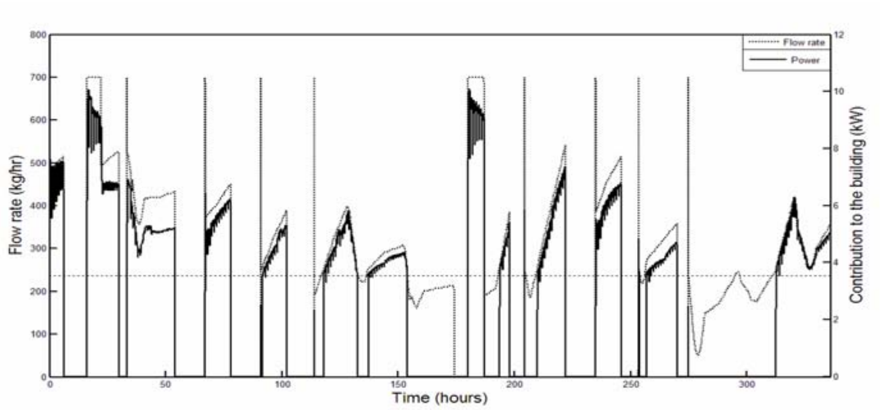

Figure 4: Flow rate and power of the heating floor

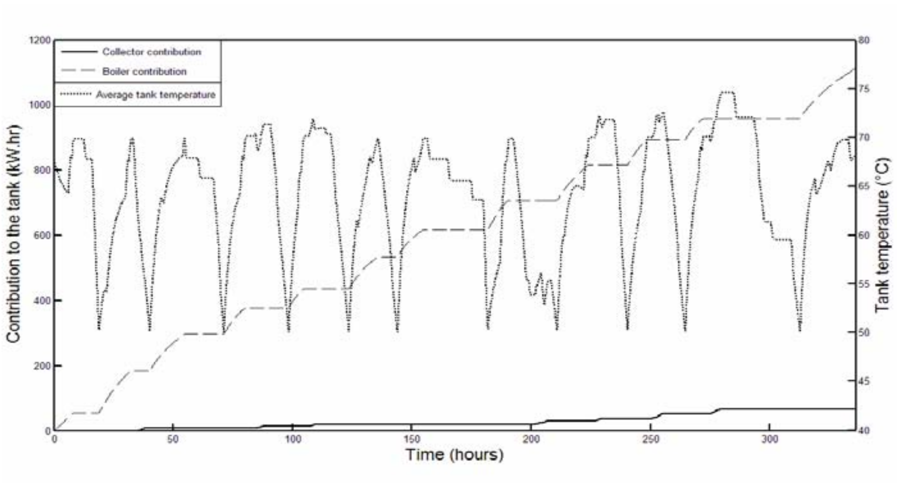

Figure 5: evolution of the temperature of the tank

Analysis

We notice on figure 3 that our criterion is verified during the week, excepted during the first Tuesday and the second Thursday when some overheat can be noticed. The control during the week end is more delicate, the temperature is sometimes too high. We also observe that the temperature doesn't necessarily decrease during the occupation period. The solar gains are sometimes so high that the temperature increases even if the heating floor is turned off. This is the same phenomenon which produces overheats during the week end, since these solar contributions aren't taken into account by the controller during the occupation period. If the temperature somehow decreases during the inoccupation period like during the first and the second Monday, the restart time is well evaluated and the following occupation period begins with a 20°C temperature.

We notice on figure 4 that the heating power given to the building is proportional to the flow rate in the heating floor. This power is null during the inoccupation period, excepted during the restart stage. If the temperature doesn't decrease during the inoccupation period lower than the high temperature level, the restart stage is skipped. The controller actually considers this stage during one time step, that's why we can notice thin flow rate peaks on the beginning of some occupation periods.

The biggest problem of the controller is the fact that it keeps heating the building whereas the temperature is already too high. This is the consequence of the minimal flow rate limit, we have to make sure that the calculated flow rates remains above the limit most of the time, which sometimes leads to overheats.

The total energy contribution of the heating floor to the building is 890 kW.hr

We notice on figure 5 that during these two weeks, the solar gain to the storage tank is low. This could be expected since in this season the solar radiations are weak. We will however notice that the final solar gain is 64 kW.hr and the boiler gain is 1115 kW.hr.

COMPARISON WITH A LINEAR CONTROL METHOD

Control method

Our fuzzy controller passed the test of our criterion, despite some high temperature sometimes and some limitations from our model. We will now compare its performances to a standard linear control.

The flow rate is calculated in function of the external temperature the following way :

$$Q_c(T_e) = Q_{\max}\frac{T_{\max} - T_e}{T_{\max} - T_{\min}}, \qquad (4)$$

$Q_{\max}$, $T_{\max}$ and $T_{\min}$ are empirically determined in order to get the best performances. $T_{\max}$ and $T_{\min}$ delimit the usual variation interval of the outside temperature.
In our case, we respectively chose for those three values =450 kg/hr, 20°C and -15°C.
The goal is to constantly maintain the temperature around the same temperature level of 20°C.

Results

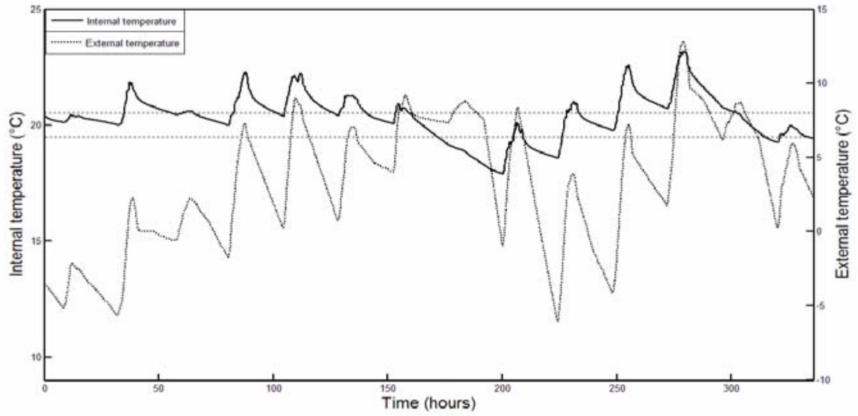

Figure 6: Evolution of the temperature with a linear control

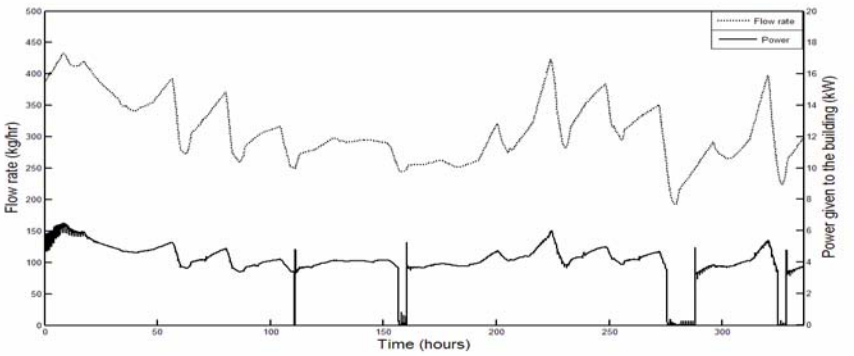

Figure 7: Flow rate and power of the heating floor with a linear control

Analysis

The performance criterion in this case for the temperature control is still to keep the temperature in the $20 \pm 0{,}5°C$ area. But this time, this criterion prevails during the whole simulation, when it only mattered during the occupation period for the fuzzy controller. We can't therefore make a quantitative comparison to determine which controller best respects the criterion but we can somehow make qualitative comparison.
We can see on figure 6 temperature risings on the same moment like on figure 3. The cause of these peaks is the same: the solar contribution isn't taken into account by the controller. However, the amplitude of these peaks is greater with the linear controller. Since the solar contribution happens during the inoccupation period, the fuzzy controller turns the heating

floor off and compensates the solar gain. The linear controller makes no difference and keeps heating while the internal temperature is already too high.
We also notice that the temperature decreases twice below 19,5°C. We can conclude that the fuzzy controller produces a better control than the linear controller.
The calculated flow rate remains between 200 and 450 kg/hr. We can confirm the proportional relationship between the flow rate and the heating power.
The heating power contribution to the building is 861 kW.hr, 3,7% less than the fuzzy controller. The boiler contribution to the tank is 914 kW.hr and the collector contribution is 69 kW.hr.

CONCLUSION

In our study, the fuzzy controller doesn't save more energy than the linear controller.
However, its control is better than the linear controller since it takes profits of the solar gains and the inertia to lower the restart step duration.
The fuzzy controller is a powerful controller but it is limited by its simple rules. It handles well the inoccupation periods, but this capacity doesn't save much energy, because of their short duration. It will in spite of that be our reference to compare the performances of a new controller which we started to design.
Our strategy consists in using advanced automation techniques such as optimal command [5] to improve the exploitation of inoccupation periods and predictive command to better anticipate external gains such as solar gains and internal gains, caused by inhabitants and electric devices. We will also use a state system modelling to anticipate on its behaviour.

ACKNOWLEDGEMENT

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Analyze performances of a Discrete Controller and a Fuzzy Regulator for heating with intermittency of a dwelling

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SUMMARY

In this paper, we choose to present the generic form of an optimal control law for heating building. To build it, three successive steps are necessary: (i) the first one allows to determine the inside temperature set point that takes into account intermittency and tends to reduce the cost control; (ii) the second that corresponds to the discrete controller design is based on a quadratic mixed criterion “cost and efficency”; (iii) the third corresponds to the design of the virtual sensor. Efficiency of this approach is shown via a numerical example and compared with a Fuzzy regulator (marketed regulator).

INTRODUCTION

Taking into account the intermittency of buildings’ occupation allows saving a significant amount of energy. Many studies are related to saving heating in buildings, because these ones in tertiary sector are only occupied for 30% of time. Today, it is also necessary to work on the residential sector where intermittency exists: dwelling houses being usually empty during the day and may be slowed right down at night.

One way to improve these installations is working downstream of heating sources and defining a strategy to minimize the energy required for heating the dwelling houses. This last one has to take into explicit account of intermittency by stressing the passive contributions and storage in vacant periods. Thus during the day, solar contributions must be exploited as much as possible and energy storage must be optimized in order to use it during evening, when the outside temperature is the less favourable.

An early study in continuous time [1] has already shown the interest of an optimal control strategy (via simulation) [2,3]. In order to take into account technical considerations such as the choice of sampling, we propose to develop in discrete time a similar strategy, based on optimal control for sampled systems. Searching an extremum for the discrete performance criterion leads to a command in the form of recursive equations ready to set up. In [1], a Feed-Forward using direct measurement of outside temperature is also used. To take into account solar contribution, Feed Forward needs an outside sensor and a model of behaviour between solar radiance and inside temperature. This last task is hard to carry out because of the non stationary and non linear feature of solar radiance and also due to solar contributions trough wall and windows. Indeed these last ones show significant difference in term of dynamic behaviour. Thus, we proposed here an elegant solution of this problem.

First, a cost criterion is used to determine the optimal trajectory of inside temperature during inoccupation stage. Searching an optimum of this criterion leads naturally to define the right time to come back to the occupation set point. Thus, a complete profile of temperature set point is built taking into consideration the specific constraints of intermittency (high and low set point).
Second, the structure of the regulator is obtained using a performance criterion composed of quadratic terms related to the quality of the heating control and cost.
Third, a virtual sensor is defined to estimate states required to control the system and the equivalent control that will be delivered by an exact real dynamic Feed-Forward using a direct measurement of disturbances. However, the virtual sensor uses only measurement of inside temperature (call virtual dynamic feed-forward later). It uses an optimal estimator in sense of the minimum variance of estimation error. This step is treated as a deconvolution problem.
Last, to show efficiency of this approach, a numerical example is treated considering only the outside temperature as an outside source. After having presented the cell test (Minibat [4]) and the behaviour model we selected, our new approach will be compared with an existing fuzzy regulator (marketed regulator). To end, we will compare our virtual Feed Forward with an exact real dynamic Feed Forward.

METHOD DEVELOPPMENT

In this part, we choose to present the generic form of the control law. To build it, three successive steps are necessary: (i) the first one allows to determine the inside temperature set point that takes into account intermittency and tends to reduce the cost control; (ii) the second that corresponds to the discrete controller design is based on a quadratic mixed criterion “cost and efficency” [2,3]; (iii) the third corresponds to the design of the virtual sensor [5].

Let us consider the general representation of state of the sampled linear systems, as follows:

$$\begin{aligned} x_{k+1} &= Fx_k + Gu_k \\ y_k^m &= Cx_k + d_k \end{aligned} \qquad (1)$$

With x_k the state vector, u_k the control vector, y_k^m the measurement, d_k the contribution of outside source. F, G and C are matrices with appropriate dimensions.
It is right to consider that the inside temperature of a dwelling is the sum of all sources’ contributions (heating device, outside temperature or solar radiance). Thus the corresponding model d_k will be considered in a first assumption as a white stochastic process, non-zero mean. In this paper, the variance of such process will be considered as constant.

Step 1 – Intermittency and set point profile

In this step, as we are interested in an ideal set point profile, a representation without disturbance must be considered:

$$\begin{aligned} x_{k+1} &= Fx_k + Gu_k \\ y_k &= Cx_k \end{aligned} \qquad (2)$$

The aim is to define the optimal couple (u^*,y^*) for the inoccupation phase respecting specific constraint previously stated. So a quadratic cost criterion is used

$$J_1 = 0.5\left[y_{N_2}^r - y_{N_2}\right]^T Q_{N_2}\left[y_{N_2}^r - y_{N_2}\right] + 0.5\sum_{k=N_1}^{N_2-1} u_k^T R_k u_k \qquad (3)$$

With N_1 and N_2 the first and the last sample of inoccupation phase. R_i and Q_i are matrices respectively definite positive and semi-definite positive. The first term of the criterion is used to find the optimal restart instant. The second allows the determination of the output trajectory that requires the lowest command.

Let us consider now the cost function Φ_k

$$\Phi_k(u_k) = 0.5 u_k^T R_k u_k \tag{4}$$

And the Hamiltonian H_k such as:

$$H_k = -\Phi_k + \beta_{k+1}^T [F x_k + G u_k] \tag{5}$$

with the adjoin state β_k.

Using the Discret Maximum Principe [3], we obtain the minimal control u_k^* :

$$\beta_k = -\frac{\partial H}{\partial x_k} \quad \Rightarrow \quad \beta_k = F^T \beta_{k+1} \tag{6}$$

$$\frac{dH}{du_k} = 0 \Rightarrow u_k^* = R_k^{-1} G^T \left(F^T\right)^{N_2-k-1} \beta_{N_2} \tag{7}$$

Using (7) in (1), some manipulations lead to:

$$\beta_{N_2} = \left(\sum_{k=N_1}^{N_2-1} F^k G R_k^{-1} G^T \left(F^T\right)^k \right)^{-1} \left(I - F^{N_2}\right) x_{N_2}^r \tag{8}$$

with $x_{N_2}^r$ the wished final value of state.

Using (7), (8), we obtain the ideal set point trajectory y_k^* for the inoccupation phase:

$$\begin{gathered} x_{k+1}^* = F x_k^* + G R_k^{-1} G^T \left(F^T\right)^{N_2-k-1} \left(\sum_{k=N_1}^{N_2-1} F^k G R_k^{-1} G^T \left(F^T\right)^k \right)^{-1} \left(I - F^{N_2}\right) x_{N_2}^r \\ y_k^* = C x_k^* \end{gathered} \tag{9}$$

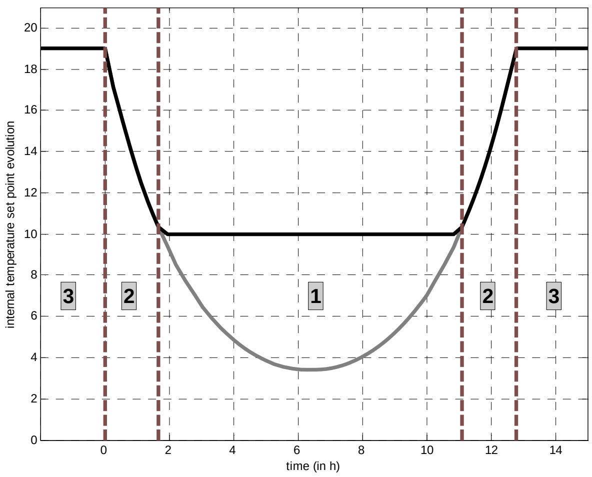

Figure 1: set point trajectory (gray – without constraint, black – with constraint)

Thus, the set point profile without constraint corresponds to a hyperbolic trajectory (9). Taking into account the specific constraint of each phase, the global trajectory of temperature is illustrated Figure 1**.** The evolution is finally composed by 3 parts. In part 2, if a regulator ensures *y**, the corresponding control is equivalent to the lowest *u**. Indeed, the dwelling is considered as a mono-zone space: the input is the heating power and the resulting output is the global inside temperature (considered as homogeneous). This SISO characteristic (Single Input Single Output) leads to a single solution. As illustrated Figure 1, constant set points are used (high and low set point) in parts 1 and 3 according to occupation and inoccupation constraints.

Step 2 – Design of control parameters

From the global set point profile defined in step 1, we are going to treat a pursuit problem in order to design the discrete controller. Thus, let us consider the following criterion:

$$J = 0.5\left[y_N^r - y_N \right]^T Q_N \left[y_N^r - y_N \right] + 0.5\sum_{k=0}^{N-1}\left\{\left[y_k^r - y_k \right]^T Q_k \left[y_k^r - y_k \right] + u_k^T R_k u_k \right\} \tag{10}$$

where y_N^r is the high set point and N is the total number of samples included in the whole cycle.

In order to ensure the reduction of the effect due to the disturbance, a new state z_k is introduced. That leads to an increased discrete representation equivalent to the representation of Johnson into the continuous case [1].

$$\begin{aligned} X_{k+1} &= \begin{bmatrix} x_{k+1} \\ z_{k+1} \end{bmatrix} = F_a X_k + G_a u_k^c \quad \text{with} \quad F_a = \begin{bmatrix} F & G \\ 0 & I \end{bmatrix} \\ Y_k &= C_a X_k \quad \text{with} \quad C_a = \begin{bmatrix} C & 0 \end{bmatrix} \text{ and } G_a = \begin{bmatrix} 0 \\ I\ \mathrm{T_e} \end{bmatrix} \end{aligned} \tag{11}$$

with Te the sample rate, I the identity matrix and subscript “a” to identify the increased representation for control. We introduce the Income function Ω_k defined by:

$$\begin{aligned} \Omega_k &= 0.5\left[X_k^r - X_k \right]^T C_a^T Q_k^a C_a \left[X_k^r - X_k \right] + 0.5\, u_k^{cT} R_k^a u_k^c \\ &\text{with} \quad Q_k^a = \begin{bmatrix} Q_k & 0 \\ 0 & R_k \end{bmatrix} \text{ and } R_k^a = \theta_k \end{aligned} \tag{12}$$

with θ_k a weighting matrix, definite positive and symmetrical.
The optimal value of the criterion is defined by a recurring form of the type:

$$J_k^* = \min_{u_k^c}\left(\Omega_k + J_{k+1}^* \right) \tag{13}$$

Where J_k^* is supposed to be a quadratic form. Thus, we define it as follows:

$$J_{k+1}^* = 0.5\left(X_{k+1}^T P_{k+1} X_{k+1} + 2\lambda_{k+1}^T X_{k+1} \right) \tag{14}$$

With λ_k a adjoin vector and P_k a definite positive and symmetrical matrix. By taking account (11), (13) and (14), the minimum of the criterion (10) with respect to u_k^c is defined as follows:

$$u_k^{c*} = -\left(R_k^a + G_a^T P_{k+1} G_a \right)^{-1} G_a^T \left(P_{k+1} F_a X_k + \lambda_{k+1} \right) = -K_k^c X_k + L_k \lambda_{k+1} \tag{15}$$

As J_k^* is a quadratic form that implies:

$$P_k = \left(F_a - G_a K_k^c \right)^T P_{k+1} \left(F_a - G_a K_k^c \right) + K_k^{cT} R_k^a K_k^c + C_a^T Q_k^a C_a \quad \text{with} \quad P_N = C_a^T Q_{N_1}^a C_a \tag{16}$$

$$\lambda_k = \left(F_a - G_a K_k^c \right)^T \lambda_{k+1} - C_a^T Q_k^a C_a X_k^r \qquad \text{with} \quad \lambda_N = C_a^T Q_{N_1}^a C_a^T X_{N_1}^r \tag{17}$$

Let us note that these two last equations must be solved backward in time. We must take care to solve them before using the controller. The real command of the process is then defined by:

$$u_k^* = z_k \qquad \rightarrow \qquad u_{k+1}^* = u_k^* + T_e\left[L_k \lambda_{k+1} - K_k^c \begin{bmatrix} x_k \\ u_k^* \end{bmatrix} \right] \tag{18}$$

Step 3 – Design of virtual sensor

Considering (1) and step 2, one can notice that the problem is equivalent to use a LQG approach [3]. At this step of design, x_k is required but not available, only the measurement y_k^m is available. Moreover, (1) is a stochastic process, thus, it is necessary to use an estimator to obtain an estimate of real trajectories followed by the states of the system [5]. We remind that our aim is the design of an estimator which estimates both the states and an equivalent contribution of disturbances. Thus, after introducing a deconvolution problem, we summarize a Kalman estimator.

We first define an equivalent system (with equivalent output) in which disturbance is considered as a stochastic unknown input.

$$\begin{matrix} x_{k+1} = Fx_k + Gu_k \\ y_k^m = Cx_k + d_k \end{matrix} \rightarrow \begin{matrix} x_{k+1} = Fx_k + Gu_k + Gw_k \\ y_k^m = Cx_k + v_k \end{matrix} \tag{19}$$

Where w_k is a non zero mean white noise with constant variance and v_k the noise measurement considered as a zero mean white noise with constant variance.
To overcome any estimation bias, we can consider that w_k is the output of a Wiener process generator driven by a zero mean Gaussian white noise with constant variance. That lead to the following representation:

$$\begin{gathered} x_{k+1}^e = \begin{bmatrix} x_{k+1} \\ w_{k+1} \end{bmatrix} = F_e x_k^e + G_e u_k + M_e \omega_k \text{ with } F_e = \begin{bmatrix} F & G \\ 0 & I \end{bmatrix}, G_e = \begin{bmatrix} G \\ 0 \end{bmatrix} \text{ and } M_e = \begin{bmatrix} 0 \\ I\ \mathrm{T_e} \end{bmatrix} \\ y_k^m = C_e x_k^e \text{ with } C_e = \begin{bmatrix} C & 0 \end{bmatrix} \end{gathered} \tag{20}$$

with subscript “e” to identify the increased representation for estimation.
From (20), an optimal estimator in sense of the minimum variance of estimation error can be defined (Kalman Theory []). Thus, in the stationary case, we obtain:

$$\Pi = F_e \Pi F_e^T + M_e Q_\omega M_e^T - F_e \Pi C_e^T \left[C_e \Pi C_e^T + R_v \right]^{-1} C_e \Pi F_e \tag{21}$$

With Π a definite positive and symmetrical matrix, Q_ω and R_k are respective input and measurement noise variance.

$$K_e = \Pi C_e^T \left[C_e \Pi C_e^T + R_v \right]^{-1} \tag{22}$$

With K_e the estimator gain
The “a priori” estimation is defined by:

$$\hat{x}_{k/k-1}^e = F_e\ \hat{x}_{k-1/k-1}^e + G_e u_{k-1} \tag{23}$$

And the “a posteriori” estimation is defined by:

$$\hat{x}_{k/k}^e = \hat{x}_{k/k-1}^e + K_e \left(y_k^m - C_e \hat{x}_{k/k-1}^e \right) \tag{24}$$

Step1 + Step2 + Step 3 = Control law

The final expression of control is as follows:

$$u_k = u_{k-1} + T_e \left[L_{k-1} \lambda_k - K_{k-1}^c \begin{bmatrix} \hat{x}_{k-1/k-1} \\ u_{k-1} \end{bmatrix} \right] - \hat{w}_{k/k} \tag{25}$$

RESULTS

Performances of the new controller are shown thanks to the model [] of the test cell Minibat developed by the CETHIL [4]. This test installation is a 2 identical contiguous area with a controlled climatic environment (temperature, sunning). The rooms dimensions are *3.10 m* $\times$ *3.10 m* $\times$ *2.50 m* (L x l x h). The external jacket consists of standard insulating concrete (Siporex) with *20 cm* thickness and has following dimensions: *7.5 m* $\times$ *4.50 m* $\times$ *3.43 m.* Walls of the cell are composed of agglomerated wood panels of *5 cm* thick, covered with a pressure-sealed plasterboard of *1 cm*. Only the southern wall is equipped with a window of *1 cm* thickness. The floor is made of a concrete flagstone of *20 cm* thickness. A set of runs was carried out with the cell in order to identify its dynamic and static behaviour. A first order continuous model is obtained:

$$F = e^{-\frac{T_e}{\tau}}, \quad G = b\left[1 - e^{-\frac{T_e}{\tau}}\right] \quad C = 1 \qquad (26)$$
$$\text{with} \quad T_e = 1000 \text{ s}, \ \tau = 9564\text{s} \ \text{ and } \ b = 0.008413$$

F, G and C are defined as in (1).

In our case, the outside temperature is the only disturbance considered. The weather data come from the Météonorm station located at Lyon – Bron, France. Simulation runs on two days with occupation range 8h00-19h00

Figure 2 shows results we obtained with a controller using an optimal control strategy combined with a real dynamic Feed-Forward (RFF) when all information are known (states and disturbance). Figure 2 shows also results we obtained with the same kind of control strategy but information needed is estimated with a virtual sensor. It clearly appears that performances of both controllers are quite similar.

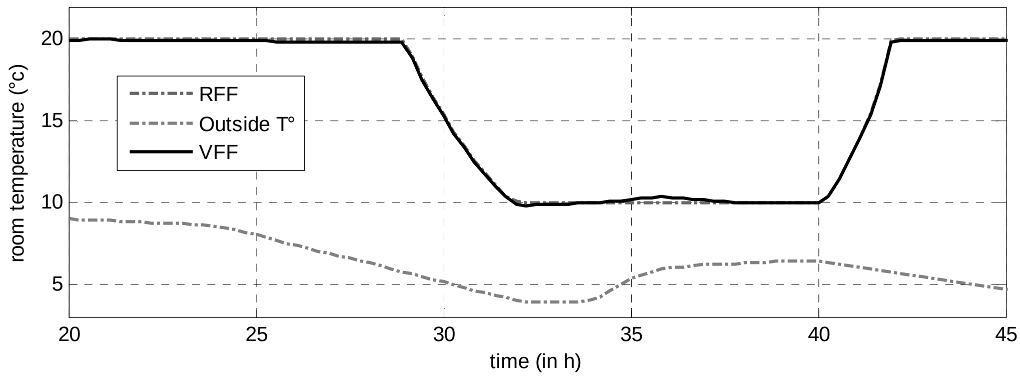

Figure 2 ***: Comparison of our controller (VFF) with an optimal control with all states available and real dynamic feed forward (RFF)***

Figure 3 gives evolution of inside temperature we obtained with a fuzzy controller [6, 7, 8, 9], and our discrete controller. Our new strategy makes it possible to satisfy with a great efficiency the constraints. Let us consider now the first inoccupation phase on Figure 3 and Figure 4. When the inoccupation phase starts, the Fuzzy controller is designed to deliver a constant control whatever external contributions. As a result, this technique spends more energy than necessary. In our case, the control is automatically reduced or stopped when no energy is required and restarts automatically when the low set point is not satisfied. Moreover, for the Fuzzy controller, the time to come back is defined a priori by the operator and the associated control is the maximum control. Thus, room temperature is equivalent to the high set point too much early. As a conclusion, energy expenditure with the Fuzzy controller is more important.

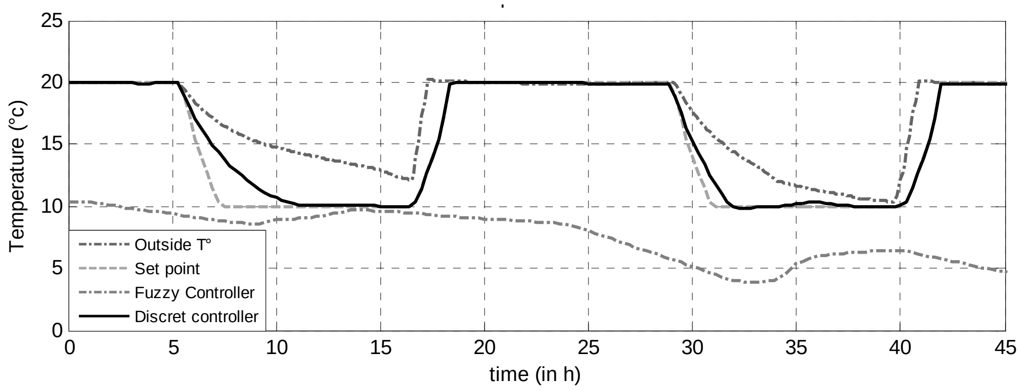

Figure 3 ***: Comparison of our controller (VFF) with a Fuzzy controller during two days (room temperature)***

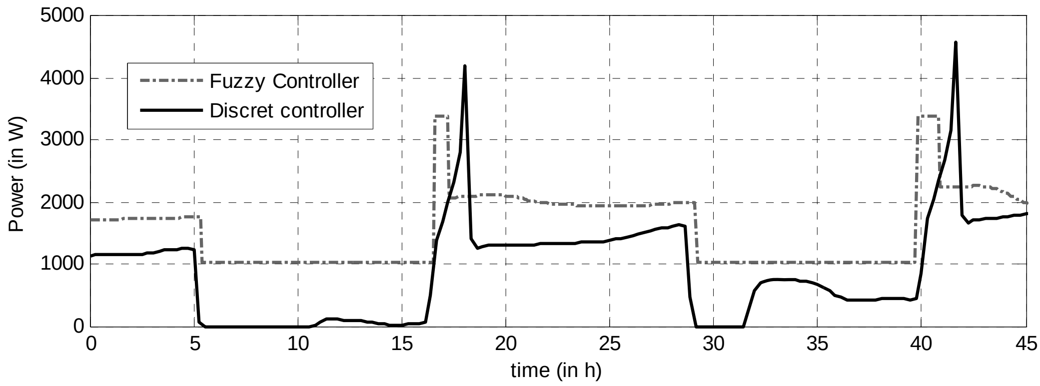

Figure 4 ***: Comparison of our controller (VFF) with a Fuzzy controller during two days (control)***

CONCLUSION

In this article, the synthesis of a discrete controller for the control of the temperature in a building was presented. After introduction, a Cost criterion makes it possible to characterize the specific behaviour to each phase (occupation and inoccupation). Parameters of the control law are next obtained using an optimal control analysis from a Cost-Efficiency criterion. A virtual sensor is then defined to take into account outside contributions from only room temperature measurement. Results of simulations are particularly hopeful and show, in comparison with a fuzzy regulator, the interest to use a model of behaviour of the test cell and more generally of a building. Our controller allows to decrease significantly the amount of energy needed to heat the building.

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Suggestion and verification of thermal storage heating and cooling systems by a simplified predictive control

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SUMMARY

In this research, we suggested thermal storage heating and cooling systems by a simplified predictive control for highly insulated office buildings and verified the usefulness of this control technique on the basis of energy conservation and thermal comfort in offices. At first, the calculation procedures of a simplified predictive control were shown. From the viewpoints of energy conservation and thermal comfort, it was very effective to supply heat continuously, such as continuous operation or nighttime operation, in the heating season, and to supply heat intermittently, such as intermittent operation or nighttime operation, in the cooling season.

INTRODUCTION

Recently, bills related to energy conservation were enforced and revised in Japan, and buildings have become highly insulated and air tightened. In contrast with European countries, external insulation has gradually spread around cold districts in Japan, like Hokkaido, where internal insulation has generally been used. However, under the present conditions, even with externally insulated buildings of reinforced concrete, problems related to control techniques such as suspending operations on Saturday and Sunday and overheating room air temperature in the daytime, could not be always solved. From the above, the main purpose of this research is to suggest a newly simplified control technique and to verify the usefulness of this control technique on the basis of energy conservation and thermal comfort in offices. Heating and cooling systems used in this research utilize thermal storage heating and cooling systems consisting of piping laid inside the floor slab and transporting water to pipes with minimal power.

OUTLINE OF BUILDING MODEL AND SIMPLIFIED PREDICTIVE CONTROL

In this research, we postulated a building model and reproduced the thermal fluctuations of an actual building by numerical simulation. This model is of an externally insulated office building that had introduced thermal storage heating and cooling systems, built in Sapporo (Figure 1). The scale of this model was the standard scale of government buildings in Hokkaido (total floor area: 3000 [m^2]), the details of which are as follows: four-storied building, total floor area 3168 [m^2], length of east-west direction 45 [m] and length of south-north direction 17.6 [m]. Concrete volume of this model is shown as the thickness of exterior walls 120 [mm] and slab 150[mm] by adjusting the direction of the depth (z) to a two dimensional model (x-y) in the direction of cross section in each direction. As insulation specifications, EPS external insulation with exterior walls (100 [mm]) and roof slab (150 [mm]) were adopted. Insulation performance became Perimeter Annual Load (PAL) 201 [MJ/m^2 year] (heating PAL 74 [MJ/m^2 year], cooling PAL 127 [MJ/m^2 year]). This value is equivalent to a 30% reduction in the standard value of bills related to energy conservation, and means that this is a very highly insulated building model.

Thermal storage heating and cooling systems are constructed by diverting hot and cold water to cross-linked polyethylene piping (13A) laid inside a floor slab. The depth of piping laid inside the floor slab is 70 [mm], at an interval of 150 [mm]. We have explained a simplified control technique for these systems. We transcribed the simplified control technique suggested in this research into a simplified predictive control, and show the calculation procedures below (Table 1, from a) to d)). Three case studies were done in which the operation methods presumed intermittent operation, continuous operation and nighttime operation (Table 2).

a) Calculation of predictive thermal load: Predictive thermal load was calculated by using mean values for a day of HASP meteorological data. The 2005 calendar was adopted, and thermal load caused by heat gain and ventilation was defined as zero on Saturday and Sunday. When natural room air temperature changed between 22 and 26 [degrees centigrade], this period was defined as the moderate season (*M*). As a result, the maximum predictive heating load became about 1292 [kWh/day] (January 20 (Thursday)) in the heating season (H), and also the maximum cooling load became about 1475 [kWh/day] (July 27 (Wednesday)) in the cooling season (C) (Figure 2).

b) Calculation of equipment capacity of heat source: Equipment required as capacity of heat source for 10-hour intermittent operation or 24-hour continuous operation was calculated by using predictive thermal load. As a result, in the case of intermittent operation, maximum heating and cooling capacity became 153.3 [kW] and 151.9 [kW], and also in the case of continuous operation, 143.1 [kW] and 76.2 [kW] (Figure 3).

c) Calculation of water temperature setting: It was presumed that floor area, which evolved or absorbed heat for heating and cooling, was 3168 [m^2] and ceiling area was 2376 [m^2]. Water temperature setting was calculated to supply predictive thermal load based on the thermal resistance from the center of the pipe to the surfaces of the floor and ceiling.

d) Calculation of approximate water temperature setting: As thermal storage heating and cooling systems were presumed so as to control the constant flow rates of water, water temperature setting repeatedly change throughout the day corresponding to predictive thermal load. However, in order to make it usable as a simplified predictive control, we defined it approximately by the change throughout the year as approximate water temperature setting.

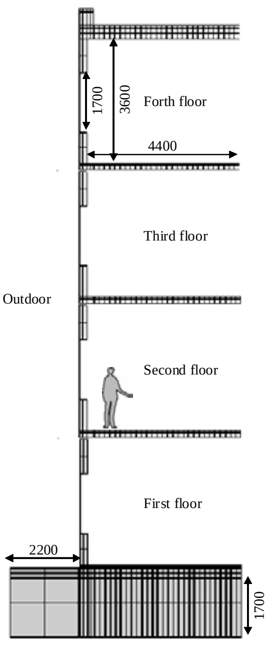


Figure 1. Building model

Table 1. Calculation processes of predictive thermal load

Calculation processes
Predictive heating load [Wh/day] = $Q_1 - Q_2 - Q_3 + Q_4 + Q_5$
Predictive cooling load [Wh/day] = $- Q_1 + Q_2 + Q_3 - Q_4 - Q_5$
Thermal load of walls [Wh/day] = Q_1 = *Siguma* (*Siguma* $K_n \times S_n \times (T_i - T_o)$)
Thermal load caused by solar radiation [Wh/day] = Q_2 = *Siguma* (*Siguma* $S_n \times J_n \times SC$)
Thermal load caused by heat gain [Wh/day] = Q_3 = 87859.2 [W] × 8 [h/day] (9:00 ~ 17:00)
Thermal load caused by infiltration [Wh/day] = Q_4 = 0
Thermal load caused by ventilation (V) [Wh/day] = Q_5
= 0.34 × 12672.0 [m^3/h] × ($T_i - T_{od}$) × 8 [h/day] (9:00 ~ 17:00)
Notes
K_n : Thermal transmittance of n [W/m^2 K], S_n : Area of n [m^2]
J_n : Value of solar radition of n (HASP, 0:00 ~ 24:00) [W/m^2]
SC : Shading coefficient (0.46)
T_i : Room air temperature for design
(Heating season (H): 22 [degrees centigrade], Cooling season (C): 26 [degrees centigrade])
T_o : Mean outdoor air temperature for a day (HASP, 0:00 ~ 24:00)
T_{od} : Mean outdoor air temperature during the daytime (HASP, 9:00 ~ 17:00)
Thermal load caused by exterior walls (W) [Wh/day] = $Q_1 + Q_2 + Q_3 + Q_4$
T_o on Q_1 in the cooling season ⇒ T_{SAT} (Equivalent outdoor air temperature)
87859.2 [W] = 41.6 [W/m^2] × 3168 [m^2] × 2 / 3
12672.0 [m^3/h] = 30.0 [m^3/person h] × 0.2 [person/m^2] × 3168 [m^2] × 2 / 3

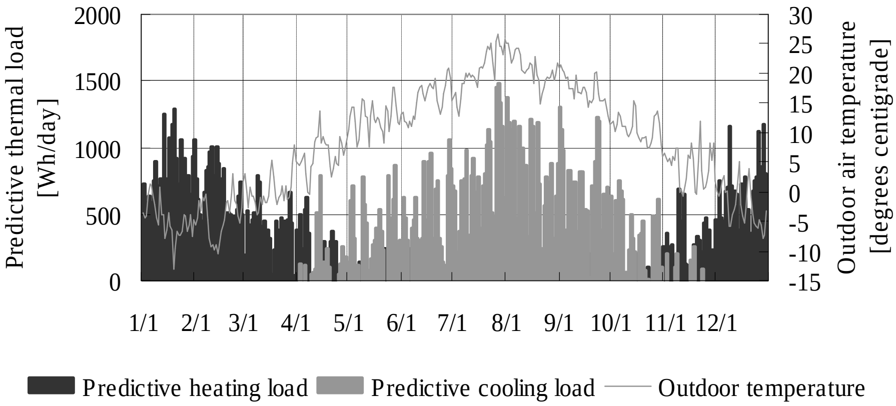
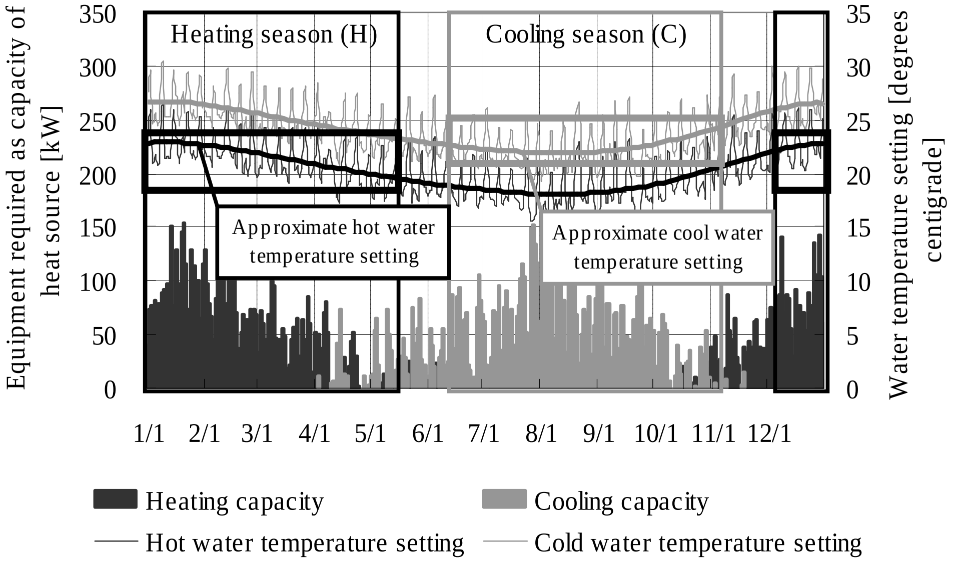
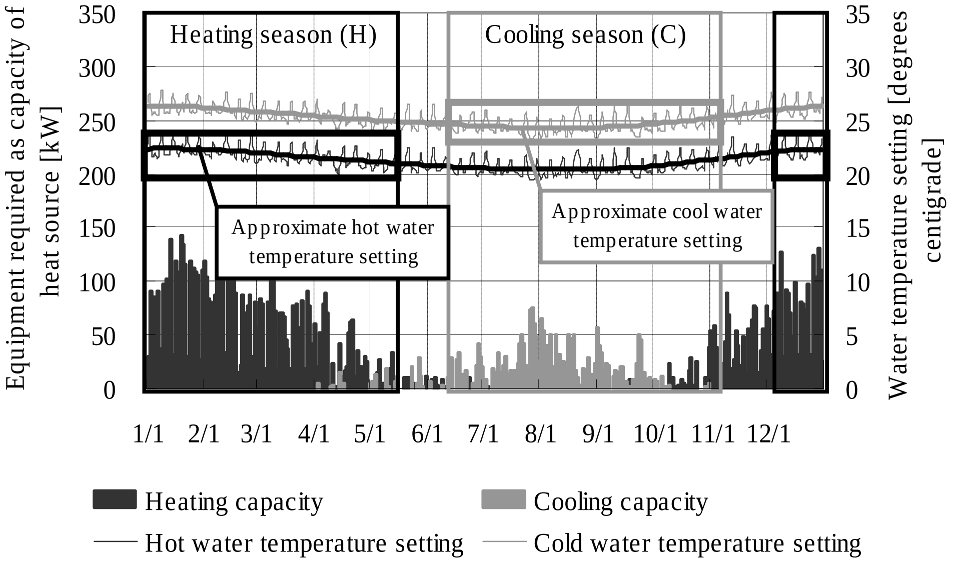


Figure 2. Predictive thermal load throughout the year



a)



b)

Figure 3. Equipment required as capacity of heat source and water temperature setting
a) 10-hour intermittent operation, b) 24-hour continuous operation

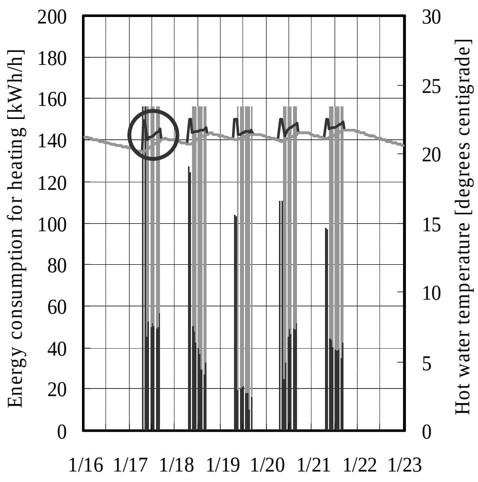
Table 2. Case Studies

Case Studies	Operation methods	Operation hours for a day	Operation start time	Operation stop time	Equipment capacity of heat source[kW]		Water inlet temperature [degrees centigrade] (y: Approximate water temperature setting, x = 184 (January 1)～548 (December 31))
CASE1	Intermittent operation	10 hours	7:00	17:00	Heating season (H)	156.5	$y = 2.6E\text{-}15x^6 - 1.5E\text{-}11x^5 + 1.9E\text{-}08x^4 - 8.9E\text{-}06x^3 + 1.7E\text{-}03x^2 - 9.8E\text{-}02x + 19.3$
					Cooling season (C)	151.5	$y = 3.8E\text{-}15x^6 - 1.7E\text{-}11x^5 + 2.0E\text{-}08x^4 - 9.2E\text{-}06x^3 + 1.8E\text{-}03x^2 - 1.0E\text{-}01x + 23.1$
CASE2	Continuous operation	24 hours	0:00	24:00	Heating season (H)	140.9	$y = 1.1E\text{-}15x^6 - 6.2E\text{-}12x^5 + 7.7E\text{-}09x^4 - 3.7E\text{-}06x^3 + 7.2E\text{-}04x^2 - 4.1E\text{-}02x + 20.9$
					Cooling season (C)	77.7	$y = 1.6E\text{-}15x^6 - 6.9E\text{-}12x^5 + 8.2E\text{-}09x^4 - 3.9E\text{-}06x^3 + 7.4E\text{-}04x^2 - 4.2E\text{-}02x + 24.8$
CASE3	Nighttime operation	24 hours	0:00	10:00	Heating season (H)	156.5	$y = 2.6E\text{-}15x^6 - 1.5E\text{-}11x^5 + 1.9E\text{-}08x^4 - 8.9E\text{-}06x^3 + 1.7E\text{-}03x^2 - 9.8E\text{-}02x + 19.3$
					Cooling season (C)	151.5	$y = 3.8E\text{-}15x^6 - 1.7E\text{-}11x^5 + 2.0E\text{-}08x^4 - 9.2E\text{-}06x^3 + 1.8E\text{-}03x^2 - 1.0E\text{-}01x + 23.1$

VERIFICATION OF ENERGY CONSERVATION

Thermal load caused by exterior walls (W) was calculated from [specific gravity * specific heat * flow rates of water * (water inlet temperature - water outlet temperature)]. Thermal load caused by ventilation (V) was calculated from [specific gravity * specific heat * flow rates of fresh air * (room air temperature - outdoor air temperature)]. As thermal load caused by heat gain was considered heat source, approximate hot water temperature setting was very low. However, when the time to operate thermal storage fell on the time to intake fresh air, approximate hot water temperature setting had not arrived at the expected line even if hot water was heated with equipment of heat source(Figure 4 a), c)). Thermal load caused by exterior walls on January 17 (Monday) became larger than that on other days, so it was confirmed that suspending operations on Saturday and Sunday and continuous operation on weekdays were not reasonable. Therefore, it seemed that it is desirable to control by continuous operation on Saturday and Sunday for minimization of equipment capacity of heat source if it is possible to control by continuous operation on weekdays (Figure 4 c)).

On the other hand, nighttime operation obtained a good balance of heat supply between the time to operate thermal storage and the time to intake fresh air. So it was very effective for the problem of stopping operations on Saturday and Sunday (Figure 4 e), f)). Moreover, concerning the relationship between thermal load caused by exterior walls and differences between outdoor and indoor air temperatures, both showed a positive correlation. However, as outdoor air temperature was too cold in the heating season, thermal load caused by exterior walls on Monday increased about 2 times more than those on other days as a result of suspending operations on Saturday and Sunday (Figure 5). Although energy consumption for heating and cooling throughout the year for 24-hour continuous operation was about 5 % larger than that for 10-hour intermittent operation, it must be considered to have minimized equipment capacity of heat source in CASE2 (Figure 6).



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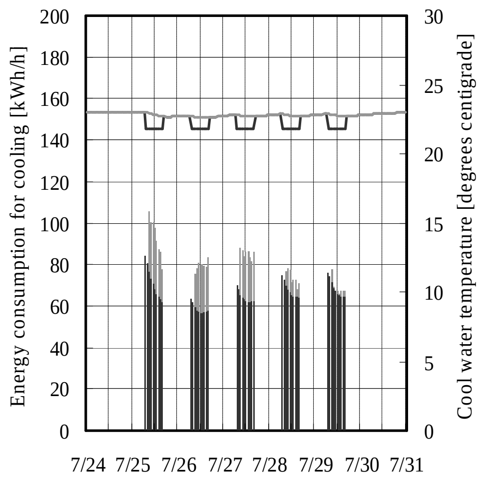

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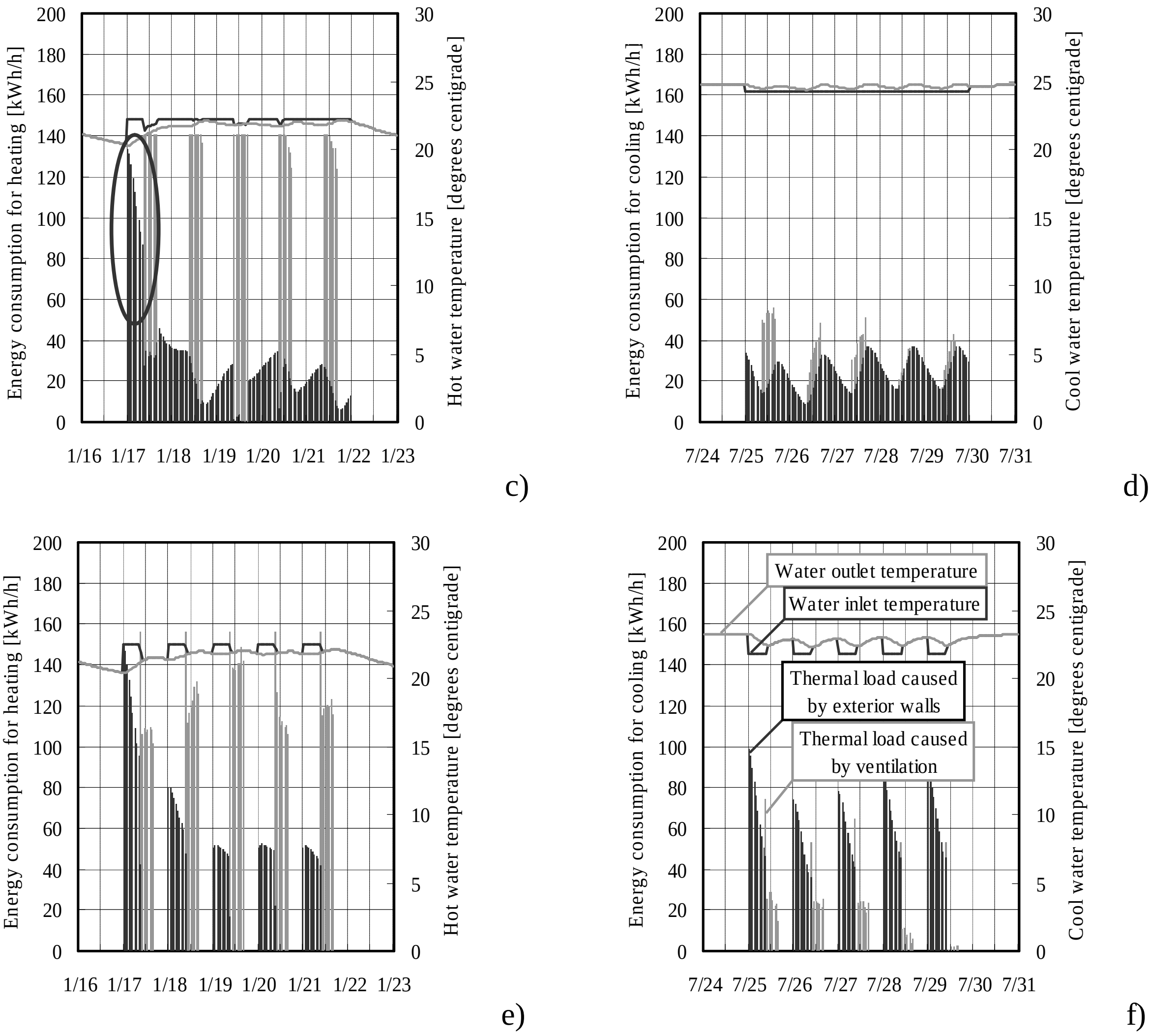


Figure 4. Energy consumption and water temperature

a) CASE1 (H), b) CASE1 (C), c) CASE2 (H), d) CASE2 (C), e) CASE3 (H), f) CASE3 (C)

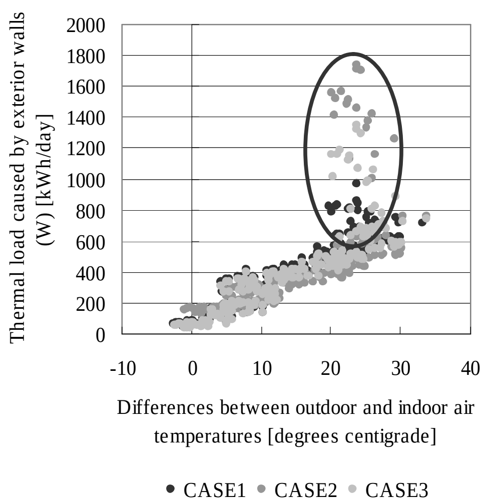


Figure 5. Relationship between thermal load caused by exterior walls (W) and differences between outdoor and indoor air temperatures

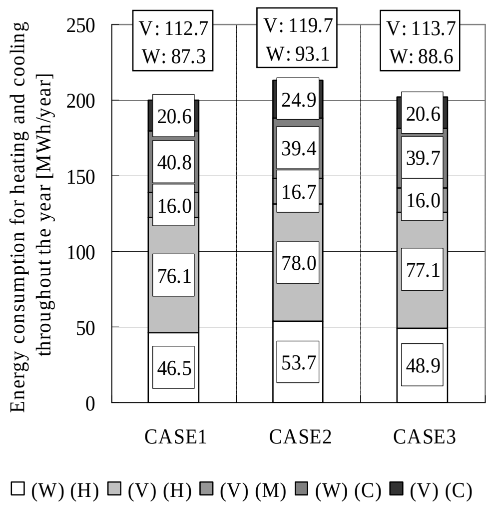


Figure 6. Energy consumption for heating and cooling throughout the year

VERIFICATION OF THERMAL COMFORT

In the case of approximate water temperature setting, it was possible that the amount of heat required for a day was surplus or short. Yet, if room air temperature fluctuations could be accepted in the range in which office workers feel comfortable, this control technique may be positioned as a simplified predictive control premised on the permissible range of room temperature fluctuations in highly insulated office buildings. As a result, room air temperature (R) and floor surface temperature (F) during office hours were mostly stable, and the range of room air temperature swings settled at about 1.2 [degrees centigrade] (Figure 7, Figure 8).

Thermal comfort in CASE1 became PPD > 10% (PMV < -0.5) in only the heating season, which was affected by the problem of suspending operations on Saturday and Sunday (Figure 9 a)). In contrast, thermal comfort in CASE2 became PPD > 10% (PMV > +0.5) in only the cooling season, which was affected by the problem of overheating room air temperature in the daytime (Figure 9 a)). When room air temperature surpassed 26.5 [degrees centigrade], PPD increased rapidly, and the maximum PPD reached more than 20% (Figure 9 b)). Thermal comfort in CASE3 was very effective because PPD was not more than the recommended value even if we admit the permissible range of room temperature fluctuations (Figure 9 c)).

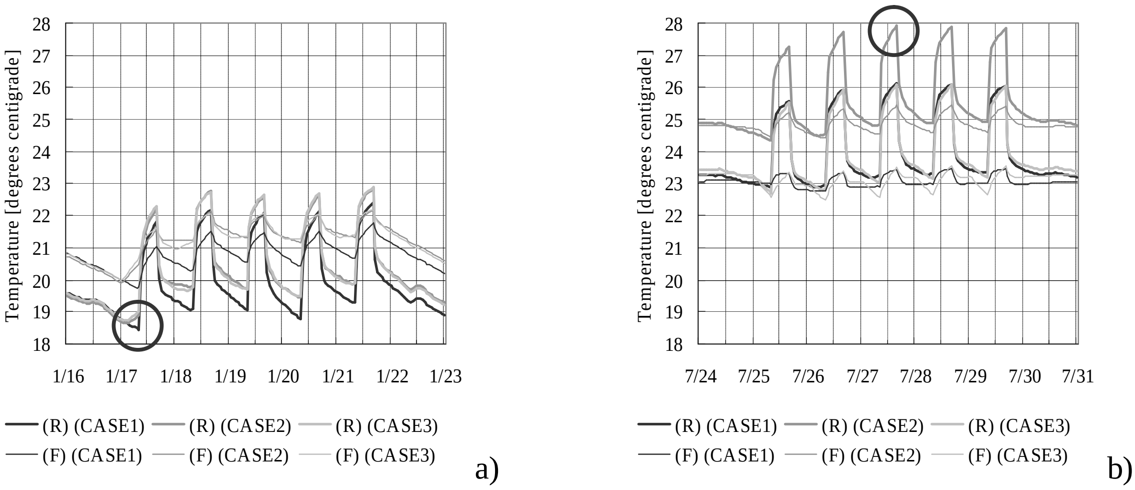


Figure 7. Room air temperature and floor surface temperature

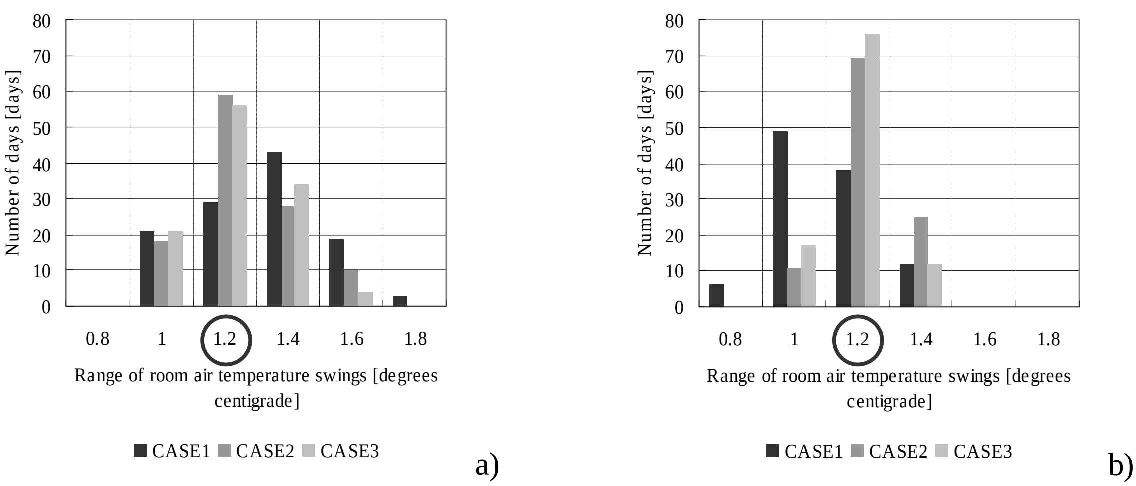
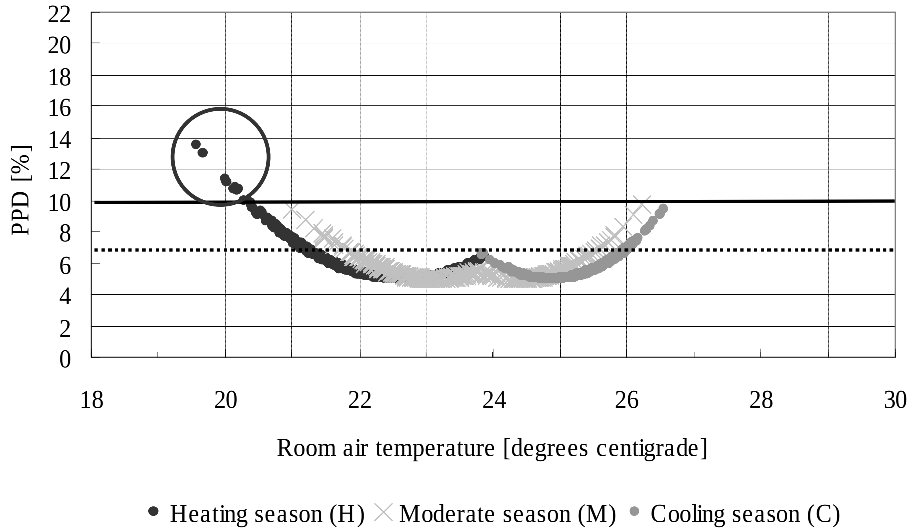


Figure 8. Frequency distribution of range of room air temperature swings
a) Heating season (H), b) Cooling season (C)



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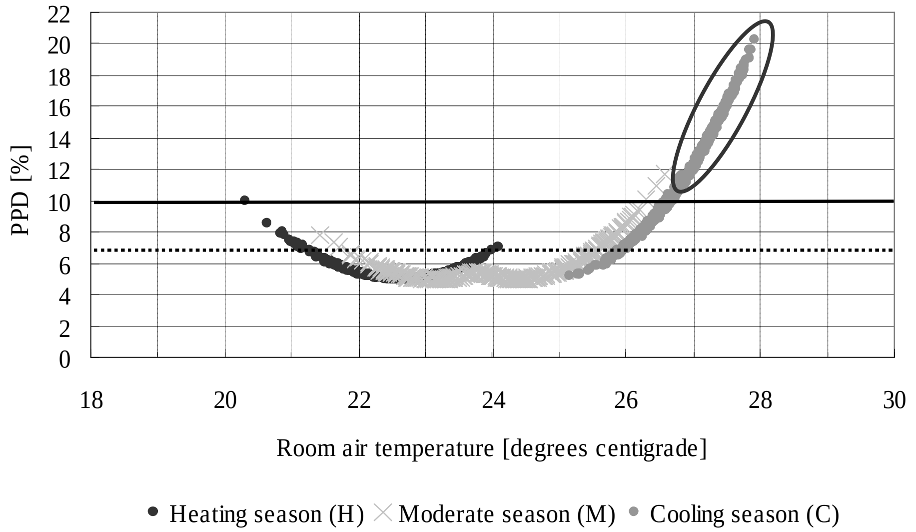

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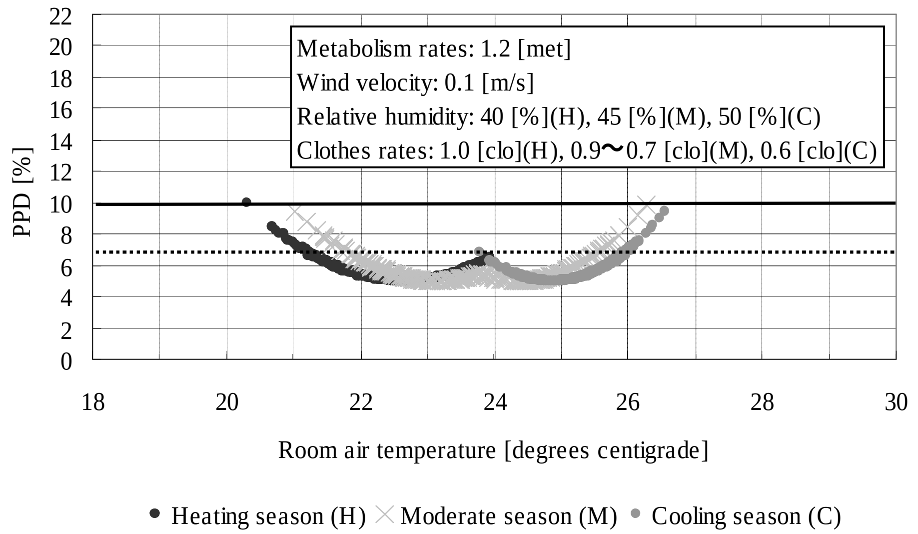

c)

Figure 9. Relationship between room air temperature and PPD
a) CASE1, b) CASE2, c) CASE3

CONCLUSIONS

In this research, we suggested thermal storage heating and cooling systems by a simplified predictive control for highly insulated office buildings and verified the usefulness of this control technique on the basis of energy conservation and thermal comfort in offices. Principally, we considered the usefulness of continuous and nighttime operations in these systems. The results of numerical simulations were as follows:

1. The calculation procedures of a simplified predictive control were shown, and when the time to operate thermal storage fell on the time to intake fresh air during office hours, approximate hot water temperature setting had not arrived at the expected line.

2. Suspending operations on Saturday and Sunday and continuous operation on weekdays were not reasonable in the heating season because thermal load caused by exterior walls on Monday increased about 2 times more than those on other days.

3. Energy consumption for heating and cooling throughout the year for 24-hour continuous operation was about 5 %, for example about 4 [kWh/m^2 year], larger than those for 10-hour intermittent operation. However, it is possible to introduce continuous operation, though operation methods should be improved.

4. From the viewpoints of energy conservation and thermal comfort, it was very effective to supply heat continuously, such as continuous operation or nighttime operation, in the heating season, and to supply heat intermittently, such as intermittent operation or nighttime operation, in the cooling season.

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Outside the Box- HVAC Management Techniques That Consider the World Around Them

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SUMMARY

This paper summarises CSIRO's work on intelligent HVAC management techniques, with a particular focus on techniques designed to participate in demand management programmes. Such programmes are of significant interest, seen as a way of reducing expenditure and minimizing energy consumption in electricity systems around the world. CSIRO's aim is to design intelligent HVAC controllers that can enable automated participation in energy demand management programmes, benefiting both the local system owner as well as the electricity business, whilst still considering factors such as occupant comfort. This paper describes our investigations into the application of machine learning techniques towards such a controller. It also summarises our recent experiments trialling demand management techniques (through setpoint variation) in a multi-story building HVAC deployment.

INTRODUCTION

As concepts such as carbon trading, electricity demand management and renewable energy integration become a significant consideration for modern society, there is an increasing focus on more intelligent control of heating, ventilation and air-conditioning (HVAC) plant. At all scales of the HVAC industry, from relatively small residential air-conditioning systems, to large commercial HVAC plant, there is a growing awareness of a need for intelligent, integrated HVAC plant that considers the world around it.

In Australia, a number of recent examples highlight the growing external pressures on the designers, owners and operators of HVAC equipment:

- Air-conditioning systems are being blamed in popular media as one of the causes of demand-related electricity network failures across parts of Australia [1].
- Programmes are being introduced by electricity retailers that are specifically designed to control or limit the energy consumption of air-conditioning units at certain times of the day[1].
- Recognising issues such as those above, standardisation work has already started trying to standardise the interfaces used to control air-conditioning systems [2].

Further, a recent new approach to the operation and management of electricity networks brings even greater challenges, and also opportunities, to HVAC organisations. Part of a wider concept referred to as *distributed energy* (DE), the recent growth in *demand-side* consideration of electricity networks brings a lot of attention towards intelligent load control ~~(or *demand management*), wit~~h local consumer consideration of new factors such as

[1] There are many examples of such programmes, however at this stage they can generally be grouped into two categories- programmes designed to control energy consumption by price signals [3], or programmes that use a physical control technology to directly manage air-conditioner consumption, away from the consumer [4].

electricity network operating condition, electricity unit price, availability of local electricity generation, and short-range weather predictions, as well as more common factors such as operating cost and maintenance issues.

In general, the reason demand-side technologies are of significant commercial, research and regulatory interest is that they are seen to have a number of benefits to electricity systems worldwide. Trials in this area conducted in Europe and the U.S.A have been shown to assist in demand-supply constraint situations, reduce energy consumption, help manage electricity retailer's commercial risks, and assist in the infrastructure investment decisions of electricity network operators. Reflecting such benefits, the U.S.A has recently mandated that distributed energy approaches be encouraged throughout their network [5], other studies have suggested DE type methods are some of the most cost-effective ways to meet peak load growth [6], and new distributed-energy related businesses are beginning worldwide.

Whilst benefits such as increased network stability and reduced financial risk help electricity retailers and network operators, we believe there are two fundamental reasons why demand management should be of interest to HVAC companies:

- Demand-side focussed programmes generally require better HVAC load control, requiring greater sophistication in HVAC operation and control strategies.
- Often, demand management programmes will include a *user-paid* component, where HVAC operators are financially rewarded for taking a particular management action with their plant, thereby facilitating additional income streams for operators of large HVAC plant.

CSIRO's work in this area takes advanced concepts from the artificial intelligence and computer science domains, and applies them to the intelligent control of HVAC plant. Along with optimized end-use efficiency benefits, such technology enables automated participation in energy demand management programmes, improved occupant comfort consideration, and more flexible control of HVAC operating expenditure.

This paper details our work in this area. We first discuss the recent growth in demand management related work, and the implications of this for the HVAC industry. We then detail the technical aspects of our work in this area, discussing CSIRO's technology, and the advanced modelling techniques we are researching for improved HVAC control. Lastly, we consider some results from a recent real-world trial of these types of strategies in a large office building, and discuss the sometimes surprising outcomes resulting from such a trial.

DEMAND MANAGEMENT

Recently, there has been a significant amount of media attention focussing on what is a perceived electricity supply problem in parts of Australia, and indeed many other parts of the world. In short, with aging network infrastructure, a growing peak electricity demand from loads such as air-conditioners, and ever increasing base-load energy consumption, electricity generation and distribution systems are being seen as unable to cope with the demand placed on them. Whilst the traditional solution to such issues has been to build more supply infrastructure, there is an increasing interest in solving these problems from the demand side. For example, consider the peak-load growth issue - addressing this through supply-side augmentation is an incredibly inefficient approach - whilst peak loads can be over double the average base load on a network, they often occur for only a few days per year.

In addition to supply and distribution issues, countries like Australia trade electricity in a relatively volatile energy market, where the price for a unit of electricity can vary from $10 per megawatt-hour, to $10 000 per megawatt-hour, in the space of one day. Such price spikes can represent over 20% of the turnover of our electricity market, but are generally less than 1% of the energy traded.

These relative inefficiencies in our electricity supply systems and economic inelasticities in electricity trading markets can be at least partly addressed by the demand-side solutions embodied in the distributed energy concept- essentially, being selective in when energy is consumed, so as to minimise electricity peaks, and manage financial risks. For the end user, this essentially means that an electricity utility would ask the user to be time-selective in when they consume electricity; assisting the utility in their network operation, and perhaps even helping them deal with unexpected occurrences. Importantly, in using demand-side solutions to address the issues discussed above, we expect the organisations that will benefit most from such solutions (electricity distributors and retailers) to pass such financial benefits on to the end-user.

Considering HVAC systems, demand side benefits are available from more optimal *load* control- fundamentally, in an HVAC system it is possible to reduce electrical consumption by trading off human comfort against the cost of operation of the HVAC plant. Alternatively, it may be possible to simply *shift* electrical consumption, preserving human comfort, by pre-cooling or pre-heating the offices in advance of an electricity price spike, and then allowing the office temperature to drift gradually during a high price period, thereby minimising electricity consumption during expensive periods.

Peak-reducing technologies are not entirely new to HVAC systems. An alternate way to manage peak energy consumption by HVAC plant is to charge ice-tanks in the building during off-peak times, and use the stored thermal energy of these tanks to cool the building during peak times [7]. Whilst effective, these techniques rely on relatively expensive additional plant, and are not well-suited to responding to dynamic, rapidly occurring network contingencies. Existing peak-management techniques that can respond dynamically to changing electricity prices generally rely on a simple load-shed table within a building management system (BMS), and are relatively unsophisticated, removing particular electrical loads once certain thresholds are crossed.

For optimal performance, we believe the HVAC management system should be dynamic and intelligent, responding to changing events, and considering a variety of external factors such as occupancy, human comfort, electricity price, and weather forecast. Further, rather than relying on additional thermal storage plant such as ice-tanks, our system pre-cools or pre-warms a building by taking advantage of the thermal storage inherent to the building fabric itself. Similar attempts to optimise the operating cost of air-conditioning equipment using electricity price signals and the building thermal mass have been attempted previously in at least one other study we are aware of [8]. However, this study used fixed temperature setpoints and the cost function did not consider occupant comfort. Furthermore, it was only possible to minimise either peak loading, electricity cost or electricity consumption but not a weighted combination of these.

An HVAC management system that accounts for all these factors needs to be quite sophisticated- it needs to be able to predict the consequences of a particular action, and decide a long-term HVAC management strategy. Our work in this area is discussed in the following sections.

HVAC PERFORMANCE MODELLING

In evaluating the cost of a particular HVAC control action, it must be determined how many people may be dissatisfied by a particular action, or, how a particular action will affect human comfort. We use the well-cited human comfort metric from Fanger [9] to evaluate how a particular HVAC action will affect human comfort, based on factors such as air temperature, humidity and air velocity.

Clearly, whilst techniques such as Fanger's predict human comfort for various environmental parameters, we need to also obtain a prediction of how these environmental parameters will be affected by a particular HVAC demand management action. Fundamentally, a thermal model of the building is needed. This model should provide some basic time constants of the building- the time to heat up or cool down, with the HVAC plant on or off.

Whilst there is a massive amount of literature available providing thermal modelling techniques for modern office buildings, we have taken a slightly different approach. Wanting our HVAC management technique to be as flexible as possible, we have taken some advanced techniques from the field of *machine learning* to automatically form a basic thermal model of the building, by observing basic building thermal behaviour over time. This field includes the use of techniques such as Neural Networks (NNs) [10] and Support Vector Machines [11], and these have been applied in such diverse fields as renewable power forecasting, financial market prediction, lightning strike forecasting, and fault tolerant robotics.

In this work we have chosen to use Support Vector Machines (SVMs) to learn the features of the thermal model of a room. Essentially, Support Vector Machines can be thought of as a more general version of Artificial Neural Networks. Like Neural Networks, SVMs are a supervised learning technique that can be used for classification or regression. They are "trained" on a particular set of data, this training being the basis for predictions of future data. Ideally these training samples will encompass all possible states of the system, but in reality, this is rarely possible, so the primary aim of supervised learning methods is to build a model that can *generalise* well, that is, it can produce a good model of the system from only a small number of training samples. SVMs take advantage of probability and statistical learning theory to maximise the generalisation ability of their fitted model, and often perform very well with noisy or sparse data sets – an advantage when used for quickly fitting the unfiltered real-time data our system supplies [11].

To simplify the analysis in this early stage of our work here, we have tested the our approach using a cool room, rather than an actual building. This strategy allows us to isolate the thermal behaviour of the room to a relatively small number of inputs and outputs, simplifying the design and analysis of the modelling techniques chosen. The cool room controller system with which our SVM model is used collects several heating/cooling cycles worth of training data, and then fits a model to these samples, essentially finding a characteristic system temperature curve. This model can then be time-stepped into the future, providing accurate temperature predictions of the system.

This behaviour can be seen in figure 1, which shows the training samples, and the predicted and actual temperatures for a period of cool room operation. It is interesting to observe the

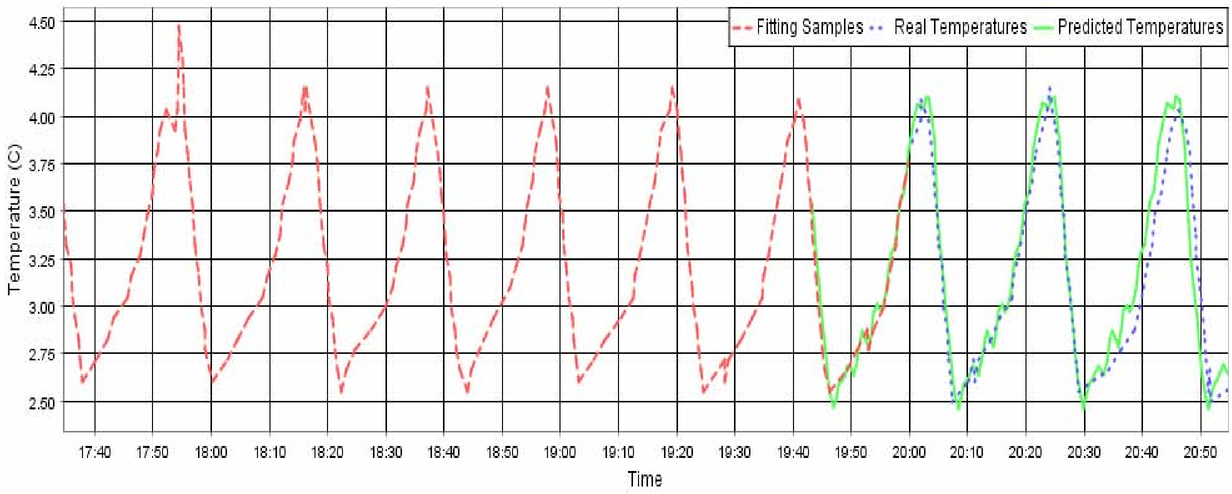

Figure 1: Operation of machine learning technique for cool room thermal model

behaviour of the system at time 17.54, when the cool room door being opened causes a spike in the internal temperature. The learning model was able to identify this sample as having a minor effect, and so the contribution of this training data to the predictive model is minimal. This is a key advantage of SVMs- their ability to intelligently filter outlying data points, and model non-obvious system subtleties like overshoot and leakage which affect the fitting samples.

Whilst these experiments have been run on a relatively simple thermal scenario- the thermal response of a cool room, we believe these results are valid across a variety of scenarios, and are now conducting almost identical modelling experiments in a true office-building setting. We have yet to investigate the performance of the modelling aspects of our work in this office building- the initial aspects of our trial have focussed on the more general impacts demand management approaches such as pre-cooling might have on a building. The results from this trial are very interesting, and sometimes surprising- they are outlined in the following section.

EXPERIMENTAL EXPERIENCES

In order to gain a quantitative assessment of the potential of intelligent demand management techniques, we conducted a variety of demand management experiments on an eleven storey commercial office block in Melbourne, Australia. The experiments were performed during summer 2006/2007 and were specifically designed to assess:

- The effectiveness of a building pre-cool prior to a DSM load shed event; and
- The load reduction, profile and duration achievable using direct adjustment of the building thermostat setpoint.

Experimental Setup

The experimental site is a standard office building comprising eleven floors and a total floor area of around 10555m^2. The HVAC system comprises two chillers (a screw compressor and centripetal compressor with capacities 780 and 600 kW thermal respectively), two boilers (each around 700kW thermal) and eleven air handling units – one on each level. In addition to the main HVAC system, there is a supplementary unit that provides additional cooling on levels 6 and 9 where call centres operate.

For this trial, the demand response was obtained by directly modifying the temperature setpoints of all zones throughout the building. The building automation system was upgraded to allow this to be performed remotely and in response to external stimuli – for example a price peak in the Australian national electricity market. Trials were particularly targeted at hot days (around 35°C) during which times the electricity network is stressed and hence representative of the conditions under which a demand response is typically requested.

In order to allow a full analysis of the building response to the demand reduction trials, additional instrumentation and data logging facilities were added to the building. A full measurement set consisting of:

- all building zone temperatures, around six per level and 68 in total;
- ambient temperature conditions;
- current consumption of each individual chiller;
- current, voltage and power of the total building HVAC system; and
- current, voltage and power of the supplementary systems on level 6 & 9

was recorded every five minutes throughout the summer. This extensive data set has allowed an assessment of both the demand reduction trials and baseline energy consumption as a function of setpoint temperature and ambient conditions.

Experimental Results

A variety of experiments were conducted throughout the trial. Focusing on one particular experiment, which was representative of the others, in figure 2 results can be seen for a 'load reduction with pre-cool' experiment. The figure shows the building setpoint temperature, two zone temperatures and the total HVAC power consumption. This experiment was conducted on Jan 3rd, 2007, which had a peak ambient temperature of 32.8°C and consisted of:

- Starting the HVAC system at 8am with a 22.5°C setpoint (this is normal);
- 2 hour building pre-cool from 11:45am to 1:45pm with 21°C setpoint;
- 2.5hr demand reduction event from 1:45pm to 4:15pm with 24°C setpoint; and
- Reverting to the normal 22.5°C setpoint until the system shut down at 6pm.

In this case, there was a peak load reduction of 35%, and average load reduction of 15%, and a total energy saving of 127kWh.

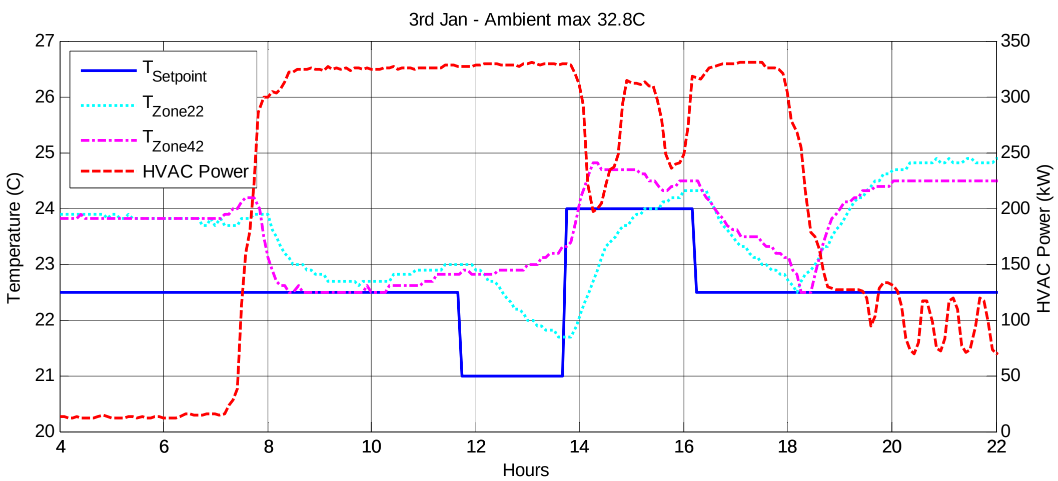

Figure 2: Typical results for a demand response load reduction with pre-cool.

A key observation from these results is that the pre-cool is ineffective when the chillers are already running at full capacity prior to the pre-cool - as is likely to often occur on hot days when load reductions are required. In this case, the pre-cool only serves to redistribute the cooling between zones, providing additional cooling to some, while consequently taking it away from others and resulting in some zone temperature *increases*. This effect can be seen with the two zone temperatures (T_{Zone22} and T_{Zone42}) in figure 2. Clearly, any "intelligent" HVAC management system will need to take into account such behaviours, providing the mechanism for an effective pre-cool that is unachievable with the simple setpoint adjustment used in this experiment.

Other results learnt from this deployment include:

- Following a setpoint change, there is a significant time delay before there is an appreciable change in the building energy consumption. Since the chiller is controlled to maintain a chilled water setpoint, a power reduction does not occur until the zone temperature setpoint change causes the chilled water temperature to drop. In the case of a constrained system, the chilled water temperature can be operating significantly above setpoint and in the case shown in figure 2, it took around 30 minutes from zone setpoint change to peak power reduction.
- It is imperative that a building control scheme is tolerant to fault in sensors/remote units. During the trial, there were typically 5% of sensors providing erroneous data, rising to 10% at times.
- Building power consumption can not be adequately assessed based on aggregate temperature data – it is important that data be intelligently integrated in order to successfully predict behaviour. As an example, in figure 2, the chillers start to pick up load at around 2:30pm specifically to meet the cooling demands of the zones whose temperatures increased during the pre-cool. This behaviour could not have been predicted based on aggregate temperature data.

CONCLUSION

Electricity industries around the world are undergoing significant changes that will have equally significant effects on the end user. In particular, the growing number of demand management initiatives will be of great interest to the owners and operators of large HVAC plant- such initiatives will have important technical *and* financial implications.

Realising this, CSIRO has initiated a significant research programme into intelligent HVAC control systems. Importantly, whilst striving to improve the operation of the HVAC system operating in isolation, we are also designing our strategies to be able to include consideration of the world around them, and thereby participate in demand management programmes.

Our first steps in this area focus on being able to model the thermal performance of a building, and thereby predict the affect of demand management actions on the building and its occupants. Such a prediction is critically important, to enable an HVAC operator to determine the cost and the benefits of a particular operation strategy. Our work in this area differs from traditional modelling techniques, in that we are investigating the performance from so-called "black box" machine learning systems, to form a suitably accurate model of the building, but in a relatively simple, automatic, and financially economical manner.

Whilst the basic principles behind demand management in an HVAC system appear relatively simple- vary the environmental setpoint strategy to change a building's energy consumption, the real-world consequences of such a strategy can be somewhat surprising. The latter part of this paper investigated the real-world consequences of a demand management programme based on peak shifting in a typical office building. Some of the observations made in this trial, such as the varying effects of a changing setpoint on certain zones of a building, were not expected, and need to be kept in mind for any demand management programme under consideration.

Our work in this area is continuing- we continue to enhance our HVAC modelling work, we are experimenting with new hardware platforms for increased granularity data gathering in an HVAC deployment, and we are working on HVAC management strategies that take advantage of this new data. Lastly, we are also looking at the benefits possible through integrating an HVAC system with other loads and generators on site- effectively obtaining a site-wide energy response that can then interact with the demand management programmes mentioned in this paper.

ACKNOWLEDGEMENT

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Impact of automation concepts for better performance and monitoring of sustainable energy systems

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SUMMARY

The integration of sustainable energy systems (e.g. earth-to-air heat exchanger (EAHX), solar air collector (SAC) into conventional energy systems is of increasing interest. When more than one of these systems are used simultaneously (e.g. an EAHX and an air handling unit (AHU) with heat recovery) improved automation concepts can help to use environmental energy more efficiently. Consequently, there is also a need of using simulation models not only for building design but also for energy optimisation and management during building operation.

At Biberach University of Applied Sciences a classroom and laboratory building is equipped with different energy systems like an air-handling unit with heat recovery that provides one classroom as a test room. For research and development issues, the external air can be taken directly from the outside or can be preconditioned using an EAHX or a SAC panel. The air can then supply the test room with or without being additionally conditioned by recovering heat from the exhaust air within the air handling unit's cross flow air-to-air heat exchanger. Finally, there is a water-to-air heat exchanger for heating or cooling the air when required.

An advanced automation system controls these facilities with respect to energy efficient operation by taking into account actual weather data or weather forecasting data. The control algorithm applied uses characteristic diagrams which consider energy gains of the EAHX (depending on earth and outside temperature and air mass flow), energy gains of the solar air collector (depending on outside temperature, solar radiation and air mass flow) and fan energy demands (depending on used system and air flow rate).

In a first step the characteristic diagrams were generated by applying simulation models of every energy system to the whole range of relevant conditions. In a next step the diagrams will be adjusted with measured and monitored data, which were collected and achieved during past system operation.

In this paper the corresponding model of an EAHX system is described as an example, and a comparison of simulated and experimentally gained results is discussed.

The impact of improved automation concepts based on such simulation models and measured data during building operation is also described.

INTRODUCTION

The aim of this study is the development of an advanced automation concept for an energy optimised supply of a room with conditioned ambient air by selecting between three options of air preconditioning. These options are: using unconditioned ambient air, preheating of the external air using a solar air collector (SAC), and preheating or precooling the external air using an earth-to-air heat exchanger (EAHX).

At the Biberach University of Applied Sciences a new classroom and laboratory building – “Technikum G” – was built. The building itself serves as a “laboratory-in-a-building” containing a variety of novel facility systems for testing and research purposes. One of these systems is an air handling unit, (Fig. 1), that includes the above-named options for receiving outside air. A detailed description of the building, its energy system and equipment can be found in 1, 2 and 3.

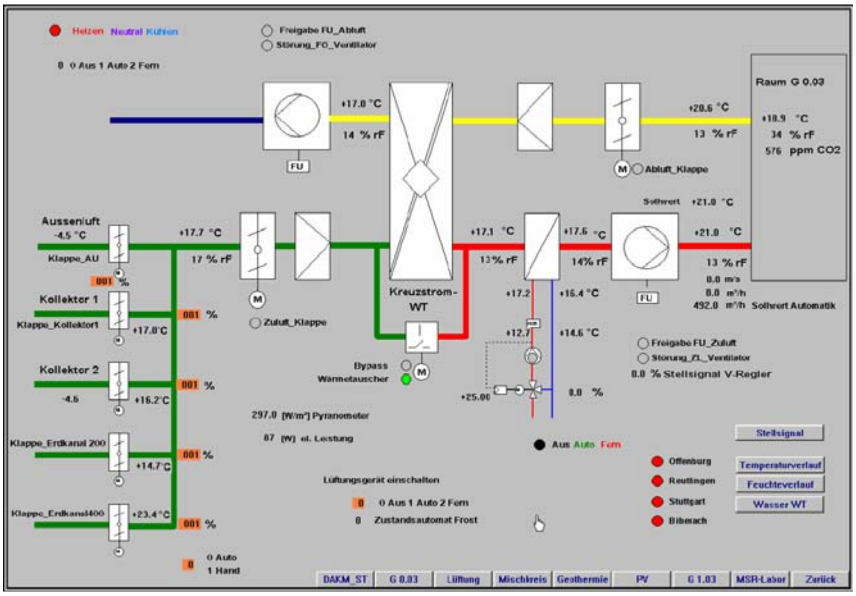


Figure 1. Visualisation of the air handling unit (AHU) of “Technikum G”.

A development environment was built up, using “Technikum-G” as a test facility for validating simulation models, as well as part of this environment. Based on this development environment new control strategies can be developed and tested. There are two possibilities to realise such a control strategy. On the one hand, the decision which air supply is used can be found in an online-simulation, calculating the most energy-efficient way. On the other hand the energy efficiency of each method (by given boundary conditions) can be taken from previous created characteristic diagrams. This paper refers to the latter possibility.

METHODS

To develop and validate overall control strategies for building automation and management an open and flexible development environment is needed. The approach here is to set up this development environment in three levels (Fig.2):

a) **Process with basic control strategies:** the process is represented by the air handling unit (AHU) and the test room G0.03 with standard controllers (e.g. PI controller)
b) **Overall control strategies:** the overall building management system, including the possibility of implementing adaption and optimisation strategies using the development level
c) **Development:** a collection of simulation models and tools to emulate and monitor the real process

All this parts are designed and implemented in a flexible development environment at Biberach University of Applied Sciences. Each level can be coupled to or decoupled from each other level. All parts are in a temporary or permanent data exchange process (offline/online).

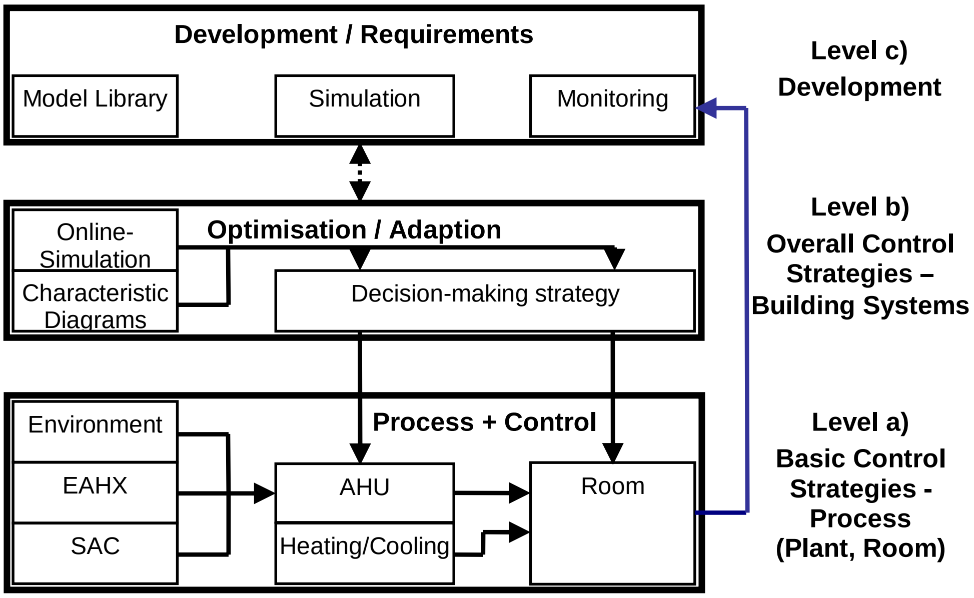


Figure 2. Development environment in three levels.

New overall control algorithms are developed in the overall control strategy level b) in connection with the development level c).
In this context simulation can play different roles:

First, simulation can be used for direct online simulation. In this case simulation runs online parallel to the real process. As a result of the online simulation the decision-making strategy optimises e.g. control parameters of a basic controller (e.g. PI-controller). Therefore a permanent monitoring of the process (plant, room) is necessary.
Second, simulation can be used for building up characteristic diagrams offline. With the help of this characteristic diagrams it is possible to adapt or optimise the basic control strategies in level a) online during operation.
Third, simulation models can be used during design to simulate the process of itself to design and test advanced control strategies in a simulation environment.

Example for Simulation Environment of Process and Overall Control Strategy

Fig. 3 shows an example of implementing level a) and b). In this example the task was to design and implement a damper controller to decide, based on energy criteria, whether to activate the dampers for using direct ambient air, the solar air collector (SAC) or the earth-to-air heat exchanger (EAHX) for preheating or precooling air for the air handling unit (AHU), based on the actual weather data (temperature, solar radiation).
As mentioned before there are two possibilities to realise the control strategy: First, the decision which air supply is used can be found by calculating the most energy-efficient way in an online-simulation. Second, the decision for the most energy efficient condition can be taken from previous generated characteristic diagrams. This paper describes a decision maker based on characteristic diagrams generated by offline simulation of the different sub-systems. This will be discussed later.
To develop an energy management strategy for an energy optimised supply with ambient air, simulation models of the process were built up which includes also all parts oft the air-handling-unit and the classroom G 0.03. In this application the simulation models are used as simulation environment of the process instead of the real process of itself.

Advantage of this approach is the possibility to simulate all interesting scenarios in a short time and under repeatable conditions. However a lower accuracy has to be accepted, as the process can be represented only in a less exact way. Therefore a deviation to measured data exists.

In further work the simulation model will be replaced by the real process and the energy management (as software) controls the dampers and the AHU in "Technikum G". Also the data of the characteristic diagrams – up to now gained by simulation – can be replaced by measured data.

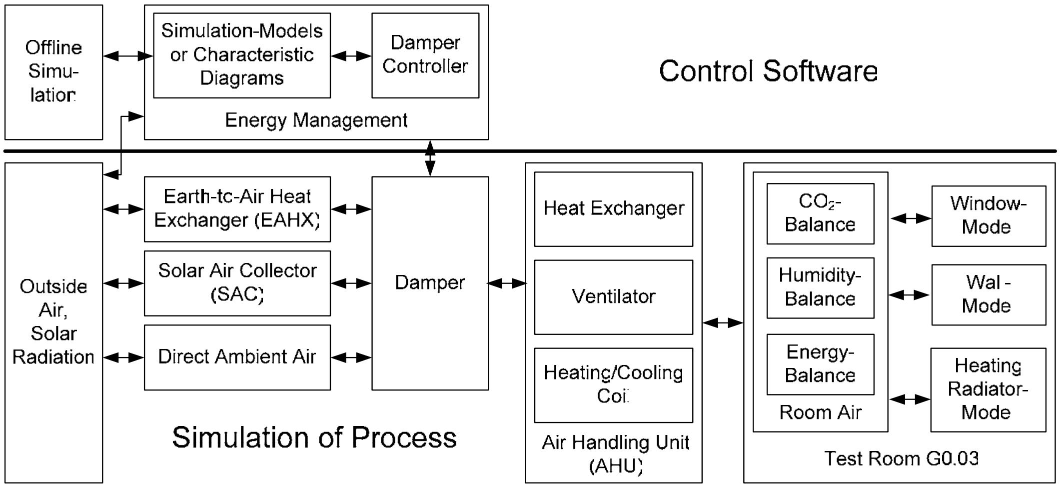


Figure 3. Simulation environment for damper controller development.

The simulation models of the process differ in level of detail. Room, solar air collector and earth-to-air heat exchanger are modelled with detailed physical properties. Heat exchanger, heating/cooling coil and fan of the air handling unit are represented by simple mathematical correlations. If necessary these sub-models will be developed in further work in more detail.

In the following, the sub-system earth-to-air heat exchanger will be discussed in more detail. A description of build-up and evaluation of the solar air collector model can be found in 2, the room model is discussed in 3.

RESULTS OF EARTH-TO-AIR HEAT EXCHANGER (EAHX) MODELLING

Based on preliminary investigations and testing of alternative modelling approaches, the following model of the EAHX was developed and implemented into the development environment (see Fig. 4):
The resistance-capacity model consists of a single air temperature node and several soil temperature nodes arranged along the radial direction from the heat exchanger pipe's centre. Each soil node represents a concentric cylindrical soil layer with a capacity according to the thermal mass of the corresponding soil volume. Heat exchange between adjacent soil nodes takes place by conduction with the innermost node taking part in the heat transfer through the pipe wall to the air. The outer boundary of the system is the undisturbed ground temperature. The air node represents the logarithmic average air temperature within the EAHX pipe. Compared to the soil, the thermal capacity of the air inside the pipe is negligible. Therefore, its temperature is calculated from a steady state energy balance including air flow through the pipe and heat transfer to the pipe wall.

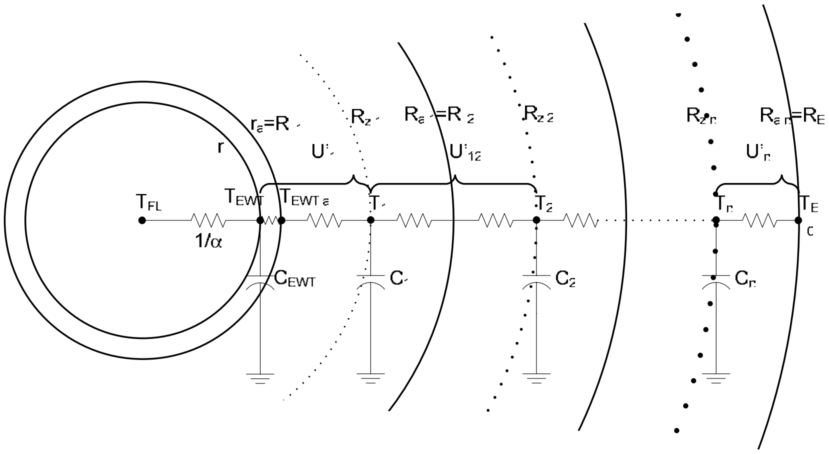


Figure 4. Resistance-capacity model of air-to-earth heat exchanger (EAHX).

During the test of the EAHX model simulations with different parameters were conducted. The results are compared to measured data of the existing EAHX at Biberach University of Applied Sciences. Tab. 1 displays the different parameters used.

Table 1. Different parameters for EAHX simulations.

	Film Coefficient (Air to Pipe)	Consistence of Earth
Test 1	2.90 W/m²K	massive, wet
Test 2	20.0 W/m²K	massive, wet
Test 3	2.90 W/m²K	light, wet

The simulated air outlet temperatures differ about 2 – 4 K from the measured temperatures (Fig. 5). The reason for this deviation is not solely due to the simulation model and its parameter values, but results also from the position of the air outlet temperature sensor which had to be placed within the building at a certain distance from the EAHX outlet.

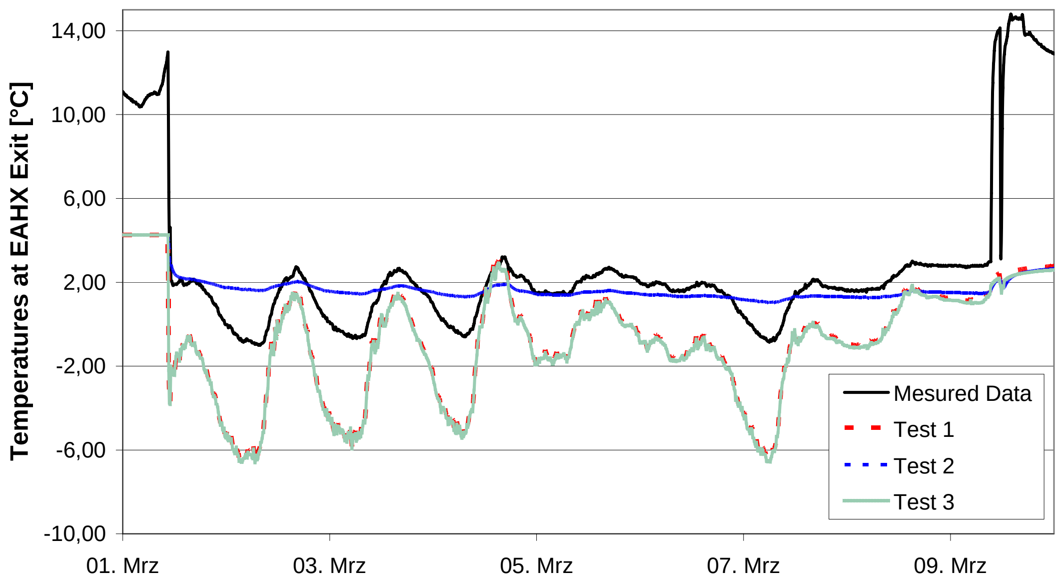


Figure 5. Results of EAHX test simulations and comparison with measurements.

The dynamic behaviour of the EAHX is well reproduced by a chosen film coefficient (air to shell) of 3.5 W/m²K. This value accords to calculations with standard heat transfer coefficient correlations for this particular air flow rate and design. Regarding the aim of developing a control strategy the simulation results are sufficient. Since a control strategy must be able to work properly even when the controlled system deviates from its design point to a certain extend, a moderately inaccurate simulation model is still applicable for testing control strategies.

RESULTS AND DISCUSSION OF THE ENERGY OPTIMISED STRATEGY

To test the abilities of the simulation model as development environment, an energy management strategy including a damper controller was developed. This strategy allows an energy optimised ambient air supply to the AHU by selecting between three options of air preconditioning.
To control the damper the decision criterion is the way to precondition external air with lowest energy consumption. As described before, there are two possibilities to realise the control strategy. The decision which air supply is used can be found by an online-simulation or by using characteristic diagrams. The implemented decision maker (Fig. 6) is composed of the following elementary parts: the characteristic diagrams and the damper controller as a rule based controller, which can be extended by additional features (e.g. weather forecasting, levels of occupancy) in future.

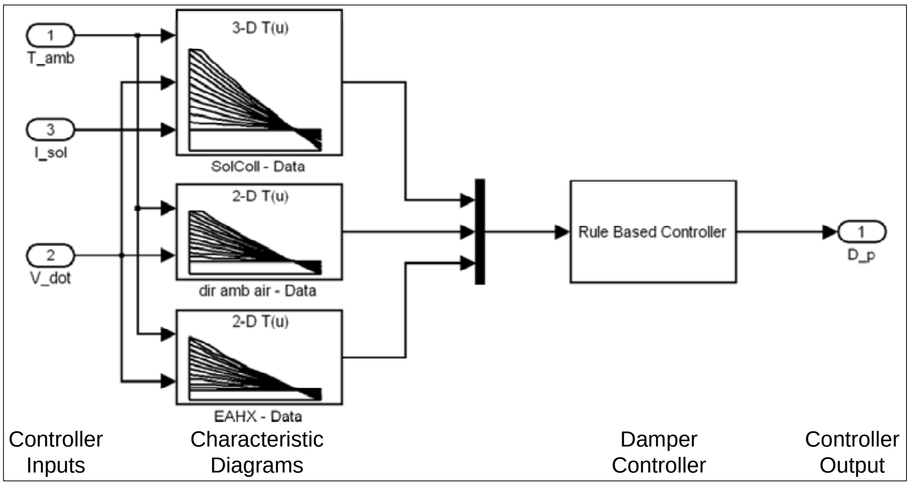
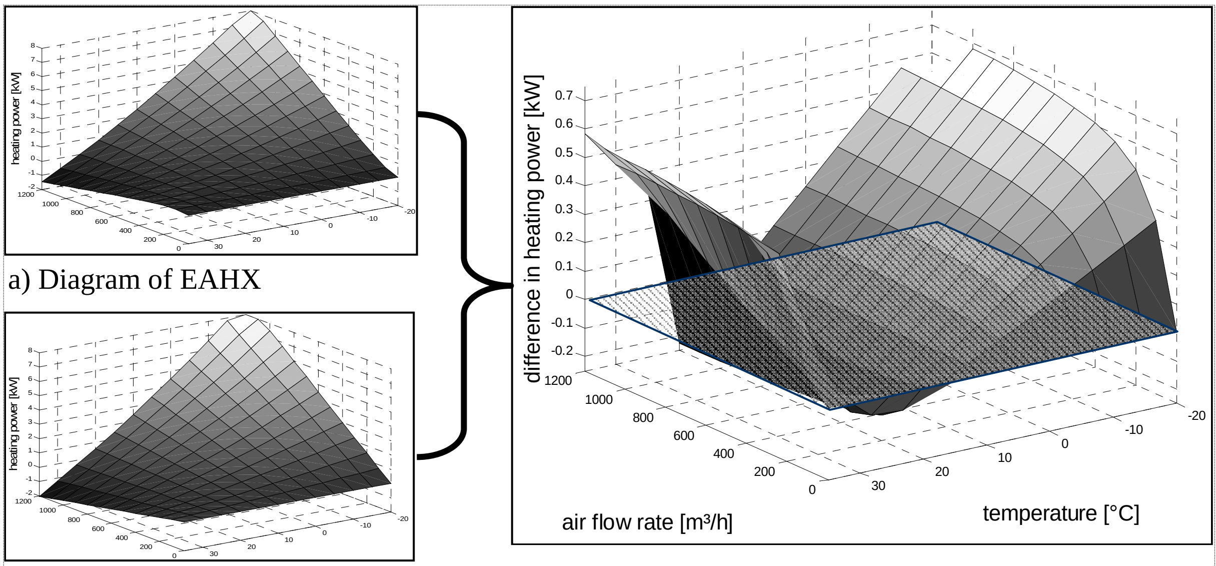


Figure 6. Layout of the energy management block including a damper controller as a rule based controller. (T_amb: ambient temperature, I_sol: incident solar radiation, V_dot: air flow rate, D_p: damper position)

In a first step the characteristic diagrams used within the control system are created by simulation results.
Relation criterion is the heating/cooling demand of the heating/cooling coil to transform the preconditioned air to a supply air of 22°C.
This data were generated as characteristic diagrams for boundary conditions of ambient air from -20 to 35°C (step size 5 K) and air flow rate from 0 to 1200 m^3/h (step size 100 m^3/h) for direct ambient air and EAHX. For the SAC exits a matrix for each of the incident solar radiations on surface (0, 200, 400, 600, 800,1000 W/m^2). Interim values are interpolated during control process.

In Fig. 7 the characteristic diagrams of EAHX (a) and direct ambient air (b) are shown. Also the difference of this values as generated by the decision maker is demonstrated. As result positive values indicate a better efficiency of the EAHX, negative values for direct outside air suction (c).



a) Diagram of EAHX

b) Diagram of direct ambient air

c) Difference of EAHX and direct ambient air

Figure 7. Examples of characteristic diagrams.

The damper controller was also tested within the simulation environment instead of the real process. For first examination this approach is faster than testing the controller directly on the plant. Object of interest is the heating/cooling demand of the heating/cooling coil to transform the preconditioned air to a supply air of 22°C. This is recorded for four cases: uncontrolled air preconditioning using the SAC (1), the EAHX (2) or direct ambient air (3) and controlled air preconditioning using energy management strategy (4). All these tests are simulated with Biberach weather data (first 6 month of 2005, hourly values) and a permanent air flow rate of 500 m³/h.

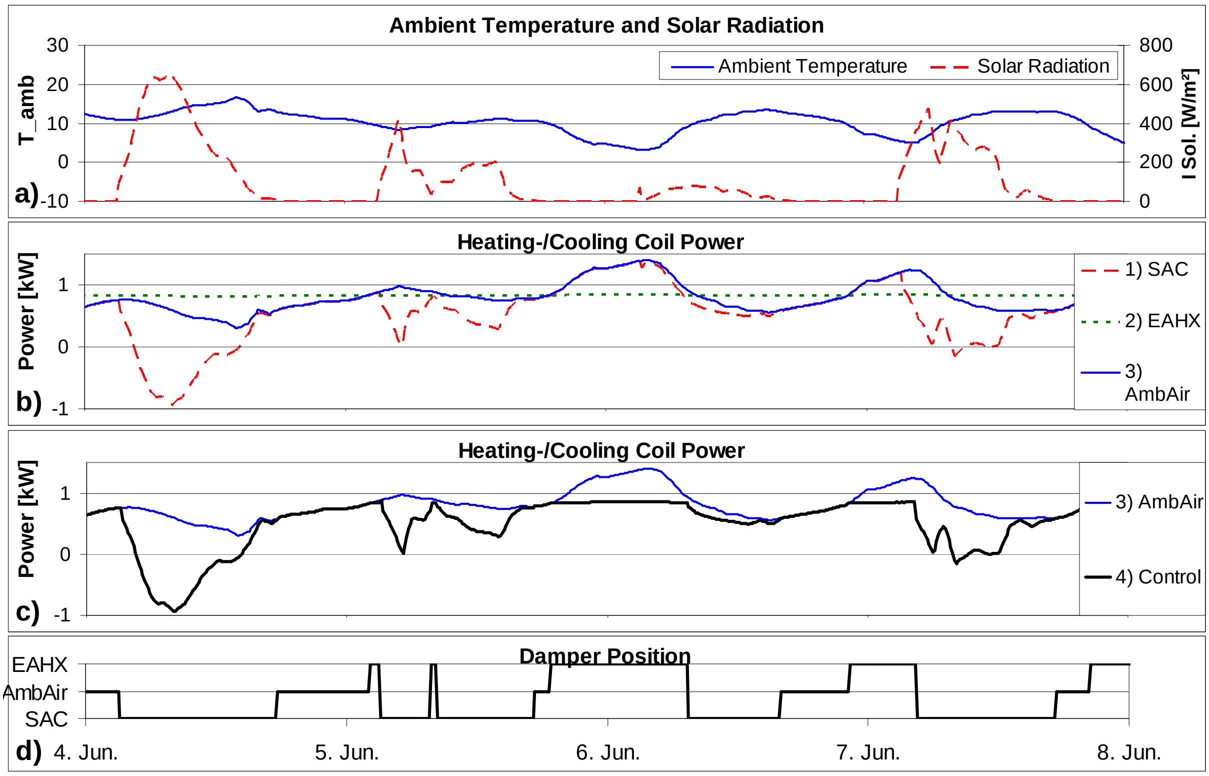


Figure 8. Simulation results without and with energy management strategy.

Fig. 8 shows the simulation results from 4th to 8th June for instance. In Fig. 8a) the boundary conditions measured ambient air temperature and incident solar radiation on SAC surface are given. Fig. 8b) indicates the heating/cooling coil power (negative values means cooling power) for the uncontrolled ventilation strategies. Fig. 8c) shows the heating/cooling coil power of the controlled ventilation strategy and of direct ambient air ventilation in comparison. The corresponding damper positions are shown in Fig. 8d).
The damper controller selects the way of air preconditioning with the lowest energy consumption.

Table 2. Energy demand at heating/cooling coil by different ways of air preconditioning.

	Simulation	Heating [kWh]	Cooling [kWh]	Sum [kWh]
(1)	Solar Air Collector (SAC)	3942	574	4516
(2)	Earth-to-Air Heat Exchanger (EAHX)	3697	0	3697
(3)	Only Ambient Air (AmbAir)	4729	103	4832
(4)	With Energy Management	2423	404	2837

Tab. 2 shows the energy consumption of the heating/cooling coil for the first six month for all the uncontrolled and the controlled air preconditioning. In this example the overall energy consumption of the controlled ventilation strategy is the lowest. However cooling energy demand is higher than in some other cases. This is caused by the fact that in some cases the power demand for cooling is lower than that of heating. In this first approach it is only possible to switch between the three sources. Of course in the real plant the preconditioned air can be mixed. In further work the control strategy has to be enhanced in a way taking into account the possibility of mixed air and will be implemented in the existing plant for test and measurement purposes.

ACKNOWLEDGEMENT

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Improving the Possibilities to Monitor Energy Consumption at Home

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SUMMARY

Inhabitants have low possibilities to monitor their energy and water consumption. The energy and water meters are typically read only once a year in Finland. The monthly bills are based on estimated consumption, i.e. the consumption of the previous year. A quantitative interview survey with a sample size of 3,094 people was performed to study attitudes and opinions about energy conservation and energy monitoring. The results show that 63% of the Finns think that it is very important or important to improve the possibilities to monitor energy consumption at home. A new prototype for monitoring energy consumption was developed at VTT. It gives occupants real time feedback on their energy consumption and the costs. In addition to the total consumption and costs, the prototype is able to present electricity consumption and costs of each device. The target is to create a system that gives occupants relevant information in a way that is easy to understand. The system is installed in two experimental houses.

INTRODUCTION

Inhabitants have low possibilities to monitor their energy and water consumption. In Finland, the monthly bills are based on estimated consumption, i.e. the consumption of the previous year, because the energy and water meters are typically read only once a year. This means that a household receives an additional bill or is paid after the meters are read. Some households may then get a bad surprise: a newspaper article [1] told about a family that received an additional 1800 € water bill.

Manual reading of energy meters will be replaced with remote reading in future. In Sweden, a new law demand monthly reading of energy meters from 1st of July 2009. Also in Finland and other countries, remote reading of energy and water meters is becoming more and more common. At first, remote reading becomes common in densely populated areas and apartment buildings. Remote reading will make at least some improvement in our possibilities to monitor consumptions.

The aims of our work are to study attitudes and opinions of Finns about household energy conservation and energy monitoring, and to develop a new prototype for monitoring energy consumption.

METHODS

Attitudes and opinions of inhabitants about energy conservation and energy monitoring were studied by a quantitative interview survey with a nationally representative sample. The target group of the study was the population of Finland. The interviews for the survey were carried out by telephone (computer assisted telephone interview, CATI). A random sample of the Finnish population aged between 15 and 74 was selected with quotas set according to gender, age and province. The total number of respondents was 3,094. A well-known Finnish data collection agency (Taloustutkimus Oy) was responsible for the practical realisation of the telephone interviews according to its quality system. The interviews were performed in November and December 2004.

In addition to the quantitative interview survey performed, a new prototype for monitoring energy consumption was developed in this work.

RESULTS OF THE INTERVIEW SURVEY

The results of the interview survey are presented in Figs 1-4. Fig. 1 shows that people think that it is important to save energy at home. 82% of the respondents think it is very important (5) or important (4) to save energy at home. Own strive to save energy is, however, somewhat lower (Fig. 2).

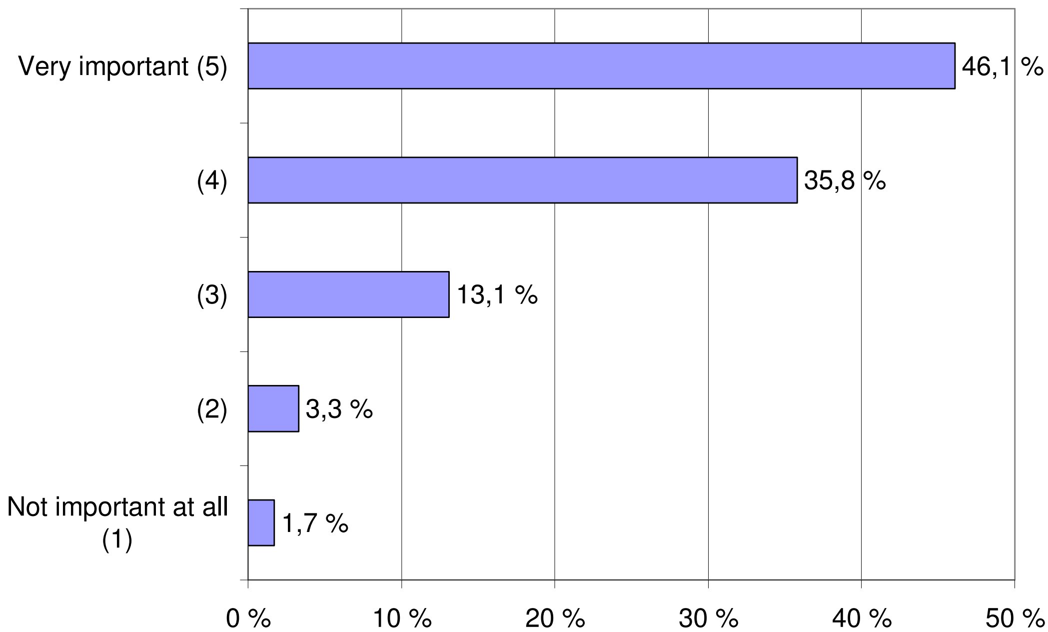

Figure 1. Importance to save energy at home (N=3075, mean: 4.2).

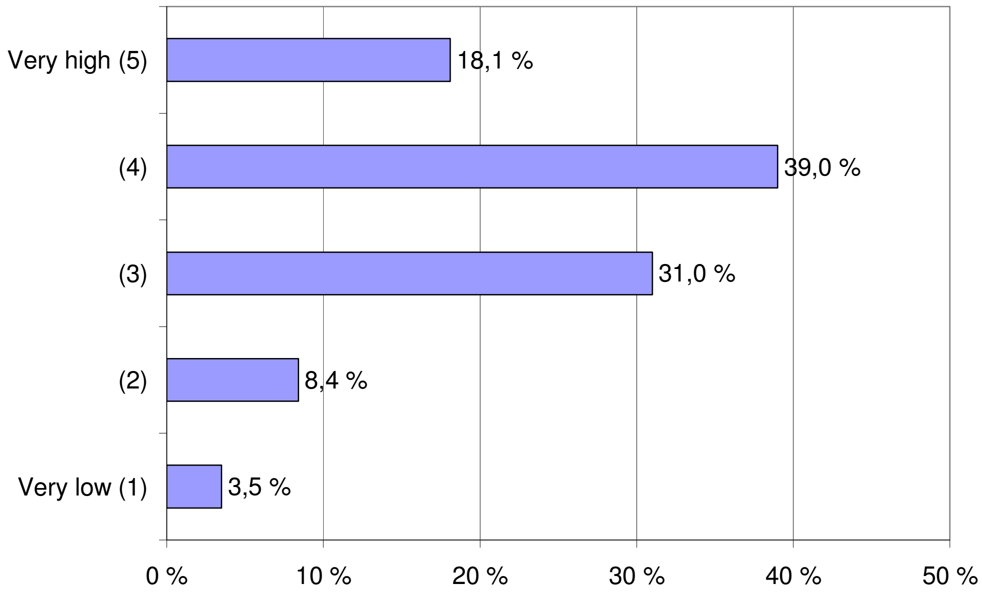

Figure 2. Own strive to save energy at home (N=3067, mean: 3.6).

People think their own actions have an influence on energy consumption (Fig. 3). Mean value is 4.0 in a scale from 1 to 5 (1, very low, … 5, very high).

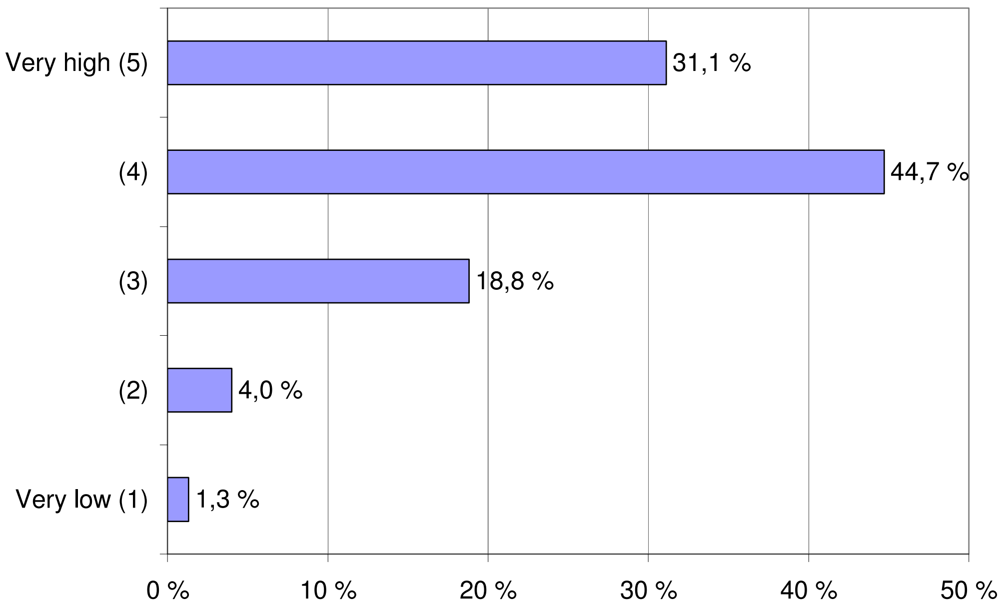

Figure 3. Idea of inhabitants' influence on energy consumption at home (N=3015, mean: 4.0).

Fig. 4 shows that 63% of the Finns think that it is very important (5) or important (4) to improve the possibilities to monitor energy consumption at home. Only 6% think that it is not important at all.

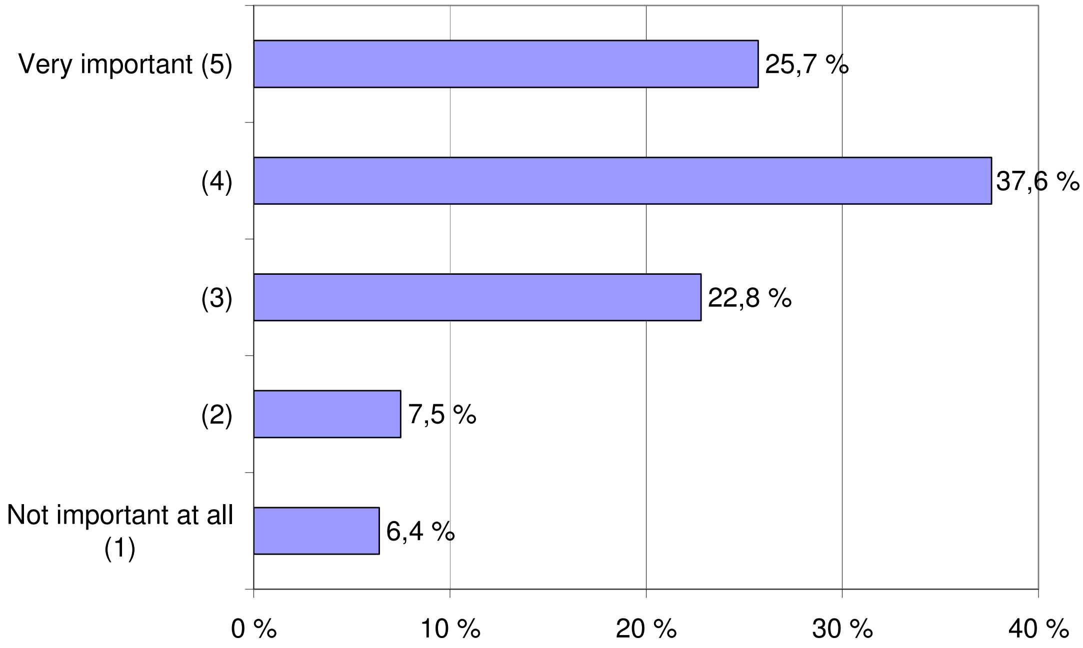

Figure 4. Importance to improve the possibilities to monitor energy consumption at home (N=3008, mean: 3.7).

PROTOTYPE

The results of the interview survey show that it is important to improve the possibilities to monitor energy consumption. A new prototype for monitoring energy consumption was developed in this work. The prototype gives occupants real time feedback on their energy consumption and the costs. In addition to the total consumption and costs, the prototype is able to present electricity consumption and costs of each device. The target is to create a system that gives occupants relevant information in a way that is easy to understand. Main components of the prototype are presented in Fig. 5 and the user interface of the prototype is presented in Fig. 6.

In the prototype two LonWorks technology based electricity meters were connected to a computer that works as a home server. The measured values were true RMS power, reactive power, power factor, true RMS energy, reactive energy, AC frequency, peak demand and computed values of RMS voltage and current. These values were saved in a database (home service book) once a minute.

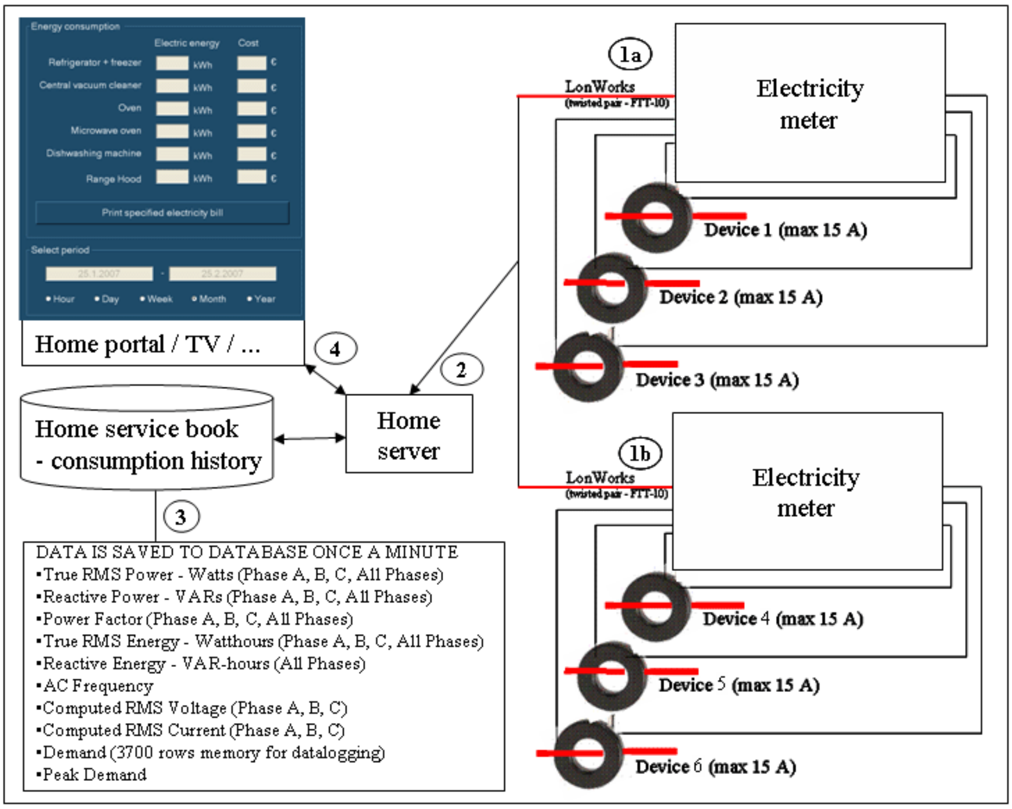

Figure 5. Main components of the prototype for monitoring energy consumption at home.

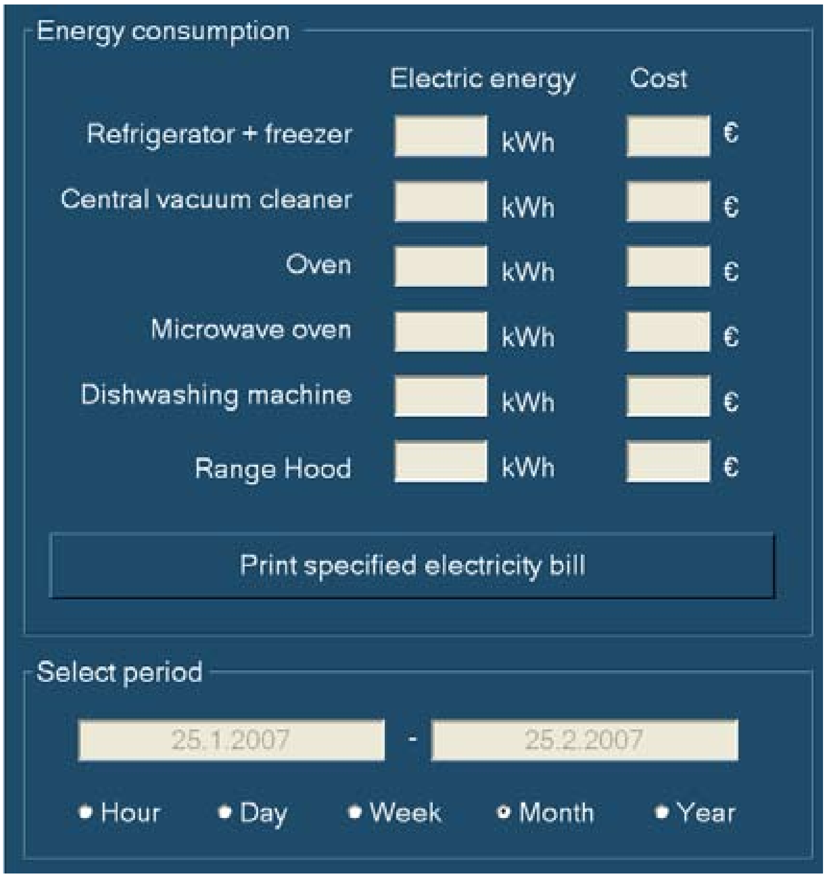

Figure 6. User interface of the prototype for monitoring energy consumption at home.

The user interface in Fig. 6 presents electricity consumptions of kitchen equipment and vacuum cleaner. Depending on the case, it would be very valuable to include also lighting, heating and air conditioning etc.

The prototype system is installed in two houses in Finland, in VTT's experimental low energy house Metop in Espoo and in Work Efficiency Institute's (TTS) experimental house in Nurmijärvi. A study of usefulness of the prototype was performed in Work Efficiency Institute's experimental house with twelve test persons. The participants were between 29 and 84 years old. The test persons think that electricity monitoring system is necessary in future, for example, in the case of selling the house. The user interface of the prototype was found to be clear, and it was easy to monitor electricity consumptions of domestic appliances. The test persons believe that this could affect their use of electricity. A forthcoming publication will present the results of the tests in more detail.

Next the consumption monitoring system will be improved with several new features. The main improvement concerns non-intrusive appliance load monitoring (NIALM) [2-5].

DISCUSSION

Public discussion on climate change and global warming has been very active in Finland in this year and in the end of the previous year. The interviews were performed before that. It is probable that people are currently even more interested in saving energy and monitoring energy consumption.

A review [6] of intervention studies shows that giving people information does not necessarily lead to behavioural changes or energy saving. Instead, according to the review, feedback appears to be an effective strategy for reducing energy consumption. The more frequent the feedback is, the more effective it is. For example, continuous display of electricity consumption was associated with an average saving of 12% in USA [7]. In another study [8], the effects of daily and monthly feedback on gas consumption were compared. It was found that those who received monthly feedback saved less that those who received daily feedback (7.7% vs. 12.3%).

The prototype developed in this work targets energy saving by means of feedback, i.e. by continuously informing how much energy each device consumes and what does that cost. There is clearly a potential to save energy with the system. In this phase, not enough evidence is, however, available, because the system is not installed to any real household yet.

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Development environment for model and automation based building management

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SUMMARY

The aim of sustainable building operation and management can be approached considering criteria such as minimized CO_2 emission or reduced energy costs. To consider these points, while also taking into account aspects such as well indoor air quality, improved control and automation strategies are required. One approach is to design decision-making strategies (e.g. for improved energy performance) calculated by an overall building management system that is based on simulation results and which e.g. sets optimised parameters for the local controllers of an energy subsystem for heating, ventilation or cooling.

To design, test and validate new or improved decision-making strategies for a building management system a flexible and open development environment is needed. "Development environment" in this context refers to a collection of simulation models and simulation programs which represent the building and the most important parts of the energy systems within the building, including building control and automation systems. At the Biberach University of Applied Science several building simulation programs (e.g. TRNSYS) and control simulation tools (e.g. Matlab/Simulink, LabView) are connected together in an open development environment.

During the design phase of a building this development environment can be used to simulate all energy subsystems including building control and automation systems. During building operation the development environment can also be used as part of a model and automation based building management system with the aim of optimised control and energy performance.

To test and validate the development environment experimentally a test facility has been set up with different energy systems such as a solar air collector and an air handling unit with a cross-flow heat exchanger. These energy systems supply a classroom which can be used as an experimental test room equipped with different sensors and actuators and a freely programmable room controller.

In the paper the structure of the development environment is described and some examples are presented to show the benefits of model and automation based management systems. The room model and some simulations results are described. Finally the benefits to energy reduction of a CO_2 demand-controlled strategy (DCV) are presented and compared to fixed air change ventilation strategies.

INTRODUCTION

At the Biberach University of Applied Sciences a new classroom and laboratory building was built in conjunction with the new program “Gebäudetechnik / Gebäudeklimatik” (building services engineering / architectural engineering). The building of itself serves as a “laboratory-in-a-building” containing a variety of novel building systems for testing and research purposes. For this reason, the building is named “Technikum G.”, see Fig. 1. A detailed description of the building, its energy systems and equipment, as well as details of the facility can be found in [1] and [2].
In the work described here, an air handling unit (AHU, Fig. 3) together with the classroom G0.03 (Fig.2) as a flexible test room serves for testing of simulation-based building control strategies. The fans of the AHU are equipped with variable frequency drives and dampers are equipped with electric actuators. Ventilation air can be introduced either directly or in a preconditioned fashion by preheating of outside air using a solar air collector, or by preheating or precooling outside air using an earth-to-air heat exchanger, Fig.. 4.

Figure 1. View of “Technikum G”.

Figure 2. Classroom G0.0 3 as test room.

Figure 3. Air Handling Unit (AHU).

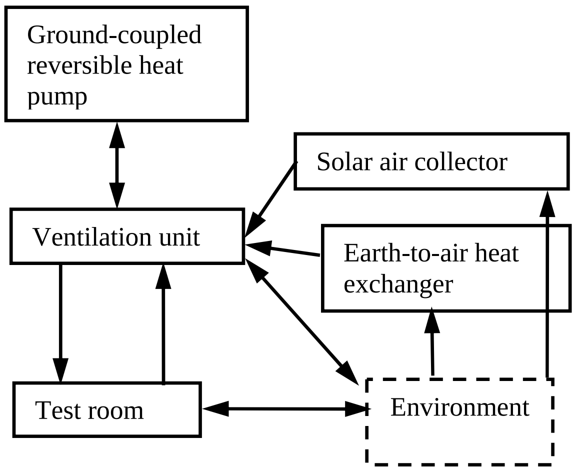

Figure 4. Overview of the test environment [1].

The classroom G0.03 is equipped with a wide range of sensors for measuring temperature, relative humidity, indoor air quality (CO_2, VOC), illuminance, occupancy, and window contacts [1, 2]. Moreover, numerous actuators are provided including ballasts for lamps, dampers for the supply air and a zone valve for heating. All sensors and actuators are connected to a room controller (PLC, programmable logic controller), which is directly connected to the overall building automation system via an Ethernet TCP/IP connection. More than 90 data points in the building are visualized and continuously monitored by the overall building automation system.

MODELING RESULTS

Work has been done and is still in progress to model and simulate all experimental systems in the flexible simulation environment. The approach chosen is to model all components of the air ventilation system, the test room and its facilities including the sensors and actuators. For the purpose of investigating and comparing improved control and automation concepts, there is a high demand for appropriate simulations models.
For maximum flexibility, an open-simulation approach was adopted. A variety of building simulation programs (e.g. TRNSYS), control simulation tools (e.g. Matlab/Simulink, WINFact), and building visualization systems (e.g. LabView, BCON, Wizcon) are implemented while dedicated automation stations (e.g. programmable logic controller) are associated with the various subsystems (air ventilation system, heat pump, rooms) in the "Technikum G". All tools and components are linked together by different communication systems such as the EIB, LON, or Modbus etc. via OPC-Server, [1, 2]. Furthermore work has been done to couple different simulations tools like TRNSYS and Matlab/Simulink, [3].
With this setup, it is possible to develop and test new control strategies in the simulation environment as well as in the test room. Subsequently, improved control concepts can be tested in the actual experimental facility. Moreover, it is possible to visualize and monitor all process data as well as simulation data on a building automation system.
Fig. 5 shows the structure of the simulation models of the test environment as shown in Fig. 4 with the sub-models earth-to-air heat exchanger (EAHX), solar air collector (SAC), air handling unit (AHU) and the model of the test room G0.03 with room air, walls, radiator and windows.

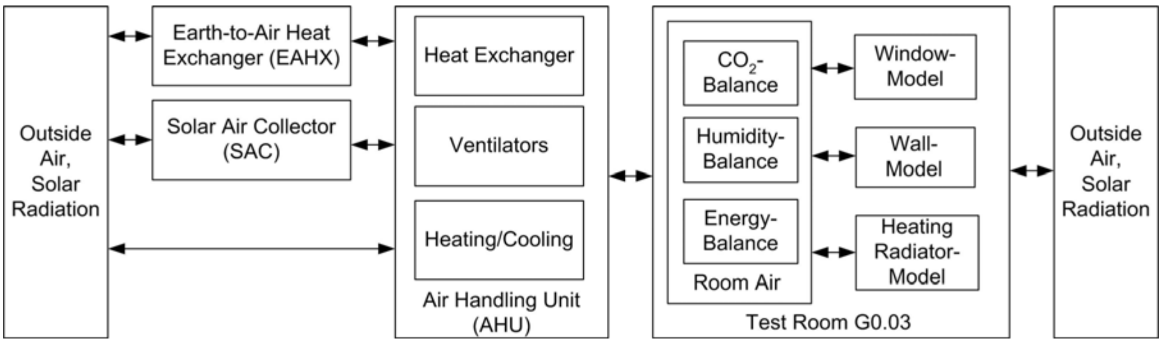

Figure 5. Overview of the simulation models.

In this paper the sub-system "room model" will be discussed in more detail. The sub-systems air handling unit and earth-to-air heat exchanger are described in [4], the sub-system solar air collector is described and discussed in [1]. The idea is to set up a real test room equipped with sensors and actuators as well as a flexible simulation environment for testing new improved room automation concepts and room control strategies which consider not only the room air with room air temperature, operative room temperature and relative humidity but also indoor air quality (CO_2, VOC). Thereby it is possible to investigate and evaluate different room control strategies in terms of energy efficiency, thermal comfort and indoor air quality (IAQ).
In a first approach the simulation model of the test room is developed in a top down way and kept as simple as possible. The main components are the room air, the walls, radiators for heating and windows. Coupling the air handling unit to the room model for air exchange of heated or cooled supply air is also considered. All walls are modeled as equivalent electric R-C circuits. For the air room the temperature is calculated based on the energy balance, the humid air on water mass balance. Additional a CO_2 balance equation is implemented to calculate the actual CO_2 concentration in the room air.

DISCUSSION OF THE ROOM MODEL

The validation of the room model is based on procedures defined by VDI 6020, [5] with two test rooms, standardized boundary conditions and a given set of simulation examples. Because the test room G0.03 is not identical to the test room in VDI 6020 another way had to be found to validate the room model of G0.03. According to this, validation of the room model was conducted in several steps, see Fig. 6.

1. The simulation tool TRNSYS was validated against test procedures based on VDI 6020, [3].
2. Generation of a validation data set. This was done with a simulation model of test room G0.03 in TRNSYS. Data sets are created for four different cases of internal and external gains. For creating this set of data, boundary conditions and constructions elements are chosen equal to the developed Matlab/Simulink model, [5].
3. Validate the room model G0.03 with this data set against the reference model in TRNSYS and weather data of Biberach, [5].

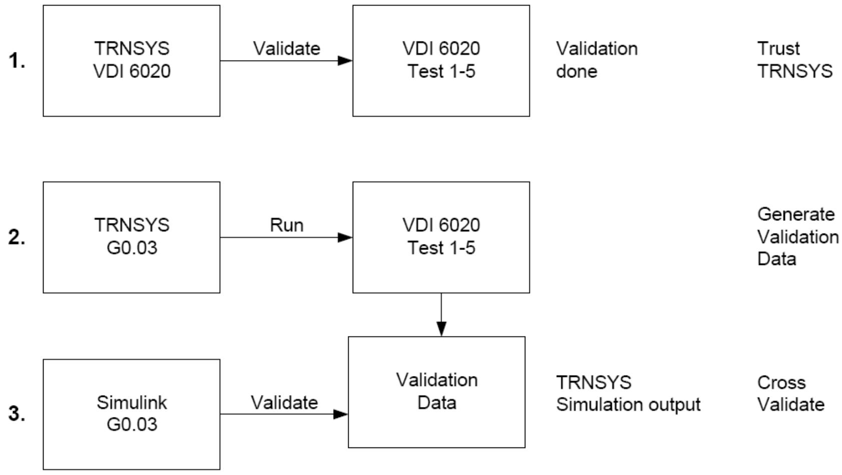

Figure 6. Approach for validation of room model versus VDI 6020, [5] and TRNSYS [6].

The simulation results show in general a satisfying match but also some significant differences. Based on a three day simulation result (Fig. 7) we will discuss some important details: There are only small differences in the dynamic run of the curves. The differences are caused e.g. due to different simulation time steps. The offsets in the air temperature and operative temperature of abut 1 K between the two simulation results can have several reasons. First, in the Simulink model the radiative gains to the construction elements are assumed to be absorbed in the centre of the first wall layer, instead of on their surfaces. Hence the absorbed radiation is subject to a larger heat transfer resistance towards the room air than in the TRNSYS model. The TRNSYS model assumes direct absorption of the radiation in the construction elements surfaces. This leads to lower air temperatures in the Simulink room model due to higher heat losses to the surroundings. Second, in the Simulink model a simplified assumption for radiative exchanges between walls and the surrounding was used. In contrast the TRNSYS model works with a radiative node model.

Conclusion:
With the respect to the aim of developing improved control concepts the quality of the newly developed Matlab/Simulink model turns out to be satisfactory. The final step, the validation of the room model against measured data in the test room G0.03 is still in progress.

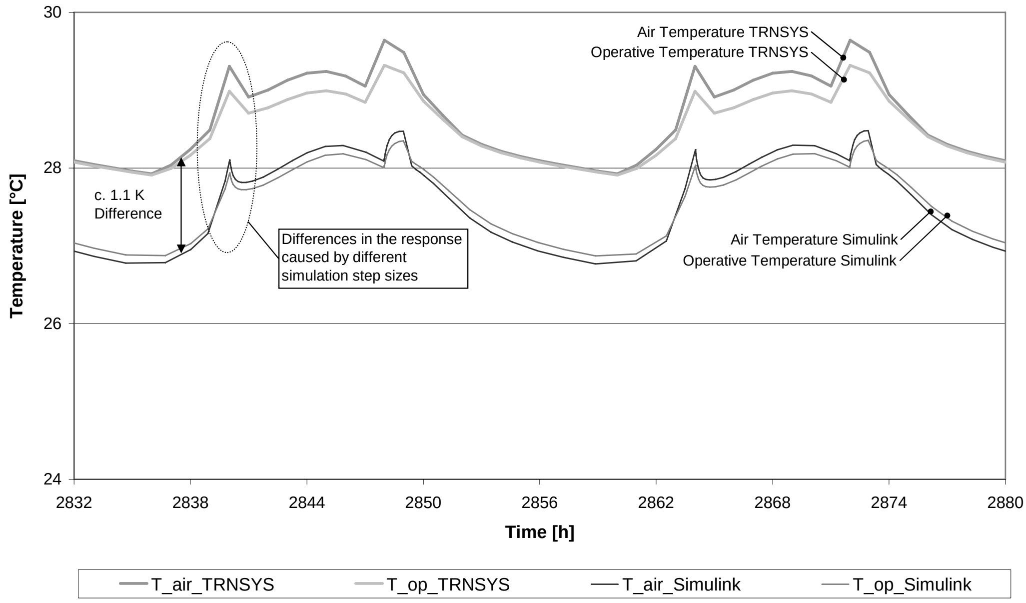

Figure 7. Comparison of test room model G0.0 in TRNSYS and Matlab/Simulink. (T_air: room air temperature, T_op operative room temperature)

DISCUSSION OF CO_2 CONROLLED INDOOR AIR QUALITY STRATEGIES

Based on the presented simulation environment different CO_2 concentration controlled strategies for a classroom were developed, investigated and compared. Two strategies will be discussed in the following.

The volume flow rate of the ventilation air as well as the corresponding air change rate that are necessary to maintain a certain CO_2 concentration mainly depends on the number of people in the classroom. In practice the number of people and their duration of stay in a classroom is rarely the same as the level of occupancy assumed during design. As a result a higher air change rate than necessary is often applied in the classroom which leads to higher energy consumption and higher energy costs. With a required air change rate of e.g. 30 m³/h per person and a CO_2 production of 20 l/h per person the calculated CO_2 concentration in the classroom room G0.03 with an occupancy of 35 people leads to a value of 1000 ppm, based on an air volume rate of 1050 m^3/h and an air change rate of 4.38 1/h respectively.

Fig. 8 shows a typical fixed time schedule for the classroom during the daily operation time from 8.00 to 17.30. The time schedule is defined with three lectures in the morning and two in the afternoon with short 15 minutes breaks and a one hour lunch break. In a first step it is granted that a fixed number of 35 people are in the classroom during the lectures.

As a simulation result Fig. 9 shows the curve of the CO_2 concentration under these conditions. Because the AHU system operates to a fixed time schedule, the CO_2 concentration slowly rises and reaches under these conditions the maximum at 1000 ppm. Because the ventilation system is running continuously during operating time with a fixed air ventilation rate the CO_2 concentration sinks during breaks. At the end of lunch time the CO_2 concentration reaches even the value of the outside air again.

Using a CO_2 sensor as an indicator for well indoor air quality (IAQ) the control task is to limit the concentration to a defined maximum value, e.g. 1000 ppm. This can be achieved by using an AHU system to control the air change rate into the classroom. This strategy in the following is called "CO_2-based demand-controlled ventilation (DCV)". In comparison to a fixed ventilation rate, DCV offers considerable advantages because it considers partial occupancy and the actual air quality conditions in a room. Fig. 10 shows the curve with a DCV strategy under the same scenario as discussed before.

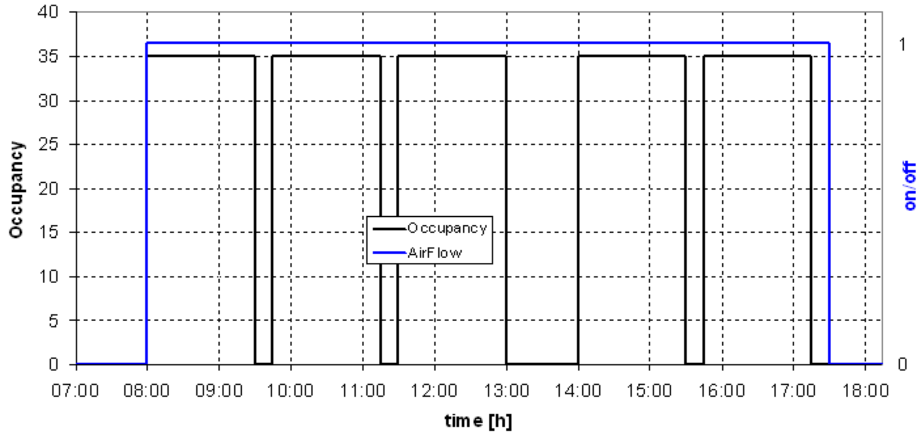

Figure 8. Example for a typical **fixed time schedule** for a daily operation time from 8.00 to 17.30 in the classroom with five lectures per day and an occupancy with 35 people.

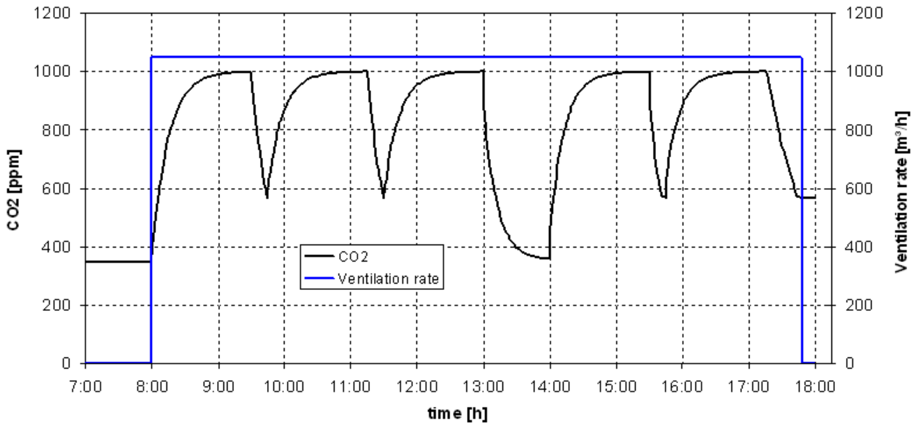

Figure 9. CO_2 concentration in a classroom with a **fixed air ventilation rate** of 1050 m^3/h and the scenario as shown in Fig. 8.

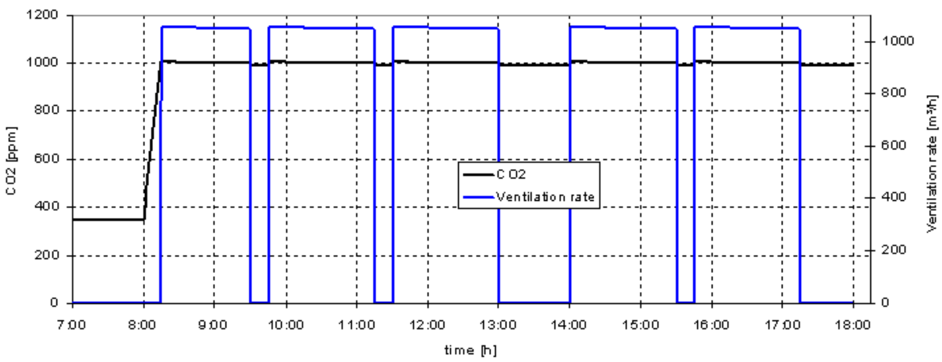

Figure 10. CO_2 concentration in a classroom with a **demand-controlled air ventilation strategy (DCV)** (set point of 1000 ppm) and the scenario as shown in Fig 8.

During usual operation the occupancy is normally not fixed to 35 people, but will be less most of the time. Taking the actual level of occupancy into account improved control concepts significantly reduce energy consumption. Fig. 11 shows a scenario with different levels of occupancy and a cancelled last lecture in the afternoon. Fig. 12 shows the resulting curve of the CO_2 concentration with a fixed air ventilation rate of 1050 m^3/h and Fig. 13 with the demand-controlled air ventilation strategy (DCV). The results in Fig. 13 are based on the same scenario as in Fig. 12. The ventilation control is activated only if the CO_2 concentration reaches the 1000 ppm limit. During breaks the ventilation switches off itself, if no occupancy is detected in the classroom. Nevertheless, during off state a minimum air change rate is supported to supply the room with a minimum fresh air volume.

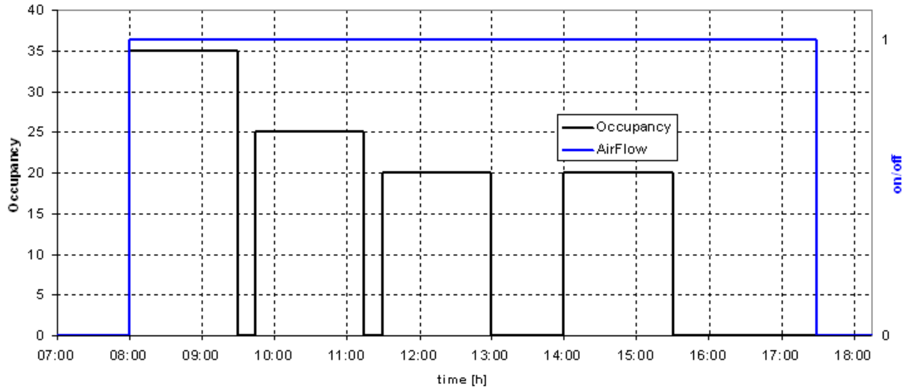

Figure 11. Example for a typical **fixed time schedule** for a daily operation time from 8.00 to 17.30 in the classroom with a **partial occupancy** with people and a cancelled lecture.

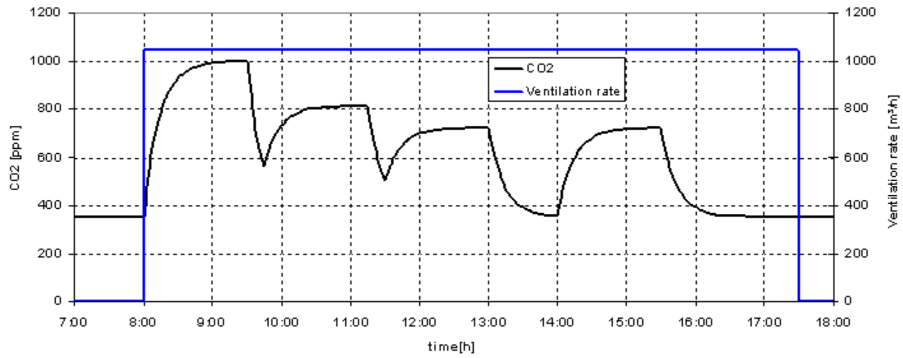

Figure 12. CO_2 concentration in a classroom with a **fixed air ventilation rate** based on scenario shown in Fig. 11.

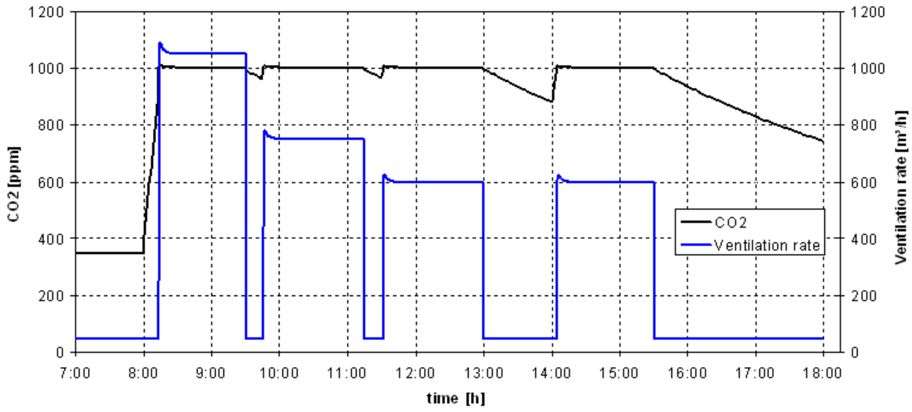

Figure 13. CO_2 concentration in a classroom with **demand controlled air ventilation strategy (DCV)** (set point of 1000 ppm) based on scenario shown in Fig. 11.

The energy consumption due to ventilation is a direct function of fresh air volume. Fig. 14 shows the resulting comparison of the two ventilation strategies shown in Figures 9, 10 and 13. The energy consumption for the demand-controlled ventilation strategy in case of a fixed occupancy of 35 people (Fig.9) is 28% lower and in case of different levels of occupancy (Fig. 13) even 59% lower compared to the fixed time strategy (Fig.9).

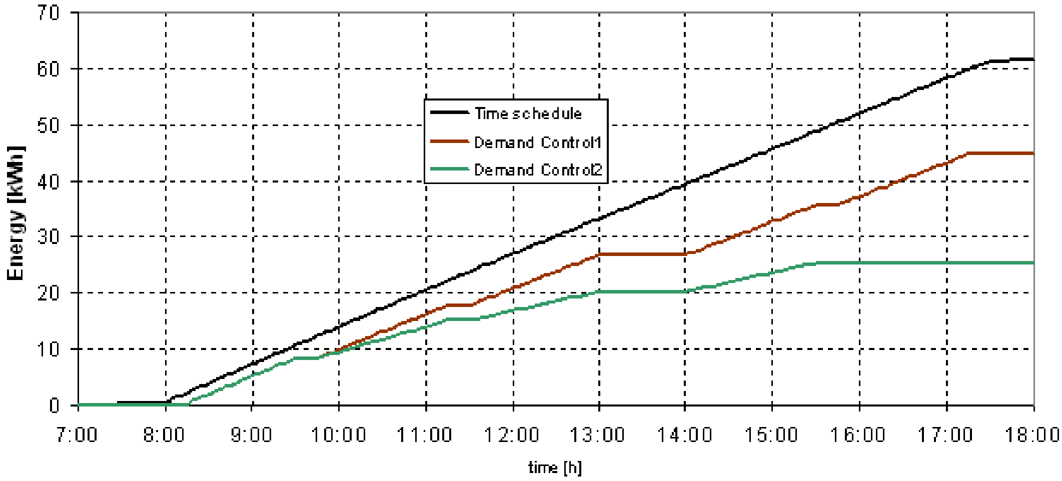

Figure 14. Comparison of energy consumption of the different control strategies based on fixed and demand-controlled air ventilation respectively.

Further work will be done to validate all sub models with data measured in the test room G0.03. Further investigations to improved automation concepts for energy efficient strategies will be carried out by simulations studies and more sophisticated control concepts (e.g. fuzzy control) which can be tested and evaluated in the test environment under real operating conditions.

ACKNOWLEDGEMENT

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The Evaluation And Analysis of A Multiple Home Trial Based on An Energy Management System

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SUMMARY

The power demands from residential and commercial purposes are increasing due to the life quality improvement. As the charge of power is based on the time-of-use, a proper solution to cut off the bill by reducing power consumption is highly proposed. In order to deliver this solution, it is essential to build up a power consumption model which contains useful information such as power usage during each time period and device type. Consequentially, a residential and commercial energy management system was designed to meet this need. The functions of this energy management system such as meter reading, data collection, reading analysis, energy saving potential and additional functions were tested during this trial. It was found that the system can help a user to reach 10.3% in energy saving. More details related to the trial and the testing results will be shown later in this article.

INTRODUCTION

There are two major factors to influence the residential and commercial power management system and they are the competition between power suppliers and the advance of network technologies. The first factor is because more suppliers are entering the power industry, and the intense of competition is fierce than ever. Hence, each supplier wants to offer users more value-added services such as energy management, home automation, direct load control and real-time pricing program to maintain its own competitiveness. From the report of Cutter Information Corp, the biggest support of home automation will be from the power industry. In addition, the Utility Customer Interface System growth reached 69.7% to 17.8 million sets from 1997-2004 from Frost & Sullivan. It can be seen that the competition of the power industry is the key to accelerate the applications of the information network technologies such as power management and other kinds of value-added services, because these applications can be delivered through the networks of the power suppliers or the other public communication networks.

For the second factor, the pervasion of network technologies which include the growth of communication network, the reduced costs and the appearance of embedded systems, increases the communication functionality and intelligence of devices, and also achieves the integration of separated database systems with networks. This trend can lead the power management for residential and commercial purposes to the next stage. According to the development trend of power management system around the world, new applications and services related to the information network are often delivered with the power management programs. It is believed that the integration of e-life and power management will come true due to the trend. Furthermore, since the competition intensity of power suppliers is getting fierce, services which can offer delicate solutions to users and users who will automatically save power consumption are getting more attention now. In Japan, the Energy Saving Navigator designed by Nado was capable of showing real-time power pricing due to the digital meter with wireless communication function. It was offered to 1700 families to use for two years, and power bill saving could reach 20% in average. The result of home power consumption survey showed that obvious saving can be found after a home energy management system was installed at home.

A “Home Energy Management System (HEMS)” project which is organized by EL Quest Corp, initialized by Panasonic and Hitachi, and supported by Japanese government aimed at applying information technologies onto the field of power management to achieve power saving by increasing users’ awareness. With the full support of the power supply company, this project carried out a multiple home trial involving 300 families in 2002. Within this trial, air conditioner, home appliances, and sensors were covered within the HEMS project, and a total power saving was set to 30%. In Italy, Enel Corp. replaced all the traditional analogue meters by digital meters integrated with LonWorks network technologies and aimed at building up the automatic meter reading system and offering value-added services to the users in 2003. According to the DSM Spotlight reported by the International Energy Agent (IEA), Bundle energy services were also carried out in Finland and the UK in 2003. The bundle energy services include remote energy management, appliance remote repairing, HVAC monitoring and control, and automatic meter reading. The Finland system aimed mainly for apartments, and the UK system offered services to detached houses.

According to the Energy Handbook published by Bureau of Energy, Ministry of Economic Affairs in Taiwan, 206,134.56 million Watts were consumed in 2004, and 19.20% which is about 39,587.86 million Watts were demanded by residential needs. Comparing with the residential needs in 2003, 2004 consumed 440 million watts more and the increasing rate was 1.1%. Comparing the previous recorded data, the power demands from residential and commercial needs increase about 1-5% per year.

Power consumption is found not directly related to the building design, climate difference and members of a family, as it is strongly related to size of a house, numbers of home appliances, the habit to operate the HVAC system and the life style. After analyzing the power consumption of a family, it is found that the numbers of home appliances and lighting devices are not the key factors to cause more power consumption, as it is more related to the using rate, usage and efficiencies of home appliances. Hence, how to choose the right applications and guide users to use them properly can then lead to the reduction of power consumption. In addition, how to deliver power consumption data to the users through the communication channels by integrating information technologies and to offer more value-added services can lead to an effective power management. This research aimed to achieve power demand reduction during the daily life with a power management system by causing users attention and then save power automatically.

METHODS

The function and performance of a set of energy management system was examined based on 100 homes. The energy management system can cooperate with existing devices to offer device management, home security and internet surfing for the purpose of e-life style. The architecture of this system is shown in Figure 1, and the key parts are the central control center, local control center and end user interfaces. The connections in between these three parts are through existing networks such as Ethernet, powerline network, and security network, and the explanation of these three parts can be seen as follow:

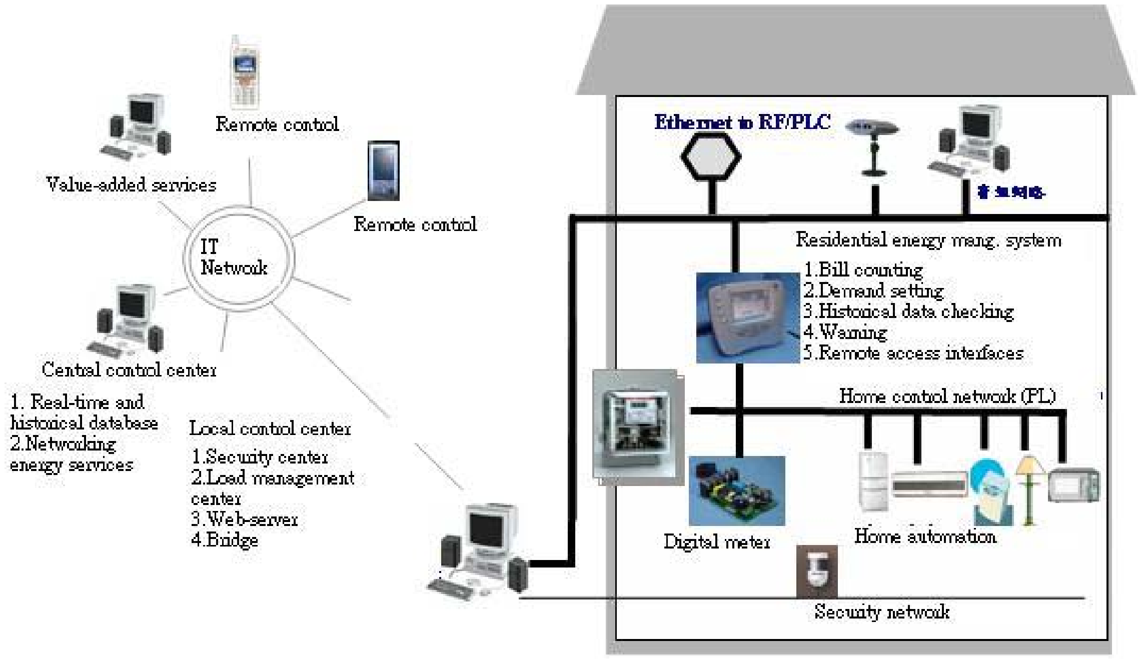


Figure 1: Architecture of residential energy management system

1. Central control center: to offer users services to access real-time and historical data, including power consumption data, data management, load checking and interfaces for vale added services.
2. Local control center: to extend the functions and interfaces from central control center, such as community security gateway, local server, end user device control server, community device management.
3. End user interfaces: to offer users proposed functions such as displaying power consumption data, power bill budgeting and home security.

The aim of this research is at testing and evaluating the function and performance of this energy management system for further promotion, and the testing and evaluating items are:

1. Long term operation testing: to operate this system for 6 months and collect data such as power consumption, historical power usage investigation and system stabilization test.
2. Load analysis: to analyze the collected power consumption data according to the family types.
3. Power saving potential investigation: to compare different power usage modes and find out the potential of power saving.
4. User satisfactory reviewing: to understand the usage of this system and its capability.

RESULTS

1. Energy management operation test

This energy management system includes embedded power meter, energy bill counter and control device and real-time and historical data base. After this system was established, 100 homes including detached houses, apartments, flats and offices were invited to carry out a

long term test, about 6-12 months in Taipei city. Several types of data were collected such as power consumption data, power usage and historical power usage habit.
Figure 2 & 3 shows the installation of power meter, which can collect required data and then send to count the power bill in the energy management system. In addition, this system can also set the desired power usage and warming signal will be sent if the power usage exceeds the desired level.

Figure 2: Energy management system used in this trial

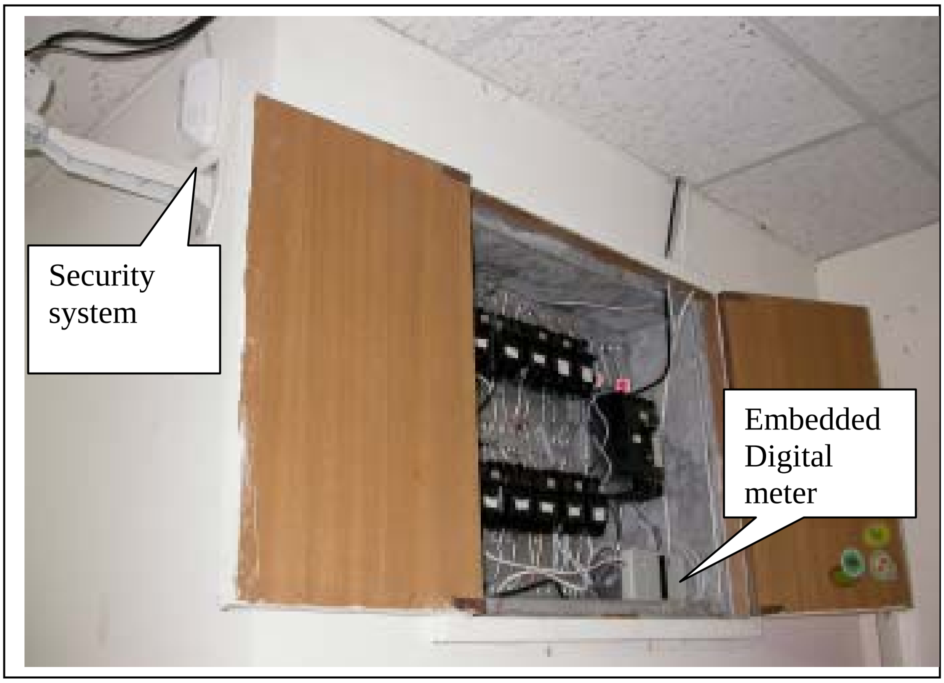
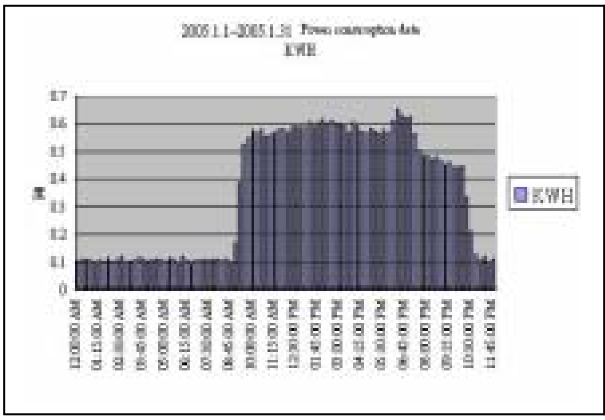


Figure 3: Installed embedded digital meter and security system

After 6 to 12 months operation, power consumption data collected from different types of home are used for analysis to find out the characteristics. The collected will also be compared with the situation in the previous year to understand the power need per month, and the power saving ratio. Related data such as education level of the users, size of the house, members of the family are also collected to evaluate the potential for further power saving.

General residential power usage characteristic

This energy management system can sample the power demand every 15 minutes. Figure 4 is the power usage of a house (sample A) which also joined the trial. It can be found that the power demand takes place from 9AM to 10 PM, and peak demand appears at 6-7PM. It can also be seen that the demands during week days are higher than weekends in figure 5.



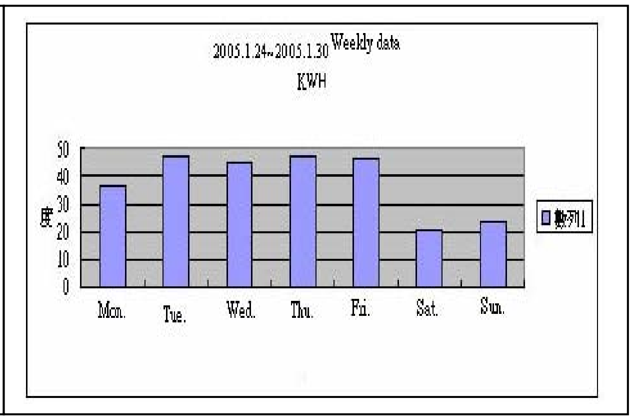
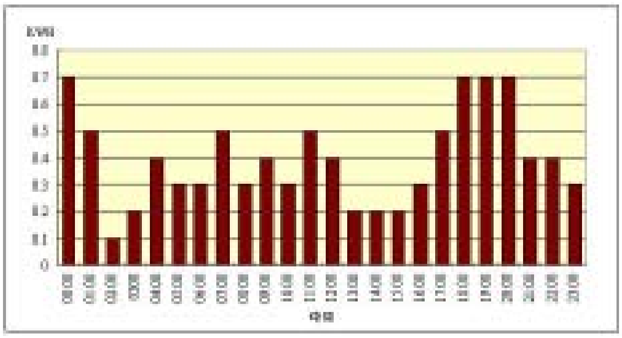


Figure 5: daily power consumption

Figure 6: Weekly power consumption

Here shows another example (sample B) by Figure 6 to Figure 9, the peak demand appears from dinner hours to bed hours, and it also shows that the demand during the weekend is higher than weeks. It also shows that the power demand of the Chinese New Year Eve is higher than rest of the February, and the average power demand in August is about 1.4 times in February.



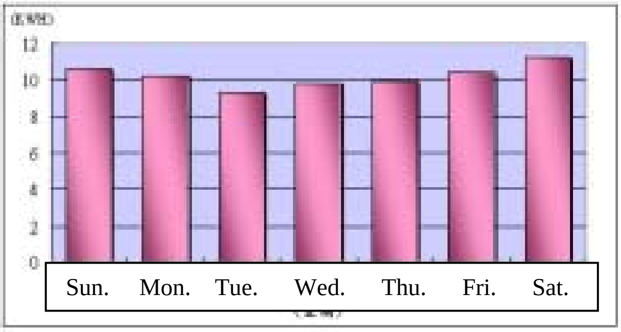


Figure 6: daily power consumption

Figure 7: weekly power consumption

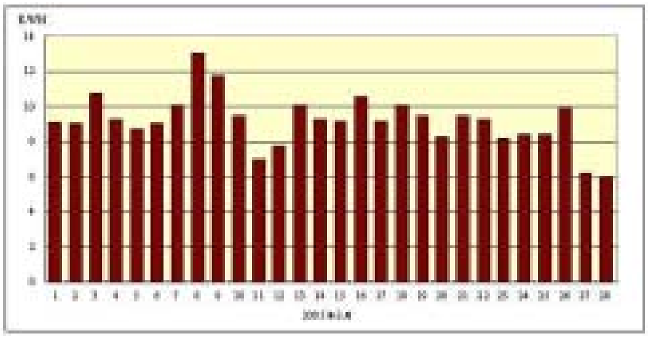


Figure 8: monthly power consumption

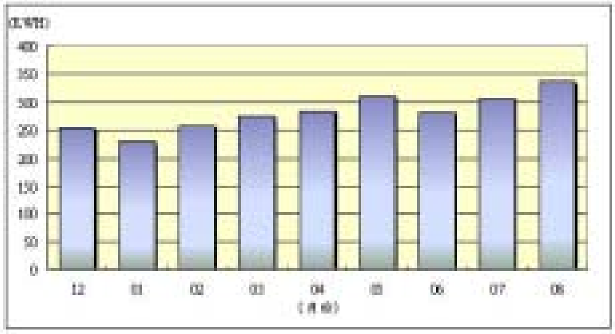


Figure 9:yearly power consumption

In Taiwan, some of the offices are allocated in a building next to an apartment. For Figure 10, it shows the power demand of an office allocated in a building. It can be found that the power demand increases in the office hour, 8AM to 7 PM or 10AM sometimes. The difference of power demand in between week days are not obvious, but during the off office hour and weekend, the demand decreases to minimum for standby power only. Basically this trend matches the demands received by the power supply company.

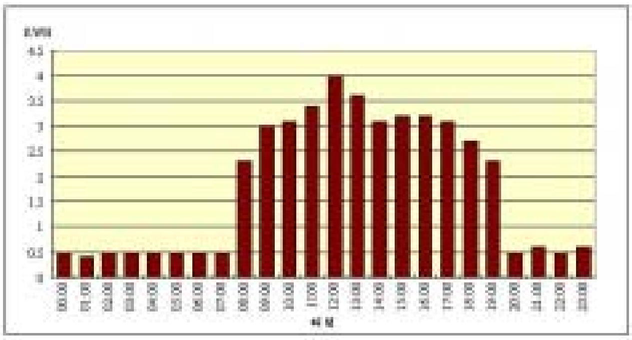


Figure10: Office daily power consumption

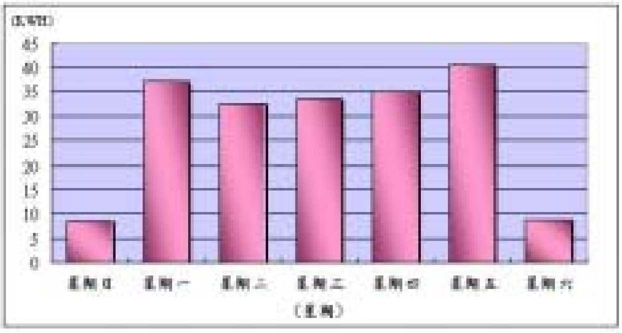


Figure11: Office weekly power consumption

The evaluation of the energy management system

After installing the system within the 100 trial homes, the power saving ratio can be seen from Figure 11&12. In sample A, the ratio is about 1.8% after comparing with the power usage of the previous year: the ratio of sample B is getting better and better. Both samples are showing positive result after adopting the energy management system.

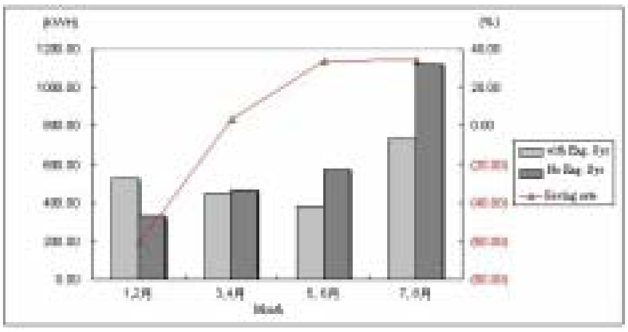


Figure12: power demand reviewing(sample A)

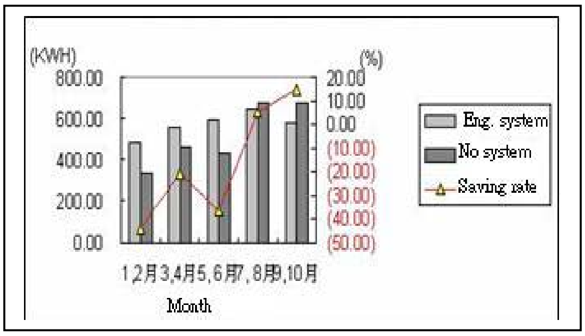


Figure 13: power demand reviewing(sample B)

In spite of the power usage investigation, this research also use questionnaire to understand the changes of the energy management system users to evaluate the potential of power saving. The target of this questionnaire is to understand the usage of the home appliances such as air conditioners, refrigerators, TV sets, dehumidifiers, lighting devices, microwaves, rice cookers and washing machines. In addition, it is also eager to know the satisfactory level and the pricing of this energy management system for further promotion.

Residential power usage analysis:
After the energy management system was installed, users did change many of their power usage habits themselves due to the warming signal. For some issues, users have changed their habits, but the improvement is not obvious, as the saving rate is about 11%. However, some of the issues can save 16-41% after proper advertisements and introduction, such as “cool off the food before put into the refrigerator” and “switch off the light as you don’t need it”. Form the experience of Japan, the issue “unplug or switch off the appliance as it is not operating” can do better power saving than other issues.

Before the energy system was installed, the energy saving issue gets more attention from users that has been awarded with first degree or above by a cross analysis. After the promotion and introduction, users with different education level realized the importance of energy and the users have master degree pay most attention to this issue.

The pricing of the energy management system
After analyzing the questionnaires and interviewing the users, the reasonable price of the energy management system is around NT 16000 dollars. It is roughly the cost of this system and the system has great chance to be accepted by its users. In addition, many users will like to have the system if extra benefits can be provided with this system.

DISCUSSION

This energy management system was tested in a multi-home trial for 6-12 months, and the target is to understand the family power requirement and the potential of power saving. The data of power requirement can be used to manage the power needed or set the contract demand with the power supply company. The power saving achieved by this system is about 1.8% recorded in a detached house, and the saving rare is around 4-21% in community centers. The saving rates recorded in different houses are different, and this can be caused by the difference level of behavior change or change of life style. The found result is quite different from 20% saving rate published by Japan Energy Saving Navigator. To find out the reasons, it can be because of higher energy price of Japan and culture difference. However, this system can actually achieve power saving as it was designed for. Even though the tips for power saving has been published and set inside the energy management system, users did not pay much attention to it, and face to face communicate can achieve better promotion results.

The promotion of residential energy management system requirement further improvement in software and hardware, and it also need some other supporting methods such as promotion through community activities to achieve a better result of saving. In addition, this system should lead users to save energy by changing their behaviors in energy using or add additional functions such as monitoring home power consumption, control the power demand, reduce standby power to extend the functionality of this system.

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Digital Convergence and Building Automation Systems

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SUMMARY

In this paper we present our (possibly controversial) view, as Internet devotees, on the state of Building Automation Systems (BAS's). With one exception, the authors were not BAS practitioners until very recently, when we became engaged in an experiment of building an Internet Protocol (IP) based and open-source based BAS. This work is described in our other paper in these proceedings [1]. Being new to the field we started by studying the BAS state-of-the-art and by familiarizing ourselves with the technological, legal (mostly in the context of intellectual property protection), socio-technical, and business aspects of the BAS industry [2-7].

This paper summarizes our "lessons learned" in two parts. First, we discuss our view of the BAS industry from the point of view of outsiders who are expert in broad aspects of the Internet: IP networks and protocols, Internet-application implementation frameworks, and Internet-related business models, including open source. Second, we take a more critical view in our own work of implementing a BAS based entirely on an Internet technology stack. As a whole, our two papers lie within the genre of systems research.

INTRODUCTION

Digital convergence of communication networks has advanced very far in recent years. Voice and video communications are now well established on IP networks [8], and are quickly becoming as ubiquitous as basic information retrieval and e-commerce services. The global information infrastructure is essentially Internet Protocol based, with all higher layers of network stack, diverse as they are, utilizing the same IP transport layer [8,9].

In contrast, convergence in the field of building automation and control is much slower, to a point that BAS's seem not to be participating in the communication and media exchange convergence. The world of building and process automation is dominated by BACnet/LONTalk [2,3] networks, which, although packet-based, implement a design philosophy alien to IP, with application logic supported by low-level network layers. Web-based interfaces to some of the existing systems and BACnet migration to Ethernet transport are superficial at best. The industrial application development frameworks (ADF's) are mostly proprietary, and carry substantial royalties or even patent protection. We believe that this situation stifles innovation and makes true digital convergence of BAS's an arduous process, with a significant negative economical impact.

In this paper we present our observations of differences between the mainstream Internet technology stack and the approaches adopted by the BAS industry, and we point to weaknesses of the latter as we perceive them. Then, we discuss the architecture of a fully converged BAS system based entirely on open software solutions, with functionality matching typical current BAS ADF's. We believe that he system has the potential to liberate the industry from proprietary solutions and bring BAS's into the mainstream of the digital convergence process.

NOT ALL PACKET NETWORKS WERE MADE EQUAL

Pre-1995, BAS's were closed, isolated, inaccessible, and inward-looking. Faced with the Internet phenomenon, the BAS industry started supporting an interoperability process. BACnet[1] [2] and LONTalk/LONMark [3] were created. Both standards are packet networks. How similar are they to Internet Protocol?

We will focus here on the BACnet standard [4]. Intellectually, we consider BACnet an admirable piece of engineering work and we recognize its superiority over proprietary solutions. However, from the standpoint of the Internet community, the BACnet design is a never-ending source of surprises and conceptual difficulties. One of the differences is the approach to protocol layering: BAS networks define all seven layers of the OSI Reference Model. In the IP world, development moved forward in small, incremental steps. Each protocol was kept simple and designed to do one thing well. Each protocol could have been tested separately, and in fact has been tested until perfection is achieved. With a lower level protocol in place, the Internet Engineering Task Force (IETF) community [8] moved to a higher level, while low-level network engineers were designing carrier networks that could carry IP. This process has been repeated hundreds of times, leading to the current net of compatible, seamlessly interoperable protocols by which anything can be connected to anything, either directly or via a suitable adapter. The history of Internet protocols and of the IETF process is a history of stunning technological success.

BAS networks represent an orthogonal approach. BACnet specification [2,4] is a large library addressing every aspects of the networking system that the working group engineers and their corporate oversight could think of. The complexity of the design seems to reflect a trend to think about interoperability at every single protocol layer, as well as a desire to ensure that the definition will directly support existing products in all their proprietary diversity. The engineers did admirable work by unifying these requirements into a coherent design, but nevertheless the BACnet is an extremely complex design addressing issues that were best left outside of the specification. This, unfortunately, has grave economic consequences. Most devices connected via BACnet are relatively simple, sometimes as trivial as temperature or pressure sensors. The requirements for a BACnet device resulting from the complexity of the network are often far more complex than the sensor itself. We see this as a problem in the world in which wireless sensors can be made as small and ubiquitous as sand particles.

In [1], we describe a moderately complex Personal Environment Module with several sensors and actuators. This module can be purchased with an optional network interface. The cost of the network interface is approximately seven times that of the module itself and rivals the price of a small automobile. The interface designed and built by us following general methodologies perfected by the Internet technology stack can be manufactured for under US$30. This is, of course, anecdotal evidence. However, it is rather obvious that a) manufacturing

[1] Work on BACnet started in 1987. In 1995 BACnet standardization process was taken over by ASHRAE.

BACnet compliant devices is expensive due to the complex network interface, and b) writing fully compliant device drivers is difficult and hence costly.

Other concepts of BACnet that are surprising for Internet community are: a) a fuzzy notion of device address or identifier, b) amount of the state that BACnet devices are required to handle, c) low-level support for sensor and service discovery, and d) a large but closed list of services that the devices are supposed to render on behalf of other network entities.

The first concept hatched and solved by the IP community was device addressing, defined in absolutely unambiguous terms. This was separated from hardware, by simply allowing for two lower layers — data and physical links — with the issue of identifying devices in these layers left outside of the IP concept. This allowed implementation of efficient routing protocols working uniformly across the entire domain, from LANs to the global Internet. Indigenous BACnet routing exists, but it now seems clear that BACnet domains are best connected using IP [7].

The amount of state that network nodes must carry is of paramount importance. It is well known that Internet devices and applications limit it to an absolute minimum. This concept permeates IP networks at all levels, from routers to HTTP-protocol web servers. BACnet devices carry a lot of state, and this *per se* suggests that large BACnet networks cannot ever reach IP robustness and stability. Why would an industry supporting enormous real-estate investments settle for anything but the best?

At the heart of the BACnet specification, there are discovery services, such as “Who-Is” and “Who Has” [10]. The presence of these services forces implementation of a bi-directional client-service model for network nodes (see our comment above regarding amount of state). While various data-link layers may implement discovery services, above the IP layer there are separate, high-level directory services such as Active Directory/LDAP [11,12], or UDDI [13,14] for web services, or NIS [15], to name a few. These services are completely separate from low-level network services and enjoy the reliability of the underlying transport protocols. They also typically use client-server rather than peer-to-peer (P2P) architecture. One could argue that the location of the service in the protocol stack does not matter. However, we have had negative experience with networks that rely on self-discovery, with Sytek/Microsoft NetBEUI/NetBIOS [16] being the most notorious example. NetBIOS is finally being phased out in Microsoft products such as Windows Server 2003, after having caused users grief for over a decade. This aside, we believe that self-discovery of devices is a useful, but not critical feature. Many simple applications do just fine with manual configuration. We are inclined therefore to see BACnet reliance on discovery services as a liability rather than strength.

Discovery services are just few (albeit important) examples of services defined by the BACnet standard. Overall, there are at least 32 services in five categories. In addition, the standard defines multiple types of objects that devices have to support. The overall impression for an outside reader is that BACnet designers were focused on a network that would easily support all possible existing sensors, devices, and applications in the protocol itself, instead of providing the basics and leaving the applications to developers. Again, one can argue that the approach taken is beneficial for interoperability. We remain skeptical. Interoperability is far harder to achieve for complex protocols than for simple ones. Further, any attempt to encapsulate functionality of applications available at a given time is futile, as it invites obsolescence once the standard is defined. BACnet proponents would probably respond that the protocol is extensible, but this is a self-defeating measure: the very mechanism of extensibility and the possible extensions themselves increase the already significant complexity, and the extensions do not guarantee interoperability.

BAS APPLICATION DEVELOPMENT FRAMEWORKS

Let us consider now to the next layer of the technology stack: the application-development frameworks. This is a large and complex field, discussion of which exceeds the space available in this paper by orders of magnitude. The general trend is a division of the market space between a few major players. In our opinion, several vendors were able to attract a very large following for their products, precisely because interfacing to complex BAS networks is difficult. Two frameworks, supported by respective organizations, seem to dominate: a) Niagara Framework [17] and the industrial consortium behind it [18] and b) the OPC (OLE for Process Control) Foundation [19-21] with over 300 members supported single-handedly by Microsoft, as the technology is tied to the Microsoft DCOM. The Niagara Framework is used by many of the largest manufacturers of HVAC equipment. It is a complete software framework and development environment. Recent software releases clearly gravitate towards better integration with the Internet [22], but this is done by layering web services functionality on a rather thick stack of components already dealing with complex BAS networks. Interestingly, the framework description cites “device data unification” as one of core benefits. In other words, the complex networks are now being simplified to a level that applications really need. Of course, there is a price for any of the elements implementing this “standardize-complicate-simplify” seesaw cycle, and the end users eventually wind up paying the bill.

INTELLECTUAL PROPERTY PROTECTION

An additional interesting facet in this thread is that the Niagara Framework is patented in the United States (US Patent # 6,832,120 [23]). This patent is one of a multitude of US Patent and Trademark Office (USPTO) patents for software systems. In the simplest terms, the patent describes a methodology for integration of elements provided either by other vendors or by public standards using a pervasive, well known, and generic software infrastructure. The patent is an example of the recent trend followed by the USPTO for software patents: it is possible to obtain protection for arbitrarily complex systems built from a mixture of generic and widely used modules that do not need to contain any intellectual property of the patent author. US Patent #6,832,120 essentially claims that any software system used for building control that uses the notion of software objects to represent devices infringes on the intellectual property rights of the patent owners. This is a very broad claim that certainly discourages implementation of competitive frameworks.

OPEN SOURCE BASED BAS

The preceding sections should have made it clear that we are not particularly fond of the current state-of-the-art BAS technology. To be constructive, rather than merely critical, we have decided to implement a functional BAS completely from scratch using only pervasive Internet technologies and methodologies. Some aspects of this work have been described in our other paper in these proceedings [1]. Here we would like first to discuss certain recent developments that, in our opinion, justify such work. Next, we will discuss problems and difficulties we have encountered in the design and implementation process.

First, we are strong believers in the open-source phenomenon [24-33]. Before embarking on the implementation, we have searched for an “open” BAS system, or elements thereof [34]. Not only have we failed to find any significant work, but also we have also found it unusually difficult to obtain information about commercial frameworks. In several cases, we were told that access to information is contingent upon a purchase of an expensive membership in a consortium and/or disclosing project goals and details. A study of the “open” projects usually revealed a lack of substance behind publicly stated claims. For a research group immersed in the Internet culture, this has been an unexpected and sobering experience, but also a stimulating challenge.

Our decision to implement a BAS the “Internet way” was reinforced by several technological developments:

Sensors are getting smaller: the “intelligent dust” is no longer a domain of science fiction. BACnet design philosophy does not appear suitable to handle this situation.

There has been profuse development of diverse types of wireless networks: WiFi promises blanket coverage in metropolitan areas [35]; satellite connectivity providing global reach is becoming both affordable and technically straightforward due to availability of Ka band [16] and low-orbit constellations, such as GlobalStar [36]; and mesh networks such as ZigBee [38] provide support for mini-sensors. All these technologies support IP by definition, and can serve as access media for BAS’s.

Construction of pure IP controllers for sensor and actuator devices has become very easy thanks to the availability of miniature, dense, highly reliable components. An HTTP server with processing power sufficient to implement web services is available in enclosures that include an RJ-45 Ethernet jack [39]. This makes the entire TCP/IP stack and a sophisticated application layer available off-the-shelf for any type of sensor assembly

There has been very substantial increase in processing power for embedded applications. Note that the entire concept of LONTalk was built around a Neuron processor [40-42]. Last year, development of ultra low-voltage CPUs such as Intel Celeron M enabled, for the first time, processing power comparable to that of a desktop PC in mobile and embedded platforms. This allows implementation of complex sensing devices. An example of such a complex sensor could be a high-resolution still-image camera capable of real-time image processing, change detection, and object/face recognition, or a video camera/GPS/IMU assembly (where IMU is short for Inertial Measurement Unit)) supporting real-time geo-rectification, object tracking, image stabilization, and feature extraction. Such complex sensor assemblies appear to be outside of what has been envisioned for BACnet domain. For our pure IP-based BAS handling of such devices is as natural as the handling of a temperature sensor.

There has been explosive growth of web services in form of Service-Oriented Architecture (SOA) [43], which at this point ensures that this particular technology will form the foundation of Internet application architecture for many years into the future.

SMART BUILDING SYSTEM

In this section we will briefly look at the way we approached design and implementation of our BAS, based on Internet Protocol (IP) and open-source technologies (for more details see [1]).

System Goals
While researching the feasibility of creating a BAS with IP-based, open-source technologies we focused on building a prototype of the system for Smart Buildings (hence its name: Smart Building System). The system should provide support for the following features:

Heterogeneous sensors and devices: It should be possible to connect arbitrary IP-based sensors and devices to the system with minimal implementation, configuration, deployment, and maintenance effort.

Personalized and dynamic environment: The system should be able to customize environmental properties of the building spaces to align them with the preferences declared by the individuals. Environmental preferences should form an environmental profile of an individual that would 'travel' with the individual across the building space.

Customizable goal-oriented control: The system should be able to apply 'reasoning' to maintain the environmental parameters (temperature, humidity, etc.) of the building within specified parameters. The reasoning logic should be customizable, so that various strategies can be implemented, analyzed, and compared.

Manual administrative control: The system should provide means for the authorized personnel to monitor the status of the system and set the global, overriding goals for the system., means for direct control of specific devices should also be provided.

Potential of integration with IT infrastructure: The system should provide interfaces for embedding it in the larger enterprise IT infrastructure.

Extensibility: It should be possible to add new or replace existing system components using common software engineering skills.

Technology
We decided to base our implementation on the following technologies:

Java 2 Enterprise Edition (J2EE [26]) is used as a general programming platform for most server-side components of the Smart Building System. This choice was not related to Java's being an object-oriented language, but rather due to the following features and properties of J2EE:
a) support for transactional access to the relational database management systems (DBMS) [31]; b) well-defined communication mechanisms such as RMI, HTTP, Web-Services, and JMS [32]; c) mechanisms for accessing directory services such as LDAP [12]; d) J2EE is a set of specifications instead of products; the code is therefore easy to port to various implementations of J2EE; e) Inclusion or integration with technologies that provide scalability, security, and fault tolerance; f) J2EE is widely used as a platform for enterprise software systems; g) a selection of mature open-source implementations of J2EE platform is available.

Architecture
We adopted a modular approach, where the system components can reside on separate IP network nodes. The Smart Building Framework is at the core of the system. It constructs and maintains an up-to-date model, which represents the controlled environment. It allows components of the system to accurately interpret the data gathered from the devices and influence the environment as intended. The Smart Building Framework, which exposes its services

using web services, is implemented using the J2EE platform. It is deployed on a J2EE application server (JBoss [27]) and communicates with a DBMS (Hypersonic [31]). Third-party modules can connect to the Smart Building Framework over the network and access its services via a Web Services [24] application-programming interface (API).

All other system components interact with the Smart Building Framework (including devices) through its set of web services. We have implemented the following essential components:

The *Inference Engine* [25] implements the control logic of the Smart Building System. Refer to [1] for detailed description.

The *Smart Device* is a software wrapper for any real or virtual device that needs to be connected to the Smart Building System. It can be either deployed as a part of the real device software or it can run on a separate network node the real device connects to. Refer to [1] for more detailed description.

The *Administration Interface* provides a graphical user interface (GUI) for monitoring and controlling the Smart Building System. It has tools for defining target environmental parameters for the system (goal-oriented approach), as well as specific device settings (direct device control overriding settings calculated and enforced by the Inference Engine). It is implemented as a Web-based application (Java Servlet, HTML, and JavaScript) that interacts with the Smart Building Framework to access the information about the system state. It is deployed on a J2EE application server (Jboss [27]).

The *Personalization Interface* provides a GUI for regular users of the system. It allows users to specify their environmental preferences that the system tries to enforce. It is deployed on the Jboss [27] J2EE [26] application server.

DISCUSSION

During design and development of Smart Building System we identified the following benefits and drawbacks of our approach.

Benefits

Re-use of existing IT infrastructure: Most of the server-side components of the Smart Building System run on a standard J2EE-compliant application server commonly used in the enterprise. Any relational DBMS can be used. As a result, the server-side components can be deployed and integrated with an existing IT infrastructure. Standard TCP/IP-based networks can be used to connect devices.

Extensibility: Since all components of the system communicate through well-defined interfaces, they can be easily replaced or updated. New components can be created or existing ones modified using well know programming languages (Java) and communication technologies (web-services). A Java API is provided for creating new Smart Devices to simplify development even further. API’s in other languages can be easily created.

Integration with enterprise systems: The systems based on the Smart Building platform can be controlled through a web-services interface. As a result, they become one more element in the SOA [43] palette in the organization.

Support for legacy devices: Adding legacy devices (BACNet, LONTalk) is possible through adapters. Representation of devices in the platform as well as the communication protocol between Smart Device and Smart Building platform are based on BACNet concepts; therefore effort required to develop of an adapter should be reduced.

Blending of real and virtual worlds: Devices connected to the Smart Building Platform can be both real and virtual. This creates the possibility of making other components of the enterprise software infrastructure behave as virtual devices. For example, a corporate calendar can behave as a device that prepares conference room for a meeting before the meeting participants arrive.

Scalability and reliability: Making Web-based and J2EE based applications scalable and fault tolerant is a relatively well-understood field.

Drawbacks

No support for real-time/priority processing: Real-time/priority requirements are not well supported by existing Web-based technologies.

Latency: The Smart Building Framework introduces an intermediate node in the communication between control logic and target devices. Careful capacity planning is required for mission-critical systems

Higher bandwidth/CPU requirement: Protocol overhead for simple communication is significant. HTTP and SOAP add a significant amount of data (impact on bandwidth and processing power). This is mitigated by increasing CPU power.

Security-related effort increased: We do not consider technologies used to build the Smart Building System inherently less secure than the ones used to build existing BAS deployments, but we believe that more tools for exploiting TCP/IP networks and HTTP-based services are available. Therefore security must be a top priority during deployment and maintenance.

Not feasible for small deployments: Smart Building System requires setup of software as well as hardware components. IT skills are required for setup and maintenance.

Complex handling of system state change notifications: This is more of an architectural issue that increases complexity of the code and bandwidth/CPU requirements. Handling notifications in HTTP-based systems is nontrivial. Several approaches are possible to provide this capability:

- *Polling:* The components of the system learn about system state changes by sending requests to the Smart Building Framework periodically. This approach is supported by our current system. It potentially suffers from high (albeit predictable) latency.
- *Web-services [24] (server)/JMS [32] (client):* Components of the system subscribe to the JMS service maintained by the Smart Building Framework and learn about state changes via JMS messages. This approach, which we consider optimal, has been designed, and is being integrated.
- *Web-services (server)/Web-services (client):* The components of the system expose the web service used for notifications, and register it with the Smart Building Framework. The Framework calls the notification web-service, when a specified system change occurs. This approach requires more work, and we plan to implement it in the next version of the system.

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Recent Developments and Long Term Test Results for an NDIR CO_2 Sensor Based on Tunable Silicon Micromechanical Infrared Filter

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SUMMARY

In carbon dioxide based demand controlled ventilation one of the key system components is the sensor used to measure the CO_2 concentration in the ventilated space. The Non-Dispersive InfraRed (NDIR) technology that is primarily used for CO_2 sensing can provide reliable and stable CO_2 readings for periods well over 10 years, provided that certain inevitable sensor drift mechanisms are compensated for. This paper describes the latest results for an NDIR sensor technology that measures the absolute CO_2 level using a tunable silicon micromechanical infrared filter.

INTRODUCTION

One of the commonly applied Demand Controlled Ventilation (DCV) strategies is to control the ventilation based on the carbon dioxide level in the ventilated area, as the carbon dioxide concentration in a space relatively well represents the person occupancy, and is straightforward to measure compared with other parameters correlating with occupancy. [1,2] Therefore, a key component in a CO_2 based DCV control system is the sensors used to measure the CO_2 concentration in the ventilated space.

For more than a decade the Non-Dispersive InfraRed (NDIR) technology has been the technology of choice for CO_2 sensing in ventilation applications. The basis of the technology is the absorption of IR radiation in gases that follows the well-known exponential behaviour described by the Lambert-Beer equation:

$$I = I_o e^{-\alpha C d} \qquad (1)$$

where

- I_o = intensity of light incident on the sample
- α = absorption coefficient of the gas at the characteristic wavelength [cm^2]
- C = concentration of the gas [$1/cm^3$]
- d = distance the light traverses in the sample gas [cm]

For each measured gas the measurement wavelength must be chosen to match with an absorption band that is characteristic to the said gas. In the case of carbon dioxide the gas features a very strong absorption band at a wavelength of around 4.2 µm. Thanks to the fact that the absorption is strong and the absorption band practically free of interfering gases, CO_2 is one of the gases most commonly measured using the NDIR technology. Figure 1. Depicts the IR absorption spectrum of CO_2.

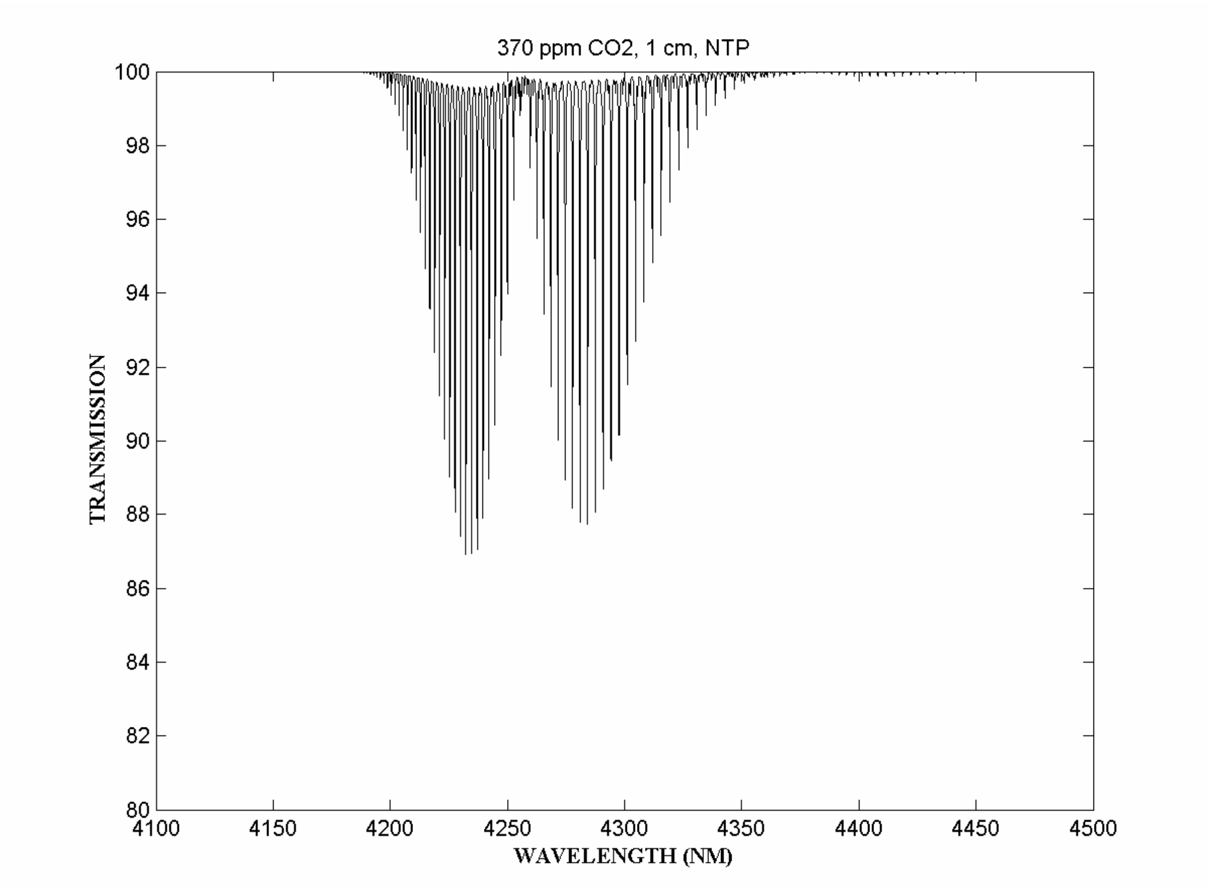

Figure 1. Absorption spectrum of CO_2 in the 4.2 µm wavelength range

Schematically, a typical NDIR based CO_2 sensor for ventilation application consists of three main functional elements: a light source, a measurement channel, and an infrared detector. In most cases the light source is a thermal emitter such as a miniature filament lamp emitting a broad band blackbody IR spectrum, with possible effects caused by absorption in the glass bulb surrounding the hot filament. When considering the long term behaviour of a CO_2 sensor the drift of the lamp intensity over several years is an important drift element that must be compensated for.

The purpose of the second functional element, the measurement channel, is twofold. Firstly, it acts as a light guide and directs light from the light source to the IR detector. Secondly, it acts as a mechanical frame for the sensor, defining the gas volume used for the absorption measurement and providing holes and possible filter elements that are necessary for the sensor to "breathe" with its environment and provide representative measurements. The measurement channel may typically be constructed of injection molded plastic or aluminum, and more often than not contains reflective surfaces used to lengthen or define the optical path of the IR light through the sensor. The optical properties of the measurement channel are crucial for the operation of an NDIR sensor, and may in the long run be affected by corrosion or contamination induced by the operational environment of the sensor.

Finally, an infrared detector is used to measure the intensity of light passing through the measurement channel and the gas therein. The spectral filtering that is necessary for identifying the 4.2 µm absorption band is normally integrated in the detector package in the form of a fixed interference filter or, as in the sensor described in this article, a more advanced tunable filter. Again, any long term drift mechanisms and effects in the sensitivity of the IR detector or in the bandpass characteristics of the filters must be minimized and / or compensated for in order to realize a stable CO_2 sensor.

In practice the different sensor structures and compensation schemes that are being used today in commercially available NDIR CO_2 sensors differ from the simple picture presented above. Dual light sources may be used to compensate for light source aging in a scheme where the second light source is operated only intermittently in order to minimize its aging and hence provide a possibility to use it in correcting for changes occurring in the other parts of the sensor. On the other hand, dual detectors furnished with different filters for measuring the IR transmittance at a CO_2 sensitive wavelength (4.2 µm) and at a reference wavelength typically adjacent to 4.2 µm may be used for compensating long term effects. Finally, software methods and algorithms can be used for post-correcting CO_2 readings from a drifting sensor in applications where the occupancy pattern in a building permits the use of the background CO_2 level at the building location as a baseline reading to which the sensor reading is adjusted, typically at night time.

In this paper we present the operation and latest measurement results of an NDIR sensor technology that uses a tunable IR filter as a means to provide a dual wavelength (CO_2/reference) operation with a single optical path and a single IR detector. The voltage tunable IR filter is made using the Fabry-Perot Intereferometer (FPI) principle, and fabricated on silicon with the methodology of silicon surface micromechanics.

METHODS

Figure 2. depicts the structure of the sensor used in the study. The light source is a miniature filament light bulb, and the measurement channel is constructed of injection molded plastic that is gold plated from the inside to provide a reflective surface which does not age even in

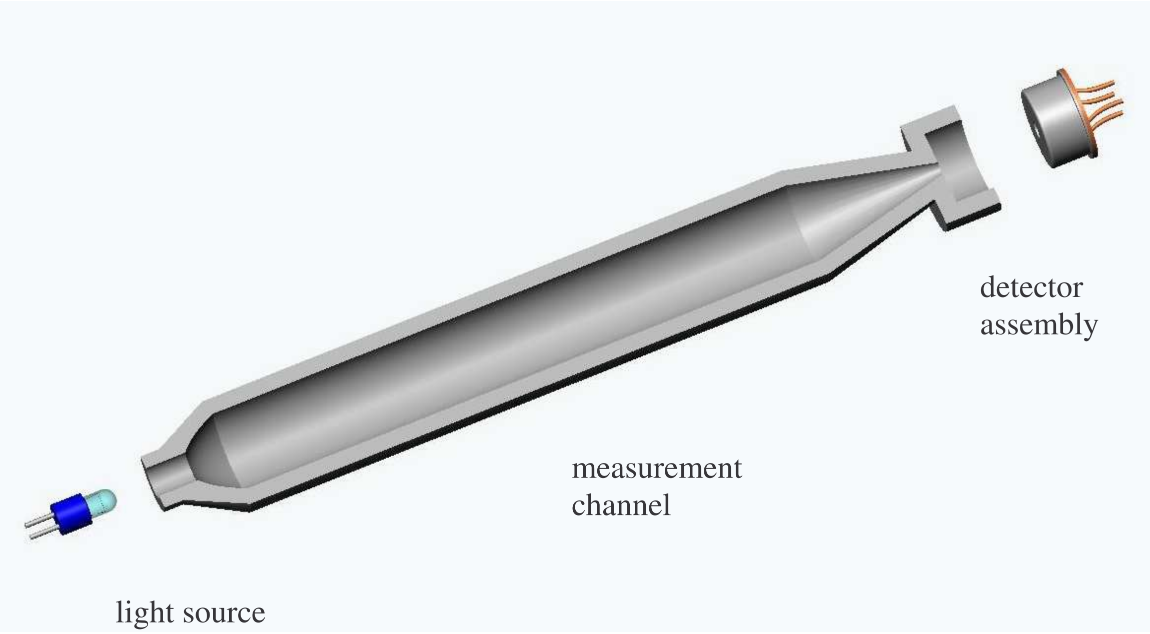

Figure 2. The structure of the NDIR CO_2 sensor used in the study.

harsh environments. The light receiving end of the sensor is built inside a hermetic TO can, which hosts a window for light entry into the can, the tunable FPI filter and an IR sensor. A schematic structure of the detector TO can assembly is presented in Figure 3.

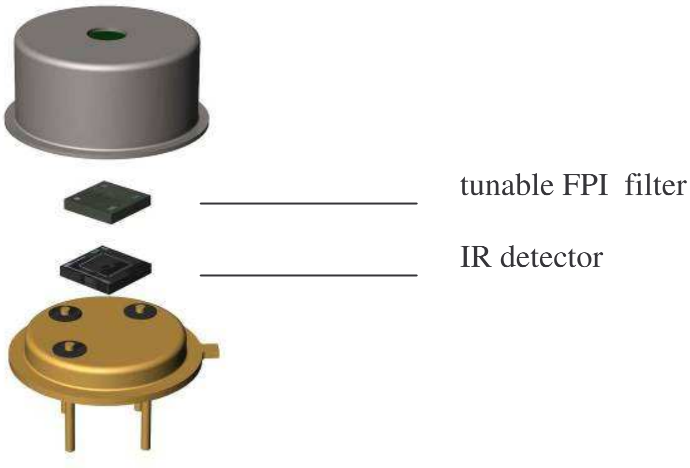

Figure 3. The detector assembly hosting the entry window, tunable FPI filter and IR detector

The tunable FPI filter that is used in the sensor is fabricated on a silicon wafer using silicon surface micromechanics. Figure 4. depicts a schematic structure of the filter.

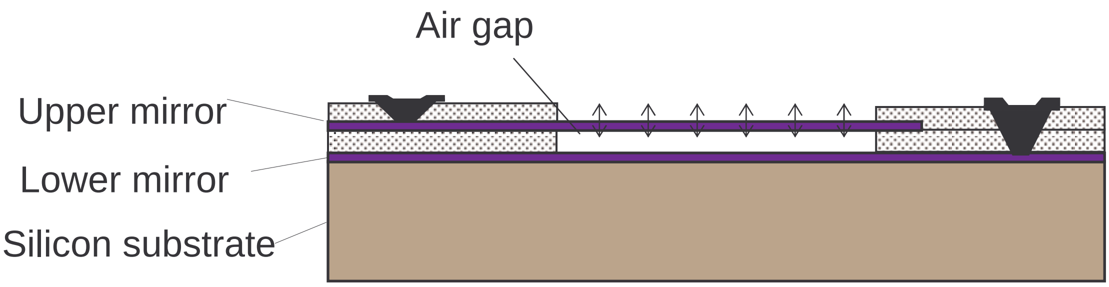

Figure 4. Schematic of the FPI filter used in the sensor

In the component a thin diaphragm is suspended in the vicinity of a silicon substrate to form a narrow air gap. According to the operating principle of a Fabry-Perot interferometer [3] the two closely spaced mirrored surfaces form a bandpass filter whose transmission depends on the distance between the said surfaces. With a judicious choice of the gap dimension the filter can be optimized to operate at the CO_2 absorption wavelength of 4.2 µm. Tuning of the filter is achieved by connecting a voltage across the air gap and displacing the diaphragm electrostatically towards the substrate and hence changing the gap.

For CO_2 sensing the IR transmission through the sensor is measured at the CO_2 wavelength of 4.2 µm and at a reference wavelength near the CO_2 wavelength. The ratio of these two readings is a measure of the CO_2 concentration that is free from many drift mechanisms of NDIR sensors such as light source aging or effects from dust, dirt or even corrosion inside the sensor.

RESULTS

Despite the fact that the operating principle of the sensor in this study inherently compensates drift mechanisms that are caused by for example deterioration of the optical surfaces inside the sensor, for extremely stable measurements it is advantageous to choose the surface coatings inside the sensor in such a way that they resist corrosion. To study corrosion resistance of the coatings we exposed one set of injection molded and gold plated production sensor parts to the following set of accelerated aging conditions, in succession:

1) 3 weeks of dry heat; 90 ^{o}C
2) 3 weeks of damp heat; 90 %RH, 50 ^{o}C
3) salt mist test according to IEC 68-2-11: two cycles of (24 hours of salt mist (5% NaCl, 35 ^{o}C), 24 hours drying (23 ^{o}C))

As a documentation of the results figure 5. shows two plastic parts, one of which is virgin with no aging treatments, the other one after exposure to the aging conditions described above. No effects are visible from the relatively serious environmental tests.

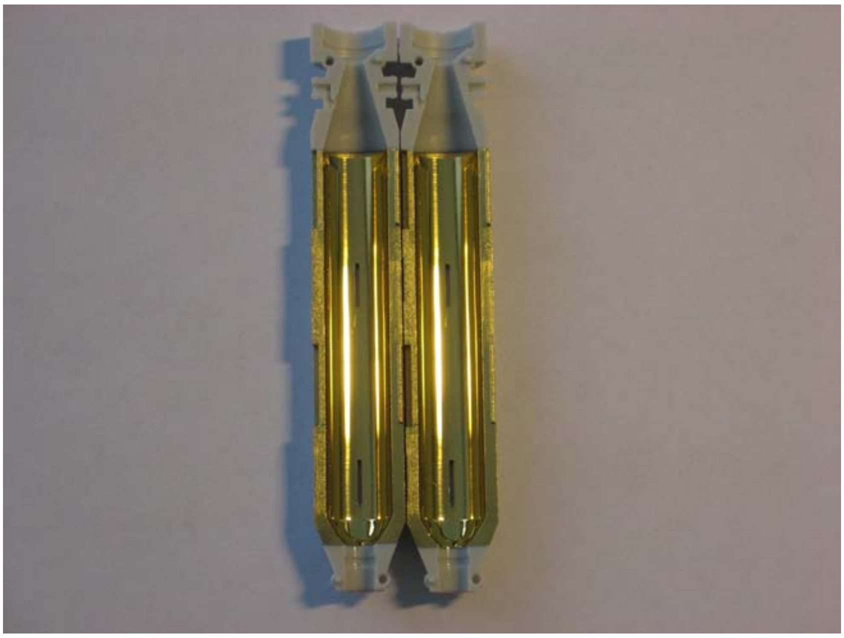

Figure 5. Plastic parts before and after exposure to accelerated aging tests. Part on the left has no exposure, part on the right has been through aging treatments described in the text. No effect is visible.

In assessing the long term stability of an NDIR CO_2 sensor also the sensing algorithm used to extract the CO_2 concentration from the transmitted IR signal(s) is of importance. Furthermore, also the implementation of the algorithm and practical realisation of the whole instrument should be taken into account. Hence for meaningful determination of the long term stability of a sensor structure and technology, a minimum unit for testing is a whole CO_2 transmitter giving out an analog or digital CO_2 signal.

We have installed four preproduction transmitters in an office environment for continuous logging. Data is recorded at 3 am when typically no people are present in the office area. The forced ventilation in the office space is on during office hours and until 8 pm, whereas during night time the mechanical ventilation is off. Figure 6. depicts raw data from the four transmitters from an ~8 month period, as recorded at 3 am daily. The data shows relatively large, +/- 50 ppm, daily variations in the background CO_2 concentration that can be seen in the data from all the devices under test. These can be understood as natural variations in the ambient CO_2 concentration due to natural (photosynthetic activity, soil respiration) and man-made (traffic, energy production) phenomena. Additionally the data would indicate that there has been an effect on the background CO_2 concentration in Helsinki (where the measurements have been made) due to the large forest fires that took place in Russia at the end of August 2006 and had a significant effect on the air quality in Helsinki at the time.

To more accurately assess the long term stability of the devices we applied a correction to the raw data, where the common mode variations that were seen in the readings of all transmitters were compensated in the readings. The resulting common mode corrected data is seen in figure 7.

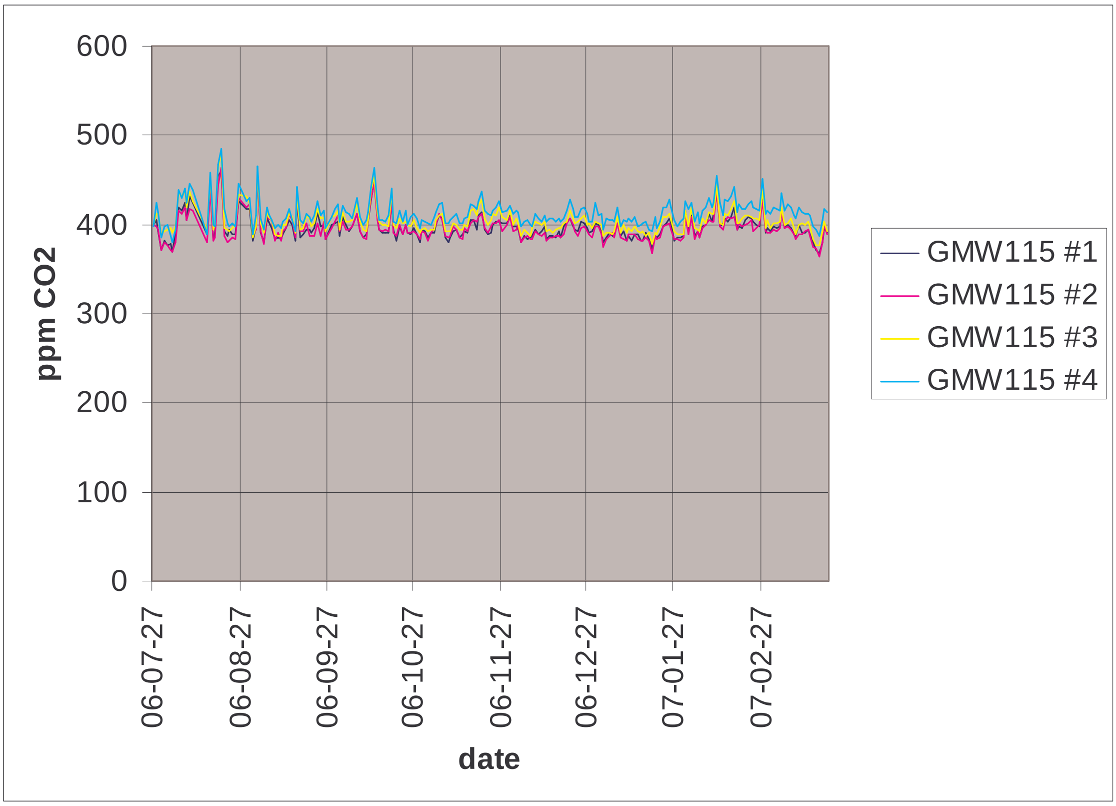

Figure 6. CO_2 readings from the tested units for an ~8 month period, raw data

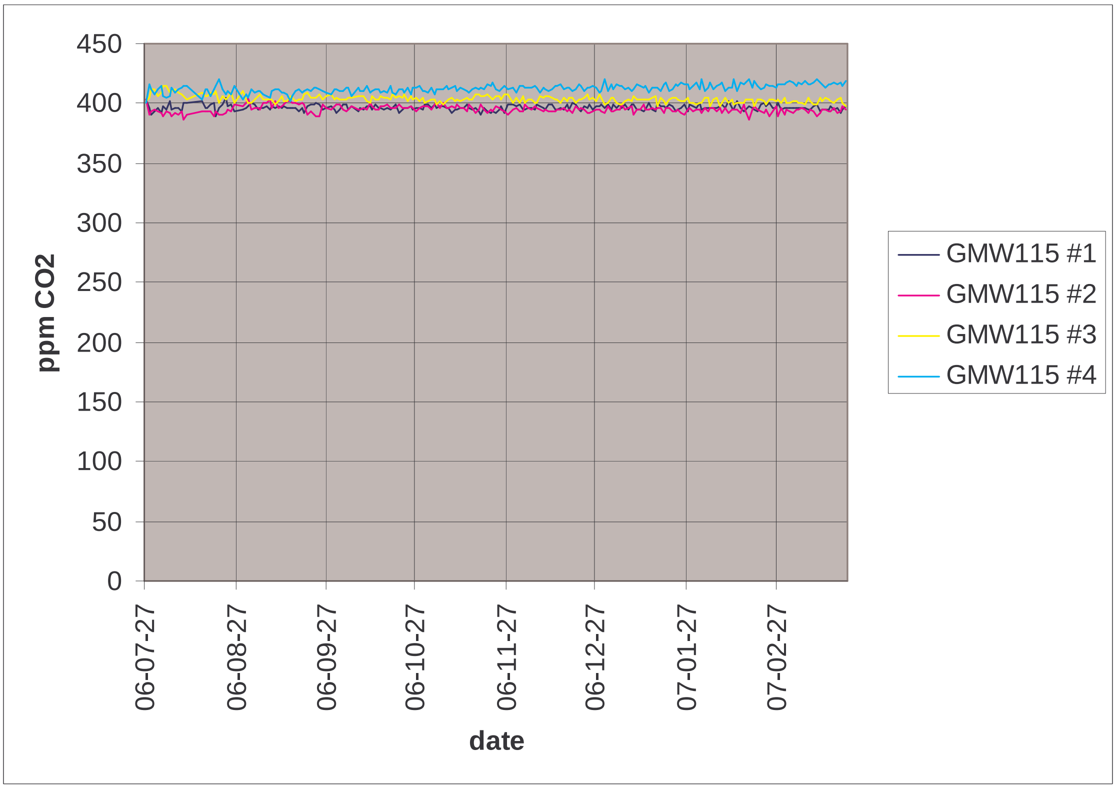

Figure 7. CO_2 readings from the tested units for an ~8 month period, data compensated for variations in the background CO_2 concentration.

In a closer analysis of the data the devices exhibit drift in the readings that vary from 0 ppm (device #1) up to 7 ppm (device #4) over the measurement period. Given a typical accuracy specification for an HVAC CO_2 transmitter (at ambient concentration) of 30-60 ppm, and a long term drift specification for the products in question of 20 ppm/yr, it is to be noted that the drift is well within specified limits and supports the long term use of this type of instruments in Demand Controlled Ventilation applications. A further notion is that in this type of devices the drift phenomena are typically not linear but tend to be faster in the early life of the instrument and then even out in a longer term.

DISCUSSION

Intermediate results from a long term stability study for NDIR CO_2 sensors for Demand Controlled Ventilation applications have been reported. An absolute CO_2 concentration measurement is achieved with a simple sensor structure with no moving parts or software compensation algorithms. In the light of the results presented here the devices show a very adequate performance for the DCV application.

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Case study of the shopping centre HVAC systems electrical loads

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SUMMARY

This paper reports on energy aspects of the HVAC systems in shopping centre. Presented analysis of the electrical loads shows possibilities to increase energy efficiency of the HVAC systems and to keep desired indoor air quality. This particular case study is done in the shopping centre located in Riga, Latvia, which is 2-floors building with the total area of 22,000 m^2. The shopping centre lighting loads are more than 50 W/m^2, which requires 1.1 MW cooling capacity in summer and just 270 kW heating load in winter.

Building management system (BMS) software stores trendlogs of all HVAC system parameters such as status of fans, fancoils, chillers, pumps and electrical heaters. The analysis of the trendlogs is fulfilled for estimation of energy consumption of the HVAC systems and for optimisation of electrical loads.

INTRODUCTION

The main objective for HVAC and control designers was to maintain 21°C indoor temperature all year round using demand controlled ventilation system (maximum air exchange rate = 1.5) and hydronic ceiling fancoil with cooling / heating system. During summer water is cooled with two chillers, during winter water is heated with electrical heater. There are no air heaters in the air-handling units. The air-handling units maintain supply of air temperature by exhaust air heat recovery and variable air volume by frequency converters. Each of six air handling units serves to an independent zone of the shop. Control is based on CO_2 gas level and temperature analysis. Hydronic cooling / heating system is splited to the same zones for maintaining the same zone temperature. All these systems are controlled from BMS. Premises are open without separated rooms, except some small office rooms on the second floor. Case study of electrical loads is carried out in January 2007 from 7.00 – 22.00, requirements for indoor climate is 21°C.

METHODS

BMS trendlogs are saved to computer as csv files. With application of Microsoft Excel and Microsoft Visual Basic these data are processed and saved as “mat” files for calculation purposes with “Matlab”.

LIGHTING HEAT POWER

The total amount of installed lighting heating power is 50 W/m². Lighting is switched on from 10.00 in the morning and turned off at 22.00. During the night there is a night lighting with heating power of 5 W/m².

AIR HANDLING UNIT ELECTRICAL DATA

Six air handling units are installed in shopping centre.

Table 1. Air handling units data

Air handling unit with the number of zone	Electrical power of supply / exhaust fans, kW	Total pressure, Pa	Air volume of supply / exhaust fans, m^3/h	Temperature efficiency of rotating heat exchanger, %	Humidity efficiency of rotating heat exchanger, %
AHU1	11 /11	891	25000	72	63
AHU2	11 / 11	891	25000	72	63
AHU3	18.5 / 18.5	842	40000	74	66
AHU4	18.5 / 18.5	842	40000	74	66
AHU5	4 / 4	702	10000	73	65
AHU6	4 / 4	702	10000	73	65

Air handling units are equipped with frequency converters. Rotating heat exchangers are rotating proportionally according to necessary supply of air temperature. Air handling units work daily from 7.00 – 22.00 Normally air handling units work with 75% air volume.

AIR HANDLING UNIT CONTROL ACCORDING TO CO_2 LEVEL IN SHOPPING PREMISES

Shopping centre is divided into six zones. Each air handling unit serves to an individual zone. CO_2 gas detectors are located in each zone, except zones 5 and 6, which work from one CO_2.gas detector, air-handling units' assignment is not to allow the CO_2 level increase more than 800ppm. CO_2 trendlogs show, that only in zone 1 and in zone 2 CO_2 increases more 800ppm (figure 1). These zones are located near to the main entrance, and require to be ventilated more than other zones.

Measurements show that CO2 level is a bit various from day to day and it increases more than 800ppm every day in zones 1 and 2. Average calculations of CO_2 gas level in each zone show, that CO_2 increases during the day and ventilation system does not allow it to overreach 800ppm.

From (figure 1) we can analyse that only AHU1 and AHU2 partly work with 100% of air volume because of increased CO_2 level in zones 1 and 2. Zone 1 has increased CO_2 level about 7h a day, and it means that AHU1 works on maximum speed 7 hours a day, 9 hours a day AHU1 uses 70% of its nominal power. Zone 2 has increased CO_2 level about 5h a day, and accordingly AHU2 5 hours a day works on maximum speed, 11 hours a day AHU2 uses 70% of its nominal power. All other air handling units are using just 70% of their nominal power.

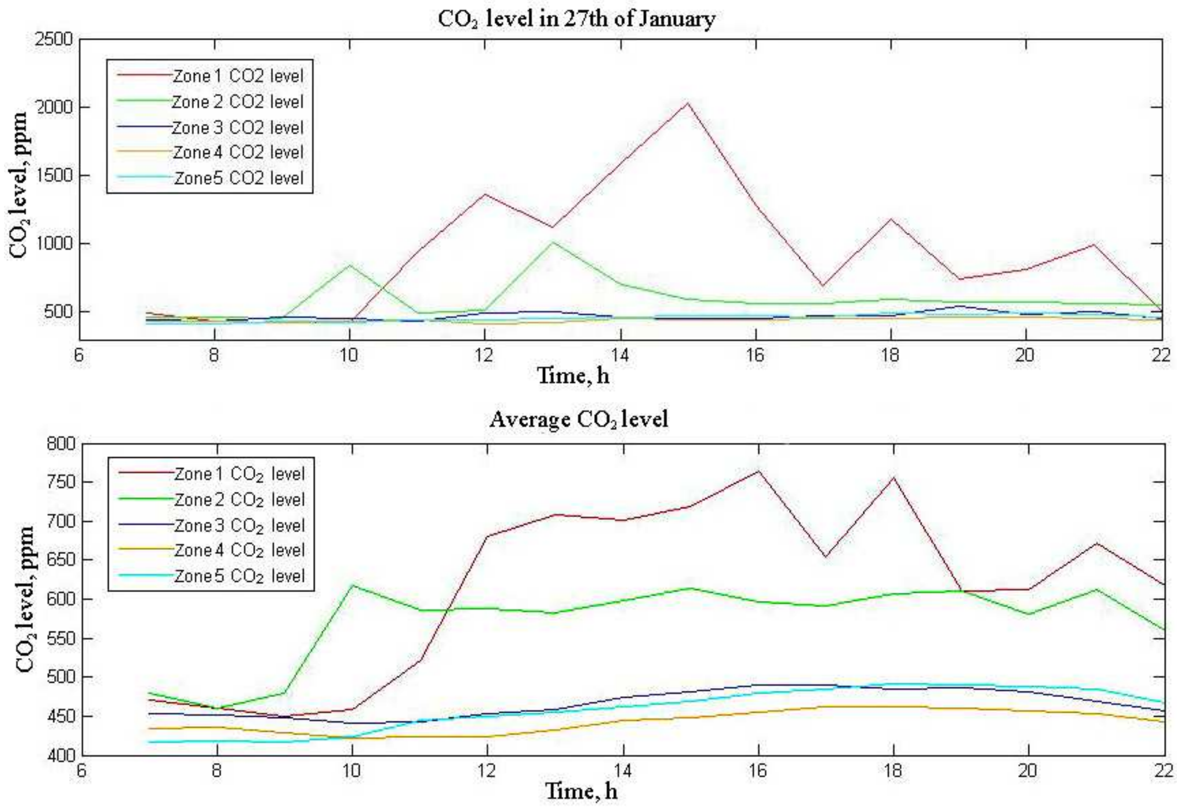

Figure 1. CO_2 gas concentration

CLIMATE IN JANUARY

Climate is measured in shopping centre with relative humidity and temperature sensors. Absolute humidity and enthalpy are calculated [3].

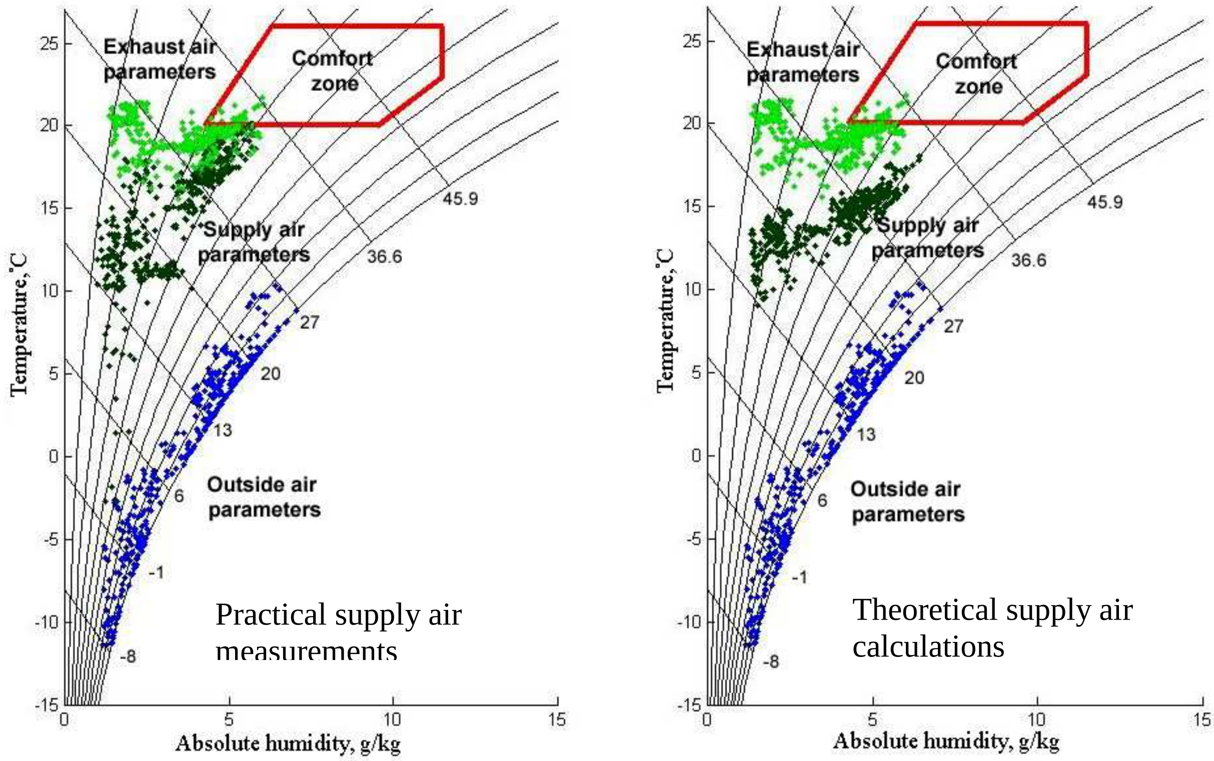

Figure 2. Air handling unit AHU2 supply and exhaust air analysis.

On the left side of Figure 2 there are data from AHU2 measurements (BMS stored trendlogs). Ceilings of premises are 6 m high, chosen air diffusers distribute air evenly, that allows to limit the supply of air temperature of air handling unit to 10 C°. On the right side of Figure 2 there is a theoretical work of air handling unit No. 2. if the speed of heat exchanger wheel is 100%. Such theoretical analyze allows us calculate air handling units effect on cooling the premises. Practically the speed of heat exchanger wheel depends on zone temperature and zone setpoint. Analyze of the average speed of air handling units' heat exchanger wheels is done in Figure 3. Temperature in premises is heated with lighting, cooled or heated with hydronic cassettes. Figure 3 shows, that heating loads are not even in zones. Air handling units No.5 and No. 6. heat exchangers work on maximum temperature efficiency, as zones No. 5 and No. 6 requires heating because of less lighting loads in these two zones. Average exhaust of air temperatures shows that heating capacity of building is enough to keep the necessary supply of air temperature.

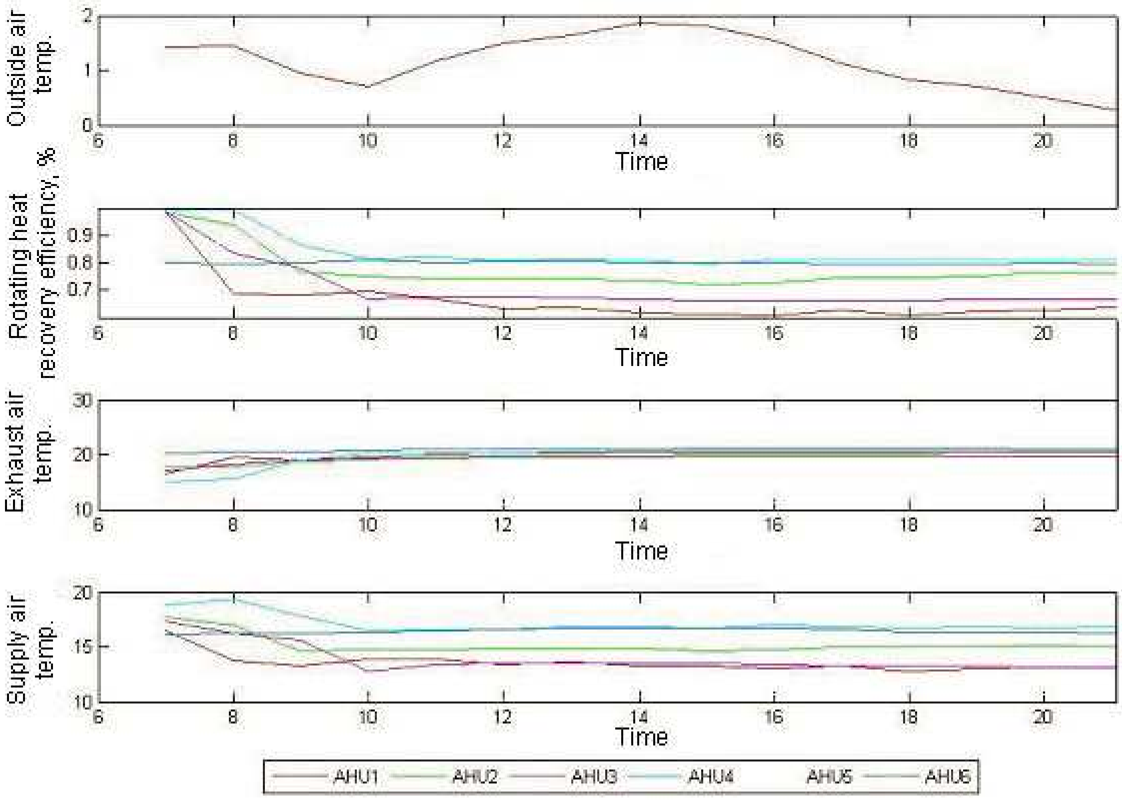

Figure 3. Air handling units average parameters in January.

Air handling units consume just electrical power, as they are not equipped with heating or cooling coils. Also as supply air temperatures are less than exhaust air temperatures, air handling units are cooling premises.

Table 2. Power analysis of air handling units

Air handling unit number	Air volume m^3/h	Electrical power, kW	Cooling power of the unit, kW
AHU1	21 560	15	50
AHU2	20 620	13	48
AHU3	30 000	15	69
AHU4	30 000	15	69
AHU5	7500	3,2	17,2
AHU6	7500	3,2	17,2
Total	**117 180**	**64**	**270**

Average supply and exhaust air temperature and humidity parameters gives information about heat and humidity emission in building premises. We can see (Figure 4) that inside air humidity depends on an outside air humidity, because humidity emission from people is quite small.

Table 3. Average supply / exhaust air parameters

	Average air temperature °C	Average air humidity, g/kg	Enthalpy kJ/kg
Supply air	14,88	3,42	23,61
Exhaust air	19,45	3,76	29,09

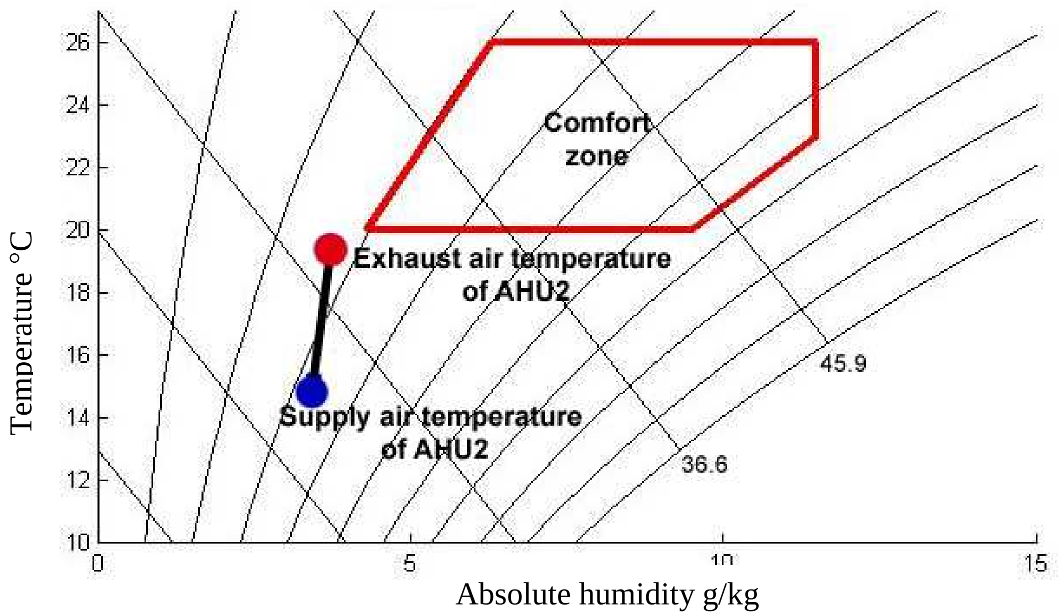

Figure 4. Graphical analysis of supply / exhaust air parameters.

Measured room temperature is a little higher than exhaust of air temperature, because technical rooms are located on the roof, air ducts to roof go through outside. Despite of air ducts insulation, exhaust of air has temperature decrease in 2°C. Such decrease of temperature decreases supply air temperature by 20%. Perfect insulation of building and air ducts could decrease average ventilation system cooling power of premises for 50kW in January.

HYDRONIC COOLING / HEATING SYSTEM

Hydronic system consists of:
Chiller No.1 with free cooling option, cooling power is 820kW (free cooling power is 387kW), electrical power is 315kW ; chiller No.2 without free cooling option, cooling power is 338kW, electrical power is 150kW; Electrical heater is 270kW, 3 steps control. Premises are heated / cooled with 180 units of ceiling cassettes, which are splitted evenly inside the building. Maximum cooling capacity of each ceiling cassette is 7kW.

During winter time, Chiller No.2 is switched off. Chiller No.1 works according to the time schedule from 10.00 – 21.00 if there is a requirement for cooling. Electrical heater is turned on, when there is requirement of heating in some zone. Before turning electrical heater on, Chiller No. 1 is turned off. Control of ceiling cassettes is divided into 6 zones, each zone consist of 30 units of cassettes. Change of speed and temperature control is adjusted from BMS for each zone. Temperature is regulated with one two-way valve for each zone. Theoretically maximum heating capacity of each cassette is 1.5kW, if all cassettes work with the same speed and two way valves are open. Practically heating power of each zone depends on the situation in other zones. Each chiller accumulation tank has its own circulation pump of 5,5kW with electronic control. Pump water flow corresponds to chiller requirements and pump work together with chiller. Three parallel working pumps with frequency converters (each 15kW) are installed after accumulation tank. Control of pumps is done with frequency converters for maintaining necessary pressure difference of 115Pa.

BMS trendlogs shows that to maintain 115Pa it is enough to run just one pump with 33Hz, which corresponds to 61m^3/h.

Cooling / heating power calculations:

$$Q = V \cdot \rho \cdot c \cdot \Delta T \, , \qquad (1)$$

where V is water flow (m^3/h), ρ is water density (1000 kg/m^3), c is water heating capacity (4.18 kJ/kg*K), ΔT is water temperature different between supply and return pipes.

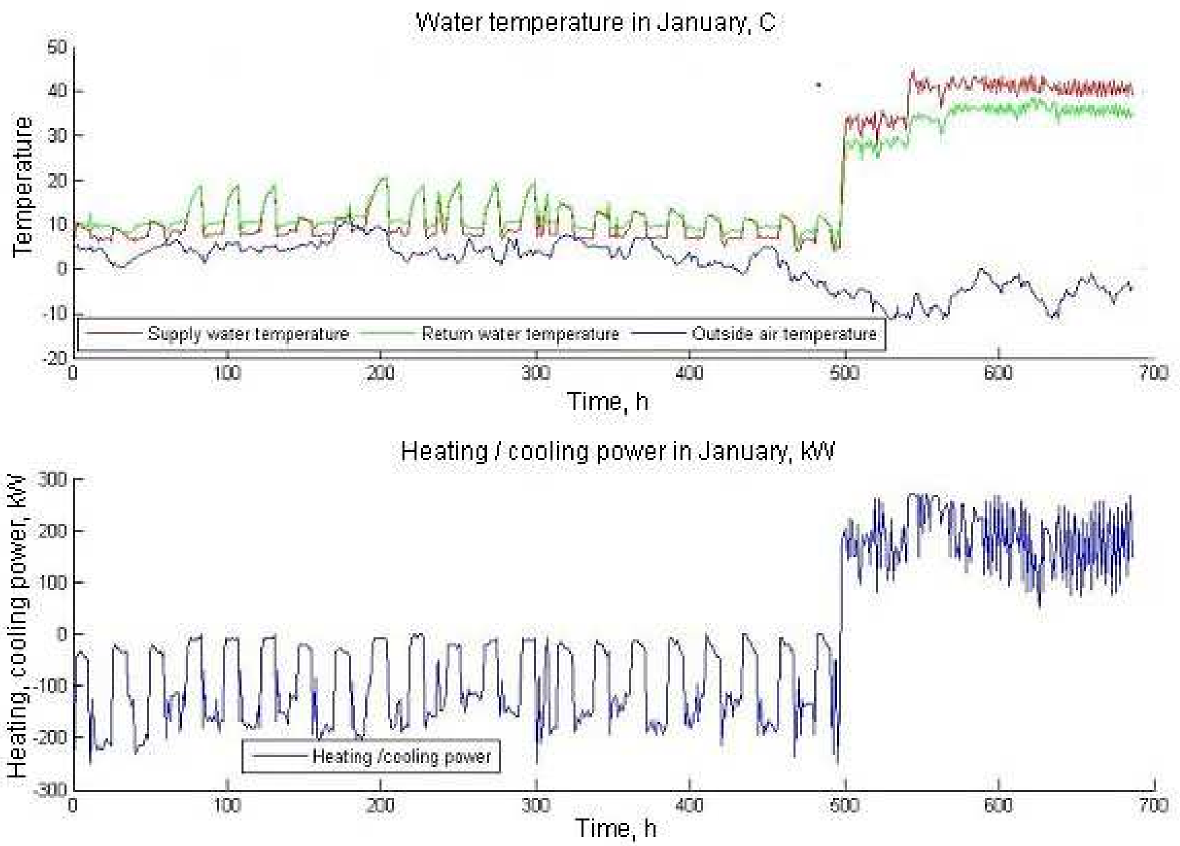

Figure 5. Heating / cooling power analysis in January.

Switching from cooling to heating is done automatically (figure 5), when the outside temperature is less than –5 °C. At this temperature there is a requirement for heating zones No.4-No.6. Shopping centre is heated also during nights. During days heating load is decreased because of high lighting loads.

RESULTS

Results show that it is possible not to use heating coils in air handling units for ventilation of shopping centers with high lighting loads. At least 30% of consumed air handling unit power can be saved with installation of CO_2 gas detectors. During design phase it is not possible to predict number of people in shopping centre, that's why air exchange volume can be higher that necessary. Inside absolute humidity level depends on outside humidity, measured process direction (figure 4) of supply / exhaust air can be used in design of shopping centers ventilation systems. Different lighting loads in building forces to use electrical heater, when outside temperature is below -5°C. Following research will investigate more deeply necessity of switching between cooling and heating. The air temperature in premises is well controlled with air handling units together with hydronic heating / cooling system and the air temperature in premises is distributed almost evenly in all zones of building.

HVAC electrical loads in shopping centre during winter time is about 70W/m².
Practical investigations show, that –5°C is a temperature, where zone 1 and zone 2 still needs to be cooled, but zones 4 – 6 need to be heated. This is the main reason, why it is not possible to keep temperature in premises just with ventilation system therefore fast switching between cooling / heating system is necessary.

CONCLUSION

Control system, which is used for ventilation, air conditioning and heating system automation in shopping centre, is best designed with effective supervision software. Energy analysis cannot be done with temperature sensors, but there are also others requirements for installation of relative humidity transmitters for outside, supply and exhaust airs. Electrical energy meters for air handling units, chillers, electrical heater and lighting electrical consumption with connection to building management system could give very important information for electrical loads.

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Evaluation of the PID and On-off control logics in the environment conditioning using a thermal storage system with ice bank

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SUMMARY

The thermal storage systems have as main objective the rationalization of the demand of electric energy, due to the use of it in periods of the low demand, beyond the reduction of the initial investment. One of the means more used for accumulation of latent heat is the ice, due to its bigger capacity of storage of energy in a single volume and also because it always casts in a fixed temperature. The conventional logics of control are widely used in accordance with the necessity of the process. In this work it was used thermal storage with ice bank and it was evaluated the efficiency of the performance of controllers On-off and PID in the control of the temperature of three different environments, after ambient disturbances not measured of thermal load provoked by the variation of the external ambient temperature and/or equipment operation, activities of people and others.

INTRODUCTION

The GREGOR *et al*. [1] showed that the equipment and systems that supply thermal comfort and internal air quality for commercial environments, consume about 42% of the used energy total in the majority of conditional environments, therefore they are the main point in the discussions on energy consumption.

A thermal storage system is that one that exchanges heat with the accumulator mean during the lower demand periods, storing energy at low temperature, “cold”, to be used in the higher demand periods [2].

The ways commonly used for latent heat accumulation are: water-ice and phase change materials (PCM). These ways differ in the amount of stored energy per volume unit, in the temperature in which they store “cold” and “heat”, and in the physical conditions of the stored energy [3].

In the case of thermal storage with ice, the frozen water temperature which flows in the air conditioners is lower, reducing the ventilated air outflow for the environments, consequently reducing the costs of the fans and in the ducts, what it allows an environment humidity reduction and an increase of the comfort and the interior air quality [4].

The main objective of a control system is to make the process operate in a steady way, in accordance with the established values for process conditions and variables [5]. The used control logics can be conventional or not conventional, and the conventional ones are On-off, P, PI and PID types, and they can be very used depending on the process necessity [6].

The buildings thermal automation increases potentiality the energy management, through the supervision graphical screens implementation that provide a general monitoring of the system, resulting in a better control of the energy use and in a higher thermal comfort to its occupants.

Considering what have been showed, the experiment proposes to evaluate the temperature behavior of the rooms 1, 2 and 3, using the PID and On-off control logics with 0,5 ºC band in operational conditions of commercial environments. These environments will be submitted to changeable thermal load, which it's directly related to the external environment, with its occupants and the materials inside the rooms, whose influence in a thermal load increase.

METHODS

In this work the ice bank was used as the thermal storage system, for temperature conditioning of three different environments. Its load was done during the night, when the refrigeration system cooled a propylene-glycol solution that was pumped internally to an immersed coil in a tank with water. Then ice was formed externally on the coil surface, resulting in a mixture of ice and water at 0ºC.

During the day, the water of the tank was pumped to the environments, where the fan-coils were located, which their function was to do the distribution of the "cold", proceeding from the frozen water, in a uniform way to all the environments. The scheme of cooling water coming from the city treatment stations and the distribution of the water frozen to the study environments can be seen in Figure 1.

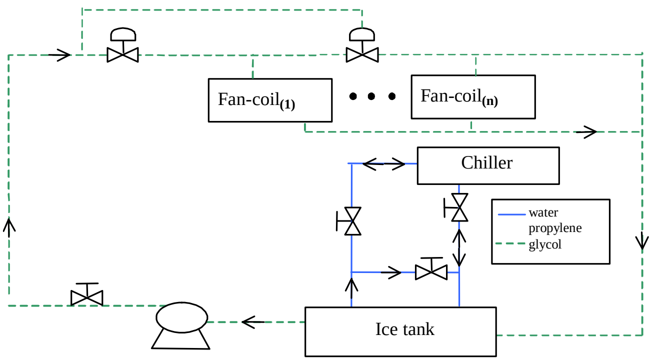

Figure 1 - Schematic diagram of the thermal storage circuit and frozen water distribution.

The temperature of each environment was controlled through On-off controllers with 0,5 ºC band and PID, where those acted directly on the control valve of frozen water outflow. This frozen water is proceeding from the ice tank, and then it's pumped and sent to the study environments, in accordance with the necessity for the desirable temperature maintenance. The PID was tuned by the Ziegler Nichols Reaction Curve method, where it makes step disturbances in the manipulated variable and observes the received reply from the controlled variable.

The On-off controllers evaluation with 0,5 ºC band and PID was made after the ice bank load, with disturbances in the frozen water distribution, then verifying the stabilization of the rooms temperatures around the set-point. These disturbances were performed during the commercial period (8:00 A.M. to 18:00 P.M.) and the imposed thermal load to environments was proceeding from the temperature variation of the external environment, daily activities of the people and equipment.

RESULTS

After the non-measured disturbances with thermal load for all the environments, and both On-off controllers with 0,5 ºC band and PID, it was gotten the internal temperatures behavior for each environment. Figure 2 shows the internal temperature behavior of room 1 under controller PID performance.

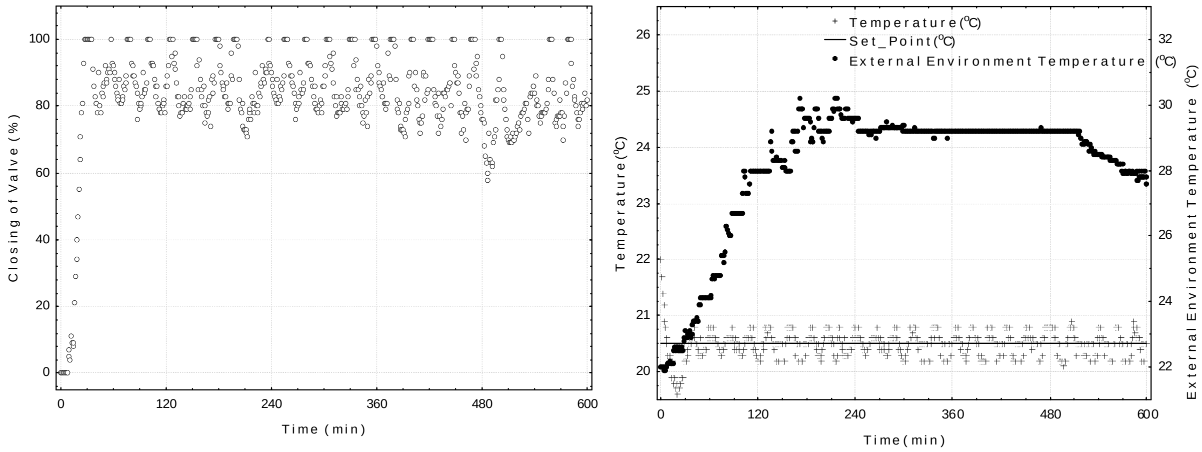

Figure 2 - Internal temperature behavior of room 1 under controller PID performance.

The beginning of the experiment occurred with the definition of the room temperature set-point in a value below the environment value. Thus the controller action, in the beginning of the environment cooling process, was opening the valve until its temperature to reach the desired value. Then the controller modulated the frozen water valve outflow so that the environment temperature was kept around the set-point, even with great variation of the external environment temperature.The Figure 3 presents the internal temperature behavior of room 2 under controller PID performance.

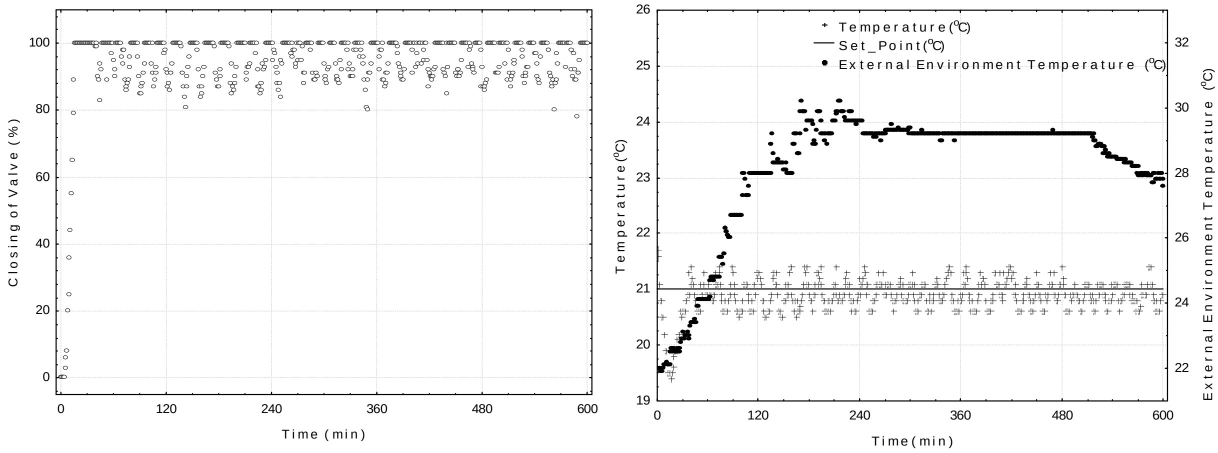

Figure 3 - Internal temperature behavior of room 2 under controller PID performance.

For the room 2 environment the controller performance is also observed so that its temperature reaches the desired initially value. However, there’s a better stabilization around the set-point due to the lower heat variation tax in this environment what it is directly related with lower solar incidence index and consequently lesser control valve modulation.

Figure 4 shows the internal temperature behavior of room 3 under controller PID performance with the respective performance of its control valve, which allows higher or minor frozen water outflow in accordance with the measured error by the controller.

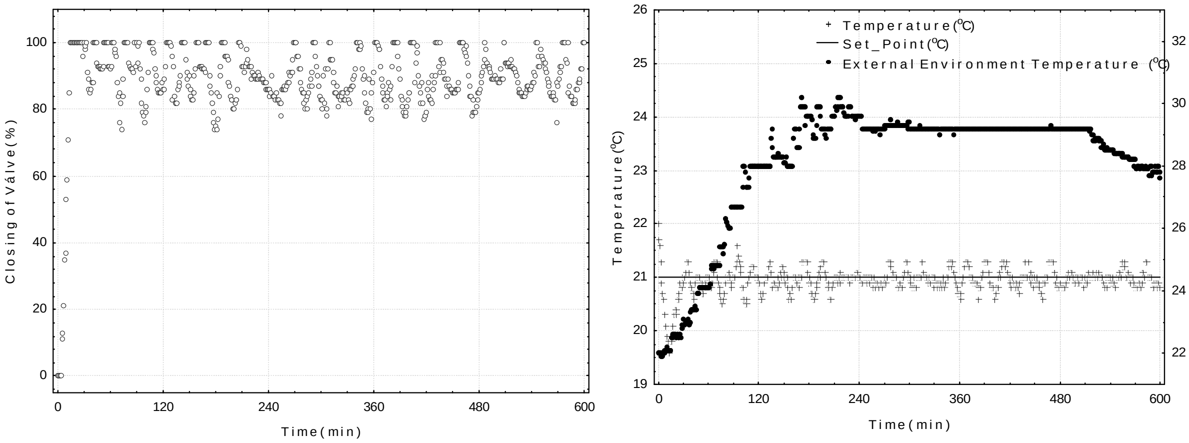

Figure 4 - Internal temperature behavior of room 3 under controller PID performance of.

Initially it was verified the control valve opening with the release of a higher frozen water outflow, occurred in function of the great value of the error (desired value little real value) in the beginning of the process. That caused the reduction of the study environment temperature.

With the reduction of the temperature, the controller compared the desired value (set-point) with the real value resulting in lower error values. In this way, the control valve modulation started to be made thus the temperature of the study environment was kept oscillating around the set-point with small variations, during all the conditioning process. In Figure 5 the internal temperature behavior of room 1 under the On-off controller performance with 0,5 ºC band can be observed.

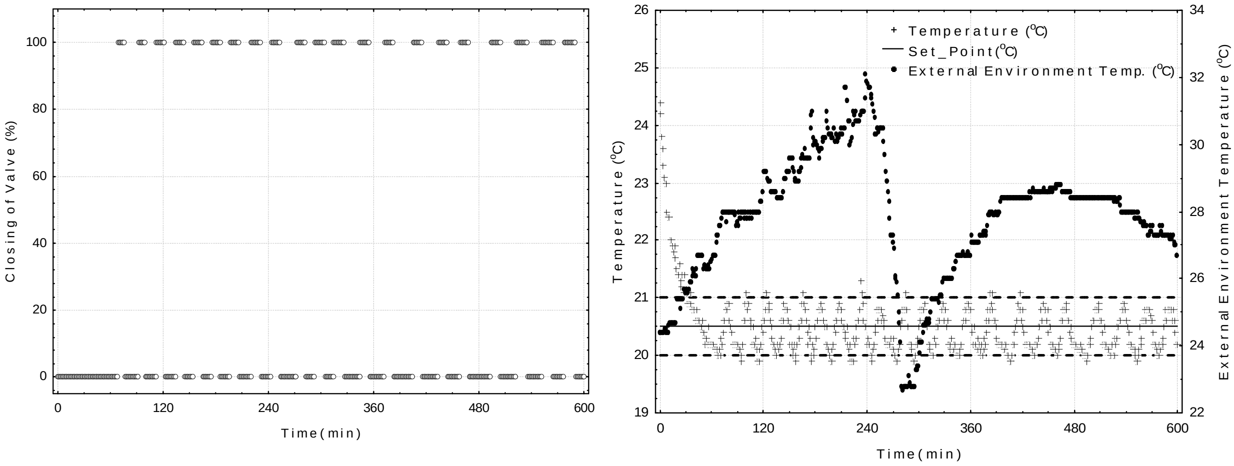

Figure 5 - Internal temperature behavior of room 1 under On-off controller performance with 0,5 ºC band.

It is verified initially that there’s reduction of the internal environment temperature until the desired value and then starts the modulation of the water frozen outflow control valve for the maintenance of the environment temperature around set-point. It can be evidenced that the controller acted in a way to stabilize the set-point between the established band, even with a brusque variation of the external ambient temperature in a certain day period, provoked by a sudden rain. In Figure 6 the internal temperature behavior of the room 2 under On-off controller performance with 0,5 ºC band can be observed.

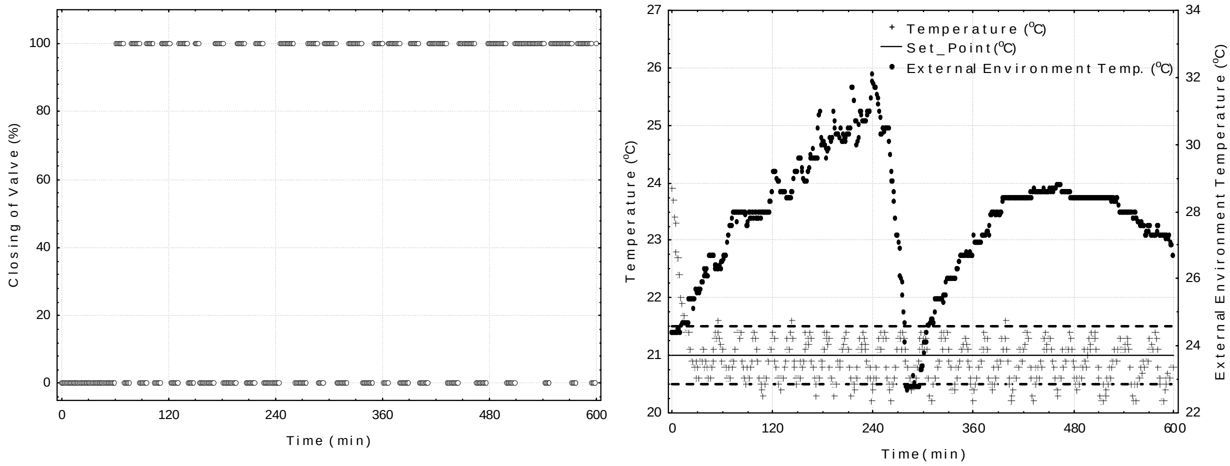

Figure 6 - Internal temperature behavior of room 2 under On-off controller performance with 0,5 ºC band.

For environment 2 it can be evidenced that even with the control valve of water outflow frozen closed during most of the experiment time, there was a higher temperature oscillation of this room comparing to room 1. It happened due to the proper controller, who is on-off type, where the temperature variation can reache values beyond the programmed band (0,5 ºC), and also due to a higher tax variation of the environment temperature, provoked by the greater number of people present. Figure 7 presents the internal temperature behavior of room 3 under On-off controller performance with 0,5 ºC band, with the respective performance of its control valve, that allows the passing of a higher or minor frozen water outflow in accordance with the existing temperature difference.

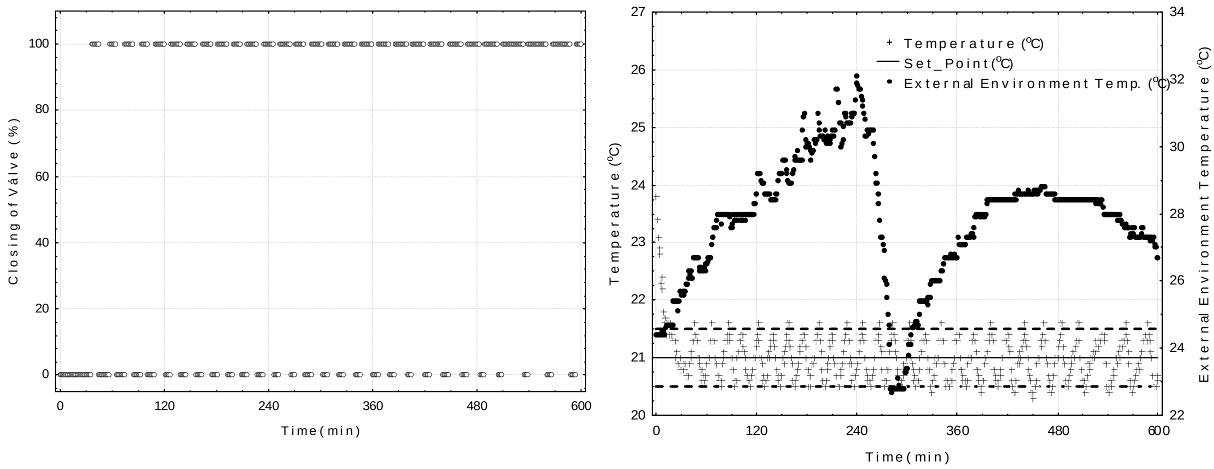

Figure 7 - Internal temperature behavior of room 3 under On-off controller performance with 0,5 ºC band.

It can also be observed for this experiment that in the beginning of the experiment the environment temperature was sufficiently high, needing a faster action of the controller in opening the control valve of frozen water outflow, thus remaining open for a higher time period, when compared with PID controller for the others experiments, that worked with conditions more suave f the internal and external ambient temperature.

DISCUSSION

From the results gotten for both controllers, On-off with 0,5 ºC band and PID, tuned by the Ziegler Nichols Reaction Curve method, can be concluded that those behaved in satisfactory way, keeping the environment temperature around the set-point with small oscillations, what it is perfectly acceptable, because the external environment that supplied most of the thermal load to the study environments that also suffered constants changes provoked by the reduction or increase of the solar light.

Applying a more detailed evaluation of the controllers efficiency as for the presented oscillations, it is evidenced the PID controller superiority, who makes a sharper adjustment around the set-point, keeping the environments temperature more steady, due to the exploitation of the particular characteristics of each action of control, in order to get a significant improve of the transitory behavior and in permanent regimen of the controlled system.

In case that the use of the controllers do not take into account a higher or minor oscillation of the temperature around the set-point (desired value), it's recommended the use of the On-ff controller with 0,5 ºC band, due to its easiness of implementation, lower operation cost during almost all tests period between the stipulated work band in accordance with the upper/lower limits of the control program.

ACKNOWLEDGEMENT

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Experimental Analysis of a Multi-zone Control System for Central Heating Plants

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SUMMARY

Central heating systems usually are driven by a control which modulates the inlet water temperature as a function of the outside temperature while no information about the actual needs of the different zones of the buildings are taken into account. In this way the maximum heat power is given with the extreme external temperature. A multi-zone control has been developed in order to satisfy the heating loads by considering the real needs. This system can be programmed to maintain different inside temperatures at different time. The control is auto learning and its fundamental aim is to reach the set point temperature as quick as possible, by using the maximum heat power, in order to heat only the air of the zone, without involving too much the building structures.
Thirteen schools of San Donà di Piave, a town near Venice, have been equipped with this system.
The results, presented in this paper, show an important energy saving which can be quantify in terms of a significant reduction of the normalized energy needs and fuel consumption. The energy saving for each school is between 12% an 30%. Obviously also the CO_2 emission is reduced. The economic performance is also very good. The financial analysis shows a pay-back period less then 4 years.

INTRODUCTION

In the interventions to optimize the energy consumption not enough attention is sometimes dedicated to the control system while, often, simple control techniques of the building energy plants can achieve good energy savings. In this paper the attention is focused in central water heating system.
Central heating systems usually are driven by an electronic control which sets the inlet water temperature as a function of the outside temperature while no information about the actual heating loads of the different zones of the buildings are taken into account. In this way the maximum heat power is always given when the design external temperature occurs. Moreover in the oldest buildings, no local controls are installed and all building zones are supplied with water at the same temperature, thus without considering the different local needs and gains. This kind of regulation causes high energy losses and local thermal discomfort conditions. Further, in some plant it is impossible to keep unheated an empty part of a building. Some improvements can be obtained using thermostatic expansion valves, but with these local control devices, no global building energy management can be developed. Better results can be obtained if the building inlet water temperature can be controlled knowing the real loads of the different zones of the building and if there is an auto learning system which can reduce the overall building energy consumption without causing local thermal discomfort [2, 4, 5, 7].

In this paper the performances of the heating plants, obtained in presence of this multi-zone control system and verified during a long period monitoring, are presented and discussed.

METHODS

The system is already on the market and it has been designed to easily set the desired internal thermal conditions by saving as much energy as possible and recording the energy consumption. This remote software system can be installed in every old and new building featured with a central heating plant to control the indoor temperature according to a scheduled plan and to compile the thermal energy bill.
The central unit consists of two main parts: one that interacts with the terminal elements in order to keep the set-point temperature and to count the energy consumption of the thermal zones and the other that drives the central boiler by considering the real energy needs. The central unit can manage up to 60 independent thermal zones but more central units can be linked together to manage buildings with more then 60 thermal zones.
In each thermal zone a converter should be installed to link the thermal zone with the central unit and to drive the terminal elements of the zone. If the temperature of a thermal zone can't be set by the user, only a temperature probe connected to the central unit should be installed, otherwise a local regulator, which is also featured by a temperature probe, to set the personal temperature scheduling should be present. This local regulator has priority in front of the default set point of the central unit. Each terminal element (i.e. a radiator) should be equipped with an electric valve to modulate the heating capacity.
The energy saving is obtained not only by controlling the internal temperature according to the real occupancy and loads of each thermal zone but also by taking "intelligent" decision. In fact this system can learn from the historical behaviour how to keep the set-point temperature without losing energy. For example, if a set-point temperature is reached before the programmed moment, the control system can modify the control constants to improve its accuracy and precision. In the same way, if a set-point temperature should be kept until a certain moment, the system can stop giving power before that moment by using the inertia of the building to keep the desired temperature. Furthermore the system can achieve more energy saving by modulating the water temperature flowing in the terminal element by considering the real heating needs and not on the basis of the outside temperature as the typical regulation method does. In this way, when internal temperature is close to its set-point value, the system operates with a low supply temperature even if the outside temperature is close to the design one.

RESULTS

This system has been installed in all the public buildings of San Donà di Piave, a town near Venice, North-East Italy. The attention in this paper is focused on thirteen schools which were equipped by this system in 1996/1997. The energy consumption data are available from 1991/1992 up to 2004/2005. These data include also the energy consumption for hot water, anyway for these kinds of buildings this energy quota is very small compared to the overall heating energy. The local heating season runs from October the 15th to April the 15th; for each season the heating degree-days (HDD), evaluated for an internal temperature equal to 20 °C, are shown in tab. 1.
In tab. 2 a number is assigned to each building, when a star [*] is present the school includes a gymnasium. The biggest building has a volume of 17701 m^3 while the smallest one has 882 m^3. In the second column, the boiler combustible is indicated: M is for methane and O for oil.

The Normalized Energy Needs (NEN) mean value, evaluated as the ratio between the primary energy annual amount of the fuel needs and the product of the heating degree-days by the building volume, is shown in the third column. The following column shows the weekly heating plant running hours: if a gymnasium is present, its heating hours are indicated after the symbol plus . As the energy consumption depends also on the scheduling of the energy plant finally the ratio between the Normalized Energy Needs and the number of hours is presented in the last column. This ratio is between 0.35 and 0.56 for eight buildings while for the other five buildings is between 2.24 and 3.06.

Table 1. Heating degree-days for the analyzed period

YEARS	HDD
91-92	2432
92-93	2573
93-94	2334
94-95	2353
95-96	2440
96-97	2341
97-98	2293
98-99	2469
99-00	2478
00-01	2225
01-02	2572
02-03	2503
03-04	2537
04-05	2496

The annual heating hours should be intended as a mean value in the monitoring period. In this way the effect due to different scheduling during different seasons can be considered not so significant. A comparison between the average seasonal values during the periods before and after the introduction of the control system is presented in figure 1 for each building.

Table 2. Characteristics of the heating plants of the buildings

N	COMB	NEN [kJ/GGm^3]	h week	NEN/h
1	M	94.31	40	2.36
2*	M	54.31	50 + 78.5	0.42
3*	M	58.67	50 + 86.5	0.43
4	M	79.48	32	2.48
5*	M	65.86	44 + 73.5	0.56
6	M	78.47	35	2.24
7*	M	60.10	50 + 72	0.49
8*	M	63.51	61 + 77.5	0.46
9*	M	48.69	61 + 76.5	0.35
10*	M	58.30	61 + 73	0.43
11	O	95.47	---	---
12	O	122.46	40	3.06
13	O	94.31	40	2.57

The bars in figure 1 show the Normalized Energy Needs mean value for every buildings evaluated before and after the installation of the control system while the line represents the obtained improvement in percentage of the same mean value. The biggest improvement is for building n. 6 (29.5%) while the smallest is for building n. 7 (11.9%), so for each building there is a remarkable energy saving. The biggest value for the Normalized Energy Needs before the control system installation is 138.3 kJ/m^3HDD for building n. 12 while the smallest is 52.4 kJ/m^3HDD for building n. 9. After the installation for the same buildings the values are 106.6 kJ/m^3HDD and 45 kJ/m^3HDD; these values are still the extreme ones. Building n. 9 is also featured by the smallest value for the ratio between Normalized Energy Needs and number of hours; this means that the small value is not due to a partial use of the building but to a good efficiency of the energy plant and to a good building envelope energy quality. It is important to underline that also for this building there is a 12% improvement; this means that, also in presence of excellent performances of the building - plant system, some further energy saving can be achievable by introducing more accurate control system.

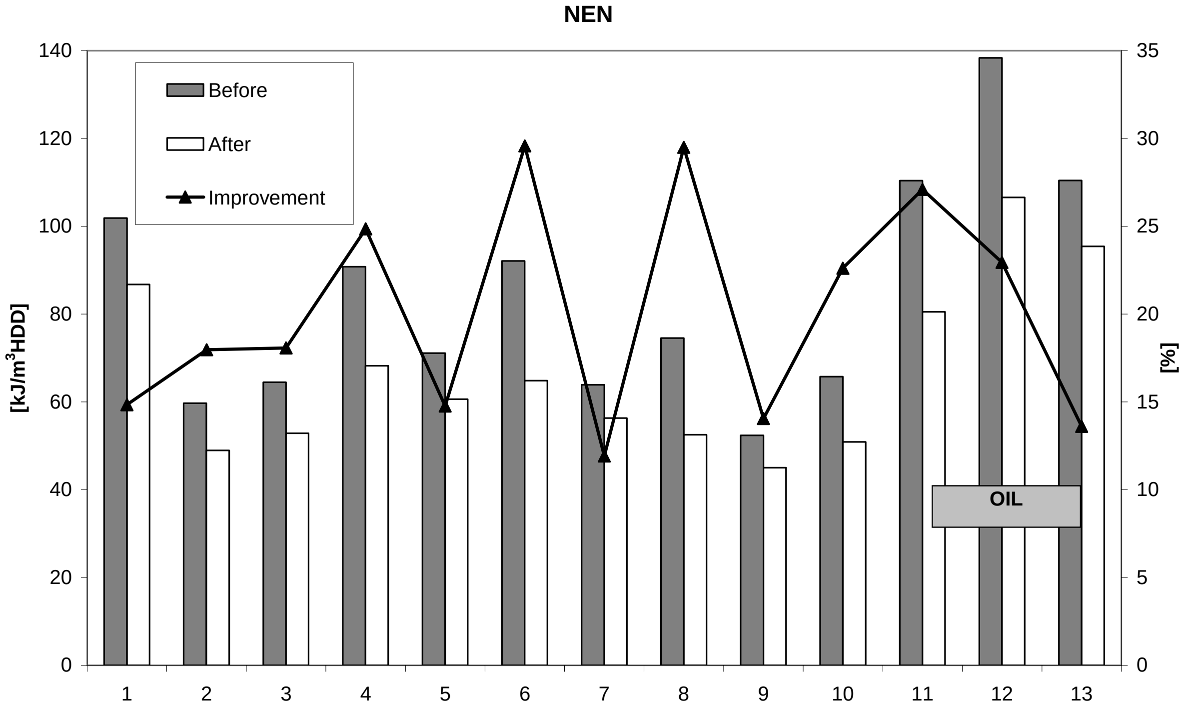

Figure 1. Comparison between average NEN values before and after the installation of the control system and their improvement in percentage for each building

Figures 2, 3 and 4 show the Normalized Energy Needs for each building for each season: the improvement from the 1996/1997 is evident; the trend is almost the same except for the oil boiler buildings.

Figure 5 shows the ratio between the energy consumption and the building size volume, expressed in m^3_{CH4}/m^3 for gas boiler buildings and l/m^3 for oil boiler buildings before and after the introduction of the control system. The line is the CO_2 emission reduction obtained by multiplying the consumption difference by the combustible emission factor (1.8 kg_{CO2}/m^3_{CH4} and 2.7 kg_{CO2}/l_{OIL}). The CO_2 annual reduction for all the buildings is about 180 tons.

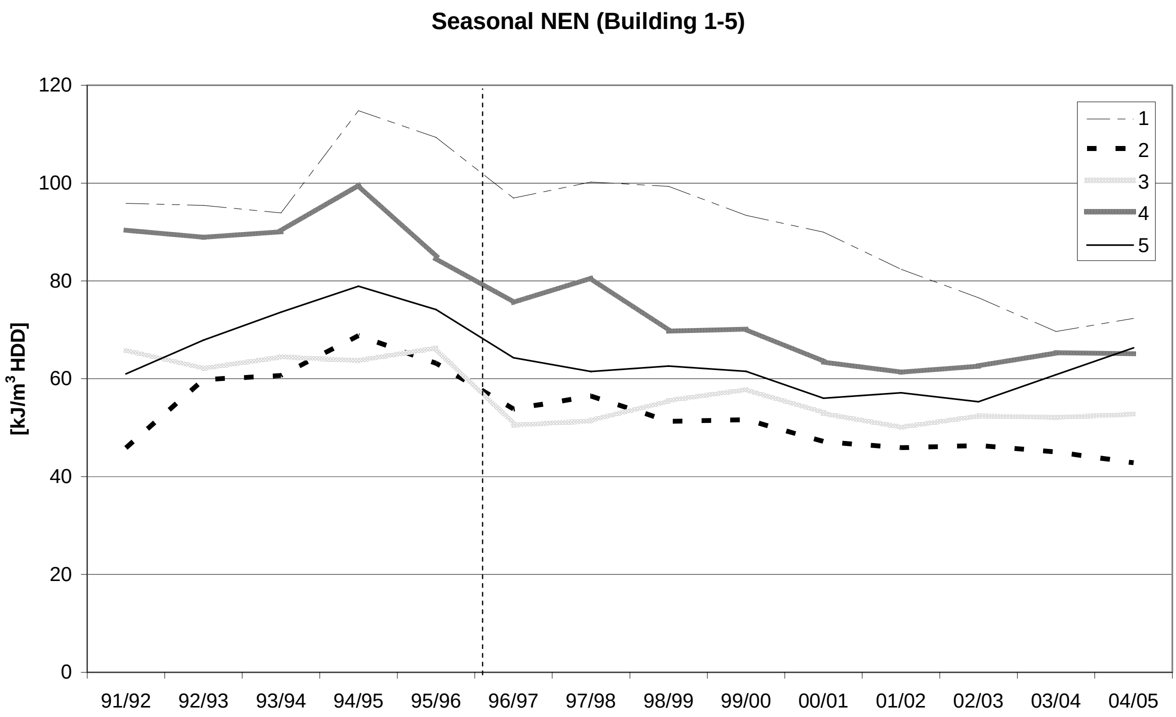

Figure 2. Seasonal NEN for building 1-5.

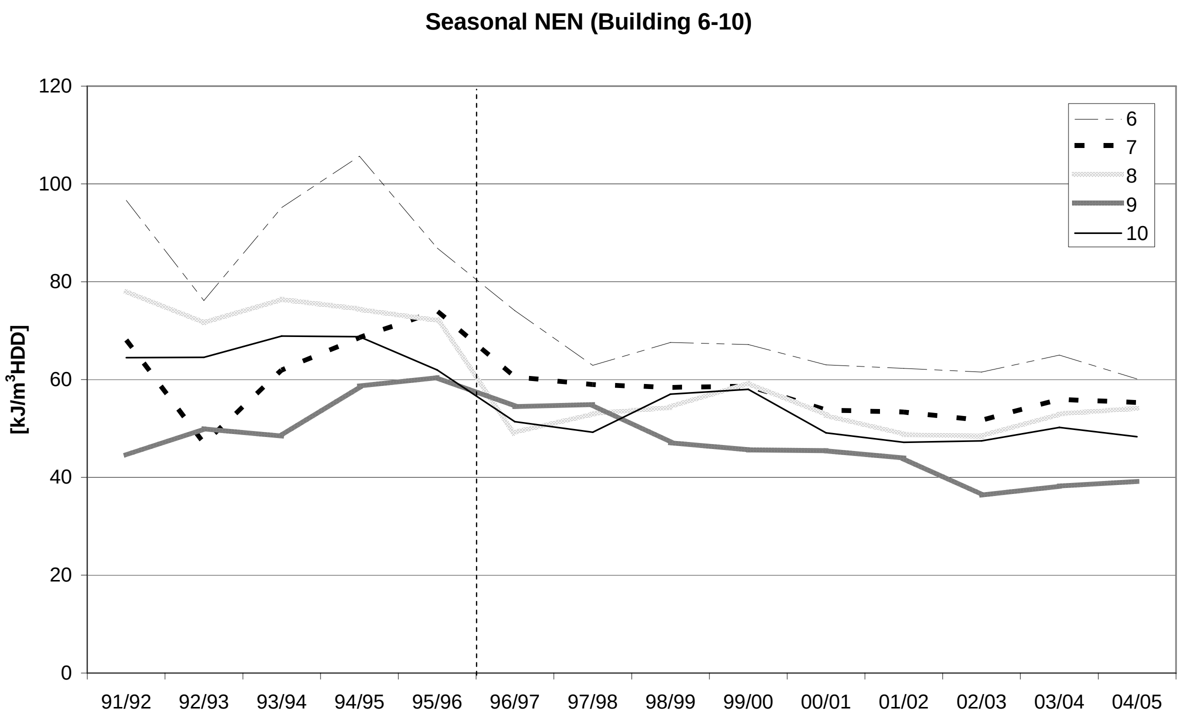

Figure 3. Seasonal NEN for building 6-10.

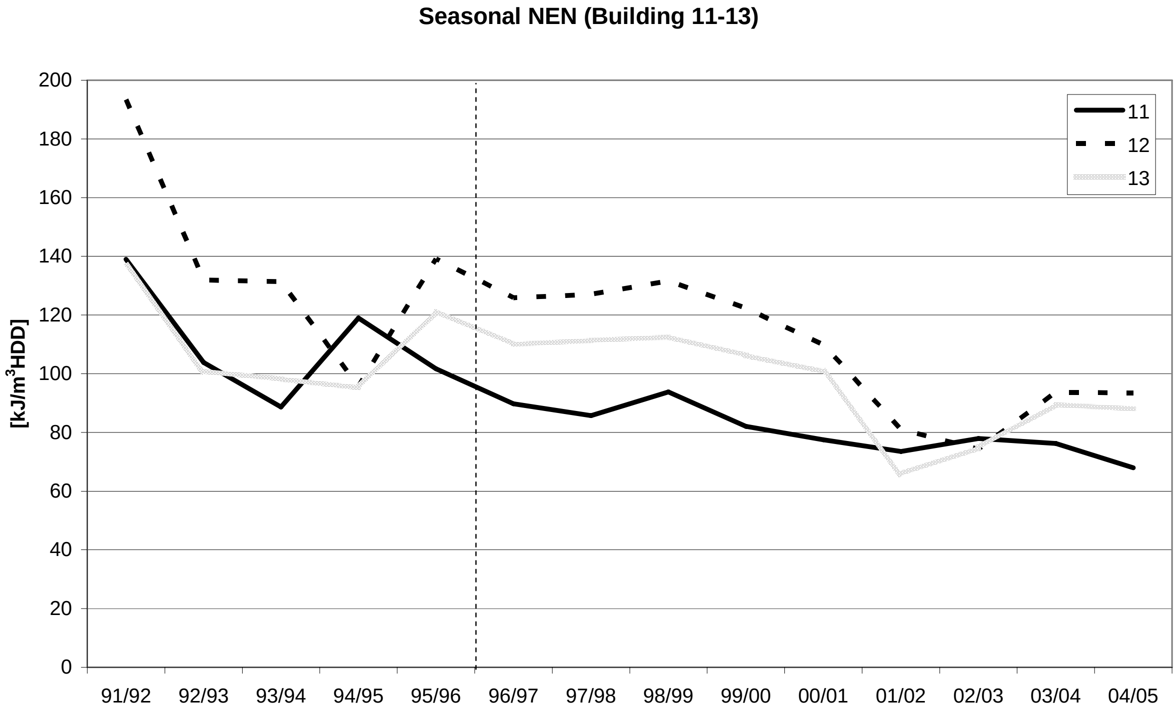

Figure 4. Seasonal NEN for building 11-13

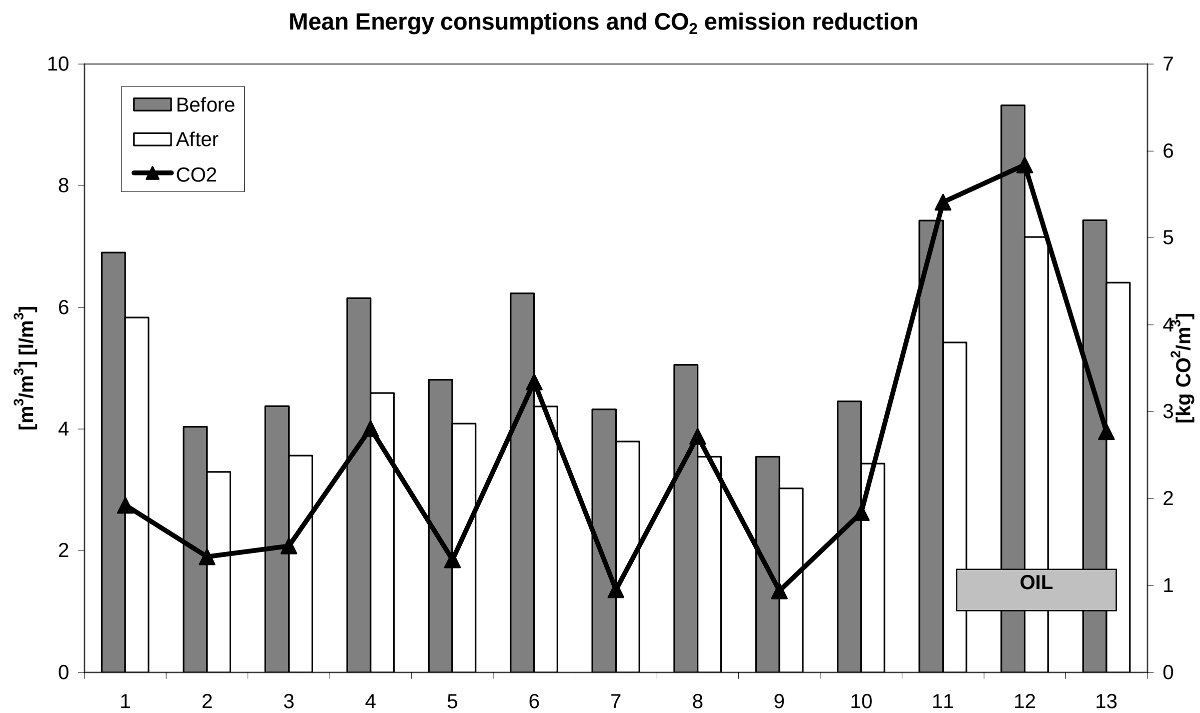

Figure 5. Mean energy consumptions before and after the intervention and CO_2 emission reduction for each building.

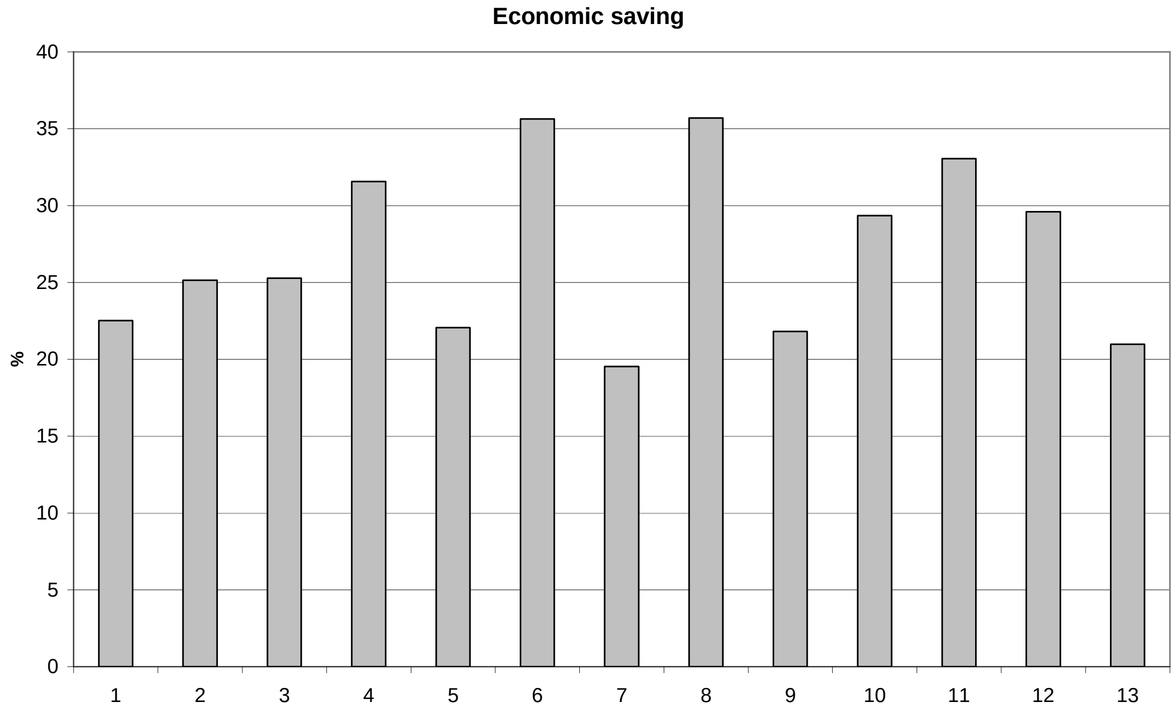

Figure 6. Economic saving in percentage for each building.

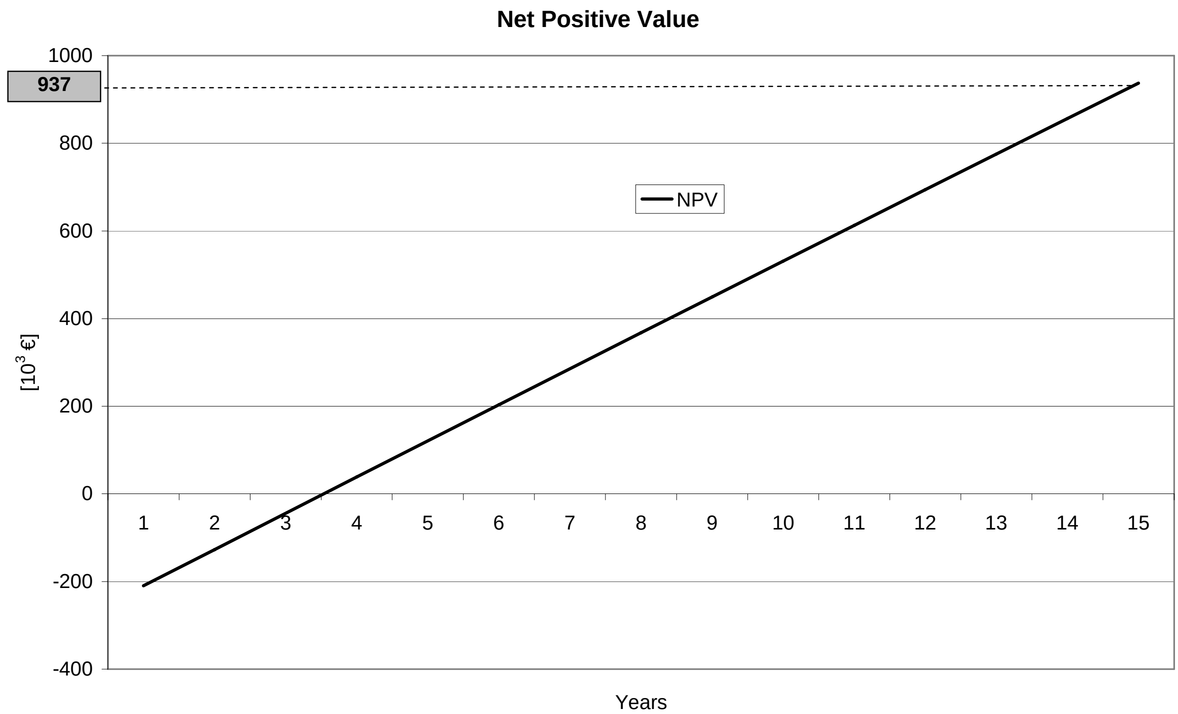

Figure 7. Net positive value of the total investment over 15 years period analysis.

Figure 6 shows the annual economic savings for each building, evaluated by considering the mean combustible reduction before and after the installation at the actual energy costs, which are 0.52 €/m^3_{CH4} + VAT and 0.92 €/l_{OIL} + VAT. It is important to underline that before the installation of the control system the VAT was equal to 20% while now it is equal to 10% as the system allows to obtain a tax reduction for its reduced environmental impact. The economic saving is between 19.5% and 35.7%.

Finally an economic analysis has been done. The cost to install the control system for all the thirteen buildings at the actual value is 293050 € while the annual economic saving for all buildings is 83296 €.

The analysis has been carried on for 15 years with an actual discount rate equal to 3.5% and a 3% inflation rate equal to the annual increment rate of the combustible cost. Figure 7 shows the trend of net positive value (NPV) for the 15 years analysis: the total value is 937197 € and the payback period is about 3.5 years while the internal redditivity rate is 27.7%.

DISCUSSION

A long time monitoring has been carried on to test a new system developed to control the internal temperature of multi-zone buildings equipped with central boiler. This control is auto learning and it is able to take intelligent decisions in order to save more energy as possible. Its fundamental aim is to reach the set point temperature as quick as possible, by using the maximum heat power, in order to heat only the air of the zone, without involving the building structures.

The system has been installed in thirteen schools of San Donà di Piave, a town near Venice. The results show a great energy saving which can be appreciated in terms of normalised energy needs and fuel consumption. The energy saving is between 12% an 30%. The annual CO_2 emission reduction is about 180 tons. The economic performance is also very good. The cost of the system can be approximated around 290000 € and it allows to save about 83000 € per year. Consequently the financial analysis shows a pay-back period less then 4 years and a total net positive value of 937197 € over a period of fifteen years.

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A Study on static-pressure control method of VAV System in Intelligent Building

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INTRODUCTION

As the development of country's economic and the improvement of living-standard in China, there are more and more buildings which are making use of the VAV(Variable Air Volume) Air-Conditioning System to make people comfortable. The Intelligent Building is making use of system integration method to combine with the Intelligence Computer Art, Communication Art, Information Art and Architecture Art in an organic way, which has the characteristics of more subjects, more technologies, high technology synthesis and so on by the automatic monitoring to the equipment, the administration to the information resources and the information service to the user and to their Building's optimization grouping, with reasonableness, safety, high-effect, comfortable, convenient, nimble characteristics to integrate and adapt the information and social demand.

In the 1960s, the Variable Air Volume Air-Conditioning System VAV was born in America. The basic principal of the VAV Air-Conditioning System is simple, which is to satisfy the room's variable load by changing the airflow supplied to the room. As the VAV air-conditioning system is running in partial load most of the time, it can improve the device and system's energy utilization by reducing the ventilator's energy consumption of decreasing the airflow capacity. At present, the usage rate of West Country's VAV air-conditioning system in high Building is surpass 90 percent, however in China, the application of VAV air-conditioning system will be more and more as the serious problem of the High Building's energy consumption. VAV air-conditioning system is composed of AHU, fresh air/exhaust air/supply air/back air duct, VAV BOX, room temperature controller, VAV BOX is the most important part of this system.

VAV AIR-CONDITIONING SYSTEM PLATFORM TASK

The introduction of the system platform

This task's VAV system mainly adopts the WEBCTRL platform as shown in *Fig 1*. Following the agreement of BACnet protocol, under the condition of compatibility of the speed, cost and future frequency width extend ability, ALC corporation's control ware network adopts the structure of three layers, which provide different controller categories including the upper computer machine, router, the controller of I/O nods, integrated controller and the local implement wares and sensors in different layers. The controller of I/O nods is divided into three kinds which is corresponding to the three layer distribution structure, for example, the structure and application of cooling/heating unit, AHU, terminal devices and so on.

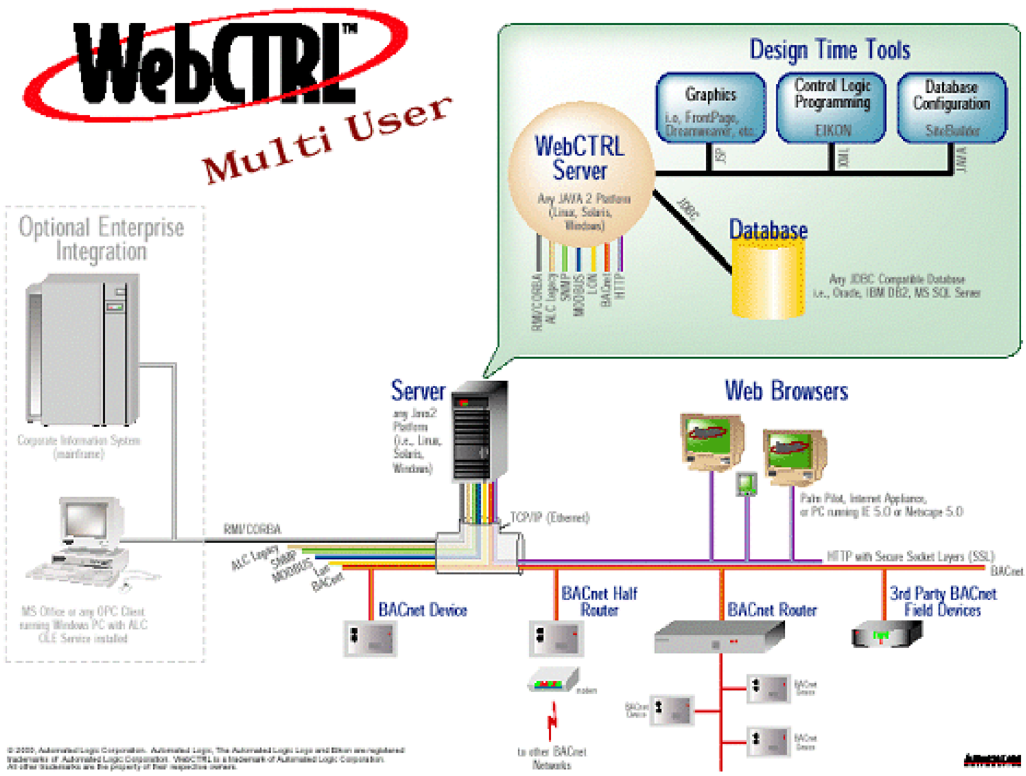

Fig 1 WebCtrl system structure picture

System Software

Building Automation Control System of ALC/WebCTRL^{TM} is composed of four parts:
(1) WebCTRL Server software it takes charge with the various controllers and network station communication as well as data exchange.
(2) SiteBuilder it is the frame tool of the system database that could quickly set up the system structure.
(3) EIKON it is the tool of graphic compiler that is used for compiling device control programs
(4) MS FrontPage2000 and its accessories are mainly used for designing the dynamic graphics and carrying out the controls and data displacements.

Here the compiler tool EIKON is the most advanced graphic compiler in the industry. By clicking the button, it can make complicated control algorithm, diagnosis faults, which can test the performance of the control programs with the real-time or simulative running parameters. EIKON is not adopting line-by-line and abstruse code so that to make the control programs easily understood

System Hardware

The hardwares of the ALC/WebCTRL^{TM}'s Building Automation Control System are as follows:
(1) Network Router LGE
LGE is the high performance net router, which is mainly used for M series, S series, UNI, WebPRTL serial control wares to communicate with the Ethernet. LGE is based on the 10Base-T Ethernet, which is mainly taken charge for the control ware subnet and sending

messages between other net router and the upper computer machine's software WebCTRL Server.

(2) M serial locale control ware
M serial control ware is suit for multi-device control, which is one part of ALC corporation 's WebCTRL™ system. It includes the S serial and U serial control wares. M serial control ware has the firmly structure and separate running ability, which is suit for multi-device application occasion.

(3) S serial locale control ware
S serial control ware is the mainly single control device, which provides the high reliability and strong design that could make the controllers accommodate various conditions, even the roof installation.

(4) Subnet locale control wares
Subnet locale control ware is mainly the U serial control ware, which has two types UNI/16 and UNI/32 at present. They are both the network router that is used for communicating with the BACnet MS/TP standard protocol control wares, which is based on the high 156K baud among the ARCnet Building Automation's communication.

The software's energy-saving device

WebCTRL system has obvious characteristics in energy-saving aspects, which makes the energy-saving devices achieve less energy consumption in a certain degree under the condition of indoor comfortable environment.

(1) Space Temperature Control (STC)
According to the cooling season or heating season, working time or free time and the condition of peak value, it will ascertain the room's temperature control value. And seven kinds of colors will show the room's temperature and PMV.

(2) TOD (Time of Day Scheduling)
Time schedule is to prescribe the building or district under a certain state, for example in the process of being used, set the device's on/off time, make the device run effectively. It can also set separate device or the device group's running schedule. Time schedule has normal, free time super time three kinds of PRI that are convenient for user to manage.
(3)TLO (Timed Local Override)
The Operating System has set the locale super control, which will allow the user to surpass the program control in requiring time and lock the import or export values.

(4) OSS (Optimum Start/Stop)
Optimum Start/Stop is mainly used to complete transition from the free time to working time. Under the normal outdoor condition, optimum start/stop will follow the temperature ascending velocity and the descending velocity of the cooling room considering to the outdoor temperature and enclosure structure's heat-gained and heat-lost coefficient. It will also assure the indoor's PMV and effectively reduce the device's running time.

(5) STO (Source Temperature Optimization)
According to the building's district demand, it will allow the temperature setting value of the cooling/ heating chain's different devices (air-conditioning unit, chiller unit, boiler device) to reset in the user's prescribed scale. The calculation of the practical setting value is processing according to the heating or cooling demand times of the present device or district feedback. In every updating period, STO program will check the network whether each device will absorb the additive heat or cool quantity from their cooling/ heating source.

(6) DL (Demand Limiting)

It will measure the using situation of the building's electric power, which will divide three electric demand grades. According to the electric demand grades to modify the temperature setting value, it will assure the system just to reduce the building's PMV instead of not be paralysis.

(7) DNS (Day/Night Setback

It allows the room's temperature to descend or ascend in a certain pre-establish scale in free time. If it reached any end of the DNS scale, heating (cooling) will start up until the room's temperature return the DNS scale.

THE EXPATIATION OF THE VAV AIR-CONDITIONING SYSTEM'S CONTROL SCHEME

The test condition of the task

This task adopts the typical VAV air-conditioning system control scheme, contrast to the full air-conditioning system, the main characteristic of which is installing a VAV terminal device in each room. This terminal device is actually a valve by which could increase or reduce the airflow supplied to the room in order to achieve separately regulation to various room.

According to the Intelligent VAV control project of this task, its terminal adopts pressure-independent control method, which is lack of plus-pressure ventilator and terminal reheat device. So the hole system only has the summer condition, the main control scheme of this task.

Intelligent temperature sensor will transfer the setting value and measuring value to the locale controller in order to compare the two values of the heat demand or cool demand. Then the airflow control module will calculate the room's airflow set point, and compare it to the actual measured value by the airflow measuring device, regulate the valve's opening and satisfy the room's load request. The valve's minimum opening should satisfy the minimum needs of the room's ventilation, terminal heating or cooling needs and transfer the limit opening value to the upper AHU through the control network. The valve's minimum opening value and the limit opening value are the foundation of resetting supply air temperature. The actual measured value is the foundation of controlling the back airflow of AHU.

Stable static pressure control scheme

At present, the stable static pressure control method is the normally used VAV system control way, which is simple and feasible. However, its fan energy consumption is high so that the terminal valve state's pressure distribution and the static pressure setting value are not easy to set in applicable engineering. Or else it will reduce the energy-saving effect and be likely to appear increscent noise phenomena.

Variable static pressure control scheme

Variable static pressure control method is the best energy-saving way. Although Variable static pressure control method could economize fan's energy consumption in the farthest degree, the control method is very complex and different to carry out because the system has increased the valve-opening sensor, which should be feedback to the static pressure controller through the communication network. The control device provided by the market doesn't offer the variable static pressure control algorithm. As a result, it needs the engineers to compile and debug in the locale case, so the workload is too heavy. Comparing to the stable static pressure control method, this kind of control method also adopts pressure control, which has the same problem of the system stability. This kind of scheme is suit for large-scale public building that adopts Building Automation System.

TRAV terminal regulation scheme

According to the advanced theory abroad, American writer T.B.Hartman has brought forward TRAV (Terminal Regulated Air Volume) control method, which creates comfortable environment by regulating the airflow. Based on the practical airflow demand, TRAV control method adopts more advanced control software to carry out the ventilator control. The concrete control method of this control system is as follows, according to the three VAV terminal airflow's differences between the total measuring value and airflow setting value, it will follow PID algorithm to control the frequency ventilator and ventilator's airflow in the opposite direction and forward feedback.

VAV TERMINAL CONTROL SCHEME

The total airflow control scheme

Based on the pressure-independent VAV terminals, the total airflow control method is a newly simple and feasible control method. According to the relation between the total airflow and ventilator's rotate speed, it can regulate the ventilator's rotate speed. The energy-saving effect is between the stable static pressure and variable pressure method. As the control way does not adopt pressure control, the control system which is a practical fan airflow control way is stable.

The total airflow control method is to calculate the ventilator's demand rotate speed in a certain degree, which is different from the typical feedback control. However, the setting airflow does not immediately set to the satisfied future airflow load (that is stable airflow) after the room's load is changed. Then the setting airflow is the middle control quantum that is gradually stabilized room temperature, which is integrated through room's temperature departure.

As a result, the ventilator's rotate speed of the total airflow control method will immediately regulate to the stable speed and not change after the room's load varies. As a matter of fact, the total airflow control method is a kind of dominating ventilator's rotate speed control method according to the room's temperature departure regulated by the PID controller.

Cooling On Demand scheme

The meaning of Cooling on demand is to carry out the energy-saving control to the cool supply equipment under the satisfying user's cooling on demand. Control system makes use of network control technique, supervises the terminal equipment, cool water user and cool source. And the cooling on demand and using cool time to request are taken on the catenulate delivery, it is shown in *Fig 2.*

Fig2 Cooling on demand control frame picture

Terminal VAV BOX calculates the cooling on demand and the cool time request according to the temperature of the control area and the Opening situation of the valve, transfers it to the superior air-conditioning unit; the air-conditioning unit sets up the supply air temperature according to subordinate cooling on demand, then it ascertains whether or not to start up the air-conditioning unit and suggests using cold time request by the superior cooling source according to the request of the subordinate cooling time; it refers to the real supply air temperature and the duct water valve's opening aperture situation to calculate the cooling demand of the superior cooling source; cooling source makes the statistic of all consumer's cooling on demand and the request of using cooling time, comes to ascertain whether or not to startup a cooling air-conditioning unit, resets the chilled water pump's supply water temperature, or adjusts compression pump again.

Terminal regulate control air volume（TRAV）is to adopt advanced control software to directly control the supply air fan according to the air-conditioning room's working condition, terminal device's real time air-flow demand and air valve's switch state. The concrete control mode is that according to the VAV terminal air-flow's different value between the total measure value and the air-flow's total set point value, it will control the fan frequency transformer and the fan airflow to regulate in the opposite direction feedback on the base of the PID algorithm. It collects VAV terminal device's valve numbers of the full-opened and full-closed, if all valves are full-closed, the fan's frequency transformer will carry out the opposite direction feedback regulation, that is to diminish the frequency of the Frequency Transformer; If at least one valve is full-opened, the fan's frequency transformer will carry out the positive direction feedback regulation. [3] The terminal regulation control theory is shown in *Fig 3*.

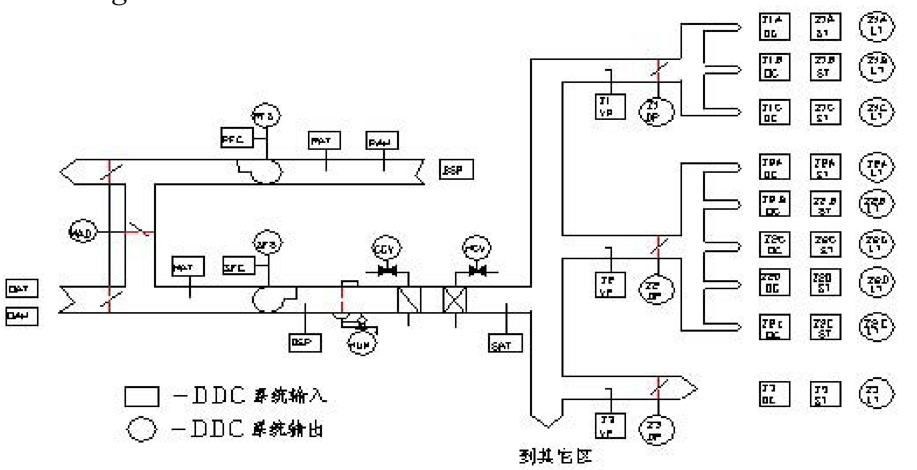

Fig 3 Terminal regulated control theory picture.
LT-use button, ST-indoor intelligent temperature sensor, OC-occupational sensor, DP-air valve, VP-airflow sensor, SAT-ventilation temperature sensor, CCV-cooling water valve, HCV-heating water valve,HUM- humidifier valve, OAT/OAH-fresh air/ temperature humidifier sensor, RAT/RAH-return air temperature humidifier sensor

Variable supply air temperature control scheme

Here we elementary set the Variable supply air temperature control scheme, the develop control program in our task, the AHU control unit could adopt cooling optimization algorithm:

(1) The initial supply air temperature set value is 12.5℃(summer condition) and the minimum value is 11.5℃ and the maximum value is 22℃

(2) If the 30 percent of the opening of the VAV terminal (Damper value)which is running has reached the maximal opening and continuing 5 min, it will regulate the cold water valve and all terminals of the unit’s service will reduce 0.5℃.
(3) If the 30 percent of the opening of the VAV terminal (Damper value)which is running has reached the minimal opening and continuing 5 min, it will regulate the cold water valve and all terminals of the unit’s service will reduce 0.5℃.
(4) AHU control unit will transit from ir-refrigeration condition to refrigeration condition, the set point’s control algorithm will change from the refrigeration condition set point in a speed of 0.5℃ per minute.
(5) AHU control unit will transit from refrigeration condition to ir-refrigeration condition, the set point’s control algorithm will change from the refrigeration condition set point in a speed of 0.5℃ per minute.
(6) If the supply air temperature is 2.5℃lower than the set point of the supply air temperature(in case of 0.5℃transfer hysteresis) in a continuously 5min, AHU will control it to bring a alarm signal, which will be effective after the ventilator is continuously running for 30 min.
(7) If the supply air temperature is 2.5℃higher than the set point of the supply air temperature(in case of 0.5℃transfer hysteresis) in a continuously 5min, AHU will control it to bring a alarm signal, which will be effective after the ventilator is continuously running for 30 min.

Software testing result

Heat district testing result

Based on the BACnet protocol, this Building Automation Platform software organizes and develops the control algorithm, and carries out software simulation test and physical system’s debugging experiment according to TROX single-duct pressure-independent VAV terminal under the condition of Building Environment and Intelligence Laboratory of Beijing Union University. At the same time, it adopts the ALC Corporation’s U341+ control ware and the test result is shown in *Fig 4*. From the chart, we can see that the indoor temperature’s trend is close to 23.8 after the air-conditioning system setup 20min. And after 55min the result is close to the set point and the dynamic departure reduced markedly. It has proved that the control algorithm is feasible and the algorithm has higher consult values.

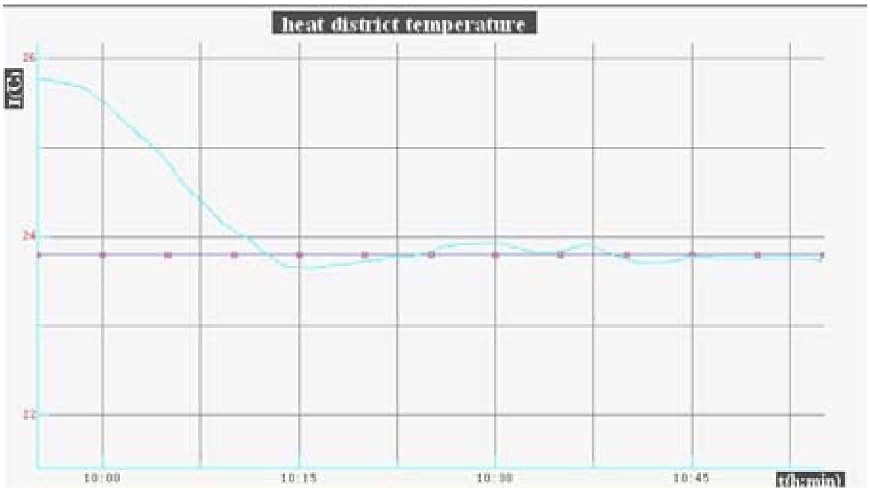

Fig 4 VAV Air-conditioning room's inspecting picture

Airflow and static pressure inspecting result

VAVBox terminal airflow device has measured the variable curve at 14:48-14:58 one day afternoon, as shown in *Fig 5*, of which the airflow is 3250 m^3/h; ventilator static device has measured the variable curve at 14:40-14:48 one day afternoon, as shown in *Fig 6*, of which the mean static pressure is 265pa; the system terminal static pressure device has measured the variable curve at 15:20-15:28 one day afternoon, as shown in *Fig 7*, of which the mean static pressure is 40pa.

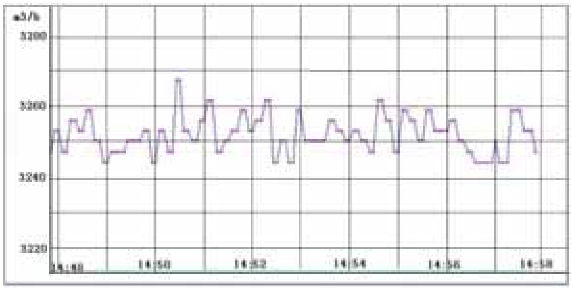

Fig 5 VAVBOX terminal airflow inspecting picture.

Fig 6 ventilator static pressure inspecting picture.

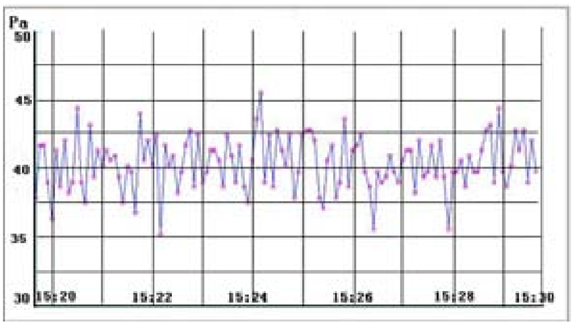

Fig 7 system terminal static pressure inspecting picture

The VAV control system is not only fixing the terminal device and gearshift air fan, but also has the total control system that is made up of some controlling loops. The VAV control system's running condition will be changed following the time varies, which could assure the system's basic demand depending on the Automation Control such as the comfortable room temperature, the sufficient fresh air, the benign airflow and the normal indoor pressure and so on.

The chiller unit's VAV control includes the loops of ventilator control, back-flow fan control, fresh airflow control and so on. We can see that both the VAV air-conditioning system and the control system are combined tightly and indiscerptibly. The main task of the control system is that it could adjust the supply airflow automatically and adapting the variable room load, which can achieve the requests of different function room's different temperature parameters and adjust the ventilator's the rotate speed so that it could reduce the running energy consumption in the air-conditioning system.

CONCLUSION

The automation control of the VAV air-conditioning system in the engineering practice mostly adopts the static pressure control scheme. However, it often exists the problem that super-cooling and super-heating of local district in the Building, the insufficient fresh airflow that dreggy air in the air-conditioning room. From the theoretical analysis, the control tactics principles of the cooling on demand and TRAV have the characteristics of energy-saving and stability, the system demands the fan power is less than the static pressure control system, when the flow drops by the fixed flow's 50%, the request fan power has been fallen below 15% of the fixed power according to this algorithm.

The task has already carried out the software testing and physical system's primary debugging in Building Environment and Intelligence Laboratory of Beijing Union University, testified its feasibility. It can be seen that the control principle and the algorithm have higher reference values. At the same time, it has the practical and immediate significance to improving the running efficiency and reducing energy consumption according to the VAV air-conditioning system's all-around energy-saving and optimal running.

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C06
Computer based design methods

BUILDING INFORMATION MODELING TECHNOLOGY FOR FULLY INTEGRATED DESIGN AND CONSTRUCTION

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INTRODUCTION

This paper explores the recent global developments in software and software protocols for Building Information Modeling (BIM) as a tool for fully integrated design and construction. Building Information Modeling is a still emerging technology that has yet to reach its anything close to its full potential. However the technology can already clearly demonstrate its overall benefits in terms of coordinated design, shorter design & construction schedules, significant cost reductions, job site safety improvements and even more sustainable construction programs.

As discussed in this paper, there is a significant amount of work being done worldwide **on software tool & protocol developments** by governmental agencies, non-profit and research organizations, as well as commercial entities, to facilitate and promote BIM technology. Also discussed are the **application software developments** currently underway. The good news is that many, if not most, of these developments are IFC compliant. However there still are many barriers to successful and commercial application.

Clearly BIM modeling has gained wider acceptance in the past two years, particularly in the architectural design side of the profession, but we are far from a fully integrated design process. Research into engineering firms specifically for information related to their experience with Building Information Modeling Programs (i.e. Bentley Microstation, Tri Forma, Factory CAD, Catia, Autodesk Revit, Graphisoft ArchiCAD, etc.) as design and construction tools, does show wider application today. But while we have seen a few companies fully integrate structural design and analysis programs into those architectural models, we have yet to find many companies that are able to demonstrate BIM model tie-ins that can fully integrate HVAC related design programs (such as Load Calculation Programs, Pipe and Duct Sizing Programs, Building Energy Modeling/Analysis Programs - DOE-2, EnergyPlus, BLAST, IBLAST, TRACE 700, etc.).

We have also yet to find many companies that have integrated Building Owning, Operations and Maintenance Programs for Facilities Management. Similarly we have yet to find many companies able to fully integrate natural day lighting design programs (such as Superlite 1.01, LUMEN , doe 2.1 or Radiance 3.4 or illumination design programs such as Autolux, AG132, esp Vision, Autodesk Lightscape, Lightcalc +Art or ALADAN) into BIM models. Yet the spatial planning, modeling and orientation technology imbedded in 3D, 4D & 5D models would seem a logical input data base for such programs and the recipient of the output.

Despite this, the tremendous opportunities for improved efficiency by integration of the design and construction process will continue to drive the technology forward.

OVERVIEW OF BUILDING INFORMATION TECHNOLOGY

The ultimate goal of Building Information Modeling is to assemble a single data base of fully integrated and interoperable information that can be used seamlessly and sequentially by all members of the design and construction team and ultimately by owners/operators throughout a facility's life cycle. The desired result is a BIM model where three dimensional (3D) graphical imaging carries real-time (i.e. immediate and dynamic access) data, and where every line and every object carries real-life intelligent physical and performance data.

The modeling technology can start with direct data transfer from the design calculation software into graphic layouts (for systems such as structural steel, fire protection or other modular elements). Alternatively it can utilize the graphic layouts as direct input to load calculations (such as pipe sizing, duct sizing, etc). Modeling programs can also link to specifications and to manufacturers web sites for data input. Either way, building information modeling technology already extends into fully integrated 4D modeling (adding the fourth time dimension for scheduling or sequencing using programs such as Primavera) and 5D modeling (adding the fifth dimension of cost for estimating and budget control (using programs such as Timberline). The building design development can continue with the provision of automatic bills of material and generation of automatic shop drawings for everything from structural steel to sheet metal duct fabrication, to fire protection and piping fabrication, to electrical cabling and bus duct layouts, etc.

CURRENT TOOL AND SOFTWARE PROTOCOL DEVELOPMENTS

A major key to the success of these efforts is establishing common software protocols. The International Alliance for Interoperability (IAI) at www.iai-na.org. has developed models to address this issue. IAI is a council of the National Institute of Building Sciences (NIBS) at www.nibs.org . The mission of IAI can be found on their Web site, which also provides information on programmable language XML data models for information transfer between disparate software packages (as an example, aecXML is a framework for using an eXtensible Markup Language standard specifically related to technology in architecture, engineering and construction). They also offer comprehensive, intelligent and universal data models through Industry Foundation Classes (IFCs) to IFC.XML2 ISO 10303-28, which incorporate HVAC schemas compatible with ifcXML - IFC2X3 code (ISO 10303-11), as well as data elements that represent entire portions of a building or system. These are used to assemble computer-readable models of the facility that contain all of the information on the parts and their relationship (ISO/PAS 16739). Table 1 Identifies HVAC schema being developed under IAI Project BS-8.

Table 1 - IFC HVAC Extension Under IAI Project BS-8

The main objective of BS-8 was to extend the IFC data model with schema needed to *fully* support the *modeling and simulation* of HVAC components and systems. The pragmatic goal was to *enable rich data exchange* among the various building simulation tools in use today and in the future.	
General Data	**IFC HVAC extension schema for support of *building energy performance simulation***
General and performance specifications of materials General specifications of HVAC and other simulation related equipment, systems and furnishings Performance specifications for the above.	Dynamic load estimation HVAC design HVAC equipment selection Measurement and verification (HVAC view) Building performance metrics (HVAC view) HVAC system and equipment commissioning HVAC system and equipment retrofit HVAC system and equipment physical layout HVAC system and equipment product data (catalogues, external data bases).
Downstream applications	**Processes that are *partially* supported by IFC HVAC extensions**
Other HVAC (design) applications Other building energy performance simulation tools (such as air-flow models) Cost estimating applications Commissioning tools Building operations and maintenance software Code-checking applications Software that serves utility companies Many other types of applications that use HVAC definitions.	Energy code compliance Interference checking Cost estimating HVAC construction documents Construction and installation Procurement Maintenance Operations Fault detection and diagnostics Emergency response Accessibility Utility billing and cost allocation

Even more importantly, IAI's international Council has established the European Integrated Project "InPro" "Open Information Environment for Knowledge Based Collaborative Processes throughout the Life Cycle of a Building"- European Project No. 026716-2.

The US National Institute of Building Sciences (NIBS) has created the National Building Information Model Standard, under their Facility Information Council. The first Version 1.0 is now available for review on their web site www.facilityinformationcouncil.org/. This is in addition to the work being done by the Building Enclosure & Environment Council and the Facility Maintenance & Operations Council. NIBS is working with IAI to develop an overall integrated program developing and expanding IFC component values under the umbrella "building SMART"- www.iai-na.org/bsmart/ . buildingSMART = BIM + IFC and combines BIM technology with Information for the Construction (IFC) global ISO Standard. Target date for the first version was the end of 2006. The annual buildingSMART Conference brings all facets of the industry together to advance the science.

As their web site indicates *"**buildingSMART** is the dynamic and seamless exchange of accurate, useful information on the built environment among all members of the building community throughout the lifecycle of a facility. **buildingSMART** is simply a smarter process for managing the project lifecycle."*

The mission of the National BIM Standard Project Committee as identified in their charter is to improve the performance of facilities over their full life-cycle by fostering a common, standard and integrated life-cycle information model for the A/E/C & FM industry. This information model will allow for the free flow of graphic and non-graphic information among all parties to the process of creating and sustaining the built environment, and will work to coordinate U.S. efforts with related activities taking place internationally.

Of related interest the National Institute for Science and Technology NIST Advanced Technology Program (www.nist.gov/atp) and Standard Reference Data programs (www.nist.gov/srd) are also very supportive of industry research and development in this area. Capital facilities industry cfiXML at www.cfixml.org is a cooperative effort of ePlant Data www.eplantdata.com ; DIPPR a chemical industries alliance and FIATECH are also developing manufacturing industries standard XMLs.
Of particular interest is the work being done by the FIATECH (a non profit consortium supported by NIST and established by the Construction Industry Institute) towards fully integrated and automated design and construction technologies. Their Capital Projects Technology Roadmap (CPTR) www.fiatech.org/projects/roadmap/cptri.htm or www.construction-institute.org/FIATECH addresses many of the critical issues facing the industry. The Projects Technology Roadmap is a cooperative effort of associations, consortia, government agencies, and industry, working together to accelerate the deployment of emerging and new technologies that will revolutionize the capabilities of the capital projects industry. This initiative, led by FIATECH, is open to all stakeholders who are committed to the future success of the capital projects industry. The Capital Projects Technology Roadmap presents their strategy for the capital projects industry in **"Developing a consensus vision for the capital projects industry and a unifying initiative to achieve the vision"**. Table 2 indicates the planned components of project development.

Table No.2 – FIATECH Capital Projects Technology Roadmap

www.fiatech.org/projects/roadmap/jobsite.html

1. Scenario-based Project Planning
2. Automated Design
3. Integrated, Automated Procurement & Supply Network
4. Intelligent & Automated Construction Job Site
5. Intelligent Self-maintaining and Repairing Operational Facility
6. Real-time Project and Facility Management, Coordination and Control
7. New Materials, Methods, Products & Equipment
8. Technology- & Knowledge-enabled Workforce
9. Lifecycle Data Management & Information Integration

The Internet Standard published by the World Wide Web Consortium www.w3.org offers interoperable technologies for information, commerce, communication and collective understanding. WWW recently released XML schema 1.1 Structures for XML documents. Still further tool technology includes Unified Building Modeling Language UML www.omg.org for specifying, visualizing and constructing the artifacts of software systems and SOLIBRI IFC Optimizer and model checker www.solibri.com for more effective data storage and transmission.

CURRENT COMMERCIAL APPLICATION SOFTWARE DEVELOPMENTS

The first challenge facing private and commercial enterprises in developing integrated information modeling is the reality of the limited size of the marketplace. Unlike basic office software where a word processing program may sell tens of millions of copies, building modeling programs (of far greater sophistication) may sell only a few thousand. The second challenge facing the industry is developing a commonality of program protocols that enable free exchange of data between disparate software systems. These are becoming incredibly complex. Here, the work being done by NIBS and IAI to develop an overall integrated program under the umbrella "building SMART" shows great promise in combining BIM technology with Information for the Construction (IFC) global ISO Standards to ISO/PAS 16793.

One of the biggest recent advances has been the development of open meld protocols that allow the integration of disparate software programs. An example of this is NavisWorks "Jetstream" © 3D Design Review at www.navisworks.com , that has the ability to manage, view and integrate disparate software programs providing real time navigation, collaborative communication and presentation of 3D & 4D Building Information Models.

Certainly the European community, particularly, and to some extent Australia, are further ahead of the USA in the widespread application of BIM. Europe is heavily committed to development of integrated building design technology with extensive work being done in the UK, Norway, Sweden, Finland, Germany and the Netherlands. Programs include Integrated IT Solutions by Nemetschek, NA noiXML whose Vector Works Fundamentals are architectural building programs that can plug into AutoCAD and CATIA, Landmark for site planning and Spotlight for lighting design. (Nemetscheck - Allplan, www.allplan.co.uk/ for architectural design and modeling software). Data Design Systems www.dds-bspco.uk/ has developed mechanical and electrical IFC compliant file systems. Meanwhile Zeit + Raum develops virtual building information models. TELKA develops structural programs. Obermeyer has project model orientated software. Octager has access an viewing technology. Mensch Machine has software for MEP CAD application. Graphisoft www.graphisoft.com ArchCAD 10 encompasses architectural design software and virtual building solutions, which will allow input of files into Energy Plus. Meanwhile EPM Technology, Norway has developed an IFC Model Server to share project model data.

Two of the major players in the architectural/engineering end of building design and modeling are Bentley and Autodesk. Both are collaborating with the buildingSMART initiative.

Bentley Solutions at www.bentley.com has Microstation and Intergraph as the primary products for building design, construction and operation. Recent enhancements include interface with gbXML for energy analysis using Trace 700 and tie into pipe and duct sizing, the latter using ddXML. Their suite of products include Tri Forma as an extension of

Microstation for 3D solid mass modeling geared to plant design. Other products under development include Bentley autoPIPE, Bentley Piping and Bentley HVAC. Also under development is Bentley Facilities for building owners and operators.

Autodesk at www.autodesk.com building solutions include AutoCAD www.autocad.com and Autodesk Revit (Building – Architectural; Structure – Structural Design; Systems Plus – MEP and Buzzsaw – Project Management Tools). Autodesk Building Systems (ABS) and Autodesk Desktop (ADT) are BIM compatible packages. ABS provides the special definition by analyzing CAD layouts identifying internal and external loads orientation, thermal zones etc. It can export to load calculations software such as TRACE and input results back to ABS It can also use the input data from TRACE for auto sizing and scheduling VAV boxes. http:/usaautodesk.com/adsk/servlet/index?siteID=123112&id=6861179

In 2006 Oracle launched a new collaborative Building Information Management Platform (CBIM) at the IAI Building Smart Conference in Munich. The Oracle CBIM www.oracle.com/openworld initiative provides a web accessed enterprise collaboration platform, which makes building information available to architects, project managers, building contractors, subcontractors, operators and facilities management firms from a single source. Oracle is working with Graphisoft to fully integrate building modeling tools with design collaboration, visualization, life cycle management and other applications.

There are even players like the Mayo Foundation (under the Mayo Clinic) are developing Mayo Graphical Integrated Computer Aided Design using MagiCAD software. MagiCAD developed by Program Oy of Sweden is IFC compliant and is used widely in Scandanavia and aimed primarily at duct design and equipment manufacturing, but including HVAC, piping and electrical design and application software (Running on an AutoCAD platform). These have the ability to feed data into Energy Plus using Olof Grandlund/LBNL developed BS Pro Com server middleware (with IFC based 3D RIM).

International Building Performance Simulation Association IBPSA www.ibpsa.org , a non profit organization, is developing building performance simulation specifically aimed at HVAC equipment, air flow in buildings, energy usage, as well as visual and acoustical environments. The annual SIM conference promotes building sustainability and performance through simulation.

Green Building Studio www.greenbuildingstudio.com offers open gbXML schema for direct data exchange offering web based building energy analysis tools integrating data from the BIM into DOE-2, BLAST, TRACE 700, EnergyPlus 1.4, etc. and exporting the results back into the BIM.

Examples of leading edge software for manufacturing and the factory floor are Pro E[6], Factory CAD[7], while for the plant facilities, Unix-based Catia at www.catia.ibm.com. , CC Plant, Tri Forma, Tri-Plan, Bentley Plant Solutions[6], Speedicon , digiPLANT, and autoPLANT, lead the way. These programs can now tackle the more complex elements of equipment layout, HVAC&R design and layout, including piping, sheet metal, equipment, and electrical services. They also are geared to support the complete life cycle from design through construction to operation and maintenance.

Other technologies are emerging, such Ansys CFX 5 computational fluid dynamic software allowing air flow simulation under laminar and turbulent states. Of even greater interest is the development of laser scanning of existing buildings and sites. Programs such as Cyra Technologies www.cyra.com and Leica Geosystems HDS Inc. www.leica-geosystems.com – can record site and as-built conditions and do geospatial imaging and engineering surveys while creating a database file that can be integrated into the building model. Disto laser scanning of existing buildings is tied to USD-M2 www.usdm2.co.uk .

Another interesting development is the work being done by the International Code Council. ICC has unveiled SMART Codes – an interoperable, automated code compliance checking system for checking compliance with the 2006 International Energy Conservation Code. This automatic checking system can yield an inspection check list or provide a 3D virtual walkthrough that tags non-compliant building components.

For a listing of current building systems related software it is worth visiting www.energytoolsdirectory.gov/

BARRIERS TO SUCCESS

The U.S. National Institute of Standards and Technology (NIST) recently published a study that identified and estimated the efficiency losses in the U.S. capital facilities industry resulting from inadequate interoperability among computer-aided design, engineering, and software systems. Although the focus of the study is on capital facilities-commercial/institutional buildings and industrial facilities - it benefits key stakeholders throughout the construction industry. This study - the conception, design, and publicity of which FIATECH was a strong and early contributor to - is based on an earlier report also done by NIST of the cost of interoperability in the U.S. automobile supply chain.

NIST GCR 04-867 estimates the cost of inadequate interoperability in the U.S. capital facilities industry conservatively to be $15.8 billion per year. These cost impacts are of interest to owners and operators of capital facilities; design, construction, operation and maintenance, and other providers of professional services in the capital facilities industry; and public- and private-sector research organizations engaged in developing interoperability solutions.

The construction industry itself is highly fragmented between design disciplines, construction trades, and material suppliers, with wide disparities in operational practices and technological development. Even large engineering companies that would, seemingly, have the resources to apply BIM technology are themselves often internally fragmented by multiple offices and operating divisions. As a result the design and construction industry significantly lags behind other industries (such as manufacturing, petro-chemical and aircraft) in exploiting technological advances.

Unlike the manufacturing industry, with its repetitive mass production processes that allows development of operational techniques to improve efficiency and reduce waste, the construction industry handles one of a kind building programs that are each uniquely designed and developed. It is a dichotomy that the very diversity and fragmentation in the industry, which could more fully benefit from BIM modeling technology, also hinders its development and acceptance. A Construction Industry Institute (CII) study indicates that the level of waste experienced in the manufacturing industry typically averages around 28% while that of the construction industry averages around 57%. Similarly the value added component of manufacturing averages 62%, while that of the construction industry is only 10%. (value added being defined as that component of design or construction team labor and cost that directly adds value for the client) . But it is this very disparity and the industry fragmentation that provides design and construction with such tremendous opportunities for improvement.

ISSUES AND BENEFITS

Of course development and use of all of this technology does not come without cost. Experience on large industrial projects ($75 to $150 million) indicates that currently the added design cost represents a 5% to 10% premium on the Architect-Engineers (A-E) fees (or roughly 0.25% to 0.5% on construction cost). The A-E cost premium can be quite a bit higher on smaller projects. More than offsetting this are all of the cost savings outlined above. Immediate savings of 3% to 7 ½ % have already been seen through improved coordination and reduced conflicts. (The Construction Industry Institute www.construction-institute.org/ analysis indicated a potential of 7% for this element alone). This can only increase as more trades come on board and BIM capabilities, such as shop drawings and quantity take-offs, are realized. Increased use of shop fabrication and elimination of waste is, itself, expected to produce savings of at least 7½ % to 10%. Recent experience in the construction of major automotive plants shows that it is possible to eliminate 20% sheet metal waste, develop programs 15% to 25% faster, reduce RFIs by 50%, eliminate 25% of all change orders and reduce construction cost by 4% to 10%. The greatest potential savings may come from the application of value added and lean construction techniques that BIM enables. Regardless, in the interim, A-Es must convince owners that this added investment justifies increased fees.

Table No.3 – Benefits of Building Information Modeling

- Provide visualization of project
- Build fundamental intelligence into drawings
- Provide a single database of information to meet the needs of all parties.
- A valuable project management tool
- Improve issue tracking process
- Provide seamless flow of information
- Provide automated bills of material Reduce the bidding time and effort
- Automatically provide shop drawings
- Facilitate off-site prefabrication
- Simplify material ordering and site management
- Improve field coordination and significantly reduce interferences
- Reduce change orders
- Save cost and time at every phase of design and construction
- Provide owner with live/intelligent file records Provide electronic links for service and maintenance Facilitate peak building performance throughout a facilities life cycle

BIM can already demonstrate all of the individual elements listed in Table No.3. However it cannot yet achieve all of these benefits incrementally without separate interface. The interoperability of software components is still not there, requiring separate and distinct input and output data. Successful widespread implementation of BIM for fully integrated design, depends upon the ability of architects and engineers, as a design team, to be able to easily input and exchange data. The key then is for the integrated system to be able to continually and dynamically model the building and all of its systems, through daily and seasonal operational cycles. This then allows the “what if” scenarios to be played out utilizing different systems and components and allow them to be evaluated on a first cost, operating cost and life cycle cost basis. We are far from that point.

CONCLUSION

There is no question that the information technology required for these processes is very complex and quite difficult to implement and is straining the limits of designers' current hardware, software, and even staff capabilities. Much more work needs to be done to enable the technology to be applied on a day to day basis and we are still far from having an interoperable system that can enable fully integrated system design. The greatest opportunity lies with fully integrated multi-disciplinary A-Es practices and where BIM integration is being done as a continuum of the design process as well as the construction process.

Building Information Modeling is clearly gaining considerable momentum as the technology evolves and greater interoperability occurs between disparate software systems. The rapidly emerging goals of green building/sustainable design, towards net zero-energy buildings, coupled that with goals for carbon dioxide emissions reduction, will require "whole building" fully integrated design and construction as a dynamic process. BIM can help provide that integration.

Benefits of Building Information Models in Energy Analysis

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SUMMARY

The importance of energy analysis in building design has grown, but it is still mostly done by simple static calculations or estimates. Accurate dynamic simulation software have been available already for decades, but these tools are still not widely used by practitioners in building projects. The main barrier of wider usage of dynamic energy analysis has been the required big manual input work. By utilising BIM as a data source for energy analysis, the data input will be more efficient and the existing data more reusable. Only by using BIM, the verification of thermal performance can truly happen in different phases of the building process. This paper describes a new concept and interoperable software environment for management of thermal performance during the whole building life cycle. Real project experiences also show that BIM based environment allows to use whole building spatial simulation instead of traditionally used zone based models.

INTRODUCTION

The importance of analysing energy performance in building design is growing, because of the increasing awareness of it's role in building life cycle costs and environmental impacts. However, energy analysis is still mostly done by statistical estimates or using simple static calculations.

Dynamic energy simulation softwares have been available already for decades. Commonly known softwares, such as TRNSYS, ESP, DOE-2 and BLAST, have been developed already in 1970's. Also new tools, such as Energyplus and IDA, have been developed for even more accurate thermal simulation. All these tools have been mostly used by researchers, not by practitioners in building projects.

Energy analysis in building design has to meet both the cost and schedule requirements of practical projects. The main barrier of wider usage of dynamic energy analysis methods has been the required big manual input work. Most of the building element specific information needed in energy simulation is described in the building information model (BIM). By utilising BIM as a data source for energy analysis, the data input will be more efficient and the existing data more reusable. It also makes possible to benefit spatial whole building models instead of zone based models often used today in energy simulation. Only by using BIM, the verification of energy performance can truly happen in different phases of the building process. Some of the public building owners, such as in USA [1], Denmark [2] and Finland [3], are starting to demand BIM in their projects, which creates more possibilities to get benefits for thermal analysis.

Other barriers for wider utilisation of BIMs in energy analysis have been the missing interoperable data interface implementations in thermal simulation tools and the lacking guidelines. The Industry Foundation Classes (IFC) [4] provides today an open standard for description and exchange of information within the life cycle of a constructed facility. To use BIM effectively however, and for the benefits of its use to be released, the quality of communication between the different participants in the construction process needs to be improved. The Information Delivery Manual (IDM) by the Norwegian BuildingSMART project [5] is developed to provide the integrated reference for process and data required by BIM by identifying the discrete processes undertaken within building construction together with the information that is required for and results from their execution. There has also been several national projects in different countries, where guidelines for the BIM based design have been developed.

Using BIM is the only practical way to verify energy performance truly in different phases of the building process. The experiences from many BIM based projects show that interoperable energy analysis software is not enough for the management of thermal performance during the building process, but it requires also tools to manage different revisions of BIMs, to compare thermal performance of these revisions and to visualise this by easy-to-understand way. This paper shows the recent developments for concept and interoperable software environment to manage spatial thermal performance. The results show also examples of using BIM based energy analysis in real projects.

METHODS

Thermal performance management can benefit a lot by utilising building information models as a data source for thermal analysis: The data input will be more efficient and the existing data more reusable. The latter is especially important for continuous thermal performance management through the whole building life cycle.

Wider utilisation of BIM requires instructions both to the actual work and to the true understanding of the potential benefits. This paper presents the results of the development of new methods and interoperable software environment to manage spatial thermal performance. The key components in the interoperable software environment are thermal simulation and spatial requirements management, but it supports also building services system modelling and the later phases of building process by linkage to commissioning and facilities management. The Finnish ProIT project [6] has produced guidelines for the BIM based design. The guidelines for building services design [7] are used here to describe the role of thermal performance management in BIM based building process (Figure 1).

The IDM has produced the most detailed guidelines for BIM based energy analysis [8]. It contains both the energy and condition analysis process description and the requirements for information exchange throughout the building process. The IDM process of developing energy analysis throughout the design stages of a project is considered to be divided into 2 key parts:

1. **Speculative**: This is the analysis work undertaken during the programming stage of the project. It is about providing advice on the potential energy performance of a building and its systems to other design roles. The aim of this advisory role is to have an impact on the overall building design, determine the feasibility of concepts in an energy context and to establish energy targets.
2. **Analytical**: This is the analysis work undertaken during the sketch, full concept and coordinated design stages of the project and assumes the availability of geometric information about the building layout. The overall process is the same at each stage of

work, the difference being simply about the extent of the information available and the level of certainty that can be applied to the information.

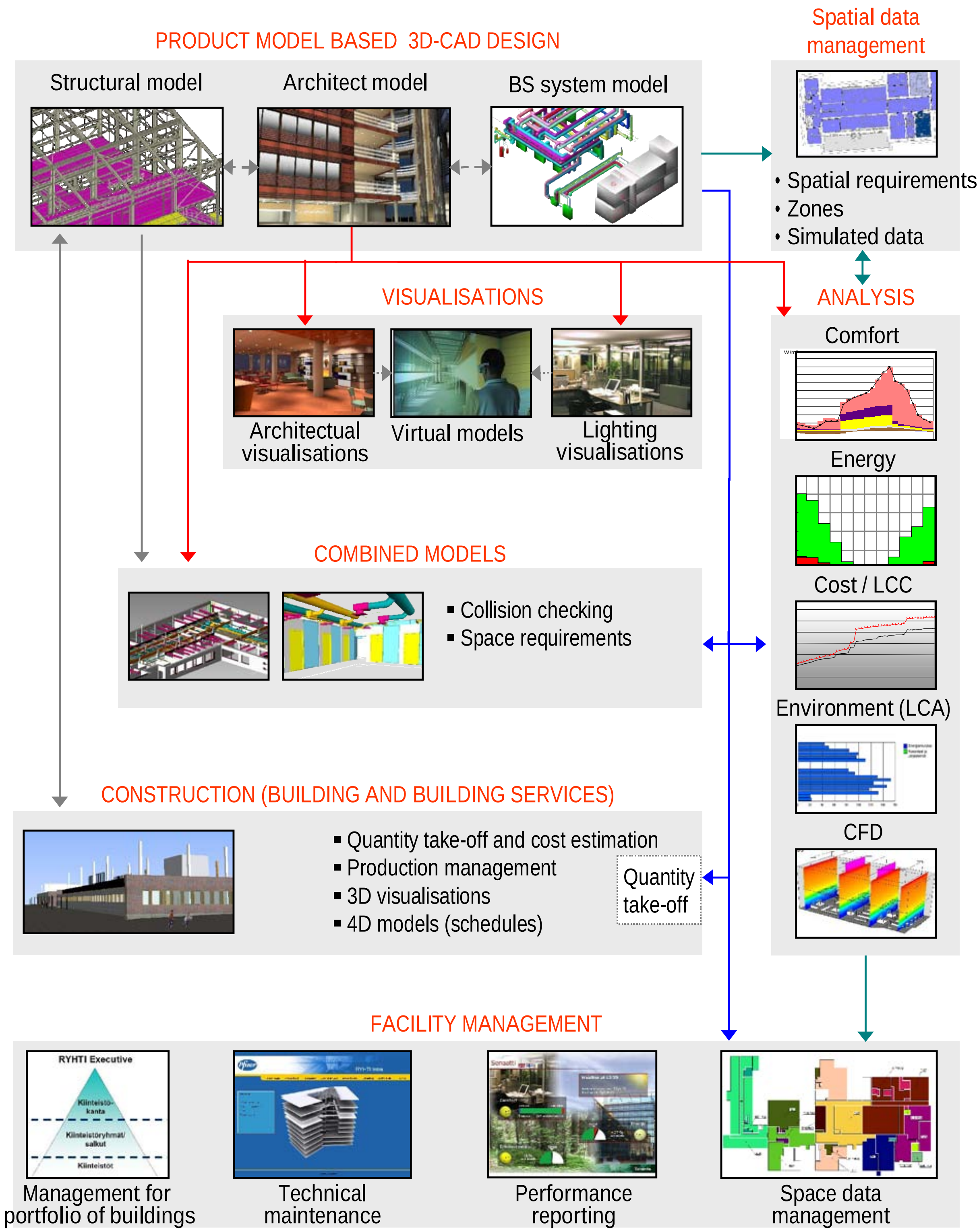

Figure 1. Energy analysis has an important role in the use of building information modelling according to the ProIT guidelines.

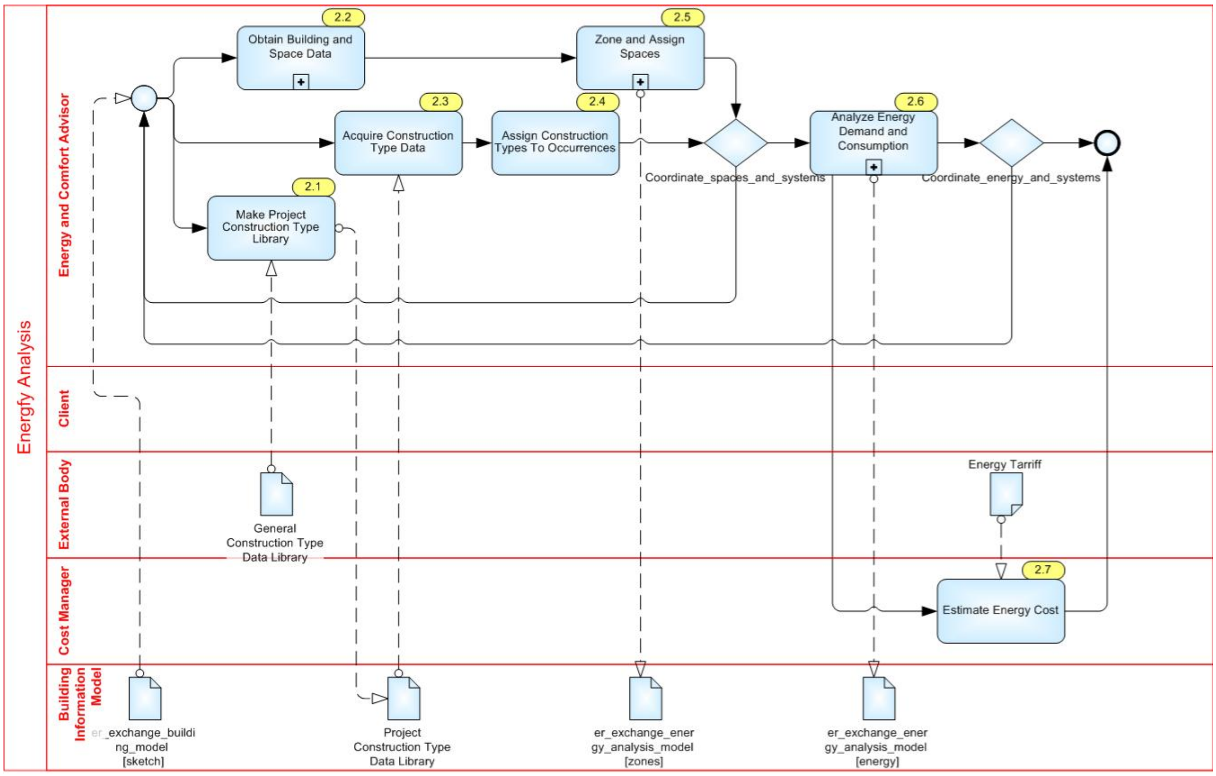

Figure 2. The IDM process map for energy analysis.

RESULTS

The main result of the development was the concept for thermal performance management utilising BIM based environment (Figure 3). The key components in the implementation of the interoperable software environment are energy simulation software RIUSKA [9] and spatial requirements management software ROOMEX. It supports also BIM based building services system modelling and the later phases of the building process by linkage to commissioning and facilities management.

Thermal requirements

Thermal requirements (task 1 in the figure 3) are defined at space type level and so this does not require 3D model of the building. It is necessary to capture client's requirements for different space types, to achieve all the later benefits of BIM based thermal performance management. This task defines both the thermal parameters and the equivalent thermal loads (Table 1). One relevant definition from the electricity calculation viewpoint is also the required lighting intensity.

Utilisation of geometry model

In the utilisation of geometry model (tasks 2 and 3 in figure 3) an architectural 3D model is imported to the environment and validated by automatic checking routines (to report of missing spaces, floors, etc.) and by visual checking (by 3D viewer).

Most of the building element specific information needed in energy simulation is described in the BIM. This includes as mandatory information: spaces with code and name specifications and building structures (walls, windows, etc.) with type definitions. To illustrate the large amount of required building model related data for energy simulation input: As an example a medium size building of 8000 m^2 contains 387 spaces, 295 exterior walls, 912 interior walls, 1439 windows, 760 doors and 57 different type of structures. All this information is needed for spatial level energy simulation, and by utilising BIM this information can be imported to the simulation software. The required information is too much for manual input, which means that traditionally whole building spatial simulation is not practical, and a simplified zone level approach is used instead.

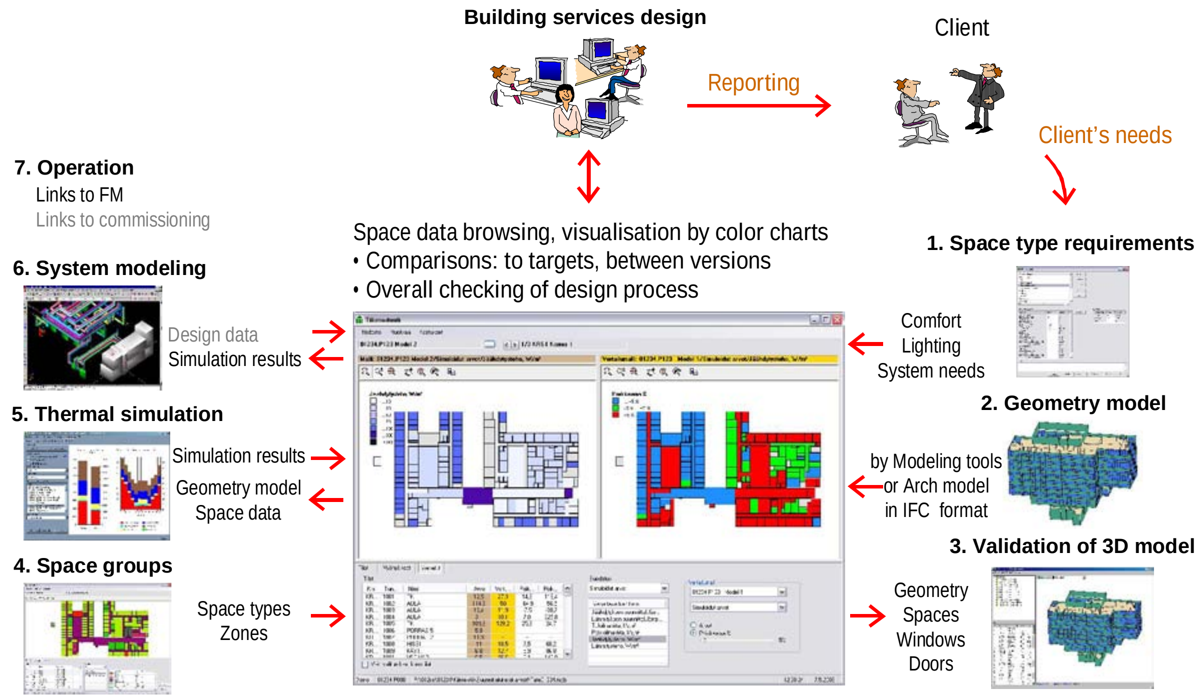

Figure 3. The developed concept for BIM based thermal performance management (the implemented tasks at black, still missing tasks at grey color). The interface in the center shows BIM revision comparison of required thermal parameters against simulated performance.

Table 1. The air quality targets and equivalent loads defined at space type level.

Air quality targets

Attribute	Value	Unit
Indoor temperature, summer	26	°C
Indoor temperature, winter	21	°C
Relative humidity, summer		%
Relative humidity, winter		%
Air velocity, summer	0,25	m/s
Air velocity, winter	0,17	m/s
Carbon dioxide content (CO2)		ppm
Supply air flow	1,5	dm³/s,m²
Supply air flow		dm³/s, per
Sound level		db(A)
Filtering class		EU
Air purity class		P

Loads

Loadtype	Design...	Max value	Unit
People	0,1		p/m²
Lighting	15		W/m²
Equipment	15		W/m²

Sizing val. = Load used in sizing
Max val. = Maximum load

Values of relative humidity are agreed together with the client

Calculated air-flow/person = 15 dm³/s, per

To utilise BIM in energy simulation, IFC model is simplified by pre-processing it. Some format of building objects (Proxys and BREP walls) do not contain the information needed for energy simulation. Figure 4 shows an example from the HITOS project [10], where the biggest part of the architectural BIM was beneficial for thermal analysis to create the required input, but some of the facades were modeled by using BREP format wall objects instead of parametric representation. By visualising the problematic parts (red color) of the model, architect was able to correct the model in the next revision to meet also the requirements of energy simulation.

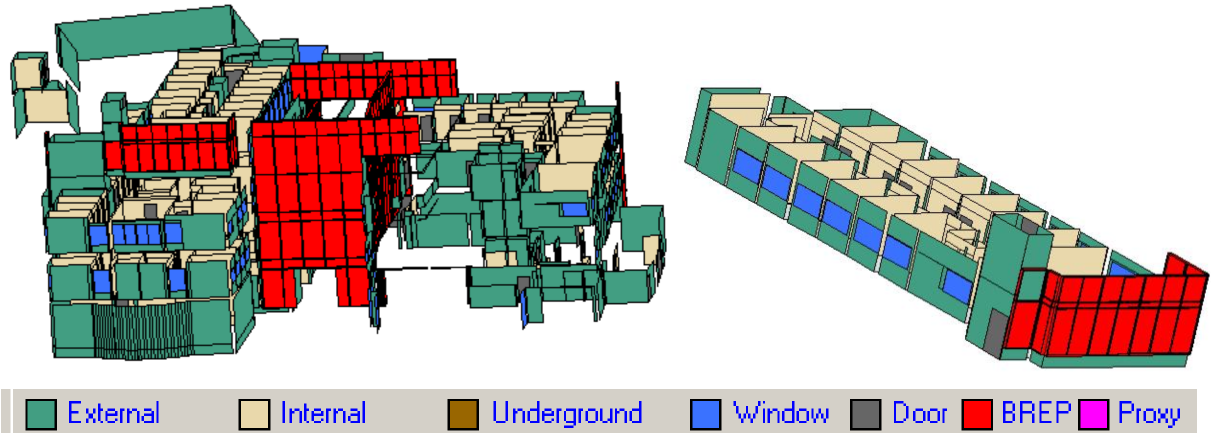

Figure 4. The architectural model of building elements can be imported to energy analysis. It is important to validate that the model meets the requirements of energy analysis.

Space grouping

Space groups (task 4 in figure 3) are created both to define spatial requirements by mapping the space type requirements to actual spaces and for zoning the spaces into different categories, such as air handling unit service areas or lighting control areas. These specifications should be done in the beginning of the building services conceptual design. The same information is needed as input data for energy analysis.

Thermal simulation

In thermal simulation (task 5 in figure 3) the building geometry related data, spatial requirements, loads and air-conditioning and lighting system zones are imported as input data. Additional input, such as usage schedules, HVAC and electrical system specifications and weather data, is defined by using the energy simulation software user interfaces. The simulation results of the spatial thermal performance are exported back to the BIM, allowing performance verification against targets and generation of input data for the HVAC system modelling.

Energy analysis in several projects (Figure 5) has been done by using IDM process descriptions and information exchange requirements. The IDM supports also the creating of energy analysis software specific training material, like the example in figure 6 to describe the required actions in energy analysis to acquire construction type data and assign it to occurrences.

Key requirements for wider use of energy simulation in practical projects are:

- modern user interfaces for practitioners instead of researchers
- efficient data input by reusing the existing information by linkage to BIM and
- using of intelligent data libraries.

Efficient computing is required to move from zone based models to spatial whole building approach. This means also balancing between solver accuracy and efficiency.

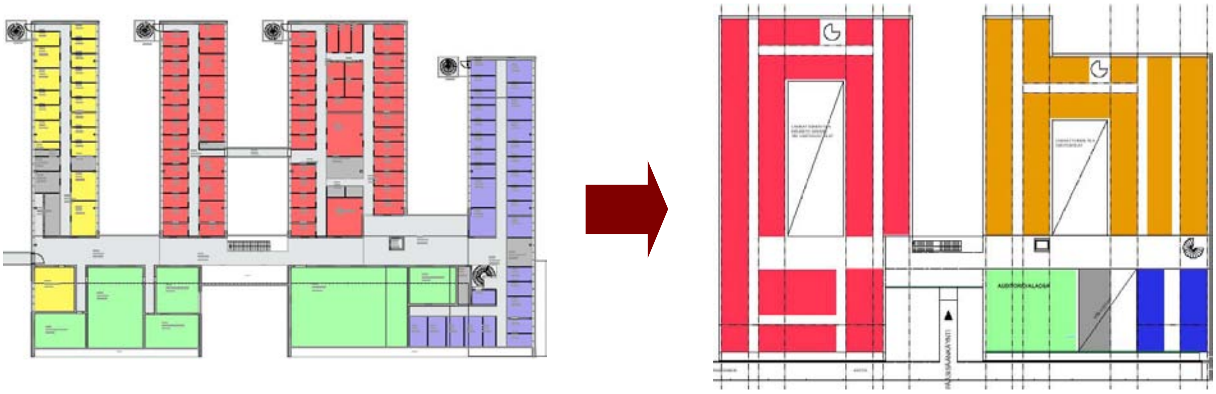

Figure 5. The energy efficiency of the architectural design of the Aurora 2 building [11] was developed during the programming phase by using BIM based energy analysis. From the first solution of cellular office plan with narrow building body (left, a “comb” type of layout), the envelope area was reduced by using glazed inner courtyards for distribution of daylight for the surrounding office spaces (right).

Building services system modelling

The results from thermal simulation can be transferred as starting point to building services system modelling (task 6 in figure 3). The information, such as spatial air flow, heating and cooling needs, is used to select air conditioning equipment for HVAC system model.

Operation and maintenance

Thermal performance management should continue during operation and maintenance phase (task 7 in figure 3). Figure 6 shows an example how the self-reporting building system [12] is used for thermal performance management during operation phase.

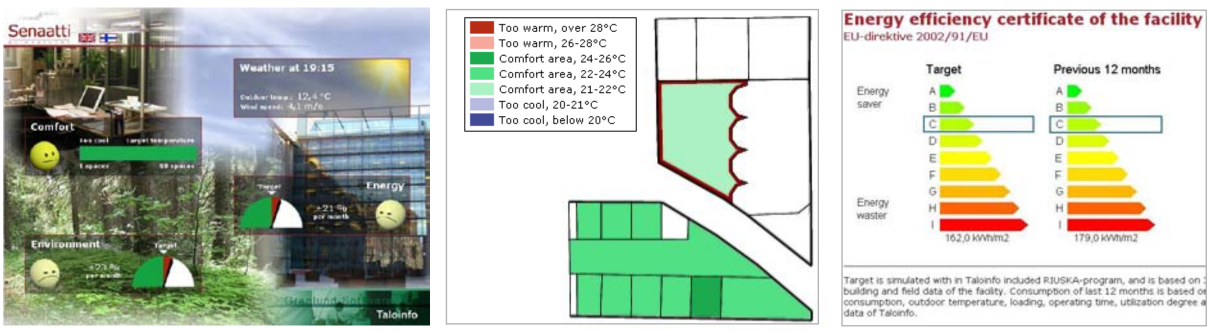

Figure 6. Self-reporting building system for BIM based performance management, including reports for top level performance evaluation (left), spatial comfort requirements verification (center) and on-line energy efficiency certificate (right).

BIM based approach also allows possibility to react on the changes in the use of the building by easy updating of energy and condition targets for monitoring and management systems.

DISCUSSION

The results show potential benefits of utilising BIM based concept in energy analysis: more efficient data input and more reuse for the existing data, possibility to use dynamic energy simulation instead of traditionally used static methods, support to use whole building spatial simulation instead of traditionally used zone based approach. However from the whole building process perspective, the biggest benefit is that by using BIM it is possible to bring continuous verification of energy performance for the whole building life cycle.

The real project experiences also showed, that successful BIM based energy analysis requires BIM revision management with comparison possibilities between different revisions, easy-to-understand visualisations of the thermal performance for clients and architect model validation for energy analysis purposes.

ACKNOWLEDGEMENT

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Flexergy; a Methodical system approach for user oriented agent based process management of energy flows in the built environment

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SUMMARY

The main challenge for future developments of renewable energy infrastructure is to reach an acceptable price level and a minimized environmental impact. Design tools for renewable energy installations are available on the level of individual installations. The potential for the application of renewable energy installations in the build environment is defined in an earlier phase of development: the design of the energy infrastructure for new build or renovated district areas. Design tools for the energy infrastructure are now lacking: renewable energy installations, tuning demand and supply in a dynamic way and tuning to the decision process. The objective is to develop design tools starting from the existing dynamic tools for single installations. New technological solutions are needed to make renewable energy solutions natural, easy and intuitively understandable for architect and consultants. The paper will discuss the approach to reach these goals.

INTRODUCTION

There is a persistent discrepancy between increasing demands for comfort in buildings and the need to decrease the use of energy. Preservation of energy resources, occupant comfort and environmental impact limitation are the key issues of modern and sustainable architecture. A major portion of primary energy consumption, about 40 %, is due to create thermal comfort in buildings by heating, cooling, ventilating and lighting. During the last decades, the main focus of research in Building Services was on reduction of energy consumption of buildings. The strong focus on the energy reduction led to situations in which health and comfort are endangered.

Integration between end-user and building is the ultimate in the intelligent building concept. “Connecting” the end-user to a building is complex. User-connectivity, the combination of usability and user interface together, is studied and developed further. Information and communication technology connects people and helps them to communicate with the building. Clements-Croome emphased transdisciplinarity and interaction: “Any consideration of intelligent buildings, whether learning, designing or managing them, requires a freedom of thinking which can embrace transdisciplinary ideas and systems. The word transdisciplinary, is a truly holistic and highly interactive concept. Intelligent building strategies are dealing with multiple criteria and attempting to integrate ideas over a wide range of issues [1].”

When the comfort control system is not working adequately, a lot of energy is wasted by too much heating or cooling. As a result of this overshoot indoor temperature is the most common issue in occupants' complaints about thermal comfort.

When design building services traditionally the main attention went to the heating demands. Due to the change in buildings, equipment and out door climate, there is a strongly growing demand for electricity instead of heating, see figure 1. So when designing the focus has to be changed.

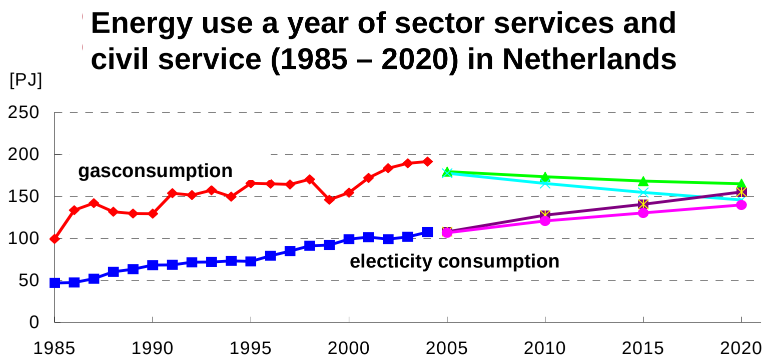

Figure 1. Development gas consumption and electricity demand the next 15 year [2].

Therefore it is important to look at energy reduction especially for this growing demand of a building. In office buildings most of the energy is needed for thermal comfort especially cooling. Present energy efficient technology is not sufficient to further reduce the energy use of buildings. New comfort control technology, such as individual control, offers new possibilities to further reduce energy consumption of office buildings. Dynamic online steering of individual comfort management and building management could save up to 20% of current energy consumption [3]. Misunderstandings and wrong conceptions about indoor comfort and energy use are common. Most office users are not even aware of the fact that they can affect the energy use. The behaviour of building occupants needs to be taken into account as it is responsible for almost half the outcome of planned energy reduction [4].

As until now the user has not been part of the building comfort system control strategy in offices, the energy consequences of the user behavior are not accounted for. New technological development is needed to incorporate the behavior of occupants of buildings.

Such novel control systems should not only improve the energy performance of the building, but should also offer benefits to users (i.e. building operators as well as workers). Comfort management should be linked with improving energy efficiency. Individual comfort management makes it possible to optimize comfort, energy efficiency and costs. This combination would be beneficial for building operators as well as occupants. Therefore in commercial buildings, the inclusion of options for individual comfort management is seen as an important feature to make such systems attractive to end users.

METHODOLOGY

Present control systems for office buildings already make use of new technical possibilities offered by computer networks. A next step in their development is the intelligent connection of the building networks with the Internet. The exploited Web is an interesting and successful storage place of information resources that can be used [3]. Comfort control systems could use dynamic real-time information from the Web about weather forecast, availability of energy and price level of energy. The information of the Web should be combined with information from the Building Management System (BMS) about the users, e.g. comfort demands or comfort preferences of the building occupants.

Agent Technology

Therefore, a new generation of control systems was developed with technology based on agent mediated communication over local networks and the Internet. Intelligent agents are autonomous and intentional pieces of software. These agents are capable of searching and sorting information from the Internet in order to perform certain tasks for the users they represent. Multi-agent systems provide the essential technology for this ICT-infrastructure[3];

- large numbers of actors are able to interact, in competition or in cooperation
- local agents focus on local interests and negotiate with more global agents
- implementation of distributed decision making by the negotiation processes between the different local or more global oriented agents
- communication between actors is minimized to generic information exchange between agents.

Previous work by Akkermans shows that the new Internet agent technology makes it possible to integrate occupants' behaviour with information sources from the Internet [3]. In two projects, SMART (Smart Multi Agent internet Technology) [5] and IIGO (Intelligent Internet mediated control in the built environment) [6] this technology was developed and tested.

Forgiving technology

A different type of technology to incorporate user behaviour, Forgiving Technology, was developed in another project, EBOB, Energy Efficient Behaviour in Office Buildings. EBOB investigated new combined technical and socio-economic solutions to make energy efficient behaviour natural, easy and intuitively understandable for the end-users of refurbished and new offices. Control scenarios for the HVAC systems were derived by analyzing occupants behavior and its effects on comfort and energy use. The EBOB project is an European 5th framework program project with eleven partners from five countries. EBOB ran from 2002 until 2005. The field test was held at Kropman's office at Rijswijk [7].

User representation

The techniques used within SMART/IIGO and EBOB made it possible to use the user representation and combine it with optimalization techniques. The representation of end-users was realized by developing an individual voting system; some results are shown in Figure 2. End-users were represented in the design by Fanger's comfort model [8]. Fanger's model predicts user's evaluations of the indoor climate in buildings. Using Fanger's model, the percentage of dissatisfied users can be predicted for a given set of comfort parameters.
The voting system allowed every user in a thermal zone to enter his vote (warmer/colder) within a voting period (e.g. one hour) while seeing the aggregated voting of other users in his zone at the moment of voting [5].

The users comfort needs dominate this control strategy. The control strategy is based on the description of the user behavior and implemented in a BMS (Building Management System). This BMS was extended with an external real-time information system to improve energy and comfort control. A learning curve is built from the user voting behavior. Responses of the user are interpreted differently depending on the overall trend of the comfort level in the building. Overall voting behavior as a function of the time of day is included in determining the action of the local comfort aspect controllers. Within this system the persistent use of user information is a leading strategy.

To work effectively with methodology, practitioners should learn to work with, and understand, the role of the user in the building design process. The building to be designed takes traditionally the central place in thinking of the design team. But in fact means and goal are mixed up. More and more the insight is growing that it is not the building to be designed that should be central but the needs of the humans for which the building is intended. This leads to a new approach in which the human needs are key aspects that have to be fulfilled, see figure 3.

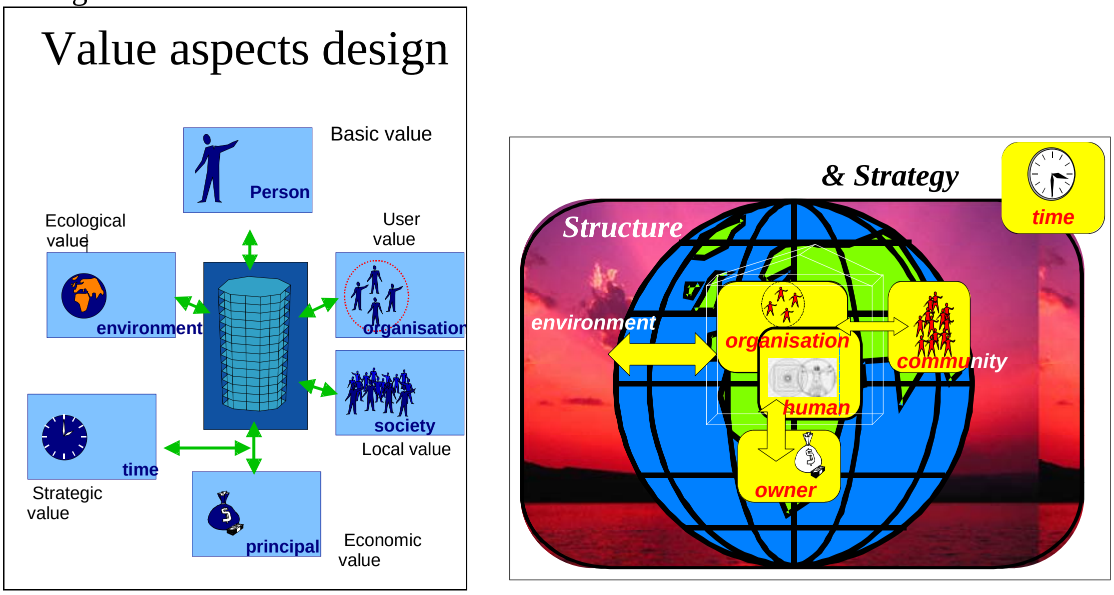

Figure 3: Strategic design, Paul Rutten [9] versus conceptuel representation of user interaction with the built environment

INTEGRAL DESIGN METHODOLOGY

Design is the key discipline that brings systems into being. In the engineering sciences, a lot of approaches have been developed to structure and optimize design processes: concurrent engineering, value engineering, design for manufacturability, systems engineering, quality function deployment, strategic design, etc. To develop our required model of design support, an existing model from the mechanical engineering domain was extended: Methodical Design [van den Kroonenberg [10,11,12] into an Integral Design methodology. The Integral design process can be described at the conceptual level as a chain of activities which starts with an abstract problem and which results in a solution. The original methodical design process is extended from three to four main phases, in which eight levels of functional hierarchical abstraction, stages can be distinguished. Starting by formulating the need, the program of demands is developed and transformed into functions to fulfill. This functional decomposition provides the means for decomposing complex design tasks into problems of manageable size. This functional decomposition is hierarchically so that the structure is partitioned into sets of functional subsystems. Decomposition is done until simple building components remain whose design is a relatively easy task, but each with each own focus, see figure 4.

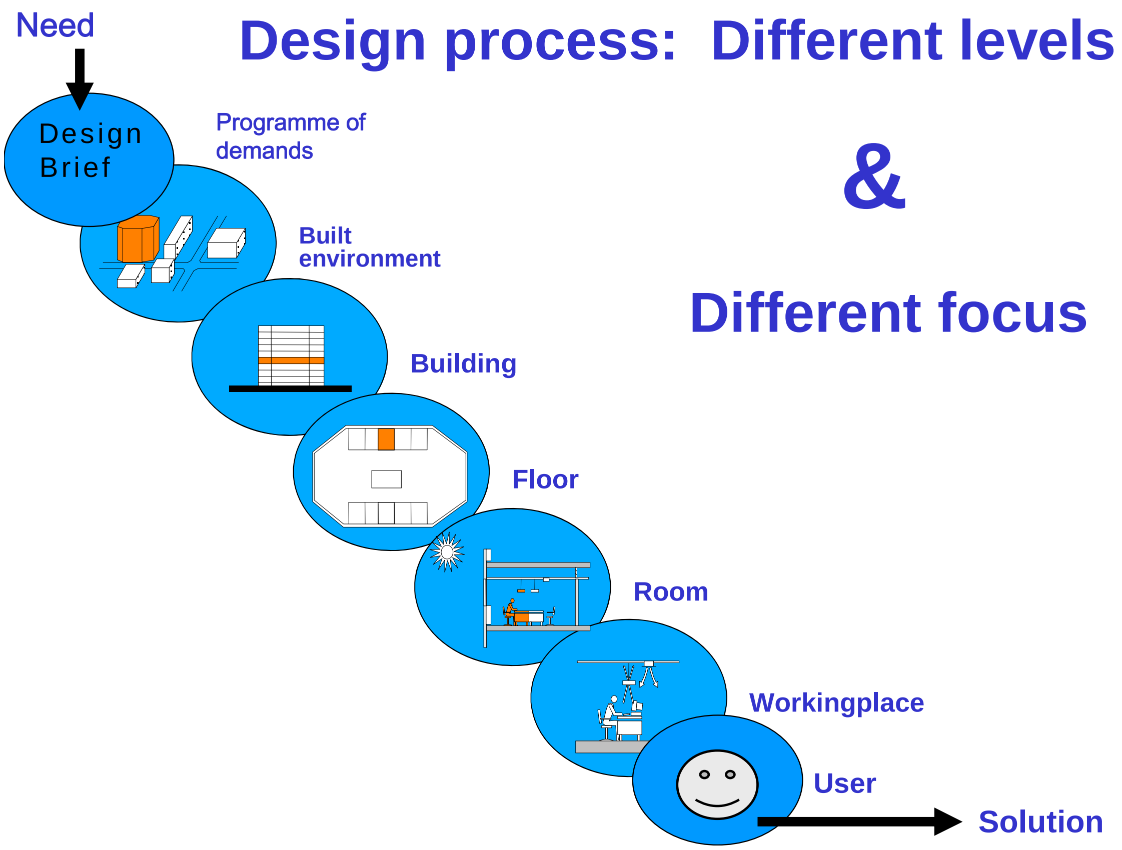

Figure 4: Hierarchical abstraction levels by functional decomposition

Hierarchical abstraction implies the decomposition of information into levels of increasing detail, where each level is used to define the entities in the level above. In this sense each level forms the abstract primitives of the level above. These higher-level terms form condensed expressions of a given relational and/or operational combination of primitives from the level below. Sets of generic components are located at distinct levels of abstraction. The contents of the layers are based on the technical vocabularies in use, technology-based layers or levels. Each layer represents an abstraction of the levels below. For a more extensive description of the models that formed the basis for the notion of technology-based layers see [13]. Separation is made between:
- Information level, knowledge-oriented, representing the "conceptual world".
- Process level, process oriented, representing the "symbolic world".
- Component level, device orientation, representing the "real world".

In addition, a new level is defined [14]: part level, parametric orientation, representing "the specification world". Thus, the four levels of aspect abstraction in the descriptive model of design are:

1. Information Level

This level deals with the knowledge of the systems by experts. One of the essential ideas behind this is that human intelligence has the capability of search and the possibility to redirect search. This information processing is based on prior design knowledge. One of the major problems in modelling design knowledge is in finding an appropriate set of concepts that the knowledge should refer to, or -in more fashionable terms- an ontology [13].

2. Process Level

This level deals with physical variables, parameters and processes. The set of processes collectively determines the functionality of the variables that represent the device properties. Modelling at the functional level involves the derivation of an abstract description of a product purely in terms of its functionality. This abstraction reduces the complexity of engineering design to the specification of the product's desired functionality.

3. Component Level

This level describes the hierarchical decomposition of the model in terms of functional components and is domain dependent. Generic components represent behaviours that are known to be physically realisable. They are generic in the sense that each component stands for a range of alternative realisations. This also implies that the generic components still have to be given their actual shape.

4. Part Level

This level describes the actual shape and specific parameters of the parts of which the components exist. Relevant technical or physical limitations manifest themselves in the values of a specific set of parameters belonging to the generic components. These parameters are used to get a rough impression, at the current level of abstraction, of the consequences of certain design choices for the final result.

Designing takes place in an environment that influences the process, it is contextually situated [15,16]. The context of the model of designing is defined by a "world view". The model of de Vries consists of 3 worlds and is extended by us to four worlds: the real world R, the symbolic world S, the conceptual world C and the specification world M, see figure 5. The Product – Process – Organization model (the PPO model), developed by Friedl [17] represents this "world view". Friedl's PPO model applies and elaborates comparable concepts as introduced in the Building Design Process Model according to Domain Theory [18]. Friedl's
PPO model discerns four essential domains of concern in the design situation:
- 'product domain'
- 'process domain'
- 'organization domain'
- 'context domain'

The relation between the extended model of de Vries and the PPO model is shown in figure 5;

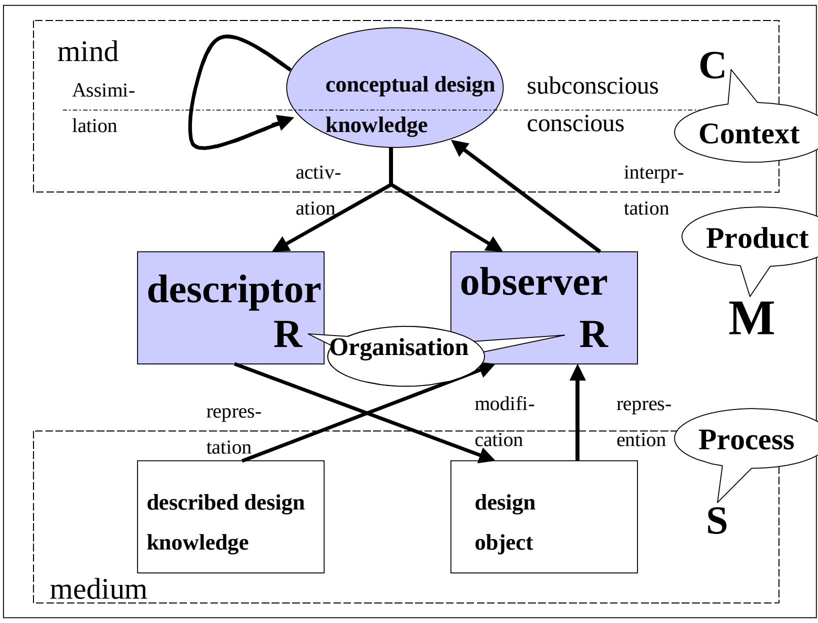

Figure 5: Extended model analytic schematic interaction model of designing

RESULTS & CONCLUSION

By implementing the insight that the user should become the leading factor within the process control, the following symbolic representation of the total process is shown in figure 6.

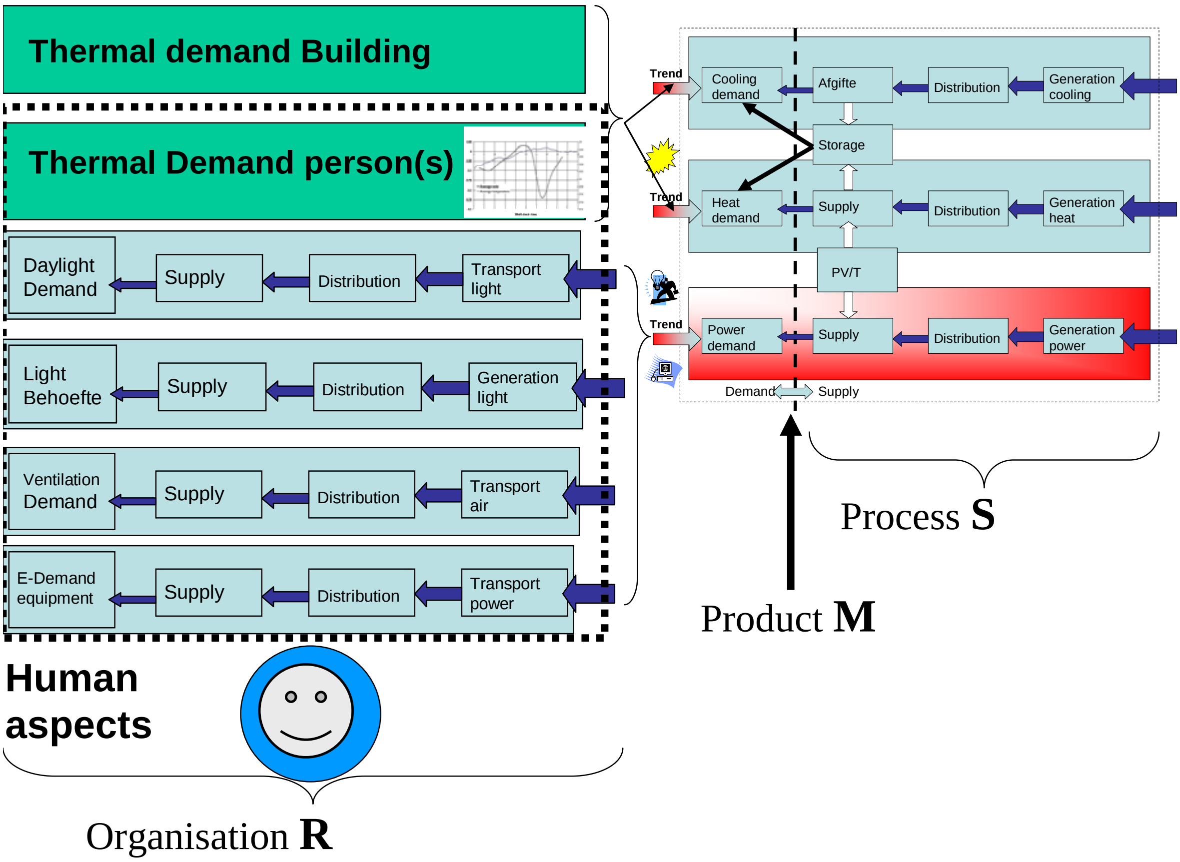

Figure 6; Process representation, the conceptual world C, presenting the context of the information model of energy flows within a building

In order to allow a stepwise approach in which each design decision has well defined implications, four different ontological levels are distinguished for designing energetic process:

* Information Model	* Conceptual World
* Physical Process Model	* Symbolic World
* Functional components Model	* Real World
* Parametric Model	* Specification World

These levels provide a structured framework for an integral design methodology. The benefits of methodical process design strongly depend on the experience of the design team members with the approach and its tools. For the Dutch building Industry workshops ‘Learning by doing’ are organized as part of the professional education of the members of BNA (Dutch Siciety of Architects) and ONRI (Dutch Society of Counsultants).

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The Design Challenges of Multipurpose Arenas

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SUMMARY

The design of the HVAC and ice rink refrigeration system in an arena is a challenging optimisation task. To ensure that the requirements for the indoor conditions and the ice temperature will be fulfilled, the HVAC and the ice rink systems have to be optimised with the interactivity of each other in mind. This challenging optimisation task is solved with advanced and effective multiphysics and CFD simulation tools. The heat and moisture balance of the arena is solved dynamically with multiphysics tool with 3D simulation of ice rink heat balance. CFD (Computational Fluid Dynamics) is used to support the design of air distribution and indoor thermal conditions. These calculations form the bases for the MEP design solutions where integration of heating, cooling, dehumidification, energy conservation and the use of outside energy resources are optimised to achieve the most economical investment.

INTRODUCTION

Most multipurpose arenas that exist today are usually used for skating in some form, for sport activities or for recreational and leisure purposes. The design of the HVAC and ice rink refrigeration system in an arena is a challenging optimisation task. Ice rink facilities share all the same concerns: energy usage, operation costs and indoor climate. Multipurpose arena design and operation are totally unique and differ in many ways from standard buildings. Each arena should be taken as an individual project. The design solution which functions greatly in the last built arena may not be the ideal solution for another new arena under construction. The project specific requirements for the indoor conditions and ice temperature during events as well as usage of the arena form the boundary conditions for the design project.

The most important to know about multipurpose arenas are to understand their features compared to other kind of buildings. These special features are due to /1/:

- Indoor conditions vary a lot depending on the usage of the arena. In an ice rink high inside temperature differences occur in the same indoor climate, from -4°C to +24°C. At the same time these internal climate zones must be controlled and stay stable. In concert situation there exists also large heat loads from equipment which should also be handled.
- Differences in indoor climate also cause humidity problems that must be under control.
- Air tightness is more important feature of the building envelope than thermal insulation.

The aim of this paper is to present HVAC and ice rink design solutions of the multipurpose arenas. The requirements for indoor conditions, in heat and moisture balance of the arena and ice rinks, CFD simulations and some guidance of HVAC design are presented.

REQUIREMENTS FOR INDOOR CONDITIONS AND ICE TEMPERATURE

The requirements for the indoor conditions and the ice temperature set a basis for design calculations of the HVAC and ice rink piping system in Multipurpose Arenas.
Indoor air design values for multipurpose arenas are presented in Table 1.

Table 1. Indoor air design values for multipurpose arenas /1,2/.

Action	Air temperature		Ice temperature	Max. relative humidity	Heat loads		Min. fresh air intake
	Rink at 1.5 m	Tribune (operative)			Lighting	People	
Hockey	[°C]	[°C]	[°C]	[%]	[W/m]		[dm³/s/ occupant]
- game	+6	+10…+15	-5	70	40	80/spectactor	4…8/spectator
-training	+6	+6…+15	-3	70	20	125/player	12/player
Figure							
-competition	+12	+10…+15	-4	70	40	80/spectactor	4…8/spectator
- training	+6	+6…+15	-3	70	20	125/player	12/skater
Other	+18	+18	-	-	20	80 W/person	8/person

The most of the Arenas are scheduled for an all-year round use with a short break during peak summer conditions. This all-year round use imposes challenges in the design work because of increasing outdoor heat and humidity loads.

In practise, the indoor temperature varies between 10-12 °C above the ice during practice session, to 12-14 °C during competition races. The relative humidity ranges between 40-65% all year around. The temperature in the spectator areas is expected to be slightly higher than above the ice, 14-15°C. The ice surface temperature is in the range of -3..-10 °C. In some cases the ice temperature is wanted to keep much lower than presented in Table 1. It is quite normal that the ice rink is dimensioned at -6°C. The air flow velocity above the ice should not exceed 0.25 m/s.

HEAT AND MOISTURE BALANCE OF THE ARENA AND ICE RINK

Quite often the ice rink piping is built “as the previous ice rink” without thoughtful analyses of the different factors that affects the ice temperature in the arena in question. To ensure that the requirements for the indoor conditions and the ice temperature will be fulfilled the HVAC and the ice rink systems have to be optimised with the interactivity of each other in mind. This challenging optimisation task can be solved with advanced and effective energy simulation, multiphysics and CFD simulation tools.

To establish the expected indoor conditions it is essential that the cold load to the space from the ice surface is taken into account. Vice versa, the ice rink cooling capacity is not possible to be calculated without taking into the account the different heat loads and the indoor conditions, its temperature and humidity. To figure out the complexity of the design work, Figure 1 shows the interactions between different factors.

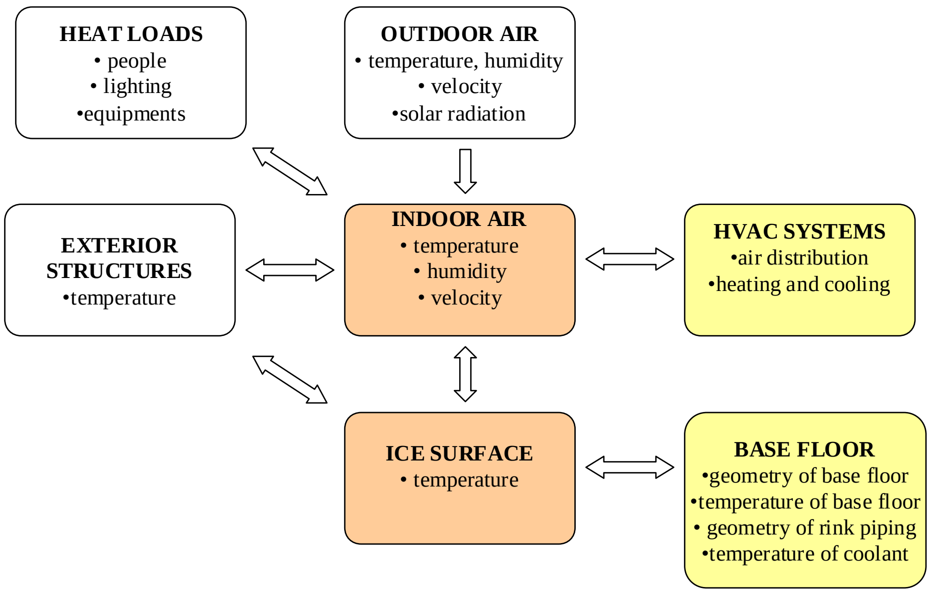

Figure 1. Interactions between different factors in Arena.

As we can see from the figure 1 there are plenty of factors that affect the indoor conditions. These are heat loads from the spectators and competitors, lighting and equipment, infiltrated and conducted outdoor heat loads, the supply air, additional heating load and the cooling load from the ice surface trough conduction and condensation. The people supply air and condensation rate to the ice surface and outdoor infiltration affects in addition the humidity of the indoor air. The moisture from the air condenses on the ice surface and has therefore a drying effect on the indoor air humidity.

Heat balance of the arena is presented in Eq.1.

$$\rho c_p V \frac{\partial T}{\partial t} = Q_{people}(t) + Q_{ligh}(t) + Q_{eq}(t) + Q_{AC}(t) + Q_{\inf}(t) + Q_{str}(t) + Q_{ice}(t)\,, \tag{1}$$

where ρ is the density of the air, c_p heat capacity of the air, V is the volume of the Arena, T is the temperature of the air, t is calculation time and Q is the heat load. The subscript *ligh* denotes lighting, *eq* denotes equipment, *AC* denotes air conditioning, *inf* denotes infiltration and *str* denotes structures.

Equation 2 presents moisture balance of the arena.

$$\rho V \frac{\partial x}{\partial t} = \dot{X}_{AC}(t) + \dot{X}_{aircirc}(t) + \dot{X}_{\inf}(t) + \dot{X}_{people}(t) + \dot{X}_{ice}(t) + \dot{X}_{other}(t)\,, \tag{2}$$

where x is the absolute humidity of the air and $\dot{X}$ moisture content. The subscript *aircirc* denotes circulated air.

The indoor conditions and the heat loads from people, lighting, equipment and building structures as well as the ground floor structure with ice rink piping have a strong effect on the ice surface temperature. In general, the final heat load to the ice surface varies between 155 to

320 W/m² depending on the activity and outdoor conditions. In practise the heat load to the ice is reasonable to keep under 220 W/m². Otherwise, the coolant inlet temperature to the ice rink piping has to be diminished radically down to -18°C to maintain the ice temperature at -6°C. In that case the energy economy of the refrigeration production will deteriorate substantially.

Figure 2 a) presents typical ice rink slab. The heat equation for the ground ice rink slab with boundary conditions for the ice surface, piping and ground are shown in Equations (3)-(6).

$$\rho_m c_{p,m} \frac{\partial T_m}{\partial t} = k_m \nabla^2 T_m \qquad (3)$$

$$k_m \frac{\partial T_m}{\partial y}(y = h) = Q_{conv,a} + Q_{rad} + Q_{cond} \qquad (4)$$

$$k_m \left(\frac{\partial T_m}{\partial y} + \frac{\partial T_m}{\partial y}\right) = Q_{conv,ht} \qquad (5)$$

$$k_m \frac{\partial T}{\partial y}(y = 0) = Q_{ground} \qquad (6)$$

where *k* is the heat conductivity and *h* is the height of the slab. Subscript *m* denotes material in the ground slab, *a* denotes air, *conv* denotes convection, *rad* radiation, *cond* condensation and *ht* heat transfer medium.

In Equation (4) the heat loads are the following:

- Convective load from the ambient air. The convective load depends on the temperature difference between the air and the ice and the heat transfer coefficient of the ice surface.
- Radiant load from the lighting and structures. The ice surface absorbs heat from the surrounding floors, walls, ceiling and lighting.
- Moisture load, condensation from the ambient air. The moisture load is proportional to the convective load and the ratio increases with increasing humidity in the air.

Additional loads, which influence on the ice rink cooling needs are the coolant circulation pumps, slab cooling during ice making, resurfacing and ice building during maintenance intervals and ice building.

To achieve indoor conditions and ice rink slab temperature distribution, the equations (1)-(6) are solved with multiphysics simulation software. In Figure 2 b) show an example of the temperature distribution of the ice rink slab. The geometry of the rink piping and the coolant inlet and outlet temperatures are essential in maintaining the ice surface at a required temperature level.

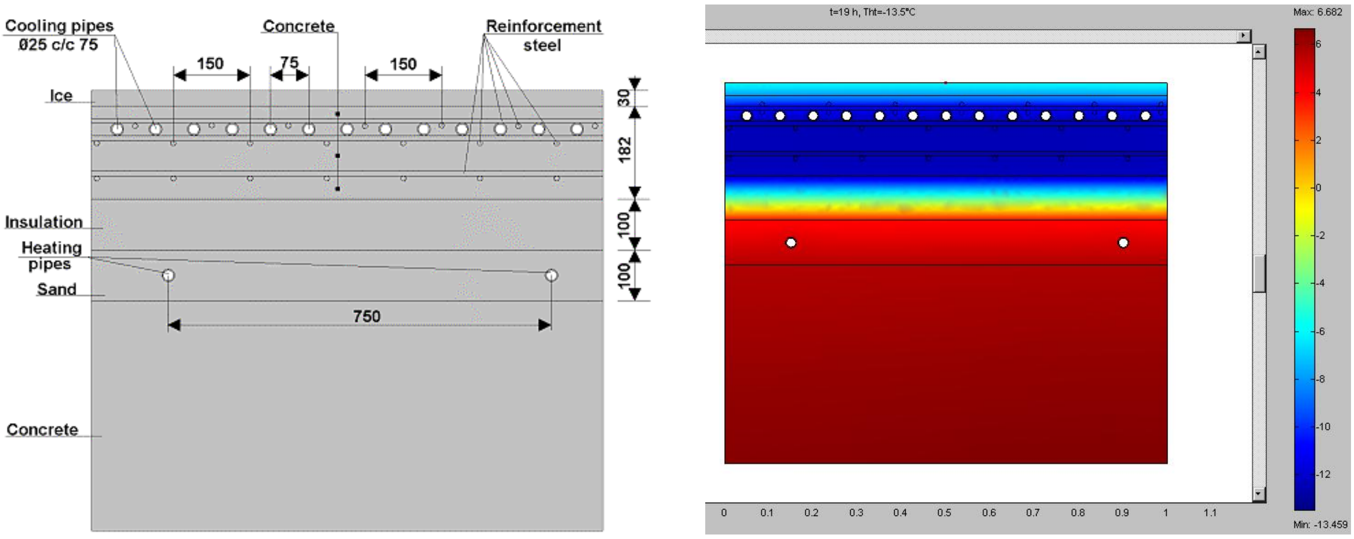

Figure 2. a) The ground floor with ice rink piping system, b) Example of the temperature distribution of the ice rink slab.

INDOOR AIR SIMULATIONS IN MULTIPURPOSE ARENAS

CFD is typically used to study indoor air conditions in spaces where design requirements are high and detailed information on flow field is important. CFD is a numerical calculation method and the realistic modelling of a indoor air environment is not an easy task. Supply air jets, forced and natural convection, and heat sources and sinks cause very complicated 3D flow field. At first, the important properties affecting flow phenomena have to be identified. Then the mathematical model is created and the space is divided into small control volumes. The conservation equations of mass, momentum, and energy are discretized and solved approximately using a numerical method. The accuracy of this calculation depends on the mathematical model, the numerical method, the calculation mesh, assumptions, initial values and iteration convergence. The typical focus is to compare different supply air distribution systems and wall constructions, and heat sources, which affect the indoor air environment. In addition, the results can be used to illustrate HVAC solutions to customers.

The first round of simulations involves individual air supply devices to test and compare operating conditions. The results allow the design team to choose the appropriate devices for each specific location. The device simulation results are compared to air jet theory and to the manufacturer’s profile data and measurements if possible. The simulated supply air jet profiles are then used as boundary conditions in whole arena simulations. The device models and simulation results are then saved to an object library for future projects.

The challenge is to design the supply air distribution so that fresh air flows to fully occupied zones and improves the thermal conditions. Draft, humidity, and temperature levels during different types of events in winter and summer conditions are considered. The benefits of the simulations are that they provide the possibility to try out different air flow device types or supply air systems, such as mixing, displacement, or a combination of both. Usually, first assumptions have to be corrected several times before the target is reached. It seems like a correctly performed CFD simulation is the only calculation method that can capture the indoor air flow field with the accuracy necessary for design purposes. In figure 3 is shown the simulated temperature distribution in arena during the ice hockey and concert events on summer time design day conditions.

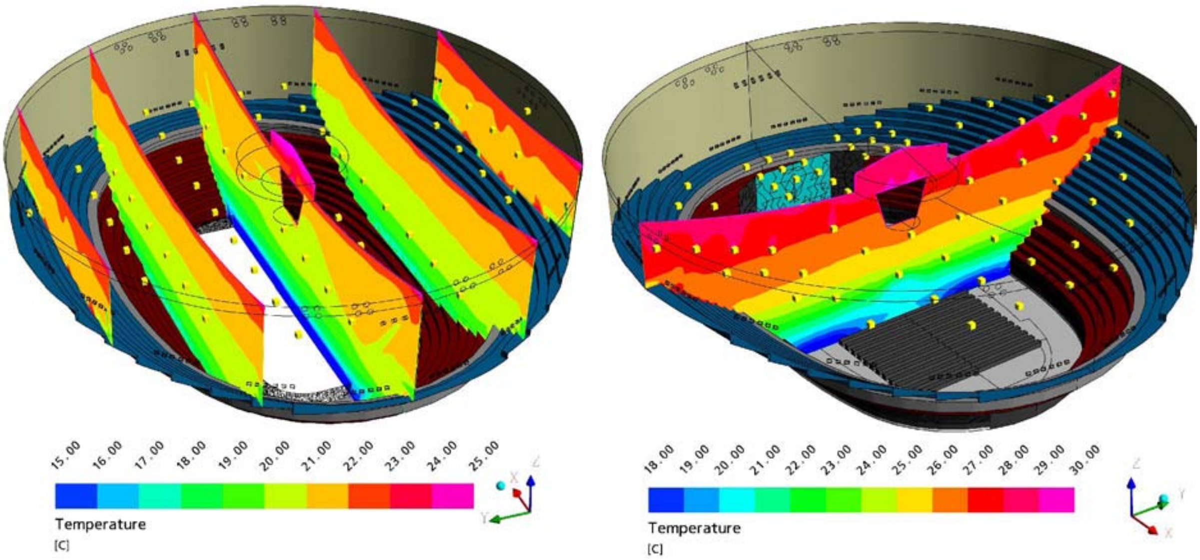

Figure 3. Temperature stratification of the Hodynka Arena in Moscow. On the left-hand side the ice hockey event and on the right-hand side the concert event.

The most challenging aspect in CFD simulations is how to have a realistic simulation results for design purposes. The conclusions should be made with care and compare different models and assumptions with experiments before the realistic results can be reached.

THE HVAC DESIGN

To maintain optimum indoor conditions all year around is a challenging task for the HVAC designer. The usage based on event varies greatly from maximum spectators to home team practice sessions. Then we have a very big open space where the occupied zone is in focus with special consideration to the ice area where a low temperature is a bonus but no condensation or fog build up is appreciated. It is obvious that for the base events such as ice-hockey, concert, small scale PR, ice-hockey practice, maintenance (empty) have to be calculated at winter and summer conditions in order to find out the design air flows and capacities for the these different functions of the air-conditioning units. It is sometimes useful to check with autumn conditions to see if the moisture balance can be obtained. The indoor condition calculations and CFD simulations set basis for HVAC design.

The moisture balance within the design set-points is as important as the temperature balance. The impact of the ice surface is essential to take into account. In cold to moderate climates the maximum air flows obtained for temperature balance are usually sufficient for moisture control. Quite often the spring conditions with ice-hockey game with maximum ventilation air demand and maximum indoor loads gives the design load for cooling. The same is for winter conditions with an empty hall, where optimum indoor conditions are maintained, gives the peak load for heating and autumn for dehumidification. If the skating activity is an yearly activity dehumidification during the summer period has to be calculated as well.

In the arena a heating demand exists as long as we have an ice surface, even during summer period.

Since the arena space itself has usually no outside walls, the only outside surface is the roof. The heat load from sun radiation to the arena is partly connective and partly radiation. The connective heat transmission has a small impact on loads, where as the radiation to the surfaces below are contributing to the total load which has to added to the total load. For some time low emissivity surface materials have been developed to diminish this factor.

The air distribution is one of the most interesting challenges in the design process. There are several parameters to consider in finding the most optimal solution. We want to achieve as low ice surface velocity as possible to keep the connective heat flow to the ice surface at a minimum level. In the spectator area with full seating we have a natural connective vertical thermal flow which will be very decisive for the arena flow pattern. This will create a high degree of stratification which with a displacement supply air distribution is an aim we want to achieve. At the same time a small vertical stratification is also desired to minimise the heating capacity demand and to avoid condensation on envelope structures during small load events such as practice sessions or maintenance periods.

There are very few options to show with design calculations how a solution will work, except with CFD modelling tools. It also gives a chance to test several air flow patterns. The air distribution solution has to selected in detail with the selection of each supply and exhaust air device. Only then is it possibly to find the optimum air distribution in the arena with a CFD simulations method.

It seems reasonable to apply several supply air methods in order to optimise the indoor climate and energy usage. During maintenance or practice periods the solution with vertical induction jets outside the ice surface in a circular pattern seems to have the best possible mixing with the space itself giving a small temperature gradient. At the same time it is possible to avoid the thermal influence of heating or dehumidification during spring and summer conditions. During events with spectators displacement ventilation around the ice rink in combination of active displacement diffuser in the perimeter of the upper spectator area seems to give the best end result.

To maintain a good indoor climate and achieve a low energy usage an advanced control strategy has to be adopted. To keep even indoor conditions at a variety of internal load situations the temperature, relative humidity and CO2 is monitored at several locations in the arena space. Usually CO2 is chosen since it is easily measured and has a rather good correlation with indoor air quality and many standards gives also level limits.
From the event calculations pre-programmed event control strategies can easily be determined.

Even if fire protections is outside the focus of this paper it is on important parameter to consider. The air-conditioning solution at best supports the fire protection solution giving small or no additional HVAC installations.

DISCUSSION

HVAC and ice rink design of the multipurpose arena is challenging optimisation task. The indoor conditions vary a lot according to usage of the hall. Different factors influences indoor conditions. To meet the requirements of energy efficiency and good indoor conditions the influences of the heat loads, ice surface, HVAC system and structures have to take into consideration. The utilisation of the advanced simulation and modelling methods together with multiphysic and CFD calculation tools ensure good background for HVAC and ice rink design.

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Virtual indoor environment model for an open-plan office

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SUMMARY

3D visualization methods have often been used in order to illustrate a building during the design process. This gives the building owner a better comprehension of the planned premises and improves the possibilities to select between different design options. The quality of the indoor environment is usually not included in the visualizations since it can be only predicted by using specialized software. However, the visualization of e.g. the air temperature, draught rating and contaminant distribution predicted with a CFD model would enable easier comparison of different room air conditioning solutions. Other indoor environment variables such as lighting and acoustics can also be predicted and included in the visualization. In this study, a virtual indoor environment model was created for a part of an open space office. The model is based on common VRML language standard for 3D presentations. It requires no custom software and can be viewed with a web browser plug-in. The model has a menu for user selection of indoor environment variables.

INTRODUCTION

3D visualization methods have often been used in order to illustrate a building during the design process. This gives the building owner a better comprehension of the planned premises and improves the possibilities to select between different design options. The quality of the indoor environment is usually not included in the visualizations since it can be only predicted by using specialized software. However, the visualization of e.g. the air temperature, draught rating and contaminant distribution predicted with a CFD model would enable easier comparison of different room air conditioning solutions.

From the standpoint of the end user it is important to set demands on the environment and to have a clear picture of the quality of the final environment. It is therefore of important to visualize the predicted environment is in such a way that it can act as an interface between the designer and the end user. Furthermore, in the design and construction phases of a building there are many partners who must co-operate with each other. There is therefore a need to enhance the speed and accuracy of the design phase and to improve the communication between different partners.

In the past, 3D visualization has been restricted to special software products and powerful workstations. At the moment, thanks to the requirements set by computer games, all standard computers have some level of hardware support to 3D graphics. This has created new possibilities to utilize 3D graphics for visualization purposes and distribute 3D models as standard documents. Applications like Google Earth demonstrate how 3D models of buildings can be made available through the internet.

A standard format for presenting 3D graphics is important for the distribution of visualization files. VRML format was developed for that purpose already in 1995. It has later been updated to VRML2 in 1997 [1] and to the present X3D format in 2002. It is an open standard designed to present 3D material in the web. VRML models can be viewed with browser plug-ins or special viewer programs that are freely available. There are several possibilities to view the model. The user can walk or fly around in the model or examine it by rotating and zooming. VRML based models have been available in the internet for more than ten years. The popularity of VRML models, however, has been hindered by practical limitations such as limited bandwidth for downloading the models and inadequate power of standard computers.

The aim of this study was to demonstrate the concept of indoor environment visualization by combining several modeling results in one interactive model. The model was created for a part of an open space office by modeling the air flow pattern and indoor conditions by CFD simulation. The model is based on VRML format and has a menu for user selection of indoor environment variables. The study was a part of the Virtual Space 4D project, where several simulation methods were applied in order to model and visualize the indoor environment [2,3].

METHODS

The modeling was done for a part of an open space office located in Helsinki (Figures 1 and 2). The modeled area of 32 m^2 was adjacent to large windows facing southwest. It included two working posts with furniture, PC:s, lamps and two supply air units. The room height was 2.3 m.

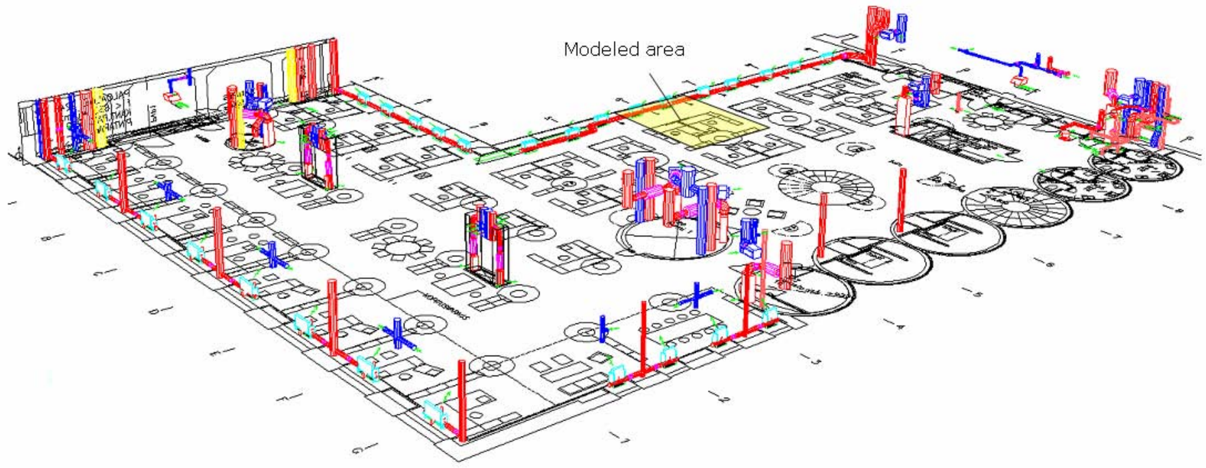

Figure 1. Layout of the open space office from the CAD drawing. The modeled area is shown.

Figure 2. Views from the modeled part of the open space office.

A simplified presentation of the studied area was created for CFD simulation with the pre-processor of the software package (Figure 3). Simplification of the actual geometry is necessary in order to reduce the computational requirements. The persons are presented by cylinders. Heat is produced in the space by the persons, PC:s and lamps. The total heat load was 18 W/m^2 floor area. The modeled situation was an early spring or late autumn day with 0 ºC outdoor temperature. The large window had a surface temperature 4 ºC lower than the room temperature. The heat loads of the model are shown in Table 1. Radiative heat transfer from the heat loads to the room surfaces was calculated separately and is included in the table.

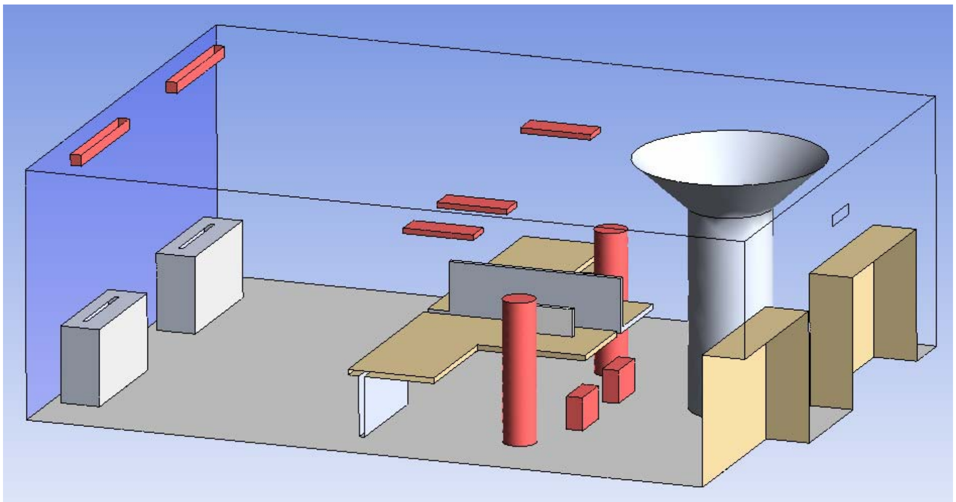

Figure 3. Simplified presentation of the modeled area for CFD-simulation.

Table 1. Heat sources and the distribution of convective and radiative heat loads.

		Convection		Radiation		
Object	Heat power (W)		To objects (W)		To floor (W)	To ceiling (W)
PC + LCD-display 1	100	50 %	50	50 %	30	20
PC + LCD-display 2	100	50 %	50	50 %	30	20
Person 1	100	50 %	50	50 %	25	25
Person 2	100	50 %	50	50 %	25	25
Lamp near working post	110	50 %	55	50 %	20	35
Lamp on the table	48	50 %	24	50 %	10	14
Lamp near the window	28	50 %	14	50 %	7	7
Total heat load	586	50 %	293	50 %	147	146
Window (dT – 4 °C)	–450	75 %	–350	25 %	–50	–50

Supply air as well as heating and cooling were produced by two induction units in front of the window. The supply flow rate was 20 l/s per unit and the flow rate of induced room air was 80 l/s per unit. The inlet boundary condition for the CFD-model was determined from the measured air flow rate and momentum flow rate of the unit. The heating and cooling of induction air was controlled by a room thermostat. In the modeled situation, most of the total head load was compensated by the cooling power of the cold window. Therefore, the induction air was not heated or cooled by the units.

The CFD simulation was carried out by using Ansys CFX 10.0 software. The computational grid was unstructured and had 220 000 nodes. k-ε turbulence model was applied. The spreading of contaminants was visualized by placing carbon dioxide (CO_2) sources above the modeled persons.

Thermal comfort was estimated by calculating draught rating (DR) values from the formula [4]

$$DR = (34 - T_a)(V_o - 0.05)^{0.62}(37 \cdot I_o V_o + 3.14) \qquad (1)$$

where T_a is the air temperature, V_o is the mean air speed (≥ 0.05 m/s) and I_o is the turbulence intensity. I_o is defined as S_o / V_o, where S_o is the standard deviation of air speed (≤V_o). Recommendations for the draught rating are given in the CEN Report CR 1752 [5] for three quality levels: 15 %, 20 % and 25 %.

The omnidirectional time averages of air speed and turbulence intensity were calculated from formulas [6]

$$\frac{V_o}{V_V} = 1 + I_V^{\,2}, \; I_V \le 0.45; \qquad \frac{V_o}{V_V} = \frac{1.596 \cdot I_V^{\,2} + 0.266 \cdot I_V + 0.308}{0.173 + I_V}, \; I_V > 0.45 \qquad (2)$$

$$I_o = \sqrt{(1 + 3 I_V^{\,2}) \frac{V_V^{\,2}}{V_o^{\,2}} - 1} \qquad (3)$$

$$I_V = \frac{\sqrt{\frac{2}{3} k}}{V_V} \qquad (4)$$

where *k* and V_V are the values of turbulent kinetic energy and air velocity magnitude from the CFD results and I_V is the turbulence intensity calculated from *k* and V_V.

RESULTS

The simulation results were analyzed with the post-processor of the CFD software. Following graphical presentations were generated:

- isosurface of draught rating value 15 % (strictest requirement) colored with temperature distribution
- isosurface of draught rating value 30 % colored with temperature distribution
- distribution of draught rating in a plane perpendicular to the window
- visualization of flow pattern by velocity vectors in a plane perpendicular to the window colored with temperature distribution
- isosurface of CO_2 concentration value 680 ppm
- distribution of CO_2 concentration in a plane perpendicular to the window

The visualization results were exported to separate VRML files and one file was exported with only the room geometry. The interactive model was then created manually from the exported files. The visualized environment variables (isosurfaces, contour diagrams and vector field) were copied from the separate files and pasted as individual objects in the file with the empty room. A floating menu was then created manually by utilizing and modifying the selection buttons that were created by the post processor to the exported VRML file. The menu buttons were then programmed to switch on and off the copied visualization objects. The selected items are indicated with red spheres in the menu. The cylinders that represented persons in the simulation were replaced with VRML models of persons in the visualization.

The resulting virtual model can be seen in Figures 4-7 with different selections of the results. The visualization is viewed here with Cortona VRML Client browser plug-in by Parallel Graphics. The viewing mode (walk, fly, examine) can be selected from the buttons on the left.

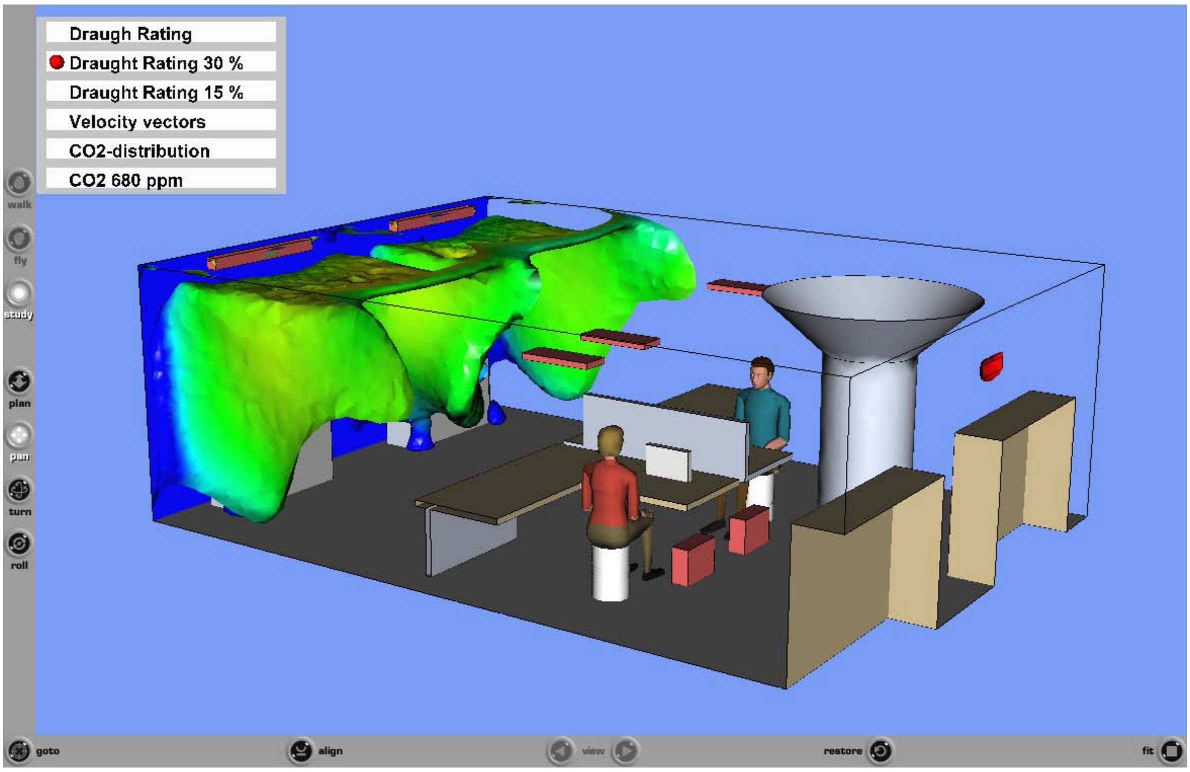

Figure 4. Virtual indoor environment model with a user menu for the selection of visualized variables.

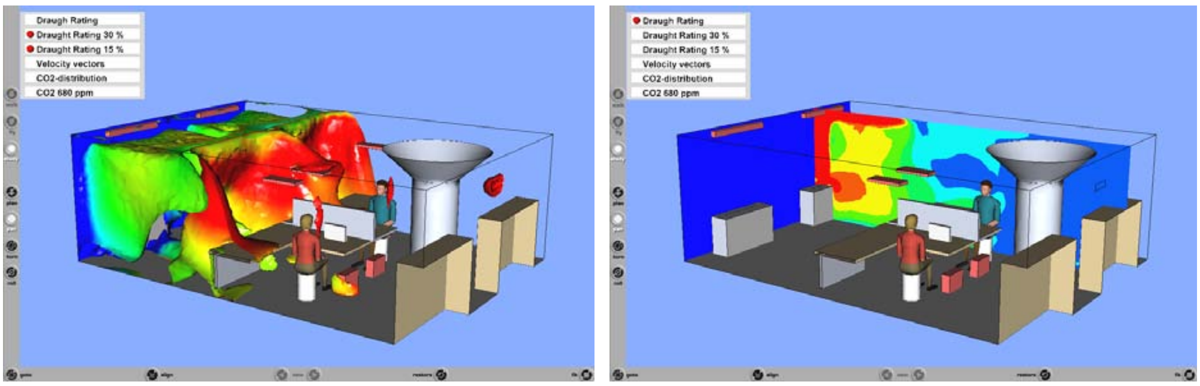

Figure 5. Two isosurfaces with different draught rating values with temperature color-coded (left) and distribution of draught rating in a plane, 5 % intervals between the contous (right).

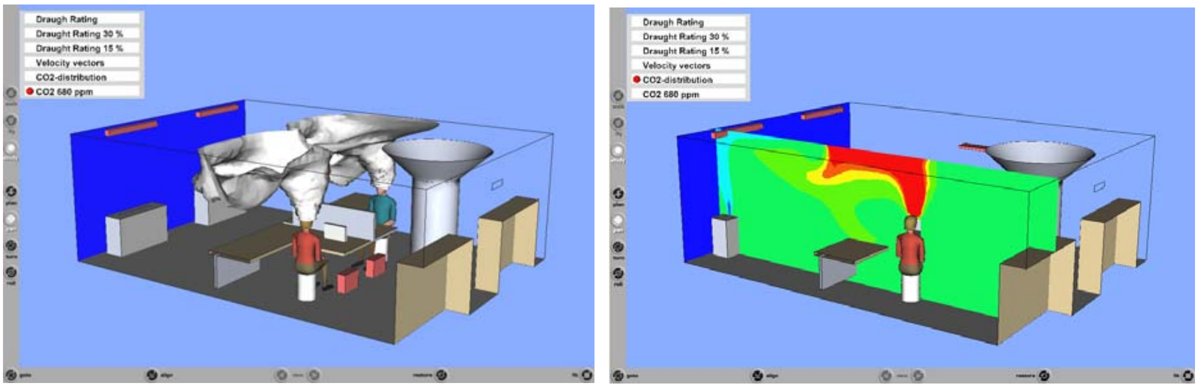

Figure 6. Spreading of carbon dioxide from the persons (left) and distribution of its concentration in a plane (right).

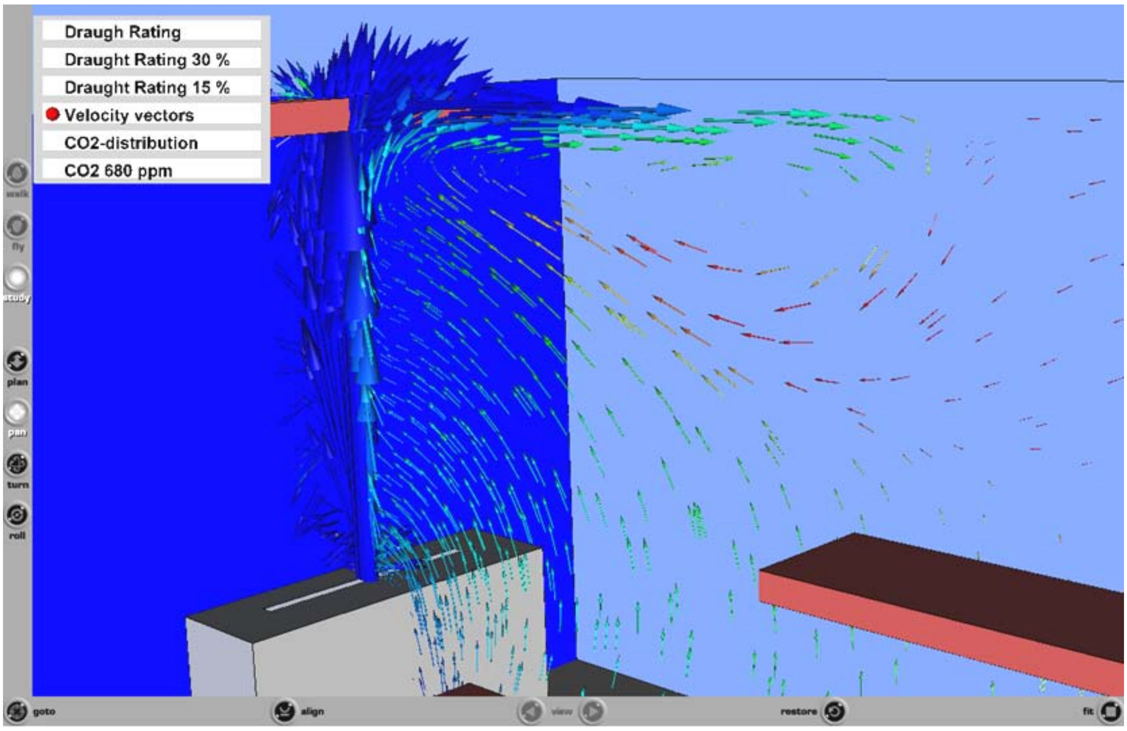

Figure 7. Flow pattern from the induction unit near the window.

DISCUSSION

The concept of virtual indoor environment model was demonstrated in this study by creating an interactive model of an office environment with a user menu for the selection of visualized variables. It represents a first step towards creating a visualization tool for enhancing the interaction between designer and end user in the design process. Such a tool can be useful e.g. in the following tasks:

- setting the user demands for the indoor environment
- visualizing the expected quality of indoor environment to the end user
- selecting between different design options
- showing the benefits of investment to the indoor environment quality
- visualizing the performance of the system in different outdoor and load conditions
- locating the room areas with a risk of discomfort
- finding optimal locations for the working posts

The VRML format offers a standard solution for the visualization of the indoor environment. The model can be equipped with a user menu for the selection of visualization variables. Other functionalities such as animations or sound can be added as well. The model can be distributed and viewed with a standard computer with a browser plug-in. VRML export is available in many modeling and visualization programs. The indoor environment model can be combined with architectural visualization. Modeling results from different applications can be combined in the same model, e.g. lighting visualization or auralization results from acoustical simulation. However, tools are needed for creating VRML models and adding functionality such as user menus or animation into the model. In this project the model functionality was built manually, which is not a practical method.

VRML is not the only available platform for indoor environment visualization. Application specific viewers are one way of distributing 3D models. These have been developed for modeling, visualization or CAD programs by the software companies for viewing the models in 3D format without having to buy and install the actual software product. They are typically meant for only one program and do not accept other formats.

A new proposal for a global standard for the presentation of 3D-graphics called Universal 3D (U3D) has recently been suggested by ECMA [7]. It has also been adopted in the new versions of Adobe Portable Document Format (pdf). It offers a promising platform for distributing 3D-models. The big advantage over specialized viewers is that pdf is a widely used de facto standard format for distributing documents. No installation of software is needed, since most computers already have a pdf reader installed. It remains to be seen, how widely this format will be adopted in the future.

The benefits of 3D visualization are obvious in many fields including the indoor environment. A lot of technical development is still needed for standardizing the file formats, building modeling tools and converters, etc. Multidisciplinary co-operation is also needed to develop the utilization of visualization in the design process in order to design the indoor environment to meet the user's needs.

ACKNOWLEDGEMENT

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A Sense Diary System for Intelligent Buildings

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SUMMARY

This paper presents a novel device called a Sense Diary for intelligent buildings. To introduce the new device and its applied system, a brief review of sensor-based systems applied in buildings has been conducted, which is based on current literature and products. The aim of this review is to understand current research and development situation of sensor-based systems that have been developed for all kinds of buildings and their systems across the world, and to further identify specific research tasks for an UK government EPSRC funded project focusing on innovative solutions of building façade systems. For acquiring the knowledge of sensor-based systems for buildings, some target information sources have been subjectively chosen. According to the literature and products review, several key research tasks of a Sense Diary System have been identified and discussed. It is concluded that the proposed Sense Diary and its applied system have advantages by providing daily real-time feedback to the facilities managers on environmental conditions thus improving control and health of occupants.

INTRODUCTION

This paper introduces a novel sensor-based subjective opinion feedback system called a Sense Diary for intelligent buildings. The initiative of this research was led by Clements-Croome in 1990 [1] and is being deployed through an IDCOP (Innovation in Design, Construction & Operation of Buildings for People) research project <http://www.idcop.soton.ac.uk/>, which is funded by the Engineering and Physical Sciences Research Council (EPSRC) in UK.

The EPSRC [2] has been conducting a research programme called Sustainable Urban Environment (SUE) since 2001. The key drivers of the SUE programme are to improve the quality of life of UK citizens, to support the sustainable development and to meet the needs of users of the built environment. The IDCOP is one of 12 research projects involved in current SUE programme, and it aims to develop innovative solutions with respect to the maintenance and refurbishment of existing buildings, specifically to find new ways to improve the performance of building envelopes over the whole building lifecycle [3]. There are two research themes:

- To develop sustainable façade technologies and enhanced methods for building refurbishment, and
- To reduce the consumption of non-renewable resources in a way that is economically viable and socially acceptable.

The research into a sensor-based system is linked to two IDCOP work packages, including embedded sensor systems and system personalisation. The research into a Sense Diary System has been deployed to achieve the following research targets, including

- a prototype of embedded sensors system with focus on occupants' well-being,
- a Sense Diary and its applied system that can be used at either work or residential places to register various degrees of either satisfaction or dissatisfaction by occupants in regard to commonly design environmental factors of buildings,
- to evaluate physical conditions of buildings while a Sense Diary system is used,
- to describe the well-being state of the occupants while they are using a Sense Diary, and
- to power a real-time facilities management process.

According to this deployment, a 4-stage plan has been made, which comprises of

- *A review of sensor-based systems for buildings*,
- *A new prototype of Sense Diary*,
- *The development of the Sense Diary and its applied system*, and
- *An experimental case study*.

This paper focuses on a brief review of current status of sensor-based systems applied in buildings across the world, and a concise introduction of the proposed Sense Diary system to be used for intelligent control of buildings. The review is based on current literature and products relevant to sensors and their applications in buildings and their systems. The aim of this review is to understand current research and development situation of sensor-based systems that have been developed for all kinds of buildings and their systems across the world, so as to further identify specific research tasks for the IDCOP project with focuses on innovative solutions of building envelop systems. For acquiring knowledge of sensor-based systems for buildings, both academic and professional information sources have been subjectively chosen, including *ScienceDirect Navigator* and IEEE Computer Society Digital Library (*IEEE Xplore*) for literature review and *ThomasNet* and *Google* for products review. According to the literature and products review, several key research tasks of the Sense Diary system are identified and discussed. It is concluded that the proposed Sense Diary and it applied system has the advantage in improving well-being and saving energy in buildings.

METHODOLOGY

Figure 1 illustrates a research framework for the review of literature and products and then the development of Sense Diary and its applied system. The research into a Sense Diary system for intelligent buildings has been divided into two stages in terms of related theoretical and methodological contexts:

- *The review stage*: Comprehensive information will be collected from three sources, including academic information sources (such as the *ScienceDirect Navigator*, the IEEE Digital Library, i.e., the *IEEE Xplore* and the *Elsevier EI Compendex*), professional information sources (such as the *ThomasNet*, the *Questex Media Group Sensors* and the *GlobalSpec Search Engine*), and general information sources (such as the *Google*).
- *The development stage*: System analysis and development method will be used to build up the Sense Diary system architecture and to program for a demo system.

With regard to each connection between a preceding stage and a following stage, which is based on a schedule for deploying each research task, a research and development strategy is being set up, which focuses on the products literature review, and the guidance for developing the Sense Diary system.

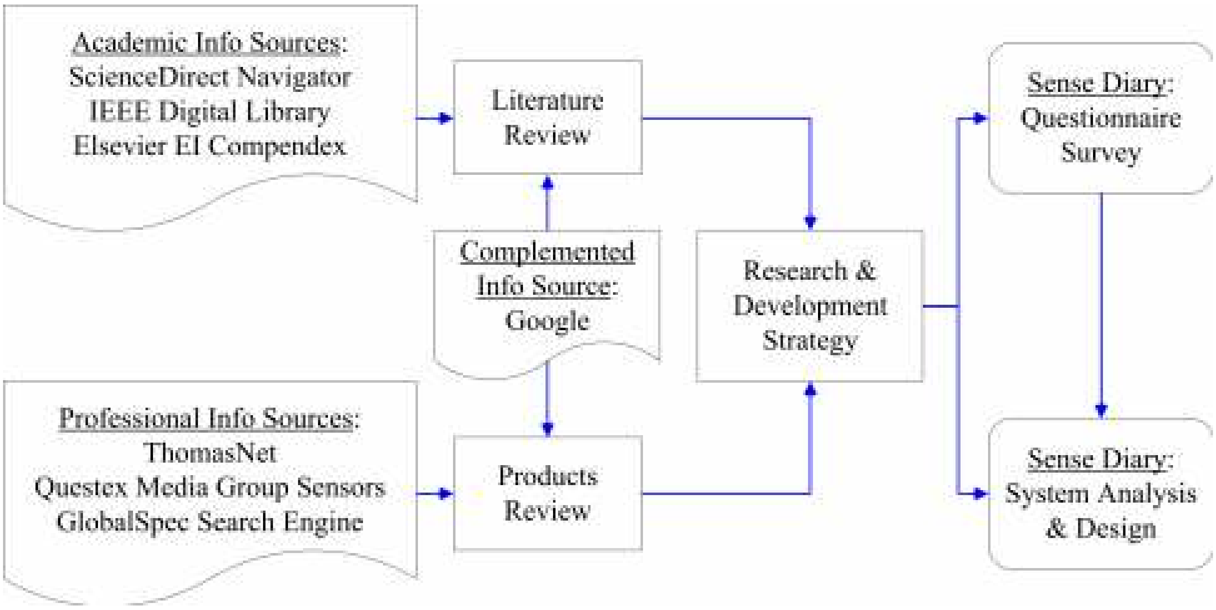

Figure 1: A framework for the research processes.

There are plenty of information sources around the world; however, to select a source of information for literature and products review, two criteria are adopted, depending on whether it is an *original source* of relevant information or whether it is an *authoritative source* of relevant information. Based on these criteria, several online databases have been selected from academic, professional or general information sources (see Figure 1).

LITERATURE REVIEW

Two main kinds of academic literature have been targeted and pursued, including periodicals and books. The reason for this selection is that periodicals can normally provide the most recent information about the review target; while books can give comprehensive information about relevant issues.

According to recent literatures such as the book edited by Gassmann and Meixner [4], the main areas of *sensor-based systems for buildings* cover the following five streams of research and development:

- *energy and heating ventilation & air conditioning* (HVAC) such as self-commissioned heating control system using Artificial Neural Networks (ANN), air-conditioning control, air quality measurement and management, sensor-based management of energy and thermal comfort, sensors in HVAC systems for metering and energy cost allocation, and pressure sensors in the HVAC systems, etc.;
- *information and transportation* such as wireless and M-Bus enabled metering devices, field-bus systems, wireless in-building networks, sensor systems in modern high-rise elevators, and sensing chair and floor using distributed contact sensors, etc.;
- *safety and security* such as life safety and security systems, biometric authentication for access control, smart cameras for intelligent buildings, and load sensing for improved site safety, etc.;
- *maintenance and facilities management* such as maintenance management in industrial installations, and worldwide facility management, etc. and

- *system technologies* such as sensor systems in intelligent buildings, and system technologies for private homes, etc.

The review has shown that modern sensor-based systems based on structured Wireless Sensor Network (WSN) have been applied in facilities management for the energy supply system, the HVAC system, and the safety and security system in buildings. In this regard, the proposed research into a Sense Diary system, which aims to set up an occupancy feedback system to be used at work or residential places to register various degrees of satisfaction/dissatisfaction with commonly design environmental factors, is within the main areas in the research and development of sensor-based systems for buildings, and the unique feature of the proposed Sense Diary system indicates that it is a novel device for aiding facilities management with regard to improving the energy efficiency of buildings and the well-being of occupants.

Davidsson and Boman [5] present a multi-agent system for controlling intelligent buildings. In a de-regulated market the distribution utilities will compete with benefit for the customer in addition to the delivery of energy. They describe a system consisting of a collection of software agents that monitor and control an office building. It uses the existing power lines for the communication between the agents and the electrical devices of the building, such as sensors and actuators for lights, heating, and ventilation. The objectives are both energy saving and increasing customer satisfaction through value added services. Results of qualitative simulations and quantitative analysis based on thermodynamically modelling of an office building and its staff using four different approaches for controlling the building indicates that significant energy savings, up to 40 per cent, can be achieved by using the agent-based approach. The evaluation also shows that customer satisfaction can be increased in most situations. In fact, this approach makes it possible to control the trade-off between energy saving and customer satisfaction (and actually increase both in comparison with current approaches).

PRODUCTS REVIEW

The literature review shows that the research and development of a Sense Diary system for buildings is a creative and innovative practice. However, this practice need to be deployed based on current level of relevant technologies. In this regard, a products review has been conducted, which aims to find out some relevant industrial systems that can be used for reference as follow.

Wireless Sensor Network

A wireless sensor network (WSN) is a computer network consisting of spatially distributed autonomous devices using sensors to cooperatively monitor physical or environmental conditions, such as temperature, sound, vibration, pressure, motion or pollutants, at different locations [6,7]. The development of WSN was originally motivated by military applications such as battlefield surveillance; however, the WSN is now used in many civilian application areas, including environment and habitat monitoring, healthcare applications, home automation, and traffic control [6,8]. The WSN can bring a broad range of diverse applications from house to commercial buildings, from simple system to complex system. The advantage of WSN set up based on wireless network other than traditional wired network can just fulfil the requirements of occupants who want much interior space from the building services systems.

To review current applications of WSN in the built environment, two commercial systems have been identified as good applications. One example is for commercial buildings and the other one is for residential buildings. For the WSN applied in commercial buildings as well as households, the example is GAMMA Building Management System from Siemens [9]; meanwhile, for the WSN applied in residential buildings, the example is the CyberEye system from Sungji [10].

Commercial Initiatives

As the proposed Sense Diary is to be integrated with a local WSN either within a residential building or within an office building, another part of products review goes to control devices; and six set of commercial systems have been reviewed, including

- The Secom building services control station [11], which is a *Building Services Panel* that tenants can use to control the turning on and turning off lighting and air conditioning according to pre-defined time zones.
- The Ambi-Rad SmartCom Control Unit [12], which is designed for warm air heater units and radiant systems and has been designed to consolidate the function of all previous control panels into one unit.
- The Echelon NES system [13,14], which is a state-of-the-art, future-proof smart metering system that brings new features and benefits to every aspect of ones utility's operations - from metering and customer services to distribution operations and value-added business.
- The TAC Satchwell MicroNet [15], which is a modular building management system that is easy to specify, operate, and tailor. It is suitable for small to medium scale operations and is sold through a locally accredited partner network.
- The Microsoft Home [16,17], which simulates a domestic environment including a front door, entry/foyer, kitchen, family room, dining room, entertainment room and bedroom. The Microsoft's technology-laden home of the future aims to feature smart appliances connected by Web services and controlled by tablet PCs or cell phones.
- The Siemens Smart House [18], which is a range of options that enhance the functionality of any home and are easy to use. Once some basic *Standard Fit*, including pre-wired cables for lighting, heating, blinds, surround sound, audio, video, data, voice and living lighting, is installed, then home owners can really make their homes 'Smart' and evolve it as they may change the way of living, and it is also believed that the choice is endless. There are three options below for home owners to choose and match in regard to the level of functionality they require, and the three Siemens Smart Homes options are:
  - The *Silver SMART Home Technology* package includes all the preliminary systems detailed in the *Standard Fit* package,
  - The *Gold SMART Home Technology* package includes *Standard Fit* preliminary cabling, plus all the features in the Silver package.
  - The *Platinum SMART Home Technology* package includes *Standard Fit* preliminary cabling, plus all the features listed in both the Silver package plus Gold package upgrades**.**

As the proposed Sense Diary system aims to develop a new control panel for intelligent buildings control in regard to real-time occupants' opinions, products review has mostly gone to devices and relevant technical platforms that all current systems adopted. Mobile phone and PDA are currently popular intermediaries applied in buildings management systems [17-19]; in addition, touch screen devices are also popularly used in some buildings [19]. For example, mobile phones are used as a controller in Siemens Smart Home [18], and PDA is also used in buildings control [20]. All these information indicates that the intermediary of the proposed Sense Diary system needs to be chosen from these devices or to be embedded into all kinds of these devices.

RESEARCH AND DEVELOPMENT

Strategy

The trend of research and development has been identified through literature and product review, including

- Multi-functional devices and systems based on WSN,
- Such devices need to be based on international standards,
- Such devices need to be with integrative functions,
- Easy to use and to be reliable and energy-efficient.

Although there are some relevant devices using desktop, laptop, PDA and touch screen, the proposed Sense Diary system is new and the idea has a very practical purpose but will contribute to sustainability.

A Sense Diary System

The research into "Embedded Sensor Technology and Occupant Behaviour" aims to help building professionals to further understand building occupancy or use patterns during a period of occupancy time. This study will explore relevant issues related to the Sense Diary System, which is expected to aid managing energy use and monitoring building performance besides assessing occupants' well-being requirements in regard to real-time performance of building services systems and building façade system. In particular, this study aims to identify the occupancy behaviour patterns associated with building use so as to find out a possible relationship among these variables, their carriers and energy use. As a result, a Sense Diary as a prototype intelligent sensor-based device to monitor these variables is introduced, and is to be developed and tested in collaboration with industrial leaders such as Siemens and Echelon, who promote interaction between UK science, engineering and industry.

To facilitate the research going further, the definitions of Sense Diary and its applied system are proposed first based on the idea proposed in the 1980s by Clements-Croome [1,21]. The Sense Diary is a user feedback panel which can be used by building occupants for inputting their opinions about well-being requirements based on indoor environmental status indication and their psychological feelings. The Sense Diary in the format of software can be programmed into different devices such as PDA, laptop, desktop PC and even mobile phone and TV. The Sense Diary within the environment of Ethernet and/or wireless network of the building can exchange data with wired network server and/or wireless network server; various sensor data parameters such as temperature, air quality, light and sound, etc. can be received by the Sense Diary, and Sense Diary can transmit occupants' feedbacks to remote control centre for building services system management or facilities management. The architecture of the Sense Diary System is illustrated in Figure 2.

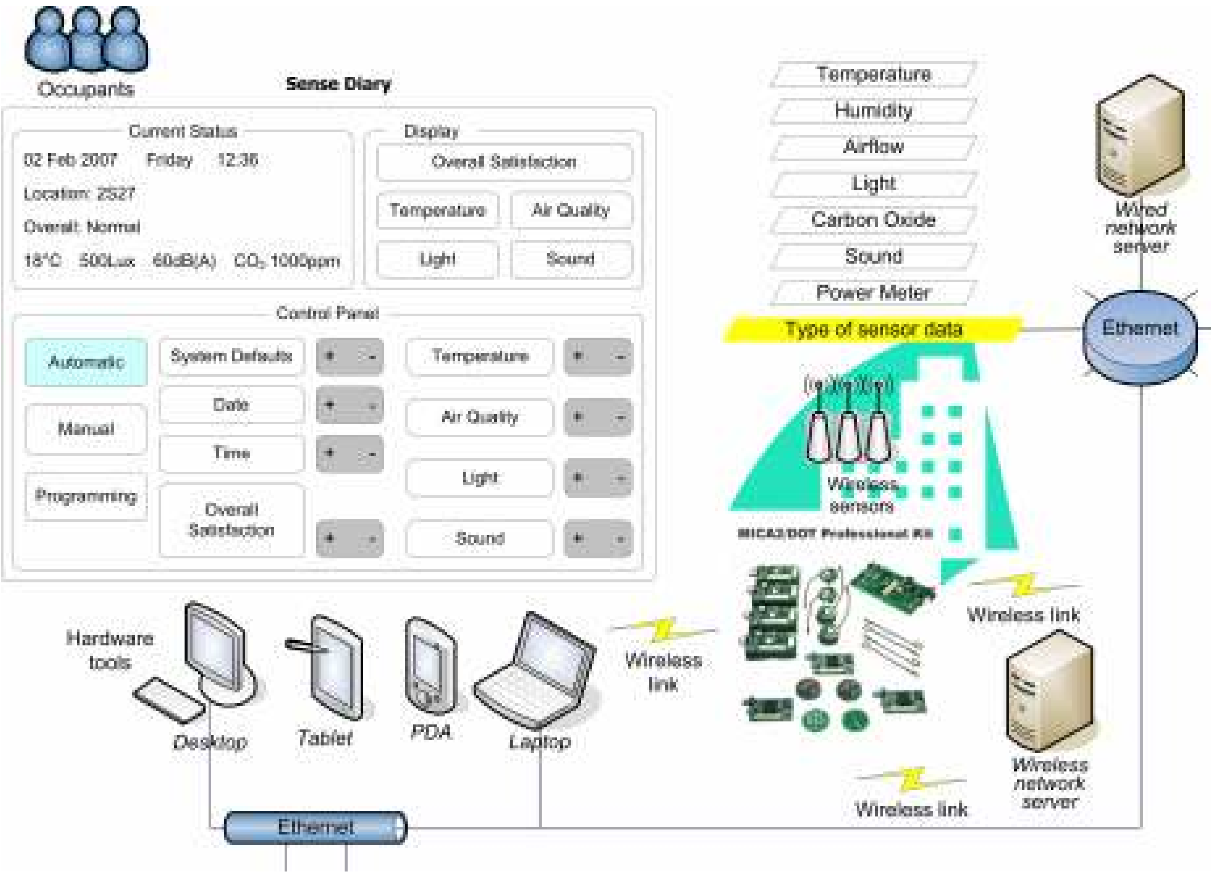

Figure 2: The Sense Diary System [1,21].

DISCUSSION

This paper introduces on-going research into Sense Diary System for intelligent real-time feedback and control to buildings, which can improve energy-saving and well-being. A system diagram about the proposed device has been described, which is based on a comprehensive review of current information from both academic and professional sources. Further research will focus on the collection and analysis of experts' opinions in terms of the design of the proposed device and its integration into diverse buildings management systems; furthermore, a demonstration system is expected to be developed at the end of this research.

ACKNOWLEDGEMENTS

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An Intelligent Decision Support System for the Design and Construction of Building Envelopes

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SUMMARY

This paper aims to introduce a prototype of an intelligent decision support system, called FacadeChoice system, for decision-making in façade solution evaluation at design stage and façade product selection at construction stage. The prototype of the proposed intelligent system comprises of several assessment tools such as a multicriteria decision-making tool using analytic network process, a façade rating tool, and an intelligent tool using artificial neural network. The integrative use of these tools can enable decision-makers to use a systemic process to deal with problems of building façade systems assessment at either design stage or construction stage. As an on-going research project, this paper does not use case studies to further demonstrate the effectiveness of the proposed system; however, it has been discussed that the adoption of the intelligent system in building façade systems assessment can facilitate the acquisition and reuse of experts' knowledge, which is essential for new buildings. This paper is an output from an on-going research project funded by the EPSRC Sustainable Urban Environment program in UK

INTRODUCTION

Building rating method has been applied in the assessment of either new or existing buildings around the world, and there have been many building assessment systems such as

- BREEAM, Building Research Establishment Ltd., UK;
- CASBEE, Sustainable Building Consortium, Japan;
- LEED, Green Building Council, USA;
- GBTool, International Initiative for a Sustainable Built Environment, Canada;
- HK-BEAM, HK-BEAM Society, Hong Kong, China; and
- IB Index, Asian Institute of Intelligent Buildings, Hong Kong, China.

However, it has also been noticed that all these building assessment systems are not specially designed to evaluate building façade systems. In other words, currently there is not a general buildings façade system assessment tool, and this leads to the research being presented in this paper.

Based on practical experience and theoretical studies of some buildings assessment systems, a new approach to the assessment of buildings façade systems is proposed using a multicriteria decision-making model using Analytic Network Process (ANP) [1] in regard to innovation and environmental sustainability in the design and renovation of building envelopes [2]. To undertake the task, an ANP model named FaçadeChoice was constructed based on a group of Key Performance Indicators (KPIs), and the model was recommended in the design of buildings façade systems for innovation and environmental sustainability. The ANP approach is difficult for practitioners to use according to a number of discussions with professionals from Arup in London. Therefore, an alternative approach integrated with an Artificial Neural Network (ANN) model was put on the schedule of research. In this regard, this paper summarizes the progress of continuing research into the assessment of buildings façade systems.

This paper aims to introduce an intelligent decision support system, called FacadeChoice system, for decision-making in either the assessment of buildings façade systems at design stage or the selection of relevant products at construction stage. The proposed system is an integrative use of several assessment tools, including

- a multicriteria decision-making tool using ANP, which is called FacadeChoice model;
- a façade rating tool, which is called Facade Progress Index; and
- an intelligent tool using artificial neural network, which is called FacadeChoice system.

As an on-going research project, this paper doesn't use case studies to further demonstrate the effectiveness of the proposed FacadeChoice system; however, it has been discussed with both academics and professionals that the adoption of this system in the assessment of buildings façade systems can facilitate the acquisition and reuse of experts' knowledge, which is essential for both new buildings and buildings refurbishment.

METHODS

FaçadeChoice model

The ANP is a general theory of relative measurement used to derive composite priority ratio scales from individual ratio scales that represent relative measurements of the influence of elements that interact with respect to control criteria developed by Saaty [1]; and there are four generic steps in the ANP, including

- model construction,
- paired comparisons between each two clusters or nodes,
- supermatrix calculation based on results from paired comparisons, and
- result analysis for the assessment.

In order to set up an ANP model, a group of indicators has to be defined first as the criteria and sub-criteria for ANP modelling. In this regard, a group of KPIs has been selected for the FaçadeChoice model [3,4] (see Table 1). There are two clusters inside the FaçadeChoice model (see Figure 1), including

- the cluster of Criteria, and
- the cluster of Alternatives.

The model is designed to support the selection of the most appropriate façade solution from two or more alternatives in the process of evaluation, or to make a comparison between a proposed building façade system and a standard/reference building façade system in a same category in the process of assessment. The cluster of Alternatives consists of two nodes in a pilot case study including Façade System A and Façade System B, which are two options of building façade systems under assessment. On the other hand, the Criteria cluster contains six subnets, i.e., the sub-cluster of KPI Group t (t=1~6), which are in accordance with the six KPI groups identified in Table 1, including

- Adaptability,
- Affordability,
- Durability,
- Energy,
- Intelligence, and
- Well-being.

Table 1. KPIs for the assessment of building façade systems

Clusters of KPIs			KPIs to evaluate building façade systems		
No.	Name of Cluster	Score* of Cluster	No.	Name of KPI	Score* of KPI
1	Adaptability		1.1	Maintenance flexibility	
			1.2	Refurbishment flexibility	
			1.3	Environmental impacts	
			1.4	Social impacts	
2	Affordability		2.1	Design cost	
			2.2	Construction cost	
			2.3	Maintenance cost	
			2.4	Refurbishment cost	
			2.5	Demolition cost	
			2.6	Recycling cost	
3	Durability		3.1	Lifespan	
			3.2	Fire protection pattern	
			3.3	Fire endurance	
			3.4	Material density	
			3.5	Structural reliability	
			3.6	Decay resistance	
			3.7	Quality	
4	Energy		4.1	Embodied energy	
			4.2	Energy performance	
			4.3	Renewable energy	
			4.4	Building orientation	
5	Intelligence		5.1	Control strategy	
			5.2	System Integration	
			5.3	Emergency response	
			5.4	Automation	
6	Well-being		6.1	Aesthetics	
			6.2	Daylight absorbability	
			6.3	Indoor daylight comfort	
			6.4	Outdoor daylight comfort	
			6.5	Indoor sound reverberation	
			6.6	Outdoor sound reverberation	
			6.7	Indoor sound absorption	
			6.8	Outdoor sound absorption	
			6.9	Indoor temperature	
			6.10	Indoor relative humidity	
			6.11	Indoor ventilation	
			6.12	Toxicity hazards	

*Note: Below please find scales for scoring with regard to sustainable (environmental, social and economic) issues relevant to People, Products and Processes:
1= Not important, 2= not to moderately important, 3= Moderately important, 4= Moderately to strongly important, 5= Strongly important, 6= Strongly to very strongly important, 7= Very strongly important, 8= Very strongly to extremely important, 9= Extremely important.

On the other hand, Table 1 can also be used to collect experts' opinions in regard to the importance of each KPI in order to set up the FacadeChoice model; and this was conducted through questionnaire survey. For example, in an experimental case study, Table 1 was used to collect information for pairwise comparisons between paired clusters and between paired KPIs.

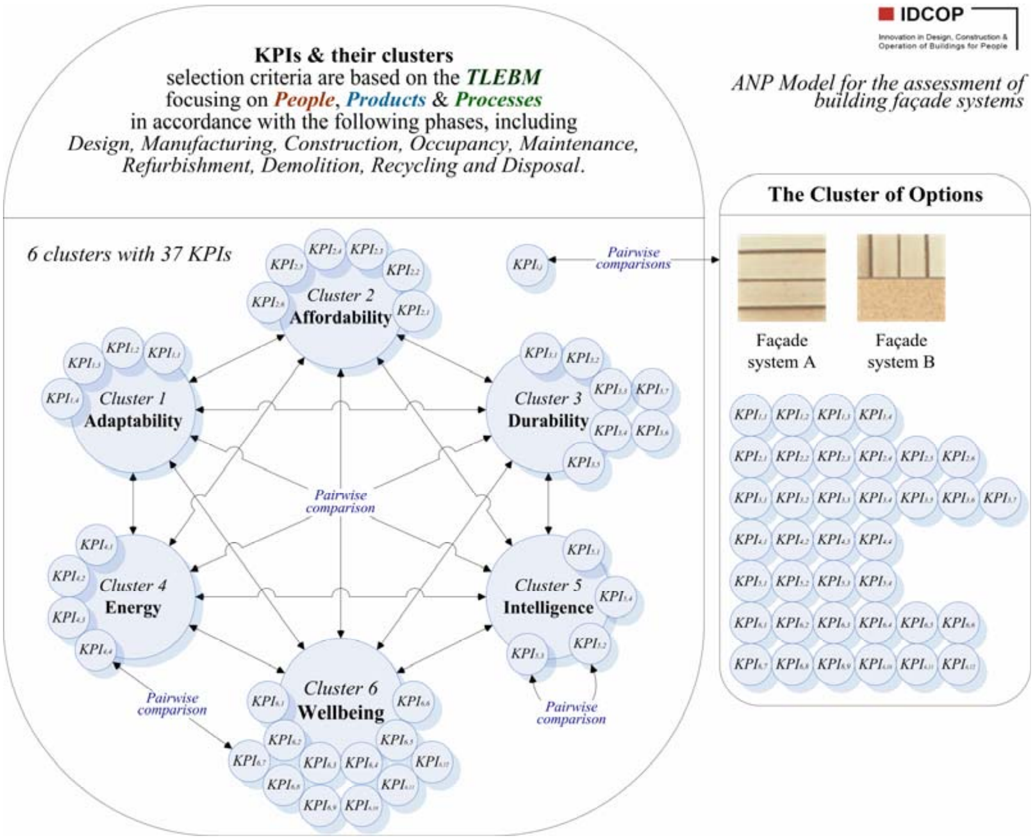

Figure 1. The FaçadeChoice model.

Façade Progress Index

Based on evaluation theories adopted by different building rating methods such as the LEED [5], it has been identified that this kind of method can only be used to evaluate one target building each time; whilst the ANP method can be used to evaluate either one target building or several target buildings [6]. To evaluate more than one building or buildings façade systems together each time, information has to be added to the ANP model in regard to details of a reference building such as a standard building/building façade system in a particular building/building façade system class as well as alternative building/building façade system plans for the same building project. However, after this pilot research into FacadeChoice ANP modelling, it has been noticed that the ANP approach is difficult for practitioners to use through a number of discussions with professionals from Arup in London; instead, it looks that rating method is much easily to be accepted. Therefore, a new rating method, called Façade Progress Index (FPI) (see Equation 1) is proposed.

$$FPI_j = \frac{\omega_{ANP\_j} \times \sum_{i=1}^{n}(w_i \times S_{Gap\_i})}{n \times S_{\max\_Focus}} \times 100\% \quad (1)$$

Where

- FPI_j is the Façade Progress Index of building façade system option j.
- ω_{ANP_j} is the ANP coefficient of building façade system option j, which is the synthesized priority weight of option j base on ANP modeling.
- w_i is the weight of KPI_i based on consensus decision-making.
- S_{Gap_i} is the Gap between current and target scores of each KPI_i.

- S_{max_Focus} is the maximum value of Gap Focus, it is 81.
- N is the number of *KPIs*.

According Equation 1, the ANP coefficient can be calculated based on the result of synthesized priority weight of options from ANP modelling. The reason of this design is that the ANP has been regarded as the most effective and efficient approach to measuring inter-relation among all KPIs, which should not be ignored in multicriteria decision-making.

FacadeChoice system

Although both ANN and ANP have been separately applied for decision-making in relevant areas for a long time, there are few research into an integrative use of ANN and ANP; in addition, it has been noticed that there has been some research initiatives in terms of the integrative use of ANN with multicriteria decision-making area focusing on how to use ANN to define the priorities of indicators for setting up decision-making models [7], and there are also few publications about either applying ANP or an integrative ANP and ANN for multicriteria decision-making purposes. In addition, it has been noticed that, among current publications, the purpose of using ANN and AHP was to use ANN models as the data supplier of AHP models. In this study, the purpose of the integration of ANP, ANN and rating approach is to train an ANN model by using results generated from ANP and the rating approach. It is clear that they are opposite processes in multicriteria decision-making between previous research and this study. As a unique decision-making model, this innovation will make it easy for practitioners to conduct assessment by using valuable expertise accumulated from ANP modeling. Therefore, the proposed FacadeChoice system provides a completely new method for the assessment of buildings façade systems.

The proposed ANN model has three layers, including an input layer, a hidden layer and an output layer. In order to achieve a synergetic effect, the group of KPIs for both FacadeChoice model and Façade Progress Index is used for the input layer; as the group of KPIs, with 37 indicators, does not involve the score of a building façade system, i.e., its value of façade Progress Index (see Equation 1), it is further adopted at the input layer as the 38th input. For the output layer, there is only one output, i.e., the value of the Façade Progress Index of a building façade system under assessment, which can be obtained from ANP modeling and Equation 1.

In order to facilitate the application of ANN, commercial software, i.e. NeuroSolutions is selected, because NeuroSolutions for Microsoft Excel is an efficient add-in that enables users to quickly develop custom neural network models without ever having to leave the Excel environment [8]; and everything users need is all contained within the NeuroSolutions sub-menu of Excel. Therefore it could be much adaptable for users who are familiar with Microsoft Office to use this software for intelligent buildings assessment.

DISCUSSION

This paper presents an integrated approach, called FaçadeChoice system, to the assessment of building façade systems for use at either design stage or construction stage for systems evaluation and selection. The FaçadeChoice system comprises of three components, including an ANP model, Façade Progress Index, and an ANN model. The FaçadeChoice decision-making model can be used to evaluate buildings façade systems. The FaçadeChoice model is developed based on the ANP containing feedback and self-loops among clusters and sub-clusters. 37 KPIs for the model are selected based on literature review and then assembled into the FaçadeChoice model. It was assumed that the FaçadeChoice has advantages over building rating methods as it tackles both values and inter-relationships among KPIs. To facilitate the use of ANP in practice, the Façade Progress Index as a new façade rating approach has been described; and it is believed that the integrative use of FacadeChoice and Façade Progress Index can provide the most reliable support in decision-making on the assessment of buildings façade systems. Finally, results from both FaçadeChoice modelling and Façade Progress Index are integrated with an ANN model in order to facilitate the application of ANP in the assessment of building façade systems.

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Effective Infrastructure Distribution, Implementing An Integrative Concept for Sustainable Office Spaces

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SUMMARY

Typical infrastructure in office buildings does not facilitate the creation of a sustainable building, effectively using energy, materials, and space. One must recognize that sustainability doesn't only refer to the use of energy, but also to spatial quality, user comfort, and ease of facility management. Different from the commonplace hierarchical systems, our goal is to distribute infrastructure such that highly networked structures arise. Our approach includes the horizontal integration of building infrastructure systems such as air, power, heating and lighting, and the vertical integration of building life cycle stages from design to implementation to management. These vertical processes are continuous, forming a "digital chain." The concept is being integrated into the renovation of the Building Services group office at the ETH Zürich. The office space has tested these concepts while utilizing this new design theory, and will continue to provide data in analysis to support the development of more sustainable office spaces.

INTRODUCTION

Among researchers in the area of building services and technology, it is common knowledge that these services and technologies are in fact the source of two-thirds of global electricity demand. This makes buildings alone responsible for a third of all anthropogenic greenhouse gas emissions [1]. Recent conglomeration of energy demand statistics has shown that when combining the portion of residential, commercial, and industrial energy demand sectors that are related to building operation and construction, buildings are directly or indirectly responsible for over half of the global energy demand [2].

Clearly, the building sector offers an immense opportunity to have an impact on future energy demand. Sustainable design, energy efficiency, ecological design, and natural systems, among many other concepts, have all been developed and used in the creation of buildings to try and address their global impact. Still, although improvements have been made, they have not been integrated into all parts of building construction and usage.

The area of building services is the direct source of energy demand and is not well integrated into the overall design perspective, especially in the important early stages of design. Interdependencies between architecture and building services are not captured. Being aware of these interdependencies is a major step toward developing more sustainable buildings.. This is done through better vertical integration along the design chain and better horizontal integration of building service components. Not only does this directly reduce the energy demand of buildings, but it also eliminates constraints imposed by separately designed infrastructure. The view of the interdependencies provides a larger range of options to improve the indoor space. Europeans spend upwards of 90% of their lives in buildings [3].

Unsatisfying spatial quality leads to less productive users. Removing constraints on spatial quality indirectly improves the sustainability of a structure.

With a holistic and inclusive planning and creation of buildings the palette of available technologies also increases. Instead of a small number of options available for one sector of design, a wide variety of systems can be considered when the entire project can be visualized through a digital design model.

Digital planning technologies gained major importance in the fields of design, construction and facility management over the last decade. However, these fields often remain separated, and information is not communicated between them. In spite of the major economic and environmental impact of building services, the usage of digital technologies in this field is not as thoroughly developed as in other fields of the building industry.

The goal of the project described herein is to define, study, and implement a framework by which seamless design processes and advanced technology can be researched, tested, and implemented in one laboratory, which will guide the creation of more sustainable buildings. In order to achieve this, a methodology was laid out to define this process based on current digital design practices and available technologies. These were applied to the ongoing renovation of the Building Services Laboratory (GT Lab). This was done in part during the real-time design process of the renovation, as well as into the renovation itself to allow future testing and analysis of design techniques and technologies. It is a functioning office space allowing for analysis of the design space from its design phase up to its usage and management. Technologies have been implemented that can be utilized in modeling processes to verify design decisions. The lab provides a system to test various permutations of how a digital chain can be linked together from design phase to operation, thus maximizing the flexibility of all phases and pieces of a building. The methods used in this design concept were applied in a collaborative network of engineers and architects.

METHODS

In the sense of horizontal integration our group picked appropriate, innovative technologies and combined these into a well performing ensemble. Every technology, before being integrated into the overall system, has been carefully chosen based on an educated selection method. The product development and implementation is done in cooperation between the chair and industry. During the selection procedure the aim was always to select technologies that contribute to a comfortable indoor environment. Again, sustainability was considered not just in a narrow, energy related sense, but in a universal sense relating to spatial quality, ease of use, etc.

Heating Ventilation and Air Conditioning (HVAC)

For the method of selection of HVAC components major quality criteria were the exergy efficiency, architectural and spatial aspects, flexibility, and the effectiveness of the ventilation system. Thermodynamic analysis based on the exergy principle has been applied to buildings in order to identify potential improvements of heating systems. The exergy concept classifies energy sources according to their “quality” or potential to do work. It is therefore appropriate to use energy sources of low “quality” to fulfill a low “quality” task such as maintaining a building at a low temperature [4]. Heating systems based on heat pumps are more exergy efficient than systems based on fuel driven burners. By choosing low supply temperatures the operation of heat pumps becomes even more efficient. For that reason the exergy design principles led to the selection of heating and ventilation technologies that are designed to operate with low supply temperatures. In order to minimize the exergy demand of the complete HVAC system, the static heating and the air supply/ventilation must be decoupled from each other. This decoupling exploits the potential energy savings of low ventilation rates without affecting the indoor temperature.

Besides energy efficiency other criterium, such as indoor air quality (IAQ), were not overlooked. Therefore, the use of low ventilation rates required ventilation systems to have high ventilation effectiveness in order to maintain good IAQ. It was decided that the best way to maximize ventilation effectiveness was through a system that comprises of a large number of distributed sensor-controlled exhaust openings, which is based on the displacement principle. Past models and measurements were observed showing that this system achieves high ventilation effectiveness compared to standard systems. The local control characteristics of this kind of system allow the airflow rate to be dynamically adjusted according to the number of occupants being present in a room or a building [5].

The supply air system was matched to this distributed exhaust. The selection of a decentralized air supply was influenced by the potential for enormous space savings and for the flexibility this approach offers compared to hierarchical, centralized systems [6]. Very compact, decentralized supply air units were chosen and optimized in collaboration with industry for integration into the complete ventilation system.

Lighting

Lighting has both a large aesthetic effect as well as a significant impact on energy demand. In this selection method, the spatial and managerial aspects of building sustainability were directly balanced with aspects relating to energy demand. An important criterion was the integration of all lighting elements in the multifunctional ceiling panels which allowed a built-in height of 10 cm only. Also, the artificial lighting was supposed to efficiently complement existing daylighting. In order to define the necessary lighting to meet the required constant amount of 500 lux, model-based lighting simulations were run, comparing different lighting variations. It was found that LED lighting in combination with fluorescent lighting could provide highly controllable lighting with very high efficacy. In order to control the lighting color within the lab and to experiment with different ambient color settings, these LEDs had to be capable to display the full RGB color spectrum.

Power Supply and Distribution

Various supply topologies have been assessed to find the most suitable solution offering the largest flexibility and ease of use. A suitable topology for power distribution was found to be a ring along the boundaries of the space with branching lines that serve the inner part of the ring. This supply structure allows maximum flexibility for the layout of desks. The design also presented an option for new construction. Electrical routing has the potential to be done through ventilation ducting directly integrated into the neutral zone of structural floor elements between steel reinforcements [7]. This integration eliminates the need for either a raised floor or a ceiling plenum. This is not possible for a renovation where an underfloor system is required, but analyzing this system for future implementation was an important consideration in its selection.

Parametric design model

One goal is to expand the digital chain into the fields of building services, starting at the early design stages of a building. The huge impact of technical systems of buildings makes it necessary to consider consequences of building service design already in these early design stages. Performance evaluation has to be integrated into the architectural design process already at an early stage, up to 80% of the decisions most important for the building performance are already made within the first 20% of the design process [8]. Common CAD-Software however does not support an integrated planning approach.

For the design and ongoing research at the GT Lab an approach was chosen based on a parametric building information model. The use of a parametric model provided the capability to design, calculate, and simulate already at very early stages of design and still be able to judge spatial quality. A three-dimensional parametric geometry model was set up. Supply requirements were directly defined in the model database. Every change in geometry or equipment resulted in an instant adaptation in performance data for the selected room cell, floor, or the whole building. The evaluated technical systems are integrated into the model in a schematic. Their dependencies are defined in the database as well as in the resulting geometric model and displayed three dimensionally. Simulations like daylight and artificial light simulations can be directly accessed, comparing light distribution and visual impression with the calculated use of energy of different lighting systems.

System Management Model

It was found that through the use of digital control every system could be addressed and controlled throughout the power network. This included the ability to control everything from ventilation to standby power delivered to a coffee machine. Sensors could be used directly in the same network integrated with the lighting and ventilation systems, which they help steer. Defining a digital power management system realized avenues to reduce control infrastructure and facilitate the optimization of the entire system. The selection method also required a highly flexible system that could be easily modified for research applications, which is provided in the digital control system. This allows for a broader analysis of the digital chain from planning to operation.

The planning model has been furthermore adapted for steering and monitoring purposes within the GT Lab. The distributed systems such as those used in the GT Lab impose a paradigm shift towards the extensive use of management software. Traditional, hierarchical systems of building services like central ventilation for example, rely on centralized control and steering. Distributed components were chosen and implemented at the GT Lab, which lack this central steering component. Instead they are highly networked and act as complex systems, reacting on local and overall conditions. This reaction is based on a large amount of information obtained from sensors and systems including user input to fulfill individual comfort needs. This information has to be synchronized, resulting in control commands created by software-based rules. The actual physical components like the small decentralized ventilation units are inexpensive and energy-efficient, but are not capable of steering the whole topology of units. Therefore, the resulting network of distributed units has to be coordinated by software.

By incorporating sensors monitoring indoor data as well as environmental conditions this network could be steered and controlled to react more sensitively and efficiently. The software would need a digital model of the built environment including geometrical and topological information of the building as well as the applied building service technologies.

RESULTS

Renovation Overview

Within the Building of the Department of Architecture (HIL) one hundred square meters has been selected by the Chair of Building Services to be transformed into a new office space and laboratory. Before the renovation, the existing infrastructure (lighting, sound absorber, air supply/exhaust) was included in a 60 cm high plenum. This means that in this particular case about 1/7 of the actual floor height was occupied by the infrastructure. During the retrofit of the office space the suspended ceiling was completely removed and a raised floor with a height of 17 cm was installed instead. The existing supply air system was substituted with compact decentralized supply air units and a ducting system incorporated in a raised floor. Electrical and IT cables were integrated into the underfloor ducting. The lighting that was previously installed in the suspended ceiling has now been integrated into multifunctional ceiling panels. These panels have a thickness of only ten centimeters and cover about 50% of the ceiling area. They combine the features of static heating with activation of thermal mass, lighting, sound absorption, and also integrated sensor-controlled exhaust openings. The control of all electrical devices is managed with a unique control system. This management system cannot only be used to control the building services but offers a simple and powerful interface for facilities management. A cutting edge microchip attached to the power supply of each device allows a unique way of digital power management. The detailed components of the renovation are described below and how each component is integrated is illustrated in Figure 1.

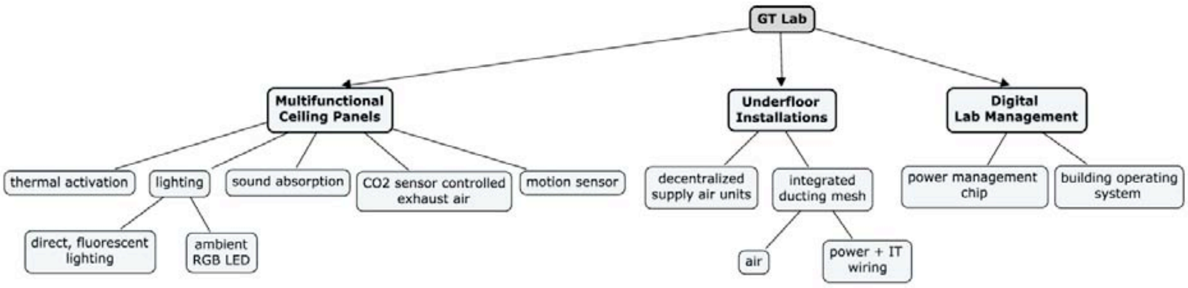

Figure 1: The components integrated into the renovation of the GT Lab.

Underfloor Installations

The fresh air is supplied to the office at a single point below the ceiling. From there two branches of the duct supply air to a U-shaped channel, integrated in the raised floor. This channel is separated into two parts and the airflow of the two supply branches can be regulated, such that different pressure conditions in the two chambers of the channel can be obtained. Since the façade could not be modified and the fresh air comes from the centralized system, the channel was designed to simulate the different pressure conditions that would arise on a buildings façade due to the influence of wind. The decentralized supply air units being installed in the raised floor are connected to the U-shaped channel on their suction side and to another ducting network on their pressure side. The ducting network connected to the supply interconnects the different sides of the channel representing the facades, and allows a regular distribution of fresh air over the entire floor space. For this network a meshed topology is chosen because of its balancing pressure characteristics. The system simulates how the ventilation network can use the wind pressure differences between the facades for air transport and hence to reduce the energy costs.

The supply air ducting network also carries cables (power, data) such that central workstations can be easily connected over a branching line to the main installation built as a ring line along the building façade. Tubing was used with an oval cross section, and newly developed compact decentralized air handling devices make the integration into the floor possible.

Multifunctional Ceiling Panels

The ventilation system with low momentum air supply through the floor and extraction at the ceiling follows the well-known displacement principle [9]. The system designed for the GT Lab utilizes this principle in combination with localized exhaust vents that use sensors to monitor local air quality. This principle of local waste air extraction is often encountered in industries and has been analyzed for buildings [7]. Sensor-controlled exhaust openings with a built-in flap, as proposed by [7] have been integrated into the panels distributed along the ceiling. Air with a higher contaminant concentration can be extracted and the over-all ventilation rate is reduced because of the demand-controlled nature of this system. This system with its autonomous local control characteristics is much simpler and more robust than a conventional system with a master-slave control.

The ceiling panels can be used to cover static temperature control. Hot or cool water in a two-wire system activates the thermal mass of the construction and convectively heats or cools the room air with a power of about 40 W/m^2 within the comfort zone. In modern, well-insulated office spaces it is enough to cover 50% of the ceiling area to meet the heating power requirements. The ceiling panel is covered with a thin, perforated sheet-metal plate that has good sound absorption characteristics. The same area as needed for heating is sufficient to reduce the sound reverberation time to a value below 0.6-0.7 seconds.

The standardized ceiling panels that incorporate the features of heating and sound absorption have been altered and the lighting equipment has been included. Two different kinds of illumination with two different technologies were implemented. Energy efficient fluorescent tubes are used to meet the higher lighting requirement while for the ambient lighting LED technology has been used. The LEDs are mounted 10 cm apart along a 1 m strip, eight of which are integrated into each ceiling panel. The power of a one of the LEDs used is 1.4 W and its luminous efficacy is approximately 40 Lumen per Watt. Compared to a common light bulb with an efficacy of about 18 Lumen per Watt the LED is a clearly superior, and also a rapidly developing technology. The LED lighting has been designed, and each strip can be controlled to compensate for light gradients occurring in the room due to daylight influences. On the other hand it serves as an ambient light providing a convenient atmosphere.

Digital Lab Management

A newly developed chip has been integrated into the lab’s infrastructure. The chip is attached to the power line just before every electrical device connected. The power line is used to carry the control commands to the chip. This transmission method makes it unnecessary to install extra cables to communicate with terminal devices. Because of the unique addressability of each terminal device this technology offers the possibility to implement very effective power management that reduces standby losses almost to zero.

For the management of the ventilation, exhaust air flaps, heat supply, power control chips, lighting, as well as sensor data a "building operating system" is implemented in cooperation with a software company. Therefore the digital chain is extended to actually control physical components seamlessly. All the systems exist in the virtual model as well as in physical reality, and simulation can be directly compared to actual physical operation data. A small server will allow access to all systems via the Internet. A web-based interface enables the user to control and monitor the GT Lab physical environment. Individual lighting schemes can be stored, and global settings for the entire environment are also available. Sensor data is recorded, which includes energy consumption for each device, local CO_2 concentrations, and activity recorded from the motion detector. With the building operating system it will be possible to group systems such as ventilation units individually, simulating the flexible management of space, reacting on changing user needs. No physical replacement of systems is necessary when spatial user requirements change, the system units can just be regrouped within the software model.

DISCUSSION

It has been found that it is possible to create a research space that employs the flexibility to study both horizontal and vertical aspects of design. Analysis can be done on the performance of new technologies for buildings systems while also understanding how these components can be integrated in the design process. By controlling and studying the preliminary design and modeling procedures that can then be implemented into a flexible, yet normally occupied office space, improvements in both physical operation of components can be made along with improvements in the ease of implementation, integration, and management. These last improvements have benefits easily overlooked in a strictly experimental laboratory. The laboratory office is both a normal working environment as well as a laboratory and has realized some unique concepts for a building research environment. In the physical renovation of this space some preliminary analysis of this design process has also taken place.

While many technical results from the laboratory are forthcoming in future research, the current research into the laboratory design provides a valuable description of how more sustainable indoor environments can be created and studied. It does not analyze one area of improvement, but instead allows for an integrated view in the research itself, as all systems are available for study through one digital interface. The use of this digital control system in particular, will contribute to the ability to study the entire system. It also provides a direct link back to the design phase and thus can be used to test preliminary design ideas and models. This is one way in which a digital chain can be maintained for analysis in the lab. As more integrated practices are being pursued for the design , construction, and operation of buildings, this concept presents a method for how these concepts of integration can be incorporated and studied in academic research.

Continued work on the design process is also planned in an expanded renovation of more space within the HIL building where the 100 m^2 lab is located. There is potential for expanded renovation of the adjoining office space, which currently houses the Chair of Computer Aided Architectural Design (CAAD) with whom collaboration studying the digital chain already exists. The entire space would then be designed to house all the chairs under the Institute for Building Technology (HBT). It has also been proposed to work toward a renovation of the entire building using concepts that would be refined in the GT Lab.

The data gathered in the GT Lab will refine the design model to be used in further designs. The direct link between the virtual and physical building model will help to validate virtual simulation results as well as optimize steering operations for the implemented services. Future research will also look at the optimization of the underfloor air distribution and power supply integration. Integrating this ducting into the infrastructure of a building will be studied. It is expected that economics savings will come from space savings, and energy savings will come from more exposed thermal mass.

One of the clearest observations made from the resulting creation of the GT Lab is the tremendous benefit of improving the spatial quality. As the office is situated directly adjacent a space which maintains the former construction, a direct comparison is possible. The higher ceiling and highly controllable lighting has substantially improved the indoor quality.

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The Method for Optimum Vertical Layout of High-rise Apartment Buildings to Improve Indoor Comfort

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SUMMARY

The objective of this paper is to develop a method for achieving optimum vertical layout of apartment buildings that can provide proper design alternatives and useful information by its application in order to improve indoor comfort and to verify the efficiency and the validity of the objective method. The result of this research can be used as supporting information to reach an amicable settlement against civil petitions and disputes. This will consequently reduce waste of the time and money, enhance the efficiency of work, and improve indoor comfort.

INTRODUCTION

In Korea, as the social phenomenon of the drift of the population to major cities began to appear from 1970's, a number of high-rise apartment buildings have been constructed in major cities such as Seoul. Following the recent improvements in living standards, there has been growing interest in indoor comfort. Especially, one of the most important issues related to indoor comfort is solar access. These social tendencies have brought about residential environment problems such as violation of solar rights. The number of civil petitions and disputes over solar rights has increased recently. These problems have negatively affected human health, work productivity, thermal comfort, and visual comfort. So, the solar rights analysis system (*HELIOS*) had been developed by research team to evaluate the solar rights of residential buildings, quantitatively (Seong et al., 2006). Furthermore, this solar rights analysis system had been applied to many practical affairs up to recently.

But, as the problems related civil petitions and disputes over solar rights are becoming more complicated and diverse, it is demanded that proper design alternatives should be suggested and active design methods of vertical layout of the apartment buildings should be studied in order to reduce the solar rights violation for neighboring area within the acceptable limits. However, the present solar rights analysis system has some limitations. That is, this system analyzes only the original design condition, and provides only the information on whether violations of solar rights have occurred or not. The information of optimum vertical layout in high-rise apartment buildings to assure the solar rights is deduced from repetitive manual process. To establish alternatives and vertical layout for solar rights assessment, for example, available alternative examined first of all, that experts must participated in. Then, solar rights analysis result about survey buildings is thoroughly examined by an expert. If the results are not acceptable, geometric information is changed and reflected in solar rights analysis system.

If the result of analysis is not satisfactory, the above process is repeated again and again until acceptable result is deduced. These problems bring about economic loss of productivity and efficiency.

For these reasons, this study is intended to develop the method for the optimum vertical layout of the apartment buildings that can provide proper design alternatives automatically and the useful information to improve solar access. The result of this research can then be used to furnish advanced information for an amicable settlement against the civil petitions and disputes, to reduce waste of the time and money, to enhance the efficiency of work, and to improve indoor comfort.

SOLAR RIGHTS AND ASSESSMENT METHODS

Solar rights

The definition of solar rights varies for different localities and situations. In Europe and U.S.A, 'solar right' is generally defined as the occupant's right to unobstructed light from the sun, i.e., the rights to light (Building Performance Research Unit, 1972). In New Mexico, U.S.A, it is considered as a form of property right for renewable energy source, and thus an owner of a real property where a solar collector is installed may claim its solar rights. In Korea, solar rights are generally associated to human health, work productivity, thermal comfort, and visual comfort in apartment buildings.

However, the Korean laws do not clearly stipulate the regulations on solar rights, and so, solar rights are interpreted in various ways based on the theories related to an occupant's solar access. First, solar right is defined as the right to remove an occupant's disadvantages from interference with direct solar access by obstructions (Choi et al., 2000). Since Korea is located in the northern hemisphere, the residents living in the northern site of the building are generally obstructed from solar access if another building is built in the southern site. Based on this definition, the residents living on the northern site of the building can file a formal complaint to the property owner of the southern site as provided in the Korean building code 53 provision. Solar right is also defined as the right to ensure that neighbors have access to natural lighting, ventilation, and field of vision by limiting the height of adjacent buildings. Also, it is defined as the right to ensure sunshine duration for usual residential living in buildings, and also, the right to enjoy high standard of living in terms of health associated with the direct solar radiation. This last definition, in turn, means that a property owner should not infringe upon another property owner's right to direct solar light access when building a new building next to an existing apartment complex.

In Korea, sunshine duration should be maintained consecutively for more than 2 hours in every housing unit during the winter solstice from 9 a.m. to 15 p.m. (during 6 hours). In addition, the distance between each block should be kept to more than 0.8 times of a building's height to the south from the adjacent apartment building. In a densely-populated city with high-rise apartment buildings such as Seoul, it is particularly important to dispute settlements over solar rights for every housing unit in an apartment building. Therefore, an analysis method for assessing the solar rights and a simulation program to quantitatively evaluate the solar rights of each housing unit in an apartment building is necessary.

Assessment methods for solar rights

The analysis methods for solar rights assessment are generally classified as sun-path diagram method (i.e. solar position diagram) (Goswami et al., 1999; Duffie and Beckman, 1991) and shadow diagram method. The sun-path diagram methods can be subdivided into the horizontal sun-path diagram method and vertical sun-path diagram method (i.e. *WALDRAM* diagram method) (Lechner, 2001). The shadow diagram methods can be subdivided into the plan shadow diagram method, the elevation shadow diagram method, the section shadow diagram method, and the perspective (3D) shadow diagram method depending on the point of view. However, only two of the methods can be used to quantitatively compute the sunshine duration for all housing units in an apartment building: the *WALDRAM* diagram method and the 3D SHADOW diagram method. Based on the comparison results (Seong et al., 2006), the *WALDRAM* diagram method was more suitable than the 3D shadow diagram method. In addition, the *WALDRAM* diagram method provided more accurate results, especially in computing the sunshine duration on a specified day. In terms of analysis accuracy, however, the *WALDRAM* diagram method could analyze more dense time interval than the 3D shadow diagram method, and in terms analysis processing time, the *WALDRAM* diagram method required less time than the 3D shadow diagram method, because the rendering process took a long time to make a 3D shadow diagram. Lastly, in terms of the analysis processing steps, the *WALDRAM* diagram method was two steps faster. As such, the *WALDRAM* diagram method was clearly more suitable than the 3D shadow diagram method for assessing the solar rights in apartment buildings.

Method for the optimum vertical layout

In this chapter, we have chosen a real case for vertical layout to develop a method for the optimum vertical layout. According to this case study, we propose a method for the optimum vertical layout in high-rise apartment buildings. An objective for the optimum vertical layout is that all housing units in survey building must be satisfied within acceptable limits for solar rights. In the case study, the survey objects were all housing units in C apartment building 202(i.e. Bldg.202) and obstruction buildings were T apartment buildings 103 & 105(i.e. Bldg.103 & Bldg.105). By figure 1, you can find out the geographical situation.

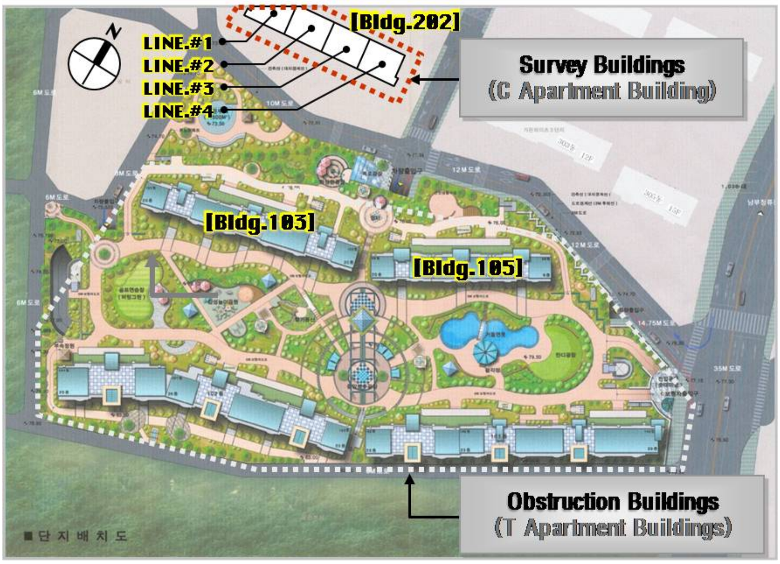

Figure 1. Geographical information of survey building and obstruction buildings

Method of Vertical Layout by Manual Process for Solar Rights

In this chapter, we briefly described the method of vertical layout by manual process for survey objects (all housing units in C apartment building) to assure the solar rights, and discussed some predictable problems in manual process.

(1) Vertical Layout by Manual Process

To find out the process of vertical layout manually, we described the process that our research team executed in the past.

A) In order to deduce alternatives that the solar rights at all housing units in survey building(C apartment building 202, i.e. Bldg.202) can be satisfied within acceptable limits, it is necessary that 1st floor housing units in Bldg.202, which have the worst condition, are investigated first. So, we selected housing units #101, #102, #103, #105(i.e. survey unit #101, #102, #103, #105) in survey Bldg.202, as shown as in figure 1, from the selected survey housing units, and derived the possible alternatives to assure the solar rights at survey unit #101, #102, #103, #105 by repetitively comparing and investigating the *WALDRAM* diagram, which is the result of the many predictable design alternatives.

B) In vertical layout for solar rights, obstruction buildings, which are the vertical layout object, consist of two Bldg.103 and bldg.105. The vertical layout of obstruction buildings should be classified in 3 parts, 103 only, 105only, and 103,105 together.

C) [Design Step.1]: First to know the influence of obstruction Bldg.105 to survey units, we performed the solar rights at survey units against various vertical layout alternatives. The number of cases expected is 24,696,000(all cases which can be generated by product with story numbers of housing lines including story number 0, that is, all cases = 21*21*20*20*14*10). We examined the *WALDRAM* diagram, continuous sunshine duration, and accumulative sunshine duration. We found that obstruction Bldg.105 did not show any improvements in solar rights for all survey units. From this, we can conclude that vertical layout of obstruction Bldg.105 did not have any influence on solar rights improvement and can exclude obstruction Bldg.105 from the vertical layout.

D) [Design Step.2] : By reviewing results of [Design Step.1], and *WALDRAM* diagrams, it is expected that proper alternatives for vertical layout which every survey units could assure the solar rights within the acceptable limits would be deduced through reducing stories of the line number 1(i.e. LINE.#1) and the line number 2(i.e. LINE.#2) in Bldg.103. So we analyzed various expected alternatives in [Design Step.2] by reducing stories of LINE.#1) and LINE.#2. But it turned out that there were no suitable alternatives. However, by reviewing the *WALDRAM* diagram (refer to figure 2) of alternatives, we found that LINE.#1 and LINE.#2 did not interrupt the solar rights in survey unit #105 and the maximum allowable stories for every survey units to assure the solar rights in LINE.#1 and LINE.#2 were 12.

E) [Design Step.3] : By the result [Design Step.1], in order to confirm an additional continuous sunshine duration, stories of LINE.#1, and LINE.#2 should remain as a 12 and we derived several alternatives("Alt.B-105", "Alt.B-106", "Alt.B-107" and "Alt.B-108") by changing the story of LINE.#3 and performed solar rights assessment for all survey units. By comparing the results, we derived the plan that minimizes the loss of story comparing with the original design condition. Results of "Alt.B-105", "Alt.B-106", "Alt.B-107" and "Alt.B-108" are shown as Table 1. "Alt.B-105" and "Alt.B-108" were suitable for every survey units to be satisfied the solar right. The loss of stories in alternative "Alt.B-105" was 24 and the loss of stories in alternative "Alt.B-108" was 22 compared to the original design condition. Finally, among the alternatives ("Alt.B-105" and "Alt.B-108"), we suggested "Alt.B-108" as the best alternative for vertical layout of obstruction buildings (T apartment Buildings). The loss of story "Alt.B-108" was 22 compared to the original design condition.

As above mentioned, by manual process, it needs professional knowledge, complex and continuous analyzing process and lots of time to deduce the alterative through vertical layout.

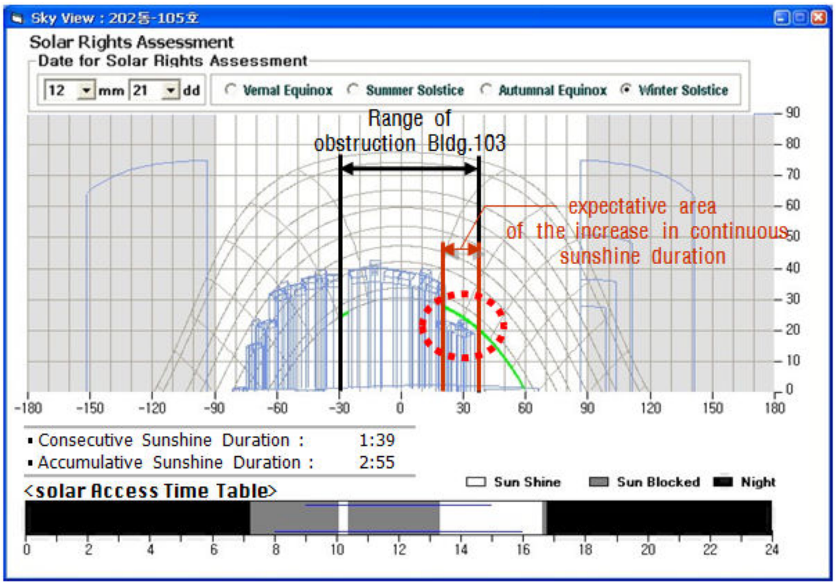

Figure 2. Analysis of *WALDRAM* diagram in [Design Step.2]

Table 1. Results of Vertical Layout by manual process

Designs	Building	Stories						Suitability	Loss*	Remarks
		LINE.1	LINE.2	LINE.3	LINE.4	LINE.5	LINE.6			
Original	Bldg.103	20	20	20	20	20	20	-	-	-
	Bldg.105	20	20	19	19	13	9			
Alt.B-105	Bldg.103	12	12	12	20	20	20	○	-24	-
	Bldg.105	20	20	19	19	13	9			
Alt.B-106	Bldg.103	12	12	16	20	20	20	×	-20	-
	Bldg.105	20	20	19	19	13	9			
Alt.B-107	Bldg.103	12	12	15	20	20	20	×	-21	-
	Bldg.105	20	20	19	19	13	9			
Alt.B-108	Bldg.103	12	12	14	20	20	20	○	-22	Best Alternative
	Bldg.105	20	20	19	19	13	9			

*: Loss of stories compared with the original design condition

(2) Problems of Vertical Layout by Manual Process

We have performed a case study through the methods for the vertical layout by manual process to assure the solar rights in survey units. As above mentioned, a manual process for vertical layout is summarized as follows:

A) Review alternatives step by step;
B) Change stories of LINEs or BUILDINGs in obstruction buildings and reflects the modified geometric information in the solar rights analysis system;
C) Analyze on the basis of reflected data, and estimate the result; and
D) To get the best result, feedback process in analyzing and reviewing should be done continuously.

By a manual process in vertical layout, it not only lowers productivity, efficiency in researching but also requires lots of time and experts. Furthermore, it also includes possibilities of mistakes such as an error of judgment. So, in order to minimize these problems, our research team suggests a methodology in automated vertical layout of high-rise apartment buildings by using computer. This helps us improve the solar rights and the waste of material and business efficiency.

Method for Vertical Layout by General Automation for Solar Rights

(1) Geometric Characteristics of the High-rise Apartment Buildings Design

In order to design proper stories of apartment building, geometric composition elements should be extracted and the hierarchy of the elements needs to be clarified systematically. The elements that compose an apartment building, the hierarchy of the elements and the organic relations are defined as follows:

A) An apartment building (i.e. **[BUILDING]**) is composed of several buildings. A **[BUILDING]** consists of housing unit(i.e. **[UNIT]**), common space(i.e. **[CORE]**) and housing unit line(i.e. **[LINE]**) (See Figure 3)

B) A **[BUILDING]** is composed of multiple **[LINE]**s, and each **[LINE]** comprises the repetitious **[UNIT]**s.

C) The height of an apartment building is determined by the vertical variation of the **[UNIT]** that composes each **[LINE]**. And the **[UNIT]**s, which are distributed uniformly along the vertical line, can be considered as elements of corresponding **[LINE]**.

D) In an arbitrary **[LINE]**, the building geometry cannot be generated if a lower **[UNIT]** does not exist. Therefore, in case of n stories, the elements of universal set "L" are 0, 1, … , n-1, n stories. And the number of possible subsets is n+1 (u_0, u_1, … , u_{n-1}, u_n).

E) A **[UNIT]** is the main space that composes the building and a **[CORE]** is the auxiliary space that serves the main space.

F) The height of a **[CORE]** is determined by the vertical variation of the relevant **[UNIT]**s.

G) In a word, the vertical space of the apartment building is determined mainly by the vertical variation of the **[UNIT]**, which decides the vertical height of the **[CORE]**.

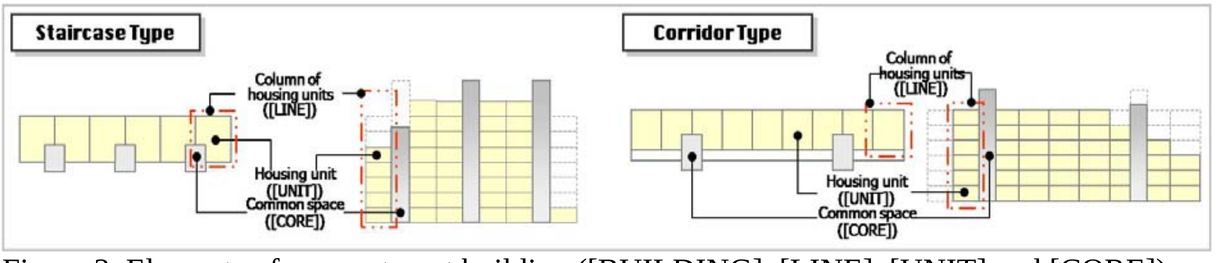

Figure 3. Elements of an apartment building ([BUILDING], [LINE], [UNIT] and [CORE])

(2) Definition of High-rise Apartment Buildings Components

To set plan of method for the vertical layout, the **[CORE]** should be excluded from the building set because it is determined by the **[UNIT]**. It is necessary that the **[BUILDING]** set should be classified by each line that is working equal vertical movement. To sum up, the set of **[BUILDING]**, ***B***, which is composed of several **[LINE]** components, can be defined as in equation (1). Each **[LINE]** set, ***L***, can be defined like equation (2), is located on the same horizontal plane, and composed by several **[UNIT]** elements which are on the same vertical line.

B = {x|x ∈ all **[LINE]**s of housing units in the **[BUILDING]**} = $\{L_1, L_2, L_3, \cdots, L_n\}$ (1)

L = {x|x ∈ all **[UNIT]**s in the **[LINE]**} = {x|x ∈ all stories which can be generated} (2)

$= \{u_0, u_1, u_2, \cdots, u_n\} = \{0_{Floor}, 1_{Floor}, 2_{Floor}, \cdots, n_{Floor}\}$

Where,

B: set of apartment building (**[BUILDING]**)

L: set of housing unit line (**[LINE]**) in the apartment building

u: element of a housing unit in the housing unit line (**[LINE]**)

(3) Algorithm for Vertical Layout by General Automation

Based on elements and sets which are the composition of the apartment defined on the Chap 2, $\boldsymbol{R_B}$ (equation (3))which is the algorithm of possible alternatives of all layer-adjustment about an arbitrary building from the apartment can be defined the subsets of Cartesian product from the given sets like equation (3). When there are many buildings, $\boldsymbol{R_T}$ (equation (4)) that is the set of possible alternatives of all layer-adjustment is able to be defined as equation (3). Figure 4 illustrates this logic in the form of a graph.

$$R_B = L_1 \times L_2 \times \cdots \times L_{n-1} \times L_n \quad (3)$$

$$R_T = R_{B1} \times R_{B2} \times \cdots \times R_{Bn-1} \times R_{Bn} \quad (4)$$

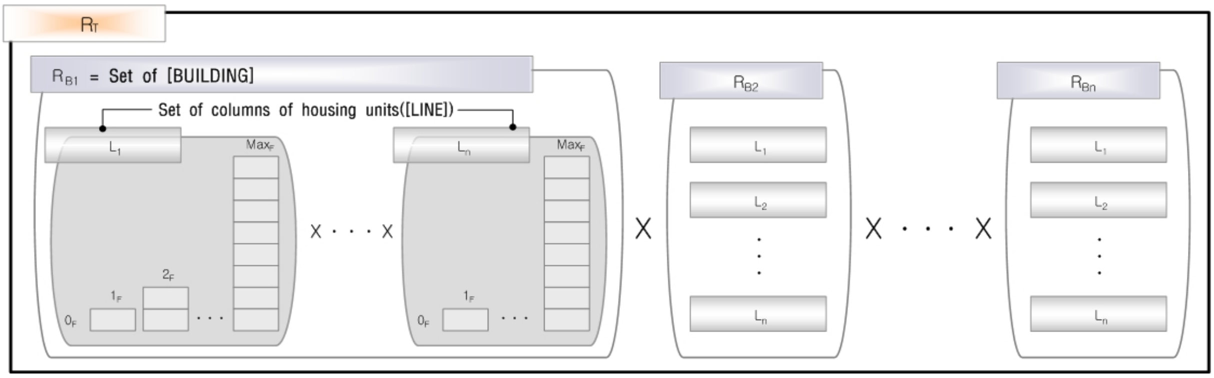

Figure 4. *L* set([LINE[set], $\boldsymbol{R_B}$ set([BUILDING] set) and $\boldsymbol{R_T}$ set

(4) Problems in Method for Vertical Layout by General Automation

Based on equation (4) which is the method of vertical layout by general automation above-mentioned, a total of 85,766,121 alternatives for vertical layout of obstruction Bldg.103 are derived by Cartesian-product of several $\boldsymbol{R_B}$. Considering the processing time of solar analysis system which is available to analyze average 10 cases per a second, it will take a total of 2.4E+03 hours to analyze all vertical layouts. The computers can be inefficient and irrational by simple computerization, and it is necessary to optimize this algorithm setting such as intuition experts can have.

Method for the Optimum Vertical Layout

In this chapter, the method of vertical layout by automation is optimized in a way to set intuition of expert knowledge about the standard sunshine duration and sun-path in the winter solstice.

A) Based on the present standard range, it is necessary to determine whether vertical layout is needed or not about each **[LINE]** to estimate sunshine duration (from 8 to 16). That is, as shown in figure 5, from 8 to 16 o'clock it should be divided into section A(range that vertical layout is necessary) which affects the calculation of sunshine duration and section B and C(range that vertical layout is unnecessary) which does not affect the calculation of sunshine duration. Also, a certain **[LINE]** which is needed vertical layout is necessary to derive information about by what floor has to be proceeded, and a certain **[LINE]** will be how many stories should be designed. Therefore, we defined a factor of "minimum allowable story", that is the minimum story number that is unnecessary vertical layout.

B) In case of section A that may affect the estimation of sunshine duration, it is separated into section A-1(vertical layout is necessary) and section A-2(vertical layout is not necessary) in a boundary of sun-path in winter solstice at *WALDRAM* diagram. Proceeding the vertical layout from the story of original design condition to 0 story in each **[LINE]**s, when all geometrical figure of the **[LINE]** is located blower than sun-path in winter solstice, that is, the story of **[LINE]** which is all geometrical figure of **[LINE]** included in section A-2 is appeared. At this point, the story of **[LINE]** does not affect estimation of sunshine duration at all, and thus it is not necessary to make a vertical layout.

C) On the other hand, a story of certain **[LINE]** in original design condition can be "Minimum allowable story", because **[LINE]** in section B and section C has no effect on estimation of sunshine duration (continuous sunshine duration, accumulative sunshine duration).

D) As mentioned above, the optimization of vertical layout automation is achieved on the basis of above theory. With this in mind, a **[LINE]** set which is defined in equation (2), ***L***, must be redefined as follows. If a certain **[LINE]** is located in section A, a **[LINE]** set, ***L***, is defined as formula (5), and if a certain **[LINE]** is located in section B or C, a **[LINE]** set, ***L***, is defined as formula (6). $\boldsymbol{R}_\mathbf{T}$ (equation (4)) that is the set of possible alternatives of all layer-adjustment is able to be deduced from this set ***L*** that can be redefined using equation (3) and equation (4)

$L = \{x|x \in$ all expected housing unit elements between the original story and the maximum allowable stories, where, the minimum allowable story $\geq 0\}$ (5)
$= \{u_{min}, u_{min+1}, \cdots, u_{Max-1}, u_{max}\}$

$L = \{x|x \in$ an expected housing unit element is the original story$\} = \{u_{max}\}$ (6)

Where,
L: set of housing unit line([LINE]) in the apartment building

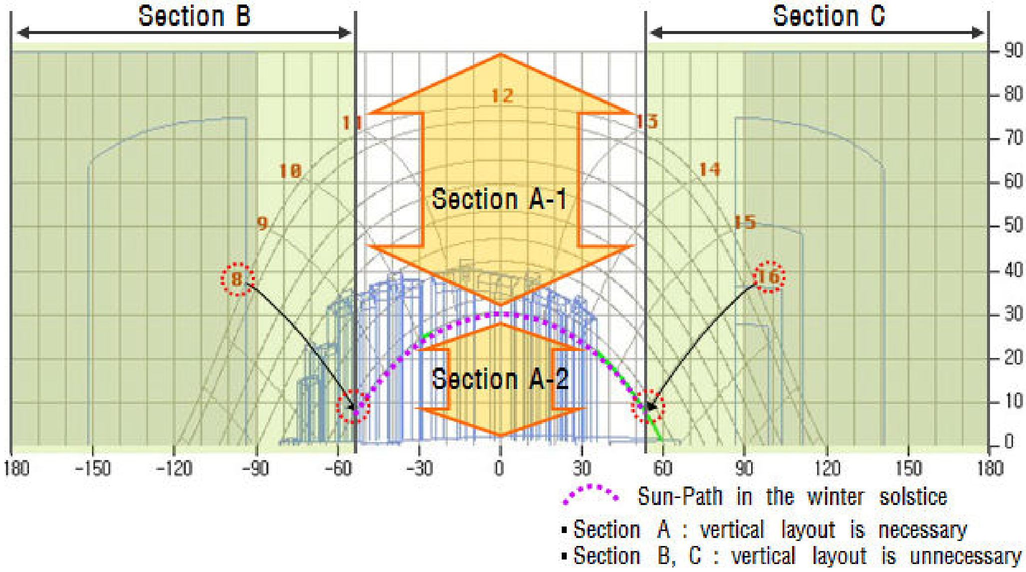

Figure 5. Classification of sections for vertical layout in WALDRAM diagram

VERIFICATION

In order to verify the objectivity of the method for the optimum vertical layout suggested in this study, we compared the results by optimum vertical layout with the results by manual process. We were able to confirm that both methods provided the same results. Also, the comparison of solar rights assessment results (continuous sunshine duration, accumulative sunshine duration and satisfaction of the solar rights) against alternatives between manual process and optimum process for vertical layout verified the accuracy of the method for the optimum vertical layout.

The ultimate goal was to suggest the optimal vertical layout of the obstruction buildings (T apartment Bldg.103 and Bldg.105) in order to assure the solar rights at all survey housing units in survey building (C apartment Bldg.202). With respect to this, however, Bldg.105 was excluded in vertical layout because Bldg.105 was determined to be unhelpful in improving the solar rights or to increase sunshine duration. In original design condition, the obstruction Bldg.103 is 20-storied building that has six [LINE]s. On this condition, the method of general automation generated 85,766,121(=21×21×21×21×21×21) alternatives. Meanwhile, the method of the optimum vertical layout generated 2,665,872(=9×9×11×11 ×16×17) alternatives. In this process, some of them were selected for the comparison in terms of analysis of solar access. The results from those layouts were compared with the results by manual process; in the result, the comparison of cases show the same result.(see Table 2) .

Table 2. Comparison of manual process and optimum process for vertical layout.

Alternatives	Loss*	Analysis results by manual design process				Analysis results by optimum vertical layout			
		Survey unit.#101 in Bldg.202		Survey unit.#105 in Bldg.202		Survey unit.#101 in Bldg.202		Survey unit.#105 in Bldg.202	
		CSD**	ASD***	CSD	ASD	CSD	ASD	CSD	ASD
Alt 1	14	239	299	54	162	239	299	54	162
Alt 2	16	239	299	99	175	239	299	99	175
Alt 3	20	239	299	99	175	239	299	99	175
Alt 4	22	255	315	125	201	255	315	125	201

* Loss of stories compared with the original design condition
** CSD : continuous sunshine duration (min)
*** ASD : accumulative sunshine duration (min)

CONCLUSION

The objective of this paper was to develop the method for the optimum vertical layout of the apartment buildings that can provide proper design alternatives and useful information to ultimately improve indoor comfort. In this study, we suggested an automatic optimum model for vertical layout in high-rise apartment buildings as well as to verify the efficiency and the validity of the objective method. To this end, we compared the methods for the vertical layout to protect solar rights through application in cases. The major results of the study are as follows.

(1) The method of vertical layout by manual process entail iteratively a complicated process of analyzing and investigating numerous possible alternatives to find out the best alternative for improving solar rights in survey units. This method considerably needs lots of time and experts. Not only that, it lowers the productivity and efficiency of work and may possibly mistakes in decision and needs some automatic process using computer.

(2) For the automation of vertical layout of high-rise apartment building, the set of alternatives, $\boldsymbol{R}_{\mathrm{T}}$, for vertical layout by automation is defined through the Cartesian product of the defined set ***B***(set of **[BUILDING]**) and set ***L***(set of **[LINE]**) and is shown in equation (4).

(3) For the improvement of inefficiency and absurdity that can arise from simple automation, we have attempted to optimize the method of vertical layout by automation in such a way to set intuition of expert knowledge about the standard sunshine duration and sun-path in the winter solstice.

(4) In the verification of the automatic method for the optimum vertical layout, first, it is expected to reduce in processing time and workforce in comparison with manual method; second, faster processing is expected in comparison with the method of general automation.

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CFD in the Design of a Music Centre Foyer

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SUMMARY

This paper presents the process how CFD was used in the design of Helsinki music centre foyer. The aim was to ensure appropriate thermal conditions in foyer occupied zones. This flow field has very challenging aspects. Supply air jets, heat loads, furniture, stairs, walkways and other objects produce a very complicated 3D flow field. Typical challenge is design the supply air distribution so that the temperature is in appropriate level and fresh air flows equally to the occupied zone without draught risk. A correctly performed CFD simulation is the only calculation method which can estimate the detailed indoor air flow field with proper accuracy for design purposes. The air flow field is solved approximately depending on mathematical model and numerical method. We learn more about air flow behaviour in different situations and this supports the design team to find the appropriate air distribution systems and constructions. The simulation process starts with individual air device simulations. Different air flow quantities and devices are simulated and the effect of air jet flow is estimated. The simulated supply air jet patterns are then used in the foyer calculation model as a boundary conditions. This reduces the number of calculation nodes and saves simulation time. The device simulation models can then be saved to an object library for future projects. This simulation process seems to be a sensible way to model indoor environments and supply air distribution systems.

INTRODUCTION

CFD is a widely used simulation method in industry and it is also an effective design tool for technical building services. We have been using CFD simulations for years to study the flow behaviour of the indoor air environment, and noticed its advantages in the design process. CFD simulations have been quite expensive, but with increasing computer efficiency it is finally reaching the level at which CFD simulation can be used cost-effectively. This paper presents CFD simulation method that was used in the design work on an air distribution system for a Helsinki music centre foyer. The special features of this space are high occupied load, large space volume and a glazed facade. The design team had a great challenge to design the indoor air environment for 1700 people, which are in occupied zones at the same time. In the design process the most important thing is the experience of the design team. The product model based design process and mathematical modelling can support this knowledge. The fact is that correctly performed CFD simulation is the only calculation method that can take account of all the important flow properties affecting the indoor air environment. CFD is used in spaces where design and quality requirements are high and detailed flow field information is important. Air distribution systems, constructions and sources can be compared, and then the results can be used to illustrate HVAC solutions to customers. CFD is a numerical calculation method and the realistic modelling of a room environment is not an easy task. The important properties affecting flow phenomena have to be identified. Then the mathematical model is created and the space is divided into small control volumes. The equations are discretized and solved approximately using a numerical method. The accuracy of this calculation depends on the mathematical model, the numerical method, the calculation mesh, assumptions, initial values and iteration convergence. CFD simulations typically start with individual supply air device simulations. The different models and air quantities are calculated and compared to test the operating conditions. This allows the design team to choose the appropriate devices for specific locations. The device simulation results are compared with the air jet theory [1,2], manufacturer's profile data and measurements if possible. This information is important for evaluation of the simulation results. The point is that a simulated supply air jet pattern can then be used as boundary conditions in the whole foyer simulations, which reduces the number of calculation nodes and saves simulation time. The device models and simulation results are then saved to an object library for future projects. The challenging part is how to take account of all the important parameters affecting the flow field. The furniture and other objects are typically modelled roughly and the occupancy and lighting loads are divided equally to specific areas. In this project the people were simulated as standing or table groups all over the occupied zone. The tables, chairs and people were modelled together as a cubic model, because it seemed to be a reasonable approximation. The complicated 3D flow field includes many parameters affecting the indoor air environment. The supply air jets, furniture, walkways, space geometry and heat sources affect the indoor air behaviour and produce complicated flow phenomena in the space. The approximations were made so that the most important flow characters were captured, and so that mass, momentum and heat balance would be realistic. This is quite challenging because the initial values are approximations as well. That is why the conclusions should be made with care and compare different models and assumptions. The important aspect is how to model this kind of flow behaviour so realistically that thermal conditions and comfort can be estimated with sufficient quality for design purposes.

METHODS

The CFD simulation model of the foyer is shown in Figure 1.

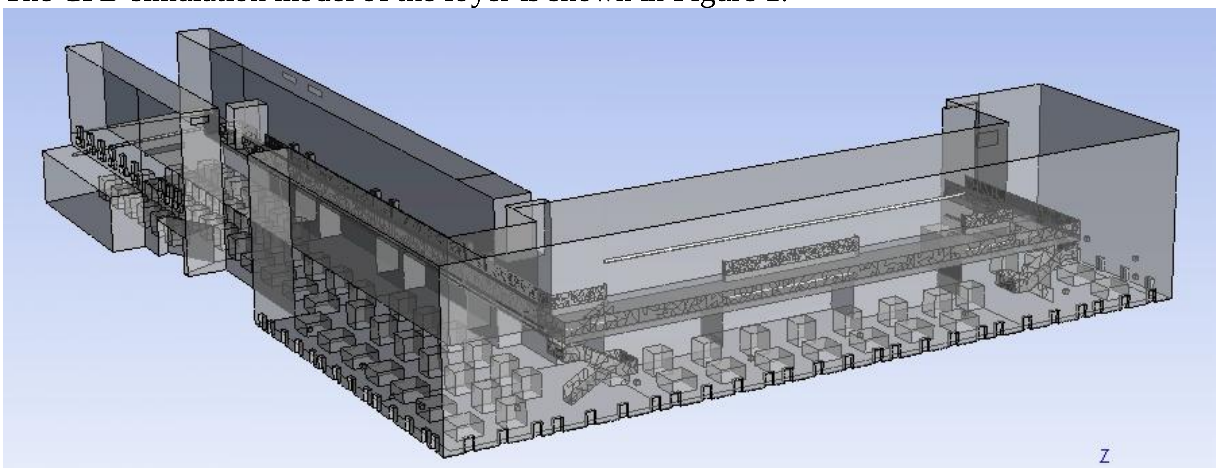

Figure 1. The CFD simulation model of the foyer

The wall geometry was imported to the CFD software from IFC product model generated for this pilot project. The imported IFC geometry was then modified for CFD purposes. After that the air distribution system, walkways, stairs, furniture and other objects were created before the calculation mesh was generated. The supply air distribution system consists of the cooling convectors near the glazed facade, supply air grilles and vortex diffusers located in the interior zone, and low velocity devices in the walkway corridors. Exhaust air grilles were located near the roof zone. The initial values are shown in Table 1.

Table 1. Initial values

Air distribution system, heat loads and constructions			
Foyer	2000 m^2	14 200 m^3	
exterior wall U-value	0.25 W/m^2,K	occupied load	1700x75W
roof U-value	0.16 W/m^2,K	lighting	20 W/m^2
wall temperature	24-39 °C		
glazed facade	total transmittance 28%, U-value 0.78 W/m^2,K		
cooling convectors	glazed facade, 54 units		
supply air grilles	inner wall zone, band installation		
vortex diffusers	inner wall zone, 16 units		
low velocity devices	corridor wall, 6 units		
exhaust air grilles	roof zone		

The lights were located in the corridors and in the facade zone. The occupancy load was simulated as standing and table groups. The heat sources were distributed equally to the surfaces of these cube-shaped models. The supply air devices were simulated individually and the resultant flow patterns were transferred to the foyer model as boundary conditions. The CFD simulation was based on the control volume method. An implicit pressure-based multigrid coupled solver was used [3]. The turbulence model was the k-ε model [4] and the incompressible flow buoyancy was simulated with an expansion factor β=1/T. The atrium foyer was divided into small control volumes using tetra elements. Prismatic elements were located in the boundary layer zone near surfaces. The calculation mesh is shown in Figure 2.

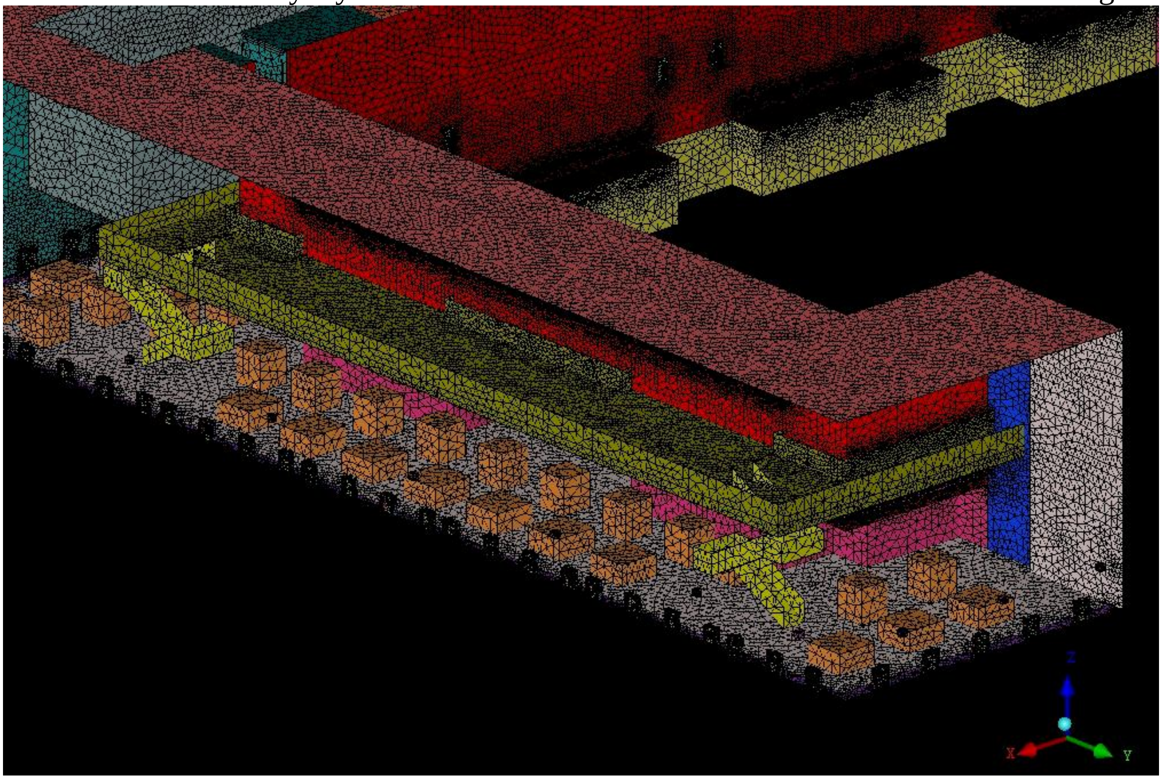

Figure 2. Calculation mesh of the foyer.

The mesh includes 824 593 calculation nodes and 3 271 333 elements. Near the surfaces there were 610 087 prismatic elements in 3 layers.

RESULTS

The results of the device simulations are shown in Figures 3-6. The results of the cooling convector simulation are shown in Figure 3.

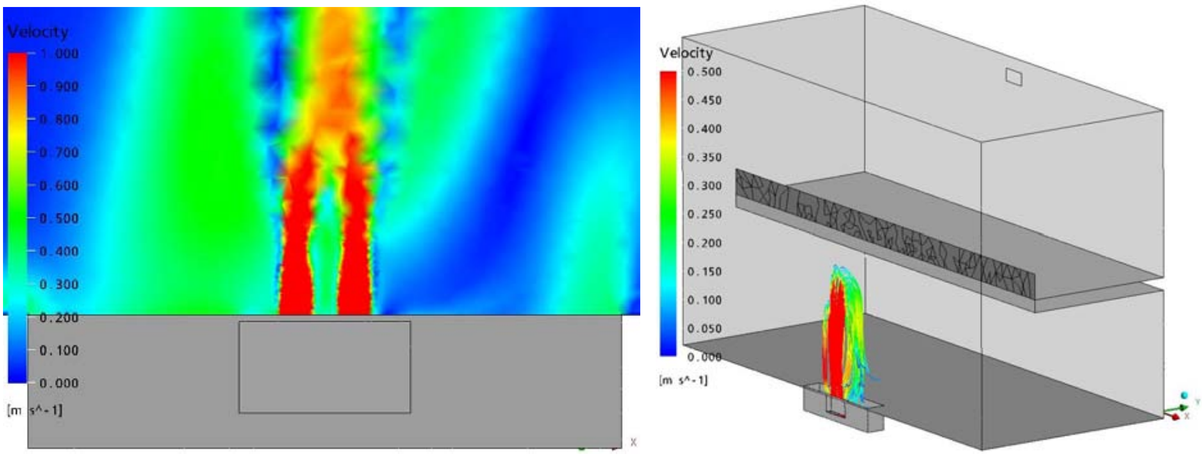

Figure 3. The results of the cooling convector simulation.

The convectors were located in a floor sink near the glazed facade. A simulated flow pattern was utilized in the foyer simulation. The supply air grille simulations are shown in Figure 4.

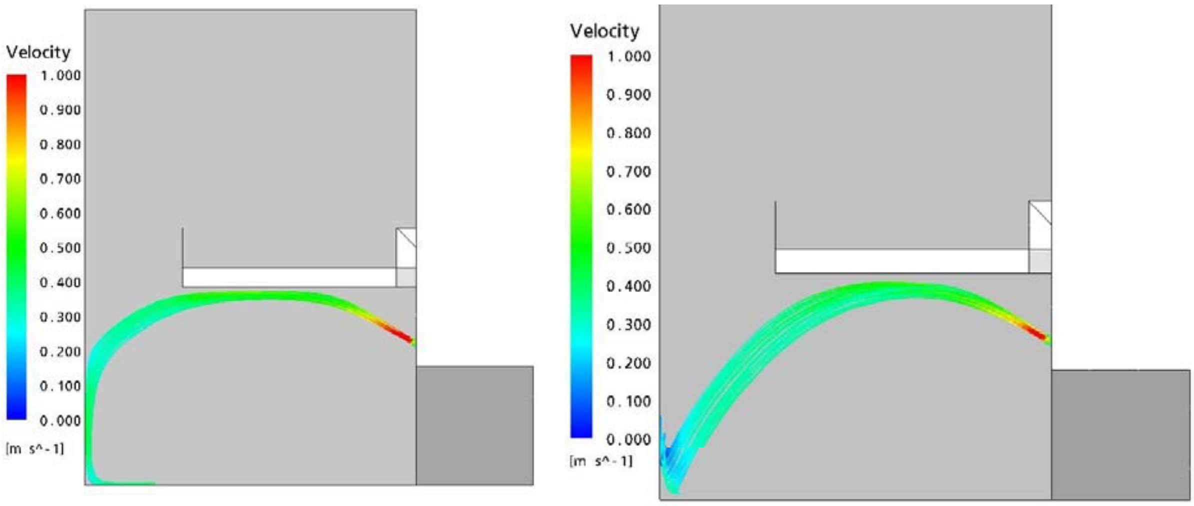

Figure 4. The supply air streamlines from the device simulations.

These devices were simulated in order to estimate the air jet length and how the jet is connected to the walkway roof. The result on the left-hand side was chosen for the foyer simulation, because the air jet seems to connect well to the roof and flows directly to the glazed facade and not to the occupied zone, where it might cause a draught. These devices are located side by side on the wall. For this reason the devices were approximated for the foyer model as a single wide device, see Figure 5.

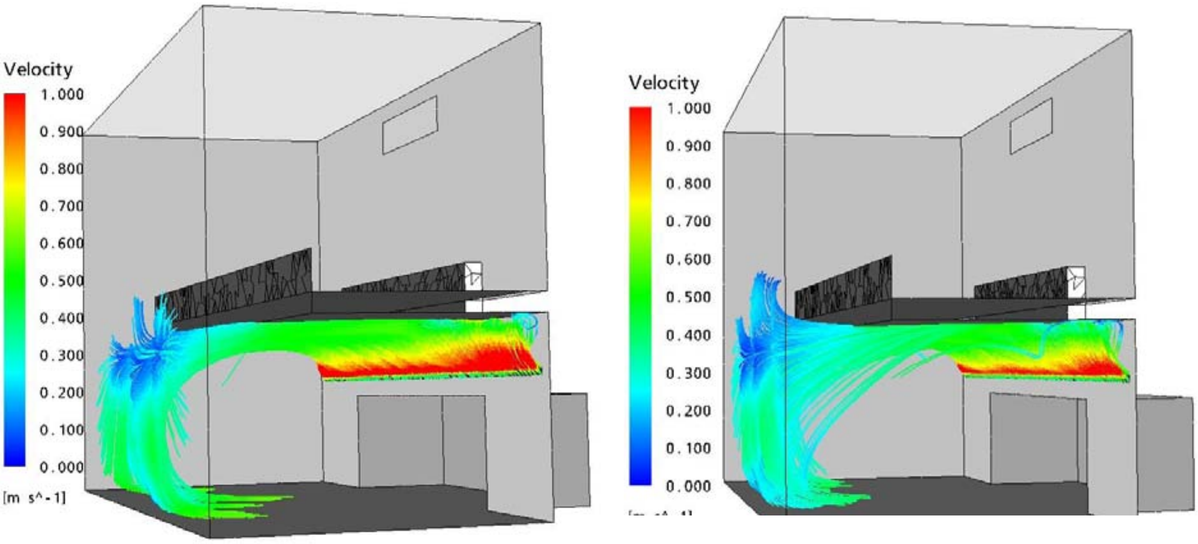

Figure 5. The supply air streamlines from the wide supply air device simulations.

The results of the swirl diffusers simulations are shown in Figure 6.

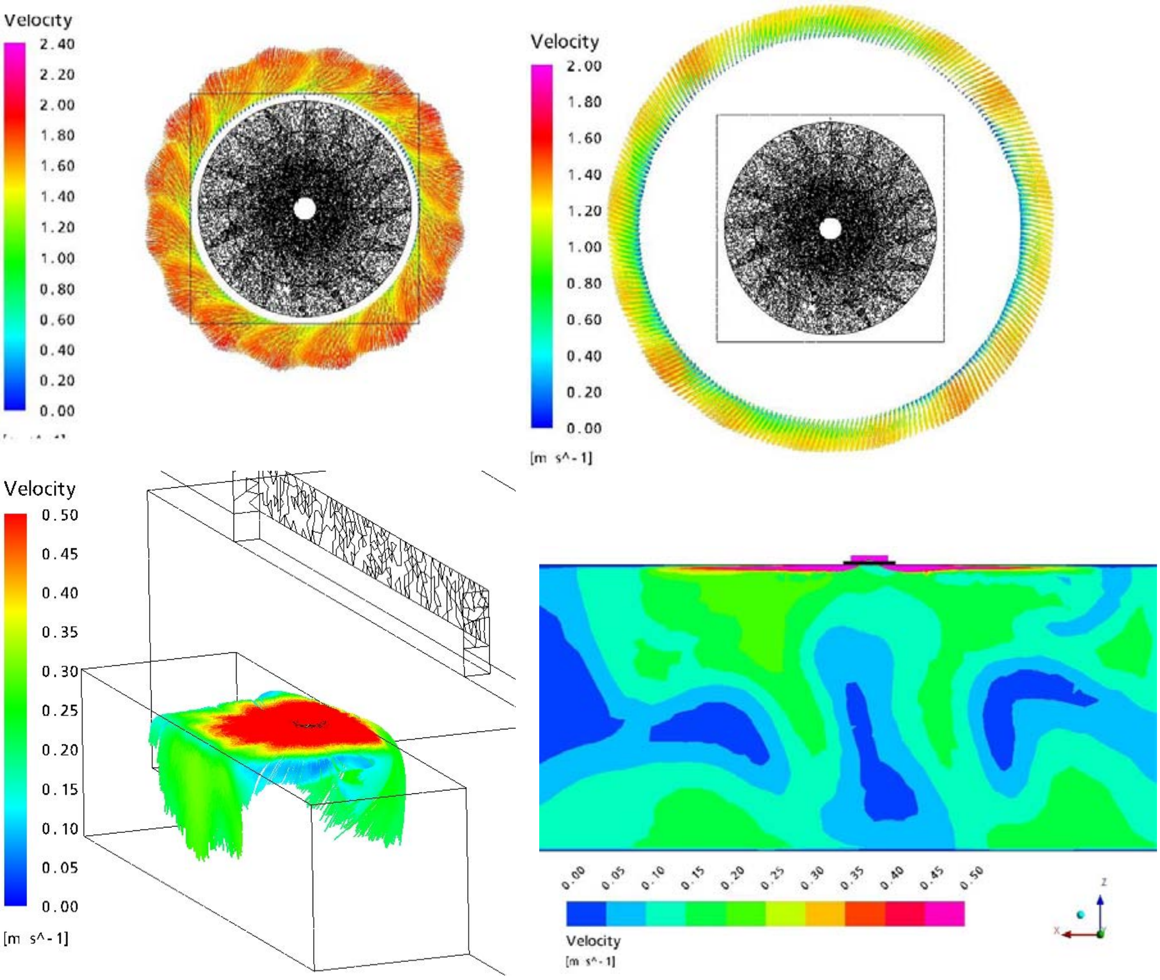

Figure 6. The results of the swirl diffuser simulations.

When the results were compared with the manufacturer's data, they appeared to be in good agreement. The foyer simulation results of the summer design day conditions are shown in Figures 7-10. The results show that the indoor air environment and thermal conditions are reasonable. The cooled air from the convectors near the glazed facade enhance thermal comfort because the air temperature falls and affects the operative temperature [5]. The supply air is distributed so that the temperature in the occupied zone is reasonable, see Figure 7.

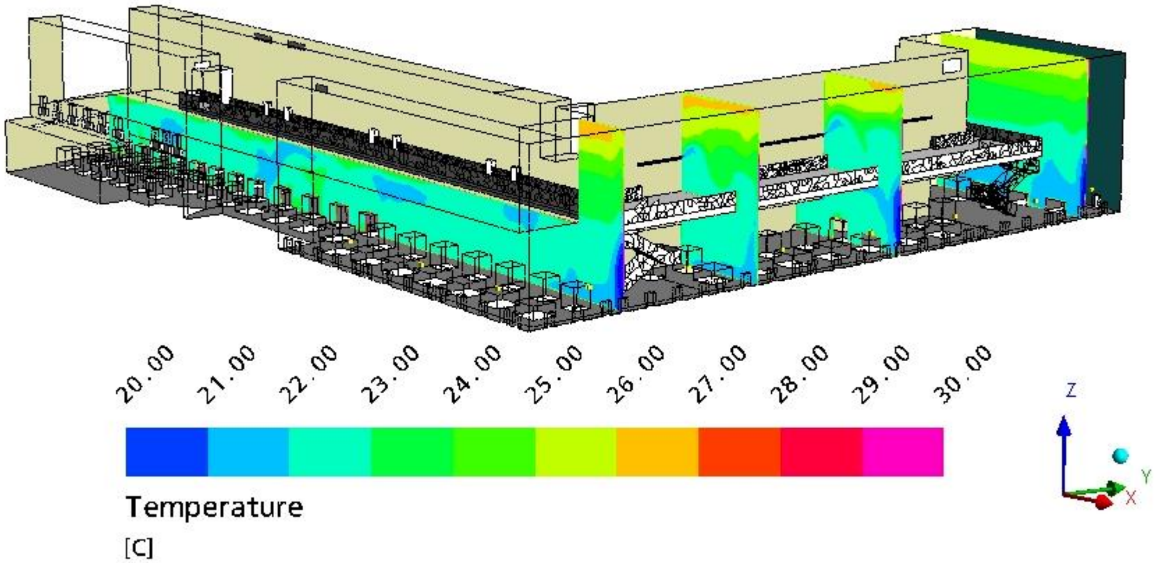

Figure 7. Summer design day temperature distribution in the foyer.

In the occupied zone the temperature is approximately 22-23°C. The temperature is higher close to the roof zone and below 22°C near the air jets. The distribution is quite even all over the foyer. The convector air streamlines are shown in Figure 8.

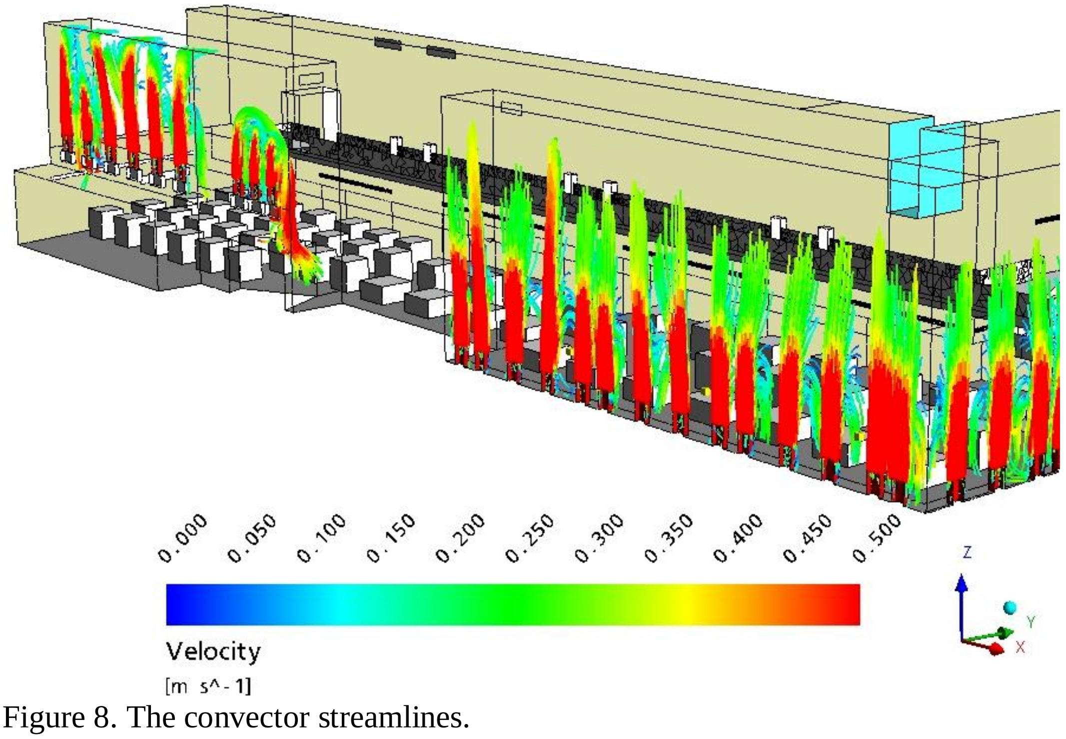

Figure 8. The convector streamlines.

The convector air flows along the glazed facade and it is then mixed with the indoor air. Figure 9 shows the supply air streamlines and velocity distribution. On the 1st floor the jet flows to the glazed facade and the jet length is much longer than on the 2nd floor because it is connected to the roof. The air jet from the cooling convector can be seen near the glazed facade.

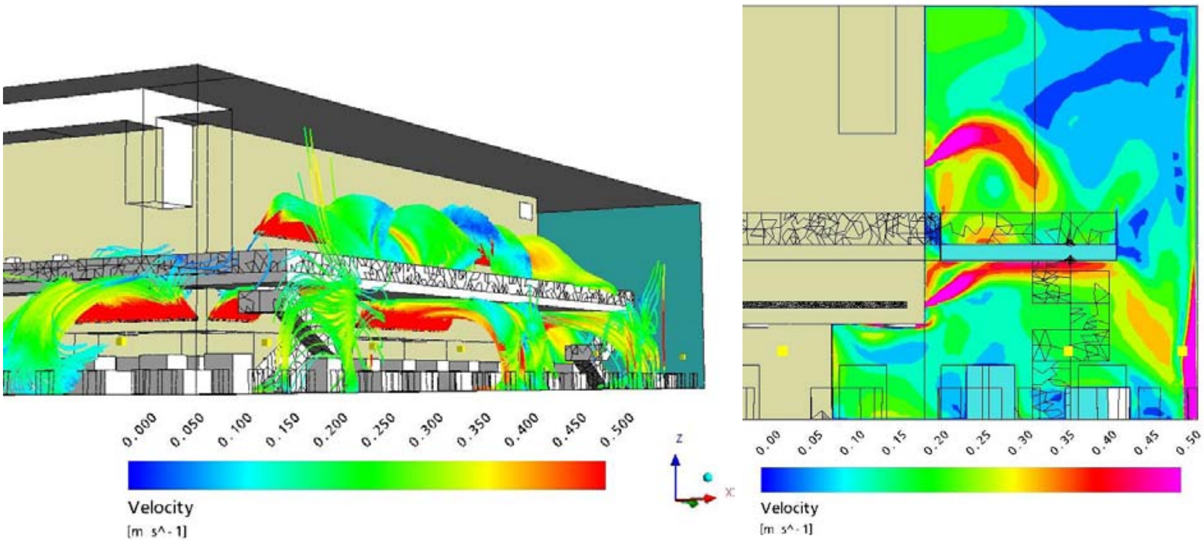

Figure 9. The supply air grille streamlines and velocity distribution.

The velocity profile is shown in Figure 10. The velocity in the occupied zone is mostly below 0.3 m/s.

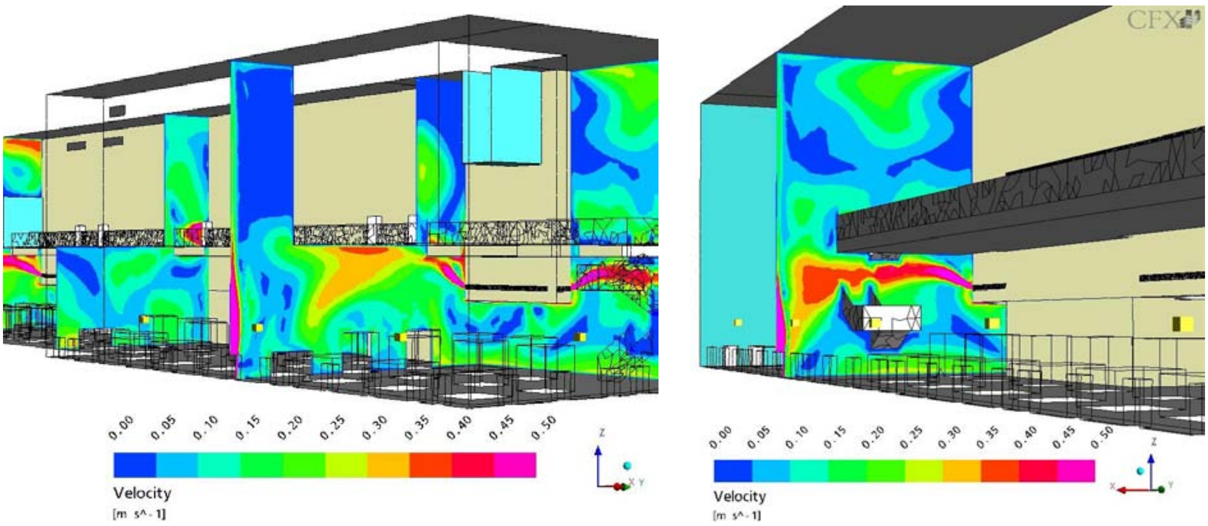

Figure 10. The velocity profile.

On the right-hand side the supply air jet flows directly to the stairs and reduces the thermal comfort in this area. It is recommended that the air jet is not thrown directly to the stairs, where it might produce a draught risk.

DISCUSSION

CFD was used to support the design team to find a functional air distribution system for a music centre foyer. The aim was to ensure appropriate thermal conditions and comfort in the occupied zones. The special features of this space were high occupied load, large space volume and a glazed facade, from where heat flows to the space in summer weather conditions. Air distribution systems, constructions and sources were compared. The individual supply air device simulations seemed to be a sensible way to find appropriate devices for specific locations, because different air flow quantities and devices could be simulated easily and these flow patterns could then be used as boundary conditions in the foyer simulations. This reduces the number of calculation nodes and saves simulation time. The device simulation models can then be saved to an object library for future projects. Approximations were made with the aim of achieving realistic simulation results for design purposes. The mesh should be sufficiently dense to ensure that all the important flow characters can be captured. The mass, momentum and heat balance should be realistic. It should be noted that the initial values are approximations, and for this reason the conclusions should be made with care and compare different calculation models and assumptions. The cube-shaped models for people and the equal distribution of lighting to the roof surfaces seemed to be reasonable approximations. The device simulation results were compared with the air jet theory and the manufacturer's data. This information is important for evaluation of the simulation results. In this study the most important effect on the indoor air environment had the behaviour of the air jet flows.

ACKNOWLEDGEMENT

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Computer Aided Design and Balanced System Selection of Split Air Conditioning Unit

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SUMMARY

The design of systems has become complex and highly specialized as Air Conditioning becomes involved in virtual with every phase of present human existence. Due to numerous applications and the ever-changing service conditions, it is imperative that we have a balanced system with all components properly selected. The split air conditioning refers to a certain physical arrangement of equipments like indoor and outdoor sections. A practical problem is dealt in this paper taking the computer program into consideration to calculate the loads and performance curves of each component are plotted on the same graph and obtained the balancing point.

INTRODUCTION

Split air conditioning system basically made up of two pieces of equipments (viz. outdoor section and indoor section) that are connected by refrigerant pipe work and control wiring. Each piece of equipment contains half of the refrigeration plant required for the vapor compression cycle (Ref: fig.1). In any system which involves many components working for that system need balance between each other. Two major problems which confront the air conditioning industry are improper system balance and misapplication of equipment. Though the air conditioning system is split into two sections, all the components are connected together in series and the refrigerant flow rate is same through all the components and hence the working capacity of all components in their own conditions must of necessarily be the same.

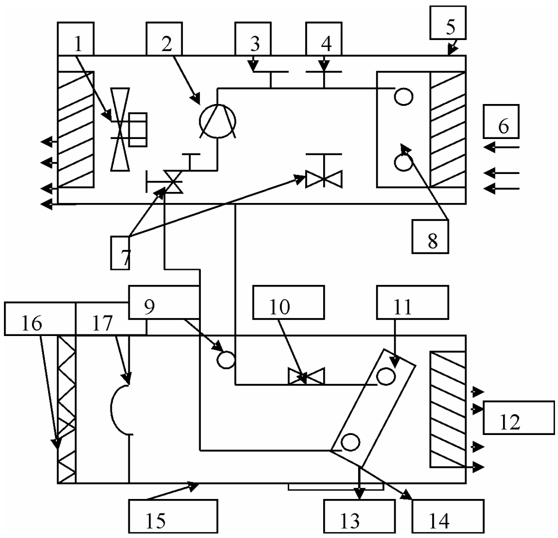

Fig.1 Simple split unit system.

The parts of Split unit are

1. Condenser Fan
2. Compressor
3. High Pressure Switch
4. Condenser Pressure Control Switch
5. Outdoor Unit
6. Air Outlet
7. Stop Valve with Service Port & L.P.Switch
8. Condenser
9. Slight Glass
10. Expansion Valve
11. Cooling Coil
12. Conditioned air
13. Condensate
14. Condensate Drain tray
15. Indoor Unit
16. Filter
17. Fan

The present paper deals with four important aspects namely (a) the cooling load calculation using computer program, (b) design and selection of compressor, (c) design and selection of condenser and (d) design and selection of an evaporator.

METHODS:

(a) The cooling load calculation using computer program:

Total heat required to be removed from the space in order to bring it at the desired temperature by the air conditioning equipment is known as cooling load. If the load estimate is not accurate, the system will not perform properly and it will be impossible to meet the customer's requirements no matter how well the individual components are balanced. Any rule of thumb that we may customarily use shall only be used as a means of check. Purpose of load estimation is to determine the size of air conditioning equipment that is required to maintain inside design conditions during periods of maximum outside temperatures. The two main components of a cooling load in a split air conditioning system are as follows:

(i) Sensible heat gain:

When there is a direct addition of heat to the enclosed space, a gain in the sensible heat is said to occur. This sensible heat is to be removed during the process of summer air conditioning. The sensible heat gain may occur due to any or all of the following sources of heat transfer: wall gain load into the building by conduction through exterior walls, floors, ceilings, doors and windows due to the temperature difference on their two sides.

$$Q = U A (T_o – T_i)$$

where

U = Overall heat transfer coefficient
A = Area of the wall
T_o = Outside air temperature
T_i = Inside air temperature
$f_{o,i}$ = outside and inside film surface conductance
K_a = thermal conductance of air space

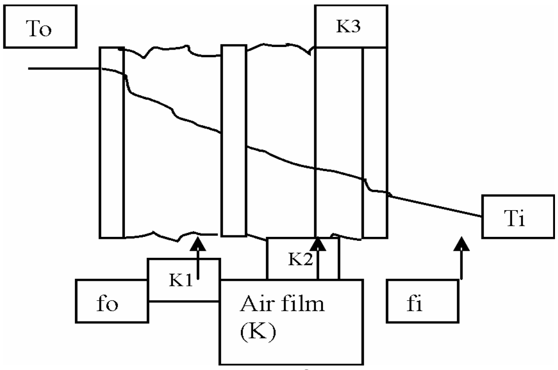

Fig. 2 Heat transfer along the wall.

$$U = \frac{1}{\frac{1}{f_o} + \frac{x1}{K1} + \frac{1}{Ka} + \frac{x2}{K2} + \frac{x3}{K3} + \frac{1}{f_i}} \quad \text{eq - 1}$$

Heat received from solar radiation consists of heat transmitted directly through window glass, ventilators etc. Heat liberated by the occupants which normally called as respiration load is also of this kind only. Heat given off by the lights, motors, machines etc. and heat carried by the outside air which leaks in is also called as air filtration load is to be considered based on openings of the doors under sensible heat gain.

(ii) **Latent heat gain:**

When there is an addition of water vapour to the air of enclosed space, a gain in latent heat is said to occur. This latent heat is to be removed during the process of summer air conditioning. Latent heat gain may occur due to moisture in the outside air entering by infiltration, condensation of moisture from occupants, heat gain due to moisture passing directly into the conditioned space through permeable walls or partitions from the adjoining regions where the water vapour pressure is higher.

Heat gain due to infiltration, due to ventilation, from occupants, from applications, from products and power requirements are to be considered to arrive at the total load on the system. Total cooling load will be the summation of the all four different loads discussed in this paper, outside conditions are 40 C DBT and 28 C WBT, and the required conditions are 21 C DBT and 60% RH and the capacity of the split air conditioning required is 3.79 KW.

(b) **Design and selection of compressor:**

The theoretical power required to drive the compressor may be found by multiplying the actual refrigerating capacity of the compressor in Kilowatts by the theoretical power required per unit capacity for the required operating conditions. The refrigerating capacity of the compressor, in general, decreases as the condensing temperature increases and raising the suction temperature will usually increase the power requirements. For the designed load of 3.79 KW, with the input of compressor ratings supplied by ASHRAE guide, computer will select a 1.5 Hp motor compressor (Model No.E036). The selected compressor ratings are as given below.

Table-1 compressor capacity vs. suction temperature**.**

Capacity of compressor (KW)	2.36	3.06	5.2	6.19
Suction Temperature(K)	279.8	282.6	294.98	298.18

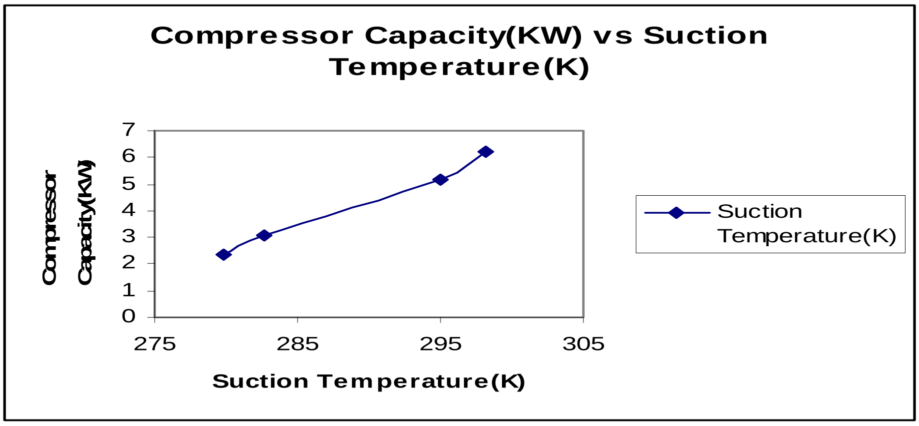

Fig. 3 Compressor capacity vs. saturated suction temperature .

The performance and merits of compressor is as shown in the figure. The reason for selecting this compressor is that it has approximately the right amount of capacity of 294.98 K suction temperature, which is just closer to the balancing point. Refrigerant selected is R-407C and a remote multi circulated air cooled condenser is selected for this requirement. Compressor performance/capacity is plotted on the ordinate and the saturated suction temperature/evaporator temperature on abscissa. Compressor operating conditions are for different condensing temperatures vary from 312K to 333K.

(c) **Design and selection of condenser:**

Capacity of the condensing unit depends upon the capacity of the compressor as the design saturated discharge temperatures depends primarily on the size of the condenser selected and upon the quantity and temperature of the available condensing medium. In this context, the rating and selection of condensing units are practically the same as those for rating and selection of compressors. Remote multi circulated air cooled condensing unit (Model no. STF 812) ratings are based on the saturated suction temperature and design ambient air temperature. Total heat rejection at the condenser includes both the heat absorbed in the evaporator and the energy equivalent of the work of compression. The performance of the condensing unit is as follows:

Table-2 Condenser capacity vs. Saturated suction temperature**.**

Capacity of condenser(KW)	2.9	4.35	6.2	7.85
Saturated Suction Temperature(K)	286.75	293.6	301.56	313.84

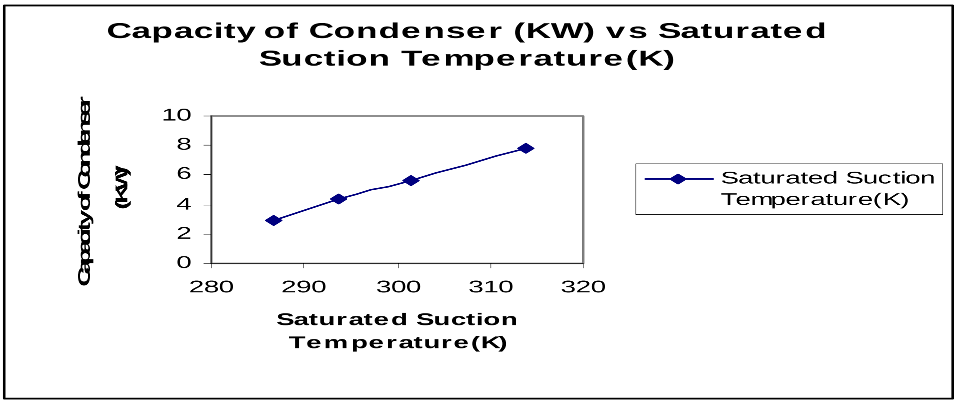

Fig. 4 Condenser capacity vs. Saturated suction temperature.

Condensing unit capacity is taken on the ordinate and the saturated suction temperature difference on the abscissa and the performance curves are plotted in the fig.4

(d) **Design and selection of an Evaporator:**

Once the conditions or the requirements of the air conditioned room are selected, the design and selection of an evaporator from the manufacturer's details is relatively simple. Evaporator temperature difference (TD) is the important aspect in the selection of evaporator and is defined as the difference in temperature between the temperature of the air entering the evaporator, usually taken as the space design temperature, and the saturation temperature of the refrigerant corresponding to the pressure as the evaporator outlet. In practice, the evaporator capacity varies directly with the evaporator TD. For the required TD in lower temperature applications, lesser will be the possibility of frost accumulation in the circuit and will result in better overall system performance. From ASHRAE table, for the required 60% RH and 8 TD, the model selected is BL 2116, whose performance details are mentioned below.

Table-3 Evaporator capacity vs. evaporator TD.

Evaporator Capacity (KW)	0	2.4	3.2	4.01
Evaporator TD(K)	0	4.21	6.4	8.45

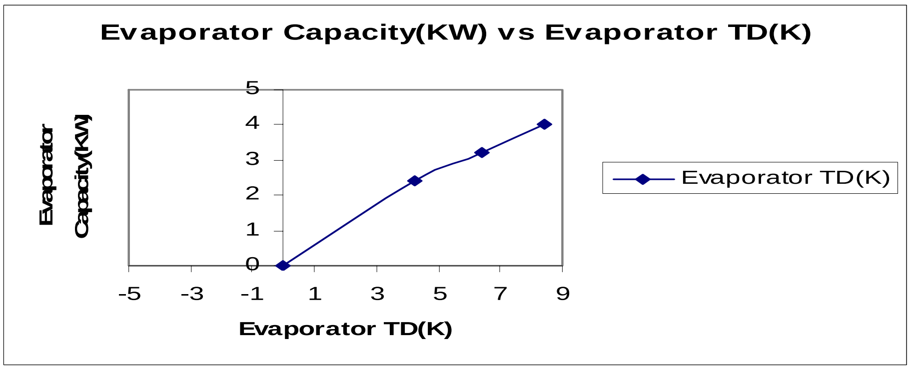

Fig. 5 Evaporator capacity vs. evaporator TD.
While plotting the performance of the evaporator capacity (KW) is on the ordinate and the Evaporator temperature difference (K) on the abscissa.

RESLUTS AND DISCUSSIONS:

It is most scientific and desirable to design and select the whole system of split air conditioning unit though practically not feasible to have a manufacturing endeavor to fabricate innumerable models economically to suit the customer's requirement. It is, therefore, the design engineers who select a proper combination for a particular situation giving the most desirable performance. For the present problem, as discussed earlier, the design load is 3.79KW. For the required conditions from the manufacturer's catalogue, compressor, condenser and evaporator are selected and they need to be balanced in their performance. Hence balance of the components in split air conditioning is observed which is highly desirable. It is also observed that the selection is on basis with selection of standard company. We can compare the balance between compressor and condenser, the compressor and evaporator, condenser and evaporator and the balancing point can be compared in all the cases as nearly 3.80 KW which is very close to our required load of 3.79KW. Finally balancing of all components on a single graph (with different scales) can be seen with the performance curves plotted and the meeting point of all the curves will be very close to the required capacity of the split air conditioning unit of 3.79KW.

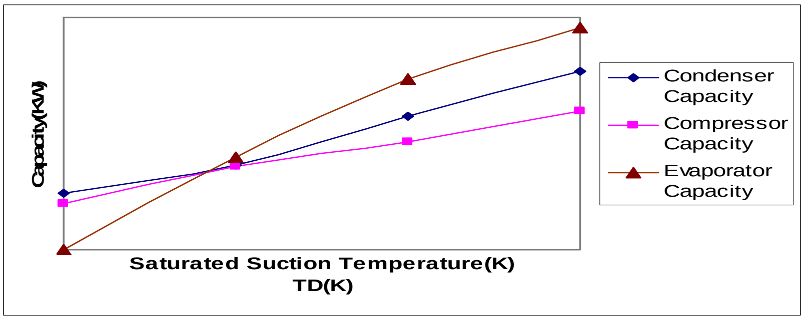

Fig.6 Capacity of Condenser, Compressor vs. Saturated Suction Temperature and Evaporator Capacity vs. TD.

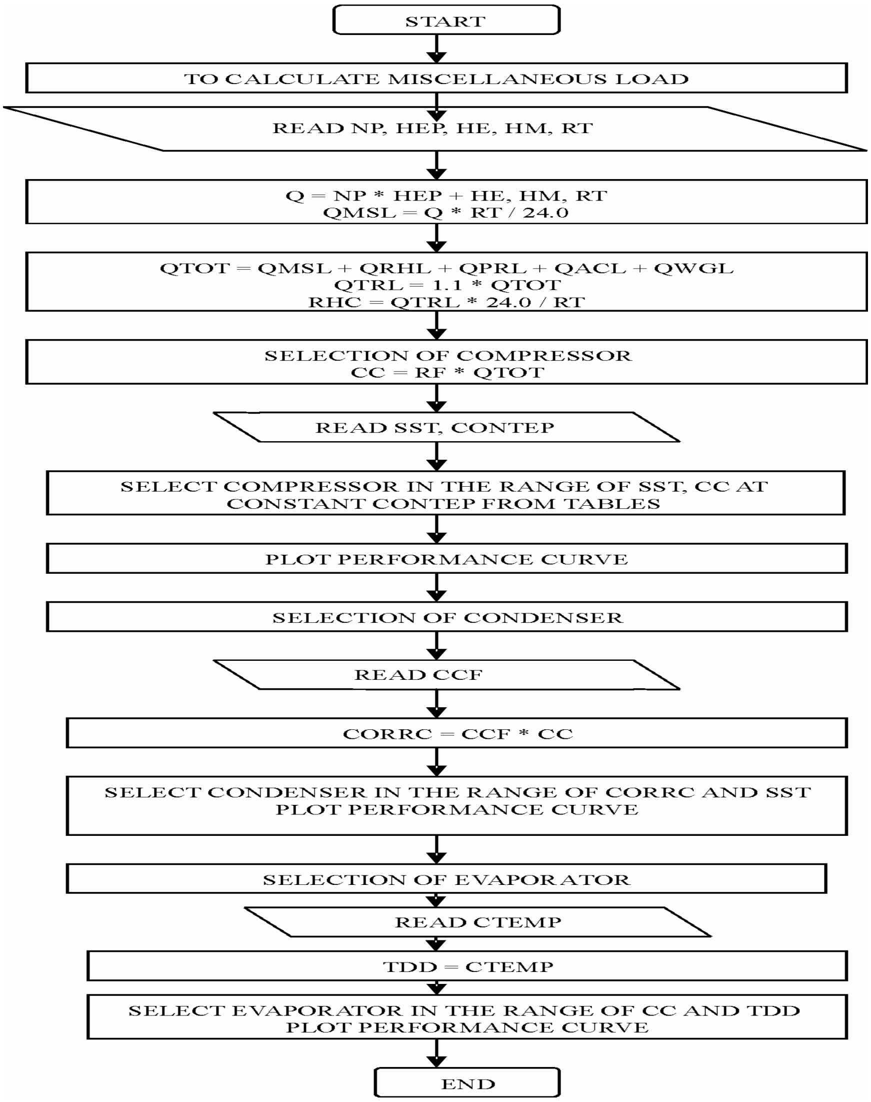

Fig. 7 Flow Chart.

Where RT – room temperature

SST- saturated suction temperature   TD-evaporator temperature difference

CTEMP-condenser temperature   CC-compressor capacity

PRACTICAL IMPLICATIONS ON THE RESULTS:

In the split air conditioning, for different applications, selection of compressor, condenser and evaporator is an important part of the manufacturer. In wide range of applications, the working conditions of the above equipment should match with practical existing conditions. For this, the present paper is very much required. After careful calculation of each load that are coming into picture at different stages, capacity of compressor, condenser and evaporator are selected using empirical equations. Once capacities of compressor, condenser and evaporator are calculated, we need to go to the catalogue of manufacturer (better to select most genuine manufacturer) and refer to the range of applications at desired conditions. In the present paper, performance curves of individual component are plotted for the combination of two and observed that there exists a common interference of the two superimposing of all the performances of the components, they are passing through a nearly common point closer to the required load. When we select the components like compressor, condenser and evaporator, this after superimposing the performance of individual and obtaining the intersecting point at the required load, then selection will be more appropriate because it has been tested and proved for wide range of applications/working conditions. In the present paper, effort is made for minimum load (3.79 KW) and if the same method is adopted for higher loads and the system is selected with proper balance (after obtaining the performance curves), it will be the best selection of the components in the system and it will be withstanding to any range of conditions that we will apply for the load that is calculated meticulously.

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Heat pumps

IEA HPP Annex 28 – Uniform testing and seasonal performance calculation of combined operating heat pump systems

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SUMMARY

Climate protection targets require the performance assessment of HVAC systems. As basis uniform component testing and subsequent transparent calculation of the seasonal performance factor is needed. Highly-integrated system layouts, however, are not covered by existing testing and calculation standards.
Annex 28 in the heat pump program of the International Energy Agency delivered recommendations for extended testing and calculation of heat pumps with combined space heating and domestic hot water production. Test procedures are an extension of the current heat pump testing for space heating-only and domestic hot water-only mode. Calculation is performed by an extended temperature class approach (bin method). Validation of the calculation method with field monitoring results show a deviation of the seasonal performance factor in the range of ±6%, approving the feasibility of the approach. Implementation of the results in European and national standards has begun in the frame of the EU Building Directive EPBD.

INTRODUCTION

Market boundary conditions

Since the mid of the Nineties of the last century, building directives set tighter limits for the energy consumption of the building leading to the development of low energy buildings with significantly reduced space heating (SH) energy needs, e.g. down to dwellings, which must not require more than 15 kWh/m^{2}a for space heating according to the German passive house standard. This enables to provide the space heating need by reheating of the ventilation air. Domestic hot water (DHW) energy needs reach fractions up to 50% of the total heat energy need. On the other hand, reduced temperature requirements of the heating system lead to new adapted system layouts, and environmentally sound technologies like heat pumps, solar and wood systems have been (re)introduced in the market and have increasing market shares. These developments lead to a generally higher system integration to multifunctional systems, which cover different building needs, in common layout space heating and DHW, often amended by the ventilation function. Currently, additional cooling and space conditioning functions are integrated due to extended comfort requirements.
However, these new system developments are often not covered by existing test procedures and calculation methods. Due to the changing system markets, different stakeholders have strong interest in guidelines to compare the performance of systems under different boundary conditions:

- *Manufacturers* need regulations for providing precise and uniform technical data of their components derived from standard testing.

- Designers need methods to enable a comparison of different heating systems or system layouts in the design process.
- Consumers need a clear indication of environmental impact and energy costs as a guideline for their purchase decision, e.g. by transparent labelling.
- Consultants and policy makers need uniform values to set targets in regulations and directives on the background of climate protection policies. The resulting energy labels or building standards may be based upon the Seasonal Performance Factor (SPF) or primary energy as well as CO_2-emissions, which can easily be concluded from the SPF.

Standarisation work

These needs are addressed by the EU-Directive on the Energy Performance of Buildings (EPBD) [1], which has the target to introduce a Building Energy Certificate, which is displayed in the building and contains the energy consumption of the building and the installed system technologies in terms of primary energy consumption and CO_2-emissions. Moreover, measures to improve the energy efficiency of the building and the system technology are included in the assessment.

As common basis for the building energy certificate the EU has mandated the revision and extension of the current building and system technology standards. Under Mandate 343 a set of standards for the building and the system calculation is currently elaborated in the European standardisation organisation CEN. Most of the standards are almost finished and currently in the final enquiry called formal vote. The introduction of the standards on the national level will probably start by 2008.

With regard to the required component data test standards for domestic hot water appliances are currently to be harmonised based on the EU Labelling Directive. Under the EU Mandate M/324 [2] test standards of household appliances for DHW production are revised. For the harmonisation of the testing common EU reference tapping profiles have been defined in the mandate.

METHODS

The scope of the IEA HPP Annex 28 was the development of test procedure and subsequent SPF calculation method for combined operating heat pump systems for SH and DHW. The nine countries AT, CA, CH, DE, FR, JP, NO, SE, USA participated in IEA HPP Annex 28. Systems found on the market can be distinguished in alternate and simultaneous operating systems. In alternate operating systems the heat pump is switched to either the space heating or the DHW operation. In simultaneous operating systems space heating and DHW energy is delivered at the same time. Most common methods for simultaneous operation are refrigerant desuperheating and condensate subcooling as well as cascade cycle layout. Examples of alternate and simultaneous combined operating systems are shown in Figure 1

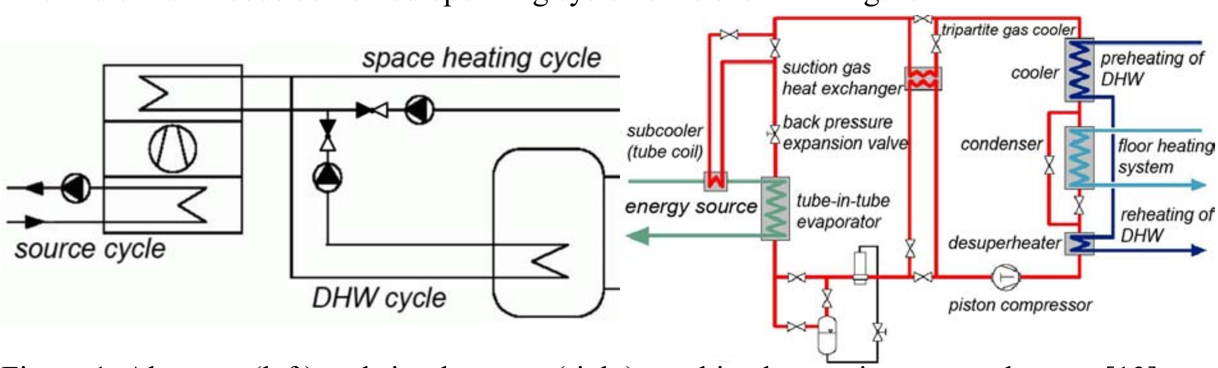

Figure 1. Alternate (left) and simultaneous (right) combined operating system layouts [13]

Ventilation compact units with exhaust air heat pumps have been treated, as well, since they have become very common in low energy dwellings. The accomplished work in Annex 28 comprises the testing of the systems according to existing and extended standards, calculation of the systems and field testing for the validation of the calculation results and to gain experience with the functionality for system optimisation. The following basic definitions have been made for the testing and calculation inline with the ongoing standardisation activities.

Test procedure
In order to uniformly cover the different system configurations on the market, a black box testing is applied, where only the values at the system boundary are used for the calculation, i.e. values, that can be measured outside the unit. This approach enables to abstract from the internal system configuration, which is difficult to consider since it varies a lot from one system to the other. The black box approach is depicted in Figure 2.

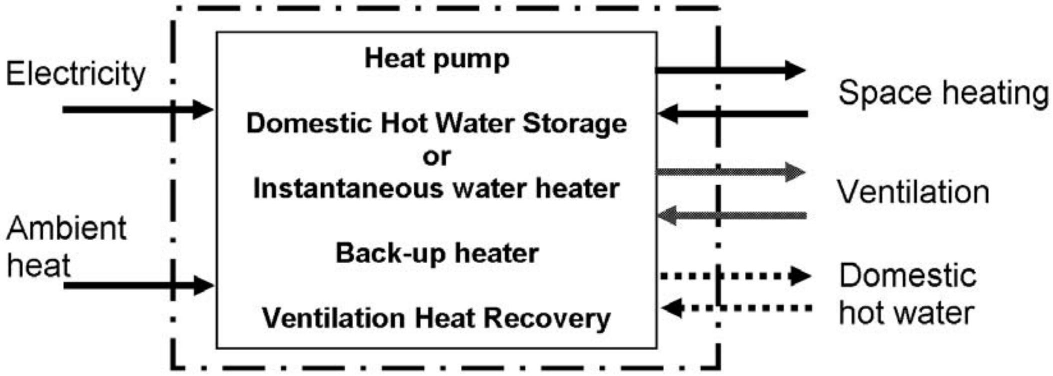

Figure 2. Black box testing as basic principle for the test procedure

Calculation method
Due to the common system testing (generator and storage) and common bivalent system layouts for heat pump systems (heat pump and back-up) the system boundary has been chosen to comprise the generator part with attached storages (both for space heating or domestic hot water) and eventually installed back-up generators. The system boundary in the calculation scheme of the EPBD is depicted in Figure 3.

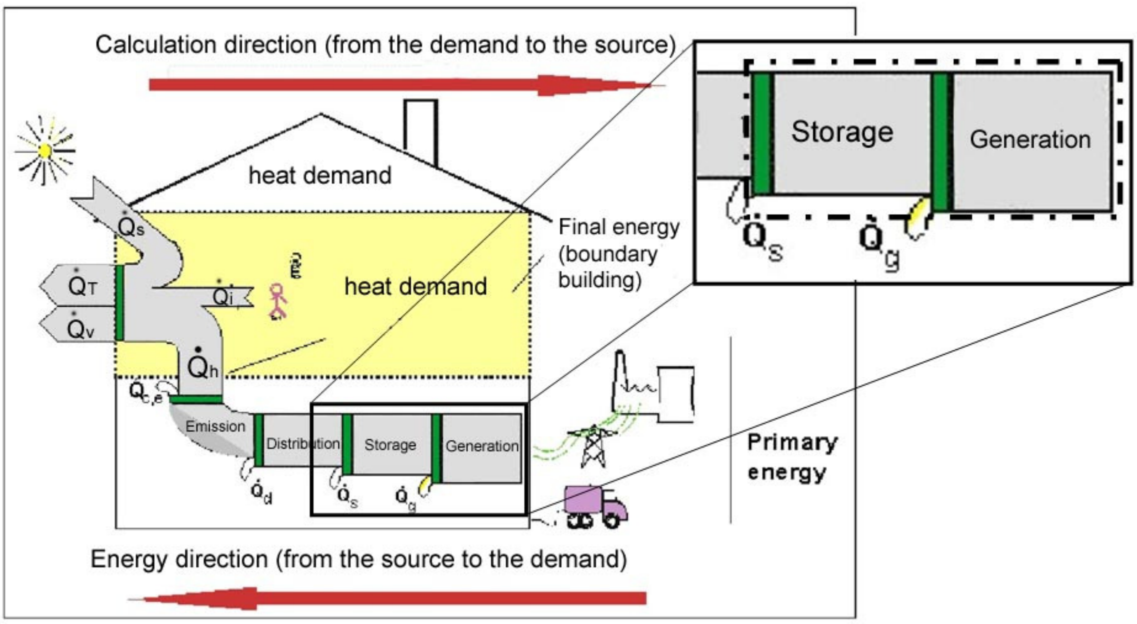

Figure 3. System boundary of Annex 28 in the calculation scheme of prEN 15316-1 [3]

RESULTS

Survey of heat pump markets and state in standardisation (state 2003)

As starting point a market survey of systems and the state-of-the-art in standardisation has been carried out.

Concerning combined operating heat pump systems, markets are different for Europe, North America and Japan. While Europe has mostly alternate combined operating systems on the market, North America mainly uses simultaneous systems with desuperheating for domestic hot water production, sometimes in combination with combined cooling/air conditioning and domestic hot water production, as well. In Japan, a combined alternate system was recently introduced in the market, where a CO_2 heat pump supplies hot water to a storage, which provides water at different temperature levels for floor heating, higher-temperature radiator heating and domestic hot water use.

In all participating countries, standards for the testing of the SH-only and DHW-only operation modes exist. In Japan, the DHW testing refers to the CO_2 heat pump water heaters, so called Eco–Cute systems, which have become very popular since their market introduction in 2002. However, testing for combined operating systems is missing except in the American standards ASHRAE 137 [4] treating the specific case of air-to-air heat pumps with DHW production by desuperheating, which is a common system configuration in the USA and Canada. Concerning the SPF calculation, no common standard for combined operating systems exists, either. For the SH-only mode, temperature class approaches (bin methods) are used in the USA and in some national guidelines in Europe, e.g. ASHRAE 116 [5] and VDI 2067-6 [6].

Evaluation of current European heat pump testing standards

<u>Space heating-only operation</u>

EN 14511 [7] performs steady state testing of the SH-only operation replaced the former standard EN 255-2 [8]. Comparison of the standards showed that resulting COP values according to EN 14511 are in the same range as EN 255, but approx. 4-5% lower due to the changes in the determination of the mass flow rate, which used to be manufacturer given in EN 255-2 and is now determined by a fixed temperature difference in the so-called standard rating point.

<u>DHW-only operation</u>

In contrast to steady-state of EN 14511, EN 255-3 [9] for the test of the DHW operation evaluates a tapping cycle, thus COP-value is derived by averaged evaluation of the energy balance for the tapping. For the determination of the COP DHW tappings of half of the storage volume are repeated, until the energy amount of the tapped hot water is within a 10% range. EN 255-3 is currently under revision in CEN/TC 113/WG 10.

Although comfortable to use, the major draw-back is a very long testing time of about 4 days for a single test point, mainly due to the testing of storage stand-by losses.

Further shortcomings of the EN 255-3, in particular with regard to combined operation, have been identified:

- EN 255-3 does not deliver a DHW output capacity, which is useful to evaluated running time in combined operation.
- EN 255-3 does only deliver one test point for the entire range of possible source and sink temperatures.
- EN 255-3 does not define a DHW outlet temperature, but uses manufacturer controller settings. Consequently, a direct comparison of tested heat pumps is not possible due to differing DHW temperatures.

On the basis of this analysis, modifications are recommended by the IEA HPP Annex 28 to be implemented during the revision of the EN 255-3.

- Storage stand-by losses are not necessary for the COP determination, as the change is about 4% and thereby in the range of the measurement exactness [10].
- Thus, if a heat loss value of the storage is known, testing of storage stand-by losses can be omitted. This can reduce the testing time significantly. If no information on the storage is available, i.e. in highly-integrated systems, the entire EN 255-3 cycle can be performed once, while the COP shall be evaluated for more test points.
- Using the period of the cycle applied for the determination of the COP, an average output capacity can be calculated.

<u>EU tapping patterns for DHW testing</u>
Delivered COP values for EN 255-3 tests using the European tapping profile according to mandate M/324 tend to be lower due to a higher mean temperature in the condenser.
Despite partly small mass flows (EU reference pattern 1), the reproducibility of COP-values is in the acceptable range of 5%, even though it may be problematic to secure the exactness of these small mass flows. Moreover, it is doubtful, if a set of 23 tappings give a more realistic result considering the differences in user behaviour.

Extension of the testing for combined operating heat pumps
Testing of alternate combined operating heat pumps approved that the overall SPF can be calculated based on the test results of the SH-only and DHW-only mode. Therefore, for alternate operating systems, no further testing is required.
For simultaneous combined operating systems, additional testing is to be performed, since the heat pump characteristic may change significantly compared to the SH-only and DHW-only operation. Therefore, the simultaneous combined operation is tested by performing the DHW tapping cycle during the space heating operation. Due to the black box testing, however, only the total electrical energy input to the heat pump can be monitored, so the allocation of the electrical energy to the operation modes is not possible. So the combined operation is evaluated as own operation mode, since in the end, the overall seasonal performance is the relevant characteristic number.

Calculation method
The calculation method is built on an existing temperature class approach (bin method) already introduced in national standards. The principle of the bin method is depicted in Figure 4.
The cumulative annual frequency of the outdoor air temperature is divided into temperature classes (temperature bins). In the centre of each bin, an operating point is evaluated with regard to the heat pump operation at these specific ambient conditions, which is determined by the respective standard component testing, e.g. for SH according to EN 14511 and for DHW according to EN 255-3. The operating point is considered to characterise the heat pump operation of the whole bin. The areas of each bin correspond to heating degree hours based on the meteo data of the site and characterises the energy consumption in the bin. Thus, a weighting of the operating conditions with this energy fraction of the bin and a subsequent summation of all bins delivers the seasonal performance factor. Electrical back-up heaters can also be considered by an evaluation of the respective area in the cumulative frequency diagram, in Figure 4 the area BU.
The overall seasonal performance can be calculated by weighting the single seasonal performance factors with the annual energy needs in SH- and DHW-operation, respectively. Thus, the alternate combined operation can be covered based on existing test standards.

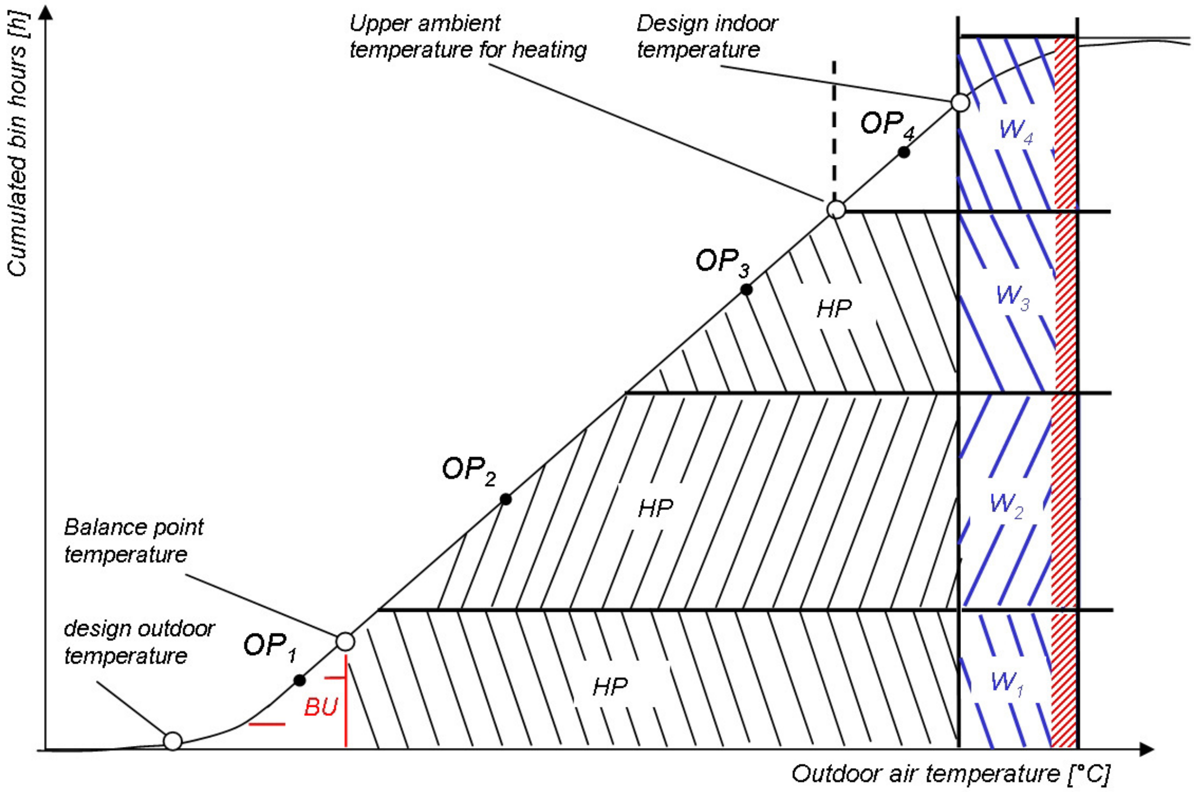

Figure 4. Principle of the bin method for alternate combined operation
(OP – operating point, BU – Back-up, HP – heat pump, W – DHW operation)

In simultaneous combined operation, however, the heat pump characteristic may change significantly. Thus, a third operation mode, the combined operation has been introduced based on the test procedure described above. The estimation, how much combined operation takes place in the bin is done by an evaluation of the running time of each mode, which is determined by the respective heating capacity of the heat pump and the energy requirement in the bin.

The overall seasonal performance can be calculated by the weighting of three operation modes, i.e. SH-only, DHW-only and combined operation.

DISCUSSION

Approximations of the calculation

Obviously, the calculation method implies some simplifications in order to keep the calculation simple. One shortcoming with regard to low and ultra-low energy house may be the redistribution of the energy requirement to the bins, that is only dependent on the heating degree hours and thereby on the outside temperature, while in low-energy dwellings the solar gains may have a higher impact on the energy distribution. This can be considered by an adjustment of the upper temperature limit for heating based on the quantity solar and internal gains as depicted in Figure 4.

Moreover, a shortcoming is that controller layout and controller setting can only be partly taken into account. In particular the energy production in combined operation may also depend on the controller settings for some system layouts. Controller settings for auxiliary components like pumps are often not known in detail and standard situation have to be evaluated. These effects can be taking into account by correction factors for typical control situations depending on the system configuration.

Comparison of calculated values to field monitoring results

To validate the calculation method with real data, field monitoring of pilot plants with combined operating heat pump systems were launched in the framework of the IEA HPP Annex 28. Even though not all types of system configuration could be monitored in field operation, the results of the comparison give a first impression of the exactness of the method.
Three direct expansion ground source heat pumps were evaluated for the SH-only operation. A ventilation compact unit with air-source heat pump and a ground source brine-to-water heat pump with alternate combined operation have been evaluated for the space heating and DHW operation. Differences between the calculation and the measured values are in the range of ±6% for the seasonal performance factor. Considering the simplifications in the approach these values are satisfactory and show the applicability of the method.

Implementation of results

Results of IEA HPP Annex 28 of have been continuously transferred to the respective standardisation working groups of the European standardisation committee CEN. The calculation approach has been implemented in the European draft standard prEN 15316-4.2 [11], which is part of the heat generator calculation standards used in the European Building Directive (EPBD) [1]. The SPF delivered by the calculation method is a basis for primary energy consumption or CO_2 emissions, respectively, which are displayed on the building energy certificate and determine the energy rating of the building.
For the implementation of the EPBD on the national level, Germany has introduced the method on a monthly basis in the pre-standard DIN V 18599 [12].
Results and recommendations for the test procedure are currently implemented in the CEN working group CEN/TC 113/WG 10 for the revision of the DHW test standard for heat pump EN 255-3. A working group committed to heat pump compact units incorporating also the ventilation is currently in constitution as a joint working group the ventilation CEN/TC 113 (Technical committee for heat pump testing) and CEN/TC 156 (ventilation systems). This gives the perspective that uniform testing of the heat pump and ventilation part of integrated ventilation compact units with heat pump can be achieved.
Concerning the international standardisation, the calculation approaches are quite similar, since the bin calculation is also introduced in ANSI/ASHRAE standards, but testing is still different due to different predominating system configurations on the market. Hence, to introduce internationally common standards, mainly the testing standards have to be further unified, i.e. on ISO level.

Information on IEA HPP Annex 28

Further information on IEA HPP Annex 28, workshop publications and related documents can be found on the Annex 28 website at http://www.annex28.net.
The final report of IEA HPP Annex 28 [13] is available at the IEA Heat Pump Centre and can be ordered via Internet on http://www.heatpumpcentre.org, category Publications/Reports.

ACKNOWLEDGEMENT

The authors would like to thank the Swiss Federal Office of Energy (SFOE) for funding and supporting the project.
It has to be emphasised that the IEA HPP Annex 28 is a team work and the results presented above are based on the effort and contribution of each member.
Hence, respect and thanks are expressed to all participants of the IEA HPP Annex 28 for the valuable contributions and for the constructive discussion and co-operation.

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Seasonal Performance and Exergy Analysis of Multi-Function Heat Pump Units

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SUMMARY

Overall energy rating of buildings including installed HVAC systems gets more important due to climate protection as demonstrated in building energy certificates presently introduced in the EU. Multifunctional system layouts for combined space heating, domestic hot water and ventilation, however, are often not covered by current product test standards and calculation methods. In this project consistent testing and calculation of multifunctional ventilation compact units with heat pumps has been elaborated based on present standard testing and calculation approaches. Different comparisons of calculated and field-monitored overall seasonal performance factor accomplished in Annex 28 show deviations in the range of ±6%.
Results are currently implemented in the respective testing and calculation CEN standards in the frame of the EU Directive on the Energy Performance of Buildings (EPBD) and related product testing.
Concerning component optimisation exergy analyses are a useful means, since different loss mechanisms can be quantified on a uniform basis.

INTRODUCTION

Due to reduced energy needs in modern highly-insulated and air-tight buildings, multifunctional system configurations have been introduced in the markets to cover different building energy needs for space heating (SH), domestic hot water (DHW) and ventilation with one unit. However, a system assessment requires standard test procedure and calculation methods, which are presently incomplete or missing for innovative multifunctional system layouts. The assessment is needed for different purposes, on the one hand for the holistic energy rating in the frame of the building energy certificate as currently introduced in the EU based on the Directive on the Energy Performance of Buildings (EPBD). On the other hand, transparent labelling concerning energy consumption/costs and environmental impact is needed to convince consumers to apply environmentally sound technologies.

METHODS

IEA HPP Annex 28 has been accomplished in 2003-2005 as co-normative research project in the frame of the Heat Pump Program (HPP) of the International Energy Agency (IEA). The nine countries AT, CA, CH, DE, FR, JP, NO, SE and US participated in the project. The scope and objective of Annex 28 were the development of

- *comprehensive test procedures* for combined operating heat pumps for SH and DHW with minimum testing requirements
- *subsequent transparent and easy-to-use calculation methods* for the SPF of combined operating heat pumps

as recommendations to standardisation organisations.

Thereby, two types of combined operating heat pump systems have to be differentiated

- *alternate combined operation*, where the heat pump is switched between SH- and DHW operation, and
- *simultaneous systems*, where SH and DHW energy is produced at the same time.

Since ventilation compact units with exhaust air heat pumps have become a standard heating system in ultra-low energy dwellings, they have been included in the scope, as well.
The accomplished work in Annex 28 comprises the

- testing of the systems according to existing and extended standards,
- calculation of the systems and
- field monitoring for the validation of the calculation results and to gain experience with the functionality for system optimisation.

Since test procedures for the SH-only, DHW-only and ventilation system already are in use, existing standards for the single operation modes have been extended to cover highly-integrated system configurations, as shown in Figure 1 and 3.

RESULTS

Results presented here refer to the Swiss national project in the IEA HPP Annex 28 [7] dedicated to the testing and calculation of heat pump compact units and field testing.

Test procedure for heat pump compact units

The testing of ventilation compact units with heat pump comprises

- Assessment of air-tightness (testing for internal and external air leakage)
- Testing ventilation
- Thermodynamic tests
- Acoustic tests
- Filter bypass leakage (optional)
- Operation/Maintenance/Safety (optional)
- Hygienic investigations (optional)

The following description refers to the thermodynamic testing as developed at HLKS of HTA Lucerne, the national Swiss test centre for residential ventilation systems. No European test standard for compact units exists, yet, but only test standards for the ventilation part and test standards for heat pumps for SH and DHW. The test procedure is based on the testing of German Institute of Building Technology (DIBt) and guidelines of the German Passive House Institute (PHI). However, the heat pump has been considered as core component of the compact unit and test points have been chosen based on the European heat pump test standards EN 14511 [2] for space heating mode and EN 255-3 [3] for DHW mode.
Since the operation conditions for the heat pump depend on the heat recovery a combined testing of the heat pump and the ventilation system is advantageous. Consequently, testing of the three basic operation modes deliver the respective characteristics of the component:

- ventilation-only operation (characteristic: temperature change coefficient, fan power)
- combined ventilation and heat pump in SH operation (COP, heating capacity SH)
- combined ventilation and heat pump in DHW operation (COP, heating capacity DHW)

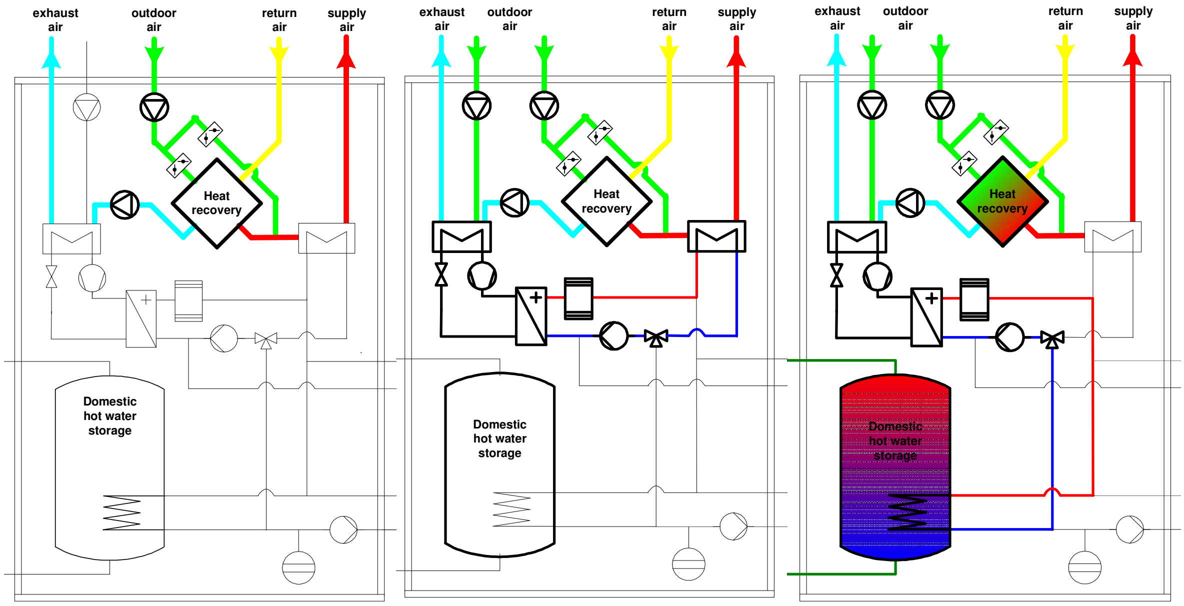

Figure 1. Operation modes for thermodynamic testing of compact units with internal circuit (left: ventilation only, middle: ventilation and SH, right: ventilation and DHW)

Figure 1 shows the different operation modes on the example of a heat pump compact unit with an internal circuit. The active circuit is depicted bold.
The component characteristic depends on the temperatures and the volume flow rates, thus test points of EN 14511 have to be tested for different air volume flow rates to be determined. To characterise the ventilation part including heat recovery unit, the temperature change coefficient and the electrothermal amplification factor are being determined.

Calculation method

The calculation of compact unit with heat pump is based on a temperature class approach (bin method) inline with the proposal for combined operating heat pump systems. Input data of the calculation method are the energy needs of the SH and DHW distribution system, test points of the unit according to the above test method, heating degree hours of the site, the system configuration and control details (e.g. heating curve, balance point).
The cumulative annual frequency of the outside temperature is divided into temperature classes (bins). In the centre of each bin, an operating point is evaluated, where the operation characteristic is known, i.e. at conditions defined in the heat pump testing standard. The operating point is considered to characterise the heat pump operation of the whole bin. The areas of each bin correspond to heating degree hours, which are proportional to the energy need in the bin. Thus, a weighting of COP values at the operating conditions with the energy fraction of the bin and a subsequent summation of all bins delivers the seasonal performance. Electrical back-up heaters and recovered heat by a ventilation heat recovery – only, if not already taken into account in the building energy calculation - can also be considered by an evaluation of the respective area in the cumulative frequency diagram.
For the domestic hot water operation, a similar calculation is performed based on standardised testing results of DHW-testing, e.g. for Europe according to EN 255-3 [3].
The overall seasonal performance can be calculated by weighting of respective operation modes, i.e. space heating-only, domestic hot water-only and combined operation.

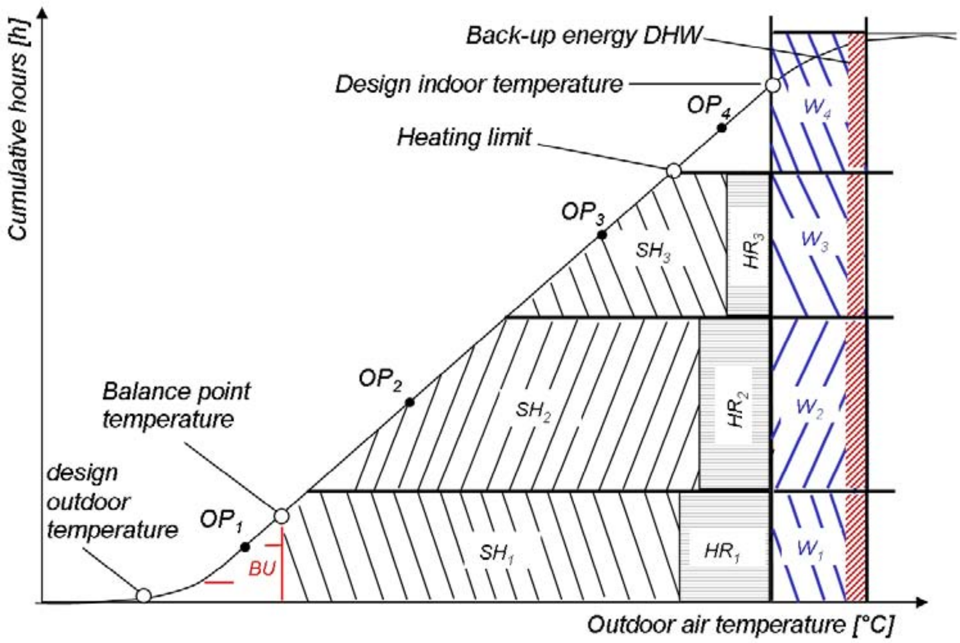

Figure 2. Principle of the bin method for multifunctional system for SH, DHW and ventilation with heat recovery unit (alternate mode) (OP - operating point, BU – back-up, SH – heat pump SH mode, W – heat pump DHW mode, HR – heat recovery unit)

Field monitoring

Calculation results have been compared to field testing results of a compact unit with air-source heat pumps installed in a single family low energy house according to the Swiss MINERGIE® standard. Figure 3 left shows the dwelling and a sketch of the principle system variant of system layouts of ventilation compact units with heat pump. In the investigated pilot plant, the compact unit works with a ventilation heat recovery and floor heating distribution system. A ground-to-air heat exchanger and solar collector are not installed.

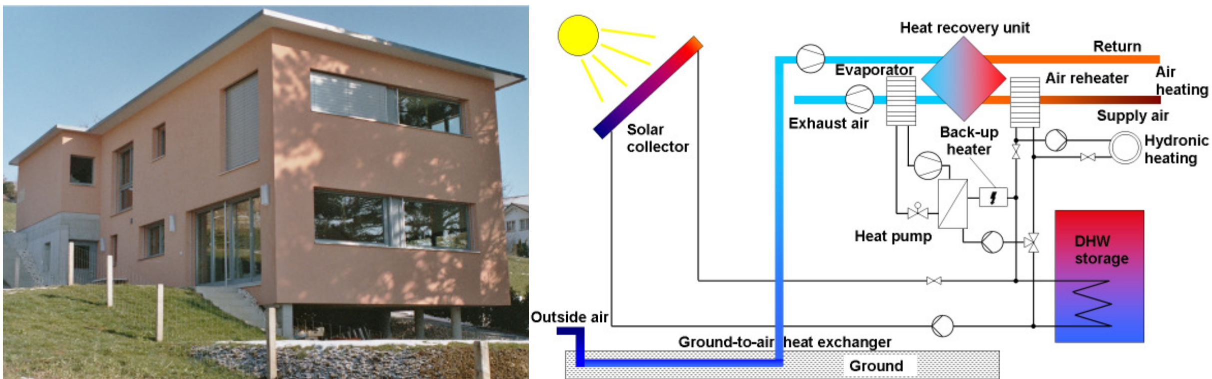

Figure 3. MINERGIE® dwelling for field testing and principle system layout of compact units

The evaluation period of the field testing is April, 26th 2004 to April, 25th 2005.
Figure 4 shows the annual electrical energy consumption of the different components of the compact unit (left) and heat energy consumption (right) for the different building needs. 75% of the electrical energy consumption is used as driving energy for the heat pump compressor, while 20% is used by auxiliaries and only 5% by the electrical back-up heater. Concerning the heat energy consumption, the remarkably low DHW heat energy consumption of only 9% explains the high fraction of SH energy of 83% despite the layout as low energy house.

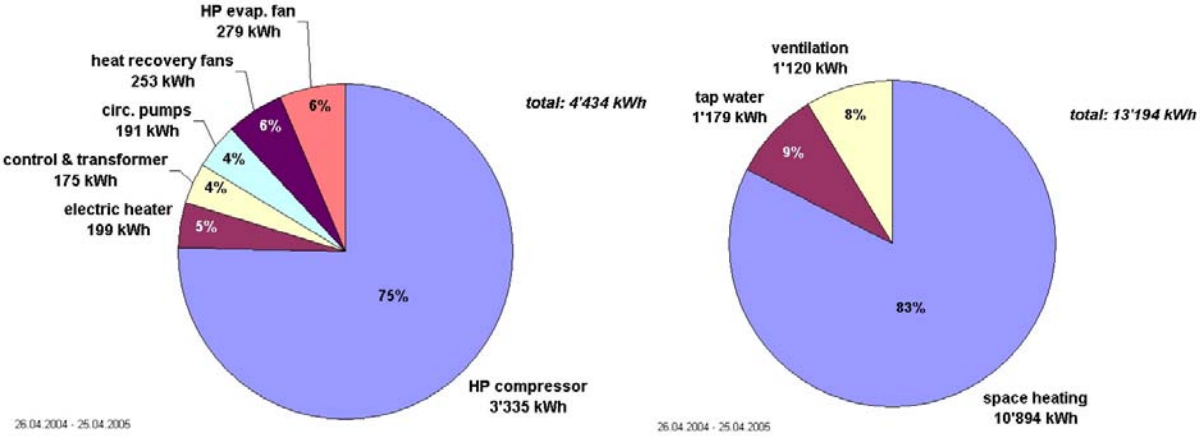

Figure 4. Annual electrical (left) and heat energy consumption (right) of the compact unit

Figure 5 presents the system boundaries used for the evaluation of performance characteristic number for the assessment of a ventilation compact unit with air-source heat pump. The same numbers have been calculated with the calculation method.

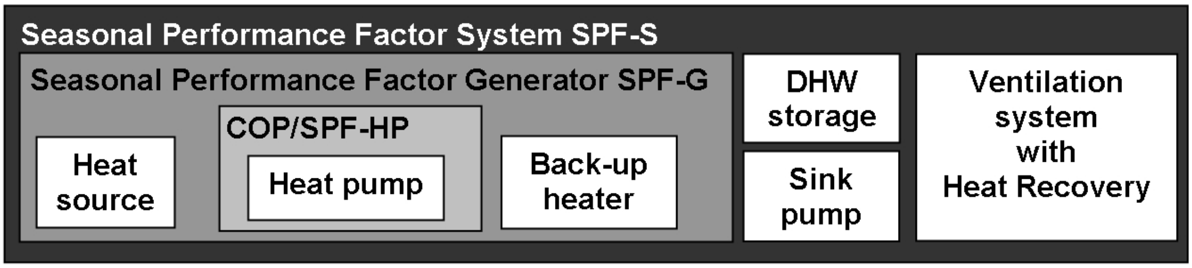

Figure 5. System boundaries for the assessment of the monitored compact unit

The SPF-HP comprises the system boundary according to the European heat pump testing. The Generator SPF (SPF-G) is the ratio of the produced energy of all generators (heat pump, back-up) to the respective electrical energy input and is well suited for the comparison to other heat generators like boilers. The System SPF (SPF-S) is related to the energy need and is calculated as ratio between the used energy (incl. the recovered ventilation energy) and the total electrical energy input to the entire system (incl. auxiliaries and electrical input to the ventilation). Results of the overall SPF for the system boundaries are depicted in Table 1.

Table 1. Monitoring and calculation overall SPF results for a ventilation compact unit with air-source heat pump installed in a low-energy house acc. to Swiss MINERGIE® standard.

	Monitoring (Reference)	**Calculation**	**Difference**
SPF-HP	3.66	3.54	-3.3%
SPF-G	3.42	3.42	0.0 %
SPF-S	3.06	2.96	-3.2%

Exergy analysis

Exergy is defined as the fraction of the energy that can be arbitrarily converted in each of the different energy forms. In all natural processes, exergy is converted to anergy, e.g. by irreversible heat transfer due to finite temperature differences or pressure drop due to friction, which corresponds to an exergy loss. Therefore, the exergy content is a measure of the thermodynamic quality of an energy flow or a process.

An exergetic consideration has been applied to characterise the impact of the temperature requirements of different source and sink systems used in compact units (cf. Figure 3 right) on the performance of a ventilation compact unit.

Pure air heating distribution systems have the limitation that the maximum temperature is restricted to about 50°C. Moreover, to avoid excessive exergy input to the fans, common designs limit the air volume flow to the hygienic necessary. Thus, air heating systems can only be applied in ultra low energy dwellings due to the limited heat capacity of the air and require high supply temperature up to the limit of about 50°C in colder winter periods.
Floor heating systems, on the other hand, have a large surface for the heat emission in the room, whereby supply temperatures below 30°C can be realised. However, additional exergy input for the circulation pump to transport the heat transfer medium is required.
The exergy comparison has been accomplished for an ultra-low energy house under the simplifying assumptions of adiabatic components, negligence of defrosting operation and a constant internal exergetic efficiency of the refrigerant process of ζ=0.5 (exergy losses of compression, refrigerant pipe, heat exchanger friction, expansion) to focus on the exergy losses linked to the source/ sink system periphery. Table 2 shows the boundary conditions used.

Table 2. Boundary conditions and resulting source and sink temperature (HR – heat recovery)

Building	
Ambient temperature [°C]/Indoor design temperature [°C]	-4 / 20
Heat load for the heat pump at ambient temperature [W]	1200
Ventilation system	
Volume flow rate ventilation system [m^3/h]	110
Temperature change coefficient (considered the same for supply and return air path) [-]	0.8
Pressure drop HR, condenser and evaporator [Pa] / air channels (each supply/return [Pa]	80 / 40
Constant fan power [W]/ Pressure drop dependent fan power [W/Pa]	10/ 0.12
Heat pump	
Pinch point condenser air heating [K]/floor heating [K]/Pinch point evaporator both[K]	5 / 2 / 5
Required supply temperature air / floor heating at ambient conditions [°C]	48 / 29
Condensation temperature / Evaporation temperature of the air heating system [°C]	48 / -21
Condensation temperature / Evaporation temperature of the floor heating system [°C]	30/ -23
Pump power of the circulation pump [W]	40

Since the COP of the heat pump can be expressed by the equation

$$\mathrm{COP} = \varsigma \cdot \mathrm{COP_C} = \varsigma \cdot \frac{T_{cond}}{(T_{cond} - T_{evap})}, \qquad (1)$$

where COP_C denotes the Carnot COP, ζ the constant internal exergetic efficiency , T_{cond} the thermodynamic condensation and T_{evap} the thermodynamic evaporation temperature, the COP is proportional to the Carnot COP and thus to the respective temperature levels.
Resulting exergy losses of the air heating are 14% higher than of the floor heating system despite the exergy input to the circulation pump and the evaporation temperature decrease due to the higher anergy extraction in the evaporator in case of the floor heating system, cf. Table 2.
Figure 6 depicts the range of temperature levels in the common components of the compact unit with common source and sink systems. The bold red arrows follow the way of the heat transfer media through the compact unit. Starting at ambient conditions the heat recovery unit raises the exhaust air temperature above ambient level, while in the evaporator it is cooled down by the refrigerant. The evaporation temperature level can be increased by a mix with an additional outdoor air flow or by a preheating of the air in a ground-to-air heat exchanger, which is often installed in order to avoid frosting of the heat recovery heat exchanger. The heat pump makes the main temperature lift from the source temperature level to the required distribution temperature level, and in the emission system the temperature level is reduced first to the room temperature, and then by the heat losses of the room to the ambient level.

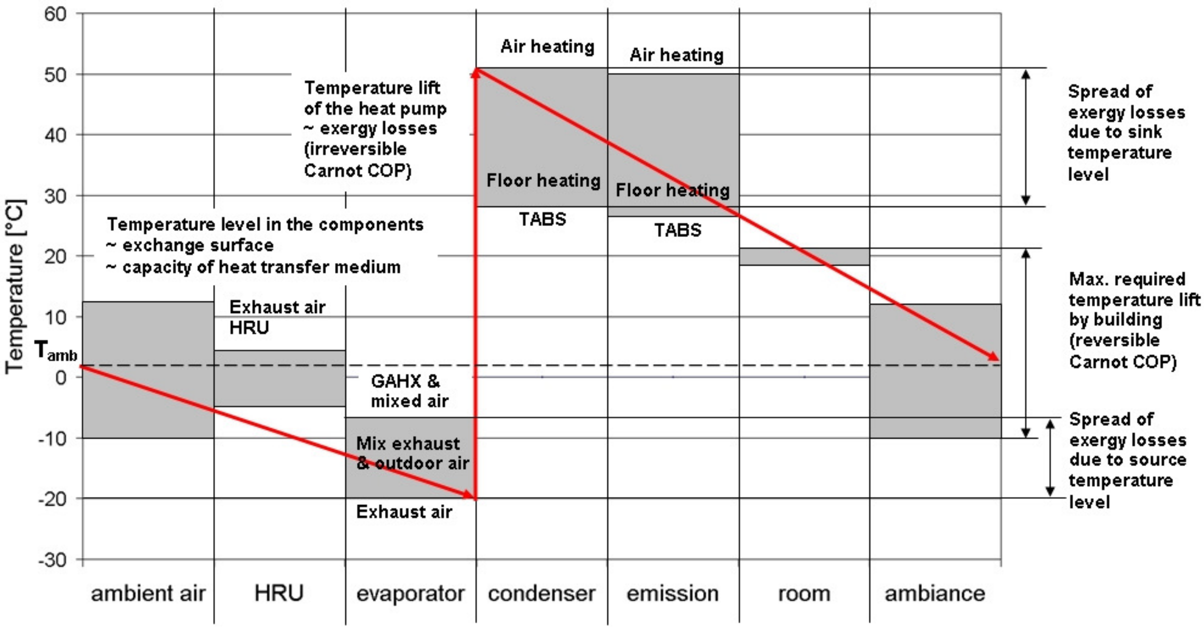

Figure 6. Temperature levels in the single components for compact unit (GAHX – ground-to-air heat exchanger, HRU – Heat recovery unit, TABS – Thermally activated building structures, COP – Coefficient of performance)

The temperature levels and corresponding exergy losses depend on the design of the heat exchangers, since larger heat exchanger surfaces reduce the heat transfer resistance and lead to lower temperature differences for the heat transfer. On the other hand, the temperature levels are also defined by the capacity of the source and the sink heat transfer medium. By the choice of hygienic necessary ventilation air as heating medium, for instance, high temperature spreads are required based to the limited heat transport capacity due to restricted volume flow rate and the heat capacity of the air.

The theoretical minimum required temperature lift for a reversible heat transfer is defined by ambient and indoor temperature (reversible Carnot COP). All limitations of the heat transfer system periphery thus have to be compensated by additional exergy input to the heat pump compressor due to the elevated temperature lift. On the sink side the maximum spread is seen between an air heating and TABS emission system, on the source side between a system with only ventilation exhaust air and a preheating of a higher outside air flow.

The general advantage of exergy analysis is, that the different loss mechanisms, e.g. due to heat transfer or pressure drop, can be quantified on the same basis as exergy loss. However, an exergy consideration gives only the technical viewpoint, and economical consideration may result in a different assessment. Last but not least environmental and comfort issues have to be considered, as well.

DISCUSSION

Simplification in the calculation approach

The calculation method implies the assumption of the redistribution of the energy according to heating degree hours which are only dependent on the outdoor air temperature. The impact of solar and internal gains on the energy redistribution can be approximated by an adjustment of the heating limit based on the quantity of used gains as depicted in Figure 2.

Moreover, controller impact cannot be considered in detail, since often, details are not known. Therefore, controller impact is taken into account by standard settings depending on the system layout, e.g. for the running time of auxiliaries.

Implementation of results

Results of IEA HPP Annex 28 for the heat pump calculation method has been implemented in the European draft standard prEN 15316-4.2 [4] used in the EPBD [1]. As national implementation of the EPBD, Germany has introduced the method on a monthly basis in DIN V 18599 [5].

A working group committed to testing of ventilation compact units with heat pump is presently in constitution as a joint working group of the Technical Committees CEN/TC 113 (heat pump testing) and CEN/TC 156 (ventilation systems). This gives the perspective of uniform testing of heat pump and ventilation part of compact units with heat pump.

Component and system optimisation

Exergy analyses are well suited for a component and process optimisation, since the impact of different loss mechanisms can be quantified on the uniform basis of exergy losses. The exergy analysis of the compact unit shows that any increase of temperature lift by the layout of the source and sink system has to be supplied as additional exergy to the heat pump. Therefore, system layout guaranteeing high source temperature and low sink temperature level lead to a significant reduction of the exergy losses and thus reduces the electrical energy input.

Information on IEA HPP Annex 28

Further information on IEA HPP Annex 28 can be found on the Annex 28 website http://www.annex28.net.

The final report of IEA HPP Annex 28 [6] is available at the IEA Heat Pump Centre. It can be ordered on http://www.heatpumpcentre.org, category Publications/ Reports.

ACKNOWLEDGEMENT

The Swiss national project [7] and the IEA HPP Annex 28 [6] have been funded by the Swiss Federal Office of Energy (SFOE).

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Model of a Reversible Heat Pump for Part Load Energy Based Optimization Design

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SUMMARY

With the increase of air conditioning needs in commercial sectors, reversible heat pumps represent a smart solution for energy consumption.
Design and sizing of reversible air conditioning / heating systems are often based on the full load performance, in order to respect thermal comfort at extreme conditions. However, the HVAC industry is switching to designs based on improved seasonal performance at part load.

In this context, this article represents a modeling of a 60 kW reversible heat pump. Detailed models of each component of the system have been developed: a plate heat exchanger (R410A/water), finned tube heat exchanger (R410A /air), two parallel scroll compressors, and an expansion valve. The model simulates the whole cycle after simulating each component separately. Results obtained are validated through tests done on a reference bench machine. Then simulations covering all the climatic conditions and regions of France for a whole year are achieved, helping to model a design of a new reversible heat pump energetically optimized at part load.
This paper will describe the main elements of the model, which has an objective to reduce the annual energy consumption of the reference commercial reversible heat pump by 50%.

INTRODUCTION

Heat pumps have got great interest during the last two decades, due to the increasing attention of the international community towards global environment and energy savings. Therefore, heat pumps are promising efficient solution for buildings air conditioning / heating compared to conventional devices, especially now when electricity production has reached efficiency about 60%.

The paper presents a model developed to assume the steady state characteristics of a commercial reversible air-to-water heat pump. In cooling mode, heat source is the ambient air and heat sink is the water cycle inside the building. Conversely, in heating mode, heat sink is the ambient air and heat source is the water cycle inside the building. Results of the simulations are validated by experimental measures in climatic chamber. Initial validation allows simulating several cases by changing the model parameters, and extrapolating their results to annual conditions in order to design an annually optimized solution. This is coherent with the tendency to introduce energy performance of buildings and HVAC systems not only based on rating conditions but also extended to whole annual building load. This means to calculate a seasonal energy efficiency ratio SEER and seasonal coefficient of performance SCOP.

MODEL DESCRIPTION

The structure of the global model is an image of the vapor compression refrigeration cycle. It is divided into modules representing the different cycle stages for different configurations.

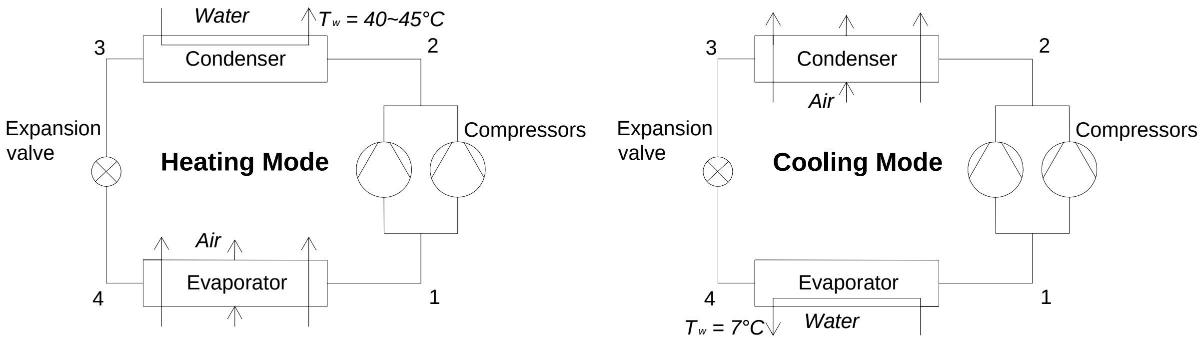
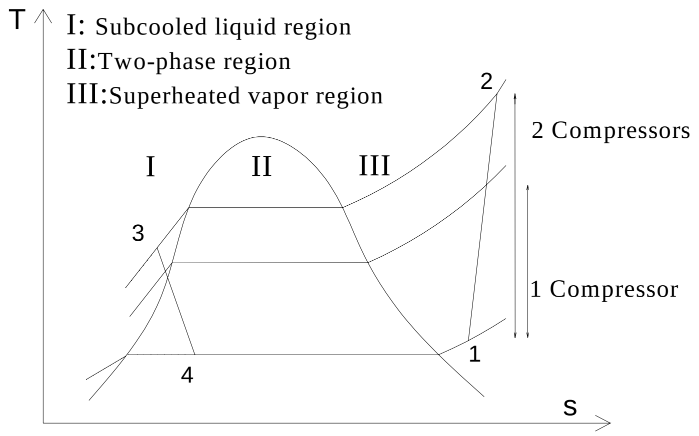


Figure 1: Layout of the vapor compression refrigeration system in heating and cooling modes



1—2 Compression
2—3 Condensation
3—4 Expansion
4—1 Evaporation.

Figure 2: R410A Vapor compression cycle on T-s diagram

Each module takes its inlet properties from the precedent module, calculates its outlet properties and transmits them to the next one. Details of the equations used are explained below.

The refrigerant used in this study is the R410A (mass fraction: 50% R32 and 50% R125). Calculation of the refrigerant, air, and water thermodynamic and transport properties is performed by correlations formulas written as computer code functions, using thermodynamic properties coming from REFPROP [1] data base. These functions are grouped in a particular module included in the overall source code.

Compressor model

A scroll compressor model is developed from the data given by the manufacturer and measurements carried out in climatic chamber on the whole heat pump. Performance curves are given using two equations fitting the compressor energy suction and the mass flow rate as a function of the evaporating temperature T_e and the condensing temperature T_c,

$$W_{comp} = C_0 + C_1.T_e + C_2.T_c + C_3.T_e.T_c + C_4.T_e^2 + C_5.T_c^2 + C_6.T_e^3 + C_7.T_e^2.T_c + C_8.T_e.T_c^2 + C_9.T_c^3 \quad (1)$$

$$\dot{m} = M_0 + M_1.T_e + M_2.T_c + M_3.T_e.T_c + M_4.T_e^2 + M_5.T_c^2 + M_6.T_e^3 + M_7.T_e^2.T_c + M_8.T_e.T_c^2 + M_9.T_c^3 \quad (2)$$

where C_1- C_9 and M_1- M_9 are constant coefficients.

Compressor calculating module inputs and outputs are:

Inputs		Outputs
Evaporating Temperature Te $[K]$ →	Eq (1)	→ $\dot{W}_{comp}[W]$
Condensing Temperature Tc $[K]$ →	Eq (2)	→ $\dot{m}_{comp}[kg/s]$
$\eta_{global}, \eta_{vol}$ →		→ $(T, h)_{discharge}$

Compressor isentropic efficiency expresses the ratio between the isentropic and real compression work:

$$\eta_{is} = \frac{\dot{W}_{is}}{\dot{W}_{real}} \quad (3)$$

where $\dot{W}_{is} = \dot{m}.(h_{2is} - h_1)$ is the real energy rate absorbed by the refrigerant during compression. Isentropic efficiency is needed to calculate the fluid properties (enthalpy, temperature…) at the compressor outlet. Compressor mechanical efficiency, assumed to be constant, relates real energy rate $\dot{W}_{real}$ transmitted to the refrigerant to power $\dot{W}_{comp}$

$$\eta_{mec} = \frac{\dot{W}_{real}}{\dot{W}_{comp}} \quad (4)$$

Compressor global efficiency expression

$$\eta_{global} = \frac{\dot{W}_{is}}{\dot{W}_{comp}} = \eta_{is} \times \eta_{mec} \quad (5)$$

Compressor volumetric efficiency

$$\eta_{vol} = \frac{V_{real}}{V_{th}} = \frac{v_{asp}.\dot{m}}{V_{th}} \quad (6)$$

where v_{asp} is fluid specific volume at the compressor aspiration [m^3/kg],

V_{th} is displacement volume given by manufacturer [m^3].

Expansion device model

A thermostatic expansion valve (TXV) is considered in the model. TXV is used as rate flow control device, which feeds back the superheat ΔT_{SH}, and adjusts the mass flow to the evaporator in order to maintain a constant superheat preventing liquid refrigerant from entering the compressor.

Inputs are the inlet enthalpy and pressure, the outlet pressure and the superheat ΔT_{SH}. The expansion is assumed to be adiabatic procedure without heat exchange neither work drawn or produced, thus it is an isenthalpic expansion: $h_4 = h_3$.

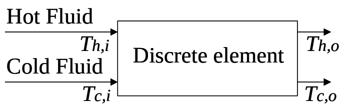
Mass flow rate governing equation depends from the difference of pressure between the condenser and the evaporator $\Delta P = P_c - P_e$, and from the superheat ΔT_{SH}:

$$\dot{m}_{TXV} = k.\Delta T_{SH}.\sqrt{\rho.\Delta P} \quad (7)$$

where k is a coefficient specific to the expansion valve considered.

Heat exchangers models

The air-to-water heat pump studied in this article involves two heat exchangers (HEs): a plate heat exchanger (R410A/water), and a finned tube heat exchanger (R410A/air). These exchangers play the role of condensing or evaporating unit according to the case of cooling or heating mode.
Three types or levels of detail exist for modeling HE depending on the required accuracy on the results: global, zonal or finite elements. In this work, HR are modeled through zonal for every type of HE wherever its working mode: parallel flow, counter flow, cross flow, in a sequence of superheat – condensation – subcooling or evaporation – superheat. However, due to space limitations, only heating mode will be considered to illustrate the different heat exchange stages of the condenser and the evaporator and the calculation procedure.
Discrete elements considered calculate the outlet temperatures $(T_{h,o}, T_{c,o})$, starting by knowing



their inlet temperatures $(T_{h,i}, T_{c,i})$. The calculation is based on LMTD approach:

$$Q = F \cdot UA \cdot LMTD \tag{8}$$

where -Q is the heat flux exchanged [W].
-UA is global heat transfer coefficient [W/K], got from the correlations applied on both sides of the heat exchange process.
- F coefficient for flow type.
-LMTD is the log mean temperature difference defined by:

for parallel flow HE: $$LMTD = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} = \frac{(T_{h,i} - T_{c,i}) - (T_{h,o} - T_{c,o})}{\ln\left(\frac{T_{h,i} - T_{c,i}}{T_{h,o} - T_{c,o}}\right)} \tag{9}$$

for counter flow HE: $$LMTD = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} = \frac{(T_{h,i} - T_{c,o}) - (T_{h,o} - T_{c,i})}{\ln\left(\frac{T_{h,i} - T_{c,o}}{T_{h,o} - T_{c,i}}\right)} \tag{10}$$

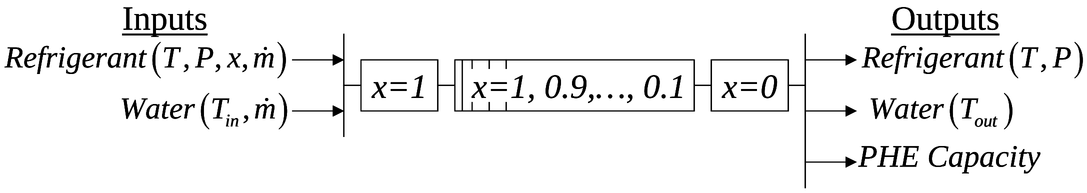
Another expression for the heat flux exchanged between the two fluids:

$$Q = C_c (T_{c,o} - T_{c,i}) = C_h (T_{h,i} - T_{h,o}) \tag{11}$$

where $C = c \cdot \dot{m}$, c is the specific heat [kJ/(kg·K)].
The procedure to solve the set of equations is as follows. An estimate of one of the outlet temperature $T_{c,o}$ or $T_{h,o}$ is taken as initial guess. Then the other outlet temperature and heat flux Q are found with equation (11). Next, a new value of Q is determined using equations (8). If the newly calculated heat flux value matches the previous one, the element computation is complete. Otherwise, Q becomes the new estimate of equation (11) and another computational sequence is performed.

<u>Plate heat exchanger</u>
Since the heat sink in heating mode is the water cycle inside the building, the plate heat exchanger PHE will play the role of condensing unit. During the condensing process, refrigerant passes through three regions: superheated vapor, condensation and subcooled liquid (figure 2). Therefore, modeling of the PHE differs between these three regions through refrigerant quality *x*. Indeed, PHE is divided to superheated zone where *x=1*, subcooled zone *x=0*, and a two-phase zone subdivided to 10 elements where vapor quality *x* is decremented by a value of 0.1 from 1 to 0.

Condenser model inputs and outputs are:



The specified geometry of the PHE includes the number of plates, plate spacing, length and width of the HE. Water side heat transfer coefficient was first taken from the single-phase water to water pre-tests done by Kim [5, 6]. First results lead to use a simple correlation function of Reynolds number and Prandtl number:

$$Nu = 0.386\,\mathrm{Re}^{0.623}\,\mathrm{Pr}^{0.369} \tag{12}$$

where Nusselt number is equal $Nu = \dfrac{h_w \cdot D_h}{k_w}$, k_w represents water thermal conductivity, and D_h hydraulic diameter.

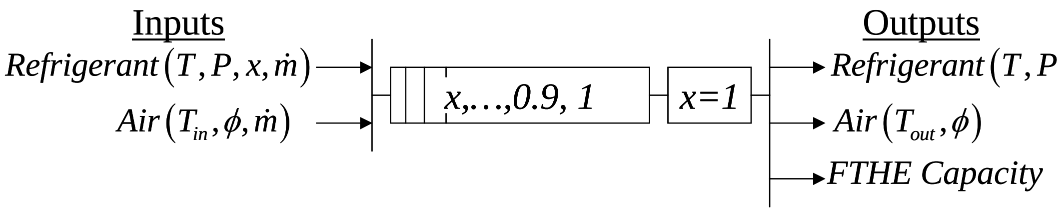
Heat transfer for refrigerant side in PHE [7] depends on refrigerant state: for single-phase flow most correlations that have been proposed by literature are of the form:

$$Nu = C\,\mathrm{Re}^{n}\,\mathrm{Pr}^{m} \tag{13}$$

while for boiling flow, correlations are more complicated like Cooper boiling correlation [6]. Thus, the model uses correlations similar to equation (13) after identifying its parameters by measurements. The parameters C, m and vary with the vapor quality x.

<u>Finned tube heat exchanger</u>

In heating mode, the finned tube heat exchanger FTHE is considered as evaporating unit. As shown in figure 2, refrigerant passes through two regions: evaporation and superheated vapor. Evaporator model inputs and outputs are:



The geometry of the FTHE is specified also, it includes the number of tubes heightwise *N*, the number of tube rows in the direction of the air flow plate *n*, the tube spacing in the direction of air flow S_L and in normal to the direction of air flow S_T, tube inside and outside diameter, spacing between fins *s*, tube length *l*, HE height. Some of these dimensional parameters are shown on figure 3, which is a side cutaway view of the FTHE.

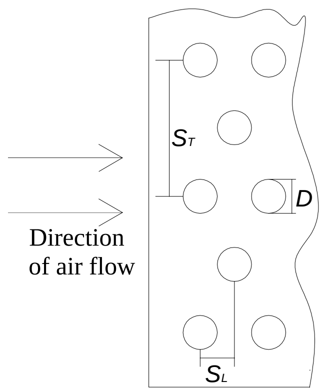


Figure 3: FTHE side cutaway view

Convergence criteria - Main computational loop

First, prior to cycle calculation, initial conditions (air, water) must be specified. Besides, a degree of subcooling must also be specified. Using the ambient temperature, the program generates its initial estimate of evaporating and condensing pressures. A flow chart outlines the model structure and the order of computation in figure 4. The subcooling is compared to the required one. According to the difference between the two values, condensation pressure is adjusted. Then, the mass flow rate calculated by the expansion device model is compared to the mass flow rate obtained by the compressor model. According to the difference, evaporation pressure is adjusted.

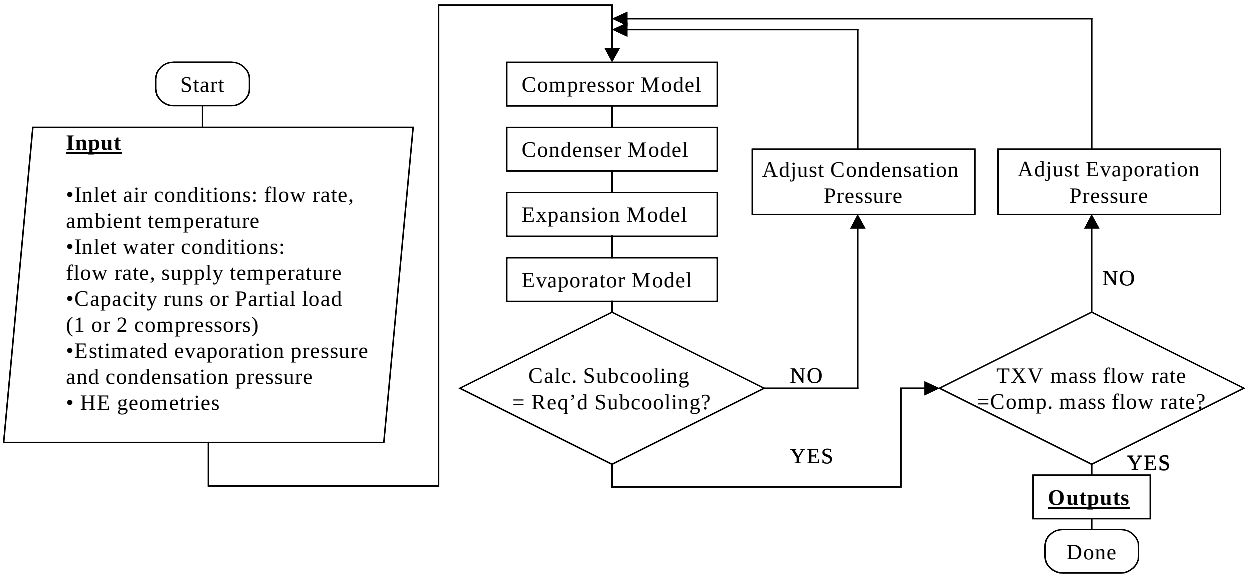
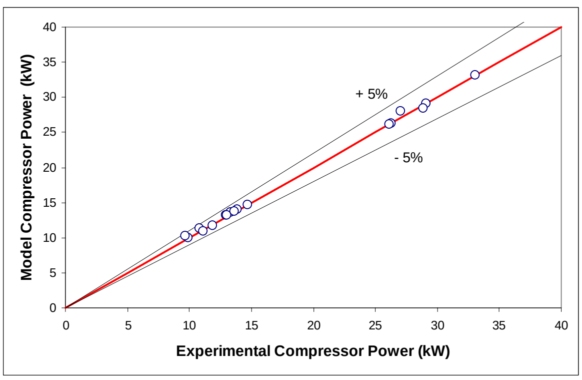


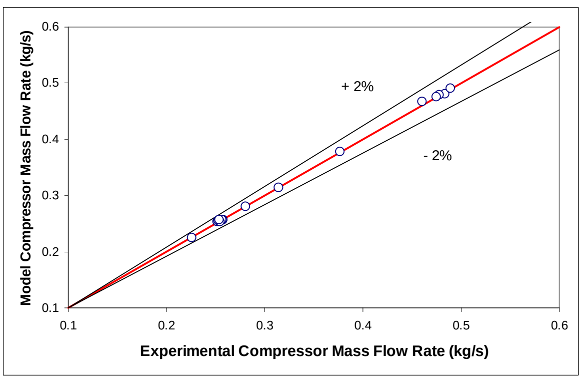
Figure 4: Overall model flowchart

MODEL VALIDATION

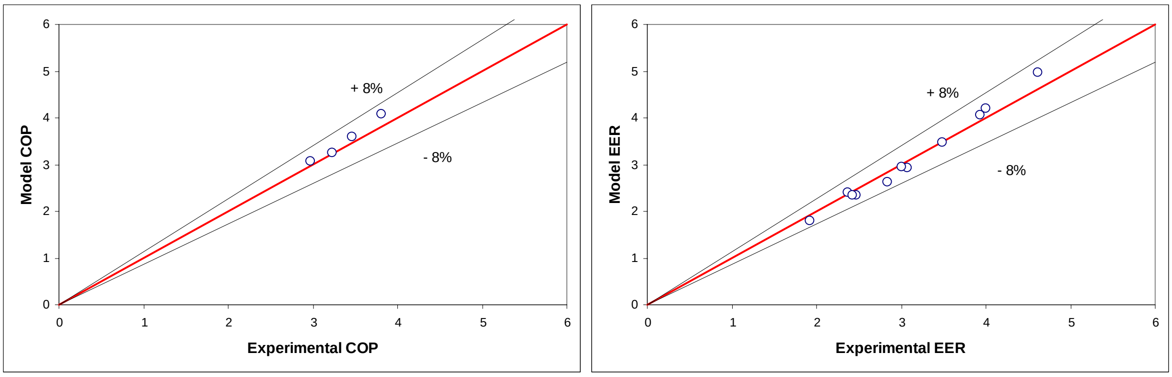
Results obtained by the program are validated through tests done on a reference heat pump machine. This test machine is mounted in a climatic chamber where ambient air is maintained at specified conditions of temperature and humidity. Moreover, compensation cycles are conceived to consume the hot and chilled water produced by the system.
The main components of the test machine are: PHE with 64 plates, 2 scroll compressors, FTHE with 24 circuit 3 tube rows and louvered fins, and two thermostatic expansion valves. Temperature and pressure measurement instruments are well distributed on the inlet and outlet of the heat pump components. Compressor mass flow rate is also measured. In addition, flowmeter and air velocity measurement devices are used to determine water and air flow rates respectively.



a)



b)



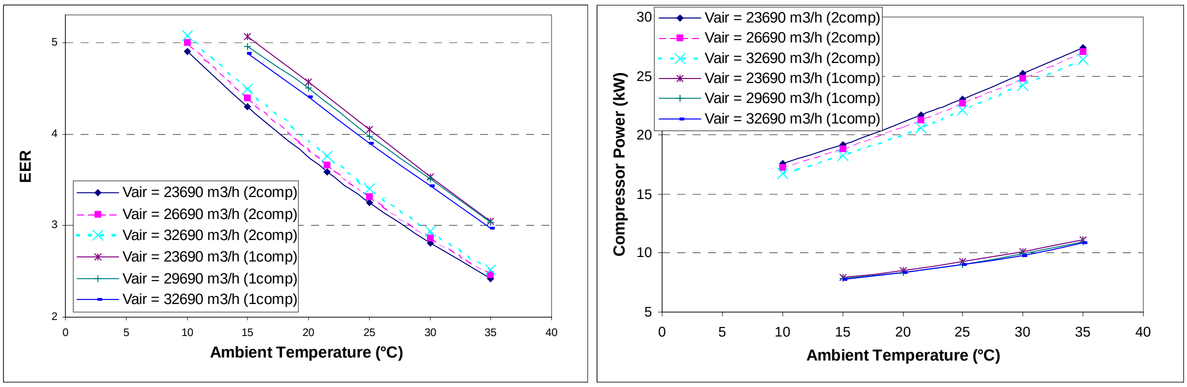
c) d)

Figure 5: Comparison between model and experiments results

Figure 5 shows comparison between the model and the experiments results. The comparison covers partial load and full load charge for a range of ambient temperature. The first plot (5.a) shows results for the power required by the compressor, which is predicted within 5% of error. Compressor mass flow rate (5.b) is perfectly computed by the model with a difference of 2%. COP and EER are predicted within 8% of error.

EXAMPLE OF RESULTS

As an example of the simulations that could be performed, the following figure shows effect of increasing air flow on the FTHE working as a condenser in cooling mode. The simulation was completed for partial load (1 compressor) and full load (2 compressors), it covers a range of temperature from 35°C to 10°C. A benefit of around 7% is noticed at full load and 3% at partial load (6.a), this is due to the decrease of condensation pressure which reduces the power required by the compressor (6.b).



a) b)

Figure 6: Effect of increasing air flow rate on EER and compressor power in cooling mode

In addition, results of simulations serve as tool for evaluating seasonal loads. Figure 7 illustrates reduced load curve for a commercial building in Nice (south east France) function of compression ratio, load and weighting factor. Weighting factor is a percentage expressing the number of functioning hours annually. The curve is divided between partial load 50% (one compressor) and full load 100% (two compressors).

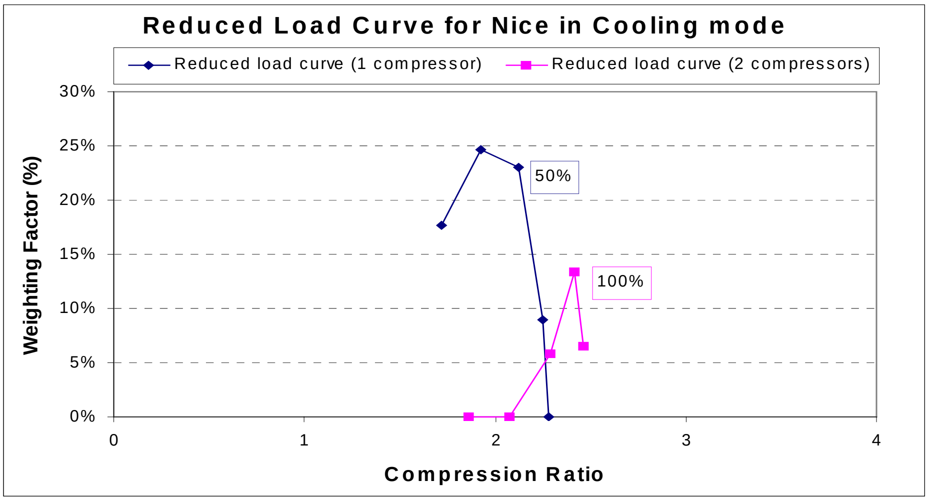


Figure 7: Reduced load curve

CONCLUSION

A vapor compression cycle model has been developed to simulate the performance of an air-to-water R410A based reversible heat pump. The global model is based on first principles and thermodynamics. It includes detailed submodels of the system components: compressor, heat exchangers, and expansion device. The model was successfully tested on a real prototype, it predicts characteristics of the cycle with only a difference of about 5% with the experimental measures. It is able to calculate a complete cycle in a few seconds.

The coupling with the building simulations enables to optimize the reversible when energy consumption has the larger impact, part load and reduced temperature as compared to standard conditions.

Further work will consist in studying different optimization paths for different building and climate conditions.

ACKNOWLEDGEMENT

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Integrated HVAC System with Direct Expansion Ground Source Heat Pumps

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SUMMARY

The electrical heating in residential sector may be optimized by using mono-fluid heat pumps with direct expansion ground heat exchangers and direct condensing through radiant floors. Such systems are simple and efficient because eliminate secondary fluids (brine, air), circulation pumps, air handlers and fans. This paper presents some design criteria and energy performances of a residential, integrated HVAC system with three direct expansion ground heat pumps and exhaust air heat recovery device. Are presented the main results obtained in a 2-story house during a typical Canadian cold climate heating season. The system seasonal energy performance depends on soil properties, ground heat exchangers depth of burial, and outdoor temperatures and precipitations. The seasonal coefficients of performance of DX ground heat pumps averaged 3.8, while the temperatures of the concrete slab (basement) and wooden floors (1st and 2nd stories) varied around 26°**C** ensuring homogeneous vertical thermal stratification and highly indoor comfort. Improvements of refrigerant circuits and controls allowed avoiding instability phenomenon and improving overall performances and reliability of the integrated HVAC system with direct expansion ground source heat pumps.

INTRODUCTION

Electricity usage for housing heating can be optimized by using ground-source heat pumps (GSHP). The ground is an inexhaustible energy source since it absorbs nearly 46% of the incident solar energy, as well as heat from natural precipitations (rain, melting snow). The well-known indirect GSHP systems use vertical or horizontal ground heat exchangers and water/antifreeze mixtures as geothermal fluid. In spite of their high energy efficiency and contribution to greenhouse gas emissions, the main disadvantages of these systems are relatively high drilling costs, additional energy consumption for geothermal fluid pumping and energy efficiency losses due to thermodynamic irreversibility of intermediate heat exchangers. On the other hand, the direct expansion (DX) heat pumps system with direct condensing inside radiant floors eliminates secondary fluids as brines and air, circulating pumps, air coils and fans. Such geothermal heat pumps have been designed and in-field tested in Eastern Canada where the average solar insulation vary from 50 kWh/m^2 in January to about 180 kWh/m^2 in July and the total heating Degree-Days averages 5 211. The residential system integrates three heat pumps with horizontal DX ground heat exchangers, direct condensing radiant floors for space heating, a central air-conditioning unit with exhaust air heat recovery heat exchanger and a 4-kW electrical back-up heater. The study aimed at determining the influence of soil properties, ground heat exchanger depth of burial and local cold weather conditions on such integrated residential heating/air cooling systems.

SYSTEM DESCRIPTION

A high-comfort 1100-m^2 (total surface area) house with half-buried basement and two stories was equipped with two DX ground-source heat pumps and one make-up air HVAC unit. The 2-story house, located along the Saint Lawrence River (Eastern Canada), is well insulated but exposed to prevailing winds. The nominal heating load, calculated for 21°C indoor and -29°C outdoor temperatures, was estimated at 20 kW of which approximately 12 kW for the basement and the 1st floor, and 8 kW for the 2nd floor. A 4-kW back-up electrical coil provides additional heat during the peak demand periods. The first mono-fluid ground-source heat pump (GSHP #1) supplies heat to the basement and the 1st floor (734 m^2 in total surface) (Figure 1). The horizontal ground-source heat exchanger #1, buried at 0.9 m depth, contains eight refrigerant loops with ½" copper tubes coated with a thin plastic layer and 720 m in total length. Up to 0.8 m depth the soil is sandy and humid, and the ground exchanger is buried at 0.9 m depth in a disaggregate schist soil mixed with a few damp clay. The nominal heating capacity (18.5-kW) of GSHP #1 is dissipated through two parallel, direct condensing heat exchangers. On the basement, the first condenser (260 m of ¾" copper tubes) is inserted inside a 100-mm concrete radiant floor thermally insulated. On the 1st story, the second condenser (350 m of ¾" copper tubes) is installed under a traditional wooden floor, thermally insulated. Precautions were taken to enhance the heat transfer between the refrigerant piping and the wood panels. Each parallel refrigerant circuit contains motorized (SV) and check (CV) valves allowing supplying heat according to each zone heating demand. The second mono-fluid ground-source heat pump (GSHP #2) supplies heat to the 2nd floor (366 m^2). The horizontal ground-source heat exchanger #2, also buried at 0.9 m depth, contains four refrigerant loops with ½" copper tubes and 360 m in total length. The heat pump's nominal heating capacity (7.5-kW) is dissipated through a direct condensing heat exchanger (350 m of ¾" copper tubes) installed under a traditional wooden floor, thermally insulated. The make-up air HVAC unit includes a heat-pipe exhaust air heat recovery heat exchanger with variable speed fans, and a direct expansion, reversible, tri-function ground source heat pump (GSHP #3) with 2.6 kW as nominal cooling capacity. The horizontal DX ground heat exchanger #3 (180 m of ½" copper tubes) is buried at 0.9 m depth. In cooling and dehumidification mode, the "cold" heat exchanger acts as an evaporator and the ground heat exchanger as a condenser. In dehumidification and heating mode, the "cold" heat exchanger operates as an evaporator and the "hot" heat exchanger as a condenser. Finally, in heating mode, the DX ground heat exchanger becomes evaporator and the "hot" heat exchanger - condenser. The electrical humidifier and a 4-kW back-up heating coil respectively provide make-up air humidification to keep the indoor air relative humidity around 50%, and additional preheating during extreme weather conditions in winter.

GROUND DX HEAT EXCHANGERS

All horizontal DX heat exchangers are buried in a compacted gravely, siliceous and damp sand, with little clay content and average thermal conductivity of 1.29W/m.K (Figure 2). The temperature of the disturbed soil was monitored at 150 mm lateral distance from the ID ½" horizontal tubes (T_2). The temperature of the not disturbed soil ($T_{infinite}$) was also recorded at 0.9 m depth, but at 10 m distance, lateral from the ground heat exchanger #1.

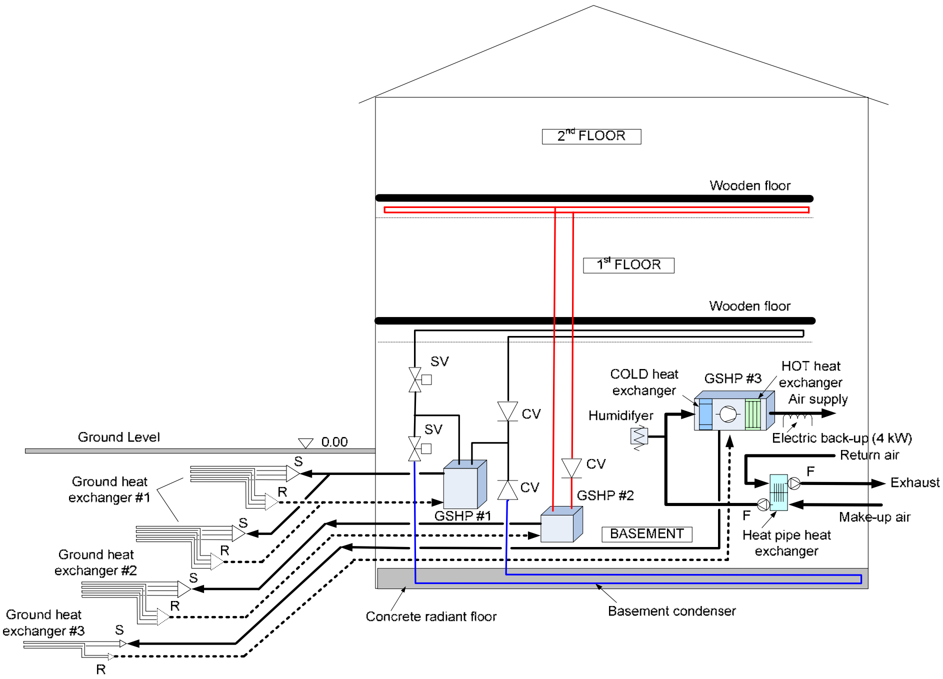

Figure 1.Configuration of the integrated HVAC system
GSHP: ground-source heat pump; SV: motorized valve; CV: check-valve; S: supply header; R: return header; F: fan

With the hypothesis that the temperature is constant along the z axis and that it vary only in radius direction, the conduction heat transfer through an infinite soil cylinder (**r_1** radius) without internal heat sources is given by the Laplace equation:

$$\frac{\delta^2 T}{\delta x^2} + \frac{\delta^2 T}{\delta y^2} = 0 \qquad (1)$$

accompanied by the circle equation: $x^2 + y^2 = r^2$. By neglecting the copper tubes thermal resistance, and under Dirichlet limit conditions ($r = r_0$, $T = T_1$ and $r = r_1$, $T = T_2$), the soil temperature field results in cylinder coordinates:

$$T = T_1 + \frac{T_2 - T_1}{\ln \frac{r_1}{r_0}} \ln \frac{r}{r_0} \qquad (2)$$

where $T_1 = T_0 + 1.5°C$ is the tube wall interior temperature (T_0 – refrigerant saturated temperature).
The thermal flux density results as follows (Fourier hypothesis):

$$\dot{q} = -k\,gradT = \frac{k_{soil}}{r} \frac{T_2 - T_1}{\ln \frac{r_1}{r_0}} \qquad (3)$$

The thermal flux per linear meter of soil (W/m) will be:

$$\dot{q}_L = \frac{\dot{Q}}{L} = \frac{2\pi k_{soil}(T_2 - T_1)}{\ln\frac{r_1}{r_0}} \qquad (4)$$

The heat transfer through horizontal tubes between the refrigerant and the interior wall surface first take place by convective two-phase flow vaporization (zone I) and then by single-phase (vapor) forced convection (zone II). The existing correlations for convective vaporization depend on the fluid properties and the range of operating parameters as regime flow, heat flux, mass velocity, vapour quality, fluid properties and surface geometry. The correlation used for convective vaporization in horizontal tubes with Froude number less than 0.05 superposes the forced convection and the nucleate boiling contributions [**1**]:

$$h_{TP} = Fr^{(0.1-2Fr)} E h_l \qquad (5)$$

where

$$E = 1 + 3000 Bo^{0.86} + 1.12\left[\frac{x}{1-x}\right]^{0.75}\left[\frac{\rho_l}{\rho_v}\right]^{0.41} \qquad (6)$$

is the enhancement factor correlating both void fraction (x) and two-phase heat transfer as a function of Martinelli parameter X_{tt} and of boiling number Bo;
$\mathbf{h_l}$ - the single-phase liquid heat transfer coefficient calculated from Dittus-Boelter equation:

$$h_l = 0.023\left[G\left((1-x)\frac{d}{\mu_l}\right)\right]^{0.8} \Pr^{0.4}\left(\frac{k_l}{d_i}\right) \qquad (7)$$

In the case of nucleate boiling (zone I), the heat transfer coefficient inside the ground DX tubes may also be estimated with the simplified equation:

$$h_{TP} = C\frac{G^{0.1} q^{0.7}}{d_i^{0.5}} \qquad (8)$$

where the constant **C** includes thermo-physical properties of the refrigerant (**0.17** for R22)

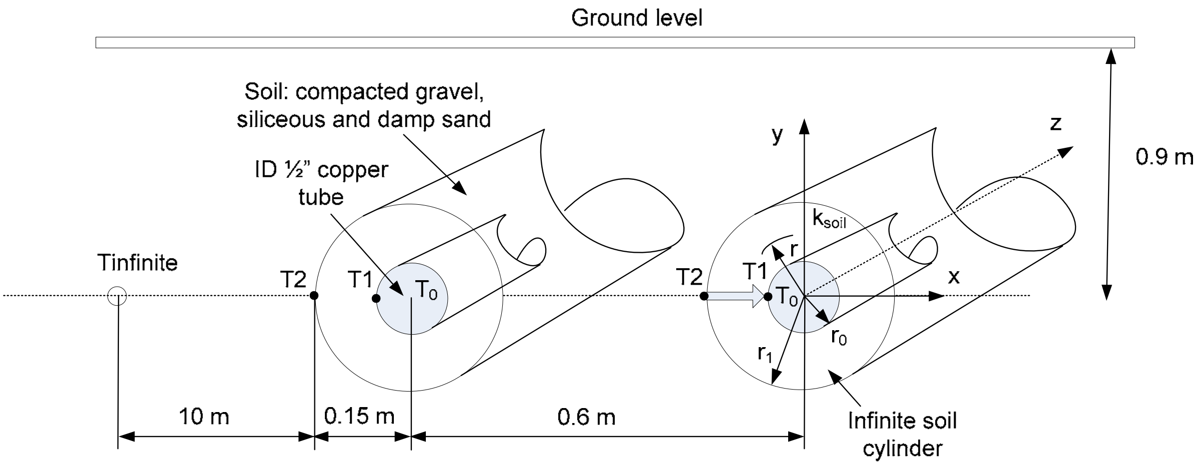

Figure 2.Relative position of ground DX refrigerant tubes buried at 0.9 m depth
T_1: temperature of disturbed soil at 0.15 m from DX refrigerant tubes; k: thermal conductivity; ID: interior diameter; r: cylinder radius; x, y, z: space coordinates

The volume flow rate of refrigerant vapour through the superheating zone II ($\dot{V}_R = \frac{\dot{m}_R}{\rho_R}$) averages 0.00043 m^3/s in each tube of ground DX heat exchangers, and the refrigerant velocity ($w_R = \dot{V}_R / A_R$) results of about 3.65 m/s. The heat transfer coefficient in zone II (134 W/m^2K) was calculated with Mac Adams equation:

$$Nu = 0.023.Re^{0.8}.Pr^{0.4} \qquad (9)$$

where the Reynolds numbers indicate turbulent flows. Table 1 brings together the predicted parameters of the horizontal DX ground heat exchangers.

Table 1.Predicted thermodynamic parameters of DX ground heat exchangers

Parameter	Symbol	Unit	DX ground heat exchanger		
-	-	-	#1	#2	#3
Tube ID diameter	d_i	mm	12.25	12.25	12.25
Tube flow cross section	A_R	m^2	$1.178*10^{-4}$	$1.178*10^{-4}$	$1.178*10^{-4}$
Mass flow	G	kg/m^2s	58	70	70.5
Froude number	Fr	-	0.016	0.023	0.023
Refrigerant velocity	w	m/s	3.65	3.65	3.65
Reynolds number	Re	-	8 052	8 506	8 614
Average evaporating temperature	T_{EV}	°C	-5°C	-5°C	-5°C
Average evaporating pressure	p_{EV}	kPa,a	420	420	420
Average (seasonal) temperature difference ($T_2 – T_0$)	ΔT	°C	6	6	6
Convective vaporization (two-phase flow) heat transfer coefficient (zone I)	h_{TP}	W/m^2K	890	905	907
Forced convective (single-phase flow) heat transfer coefficient (zone II)	h_{II}	W/m^2K	134	134	134

CONDENSERS

All mono-fluid DX ground heat pumps use condensers inserted at 40 mm inside a 100-mm concrete slab (basement) or placed under conventional wooden floors (1st and 2nd stories) (Figure 3). Both concrete slabs and wooden panels are adequately thermal insulated in order to reduce heat losses.

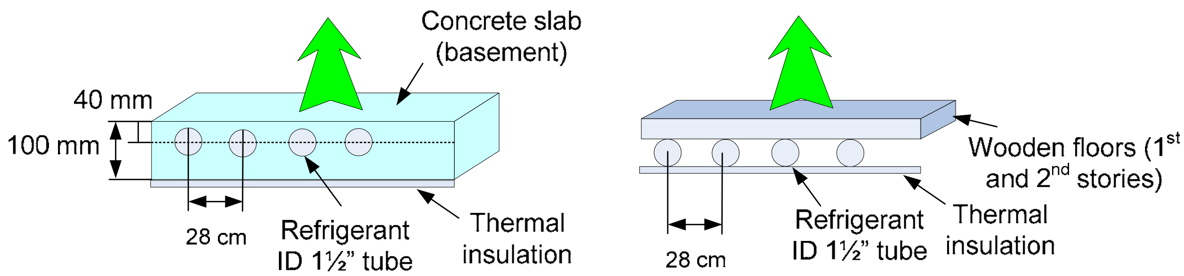

Figure 3.Emplacement of direct condensing tubes on basement (concrete slab), and 1st and 2nd floors (wooden panels)

The condensing heat transfer coefficient in horizontal tubes with stratified flow is given by the following equation:

$$h_c = 0.728 \varepsilon_v^{3/4} \left[\frac{k_l^3 \rho_l (\rho_l - \rho_v) g \Delta h_v}{\eta_l d_i (T_{cond} - T_{wall})} \right] \qquad (10)$$

Where

$$\varepsilon_v = \frac{1}{1 + [(1 - x_v)/x_v](\rho_v / \rho_l)^{2/3}} \qquad (11)$$

is the void fraction [**2**] and x_v – the vapour-phase mass flow fraction.

THERMIDYNAMIC AND COMFORT PARAMETERS

The experiment took place at outdoor weather conditions highly representative for the Canadian cold climate. During the 5-month in-field experience, the total monthly precipitation exceeded the area's usual rainfall by up to 55%. Heavy snow accumulations provided efficient thermal insulation layer during the winter and renewable energy during the spring thaw. Consequently, the average monthly temperature of the non-disturbed soil at 0.9 m depth has never dropped below 0°C, while the minimum temperature of the disturbed soil (T_2) at 150 mm lateral from DX refrigerant tubes averaged -1°C in February, March and April (Figure 4). Generally, the thermal perturbation of surrounding soil has disappeared at a distance of approximately 0.3 m around each horizontal refrigerant tube.

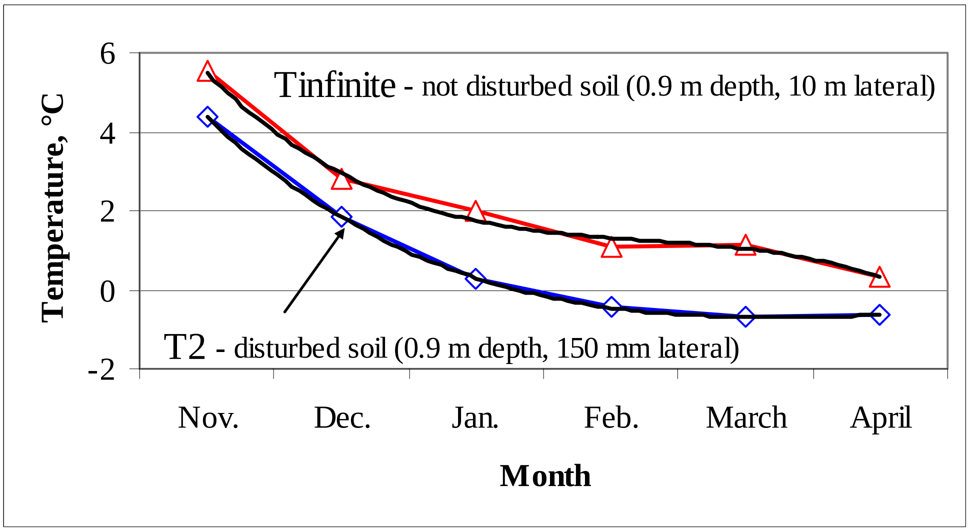

Figure 4 - Temperature profiles of disturbed and not disturbed soil at 0.9 m depth

In winter, the heat pumps #1 and #2 have operated according to typical profiles of such space heating devices. For example, during a very cold and windy night in January (Figure 5), the GSHP #1 supplying the house basement and 1st story, has continuously operated. During the day, the length of on/off cycles varied from three hours to 10 minutes according to the thermostat temperature settings and occupants comfort habits. However, further control improvements would be possible by combining the conventional indoor air temperature control with the radiant floor surface temperatures monitoring. This combined control strategy may be done, for example, on a day/night or a week basis.

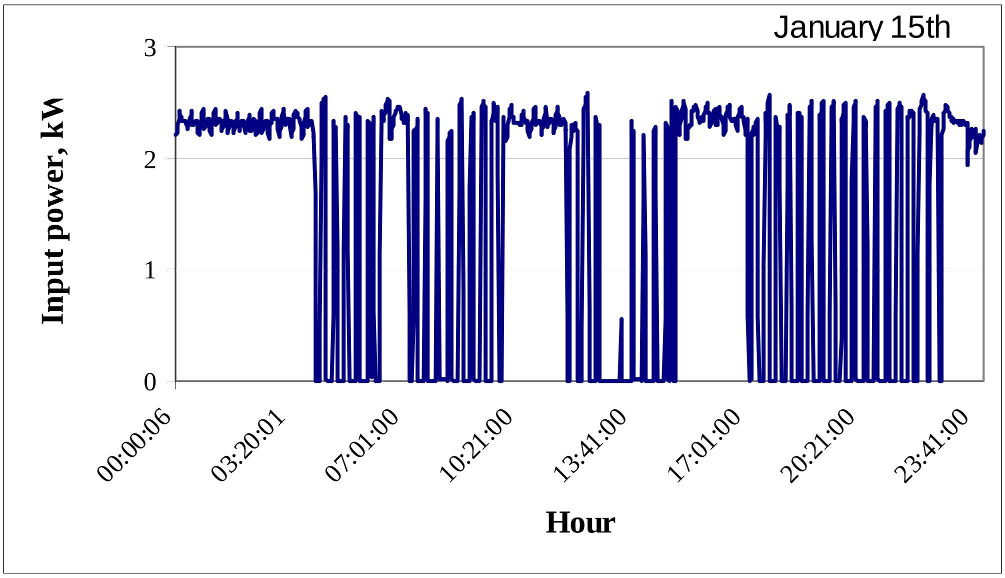

Figure 5 –Operation profile of GSHP #1 (January 15th)

The heat pumps discharge temperatures varied between 100°C and 115°C, while the condensing temperatures averaged 42 to 52°C and the evaporating temperatures were constant at around -5°C during the winter (Figure 6). Higher sub-cooling amounts were obtained with the condenser inserted in the concrete heating floor compared to condensers installed under the wooden floors. However, motorized liquid valves were installed during the tests in order to avoid refrigerant liquid hammers and/or whistling during the compressor shut-down periods. The refrigerant sub-cooling provided efficient operation of the GSHP #1 expansion valve and increased efficiency of its thermodynamic cycle. Such condensing temperatures and sub-cooling, combined to the GSHP #1 running cycles, provided stable temperatures at the basement concrete floor surface of around 26°C. It may be noted that the heating technique using direct condensing coils coupled with wooden floors is not fully developed and should be further improved. Consequently, the average temperatures at the surface of the basement concrete floor have been generally higher compared to those of wooden floors on 1st and 2nd floors. This indicates effective heat transfer and heat storage inside the concrete slab. One of the most significant advantages of radiant floor heating systems is the highly homogenous vertical air temperature distribution. The vertical temperature distribution between the surface of the basement concrete floor and the ceiling was effectively remarkably homogeneous (Figure 7). The average indoor air temperatures during the colder winter month (January) have varied from 20°C (basement) to 22°C (2nd story). During the winter, the make-up air was generally heated by 15 to 17°C through the exhaust air heat recovery heat exchanger (thermal efficiency of 57%), and by up to 13°C through the GSHP #3. The fresh make-up air has left the GSHP #3 at 18°C (December), 23°C (March) and 20°C (April), and the final adjustment of supply temperatures was provided by the back-up electrical heater.

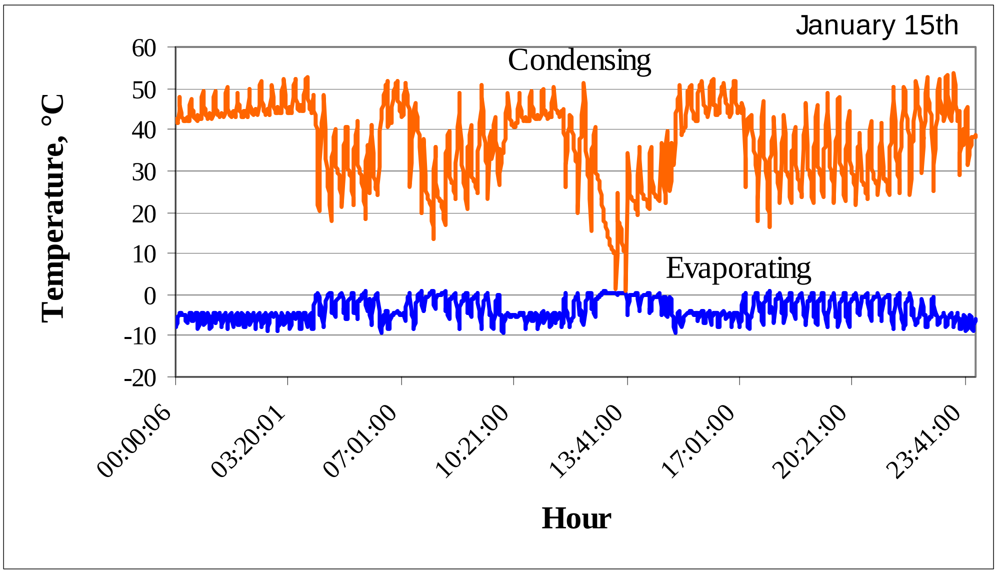

Figure 6 – Refrigerant evaporating and condensing temperatures – GSHP #1 (January 15th)

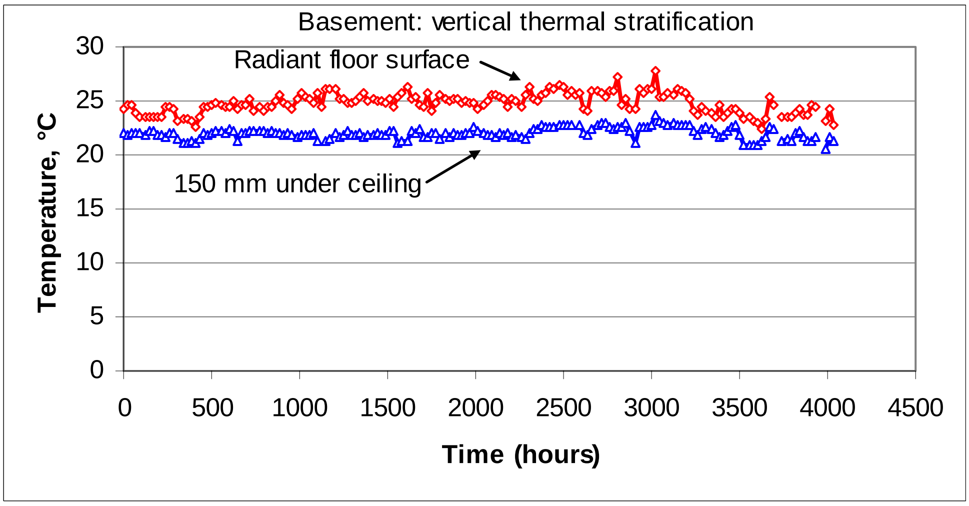

Figure 7 – Radiant floor temperature and indoor air vertical thermal stratification (basement)

ENERGY PERFORMANCES

Between December 1st and April 30th, the GSHP #1 operated 58.6% of the time. This unit has extracted from the ground 15 400 of thermal kWh. The heat pump compressor consumed 5 310 kWh with average compression ratios of 4 and electrical input power of 2.5 kW. The pressure losses through the horizontal ground heat exchanger averaged 0.4 kPa/m. The seasonal energy balance in heating mode of the GSHP #1 system shows that 74.5% of the output energy used to heat the basement and the 1st floor was extracted from the ground and 25.6% was provided by the compressor electrical energy input (Figure 9). The seasonal coefficient of performance (SCOP) of GSHP #1 unit, parameter defined as the total heating energy delivered divided by the total electrical energy input, thus averaged 3.9 (Table 2).

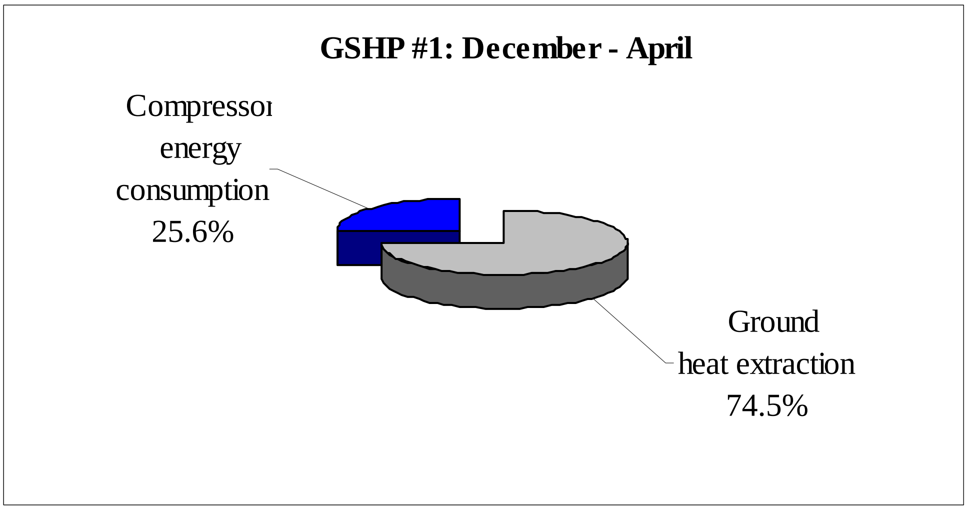

Figure 9 – Energy consumption and ground heat extraction of GSHP #3 (December to April).

The seasonal energy balance of the GSHP #2 unit in heating mode shows that 73.7% of the heating energy was supplied by the ground and the seasonal coefficient of performance (SCOP) was 3.8. This ground source heat pump operated 40% of the time and extracted 7 714 kWh from the ground. Because of temporary breaking of the acquisition system in January and February, no reliable energy data are available for the make-up air preheating system including the GSHP #3. During the entire 5-month heating period, the specific ground thermal capacity (power extraction) respectively averaged 10 W/m (GSHP #1) and 14.8 W/m (GSHP #2). At the same time, the seasonal heat dissipation basement and 1st floor radiant surfaces averaged 20.1kWh/m^2 (basement) and 21.1kWh/m^2 (1st floor). These values correspond respectively to 16 W/m (basement) and 20 W/m (1st floor) of heat flux dissipations per linear meter of direct condensing tubes (Table 2).

Table 2.Seasonal performances of ground DX heat exchangers

Parameter	Symbol	Unit	DX ground heat pump		
-	-	-	#1	#2	#3
Operating time	-	%	59	40	N/A
Ground heat extraction	-	kWh	15 400	7 714	N/A
Energy consumption (compressors)	-	kWh	5 310	2 755	N/A
Heat supplied	-	kWh	20 710	10 469	N/A
Seasonal COP	-	-	3.9	3.8	N/A
Average ground heat flux extraction per square meter of tube	$\dot{q}_{EV}$	W/m^2	260	382	N/A
Average ground heat flux extraction per linear meter of tube	$\dot{q}_{L-EV}$	W/m	10	14.8	N/A
Average heat dissipation	-	kWh/m^2	20.1	21.1	N/A
Average heat flux dissipation	-	W/m	16	20	N/A

CONCLUSIONS

The integrated HVAC system with ground-source heat pumps and exhaust air heat recovery device has demonstrated its technical feasibility in the Canadian cold climate. Predicted thermodynamic parameters for two-phase convective vaporization and condensation allowed designing optimum length of ground heat exchangers and radiant floor condensers. The space heating DX ground heat pumps have operated with seasonal coefficients of performance up to **3.9** providing high comfort with uniform thermal stratification of the indoor air. The 5-month in-field experience (December and April) showed that the average ground heat flux extractions ranged from 10 W/m (GSHP #1) to 14.8 W/m (GSHP #2) for total operating times of 58.6% and 40% respectively. Further R&D efforts will be done to develop more efficient wood-type radiant heating floors and advanced heat pump strategy controls based on dual (air & radiant floor surface) temperatures monitoring. Installing ground DX heat pumps with individual condensers for each thermal zone instead parallel condensers, optimizing the total refrigerant charge, using natural refrigerants, as propane, and develop reversible DX heat pumps for indoor air cooling and dehumidification, are future R&D objectives for this particular application.

NOMENCLATURE

Bo	boiling number ($\dot{q}/Gh_{lv}$)
c_p	specific heat, J/kgK
d	diameter, m
ΔT	temperature difference, °C
E	enhancement factor
ε	void fraction
Fr	Froude number ($G^2/\rho_l^{\,2}gd_i$)
G	mass flow, kg/m^{2}s
g	gravitational acceleration, m^2/s
h	heat transfer coefficient, W/m^{2}K
h_{lv}	vaporization enthalpy, kJ/kg
k	thermal conductivity, W/mK
L	cylinder length, m
μ	dynamic viscosity, Ns/m^2
ν	cinematic viscosity, m^2/s
$\dot{q}$	specific heat flux, W/m^2
Pr	Prandtl number ($c_p\mu/k$)
$\dot{Q}$	total heat flux, W
Re	Reynolds number ($Re = w_R d_i/\nu_R$)
ρ	density (kg/m^3)
r	cylinder radius, m
X_{tt}	Martinelli parameter
x	dryness fraction
w	velocity, m/s

Subscripts

cond	condensing
i	interior
l	liquid
TP	two-phase
R	refrigerant
SP	single-phase
v	vapour

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Research on the Air-source heat pump applications in the cold and serious cold regions

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SUMMARY

The air-source heat pump (ASHP) is a convenience and higher energy efficiency rating air conditioner; it can make the best of low-temperature energy sources of the atmosphere circumstance, and has less pollution to the region of use it. The air-source heat pump (ASHP) has energy conservation and environment protection effects; it is a kind of green air conditioning technical. But in northern, northwestern and northeastern of china, the air-source heat pump (ASHP) can not work normally in the period of heating， if it only use outdoor air as low-temperature heat source, the heating quantity of heat pump units decrease greatly, the coefficient of performance (COP) decline rapidly, the outside heat exchanger will been frosted, because of the lower outdoor air temperature.

For solved such issues, authorized by Ministry of Construction P.R.China, the Discussion Group advanced the method of the fresh air handling units in combination with the heat pump units. Make use of the exhaust fully, mix the exhaust and outdoor air, and raise the intake air temperature of heat exchanger, thereby， the heat pump units can work normally in the period of heating.

The Discussion Group analyzed the recent climate condition statistic of Changchun, China, designed the heat pump & fresh air unit, calculated the mix ratio of fresh air and exhaust at the disadvantage of air temperature and calculated heating quantity of the heat pump unit. The data shows that at the time of buildings exhaust is enough, the heat pump unit can work normally in the period of heating in cold and serious cold regions. The Discussion Group will research more about the experiment application of the units in the winter 2006-2007 in Changchun, china.

INTRODUCTION

Now， with the popularization of the decorate, the modern equipment(the computer the printer etc.) and the air condition, as well as the improved hermetic technology in the energy-saving building, the fresh air requirement of air-condition system is shortage or be canceled. It brings on the decline of indoor air quality (IAQ), and the sick building synthesis (SBS). Adjusted and controlled the indoor weather parameter (temperature, humidity, wind speed), provided the fresh air requirement that accord the indoor air quality, it could improve the IAQ. But at the same time it raised the operating energy consumption and increased the operating costs of air condition system, apparently. How to properly handle the relationship between the increased fresh air requirement to improve IAQ and the operating costs of control system are the main issues of air conditioning design. The exploration and research on new systems and new equipment suitable or compatible the thermal comfort, the indoor air quality and the energy conservation are the urgent requirement for sustainable development of energy-saving building field.

The air-source heat pump (ASHP) is a convenience and higher energy efficiency rating air conditioner; it can make the best of low-temperature energy sources of the atmosphere circumstance, and has less pollution to the region of use it. The air-source heat pump (ASHP) has energy conservation and environment protection effects; it is a kind of green air conditioning technique. But in northern, northwestern and northeastern of china, the air-source heat pump (ASHP) can not work normally in the period of heating， if it only use outdoor air as low-temperature heat source, the heating quantity of heat pump units decrease greatly, the coefficient of performance (COP) decline rapidly, the outside heat exchanger will been frosted, because of the lower outdoor air temperature.

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THE PRESENT STATUS OF APPLICATION RESEARCH ON THE AIR-SOURCE HEAT PUMP OVER THE WORLD

Energy prices and each country's energy mix affect the conditions for heat pumps to be competitive with other alternatives. Other important factors influencing how and if heat pumps are used are tradition, knowledge and the condition of existing and new buildings. Research and development, various types of political incentives, requirements for specific minimum levels of energy efficiency and the dissemination of knowledge are all important instruments for encouraging growth of the heat pump. The fact that climate, the condition of building stock, experience of the use of heat pumps and energy mix all vary widely from one country to another. [1]

The new technology of native and foreign air-source heat pump

The new technologies include that the heat pump unit performance ameliorated and advanced (for example screw compressor substituted centrifugal compressor, introduced frequency conversion timing technique, ameliorated configuration and surface performance, adopted electronic expansion valve achieved defrost circulation control, the mistiness nerve net achieved intelligent control, etc.)[2]. Heat pump exploited on the new-style exceeder critical CO_2circle system. New product exploited and extended the scope, include escalating capability for commercial building and use to exhaust heat recovery [3]. Energy storage technology applied [4].

In china, the research of minitype air/air heat pump experimentation technology and heat pump air condition computer simulation technology advance lot of innovation project. Especially, in the research of the heat pump unit worked in cold and serious cold regions.

The localization of air-source heat pump using in cold and serious cold regions

In china, climate condition is a wide variety, roughly be divided into several climate zones: serious cold, cold, hot in summer and cold in winter, very hot (tropical), with variations in humidity depending on whether it is coastal or continental climate. Technological development of Commercial air-to-air heat pumps, to expand the operating temperature range to a lower ambient temperature, has been explored sporadically and is still on going.

Theory and practice indicate that air-to-air heat pump system without any betterment not run well in winter, using it in cold region. The main issues are:

The quantities of heat making through the heat pump system decrease rapidly with outdoor temperature decline, but the quantities of heat requiring of fresh air increase rapidly. When temperature of outdoor is very low, quantities of heat making of the system not satisfy the needs of heating in winter in the region; and then the COP of machine decline rapidly, the compression ratio of compressor increase more, the temperature of exhaust air rise, if the compressor run for a long time in the condition, it will be broken.

If consider the running effect of the heat pump systems, the defrosting veracity is not well. The defrosting criterion not adapted well to the large-scale status in north of china, the phenomena of error defrosting is severity, the time of defrosting is longer, and that can make the indoor temperature greatly fluctuate; the system start up difficultly, the machine will be broken when the system start up or end of defrosting and reheat.

In sum, if only depend on the measures of heat recovery, anti-freezing, and defrosting of the air-source heat pump (ASHP) heat pumps in cold region, the issue of fresh air handling units of centre Air Condition system and heat pump units running by themselves cannot be solved. The effect of technology and economy is not well in cold or serious cold region.

THE STUDY DESIGN CONCEIVED

Analyzed the problems of using air-source heat pump in the cold and serious cold regions; we can say that the temperature and humidity is the main influence.

On the basis of the Japanese experimentation result [2], the frost weather parameter is likely to -12.8 ℃$\leq t_w \leq$5.8 ℃, $\varphi \geq$67%.That means when t_w>5.8 ℃, we can take no account of the frost influence to heat pump; when t_w<5.8 ℃, φ<67%, owing to the air dew point temperature t_{dew} under the heat exchanger surface temperature t_{fe}, it can not frost; when t_w<-12.8℃, because of the low air humidity ratio, it can not frost too.
At present, the Chinese instruction of heat pump engineering design prescribe that the outdoor design temperature in winter t_w=-3 ℃ is the air-source heat pump (ASHP) running lowest-level. In this way, in the region that outdoor design temperature for heating $t_w \geq$-3 ℃, the air-source heat pump can run normally. Or else, the air-source heat pump (ASHP) can not run. In northern, northwestern and northeastern of china, mostly region outdoor design temperature t_w<-3 ℃, that is the reason of the air-source heat pump (ASHP) can not work normally in the period of heating. Now bounded-3 ℃, statistical treatment the weather data of main city in the

cold or serious cold regions of china, quantificational take measures of the air-source heat pump(ASHP) runtime and its proportion of the whole heating period. The table 1 shows the weather data result.

Table1. The weather data of main city in the cold or serious cold regions china

City	Outdoor design temperature for heating t_w(℃)	Hours for heating $T_z(h)$ ℃	t<-3 ℃ hours for heating $T_x(h)$ ℃	t<-3 ℃ hours proportion T_x/T_z ℃	Outdoor design relative humidity (%)
Harbin	-26	4296	3089	71.9	74
Changchun	-23	4176	2879	68.9	68
Shenyang	-19	3648	2205	60.9	44
Taiyuan	-12	3456	1637	48.4	51
Beijing	-9	3096	1287	41.6	45
Jinan	-7	2544	472	18.6	54
Xi'an	-5	2424	208	8.6	67

From the result of the weather data, the air-source heat pump (ASHP) runtime is long (that about 30% heating period) in the serious cold region (Harbin). In the great mass of northern, northwestern of china, the air-source heat pump (ASHP) runtime is at least half of the heating period. If use the outdoor design temperature for winter ventilation to calculate for the system of the fresh air handling units in combination with the heat pump units, the air-source heat pump (ASHP) runtime will even more.

Accordingly, the issue of building heat recovery, prevented frostbite of fresh air handling, advanced the air-source heat pump (ASHP) efficiency in winter, prolonged the building fresh air supply time and the air-source heat pump (ASHP) runtime in winter, would been solved effectively.

THE PROJECT DESIGN OF COMBINATION UNIT IN THE COLD OR SERIOUS COLD REGIONS

Waste heat from various types of central Air Conditioning system of commercial buildings can provide a heat input to absorption heat pumps or compressor heat pumps. Another application is to recover heat from the exhaust air from mechanically-ventilated residential buildings. A requirement for heat recovery is also introduced in northeastern china, and has resulted in increased interest in exhaust air heat pumps in northeastern market.

Load from fresh air is 20%-30% of Air Conditioning. If installed the heat recovery equipment in fresh air handling unit, reclaimed the energy of exhaust air, increased energy-efficient, it would be vital importance measure of energy conservation.

At present, in china, the Air Conditioning design of middling or top grade hotels and office buildings commonly adopt primary air fan-coil system as well as exhaust air system. the energy consumptions of this type of Air Conditioning system is about 20%-40% of buildings, and that the percentage of fresh air load is20%-30 %of total load of Air Conditioning system, thus the energy consumptions of fresh air is 4%-12% of buildings. If the heat recovery efficiency is 60%, the efficiency of energy conservation will be 2.4%-7.2%.

For instance, one building in Changchun, there are 1floor underground, 10floor upside. Underground is park; the acreage is 2850m^2; 1 to 3floors are marketplace, total acreage is8550m^2; 4floor is office and restaurant, the acreage of office is 600 m^2 and 2250 m^2 of the restaurant; 5 to 9floors are hotel, the guest room of hotel is standard for 2 people, total number is 140.According as the standard and criterion, calculated the design quantity of fresh air and its load. The load of fresh air calculated by the equation below:

$$Q_{h,o} = M_o \cdot c_p (t_o - t_R), \quad (1)$$

Where $Q_{h,0}$ is the heating load of fresh air, kW and c_p is the level pressure specific heat, *kJ/(kg· ℃)*， is 1.005 *kJ/(kg· ℃)*. t_0 is the outdoor design temperature for winter ventilation, ℃, the number is－16℃ in changchun and t_R is the indoor design temperature, ℃， is 2 ℃ in the design. M_0 is the design quantity of fresh air, *kg/s*. Table 2 [5] [6] is the result of the design quantity of fresh air and its load calculation. Considered the positive pressure is 5-10*Pa* of the room, the design quantity of exhaust is 90% of the design quantity of fresh air. Considered the negative pressure is 5-10*Pa* of the park, the design quantity of exhaust is 125% of the design quantity of fresh air. Table 3 is the result of the design quantity of exhaust and its load calculation.

In Changchun, in winter, outdoor design dry-bulb temperature for winter ventilation is -16 ℃, wet-bulb temperature is -16.67 ℃, the relative humidity is 68%[7], the room parameter are, indoor dry-bulb temperature as Table3, relative humidity 40%. According to Table3, the losing heat quantity in winter is about1520.77*kW*.if the exit temperature of evaporator is 5 ℃， the recoverable heat quantity of exhaust is about 874.47 *kW*. Thus it can be seen, in air conditioning building, the discharge energy is so large, the heat recovery of discharge energy of air-condition system is very important.

Table2. The result of the design quantity of fresh air and its load calculation

Room	Standard quantity of fresh air($m^3/(h·p)$)	Area (m^2)	Number of people(p)	Design quantity of fresh air (m^3/h)	Load of fresh air (kW)
Guest room	50		280	14000	95.85
Marketplace	25	8550	2850	71250	487.84
Office	25	600	150	3750	25.67
Restaurant	30	2250	900	27000	184.87
Park	4.8time/h	2850		41040	280.99
Total				157040	1075.22

Table3. The result of the design quantity of exhaust and its load calculation

Room	Design quantity of exhaust(m^3/h)	Temperature of exhaust(℃)	Heat quantity of exhaust(kW)	Recoverable heat quantity of exhaust(kW)
Guest room	12600	22	233.71	116.97
Marketplace	64125	18	745.25	437.80
Office	3375	20	109.71	27.12
Restaurant	24300	18	325.95	165.94
Park	51300	10	386.55	126.64
Total	155700		1520.77	874.47

DESIGN AND SIMULATION OF COMBINATION UNITS

Design combination units base on the part of hotel in the building, the guest room are configured as standard room, total number is 140. The runtime is Am8:00-Pm10:00 and the heat loss of fresh air is95.85*kW* (table2). The design principle of heat pump unit is by the equations below [8]:

The equation of heat loss of fresh air related to the mean air temperature (Q_P):

$$Q_P = Q_W \frac{t_N - t_P}{t_N - t_W}, \quad (2)$$

Where Q_P is heat loss of fresh air related to the mean air temperature, *kW* and Q_W is heat loss of outdoor design temperature for fresh air heating, *kW*, (95.85 *kW*). t_N is the design temperature for fresh air, ℃, (2 ℃) and t_p is the outdoor mean air temperature during heating period, ℃, (-7.9 ℃). t_W is outdoor design temperature for winter ventilation, ℃, (-16 ℃).
The equation of refrigerating effect coefficient of refrigerating medium (*k*):

$$k = \frac{Q_k}{Q_0}, \quad (3)$$

Where Q_k is condensation heat gain of refrigerating medium, *kW* and Q_0 is evaporating heat gain.
The equation of mean refrigerating effect ($Q_{0 \cdot P}$):

$$Q_{0 \cdot p} = \frac{Q_P}{k}, \quad (4)$$

Where $Q_{o \cdot P}$ is mean refrigerating effect.
The equation of evaporator heat gain (Q_0):

$$Q_0 = K_0 F_0 \left[{(t_1 + t_2)}/{2} - t_0 \right], \quad (5)$$

$$Q_0 = \frac{1}{3.6} c_P G_0 (t_1 - t_2), \quad (6)$$

$$Q_0 = \frac{K_0 F_0}{1 + \frac{3.6 K_0 F_0}{2 c_P G_0}} (t_1 - t_0), \quad (7)$$

Where K_0 is evaporator coefficient of heat transfer, $W/(m^2 \bullet ℃)$ and F_0 is surface area of evaporator. t_0 is evaporating temperature, ℃ and c_p is the level pressure specific heat, $kJ/(kg \bullet ℃)$. G_0 outdoor air requirement pass evaporator, *kg/h* and t_1, t_2 are intake temperature and outlet temperature of evaporator.

The design parameter of heat pump unit is: Based on the weather data, calculated the outdoor mean air temperature during heating period t_P=-7.9 ℃ and the heat loss of fresh air related to the mean air temperature Q_P=52.72*kW*. Cryogen is R22; temperature of evaporation (t_0) is -5 ℃; temperature of condensation (t_k) is 40 ℃. Based t_0,t_k, get the refrigerating effect coefficient of refrigerating medium by fig P-h of R22 and equation(2) ,*k*=416.25/194.17=2.14.Calculated mean refrigerating effect,Q_{0P}=52.72/2.14=24.62*kW*, by equation(4). Make certain the outdoor air requirement pass evaporator, the air requirement/ mean refrigerating effect=400$m^3/h/kW$, the outdoor air requirement pass evaporator, G_0=1.29×400×24.62=12703.90*kg/h*,the fact quantity of exhaust is 12600×1.196=15071.77*kg/h*, that means if 85% of fact quantity of exhaust can suffice. Now used 60% of fact quantity of exhaust, calculated the exhaust outlet temperature of evaporator,t_2=8-24.62×3600/1.005×12703.90=1.06 ℃, by equation(6). When the wind speed is 3.5*m/s*, the evaporator coefficient of heat transfer, K0=32.6$W/(m^2\bullet ℃)$. Considered the influence of frost, the safety factor is 0.8, K_0=26.1$W/(m^2\bullet ℃)$. The surface area of evaporator is F_0=85.52m^2 by equation (5). Based on $Q_{0\cdot P}$=35468W, t_0=-5 ℃, t_K=40 ℃, selected reciprocating compressor. The evaporator heat gain, Q_0=1697.39(11+5) =22.07kW. Got the heat gain of compressor from the compressor stylebook, calculated the quantity for heating of condensation, Q_k=22.07×2.4=52.97kW, compare with the heat loss of fresh air related to the mean air temperature Q_P=52.72kW, it is congruence.

The parameter list in table4, the design combination unit construct is figure1.

Table4. The parameter list of combination units

Design quantity of fresh air (m^3/h)	15000	
Heating capacity(*kW*)	123.9	
Input power(*kW*)	20.45	
Power supply	3Ph/380V/50Hz	
Mode of interior side exchanger	High efficiency screw tube with aluminum fin	
Compressor	Type	Hermetic scroll compressor
	Quantity	3
Refrigerant	Control	External equalizer thermostatic expansion valve
	R22 charge(*kg*)	29.5
Exterior side exchanger	High efficiency lube-on-sheet condenser	
Outdoor heat exchanger dimension(*mm*)	2000/950/2100	
Operating noise dB(A)	68	

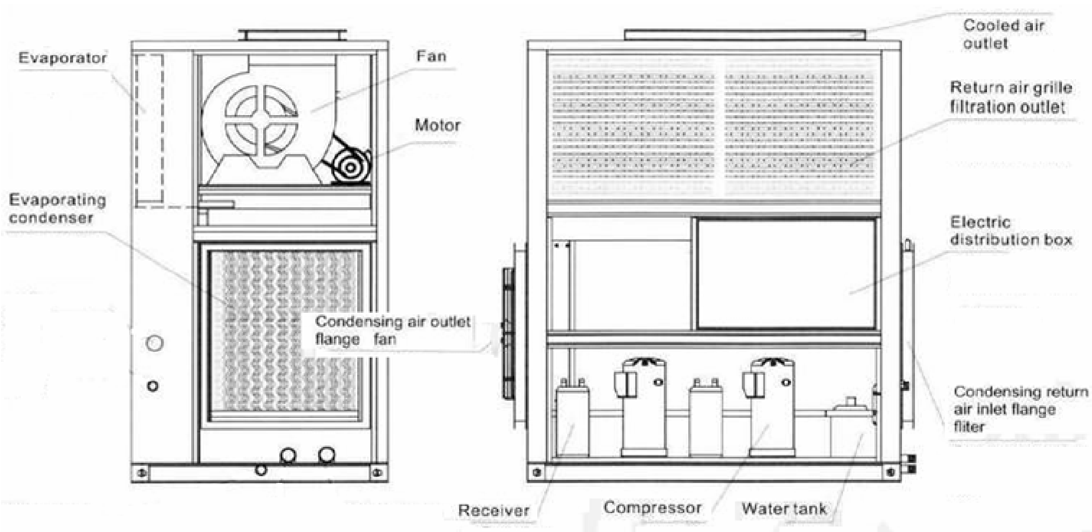

Figure1. The construct of design combination units

DISCUSSION

The unit system design was based on the hotel of Changchun, a type city of serious cold regions in northeastern of china. The design accord the design standard for energy efficiency of public buildings, the parameter gist based on the conclusions that have a series of analyzed for a long time. At the standard state, that ensured total load of fresh air and 60% quantity of exhaust, the units can run safety. At the design state, it can prolong time of the heat pump units running about 2625*h*(t<-3 ℃) hours for heating is 2879*h*, under the design parameter -16 ℃,hours for heating is 254*h*). For a year, time of heat pump unit can not run normally only 254h. it can farthest ensured fresh air of building. About running state, the defrosting control technology and assistant heating technology use could prolong running time more. We will deepen the study.

In building with air conditioning systems, the exhaust air of air conditioning systems take away much of heat quantity (in winter) or cold quantity (in summer), recovering exhaust air heat of air conditioning systems, the meaning of energy conservation is vital prominence. The air-source heat pump is equipment with benefit of energy conservation and environment protecting. In practice, the parameters of outdoor air affect the energy efficient ratio and the capability, the using of heat pump is limited by the geography condition. If lead exhaust air of air condition systems to the outdoor exchanger of air-source heat pump, the energy efficient ratio and the capability can be enhanced, and can recover the heat (cold) quantity, achieve the aims of energy conservation.

The centre air conditioning systems can create condign temperature environment and humidity environment, at the same time; it can reduce the concentration of nocuous component through the method of keeping fresh air quantity, and keep the air quality. But the fresh air handling unit of centre Air conditioning systems can't run normally in winter of cold regions, lead to the fresh air handling unit losing the function of supply fresh air. The using of air-source heat pump with exhaust heat recovery is an efficacious method.

The resulting future impact on global environment, development of advanced energy efficient and environmentally friendly technologies is of vital importance for the sustainable development in the foreseeable future.

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Environmental Assessment Method for Seawater Source Heat Pump Systems

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SUMMARY

This paper presents a study on thermal diffusion of large-scale seawater source heat pump system (SWHP).

SWHP uses seawater as heat source and sink, and discharges the same quantity of water into local oceans after use. The cooling water systems may cause a temperature rise of 3-5℃ above ambient at discharge points which is close to the upper thermal tolerance limit of most marine biota, which would not only have a certain impact on the local marine environment, but also affect the local micro-climate, the local marine life and fishing economy, etc.

Based on Navier Stokes equations and shallow water assumption, non-linear shallow water equations are established in this paper. The two-dimensional numerical simulation research of the flow velocity field and temperature field of the water space where thermal effluent is discharged is done using a case study.

And following an example of Xiaoping Island SWHP project is given. Simulation and description on the discharge of this SWHP project show that, the temperature rise/drop extends to certain area, but the temperature elevation of the influence area is far less than the standard of Grade Ⅰ of Quality Standard for State Oceanic Water.

This paper firstly introduces the hydrodynamic knowledge into HVAC design application, which shows a way to the environmental impact statement of Water Source Heat Pumps.

INTRODUCTION

In the 21st century, the energy issue has become increasingly prominent. The sea water-source heat pump (SWHP) system’s coming into China is a very important significance for the energy efficiency in buildings. Seawater source heat pump technology transfers solar energy captured in seawater which is low temperature and low quality, into high quality energy through the use of heat pump and a small quantity of electricity. At present, there’re only several experimental projects in China, and environmental impact assessment is one of the most important issues to resolve.

SWHP system uses seawater as heat source and sink, and discharges the same quantity of water into local oceans after use. The temperature of the discharge water is higher in the summer than the local water temperature around 10℃, and lower in winter about 3-5℃ than the receiving water body. The discharge temperature elevation of seawater will not only have a certain impact on the local marine environment, but also affect the local micro-climate, the local marine life and fishing economy, etc. When the large-scale application of SWHP systems come into reality in the coastal cities in China for district cooling and heating, these impacts may become particularly prominent.

The above issue can be solved by the thermal discharge research method. Thermal discharge exists in every thermal/nuclear power plant, and researchers have been working on it for tens of years. The method consists of domestic physical and numerical simulation, of which the later is most widely used. What's more, the two are often combined, for the model the model parameters of numerical equation are often obtain from physical tests.

According to different classification method, numerical model can be classified as the near-field model and the far-field model, or the 2D depth-averaged model and 3D mode, and during actual applications these models are developing toward a more sophisticated, visualization and intelligent direction. With the emergence of the sea water-source heat pump system, some new problems need to be considered.

Between the mathematical models studied by scholars at home and abroad, the depth-averaged two-dimensional fluid dynamics model is very effective to solve some problems in the oceans, coasts water level and instantaneous transport, which is widely used in the study of thermal discharge, to predict the temperature distribution of thermal discharge area[1,2].

MATHEMATIC MODEL

Based on Navier Stokes equations and shallow water assumption, non-linear shallow water equations are established. The depth averaged two-dimensional hydrodynamic model, including the effects of earth's rotation, bottom friction, wind shear and surface heat exchange are as follows [3]:

Continuity equation:

$$\frac{\partial \zeta}{\partial t}+\frac{\partial HU}{\partial x}+\frac{\partial HV}{\partial y}=Q \tag{1}$$

Momentum equation:

$$\frac{\partial U}{\partial t}+U\frac{\partial U}{\partial x}+V\frac{\partial U}{\partial y}-fV=-g\frac{\partial \zeta}{\partial x}+v_H\left(\frac{\partial^2 U}{\partial x^2}+\frac{\partial^2 U}{\partial y^2}\right)-\frac{gU\sqrt{U^2+V^2}}{HC_{2d}^2}+\frac{C_d}{H}\frac{\rho_a}{\rho}U_{10}^2\cos\beta \tag{2}$$

$$\frac{\partial V}{\partial t}+U\frac{\partial V}{\partial x}+V\frac{\partial V}{\partial y}+fU=-g\frac{\partial \zeta}{\partial x}+v_H\left(\frac{\partial^2 V}{\partial x^2}+\frac{\partial^2 V}{\partial y^2}\right)-\frac{gV\sqrt{U^2+V^2}}{HC_{2d}^2}+\frac{C_d}{H}\frac{\rho_a}{\rho}U_{10}^2\sin\beta \tag{3}$$

Transport equation:

$$\frac{\partial HT}{\partial t}+\frac{\partial HUT}{\partial x}+\frac{\partial HVT}{\partial y}=\frac{\partial}{\partial x}\left(HK_x\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(HK_y\frac{\partial T}{\partial y}\right)+S \tag{4}$$

U、V depth-averaged velocity in x and y direction; ρ_a density of air ; f Coriolis coefficient, $f=2\omega\sin\psi$, ω angular frequency of the earth, ψ the local latitude; 2D Chézy coefficient, computed by Manning Formula : $C_{2d}=\frac{1}{n}H^{1/6}$, n is the Manning's coefficient; C_d wind drag coefficient; v_H horizontal eddy viscosity; U_{10} averaged wind speed at 10 m above free surface; β the angle between the wind stress vector and the local direction of the grid line x; M_x, M_y source or sink of momentum in x and y direction; T the water temperature; K_x, K_y diffusion coefficient in x and y direction; Q, S the source and sink term.

The heat exchange flux is represented by a bulk exchange formula [4]:

$$Q_{tot} = -\lambda(T_s - T_{back}) \tag{5}$$

with T_s the water temperature at the free surface and T_{back} the natural background temperature, both in ℃. The heat exchange coefficient λ is a function of the surface temperature T_s and the wind speed U_{10}:

$$\lambda = 4.48 + 0.049T_s + f(U_{10})\left(1.12 + 0.018T_s + 0.00158T_s^2\right) \tag{6}$$

$$f(U_{10}) = \left(3.5 + 2.0U_{10}\right)\left(\frac{5.0\times10^6}{S_{area}}\right)^{0.05} \tag{7}$$

Where S_{area} is the exposed water surface in m^2.

The control equations formed by ADI method (Alternating Direction Implicit Method, Leendertse， 1967) are solved on a staggered grid. Each of the dispersed control equation group consists of three unknown variables at adjacent nodes. In the coefficient matrix, all the non-zero elements are arranged along a diagonal matrix. So triangle diagonal matrix algorithm (TDMA) is used for solution.

PROJECT AND COMPUTATIONAL ASPECTS

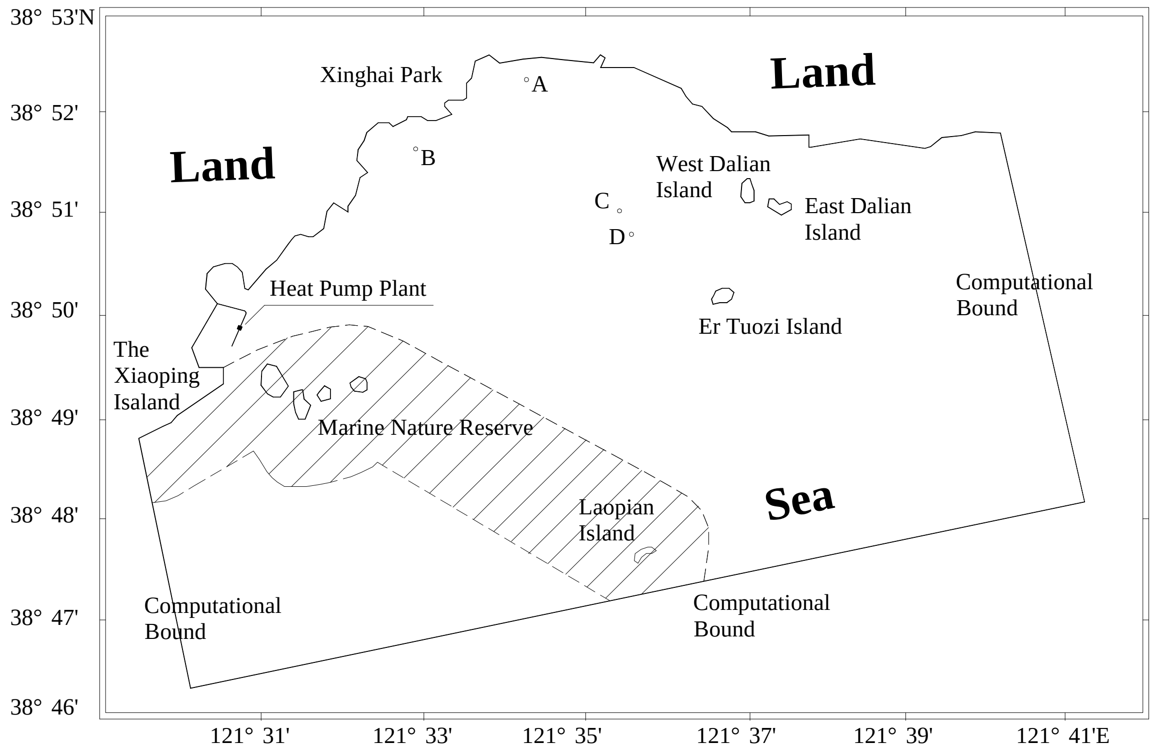

Figure1 Simulation Area and Location of the Project

Using the above method, an example of Xiaoping Island SWHP project is given in this section. This project is located to the Xiaoping Island, at the Huang hai sea of China. The heat pump plant is located at 121°30′34″-121°30′44″E, 38°49′49″-38°49′58″N, on a bulwark of 1300 meters long and L type, see figure 1. The shadow area is the marine nature reserve, which is to be considered not to be affected by this project's discharge.

The design input/output flow and temperature rise/drop at the first stage of this SWHP is as follows:
Heating Season: 1.9 m^3/s, temperature drop 2.5℃
Cooling Season: 1.14m^3/s，temperature rise 10.0℃

RESULTS AND DISCUSSION

24 hours' site tide data of four stations-A, B, C, D（From 8:00, 18th, Apr. to 7:00, 19th, Apr., 2003, sees figure 1）are used for validation of the simulation results. The comparison between the computed and observed results are shown from figure 2 to figure 5。

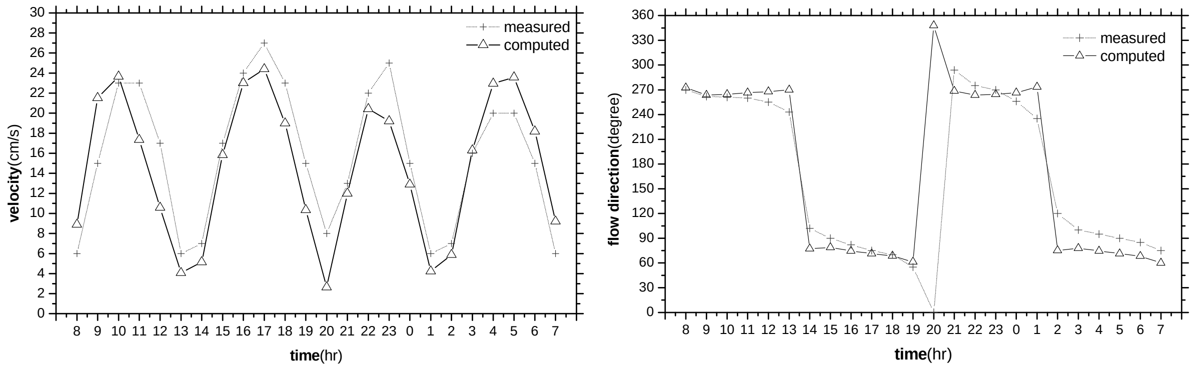

Figure 2 Computed and Observed Velocity and Phase at Station A

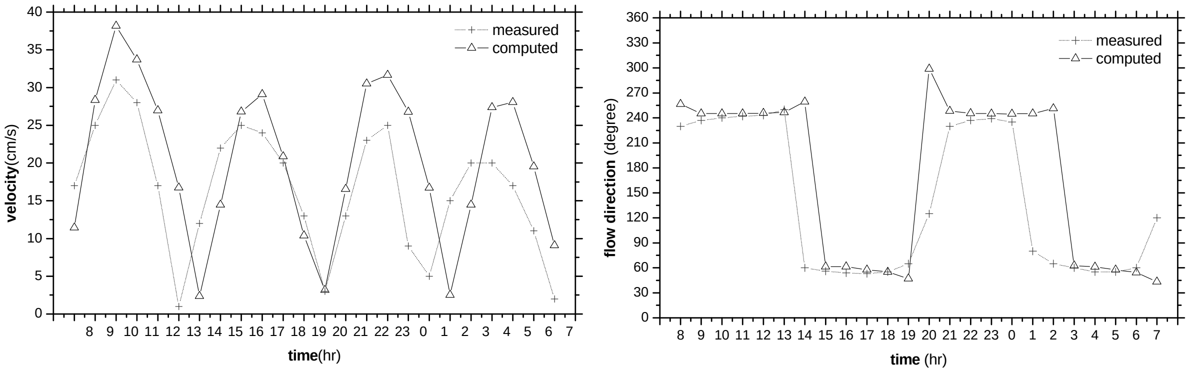

Figure 3 Computed and Observed Velocity and Phase at Station B

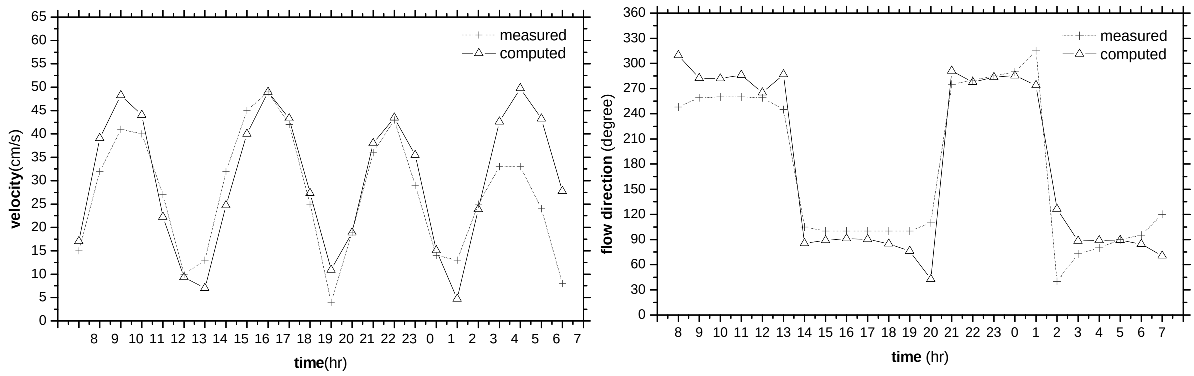

Figure 4 Computed and Observed Velocity and Phase at Station C

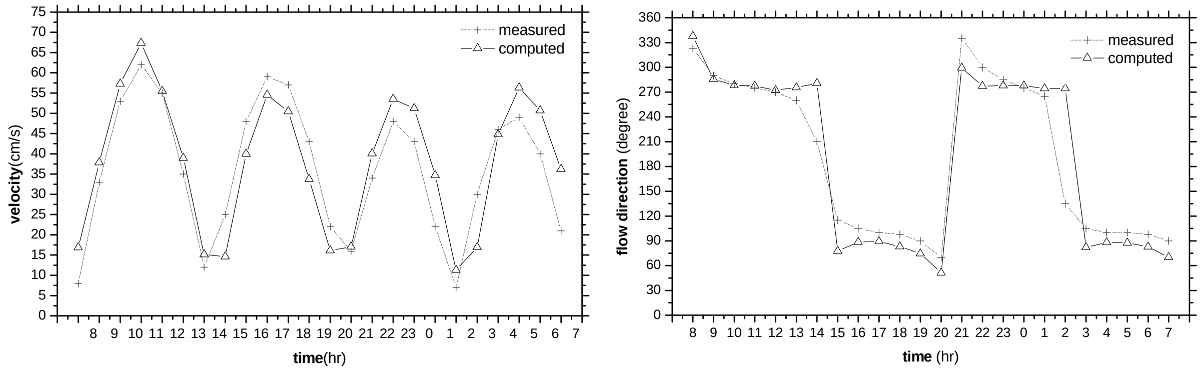

Figure 5 Computed and Observed Velocity and Phase at Station D

It can be seen that the agreement with the observation data is satisfactory. The seawater flows at pulsant turbulence, but the model in this paper is time-averaged and depth-averaged, so large warps of the phase appear at some point of the simulation periods is possible and acceptable, especially when flow direction changes (20:00 at station A for instance). What's more, flow field can be influenced by meteorological parameters-wind, air pressure, temperature etc. The model considers the regular and general instance, so the meteorological effect is ignored.

When the model is validated, transport equation can be used for temperature prediction. Conditions during neap tide and spring tide are simulated. Because of limited space, computed results only during neap tide are given in this paper. Figure 6 shows envelope area of iso-excess temperature during neap tide in summer, and figure7 shows envelope area of iso-excess temperature during neap tide in winter. Both are the worst status during 24 hours' simulation, which means that the temperature rise (drop) absolute value is maximal at the moment.

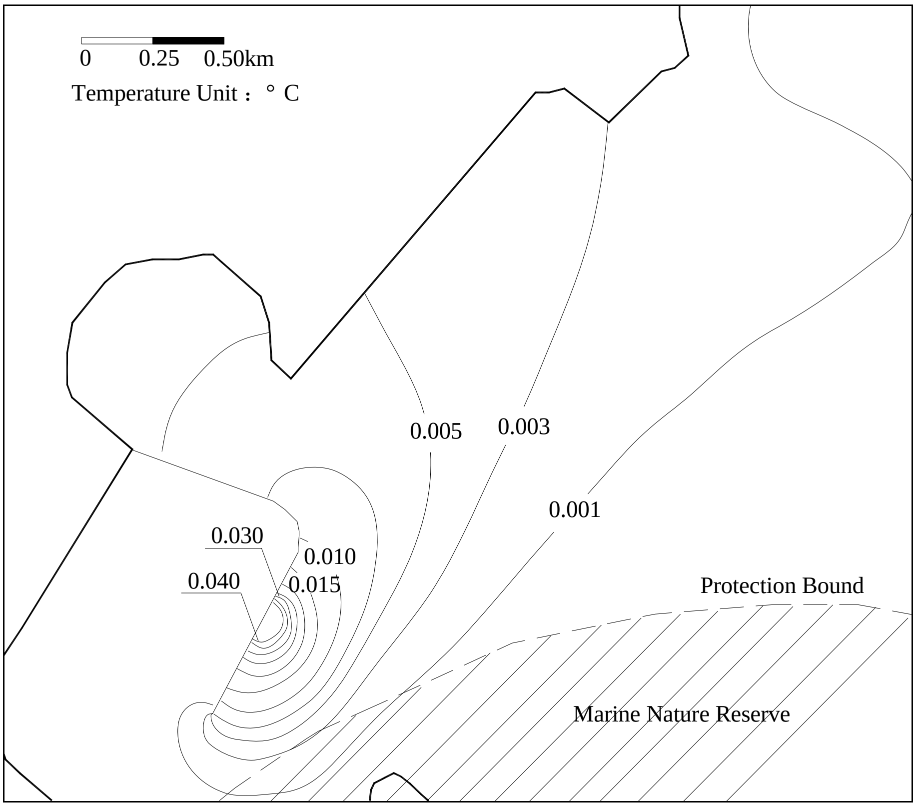

Figure 6 Iso-temperature elevation contours during neap tide in summer

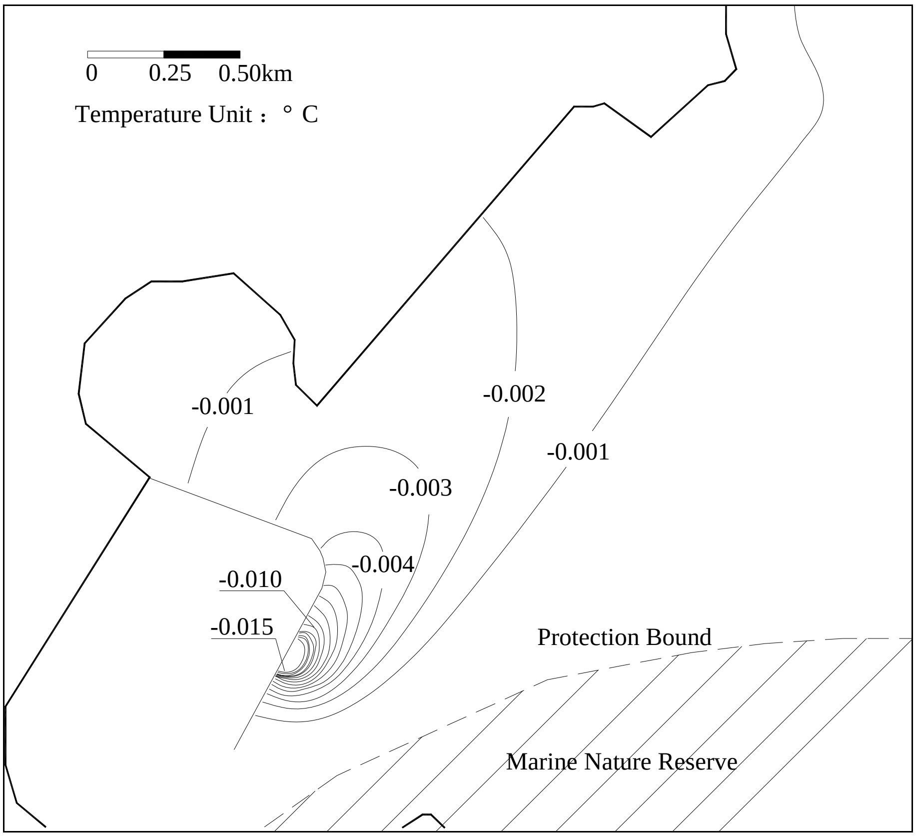

Figure 7 Iso-temperature elevation contours during neap tide in winter

In summer, the temperature rise of 0.003 ℃ would reach the marine nature reserve area (figure 6), and the maximal temperature rise is 0.040 ℃ at the discharge point. In winter temperature drop wouldn't affect the marine nature reserve (figure 7), and the maximal temperature drop is 0.015 ℃ at the discharge point. According to the Quality Standard for State Oceanic Water in China, the temperature change does not exceed prescriptive ranges.

CONCLUSIONS

The paper has presented the depth-averaged two-dimensional fluid dynamics model and its application to a planned SWHP system, which for the first time introduces the hydrodynamic knowledge into HVAC design application. Firstly the model is used for tide simulation, and it demonstrates in general an excellent agreement with the observation data. Then the model is well used for numerical prediction for the influence of temperature changes in the intake (outfall) areas of Xiaoping Island SWHP system.

The model is capable of being used as an environmental impact statement tool to aid in the design and control of SWHP system. It can also be used for the design of some other water source heat pump systems, such as Lake Source heat pump, River Source heat pump, etc.

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Design of Heat Pumps Systems using Natural Fluids

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SUMMARY

The presentation introduces new mathematical optimisation and design methodology for heat pump systems using natural fluids. A mathematical model is developed to capture thermodynamic and operating characteristics of dual-mode heat pump systems at different ambient temperatures. The multi-period optimisation framework has been developed to perform economic trade-off which minimises operating costs for heat pump, and to determine the capacity of heat pump systems. Case study is presented for demonstrating applicability of the developed methodology. Applications for energy saving in building have been considered.

INTRODUCTION

Due to increasing fuel costs and stricter environmental regulations, heat pump systems have gained great interests in these days as one of practical and realistic solutions in the near future to save energy consumptions as well as to reduce CO_2 emissions. Substituting a conventional boiler with a heat pump may save more than 50% of primary energy, while CO_2 emission from heat pump is lower than other fuel heating methods [1]. The appropriate sizing of system elements for heat pump applications is extremely important in order to achieve minimum cost at maintaining high thermodynamic efficiency. It is well known that the thermal performance of heat pump systems is directly related to its operating conditions, initial and operating costs. For residential air-source heat pumps, operating conditions varies with ambient conditions, which affect the performance of heat pump systems.

Therefore, the heat pump performance at the nominated design conditions should not be only criterion for assessing whether a heat pump is properly designed, as the heat pump is not operated at the design condition for all the time. Various researches [2, 3, 4, 5] have shown the interest the climate conditions on the optimal design of reversible heat pump systems, but there still lacks systematic studies on optimising dual-mode heat pump systems with simultaneously considering non-steady ambient conditions. By following the method [6] which is widely used in practice, a heat pump is selected such that its output balances the heat requirement of the building in most ambient conditions. This 'rule of thumb', however, does not guarantee a minimum cost solution. In this study, the integrated design of heat pump systems has been studied to address seasonal variations of ambient conditions and dual operation mode. First, a mathematical model for heat pump systems at designed conditions and sub-capacity conditions is developed. Then, the impact of compressor and heat exchanger capacity on the system capacity has been investigated to achieve cost-effective balancing the size of components, which leads to the reduction of operating cost. Variables characterising the system performance have been carefully chosen and optimised to improve economic and thermodynamic performance.

Case studies using two natural refrigerants (ammonia and propane) are carried out on the different geographic locations with different heat load, in order to illustrate benefits gained from the developed methodology.

HEAT PUMP SYSTEM MODELLING

The air-source heat pump system for residential heating and cooling normally includes five main components: compressor, indoor coil, outdoor coil, expansion valve and reversing valve (Fig. 1). Through different positions of reversing value, the heat pump system is operated in different modes, i.e. the indoor coil serves as the condenser in the heating mode, while the indoor coil is used as the evaporator in the cooling mode. This means that the heat pump extracts heat from outdoor air in the heating mode and rejects heat to outdoor air in the cooling mode.

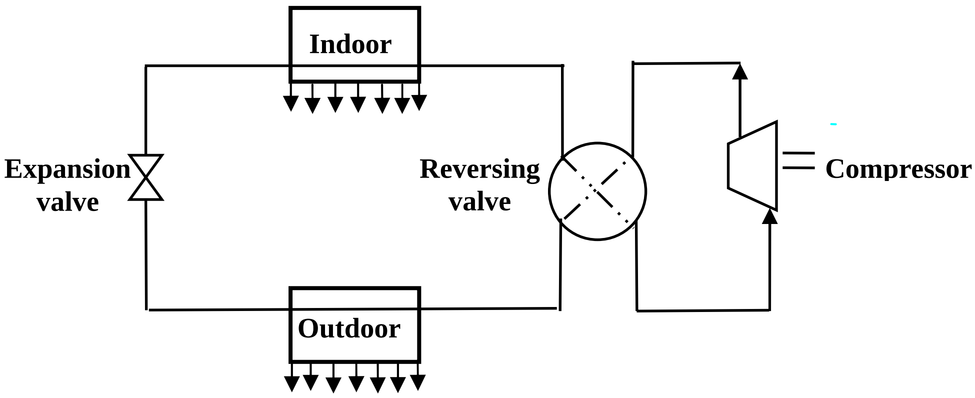

Fig. 1 Heat pump with heating and cooling modes

The important issue in the modelling of heat pump systems is that changes of ambient temperature and its effects on system capacity and power input to the compressor should be properly considered. Actual performance of a refrigeration cycle is different from that of ideal cycle because of the irreversibility occurred, for example, pressure drop of the refrigerant when it flows within the unit and temperature change from isentropic expansion and compression. Isentropic efficiency is a widely used index for efficiency of refrigeration compressors, especially for reciprocating compressors. The heat exchangers (i.e. evaporator and condenser) are assumed to have counterflow streams [7].

Compressor: A map-based method or an efficiency and loss method are mainly used in the modelling for the constant speed compressor. The former one can only be applied to the specified compressor when the performance data of compressor is provided by the compressor manufacturer. It is of higher precision, but difficult to extend to other types. In this study, it is adopted that the concept of the efficiency is used to predict the compressor performance. The compressor power is calculated by:

$$W_{in} = \frac{\dot{m}_r (h_3 - h_2)}{\eta_{isen}} \qquad (1)$$

$$H_{isen} = h_3 - h_2 \qquad (2)$$

where $\dot{m}_r$ is the refrigerant mass flow rate (kg/s), and η_{isen} is the isentropic efficiency.

As reciprocating and scroll compressors are widely used in heat pumps, in this study reciprocating compressor is chosen in the heat pump cycle.

For reciprocating compressor:

$$\dot{m}_r = \dot{V}_P \eta_v \rho_{suc} \tag{3}$$

where the volumetric efficiency η_v is the linear function of pressure ratio between compressor inlet and outlet pressure. There are many different functions to represent isentropic and volumetric efficiencies, but the following two equations are used in this study [8].

$$\eta_v = 98.6 - 5.6\gamma \tag{4}$$

$$\eta_{isen} = 0.109(\ln\gamma)^3 - 0.5247(\ln\gamma)^2 + 0.8577(\ln\gamma) + 0.3727 \tag{5}$$

Evaporator: In vapour compression refrigeration systems, the evaporator is an indirect-contact heat exchanger of the refrigerant and the air. Three equations for heat balance are used to represent the cooling capacity of the evaporator Q_e as following:

$$\begin{cases} Q_e = (UA)_e \Delta T_{e\,mean} \\ Q_e = (FCp)_e \Delta t_e = (FCp)_e (T_{e\,in} - T_{e\,out}) \\ Q_e = \dot{m}_r (h_2 - h_1) \end{cases} \tag{6}$$

where log mean temperature $\Delta T_{emean} = \dfrac{(T_{ein} - T_e) - (T_{eout} - T_e)}{\ln \dfrac{T_{ein} - T_e}{T_{eout} - T_e}}$

Condenser: The modelling of the heating capacity of the condenser Q_c is the same as for the evaporator

Expansion device: The expansion process is treated as isenthalpic.

Overall System: The following heat balance equation is used to link the evaporator, compressor and condenser together.

$$Q_c = Q_e + W = \dot{m}_r (h_2 - h_1) + \frac{\dot{m}_r (h_3 - h_2)}{\eta_{isen}} \tag{7}$$

The required physical properties of refrigerants i.e. enthalpy, density, and equilibrium pressure, can be calculated by the polynomial methods [9] regressed within the certain temperature rang or the commonly used EOS methods [10]. Heat pump efficiency is defined as the useful energy delivered, divided by the energy supplied to drive the compressor. Therefore in the heating mode and cooling mode, coefficient of performance (COP) is calculated by the following equation:

$$COP_{cooling} = \frac{Q_e}{W} \qquad COP_{heating} = \frac{Q_c}{W} \tag{8}$$

OPTIMISATION

The conventional design method for heat pumps are to choose the design condition, such as outdoor condition and indoor comport condition, to calculate heat load. Then sizing for components in the heat pump unit is made to meet heat load at the design condition so that the indoor comfort condition can be maintained. This design condition is referred as the design or balance point. When heat pump is operated at conditions other than at the design point, the heat extracted and rejected through the evaporator and condenser will be changed.

This will break the balance between heat exchangers and compressor, and the refrigerant in the heat pump cycle will adjust its evaporating and condensing temperature to reach a new balance point [9]. Below the balance point, the system has insufficient capacity to meet the demand, as shown in Fig. 2. This deficit should be supplemented from external energy source, such as an electric heater or a gas boiler. Above the balance point, the surplus capacity of the heat pump system can be stored, for example, using water or switching the system off periodically.

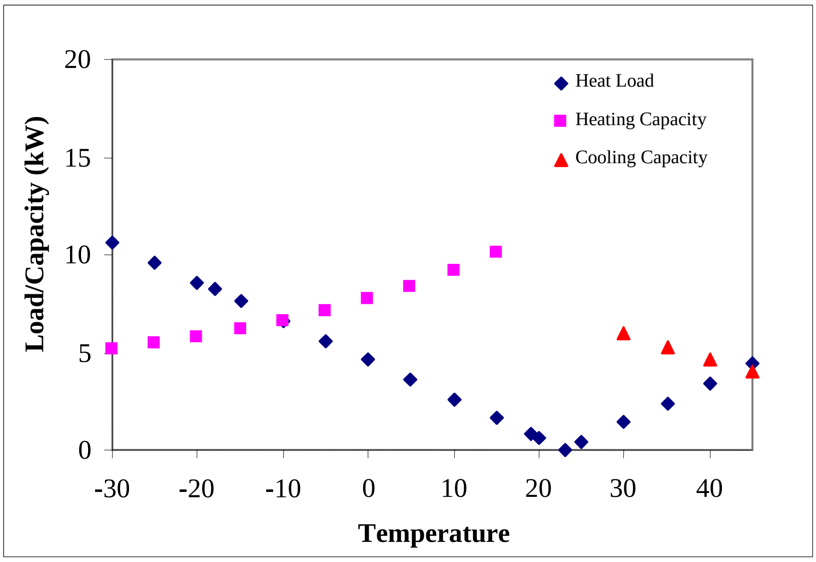

Fig. 2 Variation of system capacity with operating temperature

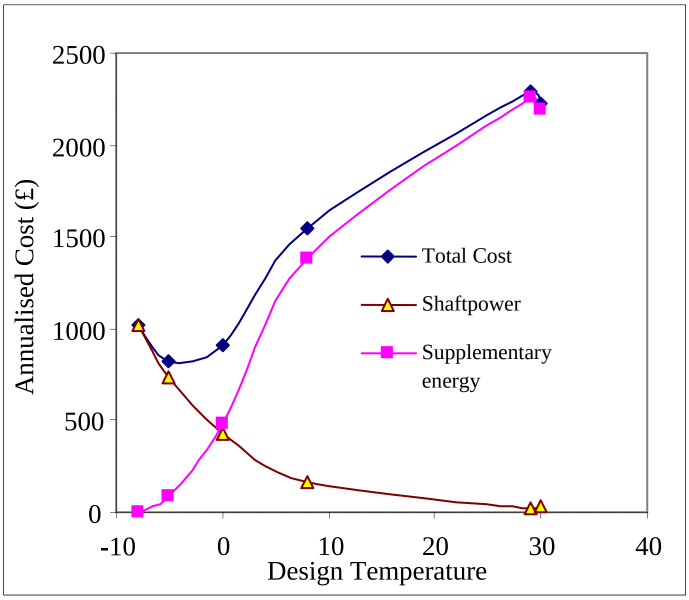

Fig. 3 Annualised energy cost at different design points

Given the geographical location and system on/off switch threshold, the heat pump is to be operated at different ambient temperature and consequently with different capacity and performance. If lower ambient temperature is chosen as the design condition, bigger capacity for heat pump system is required to meet higher heat demand, and less supplementary heating is necessary. On the other hand, the design temperature becomes higher, shaftpower required for compressor becomes less while the need for supplementary heating becomes higher. Fig. 3 shows trade-off between shaftpower and supplementary heating.

Heating and cooling load: The heat pump systems are to meet heating demand in the winter and cooling demand in the summer. Cooling load, Q_{load}^{C}, and heating load, Q_{load}^{H}, are defined in this study as the rate at which heat must be removed from and added to an indoor space so as to maintain a comfortable temperature.

In the heating mode, the condenser capacity Q_c is to meet the heating load, Q_{load}^{H}, while in the cooling mode, the evaporator capacity Q_e is to meet the cooling load Q_{load}^{C}. The general techniques used to calculate the heating/cooling load are TETD/TA method, Transfer Function Method (TFM), 1989 CLTD/CLF Method and Radiant Time Series (RTS) Method [11]. In this study, heat load (heating/cooling) is determined by the simple linear equation of ambient temperatures.

$$Q_{load} = A*[23 - T_a] \tag{9}$$

This is taken from the Canadian Building Digest [12], showing the heat demand of a residential house with modest insulation. Parameter A is 0.2.

Characterising variables of heat pump systems: For the heat pump system as a whole, system capacity is not determined purely by compressor capacity. The evaporator and condenser have their own effects on system capacity as well. Therefore, design variables should be selected carefully to characterise thermodynamic performance of units. In this study two characteristics are chosen to represent for heat exchangers.

Heat transfer capacity, *UA* (kW/K), combines the overall heat transfer coefficient and surface area of the heat exchanger, representing the effect of configuration on heat transfer. When heat transfer capacity *UA* of the evaporator and condenser is increasing, the system capacity is increasing accordingly; therefore the total cost is reduced. In overall, system capacity gain benefits from the increasing of UA of the evaporator and condenser.

Another characteristic of the air-sourced heat exchanger is air-side heat flow capacity, *FCp* (kW/K), which is used to measure how much heat is taken in and out by air flowing through the coils. It is assumed that the specific heat (*Cp*) for air is constant value (1 kJ/kg K) within the working temperature. System capacity is increasing while air side heat flow capacity is increasing, the total cost will be decreased as well.

Overall system performances are collectively influenced by sizing of compressor, evaporator and condenser. The design at minimum cost or high thermodynamic efficiency cannot be simply obtained from heuristics, experiences or engineering judgement. Optimisation should be carried out carefully to obtain the cost-effective design of heat pumps.

Objective function: The objective function is set to minimise annualised cost of shaftpower and supplementary energy cost.

$$\min \quad Cost = \sum_i W(i) \times D(i) \times P \; + \sum_i IH(i) \times (Q_{Load}(i) - Q_c(i)) \times D(i) \times P_{SH} \; + \\ \sum_i IC(i) \times (Q_{Load}(i) - Q_e(i)) \times D(i) \times P_{SH} \tag{10}$$

$$st. \begin{cases} T_{cin}(i) = T_a(i) \\ T_{ein}(i) = T_a(i) \\ IH(i) = 1 \quad if \quad T_a(i) \le 15 \\ IC(i) = 1 \quad if \quad T_a(i) \ge 25 \\ IH(i) + IC(i) \le 1 \end{cases}$$

where i is the index of the set, *IC* and *IH* are the integer variables for cooling mode and heating mode. The optimisation problem is formulated as discrete NLP problem, and the optimisation is solved in GAMS environment using CONOPT solver. It should be noted that in principle the objective function in Eq. 10 can include (annualised) capital cost, and then, economic trade-off between capital and operating costs would be performed during the optimisation.

CASE STUDY – HEAT PUMP HEATING A BUILDING

Fig. 4 shows that the monthly average high and low temperature over the whole year, and how many days at which temperature requiring heating and cooling for the case study. For simplicity in this paper, the case study illustrates the design of heat pump systems using ammonia as a fluid for one geographical location. It should be noted that the method is readily applicable to other fluids (e.g. propane) and other geographical locations with minor adjustments in the model.

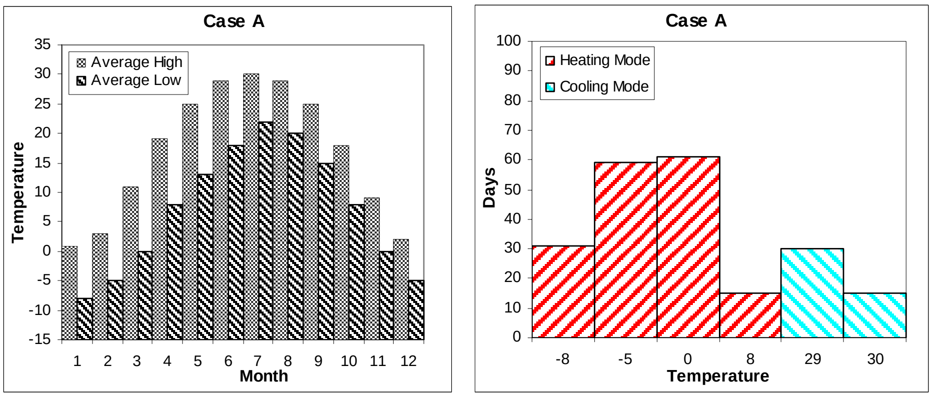

Fig. 4 Climate conditions and discrete multi-period conditions

Two scenarios are considered. Scenario 1 assumes the supplementary heat is provided by the electric heater with 100 % efficiency. Scenario 2 assumes the supplementary heat is provided by the gas boiler with 85 % efficiency. After optimisation, the annual (minimum) operating costs of the heat pump system for Scenario 1 and Scenario 2 are £524.3 and £493.4 respectively.

Fig. 5 shows the heat load and heat pump capacity. In both scenarios the heat pump cannot satisfy the heat demand at lower ambient temperature, which means supplementary heating is employed. COPs and shaftpower requirements at different operating conditions for two scenarios are given in Fig. 6, where COP for the heating mode is decreasing with the increasing of ambient temperature.

Expensive heating source (i.e. electricity in this case study) is used to provide supplementary heating in Scenario 1. Therefore heat pump system with relatively high capacity requiring low supplementary heating is able to achieve the minimum total operating cost, but COP is inevitably sacrificed. It is contrast to the Scenario 2, gas boiler can provide more supplementary heat for the minimum overall operating cost with high system COP.

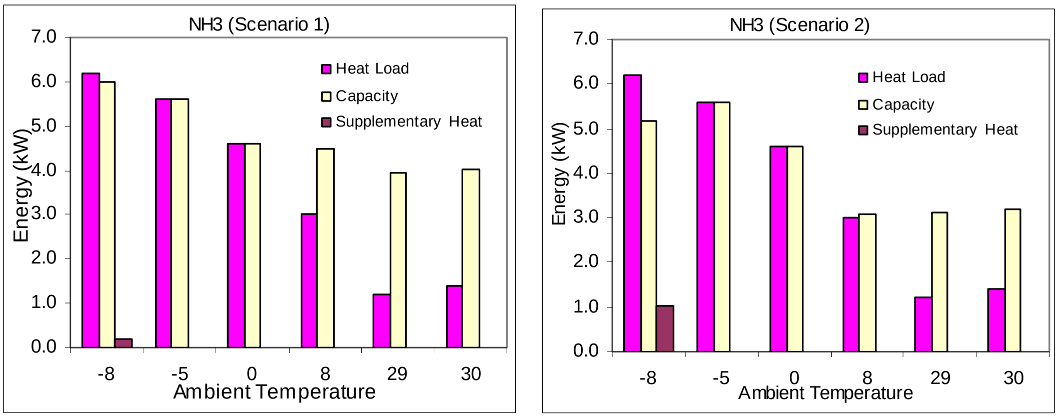

Fig. 5 System capacity and heat load: (Fluid: NH_3)

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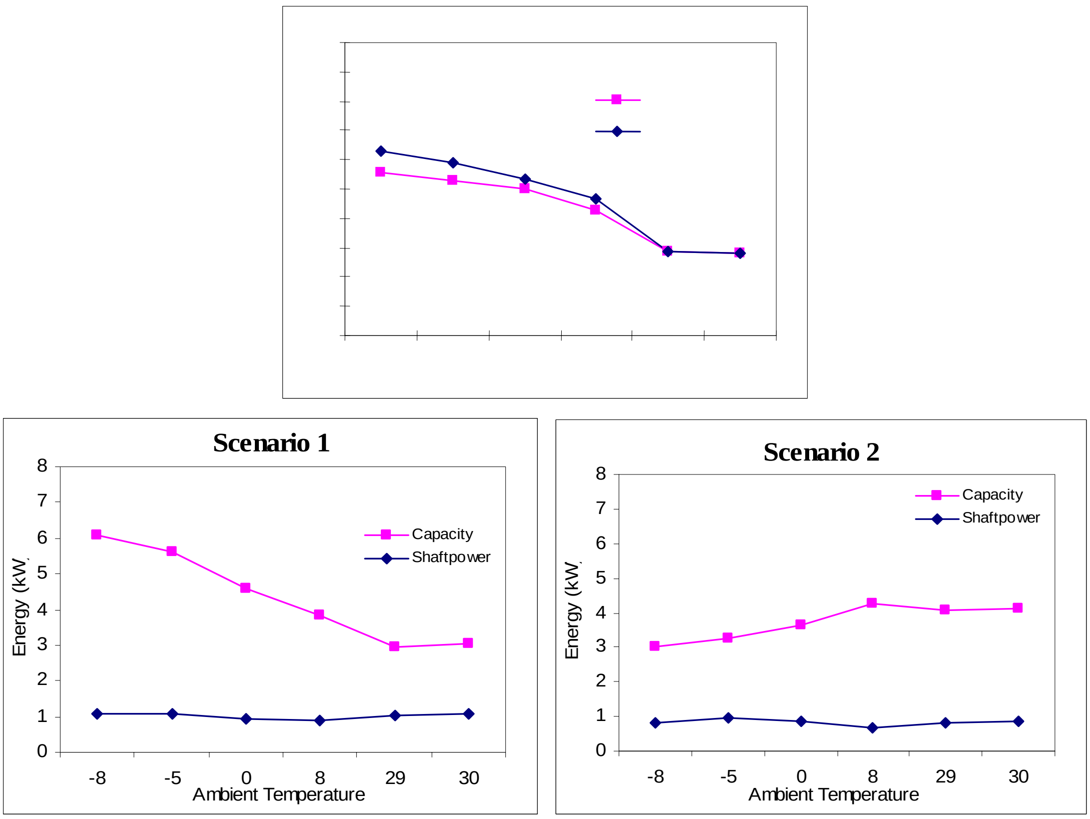

Fig. 6: COP and shaftpower (Fluid: NH_3)

CONCLUSION

In this study, the integrated design and optimisation method for heat pump systems are proposed to improve cost-effectiveness and thermodynamic efficiency of heat pump systems using natural fluids. The optimisation is based on multi-period formulation, in which different ambient conditions and its influences on heat pump performance are evaluated. During optimisation, selection of system capacity and sizing of units in heat pump systems are systematically investigated and screened, as well as economic trade-off between supplementary heating (or cooling) and operating cost for heat pump is performed to minimise annualised operating cost. Dual-mode (i.e. heating and cooling) operation and characteristics are also considered in the modelling and optimisation, which ensures that operating flexibility through dual-mode operation is fully explored in the design stage. The design approach is applied to a case study, which illustrates the developed method is able to bring significant benefits to the design and operation of heat pumps applications for buildings.

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Environmental Assessment Method for Seawater Source Heat Pump Systems

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SUMMARY

This paper presents a study on thermal diffusion of large-scale seawater source heat pump system (SWHP).

SWHP uses seawater as heat source and sink, and discharges the same quantity of water into local oceans after use. The cooling water systems may cause a temperature rise of 3-5℃ above ambient at discharge points which is close to the upper thermal tolerance limit of most marine biota, which would not only have a certain impact on the local marine environment, but also affect the local micro-climate, the local marine life and fishing economy, etc.

Based on Navier Stokes equations and shallow water assumption, non-linear shallow water equations are established in this paper. The two-dimensional numerical simulation research of the flow velocity field and temperature field of the water space where thermal effluent is discharged is done using a case study.

And following an example of Xiaoping Island SWHP project is given. Simulation and description on the discharge of this SWHP project show that, the temperature rise/drop extends to certain area, but the temperature elevation of the influence area is far less than the standard of Grade Ⅰ of Quality Standard for State Oceanic Water.

This paper firstly introduces the hydrodynamic knowledge into HVAC design application, which shows a way to the environmental impact statement of Water Source Heat Pumps.

INTRODUCTION

In the 21st century, the energy issue has become increasingly prominent. The sea water-source heat pump (SWHP) system’s coming into China is a very important significance for the energy efficiency in buildings. Seawater source heat pump technology transfers solar energy captured in seawater which is low temperature and low quality, into high quality energy through the use of heat pump and a small quantity of electricity. At present, there’re only several experimental projects in China, and environmental impact assessment is one of the most important issues to resolve.

SWHP system uses seawater as heat source and sink, and discharges the same quantity of water into local oceans after use. The temperature of the discharge water is higher in the summer than the local water temperature around 10℃, and lower in winter about 3-5℃ than the receiving water body. The discharge temperature elevation of seawater will not only have a certain impact on the local marine environment, but also affect the local micro-climate, the local marine life and fishing economy, etc. When the large-scale application of SWHP systems come into reality in the coastal cities in China for district cooling and heating, these impacts may become particularly prominent.

The above issue can be solved by the thermal discharge research method. Thermal discharge exists in every thermal/nuclear power plant, and researchers have been working on it for tens of years. The method consists of domestic physical and numerical simulation, of which the later is most widely used. What's more, the two are often combined, for the model the model parameters of numerical equation are often obtain from physical tests.

According to different classification method, numerical model can be classified as the near-field model and the far-field model, or the 2D depth-averaged model and 3D mode, and during actual applications these models are developing toward a more sophisticated, visualization and intelligent direction. With the emergence of the sea water-source heat pump system, some new problems need to be considered.

Between the mathematical models studied by scholars at home and abroad, the depth-averaged two-dimensional fluid dynamics model is very effective to solve some problems in the oceans, coasts water level and instantaneous transport, which is widely used in the study of thermal discharge, to predict the temperature distribution of thermal discharge area[1,2].

MATHEMATIC MODEL

Based on Navier Stokes equations and shallow water assumption, non-linear shallow water equations are established. The depth averaged two-dimensional hydrodynamic model, including the effects of earth's rotation, bottom friction, wind shear and surface heat exchange are as follows [3]:

Continuity equation:

$$\frac{\partial \zeta}{\partial t}+\frac{\partial HU}{\partial x}+\frac{\partial HV}{\partial y}=Q \tag{1}$$

Momentum equation:

$$\frac{\partial U}{\partial t}+U\frac{\partial U}{\partial x}+V\frac{\partial U}{\partial y}-fV=-g\frac{\partial \zeta}{\partial x}+\nu_H\left(\frac{\partial^2 U}{\partial x^2}+\frac{\partial^2 U}{\partial y^2}\right)-\frac{gU\sqrt{U^2+V^2}}{HC_{2d}^2}+\frac{C_d}{H}\frac{\rho_a}{\rho}U_{10}^2\cos\beta \tag{2}$$

$$\frac{\partial V}{\partial t}+U\frac{\partial V}{\partial x}+V\frac{\partial V}{\partial y}+fU=-g\frac{\partial \zeta}{\partial x}+\nu_H\left(\frac{\partial^2 V}{\partial x^2}+\frac{\partial^2 V}{\partial y^2}\right)-\frac{gV\sqrt{U^2+V^2}}{HC_{2d}^2}+\frac{C_d}{H}\frac{\rho_a}{\rho}U_{10}^2\sin\beta \tag{3}$$

Transport equation:

$$\frac{\partial HT}{\partial t}+\frac{\partial HUT}{\partial x}+\frac{\partial HVT}{\partial y}=\frac{\partial}{\partial x}\left(HK_x\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(HK_y\frac{\partial T}{\partial y}\right)+S \tag{4}$$

U、V depth-averaged velocity in x and y direction; ρ_a density of air ; f Coriolis coefficient, $f=2\omega\sin\psi$, ω angular frequency of the earth, ψ the local latitude; 2D Chézy coefficient, computed by Manning Formula : $C_{2d}=\frac{1}{n}H^{1/6}$, n is the Manning's coefficient; C_d wind drag coefficient; ν_H horizontal eddy viscosity; U_{10} averaged wind speed at 10 m above free surface; β the angle between the wind stress vector and the local direction of the grid line x; M_x, M_y source or sink of momentum in x and y direction; T the water temperature; K_x, K_y diffusion coefficient in x and y direction; Q, S the source and sink term.

The heat exchange flux is represented by a bulk exchange formula [4]:

$$Q_{tot} = -\lambda(T_s - T_{back}) \tag{5}$$

with T_s the water temperature at the free surface and T_{back} the natural background temperature, both in ℃. The heat exchange coefficient λ is a function of the surface temperature T_s and the wind speed U_{10}:

$$\lambda = 4.48 + 0.049T_s + f(U_{10})\left(1.12 + 0.018T_s + 0.00158T_s^2\right) \tag{6}$$

$$f(U_{10}) = (3.5 + 2.0U_{10})\left(\frac{5.0\times10^6}{S_{area}}\right)^{0.05} \tag{7}$$

Where S_{area} is the exposed water surface in m^2.

The control equations formed by ADI method (Alternating Direction Implicit Method, Leendertse，1967) are solved on a staggered grid. Each of the dispersed control equation group consists of three unknown variables at adjacent nodes. In the coefficient matrix, all the non-zero elements are arranged along a diagonal matrix. So triangle diagonal matrix algorithm (TDMA) is used for solution.

PROJECT AND COMPUTATIONAL ASPECTS

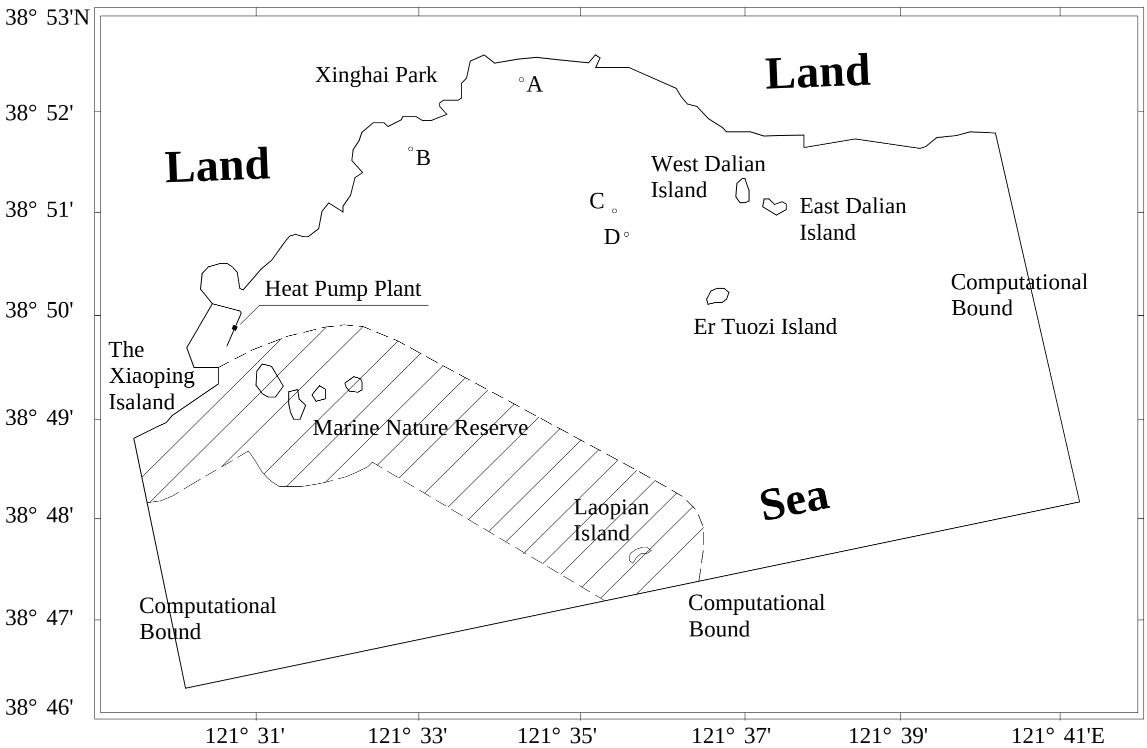

Figure1 Simulation Area and Location of the Project

Using the above method, an example of Xiaoping Island SWHP project is given in this section. This project is located to the Xiaoping Island, at the Huang hai sea of China. The heat pump plant is located at 121°30′34″-121°30′44″E, 38°49′49″-38°49′58″N, on a bulwark of 1300 meters long and L type, see figure 1. The shadow area is the marine nature reserve, which is to be considered not to be affected by this project's discharge.

The design input/output flow and temperature rise/drop at the first stage of this SWHP is as follows:
Heating Season: 1.9 m^3/s, temperature drop 2.5℃
Cooling Season: 1.14m^3/s， temperature rise 10.0℃

RESULTS AND DISCUSSION

24 hours' site tide data of four stations-A, B, C, D（From 8:00, 18th, Apr. to 7:00, 19th, Apr., 2003, sees figure 1）are used for validation of the simulation results. The comparison between the computed and observed results are shown from figure 2 to figure 5。

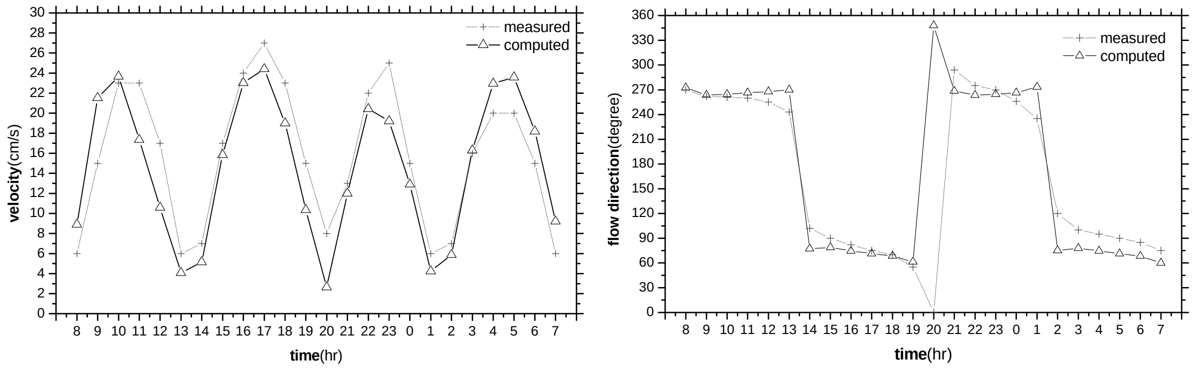

Figure 2 Computed and Observed Velocity and Phase at Station A

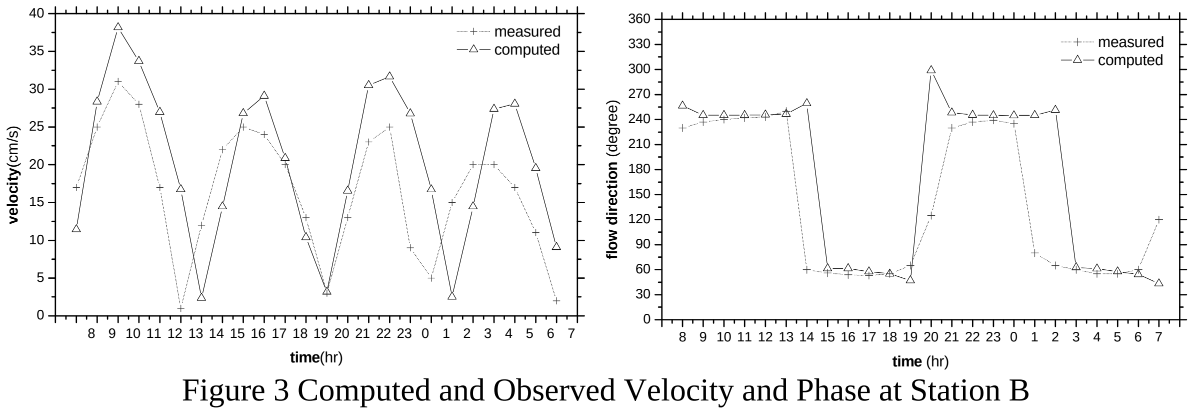

Figure 3 Computed and Observed Velocity and Phase at Station B

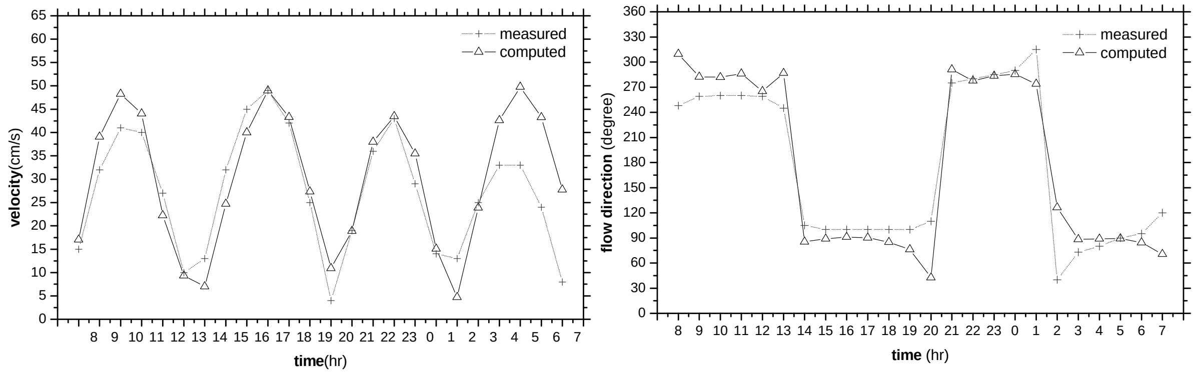

Figure 4 Computed and Observed Velocity and Phase at Station C

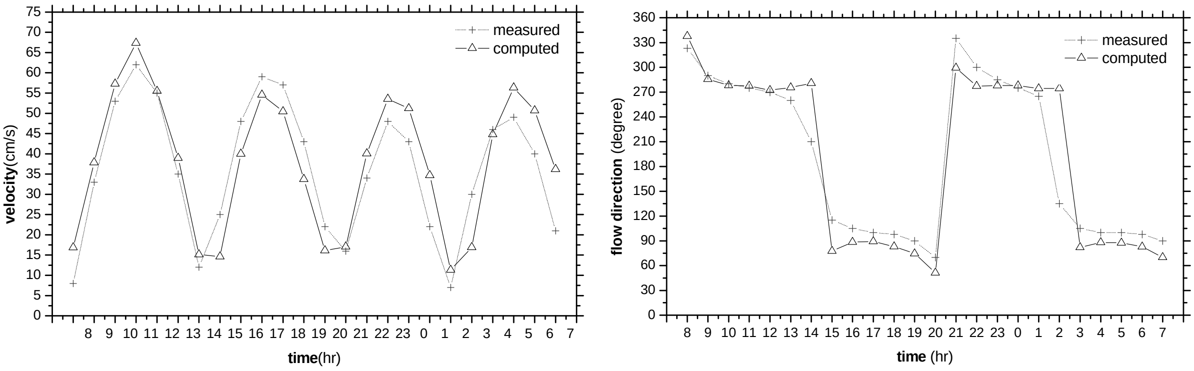

Figure 5 Computed and Observed Velocity and Phase at Station D

It can be seen that the agreement with the observation data is satisfactory. The seawater flows at pulsant turbulence, but the model in this paper is time-averaged and depth-averaged, so large warps of the phase appear at some point of the simulation periods is possible and acceptable, especially when flow direction changes (20:00 at station A for instance). What's more, flow field can be influenced by meteorological parameters-wind, air pressure, temperature etc. The model considers the regular and general instance, so the meteorological effect is ignored.

When the model is validated, transport equation can be used for temperature prediction. Conditions during neap tide and spring tide are simulated. Because of limited space, computed results only during neap tide are given in this paper. Figure 6 shows envelope area of iso-excess temperature during neap tide in summer, and figure7 shows envelope area of iso-excess temperature during neap tide in winter. Both are the worst status during 24 hours' simulation, which means that the temperature rise (drop) absolute value is maximal at the moment.

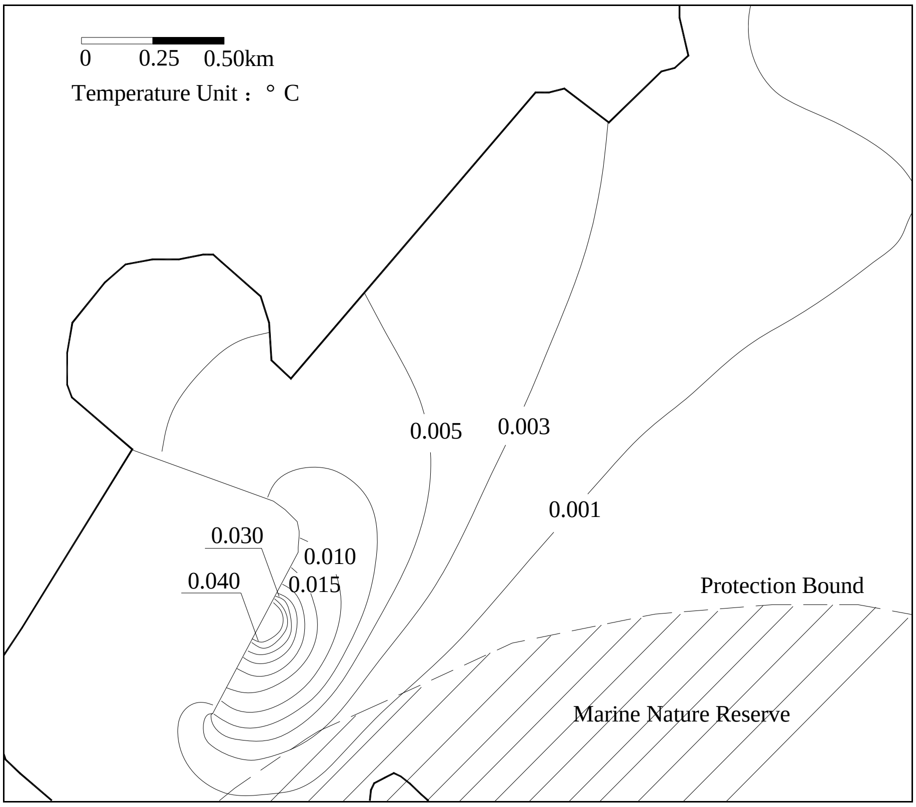

Figure 6 Iso-temperature elevation contours during neap tide in summer

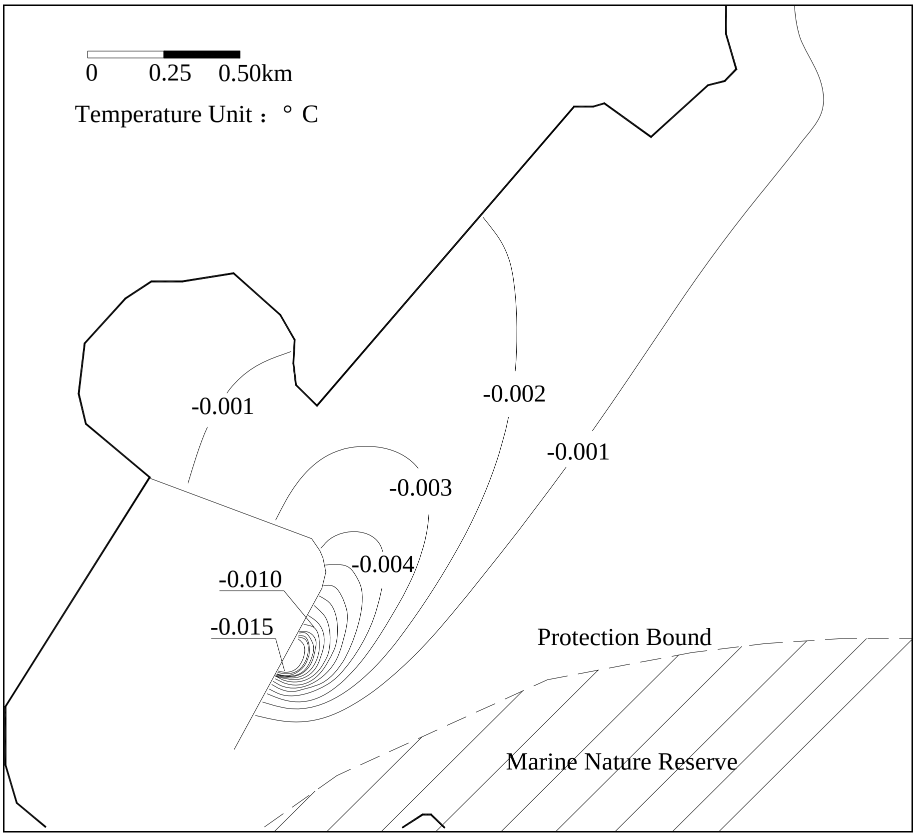

Figure 7 Iso-temperature elevation contours during neap tide in winter

In summer, the temperature rise of 0.003 ℃ would reach the marine nature reserve area (figure 6), and the maximal temperature rise is 0.040 ℃ at the discharge point. In winter temperature drop wouldn't affect the marine nature reserve (figure 7), and the maximal temperature drop is 0.015 ℃ at the discharge point. According to the Quality Standard for State Oceanic Water in China, the temperature change does not exceed prescriptive ranges.

CONCLUSIONS

The paper has presented the depth-averaged two-dimensional fluid dynamics model and its application to a planned SWHP system, which for the first time introduces the hydrodynamic knowledge into HVAC design application. Firstly the model is used for tide simulation, and it demonstrates in general an excellent agreement with the observation data. Then the model is well used for numerical prediction for the influence of temperature changes in the intake (outfall) areas of Xiaoping Island SWHP system.

The model is capable of being used as an environmental impact statement tool to aid in the design and control of SWHP system. It can also be used for the design of some other water source heat pump systems, such as Lake Source heat pump, River Source heat pump, etc.

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Exergetic performance assessment of gas engine heat pumps

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SUMMARY

A gas heat pump (GHP) or a gas engine-driven heat pump (GEHP) is a new type of heat pump (one of today's promising new technologies). In evaluating the efficiency of heat pump (HP) systems, the most commonly used measure is the energy (or first law) efficiency, which is modified to a coefficient of performance (COP) for HP systems. However, for indicating the possibilities for thermodynamic improvement, energy analysis is inadequate and exergy analysis, which has been recently widely used in the design and performance evaluation of thermal systems, is needed. This study, which is one of the limited studies, presents exergy analysis and assessment of GHPs aimed at identifying improvement potential. An illustrative example is also presented. The exergy destructions in each of the components of the overall system are determined. Exergy efficiencies of the system components are also derived to assess their performances and to elucidate potentials for improvement.

INTRODUCTION

A gas heat pump (GHP) is a new type of heat pump (one of today's promising new technologies) in which an open type compressor (the core part) is driven by a gas engine. The GHP has been paid more attention in the heating, ventilation and air conditioning (HVAC) field in recent years due to its advantage of reducing the electric consumption in the cooling and heating seasons. Another two distinguished advantages of the GHP are (1) the ability to recover the waste heat released by the engine cylinder jacket and exhaust gas in the heating mode and (2) the easy modulation of compressor speed by adjusting the gas supply. Therefore, the GHP has a different performance from that of the electric driven heat pump, especially in the heating mode [1].

Heat pump is known as a kind of energy saving air conditioning equipment, which is mostly driven by electricity. Current practice is mainly to convert fuel to electric power at a central power plant and reject the waste heat to the environment. The electricity is then transmitted to the heat pump motor, which provides the work input to the heat pump cycle. Significant losses will occur from fuels to work of the heat pumps in such a case. An obvious way to improve overall fuel utilization is to locate the fuel conversion process closer to, where the exhaust heat can be productively used. A combustion engine directly driving a heat pump compressor has about the same overall fuel-to-heat or cooling efficiency as an electric unit. An additional advantage is that the combustion engine's waste heat can be utilized at the site to provide supplemental space heating or water heating. Therefore, GHP will have a high efficiency of fuel utilization. At this time no more than 1/3 of energy consumed in a conventional heat pump is needed in GHP to achieve the same heating effect, as shown in Figure 1 [2,3].

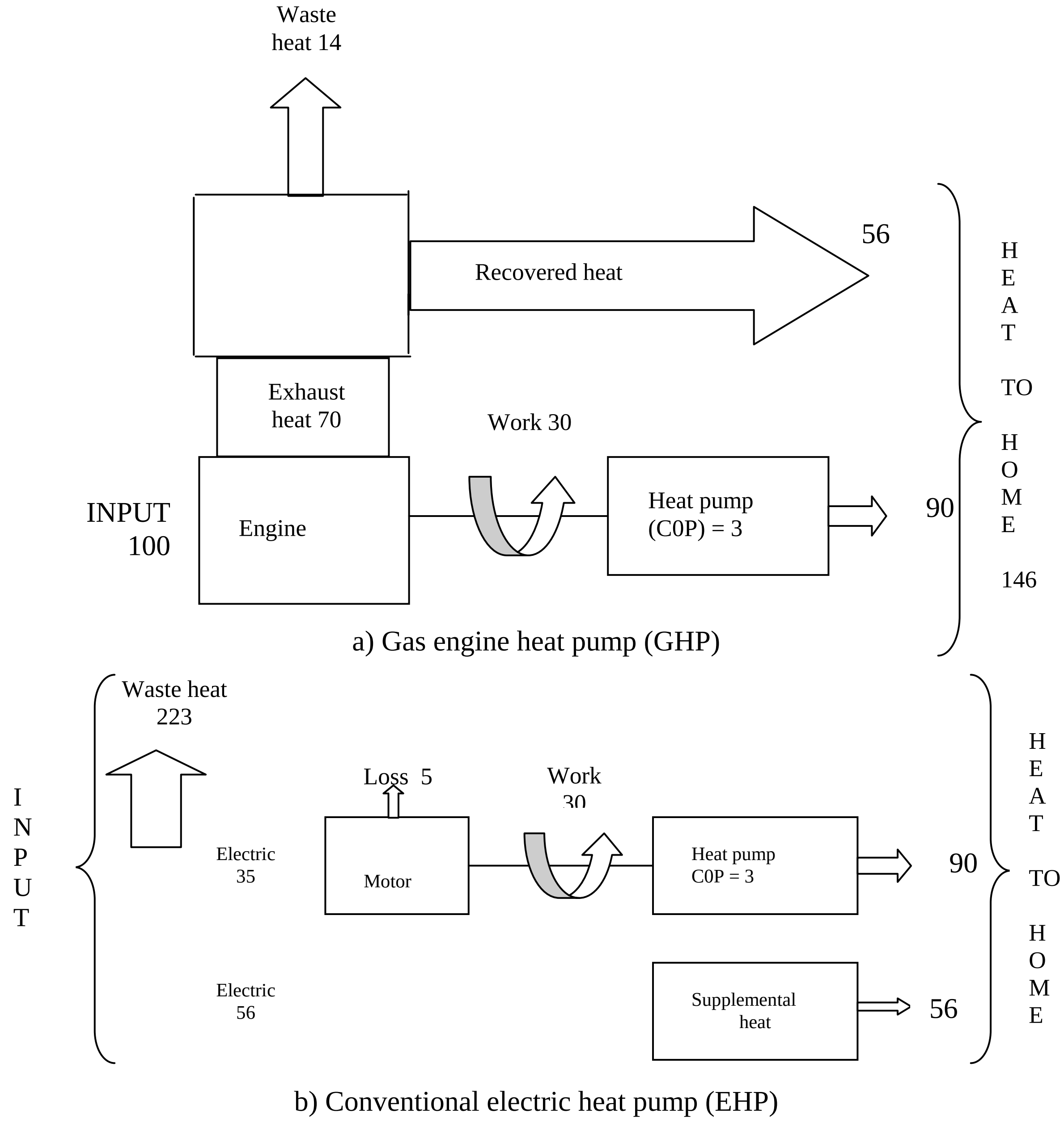

Figure 1. Comparison of energy conversion process between conventional EHP and GHP [3].

GHPs were first launched in Japan in 1987. There were 400,000 units installed worldwide in 2003, with 50,000 annual sales in Japan. The range of gas engine heat pumps, called GHPSplit, are reversible units with capacities 33 kW to 73 kW for heating and 28 kW to 56 kW for cooling [4]. Early examples of GHPs were unreliable, noisy and had a high maintenance requirements. Barriers to their introduction was overcome by research, development and support from gas utilities and governmental organizations [5].

Up to date, various studies on GHPs have been conducted by many investigators. Some of these studies include energetic performance evaluation [1], combination of GHP and water-loop heat pump [3], economic analysis [6,7], cascade fuzzy control [8] and combined refrigeration cycles driven by GHPs [9]. Although various studies on energetic and economic assessment of GHPs have been performed, no detailed studies on exergetic modelling and assessment of GHPs appear in the open literature to the best of the author's knowledge. This has provided the motivation in doing the present study.

METHODS

In the performance evaluation of GHP, energy and exergy analysis methods are used. In this regard, the GHP system may be modeled using the following relations, which are related to the heat pump unit and the whole GHP system. .

General energy and exergy balance equations

General mass, energy, entropy and exergy balance equations are given in more detail elsewhere [10].

The mass balance equation can be expressed in the rate form as

$$\sum \dot{m}_{in} = \sum \dot{m}_{out}, \ (1)$$

where $\dot{m}$ is the mass flow rate, and the subscript in stands for inlet and out for outlet.

The general energy balance can be written as

$$\sum \dot{E}_{in} = \sum \dot{E}_{out} \ \text{or} \ \dot{Q} + \sum \dot{m}_{in} h_{in} = \dot{W} + \sum \dot{m}_{out} h_{out}, \ (2)$$

where $\dot{m}$ is the mass flow rate and the subscript *in* stands for inlet and *out* for outlet, $\dot{E}_{in}$ is the rate of net energy transfer in, $\dot{E}_{out}$ is the rate of net energy transfer out by heat, work and mass, $\dot{Q} = \dot{Q}_{net,in} = \dot{Q}_{in} - \dot{Q}_{out}$ is the rate of net heat input, $\dot{W} = \dot{W}_{net,out} = \dot{W}_{out} - \dot{W}_{in}$ is the rate of net work output, and *h* is the enthalpy per unit mass.

The general exergy balance can be expressed in the rate form as

$$\sum \dot{E}x_{in} - \sum \dot{E}x_{out} = \sum \dot{E}x_{dest} \ \text{or} \ \sum \left(1 - \frac{T_0}{T_k}\right)\dot{Q}_k - \dot{W} + \sum \dot{m}_{in}\psi - \sum \dot{m}_{out}\psi = \dot{E}x_{dest}, \ (3)$$

$$\psi = (h - h_0) - T_0(s - s_0), \ (4)$$

where $\dot{Q}_k$ is the heat transfer rate through the boundary at temperature T_k at location *k*, $\dot{W}$ is the work rate, ψ is the flow (or specific) exergy, *s* is the specific entropy and the subscript zero indicates properties at the dead state of P_0 and T_0,

The exergy destroyed or the irreversibility may be expressed as follows

$$\dot{I} = \dot{E}x_{dest} = T_0 \dot{S}_{gen}, \ (5)$$

where $\dot{S}_{gen}$ is the rate of entropy.

Van Gool's exergetic improvement potentia in the rate form, denoted $I\dot{P}$, is given by [11,12]

$$I\dot{P} = (1-\eta)(\dot{E}x_{in} - \dot{E}x_{out}) , (6)$$

General energy and exergy efficiency relations

The energy-based efficiency measure of electrical HP unit (COP_{HP}) is calculated as follows [13]:

$$COP_{HP} = \frac{\dot{Q}_{cond}}{\dot{W}_{comp,elec}} , (7)$$

The COP for the GHP is calculated from [14,15]

$$COP_{GHP} = \eta_{motor,eff}\, COP_{HP} + a_{motor,u} , (8)$$

or

$$COP_{GHP} = \eta_{mech}\, COP_{HP} + (\eta_{total} - \eta_{mech}) , (9)$$

where $\eta_{motor,eff}$ is the effective (mechanical) efficiency of the gas engine, $a_{motor,u}$ is the share of gas engine useful waste heat in the primary energy utilization, η_{mech} is the mechanical efficiency of the gas engine, and $\eta_{motor,eff}$ is the total efficiency of the gas engine.

The exergetic coefficient of performance of electrical HP unit is calculated by

$$COP_{ex,HP} = \frac{\dot{Q}_{cond}(1 - \frac{T_0}{T_{cond}})}{\dot{W}_{comp,elec}} , (10)$$

The exergy efficiencies of the electrical HP and GHP are obtained, respectively, from

$$\varepsilon_{HP} = \frac{\dot{E}x_{heat}}{\dot{W}_{comp,elec}} = \frac{\dot{E}x_{in,condenser} - \dot{E}x_{out,condenser}}{\dot{W}_{comp,elec}} , (11)$$

$$\varepsilon_{GHP} = \frac{\dot{E}x_{in,condenser} - \dot{E}x_{out,condenser}}{\dot{E}x_{fuel}} , (12)$$

RESULTS

Description of an illustrative example

The general relations given in the previous section are applied to an air/water GHP shown in Figure 2 [6], which is described by Lazzarin and Noro [6] as follows: ''S. Nicola'' HVAC plant in Vicenza, Italy features innovative and significant energy savings characteristics. The nominal cooling power is 275 kW (22 Nm^3/h of natural gas is the nominal fuel consumption), while in heating mode the output power is 380 kW (19 Nm^3/h fuel consumption). These performances are labelled for summer external air 35°C and evaporator input/output 12/7°C; winter external air 10°C and condenser input/output 40/45°C. Heat recovery is taken from the lubricating oil, engine cooling water and partly from the exhaust. The nominal power thus recovered is 109 kW in heating mode and 127 kW in cooling mode, to produce hot water at about 70°C. The energy balance of the system is reported as relative units (primary energy input = 100) and in absolute values (kW) for heating mode. Two hydraulic circuits are provided. The primary one supplies chilled water in summer (7/12°C) and warm water in winter (45/40°C) produced by the heat pump and the secondary one hot water at 70°C given by the heat recovery. The control system is governed by a microprocessor that commands the engine speed regulator and the compressor cylinders chocking.

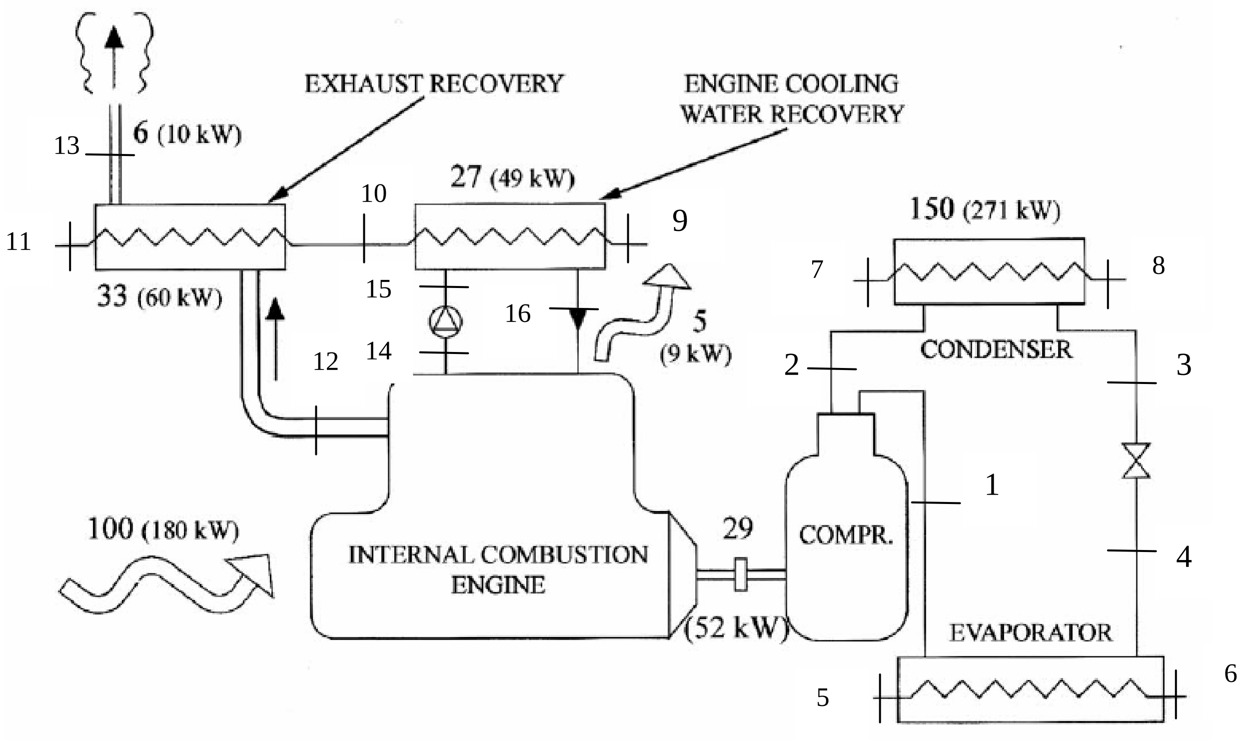

Figure 2. Schematic diagram of the gas engine heat pump studied [6]

Energetic and exergetic assessment of gas engine

The energetic and exergetic evaluation of the gas engine has been given in more detail elsewhere [16], while its exergy efficiency may be calculated using the following relation

$$\varepsilon_{engine} = \frac{\dot{W}_{eff}}{\dot{E}x_{fuel}} \text{ , (13)}$$

where $\dot{E}x_{fuel}$ is the exergy of the fuel, which may be found using the relations given in Ref. [17].

Energetic and exergetic assessment of GHP components

The GHP consists of a HP unit (compressor, condenser, evaporator and expansion valve), a gas engine and heat exchangers (i.e., exhaust recovery and engine cooling recovery), as shown in Figure 2. The energy and exergy balance (exergy destruction) relations for the components of a ground-source HP system have been derived for the author elsewhere [18,19], while they are obtained along with their exergy efficiencies for an air/water GHP shown in this figure as follows:

Compressor (I):

$$\dot{m}_1 = \dot{m}_{2,s} = \dot{m}_{act,s} = \dot{m}_r \text{, (14a)}$$

$$\dot{W}_{comp} = \dot{m}_r (h_{2,act} - h_1) \text{, (14b)}$$

$$\dot{E}x_{dest,comp} = \dot{m}_r (\psi_1 - \psi_{2,act}) + \dot{W}_{comp,elec} \text{, (14c)}$$

$$\varepsilon_{comp} = \frac{\dot{E}x_{2,act} - \dot{E}x_1}{\dot{W}_{comp,elec}} \text{, (14d)}$$

where heat interactions with the environment are neglected.

Condenser (II):

$$\dot{m}_2 = \dot{m}_3 = \dot{m}_r \text{, (15a)}$$

$$\dot{m}_7 = \dot{m}_8 = \dot{m}_w \text{, (15b)}$$

$$\dot{Q}_{cond} = \dot{m}_r (h_{2,act} - h_3) \text{, (15c)}$$

$$\dot{Q}_{cond} = \dot{m}_w C_{p,w} (T_8 - T_7) \text{, (15d)}$$

$$\dot{E}x_{dest,cond} = \dot{m}_r (\psi_{2,act} - \psi_3) + \dot{m}_{w,fh} (\psi_7 - \psi_8) \text{, (15e)}$$

$$\varepsilon_{cond} = \frac{\dot{E}x_8 - \dot{E}x_7}{\dot{E}x_{2,act} - \dot{E}x_3} = \frac{\dot{m}_w (\psi_8 - \psi_7)}{\dot{m}_r (\psi_{2,act} - \psi_3)} \text{, (15f)}$$

Expansion (throttling) valve (III):

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_r \text{, (16a)}$$

$$h_3 = h_4 \text{, (16b)}$$

$$\dot{E}x_{dest,exp} = \dot{m}_r (\psi_3 - \psi_4) \text{, (16c)}$$

$$\varepsilon_{exp} = \frac{\dot{E}x_4}{\dot{E}x_3} = \frac{\psi_4}{\psi_3} \text{, (16d)}$$

Evaporator (IV):

$$\dot{m}_4 = \dot{m}_1 = \dot{m}_r \text{, (17a)}$$

$$\dot{m}_5 = \dot{m}_6 = \dot{m}_{air} \text{, (17b)}$$

$$\dot{Q}_{evap} = \dot{m}_r (h_1 - h_4) \text{, (17c)}$$

$$\dot{Q}_{evap} = \dot{m}_{air} C_{air,p} (T_6 - T_5) \text{, (17d)}$$

$$\dot{E}x_{dest,evap} = \dot{m}_r (\psi_4 - \psi_1) + \dot{m}_{air} (\psi_6 - \psi_5) \text{, (17e)}$$

$$\varepsilon_{evap} = \frac{E\dot{x}_6 - E\dot{x}_5}{E\dot{x}_4 - E\dot{x}_1} = \frac{\dot{m}_{w,prt9} (\psi_6 - \psi_5)}{\dot{m}_r (\psi_4 - \psi_1)} \text{, (17f)}$$

Heat exchanger for engine cooling water recovery (V):

$$\dot{m}_9 = \dot{m}_{10} = \dot{m}_{recovery,w} \text{, (18a)}$$

$$\dot{m}_{15} = \dot{m}_{16} = \dot{m}_{cooling,w} \text{, (18b)}$$

$$\dot{Q}_{cooling,w} = \dot{m}_{cooling,w} (h_{15} - h_{16}) \text{, (18c)}$$

$$\dot{Q}_{recovery,w,I} = \dot{m}_{recovery,w} C_{recovery,w,,p} (T_{10} - T_9), \text{ (18d)}$$

$$\dot{E}x_{dest,engine\ cooling} = \dot{m}_{cooling,w} (\psi_{15} - \psi_{16}) + \dot{m}_{recovery,w} (\psi_9 - \psi_{10}), \text{ (18e)}$$

$$\varepsilon_{HE,cooling,w} = \frac{E\dot{x}_{10} - E\dot{x}_9}{E\dot{x}_{15} - E\dot{x}_{16}} = \frac{\dot{m}_{recovery,w} (\psi_{10} - \psi_9)}{\dot{m}_{cooling,w} (\psi_{15} - \psi_{16})}, \text{ (18f)}$$

Heat exchanger for exhaust recovery (VI):

$$\dot{m}_{10} = \dot{m}_{11} = \dot{m}_{recovery,w}, \text{ (19a)}$$

$$\dot{m}_{12} = \dot{m}_{13} = \dot{m}_{exhaust,gas}, \text{ (19b)}$$

$$\dot{Q}_{exhaust,gas} = \dot{m}_{exhaust,gas} (h_{12} - h_{13}), \text{ (19c)}$$

$$\dot{Q}_{recovery,w,II} = \dot{m}_{recovery,w} C_{recovery,w,,p} (T_{11} - T_{10}), \text{ (19d)}$$

$$\dot{E}x_{dest,engine\ exhaust} = \dot{m}_{exhaust\ gas} (\psi_{12} - \psi_{13}) + \dot{m}_{recovery,w} (\psi_{10} - \psi_{11}), \text{ (19e)}$$

$$\varepsilon_{HE,exhaust\ gas} = \frac{E\dot{x}_{11} - E\dot{x}_{10}}{E\dot{x}_{12} - E\dot{x}_{13}} = \frac{\dot{m}_{recovery,w} (\psi_{11} - \psi_{10})}{\dot{m}_{exhaust\ gas} (\psi_{12} - \psi_{13})}, \text{ (19f)}$$

Circulating pump (VII):

$$\dot{m}_{14} = \dot{m}_{15s} = \dot{m}_{15,act} = \dot{m}_{cooling,w}, \text{ (20a)}$$

$$\dot{W}_{pump} = \dot{m}_{cooling,w} (h_{15,act} - h_{14}), \text{ (20b)}$$

$$\dot{E}x_{dest,pump} = \dot{m}_{cooling,w} (\psi_{14} - \psi_{15}) + \dot{W}_{pump,elec}, \text{ (20c)}$$

$$\dot{W}_{pump,elec} = \dot{W}_{pump} / (\eta_{pump,elec} \eta_{pump,mech}), \text{ (20d)}$$

$$\varepsilon_{pump} = \frac{\dot{E}x_{15} - \dot{E}x_{14}}{\dot{W}_{pump,elec}} = \frac{\dot{m}_{cooling,w} (\psi_{15} - \psi_{14})}{\dot{W}_{pump,elec}}, \text{ (20e)}$$

Relative irreversibility is given by:

$$RI = \frac{\dot{E}x_i}{\dot{E}x_{total}}, \text{ (21)}$$

DISCUSSION

The system considered could not be evaluated using numeric values in terms of energetic and exergetic aspects since they were not available in Ref. [6]. However, the following assessments may be done: Since compressor power depends strongly on the inlet and outlet pressures, any heat exchanger improvements that reduce the temperature difference will reduce compressor power by bringing the condensing and evaporating temperatures closer together. From a design standpoint, compressor irreversibility can be reduced independently. Recent advances in the market have led to the use of scroll compressors. Irreversibilities in heat exchangers (i.e., evaporator, condenser and heat recovery components) occur due to the temperature differences between the two heat exchanger fluids, pressure losses, flow imbalances and heat transfer with the environment. The irreversibility associated with the expansion valve (capillary tube) due to the pressure drop of the refrigerant passing through it.

The only way to eliminate the throttling loss is to replace the capillary tube with an isentropic turbine (an isentropic expander) and to recover some shaft work from the pressure drop.

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Assessment of The Dynamic Working Conditions of An Electric Power Heat Pump in The Heating State by Exergy Analysis

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SUMMARY

This paper assesses the efficiencies and properties of an electric power heat pump (EPHP) system running in dynamic and stable conditions in the heating state with exergy analysis and the whole heating system **C**oefficient **O**f **P**erformance (HS COP). It compares and analyzes the differences of the HS COP, the normal COP of the EPHP set, and exergy loss and exergy efficiency between the stable and the dynamic conditions. With the mathematic models and a case study, it shows that when the devices of the EPHP work in the range of durable strengths, the normal COP of the EPHP set in the dynamic condition is higher than the other one's. It also shows that the HS COP in the dynamic condition, especially the temperature at the evaporator changes with the lower source (or environment) temperature in the EPHP heating system, is higher than the stable one, and it save more energy too.

INTRODUCTION

Generally an EPHP system often runs in stable state. The inlet and outlet temperatures at the condenser and the evaporator are often made steady in engineering [1]. The heating load changes frequently because of the outdoor temperature and lower-level source temperature. So the EPHP works in the part-load conditions. Then whether the running EPHP system benefits in saving energy?

To diagnose an EPHP performance generally uses the first law of thermodynamics. In the other words, **E**ffective **E**nergy **R**ate (EER) and **C**oefficient **O**f **P**erformance (COP) are often taken as standards. However, the whole heating system (HS) can not only use them as judgments. Exergy analysis bases on the first and second laws of thermodynamics [2]. It can analyze energy use from “quality” and “quantity”. And it can reveal how energy has been use and lost. Therefore, exergy analysis points the economic way on energy and the EPHP performance.

DYNAMIC MATHEMATICAL MODEL OF "EPHP" HEATING SYSTEM

Mathematical model of building heating load

The method of calculating the building heating load of the experiment house is simple-stable-state one. The actual heating load $Q_{k,\,j}$ depends on the temperature difference between the indoor and the outdoor. When the indoor temperature keeps 18 ℃, the actual building heating load is related to the outdoor temperature. Therefore, the building heating load can be expressed as:

$$Q_{k,j} = KF(18 - t_w) \quad \text{kW}, \qquad (1)$$

Normal COP mathematical model of a EPHP set

The EPHP COP is defined as:

$$COP_{EPHP} = Q_k / N, \qquad (2)$$

Where Q_k is released heat at the condenser (unit, kW) and N is electrical power input to the compressor (unit, kW). $Q_{k,j} = \xi Q_k$ [3]. ξ is coefficient of heat leak. And it ranges from 0.8 to1. In the paper, ξ is 0.9.

Mathematical model of the HS COP

The HS COP is defined as:

$$COP_{HS} = Q_{k,j} / (N + N_{con} + N_{ev}), \qquad (3)$$

Where N_{con}, N_{ev} are the electrical power inputs to hot water pumps (HWP) at the condenser and the sea water pumps (SWP) at the evaporator, unit, kW.

Electrical power input to the pump model [4] is given by

$$N = \rho GH / 102\eta_{se}, \qquad (4)$$

Where ρ is density of water (kg/ m^3), G is mass flow rate (kg/s), H is lift (m) and η_{se} is pump efficiency. In the paper, density of water is 1×10^3 kg/ m^3; density of sea water is 2.2×10^3 kg/ m^3; HWPs and SWPs lifts are 32m and15m, respectively; HWPs efficiency is 0.78. SWPs efficiency is 0.7.

MATHEMATICAL MODEL OF EXERGY ANALYSIS

Exergy analysis bases on the law of conservation of mass, the law of the energy equilibrium, and the second law of thermodynamics.

Exergy analysis of EPHP set

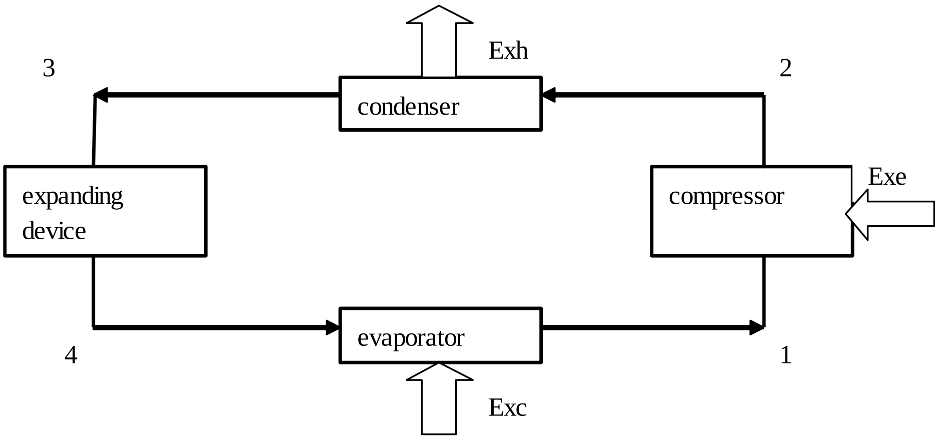

Fig.1. Schematic of an EPHP

Fig.1 Shows an EPHP. The number of the available energy (exergy) entered is equal to the total of the exergy exited and the exergy lost in the system [5].

$$E_{xe} + E_{xc} = E_{xh} + E_{xl} \, , \qquad (5)$$

Where E_{xe}, E_{xc}, and E_{xh} denote exergy at the compressor, the evaporator, and the condenser, respectively; E_{xl} is exergy loss at the EPHP.

When the EPHP works in the heating state, the exergy efficiency (η_{ex1}) is given by

$$\eta_{ex1} = E_{xh} / E_{xe} \, , \qquad (6)$$

Exergy loss of a water pump

When the water flow passes pumps, there are not physical change and chemical change, only raising height. Because pumps consume electrical power on work, exergy loss is same as the electrical power in this process.

For HWPs, exergy loss is $\Delta E_{pc} = N_{con}$.

For SWPs, exergy loss is $\Delta E_{pe} = N_{ev}$.

Exergy analysis in EPHP heating system

The theory is same as exergy analysis in the paper above, the number of the exergy entered is equal to the total of the exergy exited and the exergy lost in an EPHP heating system [6].

$$E_t = E_u + \Delta E \, , \qquad (7)$$

E_t is the total of exergy entered.

$$E_t = N + N_{ev} + N_{con} + Q_{e,j}(1 - \frac{T_0}{T_L}), \qquad (8)$$

Where $Q_{e,j} = \xi Q_e$, ξ is 0.9 in this paper.

E_u is the total exergy used (or exited) in the EPHP heating system.

$$E_u = Q_{k,j}(1 - \frac{T_0}{T_H}), \qquad (9)$$

T_L, T_H are the water average temperatures at the evaporator and the condenser, respectively.

$$T_L = \frac{T'_{w,ev} - T''_{w,ev}}{\ln \frac{T'_{w,ev}}{T''_{w,ev}}}, \qquad (10) \qquad\qquad T_H = \frac{T''_{w,con} - T'_{w,con}}{\ln \frac{T''_{w,con}}{T'_{w,con}}}, \qquad (11)$$

Where $T'_{w,ev}$, $T''_{w,ev}$ are the cold water inlet and outlet temperatures at the evaporator, respectively, $T'_{w,con}$, $T''_{w,con}$ are the hot water inlet and outlet temperatures at the condenser, respectively.

ΔE is the total exergy lost. The rate of exergy loss in the EPHP heating system is given by

$$\eta_{ex2} = \Delta E / E_t, \qquad (12)$$

APPLICATION OF AN EXAMPLE OF MODEL

Assuming the indoor temperature keeps 18℃, the actual heating load is a variable with the outdoor temperature. Using the SOLKANE software calculates the inlet and outlet parameters of the equipment in the EPHP system.

Main conditions in the EPHP running system

The case study chooses a helical-compressor sea water heat pump system (SWHP) (all kinds of the EPHP have the same running theory), uses R22 as refrigerant. The SWHP utilizes the five meters deep sea water as lower-level heat source. The standard working condition of the heat pump is that the condenser releases 879 kW heat; electrical power input to the compressor is 209 kW and energy regulation is stepless from 25% to 100%; the hot water inlet/outlet temperatures at the condenser are 40/45 ℃ and the hot water flow rate is 151 m^3/h; the cold water inlet/outlet temperatures at the evaporator are 9/4 ℃ and the cold water flow rate is 115 m^3/h. In the case, sea water runs into the evaporator directly. The SWP uses the variable frequency pump which maximum flow rate is 200 m^3/h.

Other conditions

The Fig.2 is the meteorological data in Dalian city, China, from 8:00A.M, Jan.1st to 14:00P.M, Jan.10th, 2000 (the time is Beijing time).

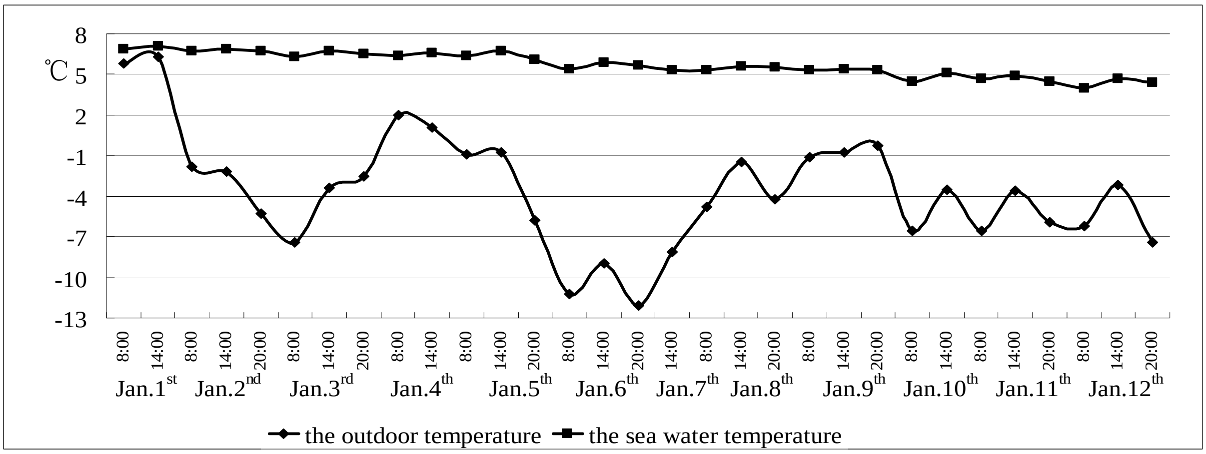

Fig.2 Outdoor air temperature and sea surface water temperatures in Dalian from 8:00A.M Jan.1st to 14:00P.M, Jan.10th, 2000. (The data come from http://www.coi.gov.cn/)

As Fig.2 shows, from 8:00A.M, Jan.1st to14:00P.M Jan.10th, 2000, the outdoor air temperature fluctuated from -12.1 ℃ to 6.3 ℃. The lowest temperature appeared in Jan. 6th. And the sea water temperature fluctuated stably from 4.5 ℃ to 7.1 ℃. Because the sea water temperatures have no difference between the 5 m deep and the sea surface, therefore the sea surface water temperature is used as the sea water temperature in 5m deeps.

Three schemes in the case study

Scheme Ⅰ: the EPHP runs in stable state. Control parameters: the hot water outlet temperature at the condenser is 45 ℃, its flow rate is 150 m^3/h; the cold water outlet temperature at the evaporator is 1.5 ℃.

Scheme Ⅱ: the EPHP runs in dynamic state. Control parameters: the hot water outlet temperature at the condenser is 45 ℃, its flow rate is 150 m^3/h; the cold water flow rate at the evaporator is 200 m^3/h.

Scheme Ⅲ: the EPHP runs in dynamic state. Control parameters: the hot water outlet temperature at the condenser is 45 ℃, its flow rate is 150 m^3/h; the difference temperature between the outlet and the inlet cold water at the evaporator is given by 4 ℃.

RESULTS AND DISCUSSION

Normal Comparison of three schemes

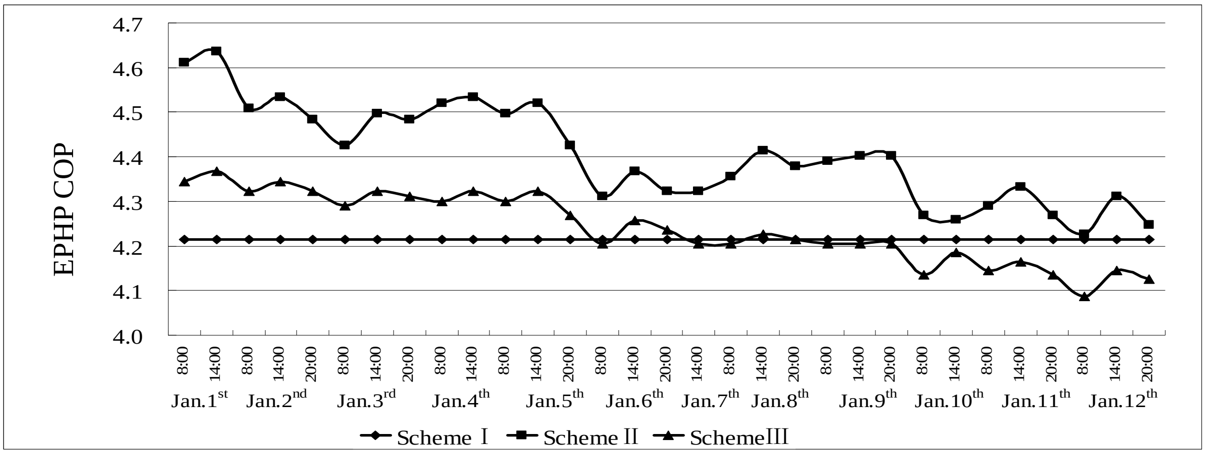

Fig.3. Comparison of normal COPs of three schemes

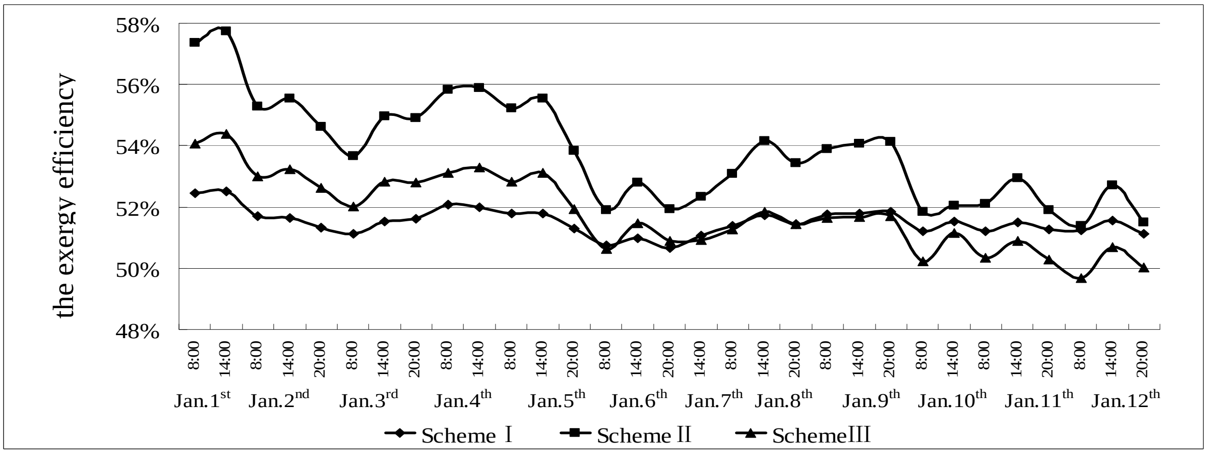

Fig.4. comparison of normal exergy efficiency of three schemes

From fig.3, when the EPHP worked in the way of schemeⅡ, COP, working efficiency, is greater than the others on majority conditions, and the exergy efficiency of schemeⅡ is the best, so obviously the schemeⅡ is the best. The fig.4 shows, for long-term working efficiency and saving energy, the normal COP of an EPHP set could be increased in dynamic state. Comparison of schemeⅡ and schemeⅢ indicates that the normal COP of an EPHP set can be improved better by reducing the heat exchange temperature difference and increasing the cold water flow rate, and the EPHP could recover more available energy from the lower source and decrease the consumption of the high-level energy source and get better working efficiency.

From Jan.10th to Jan.12th in fig. 4, the exergy efficiency of schemeⅢ is the lowest. When the sea water temperature was lower than 5 ℃ after Jan.10th, the refrigerant temperature within evaporator is the lowest. Therefore, the rate of exergy loss and the EPHP normal COP for schemeⅢ are the lowest. At that time, working and the energy efficiencies are both worse.

System comparison of three schemes

Only the EPHP normal COP is better does not mean that the whole EPHP heating system saves more energy. For saving more energy in the EPHP heating system, other equipment connected to the EPHP set such as pumps and expanse regulator should be considered as whole.

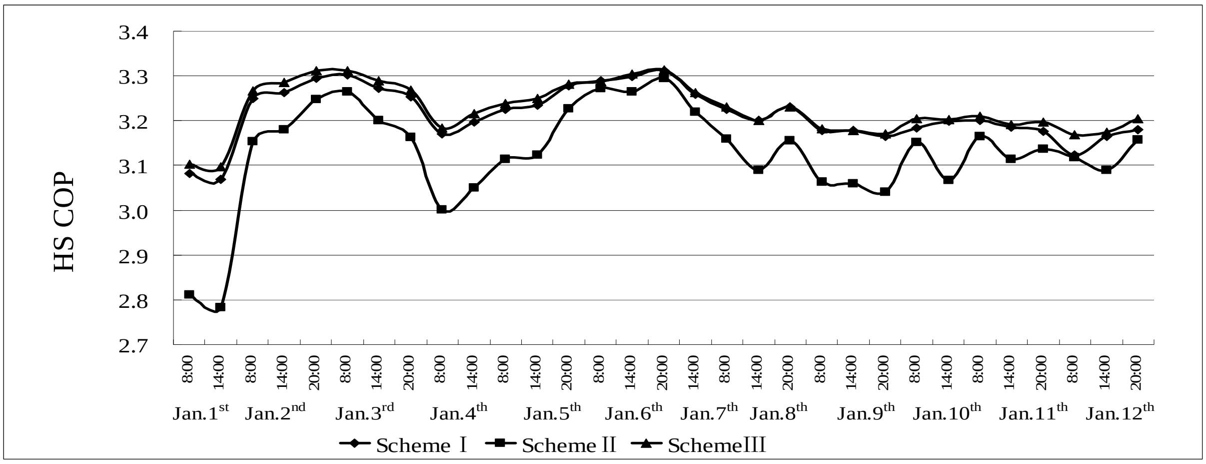

Fig.5. HS COPs Comparison of three schemes

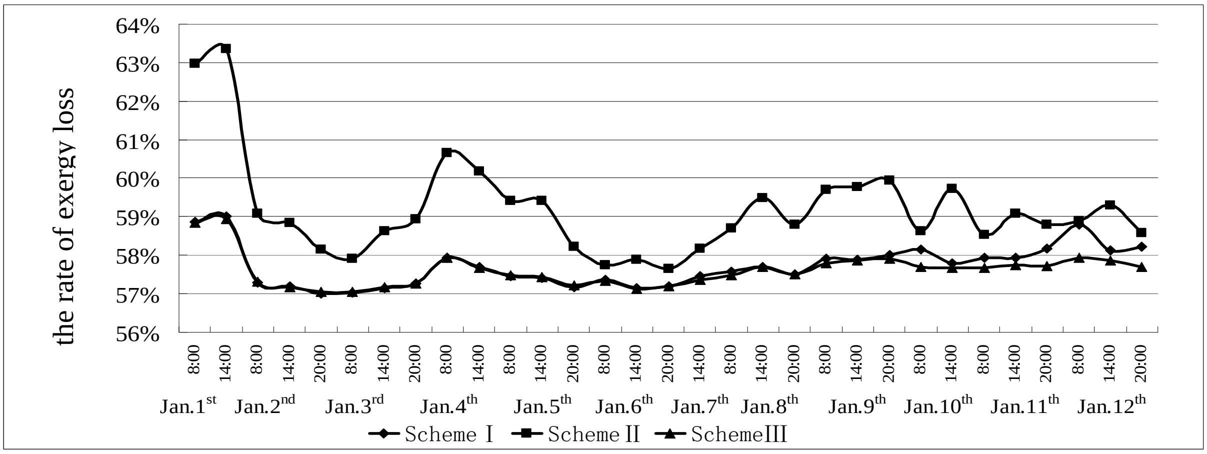

Fig.6. Comparison of the rate of exergy loss of three schemes

From the upper analysis, scheme Ⅱ is the best. However, fig.5 and fig.6 show that the rate of exergy loss of scheme Ⅱ is the highest and its HS COP is the lowest. Although its exergy efficiency of the EPHP set is the greatest, because the cold water flow rate is too much, the exergy loss of the HWP and the SWP are higher. Hence, scheme Ⅱ does not save energy according to the whole EPHP heating system analysis. Therefore fixed cold water flow rate, changed refrigerant temperature with the actual heating load, this kind of scheme does not save energy. It is suggested that engineers should avoid taking the scheme in practical project. In one word, at the same meteorological data, from property comparison of the EPHP set and the EPHP heating system, an EPHP set has the greatest normal COP may not be the best one to check the energy saving and better efficiency when we consider the EPHP set and linked equipment as a whole system. It is suggested that in practical project engineer should not only consider the better normal COP as a standard, sometimes we have to choose lower normal COP in order to save energy of the EPHP heating system and attain an better HS COP.

According to the analysis of the EPHP set, the normal COP and the exergy efficiency of schemeⅢ are the lowest, and it is the worst scheme. However, the HS COP of schemeⅢ is higher than that of schemeⅡ, and the rate of exergy loss in schemeⅢ is lower than in schemeⅡ after Jan.10th. Therefore, according to the analysis of the EPHP heating system, schemeⅢ is the best one. The whole system analysis should be considered to save energy.

From the fig.6, the rate of exergy loss between schemeⅠ and schemeⅢ is very small. It is hard to choose the better one. Hence, it needs to refer to HS COP and the exergy analysis to solve the dilemma. From the fig.5, the HS COP of schemeⅢ is higher than that of schemeⅠ from Jan.1st to Jan.5th and after Jan.10th, so the system could run in dynamic condition. From Jan.6th to Jan.9th, we suggest use the EPHP heating system in stable state because it has advantages in working and management.

CONCLUSION

The paper calculates the energy loss and different normal COP and HS COP results of several EPHP systems running in dynamic and stable conditions.
First, when the devices of the EPHP work in the range of durable strengths, the EPHP COP in the dynamic condition is higher than the other one's. Second, with a case study, it also shows that the HS COP in the dynamic condition, especially the temperature at the evaporator changes with the lower source (or environment) temperature in the HS, is higher than the stable one, and it save more energy too. Finally, using HS COP and the exergy analysis can point out the ways that save energy and improve working efficiency.

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Reducing Energy Consumption of a Dehumidifying System for a Dry Room - Basic Investigation And Experiment of a Pilot Plant Using a CO_2 Heat Pump Cycle -

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SUMMARY

The purpose of this study was to reduce the energy consumption of an industrial air-conditioning system. The energy conservation of an air-conditioning system that supplies low dew point air was investigated and validated experimentally. The consumption of energy was reduced by introducing a heat pump for both heating and cooling of the air at the same time. The authors studied working fluid that is suitable for the heat pump and found that R744 (CO_2) has higher heating performance than R410A and R22. A simulation of the system with a CO_2 heat pump was found to reduce energy consumption by 8.2% at peak conditions and 13.2% at intermediate conditions. The authors also developed a pilot plant that combines dehumidifier and heat pump cycle, and the experimental results indicated a 6.5% energy reduction at peak conditions and a 10.2% reduction at intermediate conditions.

INTRODUCTION

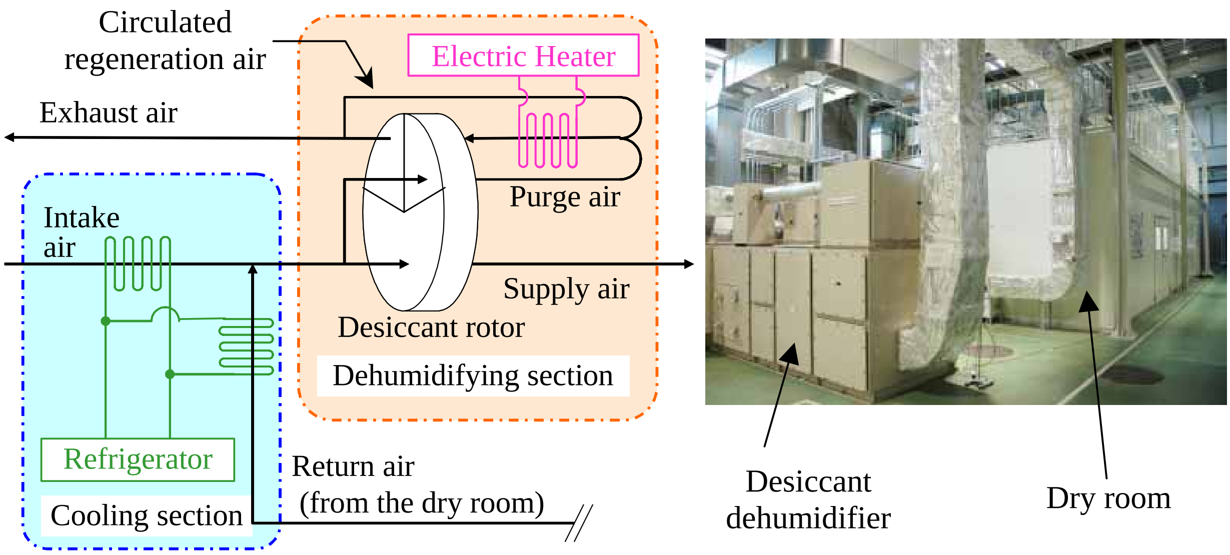
Dry rooms are used when producing electrical and chemical appliances, such as lithium-ion batteries and organic electro luminescence devices. This production process requires extremely low humidity with a dew point of approximately -60 °C. A desiccant dehumidifier is usually used for dry rooms. To provide such dry air, the desiccant must be heated to a higher temperature than a general system. This system requires that the air for the desiccant rotor be heated to over 140°C. Therefore, a lot of electricity is needed for heating the air. This system also has an air-cooling process for intake air and return air from the dry room.

Several studies have been made on the energy conservation of the desiccant air-conditioning system, and some of them have focused on applying a heat pump to both heat and cool the air. Tsay et al. studied the applicability of combining the desiccant cooling system with a heat pump and estimated the system performance [1]. Kato et al. studied the performance of a CO_2 heat pump (CO_2 -HP) with a desiccant air-conditioning system for general air conditioning system [2]. They calculated the behavior of the system and showed a substantial increase in energy efficiency.

In this paper, the authors describe a computer simulation that was carried out to increase the energy saving performance of a desiccant dehumidifier combined with a heat pump. The construction of this system is different from Kato’s one because of the difference of the target humidity. Furthermore, a pilot plant consisting of a CO_2 -HP unit installed with a desiccant dehumidifier was developed. The performance of this system was validated experimentally.

DESICCANT DEHUMIDIFIER FOR A DRY ROOM

Figure 1 shows the basic construction of a conventional air-conditioning system for a dry room and a photograph of a desiccant dehumidifier with a dry room. This system consists of cooling dehumidifying sections. In the cooling section, a refrigerator chills intake air and return air from the dry room. A portion of the moisture in the intake air is removed via the drain. In the dehumidifying section, a desiccant rotor removes the water from the air that was sent from the cooling section and supplies the dry room with the air.



a) b)

Figure 1. Conventional air-conditioning system. a) Basic construction, b) Photograph of system.

This desiccant rotor has three zones, i.e., a dehumidification zone, regeneration zone, and purge zone. In the dehumidification zone, the remaining moisture in the air from the cooling section is removed, and the dew point of the air becomes approximately -60°C. This dry air is then supplied to the dry room. In the regeneration zone, the moisture of the rotor is removed by regeneration air, and a part of this air is circulated to join the purge air at the inlet of an electric heater. The rest of the air is discharged as exhaust air. To make the supply air sufficiently dry, the temperature of the air at the inlet of the regeneration zone must be over 140°C. In the purge zone, the regenerated rotor is cooled by purge air, which is a portion of the air from the cooling section. After cooling the rotor, the purge air is mixed with the circulated regeneration air and heated electrically.

A breakdown of energy consumption of the conventional system appears in figure 2. The electric heater and refrigerator consume a large part of the whole energy, especially during peak conditions. We can see from figure 2b) that the energy reduction of the electric heater is important for this system's energy conservation. To reduce the energy consumed at the heater, the authors focused on applying a heat pump that produces sufficiently high temperature. The characteristics and possibilities of a heat pump cycle and a combined system of a desiccant dehumidifier were then investigated.

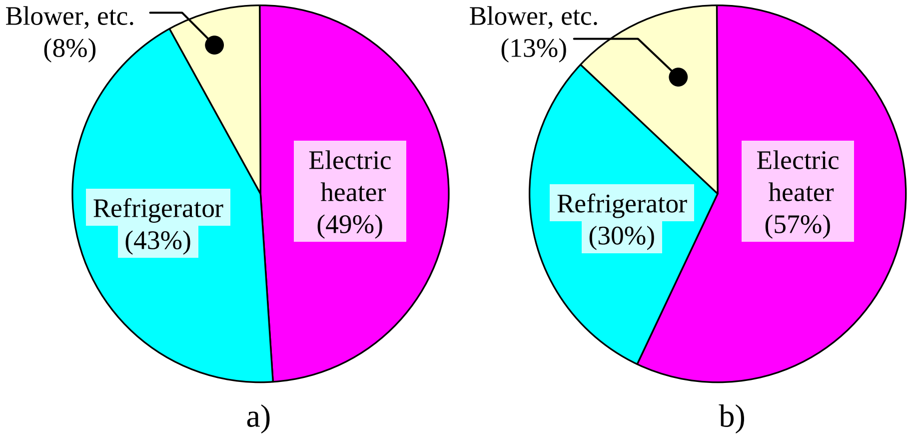


Figure 2. Breakdown of energy consumption of conventional system.
a) Peak (summer) conditions, b) Annual average.

WORKING FLUID OF THE HEAT PUMP

Selecting the working fluid is an important factor for the heat pumps. However, few studies have addressed this matter from the viewpoint of applying heat pumps to a desiccant dehumidifier. Hence, the authors focused on the development of a pilot plant and experimental work. Three kinds of working fluid, which are available on the market, were selected and compared.

Figure 3 shows the theoretical heat pump cycle of each working fluid in a T-h diagram. R410A (R32/R125=50/50 wt%) is presently used in many types of air conditioners. R744 (CO_2: carbon dioxide) was applied for heat-pump water heaters by Endoh et al. [3]. These two fluids were assumed to be available for the target system. Additionally, R22 ($CHClF_2$) was evaluated as a reference. The calculating assumptions shown in table 1 are based on the operating points of the peak conditions.

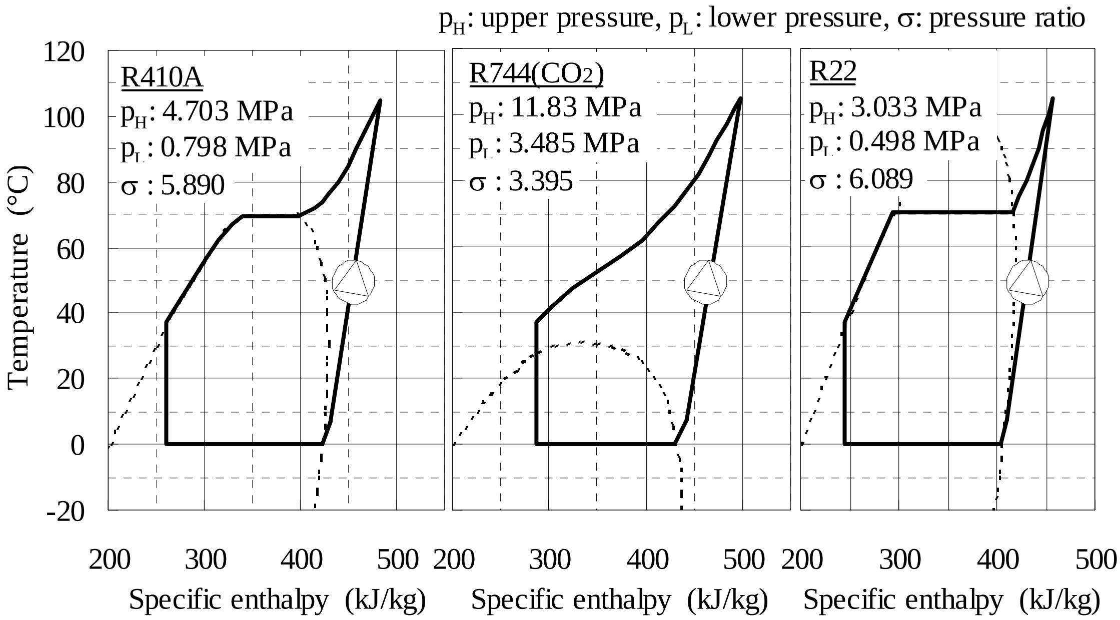


Figure 3. T-h diagram of each working fluid.

The temperature of the inlet air of the electric heater is considerably higher than the ambient air. Consequently, working fluid should have a higher heating capacity in the high temperature domain. That is, in figure 3, the decrease in specific enthalpy after the air was discharged from the compressor outlet (highest point of the T-h diagram) should be big. From this point, R410A and CO_2 have better characteristics than R22 because their shapes in the T-h diagram are not as steep as R22 after the outlet of the compressors.

Table 1. Calculating assumptions of figure 3

Compressor outlet	105°C
Evaporator	0°C
Superheat at the evaporator	7°C
Condenser outlet	37°C

This comparison is represented in figure 4. R22 has the highest value in the theoretical COP (coefficient of performance) of cooling, and the cooling performance of CO_2 is lower than that of the other two substances. The other three characteristics were calculated from the heating capacity when the temperature of the working fluid decreased from 105°C to 75°C. The theoretical COP of heating of CO_2 is obviously better than that of R22 and R410A. These results agree with the aforementioned ones for figure 3.

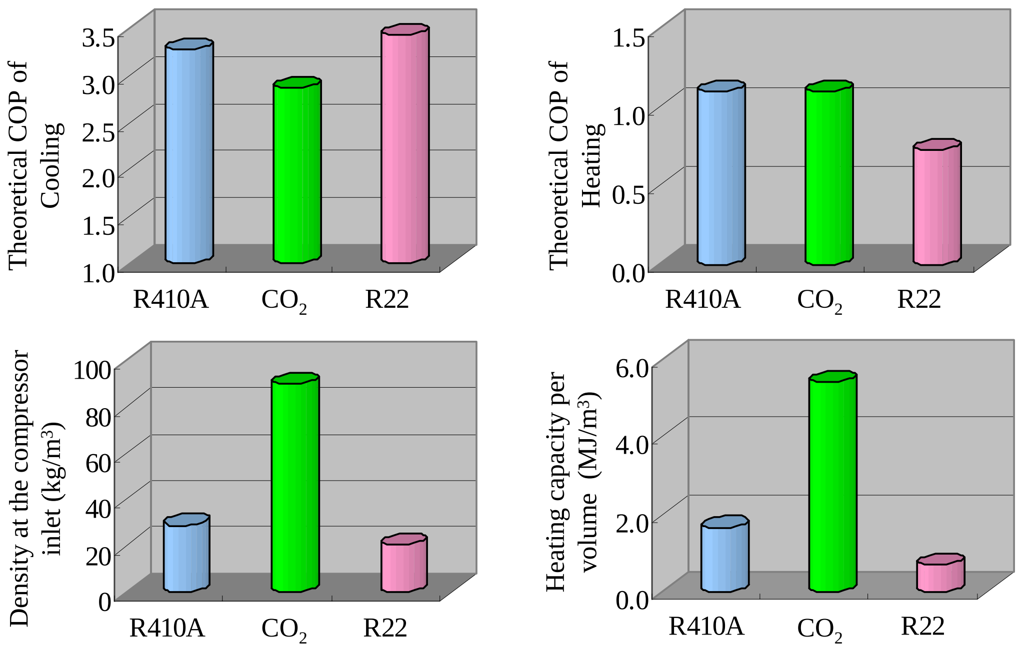


Figure 4. Comparison of working fluids by performance and heating capacity*
(When temperature of working fluid decreases from 105°C to75 °C)

The lower two bar charts present the density of the working fluids and the heating capacity per volume at the inlet of the compressor. The density of CO_2 is considerably higher than that of R410A and R22. This makes the heating capacity of CO_2 more than three times that of R410A and seven times that of R22. This means that CO_2 can miniaturize the components, in particular, the compressor.

The authors selected CO_2 as the working fluid for the heat pump of this system on the basis of these results, namely from the viewpoint of energy saving and practicality. The low global warming potential (GWP) of CO_2 was also taken into consideration.

SYSTEM DESIGN AND COMPUTER SIMULATION

Figure 5 shows the new air conditioning system for the dry room. This system mainly consists of a desiccant rotor, a CO_2-HP, an electric heater, and a refrigerator. The CO_2 -HP is used for both heating regenerating air and cooling return air.

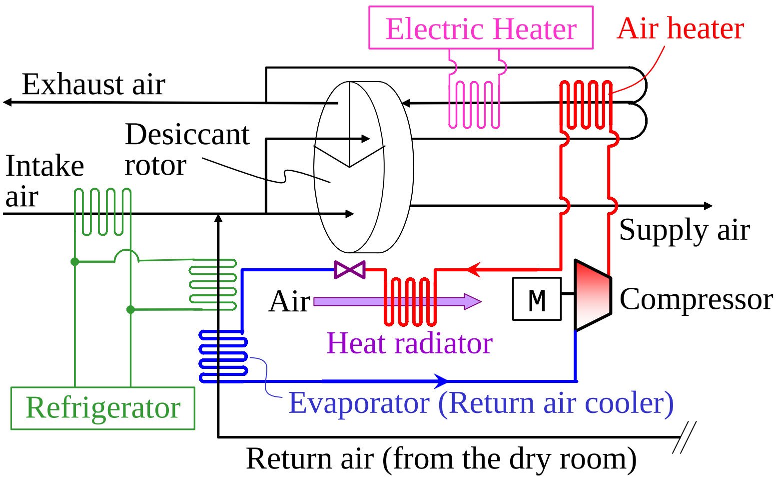


Figure 5. Air conditioning system with desiccant dehumidifier and CO_2-HP.

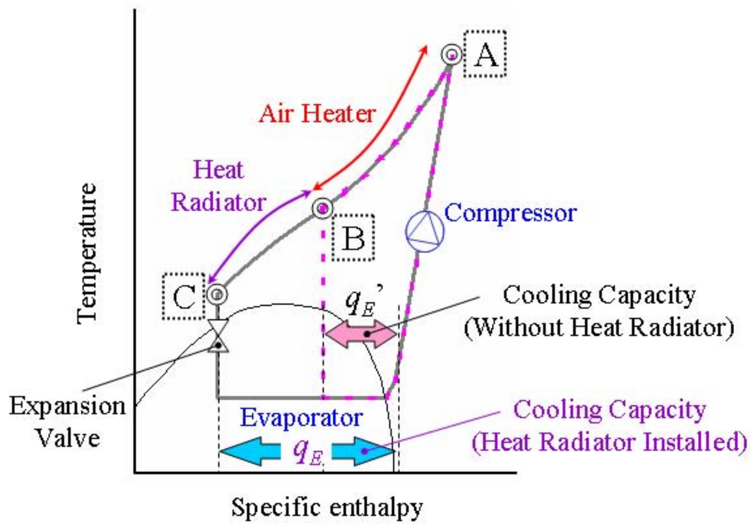


Figure 6. T-s diagram of CO_2 heat pump cycle and effects of adding heat radiator.

A heat radiator is shown in figure 5. It turns on during the cycle to boost the cooling capacity at the evaporator. The effects of the heat radiator can be explained as follows using figure 6. State "A" is the outlet of the compressor. The temperature of the inlet air of the air heater is considerably higher than that of the ambient air. Therefore, if you do not use the heat radiator, the working fluid of state "B" goes directly to the expansion valve. In this case, the cooling capacity would be only "q_E'", which is shown in figure 6. If you put the heat radiator into the heat pump cycle in the position it is in figure 5, the working fluid is cooled by ambient air. Then the inlet state of the expansion valve is state "C". In this case, the cooling capacity is "q_E", which is shown in figure 6.

A computer simulation of the system shown in figure 5 was performed and compared with the conventional system. This simulation was based on the performance data of an actual compressor and heat exchangers of a heat-pump water heater using CO_2 as the working fluid [3]. The ambient temperature was assumed to be 32.0°C at peak conditions and 19.0°C at intermediate conditions, and the return air temperature was assumed to be 25.0°C. The results are in figure 7. The new system, compared with the conventional system, reduced energy consumption by 8.2% at peak conditions and 13.0% at intermediate conditions.

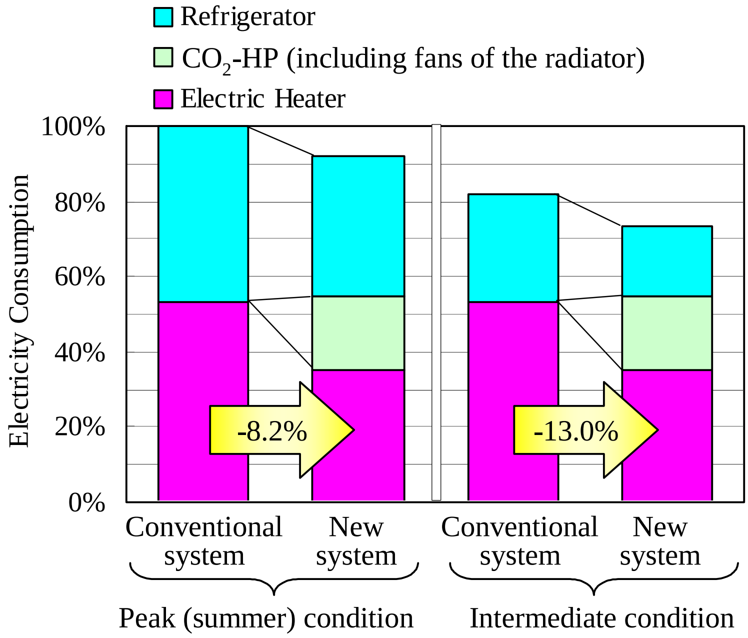


Figure 7. Simulation results of adopting CO_2-HP.

EXPERIMENTAL EVALUATION

A pilot plant, shown in figure 8, was developed by installing the CO_2-HP into a conventional system. This plant can supply 3,000 m^3/h of dry air with a -60°C dew point. The desiccant rotor is 965 mm in diameter and is 400 mm thick. The radiator unit is separated from the desiccant dehumidifier because it will be an outdoor unit when installed for actual use. Figure 9 shows a frontal view of the radiator unit as well as a sectional view of a compressor used in this system.

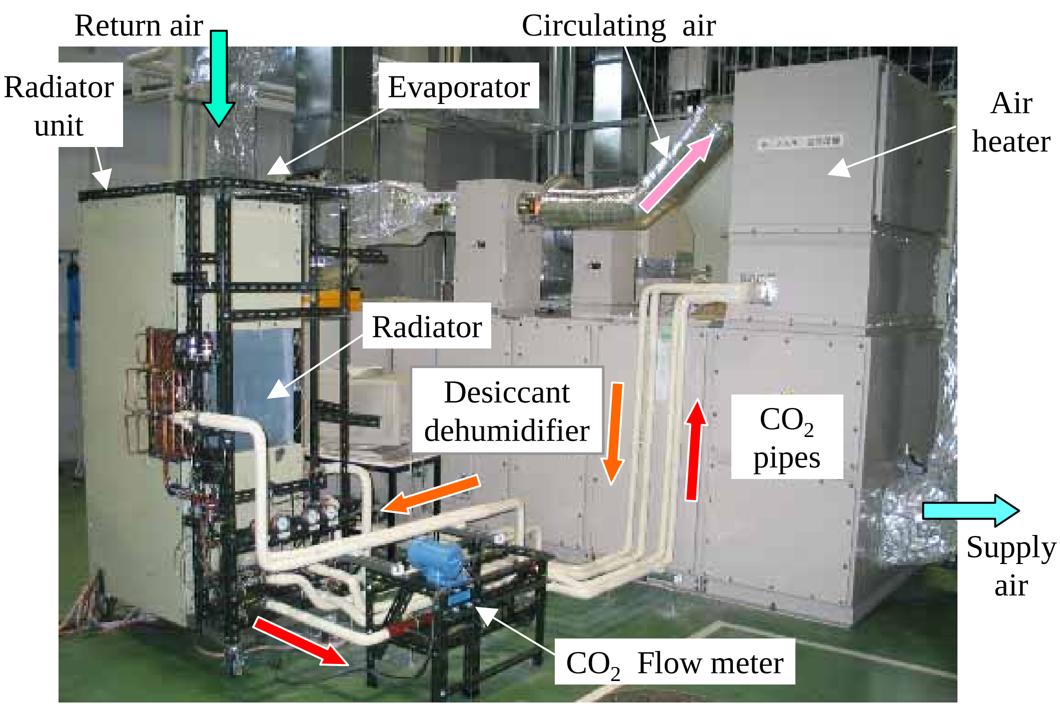


Figure 8. Pilot plant of the system with construction of figure 5.

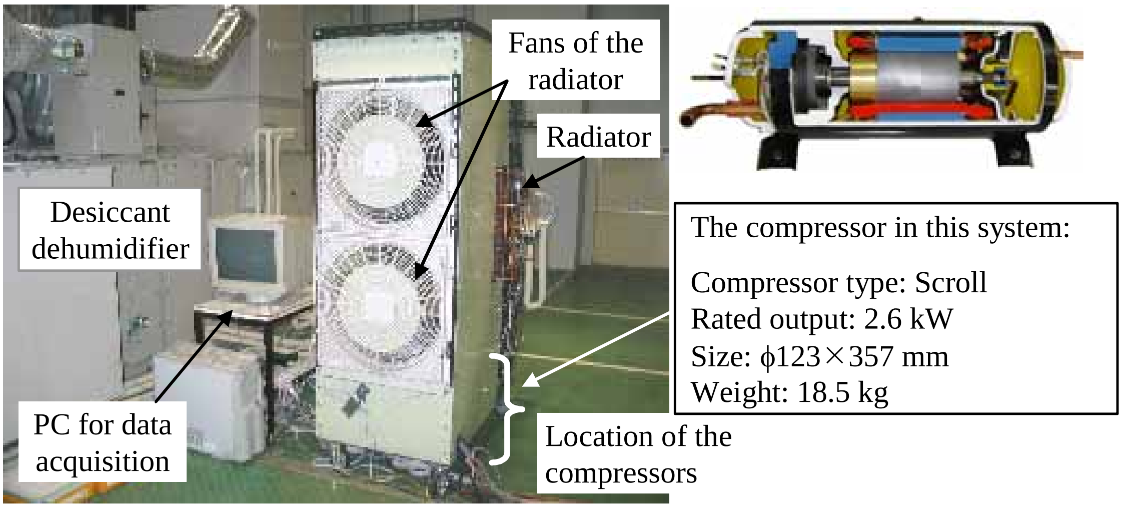


Figure 9. Front view of radiator unit and sectional view of compressor.

The heat pump has two scroll compressors located at the bottom of the radiator unit. They were developed by Endoh et al. [3] and manufactured for CO_2 heat pump water heaters. The specifications of this compressor are also in figure 9. The radiator and evaporator are in the form of a fin-tube heat exchanger.

The experiments were carried out with the same conditions as the aforementioned computer simulation. The results are shown in figure 10. Electricity consumption was reduced by 6.5% at peak conditions and 10.2% at intermediate conditions. Table 2 summarizes the operating conditions of the experiments. The dehumidification obtained at each condition as well as the energy savings were excellent.

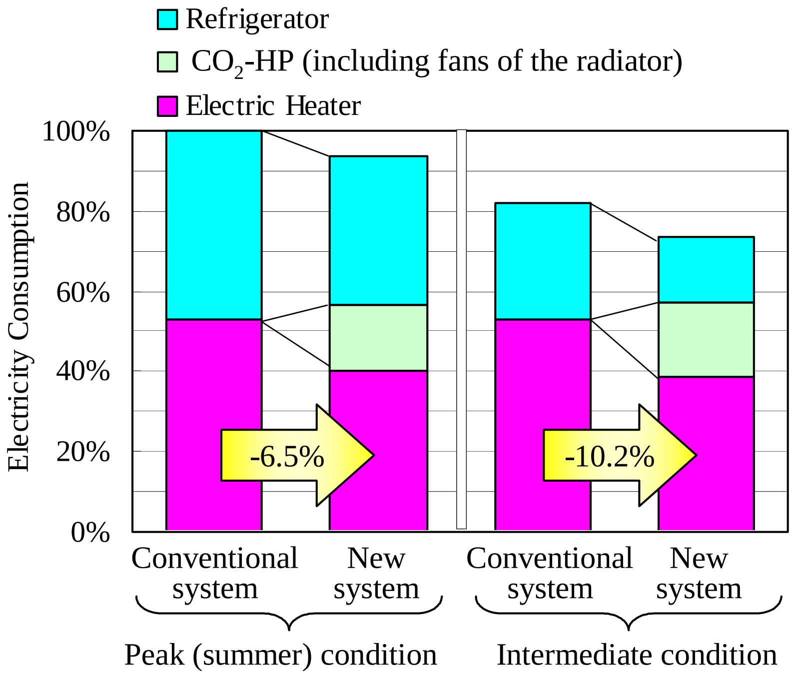


Figure 10. Experimental evaluation of new system.

Table 2. Operating conditions of experiments.

Operating condition		Peak	Intermediate
Ambient air temperature		32.0°C	19.0°C
Return air	Temperature	25.2°C	19.1°C
	Volume flow rate	100%	75.5%
Supply air	Dew point temperature	-60.2°C	-60.4°C
	Volume flow rate	2,946 m^3/h	3,254 m^3/h

DISCUSSION

Combining a heat pump cycle with a desiccant dehumidifier was found to be a practical application of the air-conditioning system for a dry room. The experimental results of the pilot plant demonstrated this. The authors compared CO_2 with R410A and R22 for the working fluid of the heat pump and selected CO_2 because of its superior heating performance and because of environmental issues. The practicality of this system is supported by the fact that the components of a CO_2-HP are already available as the components of a heat pump water heater that uses CO_2 as the working fluid [3].

However, this system can be further optimized. In particular, the matching of the desiccant dehumidifier and CO_2-HP needs further improvement. The heating capacity and evaporating temperature of the CO_2-HP, heat exchanger arrangement, and other aspects are other areas that can be improved. Various applications of the heat pump cycle are being developed, and the performance of compressors and heat exchangers are improving as a result. This study will lead us to applying these technologies of this system and optimizing it.

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Design and simulation of residential CO_2 transcritical heat pump water heater in central China

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SUMMARY

Based on the weather and groundwater conditions in central China, a computer code has been developed to simulate a residential water-to-water heat pump water heater (HPWH) which practices CO_2 transcrtical cycle; such a modeling is carried out in Engineering Equation Solver (EES). Under the design conditions, the inlet heat source(groundwater) temperature of 17℃,the evaporation temperature of 7℃,the inlet hot water temperature of 17℃,the outlet hot water temperature of 65℃, the compressor power at 3000W, a heating COP of 3.2 is achieved and the optimum high-side pressure is controlled below 8.5MPa which benefits manufacturing materials and operation a lot. The calculation results show that system with internal heat exchanger(IHE) has a 18% increase in COP. Effects of operating parameters on system performance,such as, inlet temperature of heat source, inlet temperature of hot water, compressor speed are also discussed. Compared with the conventional electrical or natural gas fired systems which are now very commonly used in urban districts in China, such HPWH with CO_2 as refrigerant is much more efficient ,energy-saving and correspondingly promising. The simulation results are found to be in reasonable agreements with simulation or experimental results published in the open literature, and offer useful guidelines for prototype design, experiments, and system optimization.
Keywords: Design; Simulation; HPWH; CO_2 (Carbon dioxide);Transcritical;EES

INTRODUCTION

The history of CO_2 commonly used as a refrigerant dates back to the late 1800s. Compared with other natural refrigerants in common use like ammonia, propane, CO_2 is completely harmless. Owing to its non-flammability, non-toxicity even if leaked to the atmosphere, CO_2 was once broadly applied to ships, the kind of unstationary fields. With the technology of the day,the lower critical temperature(31.1℃) and the higher critical pressure(7.37Mpa), CO_2 system had a lower refrigeration efficiency. Additionally, seawater,as the cooling water to evaporator, especially in the tropics approached its temperature to 30℃,therefore further reduced the efficiency of the subcritical refrigeration system, which greatly limited application fields of CO_2. The successfully developed Freons in 1930s suddenly terminated the use of CO_2 as a safe refrigerant. Interestingly, after the CFCs, HCFCs and HFCs were proved to be harmful on atmosphere in different degrees and banned by international agreements, CO_2 revived in refrigeration and heat pump applications. The pioneer researchers, late Professor Gustav Lorentzen looked early upon natural working fluids as the best replacements for the CFCs/HCFCs[1]. In 1992, He [2] and J. Pettersen firstly

Nomenclature

Bo	Boiling number($=\frac{q}{G(i_g - i_l)}$)	σ	surface tension, (N/m^2)
Bd	Bond number($=\frac{g(\rho_l - \rho_g)}{\sigma_l}$)	Subcripts	
D	diameter(m)	1-6	state point of refrigerant
g	gravitational acceleration,m/s^2	b	evaluated at bulk temperature
G	mass flux,kg/(m^2.s)	ci	inlet of compressor
h	heat transfer coefficient,W/(m^2.K)	co	outlet of compressor
i	enthalpy,J/kg	cr	critical point
k	thermal conductivity,W/(m.K)	g	gas state
L	length,m	hw	hot water r
Nu	Nusselt number,(-)	hwi	inlet of hot water
P	Pressure，Pa	hwo	outlet of hot water
Pr	Prandtl number,(-)	i	at inner wall
q	heat flux,W/m^2	in	inlet
Re	Reynolds number,(-)	is	isentropic
v	speed,m/s	l	liquid state
		o	outlet
Greek symbols		pool	nucleate boiling
ρ	density, (kg/m^3)	pc	pseudocritical conditions
χ	quality,(-)	r	refrigerant
θ_{dry}	dry angle	tp	two phase
η	efficiency	wet	wetted portion

proposed the concept ,transcritical cycle where a single-phase gas cooler replaced the traditional two-phase condenser . They considered CO_2 strongly suitable to HPHW because a higher temperature glide at heat rejection in gas cooler is readily available. Petter Neksa et al. [3] have carried out experimental studies on a CO_2 HPWH.Over the past decade,several research groups in Norway,America,Japan,Korea,India,China etc. have made extensive studies on HPWH. In particular,Japan now sells several hundreds thousand CO_2 HPWHs per year. In china, CO_2 application is still in experimental stage.CO_2 also has other unique advantages: compactness due to high operating pressures and accordingly low specific volume, excellent transport properties. The good characteristics make it much fitter for automobile air conditioning which values highly the volume of refrigeration unit[2][4-5].

In order to push the CO_2 application, theoretical system simulation work based on the modern developed computer science plays an important role, providing the true experiment with more guidance, accuracy and economization. But theoretical system simulation on CO_2 is scare in literature.[6] simulated CO_2 mobile air conditioning system.[7] performed simulation studies of CO_2 transcritcal cycle system with air as secondary fluid.[8]reported an simulation of air-to-air air conditioners and heat pumps. A theoretical and experimental study was carried out for a residential brine-to-water CO_2 heat pump system for combined space heating and hot water heating by [9].[10] developed a steady state water-to-water simulation model to evaluate the system performance of a transcritical CO_2 heat pump for simultaneous heating and cooling .Such a system is suitable in diary plants where simultaneous cooling at 4.8℃ and heating at 73.8℃ are required. In the present research, counter flow, pipe-in-pipe heat exchangers are chosen as the configuration of the heat exchangers. After analyzing and comparing in details, the appropriate heat transfer and pressure drop correlations have been used as presented below. According to the present groundwater conditions in central China, it’s such a energy-saving method to use groundwater as the heat source to evaporator, especially in North China where outdoor air at lower temperature is not suitable for low temperature heat source.Under the design conditions,meanwhile considering the national water heater standards, a water-to-water CO_2 transcritical system was designed and simulated to assess the COP, heating capacity, optimum discharge pressure. Effects of operating parameters such as, inlet temperature of heat source, inlet temperature of hot water, compressor speed are also discussed. The simulation models are developed on the following assumptions: compression process is adiabatic but not isentropic; expansion process is isenthalpic; pressure drop on water and connecting pipes, heat transfer with the ambient are negligible; only single-phase heat transfer occurs for both heat source and hot water. The simulation results are expected to offer useful guidelines for prototype design, experiments, and system optimization.

METHODS

Design

As shown in Fig.1, a one-stage CO_2 heat pump water heating system contains four main components and IHE. The corresponding T-s diagram is illustrated in Fig 2.Groundwater at 17℃ is fed as the heat source to the evaporator while city water at 17℃ is supplied as the fluid to gas cooler. Based on the national water heater standards, the hot water temperature at the outlet to gas cooler is controlled at 65℃.Both the heat exchangers are characterized of pipe-in-pipe counter flow type with working fluid in the inner tube and water in the outer annular channel. In the pipe-in-pipe counter flow IHE, hot CO_2 flows through the inner tube and cold CO_2 passes through the outer annular.

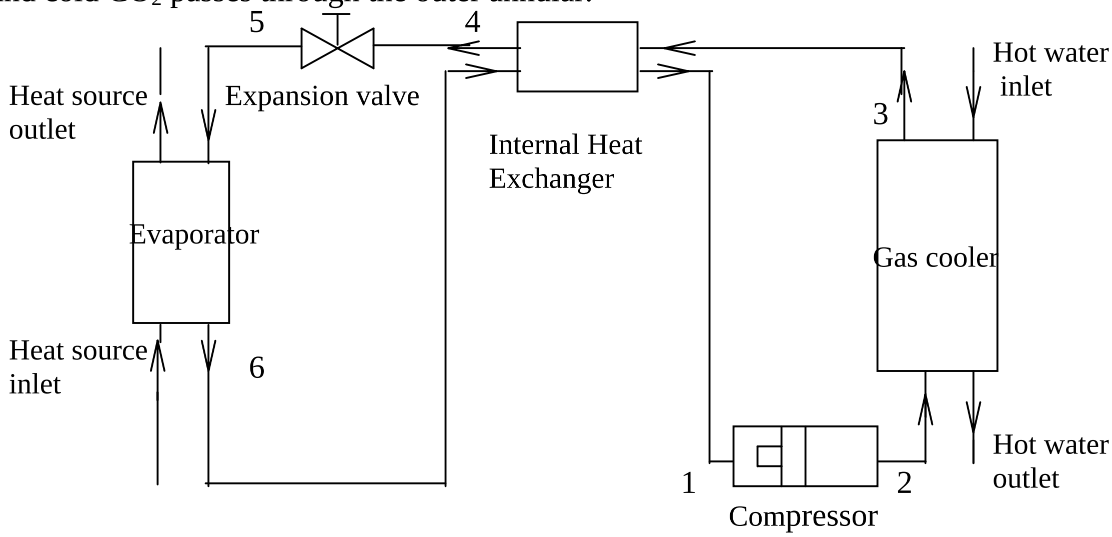

Fig 1. Sketch of a one-stage CO_2 transcritical heat pump system

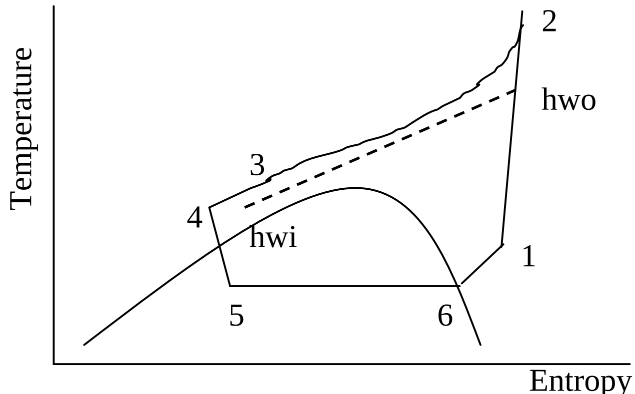

Fig 2 Temperature-entropy diagram for water heating with transcritical CO_2 cycle

is reduced from 4.1 to 2.7 if the hot water inlet temperature increase from 5℃ to 23℃ which spans the city water temperature range during the year in central China.

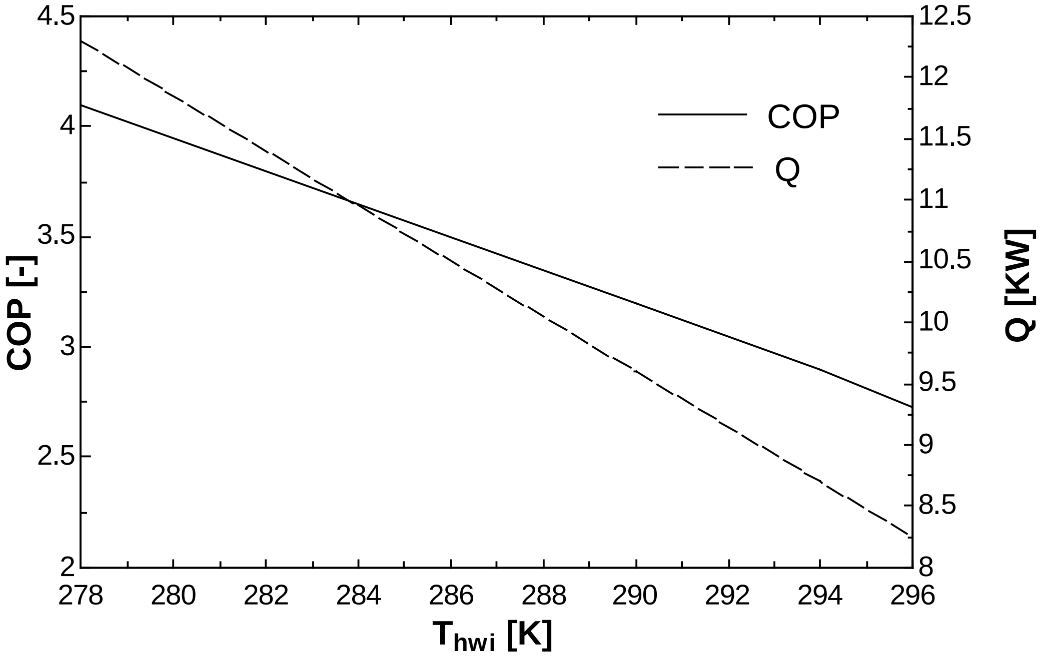

Fig. 3   Simulated COP and Q when varying the hot water inlet temperature

The effect of inlet heat source temperature on system performance is illustrated in Fig.5 .It is obvious that heat source inlet temperature exerts an opposite influence on COP and heating capacity. Considering that certain factors containing artificial recharge affects the temperature of groundwater, a wide temperature scope from 279K to 295K is chosen to model.

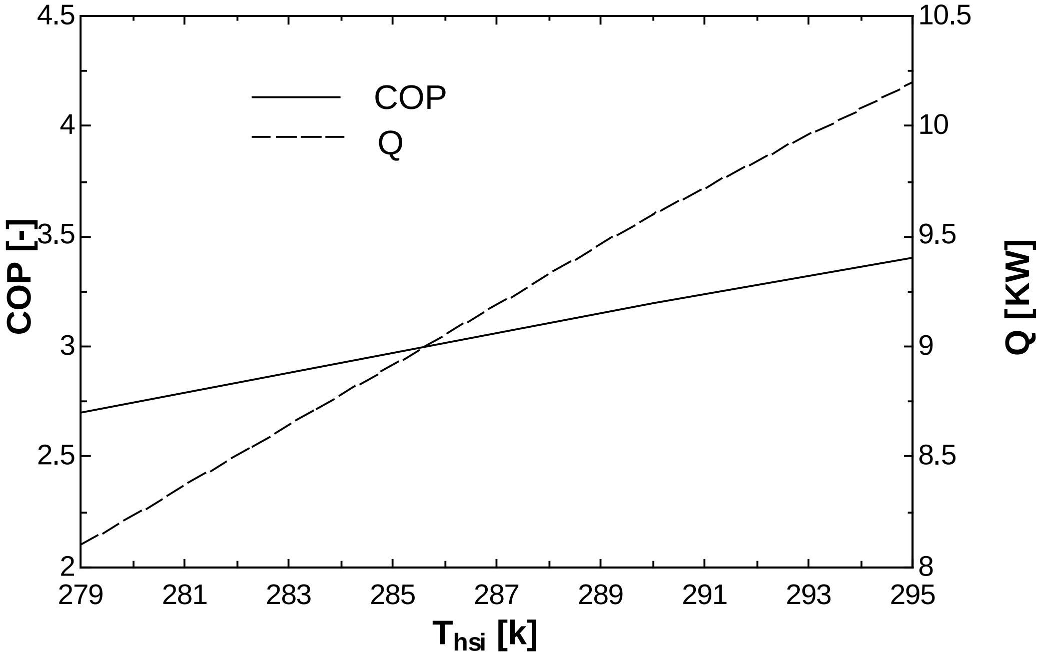

Fig. 4   Simulated COP and Q when varying the heat source inlet temperature

The simulation results reflect that the compressor speed is inversely proportional to system COP and Q.As the compressor speed increases,COP decreases. Rotation of compressor is proportional to compressor work and heating output, but compressor rises more than heating capacity. Consequently, COP falls with the increase of compressor rotate speed. Adding

$$S = \frac{1}{1 + 1.62 \times 10^{-6} E^{0.69} \mathrm{Re}_l^{1.11}} \qquad (12)$$

Secondly, if $\chi > \chi_{cr}$,

$$h_{tp} = \frac{\theta_{dry} h_g + (2\pi - \theta_{dry}) h_{wet}}{2\pi} \qquad (13)$$

Finally, if the tube is dry totally, the heat transfer coefficient is calculated as:

$$h_g = 0.023 \mathrm{Re}_g^{0.8} \mathrm{Pr}_g^{0.4} \frac{k_g}{D} \qquad (14)$$

CO_2 pressure drops are defined by Kim correlation [17] while water by the conventional Dittus–Boelter equation.

A computer code carried out in EES has been developed to design the transcritical CO_2 system for water heating. Due to the sharp variation of refrigerant properties near pseudocritical region, the entire length of the gas cooler was divided equally into n discrete segments to yield accurate simulation results. Energy equations were employed for each segments. Table 1 shows the detail design specifications for the water-to-water CO_2 HPWH.

Table 1 Design specifications for the water-to-water carbon dioxide HPWH

Compressor	Dorin TCS340/4-D:Semihermetics, reciprocating unit,1450rpm input rated power—3000W
Gas cooler	Counter-flow, tube-in-tube, single-pass, copper Length—10m,CO_2 outer diameter/thickness—0.010m/0.001m, water outer diameter/thickness—0.017m/0.001m
Internal heat exchanger	Counter-flow, tube-in-tube, single-pass, copper Length—1.9m, CO_2 outer diameter/thickness—0.010m/0.001m, water outer diameter/thickness—0.017m/0.001m
Evaporator	Counter-flow, tube-in-tube, single-pass, copper Length—3.8m, CO_2 outer diameter/thickness—0.010m/0.001m, water outer diameter/thickness—0.017m/0.001m
Expansion valve	Control high-side pressure

Simulation

Employing the same method as design procedure, a computer code based on EES has been developed to simulate the system. In the present study, simulation results of design conditions and effects of altering parameters (inlet temperature of heat source, inlet temperature of hot water and the speed of computer) on system performance are presented as below.

RESULTS AND DISCUSSION

Design conditions

Under the design conditions, namely, at the inlet heat source(groundwater) temperature of 17℃,the evaporation temperature of 5℃,the inlet hot water temperature of 17℃,the outlet hot water temperature of 65℃, the compressor power at 3000W, a heating COP of 3.2 is achieved and the optimum high-side pressure is controlled below 8.5MPa which benefits a lot for manufacturing materials and operation. A comparison between system with and without IHE was also made, and it was observed that with IHE, the system heating COP can be increased by more than 18%.

altering parameters

Fig. 4 shows the variation of heating capacity and COP when varying the hot water inlet temperature and keeping other parameters constant. It may be observed that the heating COP

frequency converter to constant speed Dorin CO_2 compressor can save energy and increase system efficiency.

4. CONCLUSION

The simulation results presented in the paper show that CO_2 is very suitable for tap water heat pumps. The prediction tends of system performance when altering the operation parameters is found to be in reasonable agreement with experimental data published. Due to the different working conditions, absolute values differ to an acceptable degree. Compared with the conventional electrical and natural gas water heater, the energy consumption of HPWH with CO_2 as working fluid can be greatly reduced. Obviously, the market potential for HPWH with CO_2 as refrigerant in central China is large,which is expected to be pushed by the present study and the further theoretical research , experiments and prototype design.

ACKNOWLEDGEMENT

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frequency converter to constant speed Dorin CO_2 compressor can save energy and increase system efficiency.

CONCLUSION

The simulation results presented in the paper show that CO_2 is very suitable for tap water heat pumps. The prediction tends of system performance when altering the operation parameters is found to be in reasonable agreement with experimental data published. Due to the different working conditions, absolute values differ to an acceptable degree. Compared with the conventional electrical and natural gas water heater, the energy consumption of HPWH with CO_2 as working fluid can be greatly reduced. Obviously, the market potential for HPWH with CO_2 as refrigerant in central China is large,which is expected to be pushed by the present study and the further theoretical research , experiments and prototype design.

ACKNOWLEDGEMENT

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Development and Performance Analysis of Two-stage Compression Variable Frequency Air Source Heat Pump

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SUMMARY

A two-stage compression variable frequency air source heat pump has been developed combining variable frequency technology with two-stage compression cycle, and two-control -model with the priority target as coefficient of performance (*COP*) or heating capacity is advanced in this paper. The mathematical model of the heat pump system has been developed and verified with the test data. The simulated optimal intermediate pressure and transition temperature between single-stage and two-stage compression heating condition are given, then the heating capacity, *COP*, and compressor discharge temperature are simulated and analyzed. It can be seen from the simulation results that *COP* of this heat pump system is over 2, the compressor discharge temperature under 120 °C, and the heating capacity can meet the heating load needed when the condensing temperature is 50°C and outdoor air temperature is -20°C.

INTRODUCTION

More and more heat pump systems have been used as the cold source and heat source of air conditioning systems due to their higher efficiency, pollution-free and multi-use. Among all kinds of heat pump units, air source heat pump is the most widely-applied equipment for its low initial investment, convenient maintenance and little influence to ambient. Since 1990's, air source heat pump had been successfully used for building indoor climate in the unconventional heating region of China including the middle and lower reaches of the Yangtze River, south and southwest China. However, when the air source heat pump is used in north China in winter, where the outdoor temperature is very low, there are some problems such as insufficient heating capacity, lower *COP*, and very high compressor discharge temperature except the frost on the outdoor heat exchanger, which block its application in those cold regions.

Besides the auxiliary heat source for the heating supply, much research had been made to solve the heating problems of air source heat pump in cold regions. A new type of heat pump with a kerosene-fired burner was developed to improve the heating capacity and the *COP* [1]. An air source heat pump with an inverter scroll compressor and liquid injection was proposed by Horiuchi [2], which increases the rotary speed under low ambient temperatures and injects liquid refrigerant into compressor to decrease the compressor discharge temperature. Based on the system proposed by Horiuchi, a system with an economizer and vapor injection was developed [3]. A two-stage coupling heat pump system was developed to use air source heat pump to absorb heat from outdoor air and to obtain water of 10-20 ºC, then make higher temperature water for building heating with another heat pump unit [4]. A ternary mixture refrigerant was used to improve the *COP* of heat pump for cold climates by Sami and Tulej[5].

Also a compression/absorption heat pump system was invented by Hulten and Berntsson [6], the waste heat is used to drive the absorption heat pump to get hot air, which is then used as the heat source of vapor compression heat pump, and finally get hot water for heating requirement. The research above has improved the heating performance of air source heat pump for cold climates with different methods.
In this paper, a two-stage compression variable frequency air source heat pump (TV-ASHP) system for cold regions is proposed combining variable frequency technology with two-stage compression cycle. With a variable frequency scroll compressor as the low pressure stage compressor and a fixed frequency scroll compressor as the high pressure stage compressor, the mathematical model of TV-ASHP system with a standard cooling capacity of 16 kW is developed, and the system performance is simulated and analyzed then.

DEVELOPMENT OF TV-ASHP

The TV-ASHP system developed in this paper is shown in Figure 1. The uncompleted intercooling two-stage compression cycle is adopted, and the system mainly includes a low pressure stage compressor, a high pressure stage compressor, an outdoor heat exchanger, an indoor heat exchanger, an intercooler, two throttling devices, a four-way valve, and a solenoid valve. Three work conditions including the single-stage compression cooling, single-stage compression heating and two-stage compression heating can be carried out by controlling four-way valve and solenoid valve and changing the refrigerant flow circuit.

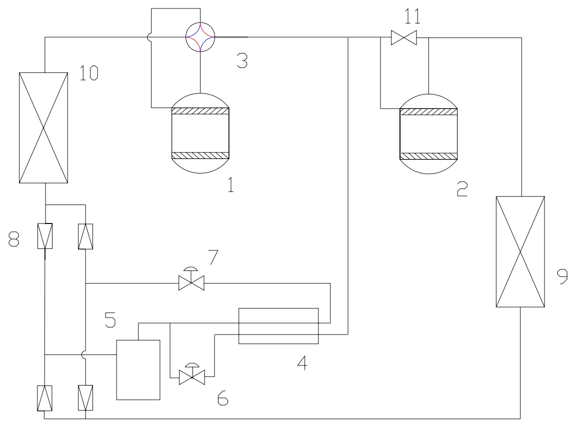

Figure 1. Schematic of TV-ASHP system
1- low pressure stage compressor 2- high pressure stage compressor 3-four-way valve
4- intercooler 5- receiver 6-throttling device I 7- throttling device II 8- one-way valve
9-indoor heat exchanger 10-outdoor heat exchanger 11-solenoid valve

Two-control-mode with the priority target as *COP* or heating capacity is advanced to assure the system higher *COP* when the heating capacity can meet the heating load. That is, when the heating capacity meets the heating load, the system works in the mode of *COP* priority and is kept the optimal intermediate pressure by changing the power frequency of low pressure stage compressor. But if the heating capacity is insufficient for demand, the control mode of heating capacity priority has to be chosen and the power frequency should be adjusted to meet the heating demand. With the TV-ASHP system, the heating capacity is enhanced with two-stage compression and increase of power frequency of low pressure stage compressor together, *COP* is improved with two-stage compression and optimal intermediate pressure by power frequency adjustment of low pressure stage compressor together, and the discharge temperature of high pressure stage compressor is decreased with the two-stage compression cycle, which are used to solve the insufficient heating capacity, lower *COP*, and high compressor discharge temperature in the common air source heat pump systems.

MATHEMATICAL MODEL AND EXPERIMENTAL VERIFICATION

Mathematical model

The mathematical model of TV-ASHP system mainly includes the sub-models of low pressure stage compressor, high pressure stage compressor, outdoor heat exchanger, indoor heat exchanger, intercooler and throttling devices.
The refrigerant mass flow rate and input power of low pressure stage compressor and high pressure stage compressor are fitted with the test data from the manufacturer:

$$\dot{m}_{\mathrm{l}} = \left(0.109 + 0.00305T_{\mathrm{e}} - 0.000253T_{\mathrm{m}} - 0.00000194T_{\mathrm{m}}^2\right)\frac{f - 2.857}{60 - 2.857}, \quad (1)$$

$$N_{\mathrm{l}} = \left(2.13 + 0.02T_{\mathrm{e}} + 0.013T_{\mathrm{m}} + 0.0009T_{\mathrm{m}}^2\right)\frac{f + 2.2}{60 + 2.2}, \quad (2)$$

$$\dot{m}_{\mathrm{h}} = 0.045 + 0.00149T_{\mathrm{m}} + 0.000072T_{\mathrm{c}} - 0.000000208T_{\mathrm{c}}^2, \quad (3)$$

$$N_{\mathrm{h}} = 0.98 + 0.000886T_{\mathrm{m}} + 0.00064T_{\mathrm{c}} + 0.000044T_{\mathrm{c}}^2, \quad (4)$$

where $\dot{m}_{\mathrm{l}}$, $\dot{m}_{\mathrm{h}}$ are the refrigerant mass flow rate through the low pressure stage compressor and high pressure stage compressor respectively, N_{l}, N_{h} are the input power of low pressure stage compressor and high pressure stage compressor respectively, T_{e}, T_{m}, T_{c} are the evaporating temperature, intermediate temperature, and condensing temperature respectively, and f is the power frequency of low pressure stage compressor.
The mathematical models of outdoor heat exchanger, indoor heat exchanger, and intercooler are all the distributed parameter ones.
With the inlet and outlet parameter relationship between every component in the TV-ASHP system, the system model is developed to combine all the sub-models. For the solution procedure, firstly set starting values of evaporating pressure, condensing pressure, intermediate pressure, suction superheat of low pressure stage compressor. Then start the simulation from the suction end of low pressure stage compressor, and call the subroutine of low pressure stage compressor, high pressure stage compressor, outdoor heat exchanger, indoor heat exchanger, intercooler and throttling devices. Finally adjust the assumed starting value of condensing pressure to keep mass conservation of heat pump system, adjust the assumed starting value of evaporating pressure to keep energy conservation, adjust the assumed starting value of suction superheat of low pressure stage compressor to keep momentum conservation of heat pump system, and adjust the assumed starting value of intermediate pressure to keep balanced refrigerant flow rate distribution.

Experimental verification

Figure 2 shows the test bench of TV-ASHP system whose standard cooling capacity is 16kW. The test bench consists of heat pump system, glycol-water system, indoor water system, control system, and measurement system. Three conditions including the single-stage compression cooling, single-stage compression heating and two-stage compression heating can be carried out at different condensing and evaporating temperatures.
The heat pump system is same with the TV-ASHP system shown in Figure 1. A 5HP variable frequency compressor is used as low pressure stage one while a 3HP fixed frequency

compressor is used in high pressure stage. The glycol-water system consists of plate-type heat exchanger, tank, electric heater, pump and flow meter. The mass concentration of glycol in the glycol-water solution is 50%, so the freezing point can be as low as -35 ºC, which can be used to simulate the very low outdoor environment. Indoor water system is constituted with plate-type heat exchanger, water tank, electric heater, pump and flow meter.

The parameters controlled in this test system are the power frequency of compressor, evaporating pressure, condensing pressure, intermediate pressure. The power frequency of low pressure stage compressor is adjusted with the variable frequency controller. The evaporating pressure is adjusted by the glycol-water temperature at outdoor heat exchanger inlet, and the condensing pressure is adjusted by the water temperature at indoor heat exchanger inlet. The intermediate pressure is controlled mainly with the power frequency of low pressure stage compressor.

The parameters measured in this test system include the refrigerant temperatures and pressures at different test points, water temperatures at indoor heat exchanger inlet and outlet, glycol-water temperatures at outdoor heat exchanger inlet and outlet, indoor water flow rate, glycol-water flow rate, compressor power consumption. There are twelve copper constantan thermocouples (measurement precision is ±0.2 ºC) set in the refrigerant circuit to obtain the suction temperature, discharge temperature, refrigerant temperature at the heat exchanger inlet and outlet, as well as the refrigerant temperature at the high pressure inlet and outlet of intercooler. Four voltage-output type pressure sensors with the precision of ±0.1% are used to measure the refrigerant pressures at the inlet and outlet of low pressure stage compressor, the refrigerant pressure at the outlet of high pressure stage compressor and throttle device. The water temperatures and glycol-water temperatures at heat exchanger inlet and outlet are measured with four platinum resistance temperature sensors, whose precision is ±0.1 ºC. The compressor power consumption is measured with the Fluke power meter. Two glass rotor flow meters, calibrated with a calibration cylinder, are used to measure the flow rate of glycol-water solution and indoor water. These sensors above all produce electric voltage or current, which are sent to a data logger and a computer to record the test data.

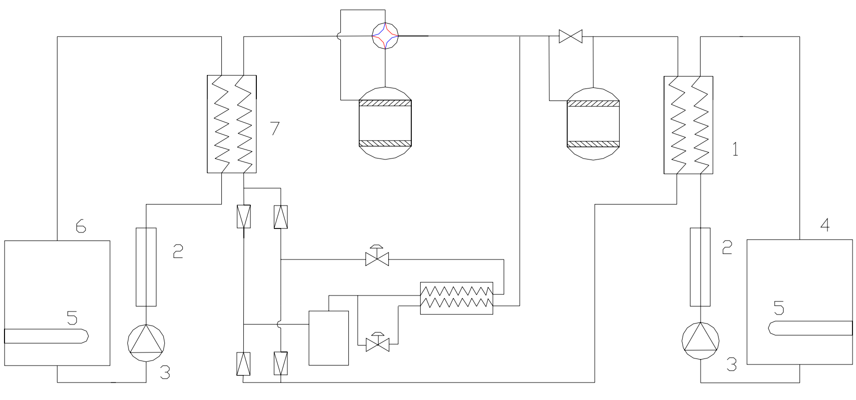

Figure 2. Test bench
1- indoor heat exchanger 2-flow meter 3-pump 4- water tank
5- electric heater 6- glycol-water tank 7- outdoor heat exchanger

Table 1 shows the comparison between the simulated results and test data when the system runs in the two-stage compression heating and the condensing temperature is 50 ºC. It can be seen from Table 1 that the deviations between simulated and measured values of heating capacity and *COP* are less than ±6%, and the simulated results agree well with experimental data.

Table 1. Comparison between simulated and measured results

Evaporating temperature (ºC)	0	-10	-20	-30
Heating capactiy (kW)				
Test data	19.86	16.82	13.4	10.02
Measured results	19.87	16.87	13.9	10.56
Relative error (%)	0.05	0.29	3.57	5.14
COP				
Test data	2.54	2.34	2.1	1.9
Measured results	2.64	2.32	2.14	2
Relative error (%)	3.88	0.85	1.79	5.24

SIMULATION AND ANALYSIS

With the verified mathematical model of TV-ASHP system, the optimal intermediate pressure and transition temperature between single-stage and two-stage compression heating condition are calculated firstly, and the heating capacity, *COP*, and discharge temperature of high pressure stage compressor are simulated and analyzed then.

Optimal intermediate pressure

When the heating capacity meets the heating load and the control mode of *COP* priority is chosen in the two-stage compression heating condition of TV-ASHP system, the optimal intermediate pressure, p_m, is the goal parameter for adjusting the power frequency of low pressure stage compressor. Therefore it is important to learn how much the optimal intermediate pressure is in the different conditions.
The simulated optimal intermediate pressure of TV-ASHP system is shown in Figure 3 when the evaporating temperature is 0 – -30 ºC and the condensing temperature is 50 ºC. The optimal intermediate pressure rises along with the increase of evaporating temperature.

Transition temperature between single-stage and two-stage compression condition

Figure 4 shows the comparison of *COP* in two-stage and single-stage compression heating condition when the condensing temperature is 50 ºC. Here the two-stage compression heat pump system is operating at the optimal intermediate pressure.

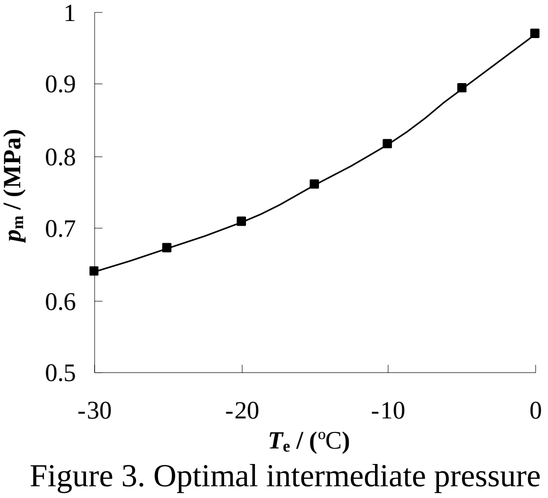

Figure 3. Optimal intermediate pressure

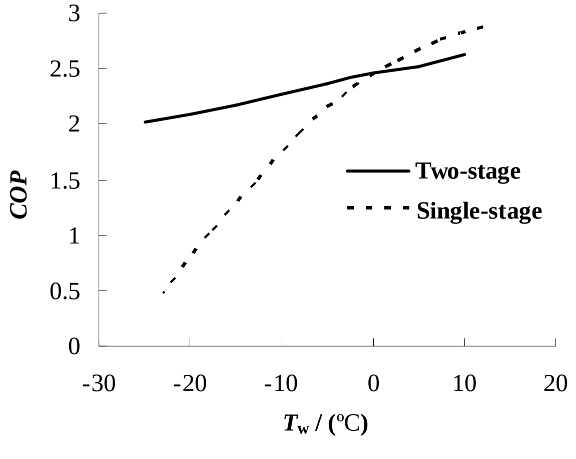

Figure 4. Comparison of *COP*

The *COP* of both two-stage and single-stage compression system goes down along with the decrease of the outdoor air temperature, T_w, but the decreasing rate is more less for two-stage compression system, especially when the outdoor air temperature is lower. As shown in Figure 4, the *COP* of two-stage compression system is equal to that of single-stage compression when the outdoor air temperature is 0 ºC. The *COP* of two-stage compression system is less than that of single-stage compression when the outdoor air temperature is higher than 0 ºC, and larger than that of single-stage compression when outdoor air temperature is lower than 0 ºC. So it can be considered the just temperature to transfer the control modes of the single-stage compression heating and two-stage compression heating when the outdoor air temperature equals 0 ºC. The single-stage compression heating mode is adopted when the outdoor air temperature is higher than 0 ºC, otherwise the two-stage compression heating mode is chosen.

COP in two-stage compression heating condition

The variation of *COP* in two-stage compression heating condition is simulated when the condensing temperature is 50 ºC and outdoor air temperature is 0 ºC, -10 ºC and -20 ºC (Figure 5). When the outdoor air temperature is kept constant and the low pressure stage compressor runs in a higher frequency, *COP* goes down along with the increase of power frequency. And the *COP* goes down along with the decrease of outdoor air temperature at constant power frequency. The *COP* is higher than 2.0 when the power frequency is lower than 70Hz, the outdoor air temperature is -20 ºC, and condensing temperature is 50 ºC.

Heating capacity in two-stage compression heating condition

Figure 6 shows the variations of the heating capacity, Q_c, along with the power frequency of low pressure stage compressor when the condensing temperature is 50 ºC and outdoor air temperature is 0 ºC, -10 ºC and -20 ºC. The heating capacity goes down along with the decrease of outdoor air temperature at constant power frequency. When the outdoor air temperature is kept constant, the heating capacity rises along with the increase of power frequency. So the TV-ASHP system can adjust the power frequency of low pressure stage compressor by the control mode of heating capacity priority to increase the heating capacity besides the improvement with the two-stage compression method when the two-stage compression in optimal condition alone cannot satisfy the heating requirement due to very low outdoor temperature. However, the *COP* goes down a little when the power frequency rises then (see Figure 5).

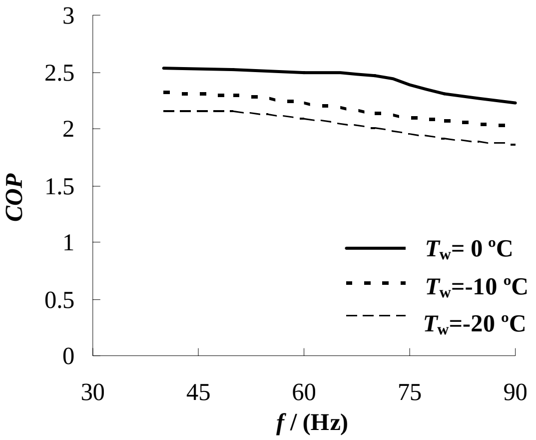

Figure 5. *COP*

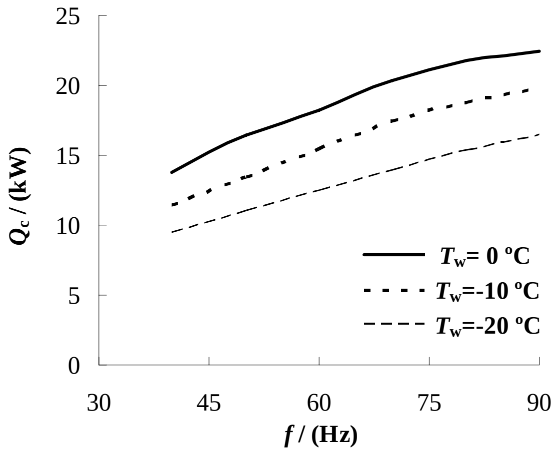

Figure 6. Heating capacity

Discharge temperature of high pressure stage compressor

The variations of the discharge temperature of high pressure stage compressor, T_d, in two-stage compression heating condition along with the power frequency of low pressure stage compressor are shown in Figure 7 when the condensing temperature is 50 ºC and outdoor air temperature is -20 ºC. The discharge temperature increases slowly when the power frequency rises, but doesn't exceed 120 ºC, below which the compressor can operate safely.

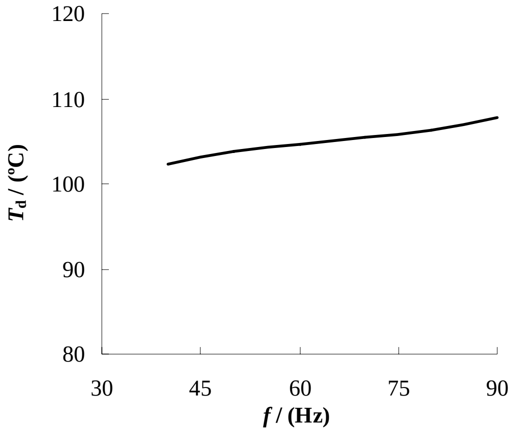

Figure 7. Discharge temperature

CONCLUSIONS

In this paper, a two-stage compression variable frequency air source heat pump has been proposed combining variable frequency technology with two-stage compression cycle, and two-control-mode with the priority target as *COP* or heating capacity has been advanced. The mathematical model of this heat pump system has been developed and verified. The optimal intermediate pressure and transition temperature between single-stage and two-stage compression heating condition have been simulated, then the heating capacity, *COP*, and discharge temperature of high pressure stage compressor have been simulated and analyzed.
It can be seen from the simulation results that *COP* of this heat pump system is over 2, the compressor discharge temperature under 120 °C, and the heating capacity can meet the heating load needed when the condensing temperature is 50°C and outdoor air temperature is -20°C.

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Refrigeration and cooling systems

Energy Efficient Operation of the Cooling Production System in a Commercial Building Under Hot Climate

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SUMMARY

Commercial buildings consume large amounts of energy under hot climates. The efficient operation of air-conditioning systems can lead to important energy savings. This work assesses the potential for savings through efficient operation of the cooling production system in a commercial building under hot climate. The cooling system of the building has a nominal capacity of 3380 Refrigeration Tons (RT) (11900 kW). Analysis of measurements from the building led to the adoption of a more efficient operation strategy for the cooling system. Implemented energy efficiency and conservation measures included: optimizing chilled water distribution, closing flow through non-operating chillers, increasing the loading of chillers, reducing the number of operating pumps and increasing the chilled water supply temperature during non-occupancy and low-demand periods. These measures produced significant improvements in the power rating of chillers and total energy consumption. Daily energy savings of 17% to 40% were achieved by implementing these measures.

INTRODUCTION

Air-conditioning systems operating under hot climates are characterized by high energy consumption. In a country like Kuwait, it is estimated that comfort cooling in buildings accounts for 45% of the annual energy consumption and 70% of the national peak demand for electrical power. Therefore, energy efficient air-conditioning systems are essential for saving energy and for meeting the peak electrical load.

A cooling production system is designed to meet the maximum cooling load during worst conditions to ensure comfort for building occupants. For most of the time, however, the cooling system operates under part-load and/or favorable weather conditions. In the case of cooling systems with centrifugal chillers, part-load operation can deteriorate their energy performance. In addition, the temperature of chilled water leaving the chiller, T_{ws}, affects efficiency. Raising the value of T_{ws} at periods of low cooling demand can improve the performance of the chiller. In hot climates, the efficiency of a chiller with air-cooled condenser is strongly affected by ambient weather temperature. Lower air temperatures lead to reduced condensing temperature and improved efficiency. Thus, the actual efficiency of an air-conditioning system depends on operating conditions.

This work assesses the potential for energy savings through the efficient operation of air-conditioning systems in commercial buildings. The goal of the study is to improve the energy performance of the cooling production system in an office building and estimate the energy savings achieved.

Building and Cooling System

This investigation studied the main office building of the Kuwait Institute for Scientific Research (KISR). The building is an energy-efficient structure with well-insulated walls and roofs and double/triple-glazed fixed windows, meeting the building energy code of Kuwait. It has an approximate air-conditioned area of 23,470 m^2. The air-conditioning system accounts for nearly 70% of the annual electrical consumption of 14,000 MWh.

The air-conditioning system in the building is designed to provide year-round comfort conditions with constant air and chilled water circulation. The cooling production system, commissioned in 1984, consists of ten air-cooled chillers. Every chiller has a 373 kW centrifugal compressor and three condenser fans of 15 kW each, providing a cooling capacity of 338 Refrigeration Tons (RT) (1190 kW cooling). The cooling distribution subsystem has four chilled water pumps (including a standby pump) feeding water to 23 Air Handling Units (AHUs) and 67 Fan Coil Units (FCUs). The major components of the cooling system and their power requirements are presented in Table 1 and a schematic diagram of the cooling system is shown in Figure 1. The building is occupied between 07:30 and 15:00 h, from Saturday to Wednesday.

Table 1. Important features of the air-conditioning system under consideration.

Component	Quantity	Connected Load (kW/Unit)	Total Load (kW)
Chiller (compressor + condenser fan motors)	10 (1+3)	373 +3x15	4180
Chilled water pump	4	75	300
Supply air fans for AHUs	23	0.55-45	258
Return air fans for AHUs	15	0.25-22	114
Fan coil units	67	0.18	12
Exhaust fans		0.18	
Total connected load			4864

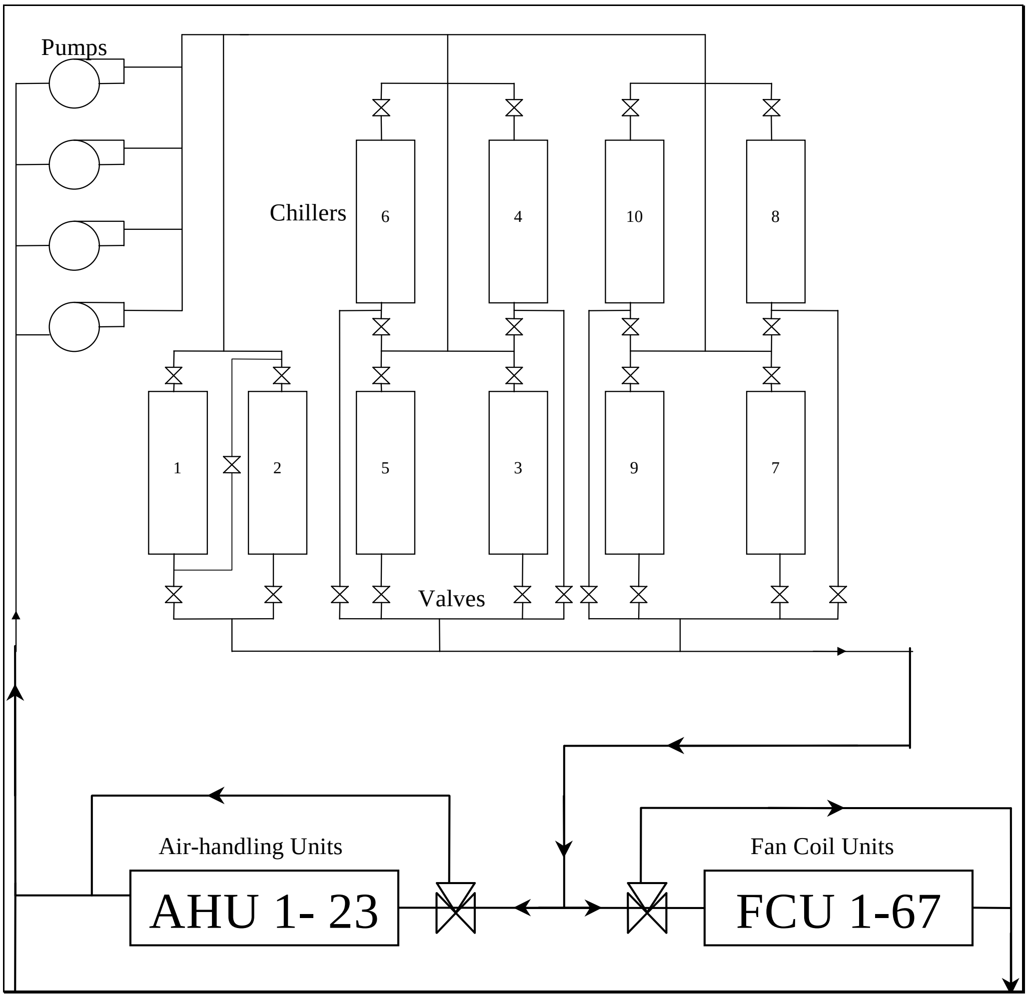

Figure 1. A schematic of air-conditioning system in the building.

METHODS

A key parameter in characterizing the efficiency of a cooling production system is the power rating (PR). The power rating of an air conditioning system and its components is the amount of power needed to produce a unit of cooling (Refrigeration Tons, RT). The power rating of a chiller ($PR_{chiller}$) is determined from

$$PR_{chiller} = \frac{P_{chiller}}{Q_c}, \qquad (1)$$

where $P_{chiller}$ is electric power used by the chiller, and Q_c is the amount of cooling produced. The power consumption of the chiller can be measured by a power meter. The amount of cooling is found from

$$Q_c = q\rho c_p (T_{wr} - T_{ws}), \qquad (2)$$

where q is chilled water flow rate, ρ is density, c_p is specific heat and T_{ws} and T_{wr} are the chilled water supply and return temperatures, respectively. The power rating of the system when chillers and pumps are considered together is

$$PR = \frac{P_{chiller} + P_{pump}}{Q_c}, \qquad (3)$$

where P_{pump} is the power consumed by the chilled water pumps and is related to the number of pumps in operation. Improving the efficiency of the system can be achieved by reducing the power rating of the chillers and the power consumption of the chilled water pumps. The air-distribution system was not considered in this study.

In order to determine *PR* for the cooling production system, the following parameters were monitored: $P_{chiller}$, P_{pump}, q, T_{ws}, and T_{wr}. The ambient temperature was also monitored as it affects the cooling demand of the building and the performance of the chillers. Current transformers and power transducers were used for power measurements. The water flow rate was measured by an ultrasonic flow meter. Chilled water temperatures were determined by high-accuracy Resistance Temperature Detectors (RTDs). Accuracy of temperature measurements was a concern because of the small rise in chilled water temperature (T_{wr} - T_{ws}), which impacts the accuracy of the cooling production in equation (2) and power rating in equations (1) and (3). Ambient temperature and humidity were logged using a weather station. A data acquisition system recorded measurements continuously every 10 minutes.

Analyzed data was used to establish the base-case performance of the cooling production system. The base-case data highlighted the opportunities for energy savings through improved operation of the system. A new operation strategy for the cooling production system was developed and implemented. The performance of the system before and after implementation of the operation strategy was compared to estimate energy savings.

RESULTS

Measurements of chilled water temperatures and flow rates revealed a problem in the cooling distribution system. One of the branches in the constant-flow chilled-water network had high cooling demand, which was originally met by increasing the system flow rate. Other chilled water branches showed low temperature rises, indicating high flow rates. Adjusting the flow rates in the branches according to the cooling demand reduced the total flow requirement and the pumping power. The number of chilled water pumps required during peak summer was thus reduced from 3 to 2. Additional pumping power was saved by removing a pump from operation during non-occupancy and low demand periods. Figure 2 compares the number of chilled water pumps in operation before and after the modification for two days of similar conditions. The figure manifests the reductions in number of operating pumps during occupancy and non-occupancy hours.

Analysis of temperature data showed that the temperature of water supplied to the air handling units (AHUs) was 1 to 3.5°C higher than the temperature of water produced by the chillers. Warmer water returning from AHUs was allowed to flow through non-operating chillers and mix with colder water before being supplied to the building. Cooling the water to temperatures below the supply temperature represented an energy loss. Closure of water flow through non-operating chillers resulted in better matching between the production and supply temperatures of chilled water.

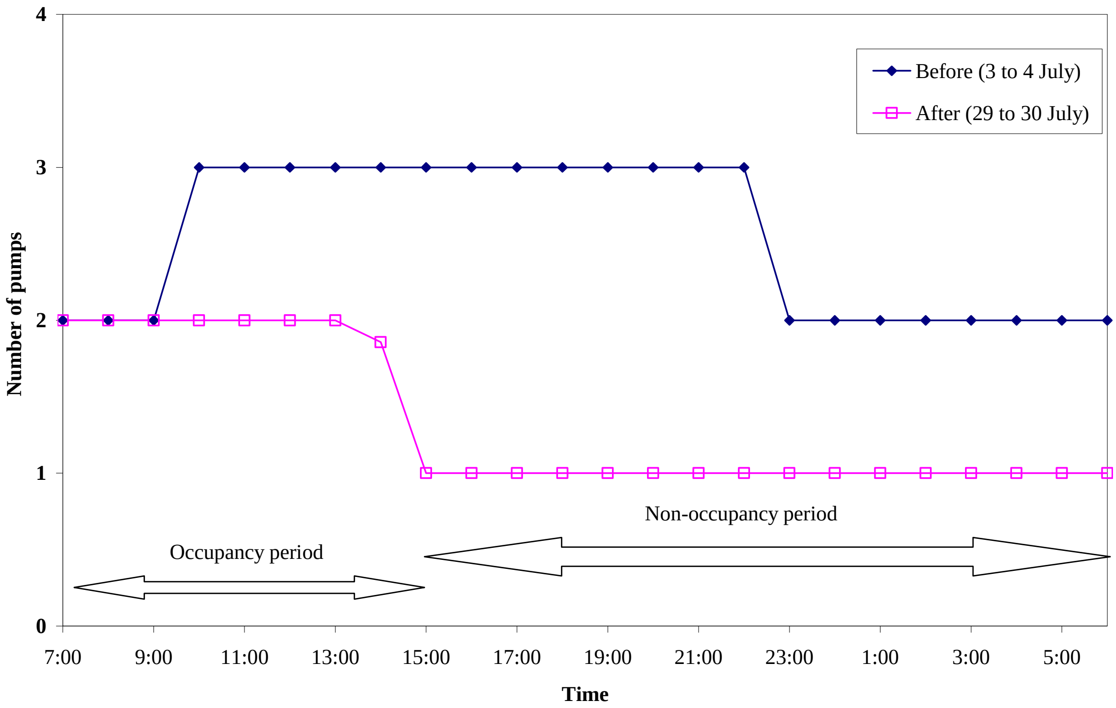

Figure 2. Number of operating pumps before and after optimization for two similar days.

Power measurements showed that chiller loading was low, leading to increased power rating. The loading of the chillers was increased by reducing the number of operating chillers when possible. Although maintenance concerns restricted maximum chiller loading to 70% during the peak summer, average loading was increased from about 55% to 65%.

Daily profiles of chilled water temperature manifested another opportunity for energy savings. The supply temperature of water was higher during the building occupancy hours and lower during non-occupancy hours. Allowing the supply temperature to rise during non-occupancy hours saves energy due to the reduction in number of operating chillers. Moreover, chiller efficiency improves at higher chilled water supply temperatures. The removed chillers were used to pre-cool the building during early morning hours, when weather conditions are milder, leading to better chiller operating efficiency.

Figure 3 shows the profile of chilled water supply temperature before and after optimization for two days of similar conditions. After optimization, the supply water temperature is lower during the occupancy period (7:30 – 15:00h), allowing for improved comfort. During non-occupancy hours, the value of T_{ws} is higher after optimization, as discussed above. Both profiles reach similar temperatures during early morning hours. The impact of increased chiller loading and supply water temperature is manifested in the chiller power rating, $PR_{chiller}$. Figure 4 shows the daily power rating profiles for operating chillers before and after optimization for two days of similar conditions. Although the power rating is high due to chiller age and harsh climate, a significant reduction in power rating was achieved.

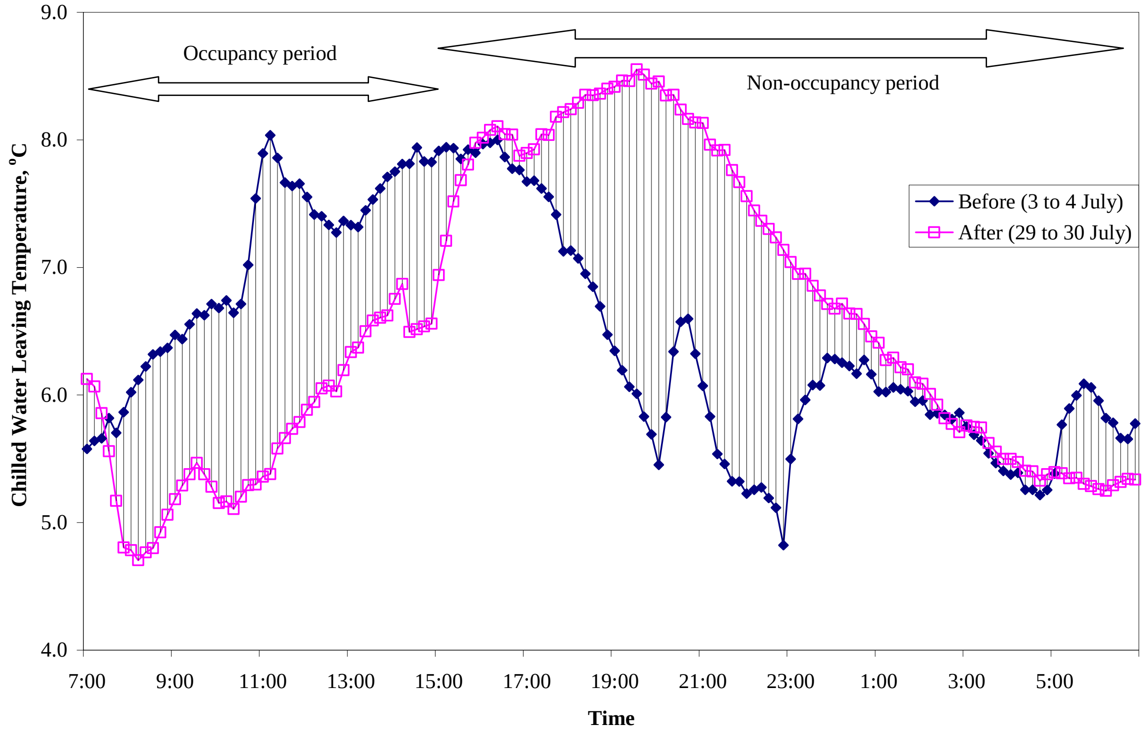

Figure 3. Profiles of chiller leaving water temperature before and after optimization.

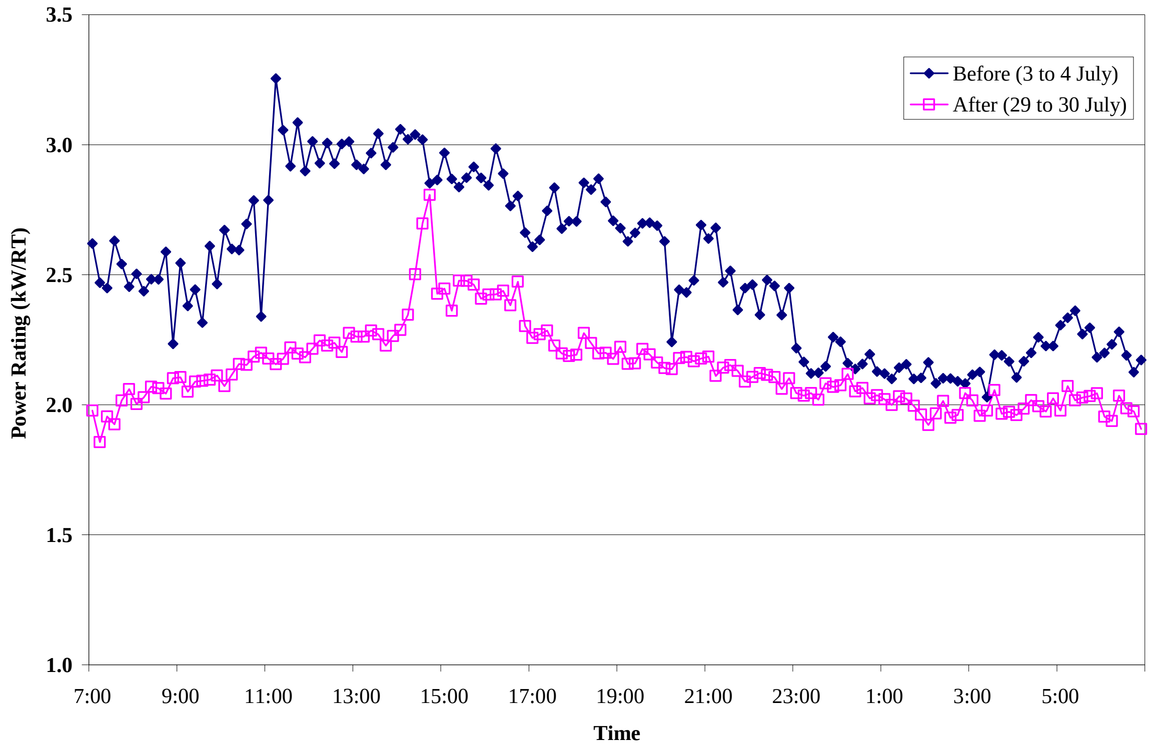

Figure 4. Average power rating (chillers and pumps) before and after optimization.

Table 2 summarizes the comparison results for two nearly identical days before and after optimization. In particular, the table shows a 15% improvement in average chiller efficiency, 43% reduction in daily pumping power, 19% reduction in daily energy consumption, and 14% reduction in peak power demand. The different temperature profiles of Figure 3 resulted in the same daily average chilled water supply temperature.

Table 2. Daily results for two similar working days before and after optimization.

Parameter	Before	After	Remarks
	3-4 July	29-30 July	
Average Ambient Temp, Deg C	36.8	36.0	
Maximum Ambient Temperature, Deg C	43.9	43.9	
Ambient Enthalpy, KJ/Kg	75.7	78.1	
Cooling Production, RTh	14930	14272	4.4 % Reduction
Average Water Supply Temperature, Deg C (non occupancy)	6.2	7.0	11.6 % Increase
Average Water Supply Temperature, Deg C (Occupancy)	7.0	5.7	18.2 % Reduction
Daily Average Chilled Water Supply Temperature, Deg C	6.5	6.5	
Total Energy (non occupancy), kWh	24953	18788	24.7 % Reduction
Total Energy (Occupancy), kWh	12660	11727	7.4 % Reduction
Total Energy, kWh	37613	30515	18.9 % Reduction
Peak power demand, kW	1806	1557	13.8 % Reduction
Average Power Rating, kW/RT	2.5	2.1	15.3 % Reduction
Average Chiller Loading, %	59.5	64.0	7.5 % Increase

Figure 5 shows the daily energy consumption before and after optimization as functions of the daily average ambient temperature. The energy efficiency and conservation measures resulted in daily energy savings ranging from 17% to 40%. Savings were higher for milder weather conditions.

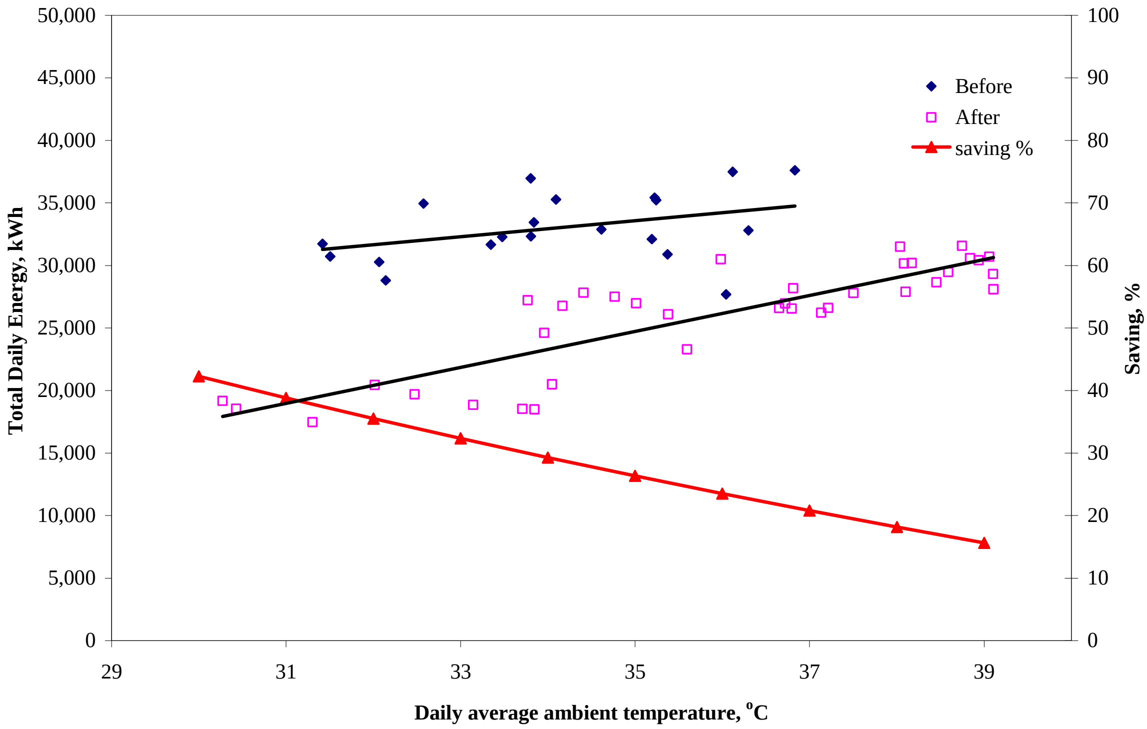

Figure 5. Daily energy consumption before and after optimization as functions of the daily average ambient temperature.

DISCUSSION

Efficient operation of the air-conditioning system in the commercial building under consideration led to significant energy savings. These savings resulted from improving the operating efficiency (power rating) of the chillers and reducing auxiliary power consumed by the chilled water pumps. The chilled water distribution problem is not uncommon in older commercial buildings. Expansions and changes in space usage within a zone can change the cooling load of that zone. In this work, optimization of water distribution improved comfort conditions as well as energy consumption. Similarly, poor chiller loading and other causes of inefficiency are often found in commercial and institutional buildings.

The energy efficiency and conservation measures implemented in this study were focused on operational issues. There are no costs associated with these measures except instrumentation and manpower, which are readily available for many commercial buildings. The implemented energy efficiency measures included optimizing the distribution of chilled water, closing the flow through non-operating chillers, and increasing chiller loading. The energy conservation measures included raising chilled water supply temperatures and reducing pumping power during non-occupancy and low demand periods.

ACKNOWLEDGEMENT

The authors would like to acknowledge the support of Kuwait Foundation for the Advancement of Science (KFAS) and Kuwait Institute for Scientific Research (KISR).

LCM Optimization by Using a Higher-Efficiency Chilled Water Supply System and Implementing Optimal Combinations of Maintenance Methods

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SUMMARY

Energy saving requirements make it imperative to cut CO_2 emissions, reduce energy consumption, and curtail costs at the same time. Life Cycle Management (LCM) evaluations are used to assess the effects of such energy saving efforts. In this study, LCM evaluations were conducted on the air-conditioning chilled water supply system installed as an ESCO facility in the cleanrooms at a semiconductor manufacturing plant. A detailed analysis of the overall system including the pumps, cooling towers, and chillers was conducted to identify the effects of using higher-efficiency absorption chillers and explore optimal mixes of maintenance methods. The analysis yielded quantitative indications of reduced Life Cycle Cost (LCC) and Life Cycle CO_2 ($LCCO_2$) emissions attributed to the use of advanced chillers. The scheme and timing of maintenance practices best suited for specific types of chillers and the degree of cooling water fouling were also identified.

INTRODUCTION

The concepts of LCM that minimize LCC and $LCCO_2$ are becoming permanent parts of modern corporate management. Many existing studies concerning LCM focus on LCA-based environmental evaluation techniques and LCC evaluations. The work done by Actacir[1], Elsafty[2], and others typifies the research conducted on LCC in the chiller air-conditioning sector. These studies are concerned at best with developing comparative evaluations of individual units of equipment, however, and have yet to launch a detailed probe into maintenance-recommended practices.

This study analyzed the effects of using higher-efficiency absorption chillers and explored optimal mixes of maintenance methods from the perspective of LCM evaluations of a chilled water supply system. The following summarizes the major aspects of this study.

- A comprehensive LCM evaluation of the overall chilled water supply system including the pumps, cooling towers, and chillers with regard to the partial load characteristics of the chillers and characteristics of cooling water temperature

- Identification of the optimal methods of maintaining absorption and turbo chillers from the standpoint of optimized LCM, by evaluating how maintenance methods make differences in their effects of energy saving on equipment and in the cost of is equipment.

SUMMARY OF LCM EVALUATION

The chilled water supply system under discussion was installed as an Energy Service Company (ESCO) facility. Because the system is committed to a contractual period of 10 years, both LCC and $LCCO_2$ were evaluated with regard to this period (see Figure 1).

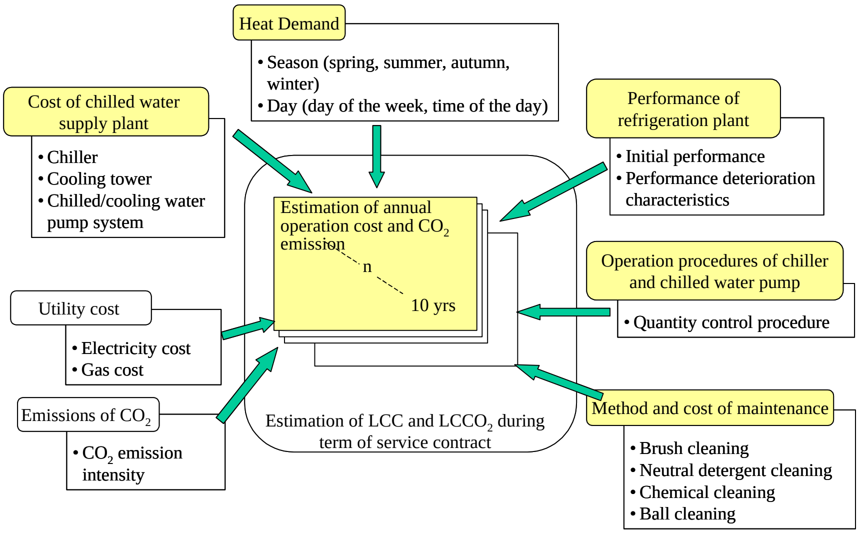

Figure 1. Evaluation system of LCM at refrigeration plant

Estimating LCC

LCC is the sum total of costs for initial equipment installation, operation and maintenance, plus the cost of equipment scrapped. The energy charge is for driving the chillers, cooling towers, and pumps. Annual costs have been evaluated on an NPV (Net Present Value) basis, with an assumed discount rate of 2% per annum.

Estimating $LCCO_2$

The $LCCO_2$ value of a chilled water supply system consisting primarily of chillers is the sum total volume of CO_2 emissions released during system construction and operation, and during the maintenance and scrapping of the chillers.

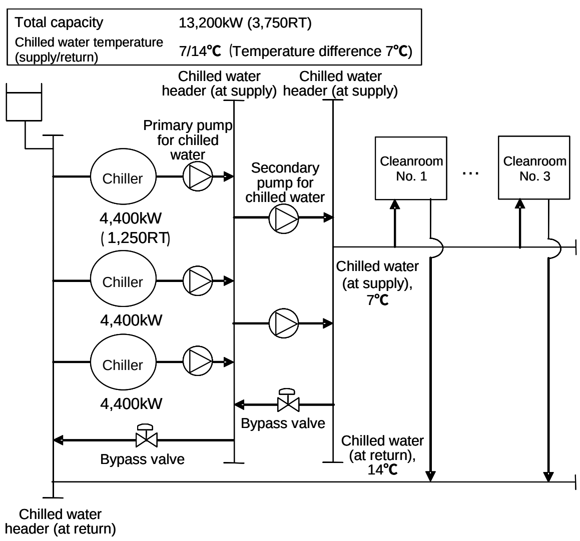

Figure 2. Schematic diagram of chilled water supply system

OUTLINE OF CHILLED WATER SUPPLY-DEMAND SYSTEM

Chilled water supply facility and heat demand

Chilled water is fed to the cleanrooms at the semiconductor manufacturing plant[3]. In the chilled water supply system shown in Figure 2, water is chilled to a temperature of

7°C when supplied and is at 14°C when returned. The maximum and minimum heat demands are 13,050 kW and 4,400 kW, respectively.
Figure 3 shows the heat load demand and annual cumulative time (in hours). The annual demand for heat load is 72.7 x 10^6 kWh. The annual demand for heat load divided by the maximum demand for heat load yields a total-load equipment operation time of 5,570 hours. This plant is running all year-round, with an average load factor as high as 64.5%.

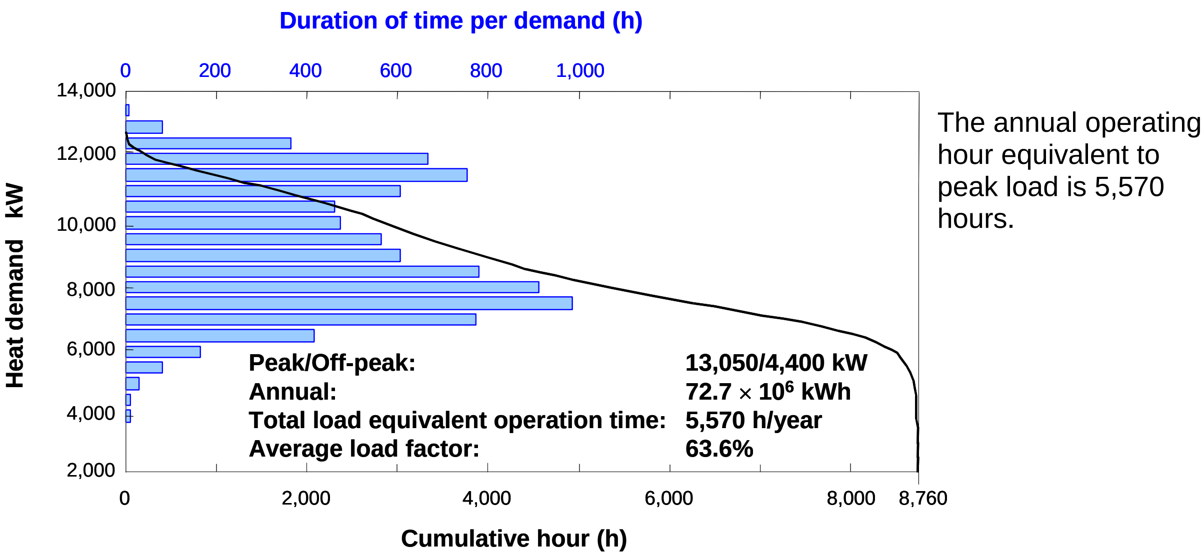

Figure 3. Head demand and annual cumulative time

Performance of chilled water supply system

(1) Turbo chillers

The turbo chiller model using HFC134a as a refrigerant has two-stage economizer and subcooler to achieve higher efficiency. This model conforms to standard specifications and has chilled-water inlet and outlet temperatures of 12°C and 7°C, respectively.

LCM evaluation requires the use of a database covering performance data on chillers working under wide operating conditions; thus we used a cycle simulator for turbo chiller performance data. The coefficient of performance (COP) was then calculated using a chiller load factor between 20% and the maximum load factor, and a cooling water temperature from 12 to 32°C, with and without using a compressor inverter.

Figure 4 plots the results of a simulation performed with cooling-water temperature settings of 24°C and 16°C, in addition to the rated 32°C.

The maximum output exceeds rated capacity in case of cooling water temperature less than 32°C. This characteristic can make the number of chillers to be run reduced.

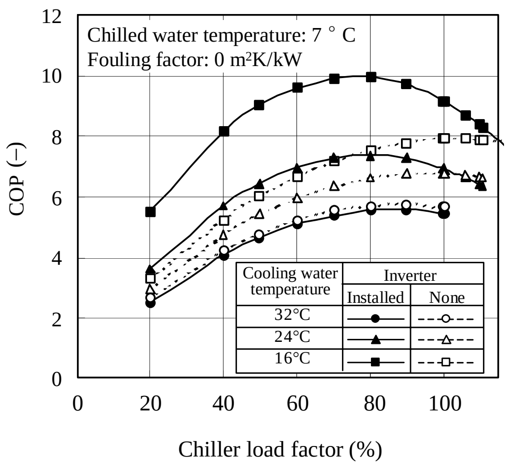

Figure 4. Turbo chiller partial load characteristics

(2) Absorption chillers (steam-driven double-effect/triple-effect)

a. Double-effect absorption chiller

The double-effect chiller (Photo 1) has evaporators and absorbers split into two stages to maintain a heat exchanger temperature difference, and recover heat from condensed refrigerant to achieve a steam consumption rate of 3.5 kg/RT.h and COP value of 1.5 during rated operations.

Photo 1. High efficiency double-effect steam-driven absorption chiller

b. Triple-effect absorption chiller

This study also considered a triple-effect absorption chiller. The first machine, which uses gas directly, achieved a COP value of 1.6 (HHV). We modified performance simulator from gas to steam medium, and an expected steam consumption rate can be 2.81 kg/RT.h and COP value can be 1.87 under rated condition.

Figure 5 shows the partial load characteristics of the double-effect and triple-effect absorption chillers. The COP value of the turbo chiller is converted to primary energy input. The power plant generating efficiency is assumed to be 41.9%.

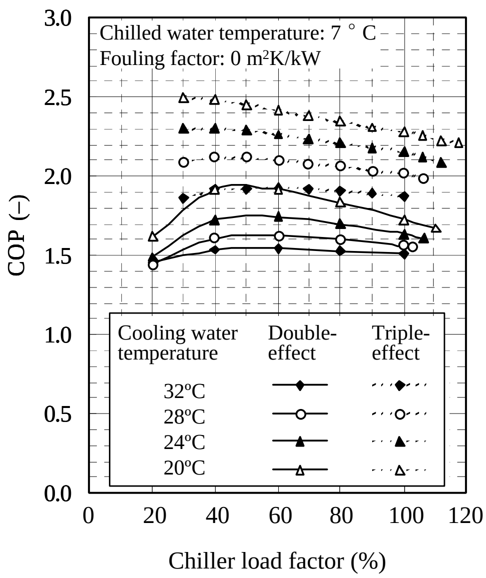

Figure 5. Partial load characteristics of a double-effect absorption chiller (steam consumption rate: 3.5 kg/RT.h) and triple- effect absorption chiller (steam consumption rate: 2.81 kg/RT.h)

Chiller performance effect due to fouled cooling water

Fouled cooling water could impair the heat transfer characteristics of the chiller condenser, resulting in degraded chiller performance. Using the cycle simulator, the chiller performance deterioration associated with fouled water has been determined by calculations (see Figure 6).

Changes in chiller performance without cleaning the CD tube have been calculated based on the assumption that the fouling factor would increase by 0.086 m^2K/kW per year and twice.

The chart reveals quantitative drops in cooling capacity and COP with increases in fouling. The absorption chiller suffers drastic drops in chiller capacity with increased fouling, but its COP only drops slightly.

The deterioration of chilling capacity with the progress of cooling water fouling is more marked on the absorption chiller than the turbo chiller. The rate of reduction in the COP is not much varied among the different chiller types.

Two kinds of cooling water have been selected to be fouled: good and poor. The effect of water fouling upon the condenser was assumed to be 0.86 (m^2K/kW) per year for good-quality water and 1.72 per for poor-quality one.

Cleaning CD tubes and resultant improvements in performance

Brush cleaning (BrC), ball cleaning (BaC), and chemical cleaning (ChC) are typically employed to clean inside the condenser (CD) tubes of turbo chillers. Conversely, cleaning using a natural detergent (NDC) instead of brush or ball cleaning is the preferred method of CD cleaning of absorption chillers. This study has extended its discussions to include four cleaning methods (see Table 1).

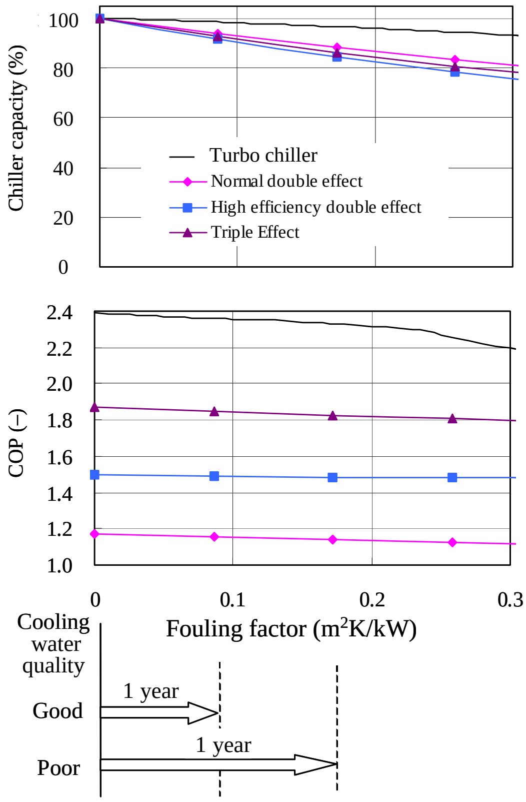

Figure 6. Chiller performance deterioration affected by cooling water fouling

Table 1. Methods of cleaning CD tubes in each chiller and their effects by cleaning methods

Maintenance method	Brush cleaning (BrC)	Neutral detergent cleaning (NDC)	Chemical cleaning (ChC)	Ball cleaning (BaC)
Applied chiller type	Turbo chiller	Absorption chiller	Turbo and absorption chillers	Turbo chiller
Cleaning method	Cleaning inside the CD tubes with a wire brush.	Cleaning inside the CD tubes with a neutral detergent.	Acid-cleaning inside the CD tubes	Inserting spongy balls into the CD tubes to remove contaminants.
Frequency	Once a year	Once a year	Once every several years	Once every several days
Effect (provisional value)	Up to 98% of the COP value obtained one year ago can be recovered.	Up to 98% of the COP value obtained one year ago can be recovered.	The condition of the chiller at initial installation can be restored.	Up to 99% of the COP value obtained one year ago can be recovered.

LCM EVALUATION

Evaluating LCM

1) Method of LCM comparative evaluation

The LCM of three types of chillers has been analyzed over a 10-year period. The following patterns of maintenance were discussed.

- Maintenance is conducted once a year.
- BaC is performed every week.

- At least three years must be allowed to elapse after former ChC practice to minimize the risks of local corrosion on the chiller condenser. (Once ChC is conducted in a given year, no other method of cleaning is employed at the same year.)

2) Method of calculating LCC

The following describes the method of calculating 10-year LCC for the chilled water supply system. The total amount of LCC (or LC) is represented in Eq. (1) as follows:

$$LC = IC + \sum_t \phi_E \overline{E}_E + \sum_t \varphi_E E_E + \sum_t \phi_G \overline{F}_G + DC \qquad (1)$$

where, IC denotes the investment cost, $\overline{E}_E$, the rate of power consumption [Nm^3/year], E_E the contractual demand [kW], $\overline{F}_G$ the gas consumption rate [Nm^3/year], t a 10-year period, ϕ the electric energy charge 7.7 [yen/kWh], φ_E the electric basic charge 1,740 [yen/kW month], and ϕ_E the gas energy charge 35 [yen/Nm^3]. The third term of Eq. (1) is zero for a turbo chiller, because it does not use gas.

3) Method of calculating $LCCO_2$

$LCCO_2$ is determined by summing up the volume of CO_2 emissions calculated from the energy emitted over a 10-year period and that of CO_2 emissions at the time of chilled water supply system initial manufacturing/installation, and scrapping. The total volume of CO_2 emissions of each system is calculated as LE by solving Eq. (2) as follows:

$$LE = E_I + \tau_E \cdot C_E + \tau_G \cdot C_G + E_S \qquad (2)$$

where,
E_I: Volume of CO_2 emissions at chiller manufacturing and installation [t-CO_2]
τ_E: CO_2 emission intensity due to electric power (0.555×10^{-3}) [t-CO_2/kWh]
C_E: Total electric power consumption rate [kWh]
τ_G: CO_2 emission intensity due to gas (2.097×10^{-3}) [t-CO_2/Nm^3]
C_G: Total gas consumption rate [Nm^3]
E_S: Volume of CO_2 emissions at equipment scrapping [t-CO_2]

Table 2 lists the volumes of CO_2 emissions [t-CO_2] at system and scrapping.

Table 2. Volumes of CO_2 emissions at chiller installation and scrapping

Type of chiller		Volume of CO_2 emissions at manufacturing and installation (t-CO_2)	Volume of CO_2 emissions at scrapping (t-CO_2)	Total volume of CO_2 emissions
Turbo chiller		564	60	624
Absorption chiller	Double-effect	345	40	385
	Triple-effect	414	50	464

4) Costs of chillers and maintenance equipment/procedures

Table 3 lists different types of chillers and maintenance equipment (BaC) and the costs incurred to maintain them by chiller condenser cleaning method. The triple-effect absorption chiller supply facility requires the cost of equipment, 1.8 times higher than turbo chillers. Among the four cleaning methods, ChC is most costly.

Table 3. Costs of chillers and their maintenance

Item \ Model		Turbo chiller	Absorption chiller: Double-effect (COP) 1.17	1.3	1.5	Triple-effect (COP 1.87)
Equipment and installation cost (million yen)	Chilled water supply facility (system as a whole)	375	453	466	488	675
	Maintenance equipment (BaC, common use)	+10	-			-
Maintenance cost (million yen/time/system)	BrC	1.5	-			-
	NDC	-	1.8	1.8	1.8	2.4
	BaC	2.4	-			-
	ChC	6.0	6.0	6.0	6.0	6.9

Evaluating the effects of LCM by chiller type and cooling water quality

The LCM values of each chiller associated with differences in water quality were evaluated. Absorption chillers optimize their LCM when serviced by annual NDC, coupled with ChC conducted at four-year intervals, and turbo chillers optimize their LCM when serviced by BaC, coupled with ChC conducted at four-year intervals.

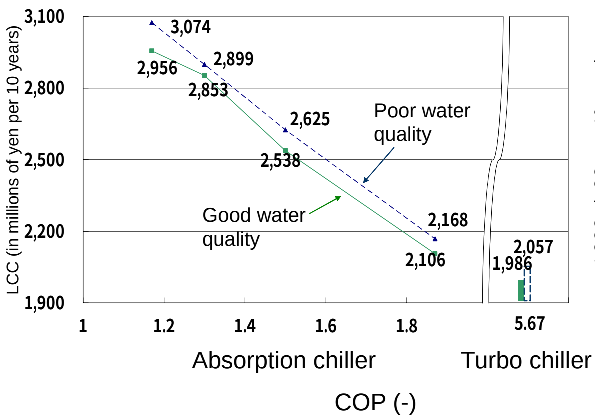

Figure 8. Evaluated effects of LCC by chiller's type

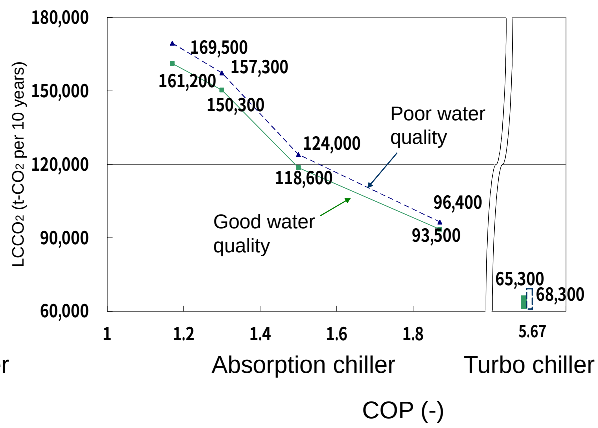

Figure 9 Evaluated effects of $LCCO_2$ by chiller's type

As shown in Figure 8, absorption chillers can reduce their LCC with increases in the COP. Turbo chillers are more cost-effective than absorption chillers. As long as the water is good, the difference between a triple-effect absorption chiller and a turbo chiller remains at about 120 million yen. If the water quality worsens, however, the difference would escalate by 11 million yen. Hence, turbo chillers are more affected by water quality than triple-effect absorption chillers.

As can be seen in Figure 9, turbo chillers are superior to absorption chillers in terms of $LCCO_2$. $LCCO_2$ affected by water quality is almost same as triple-effect and turbo chiller.

Figure 10 shows the relation between LCC and $LCCO_2$ in terms of maintenance and water quality. The arrow marks indicate the direction of an increasing degree of water fouling. LCC/$LCCO_2$ are found to be more noticeably affected by differences in water quality than differences in the maintenance method (standard or optimal maintenance).

Risks imposed by varying energy unit prices

Because the ESCO project is committed to a contractual period of 10 years, the electric power and gas rates are likely to vary in the meantime, resulting in changes in LCC and hence in business economics. Figure 10 presents an evaluation of the cost advantages of triple-effect absorption chillers and turbo chillers against changes in the electric energy charge and gas unit price.

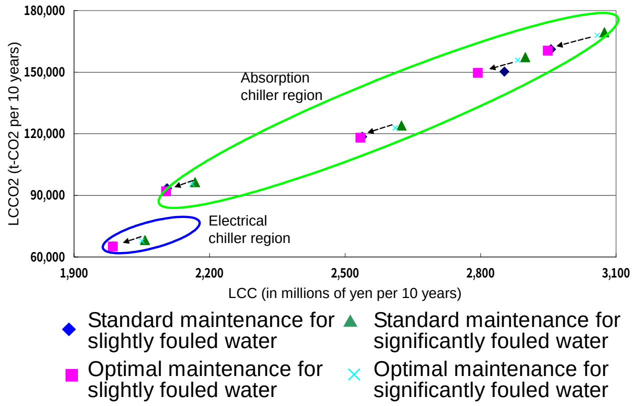

Figure 10. Relation between LCC and $LCCO_2$

In the figure, the upper-left region above the continuous solid line denotes the region in which electric turbo chillers come more advantageous than gas chillers. Figure 11 demonstrates that turbo chillers are more advantageous under the present conditions of review. Moreover, the less the cooling water is fouled, the wider the area where turbo chillers are effective.

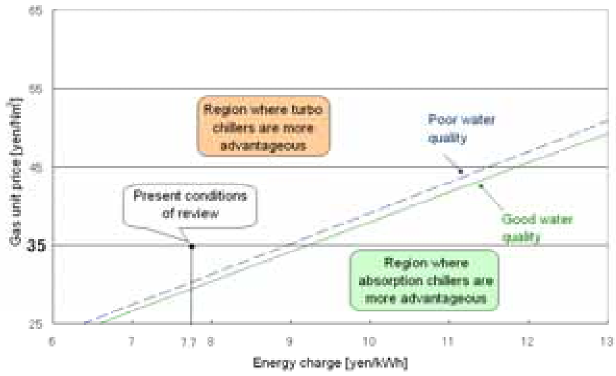

Figure 11. Decision criteria for making energy choices

CONCLUSIONS

This study has conducted an LCM evaluation of turbo, double-effect absorption and triple-effect absorption chillers and, what is characteristic, has demonstrated quantitative measures of reductions in both LCC and $LCCO_2$. Information derived from this study includes:

1) Depending on combinations of cleaning methods and their frequencies, optimal maintenance patterns exist to suit specific refrigerator types.
2) Absorption chillers coupled with neutral detergent cleaning conducted every year and chemical cleaning conducted at 4-year intervals minimize both LCC and $LCCO_2$.
3) Turbo and triple-effect absorption chillers virtually equaled in their evaluation of LCC, but electrical chillers prove better when it comes to minimize $LCCO_2$.

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Benchmarking Cooling Efficiency in Server Rooms

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SUMMARY

This paper discloses a new set of design guidelines for choosing an energy efficient server room design. Choosing the right server room design can secure very fast money recuperation when selecting a better cooling system than the cheapest available.

The ideal server room has a physical barrier between the hot and cold side. If a physical barrier is not possible CFD-calculations show a large savings potential of electricity for cooling machinery just from choosing the optimal room and server positioning design. With a demand of a specific maximum temperature at the server inlet and a estimated savings potential of 65- 130 Euros/(year*kW*K) and the difference between the best and the worst model design is 8 °C. The servers are consuming 30 kW electricity all year round generating equal amount of heat. The savings potential for this room is 15,600 - 31,200 Euros per year.

INTRODUCTION

Server rooms have large energy consumption thus demanding large cooling effects. The distribution of cooling air was thought essential in order to reduce the power consumption by the cooling machinery for this purpose. When the evaporation temperature of a cooling system is lowered the COP is reduced thus increasing the power consumption for removing the heat from the server room. With a demand of a specific maximum temperature at the server inlet and a estimated savings potential of 65- 130 EURO/(year*kW*K) it was considered essential to be able to control the air flow in the server room avoiding short circuiting of hot exhaust air from the server outlets to the server inlets.

In collaboration with the Danish Electricity Savings Trust, Birch & Krogboe has performed large series of CFD calculations testing the consequences of different common ventilation lay-outs in a small room with 2x3 rack central in the room. The scope of these calculations was creating a benchmarking or an energy classification labelling for server room design. Later the results will be used for guidelines for server room design at the Danish Electricity Savings Trust homepage and will probably be launched late 2007.

Different layout where tried such as fan coils placed in different positions, servers positioned either face to face, face to rear or rear to rear, inlets placed in an elevated technique floor on the server face side and outlets in different positions in the ceiling.

Ideal configuration of server room would be a room with a complete separation of the cold inlet side of the servers and the hot exhaust side and a critical amount of cooling air supplied to the room. Restrictions against mechanical obstructions in the room meant that the exhaust air from the servers had to be guided to the ventilation outlet by the air itself without creating too much short circuiting between the server's outlet and inlet.

Parameters such as air mixing index, scattering and/or enveloping of the hot air, and necessary inlet temperature of the cooling air was derived from the results of the CFD-calculations for different server room arrangements.

METHODS

All calculations were carried out with a commercial general purpose Computational Fluid Dynamics code (CFD). All CFD calculations were steady state, the turbulence model is a Reynolds Average Navier Stokes model the RNG- $k\varepsilon$. 3 different models have been used in a number of variations. Models are processed for 100 iterations. This gives a convenient balance between accuracy and calculation time. The calculations are not fully converged in accordance with the convergence criteria, but solutions are found to be sufficiently precise for a qualitative as well as quantitative comparison between the different setup.

There are 3 basic models, SK2, SK3 and SK4. Each calculation has a name describing what model is used, the orientation of the servers and finally the position of the fan coil/outlet eg. SK2FRS means model SK2 with servers orientated front toward rear (FR) in the two rack lines. The S describes the position of the fan coil outlet at the side of the room. Other positions are at the end of the room (E) and in the ceiling (C).

The SK2 model measures lxwxh 6000x4200x2400. The blue blocks are each 3 racks placed side by side. The racks are wxdxh 1800x1200x2000 mm. Servers are modelled as square tubes, 100x100 mm, passing through the racks. Upper, middle and lower row are active server, each producing 1667 W, total for the servers 30 kW. The air direction kan be selected in both directions through the active servers. The remaining tubes are leaks through the racks where the air is free to pass in any direction. A fan coil can be activated either at the end or at the side of the room, only by the ceiling. Initially airflow through the fan coil are 4 times the airflow through the servers in order to let the temperature difference through the servers be 20 °C and 5 °C through the fan coil/cooling system. Inlet temperature is 20 °C for all calculations. The SK2 model has a calculation grid of 385.330 cells.

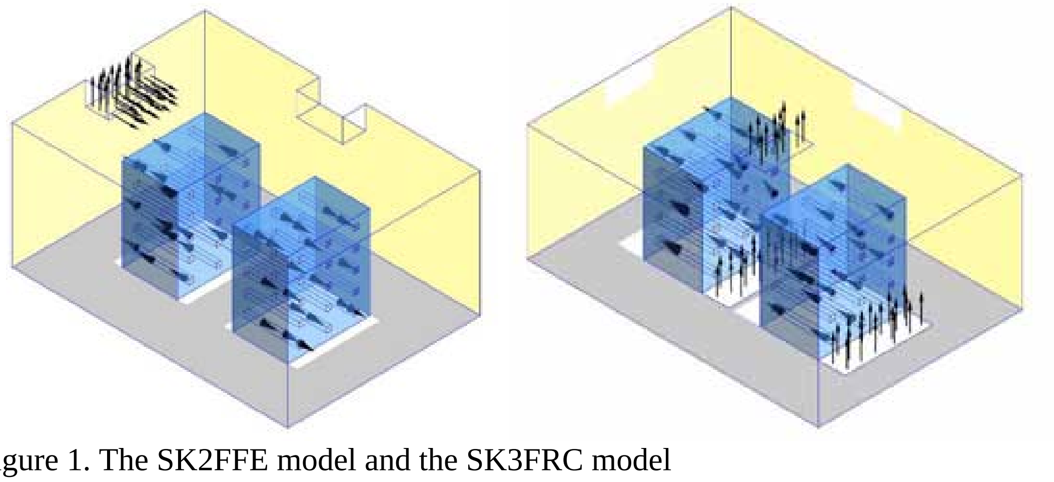

Figure 1. The SK2FFE model and the SK3FRC model

The SK3 model is almost equal to the SK2 model. The fan coil boxes have been replaced with openings in the wall for outlets. A third outlet possibility is situated in the middle of the ceiling. Cooling air inlets are placed in the floor by the front of the servers. The SK3 model has a calculation grid of 332.991 cells.

The SK4 model is similar to the SK3 model having more outlet positions in the ceiling in order to place the outlet opening in the most appropriate position. The model has furthermore a reduced room height, only 2100 mm. The SK3 model has a calculation grid of 407.205 cells.

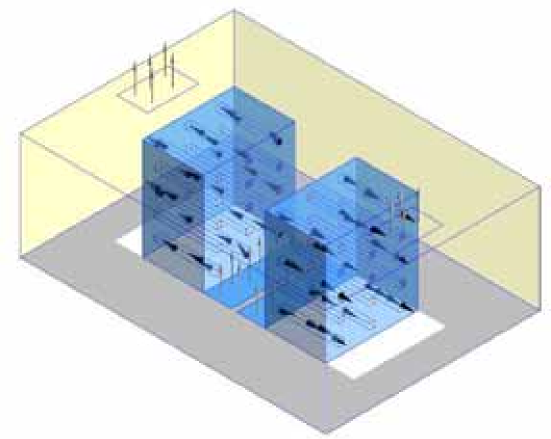

Figure 2. The SK4RRC model

5 main rounds of calculations were carried out:

1. The SK2-model. Fan coil placed at 2 different locations 3 positions of servers in racks (FF front to front, FR front to rear and RR rear to rear), room height 2400mm
2. The SK3-model. Inlet in installation floor by front of servers, outlet at 3 different locations, 3 positions of racks, FF, FR and RR , room height 2400mm
3. The SK4-model. Inlet in installation floor by front of servers, outlet at different locations (ceiling only), 3 positions of racks, FF, FR and RR , room height 2100mm
4. All models. Effect on reduced air volume ratio of cooling air. Inlet in installation floor, outlet at 3 different locations, 3 positions of racks.
5. The SK4FFC-model. Effect on different air volume ratio cooling: servers 4:1 VS 3:1 VS 2:1 VS 1:1; Inlet in installation floor. Racks only FF. Outlet only in ceiling.

All results were checked for all server inlet temperatures, best and worst value chosen, model maximum temperatures, flow speed and flow patterns. From all data was various parameters calculated. The following 3 are the most significant:

The Mixing Index (MI) describing the air quality for the most critically placed server. MI=0 means all server inlet air comes directly from cooling inlet with no mixing af warmer air. MI =100 means a 100 % short circuit where all server exhaust air are led to a server inlet with no mixing of fresh cooling air. The MI is divided in the index classes below:

A	0-4.9
B	5.0-9.9
C	10.0-19.9
D	20.0-29.9
E	30+

The Sub Cooling(SC) gives a direct indication of how much the server room cooling system ha to sub cool the inlet air compared to an ideal system in order to secure all servers in the room are supplied with air below the required maximum temperature. In the ideal system SC is 0 ºC. For increasing values of SC the evaporator temperature and COP of the cooling system are decreasing causing increased energy consumption for the cooling of the server room. In fact the SC is probably the most important index on the server room energy efficiency. The SC is divided in the index classes below:

A	0 - 0.99
B	1 - 1.99
C	2 - 2.99
D	3 - 4.99
E	5+

The overall Energy Class Index is derived from MI and SC and server inlet and mean temperatures and model maximum temperatures. The Energy Class Index is divided into equivalent 5 classes as seen above for MI and SC.

RESULTS

As an example model SK2FFE will be demonstrated thoroughly. Other results will be listed in table 2.

MODEL SK2FFE:

Servers placed front to front. Cooling air enters room by ceiling from fan coil on end wall. Exhaust at the same location from the down side of the fan coil. See figure 1.

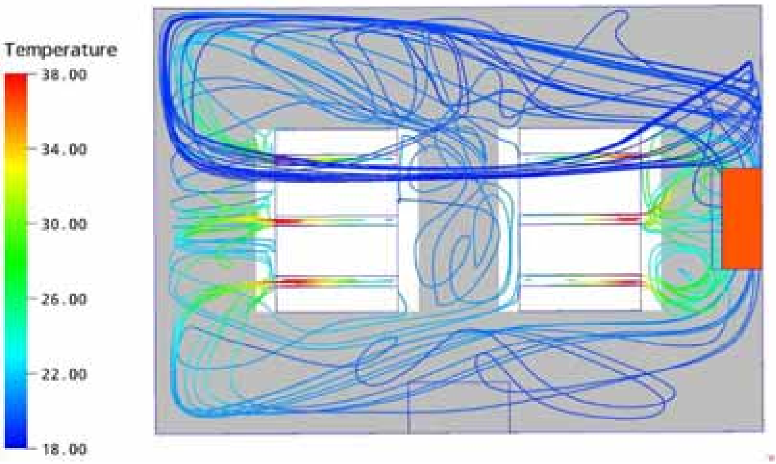

Figure 3. The SK2FFE model seen from above. Fan coil to the right

Table 1. The evaluation of the SK2FFE model

Model	DeltaTmax	Mixing index	Sub cooling	Energy Class
SK2FFE	24.5	17.0	3.4	C

This design gives a substantial mixing of hot air to the servers placed near the outlet of the fan coil. If all servers must have air supply below 20 ºC fan coil have to supply inlet air to the room at maximum 16.6 ºC.

When you have to use a fan coil placed near the ceiling it doesn't matter whether it is placed on the side wall or on the end wall. The significantly best results come from placing the servers front to front. MI 16 – 17 and sub cooling 3.2 – 3.4 °C. All other server configurations give MI 21.5-40.0 and sub cooling 4.3 – 8.0 ºC.

Table 2 Results of initial server room index calculations

MODEL	MIXING INDEX	SUB COOLING	ENERGY CLASS	COMMENT
SK2FFE	17.0	3.40	C	
SK2FFS	16.0	3.20	C	
SK2FRE	40.0	8.00	E	
SK2FRS	23.5	4.70	D	
SK2RRE	21.5	4.30	E	
SK2RRS	24.0	4.80	E	
SK3FFE	2.0	0.40	A	
SK3FFC	8.0	1.60	B	
SK3FFS	1.3	0.25	A	
SK3FRE	7.0	1.40	B	
SK3FRC	14.0	2.80	C	
SK3FRS	11.0	2.20	C	
SK3RRE	21.0	4.20	D	
SK3RRC	1.3	0.25	A	
SK3RRS	39.0	7.80	E	
SK4FFC	0.3	0.05	A	
SK4RRC	0.5	0.10	A	

Series of calculations with reduced cooling air or/and increased flow through servers all showed lesser results than the one listed in table 2 when comparing each setup.

One SK4-model (SK4FFC) was chosen in order to examine when a critical change in air supply compared to the air flow through the server racks.

Effect of different air change rates:

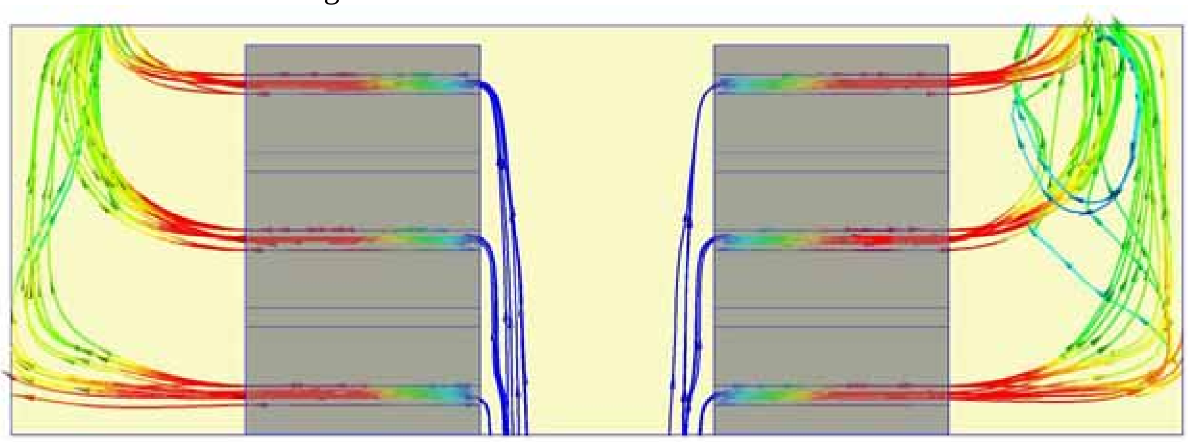

Figure 4 The SK4FFC100 model. Flowlines showing temperature

The SK4FFC100 model has an air change ratio between cooling system and through servers of: 4:1 In this case there is full control with the hot exhaust air which is completely engulfed in an envelope of cold cooling air. MI: 0.25, SC: 0.05°C, Energy Class A.

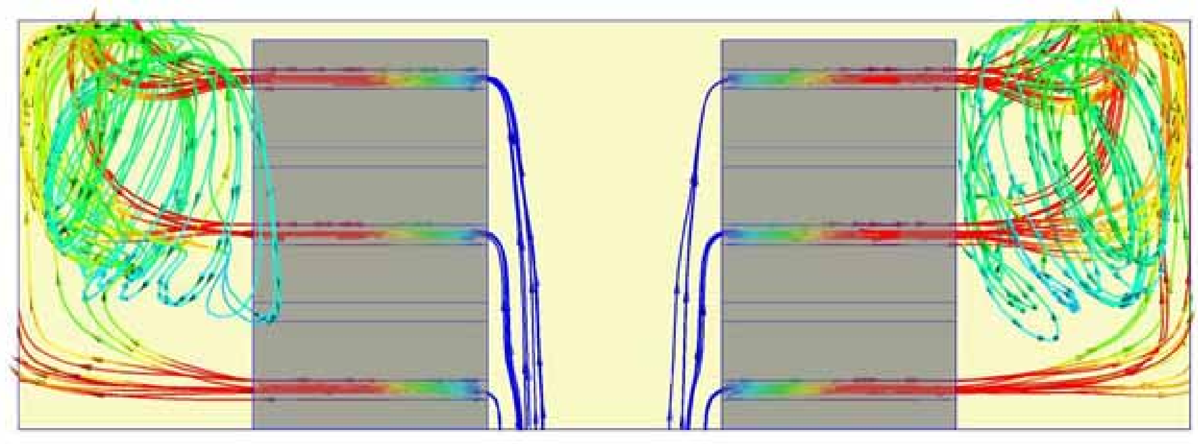

Figure 5 The SK4FFC75 model. Flowlines showing temperature

The SK4FFC75 model has an air change ratio between cooling system and through server of: 3:1 In this case there is control with the hot exhaust air which is partly engulfed in the cold cooling air. The flow patterns show some scattering and mixing but no back flow of hot air to the server inlet side. MI: 0.50, SC 0.10°C, Energy Class A.

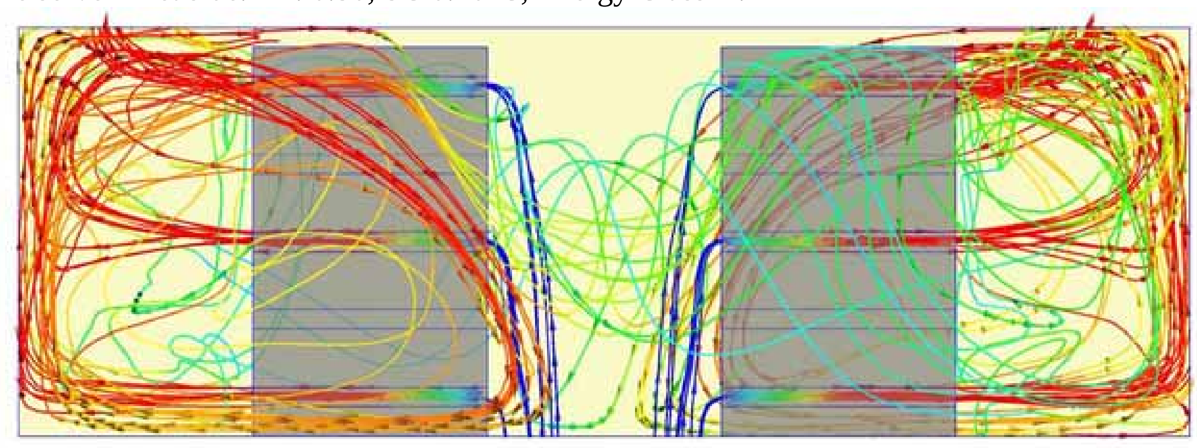

Figure 6 The SK4FFC50 model. Flowlines showing temperature

The SK4FFC50 model has an air change ratio between cooling system and through server of: 2:1 In this case there is control with the hot exhaust air which is no longer engulfed in the envelope of cold cooling air. The flow patterns show scattering of the hot air and. It is primarily the servers by the edges of the rack that receives premixed thus warmer air.MI: 3.0, SC 0.6°C, Energy Class B.

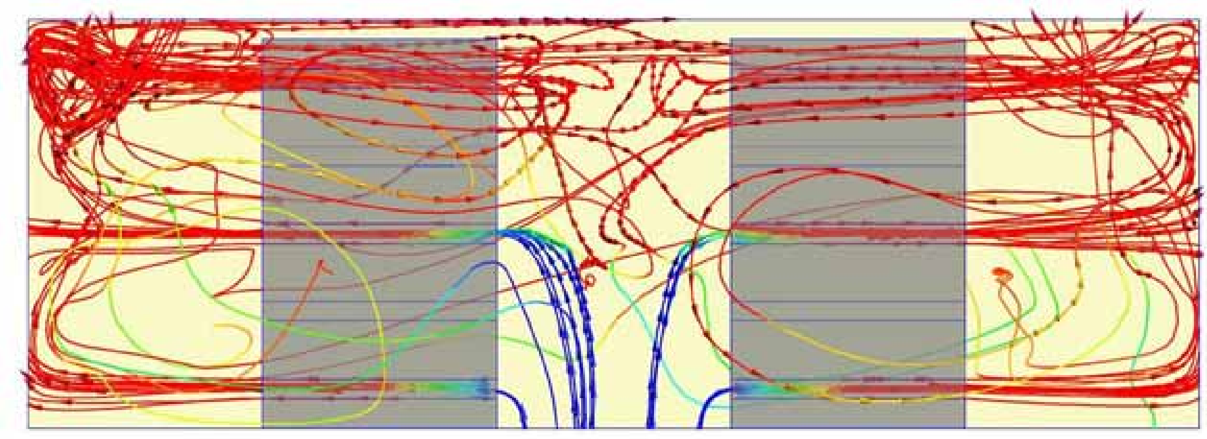

Figure 7 The SK4FFC25 model. Flowlines showing temperature

The SK4FFC25 model has an air change ratio between cooling system and through server of: 1:1 In this case there isn't cold air upper half of the servers. The hot exhaust air is traveling freely between the 2 exhaust sides. There is outspoken scattering and mixing of the hot air and most servers receives premixed thus warmer air.MI:80, SC 16.8°C, Energy Class E.

Open rooms with a narrow space above the racks needs a supply of cooling air at least 2 - 3 times the air flow through the servers giving a holding pressure large enough to keep the exhaust air from mixing and getting to the server intake side. For this air ratio to work it is necessary to have physical barriers between the intake and the exhaust side of the servers.

DISCUSSION

From the results of the CFD-calculations it is now possible to setup some basic guidelines for selecting the optimal server room design.

It is always better to build up a server room with an installation floor and concealed ceiling and this way guide your cooling air inlet and outlet to the optimal position as close to the server than building a server room with fan coils for cooling. If it's impossible to separate the intake and exhaust side physically it should be attempted to have as little room above the rack as possible and reduce the number of gaps in the rack. Demand for cooling air increases with increased opening areas between the two sides.

If you are forced to use fan coils, it is better to have some open space above the racks. When having a fan coil placed near the ceiling it can be at either side or end wall. The servers just have to be placed front to front for the most efficient cooling. MI 16 – 17 and sub cooling 3.2 – 3.4 °C. All other server configurations give MI 21.5-40.0 and sub cooling 4.3 – 8.0 ºC.

Table 3 Comparing model results

	MI-range	SC-range
SK2	16-40	3.2-8.0
SK3	1.3-39	0.25-7.8
SK4	*	*

* SK4 models seem to be better than SK3- models. This has only been tested for a few configurations.

Comparing model SK3 and SK4 can also be used to optimize the cooling air flow rate. It is possible to control and guide the exhaust air from server to room outlet in an SK4 setup with as little as half the flow rate of a SK3 saving half the power consumption for ventilating the cooling air.

With a demand of a specific maximum temperature at the server inlet and a estimated savings potential of 65- 130 EURO/(year*kW*K). As the difference between the best and the worst model design is 8 °C, and the server room is emitting 30 kW all year round, the savings potential for this room is 15,600 - 31,200 Euro pr. year. Further savings can be found if the system can operate at a higher temperature. This gives exactly the same savings pr. °C. If the system can be designed to use fresh air for free cooling during periods with outdoor temperature below the inlet set point even more electricity for cooling can be saved.

About the model it has been created in a well verified General Purpose CFD code from one of the leading vendors. The models have all been checked for grid independency. Though full convergence was not reached, 100 iterations brought the calculations very close to the convergence criteria. Measurements of server inlet temperatures were taken 1cm from inlet of tube inside the tube. Thus best mixing index 0.25 equals ideal 0.00. Comparative testing is a future challenge, but will probably not give any substantial deviation from the CFD-results.

No free cooling considered since it was the mere room design that was the scope of this work.

ACKNOWLEDGEMENT

The Danish Electricity Savings Trust

Study on A New Ventilation System Using Ground Heat Collector of Vertical Double Pipe (Part 1 Outline of A New Ventilation System And Its Performance)

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SUMMARY

Geo Power System [1] [2] uses the geo thermal heat from the very shallow area of the underground temperature down to 5 m depth. Though the conventional Geothermal Heat Pump system uses the deeper soil like 30 to 100 m from the ground, because of the slow heat transfer in the soil, the temperature of 3 to 5 m depth is warmer in winter and cooler in summer than the average temperature of the site. This system is more efficient and lower cost. Now this system was introduced in Japan for more than 600 buildings including 27 large size facilities. This paper explains the system and the typical performance data of the house, kindergarten and the office.

The test in Canada was done in Ottawa with Natural Resources Canada for the application to the different climate from Japan.

INTRODUCTION

The temperature profile of the underground

The annual temperature profile of the underground is shown in Fig.1 and one day profile in summer is in Fig.2.

The temperature below 6m is constant as the average ambient temperature of the site and at 3 to 4m below the ground level is lower in summer and higher in winter than the average temperature because of the slow heat transfer of solar energy from the sun.

In western part of Japan, we have investigated and worked with Yamaguchi University and applied this character in this system for efficient and cheaper heat exchanger of house ventilation air.

Fig.2 one day temperature profile in summer
30 °C air in summer can heat exchange with shallow soil of 15 °C and installation cost can be cheaper because of the digging cost can be reduced by auger drilling down to 5 m is easy comparing with the cost down to 30 m or 100 m depth from the ground.

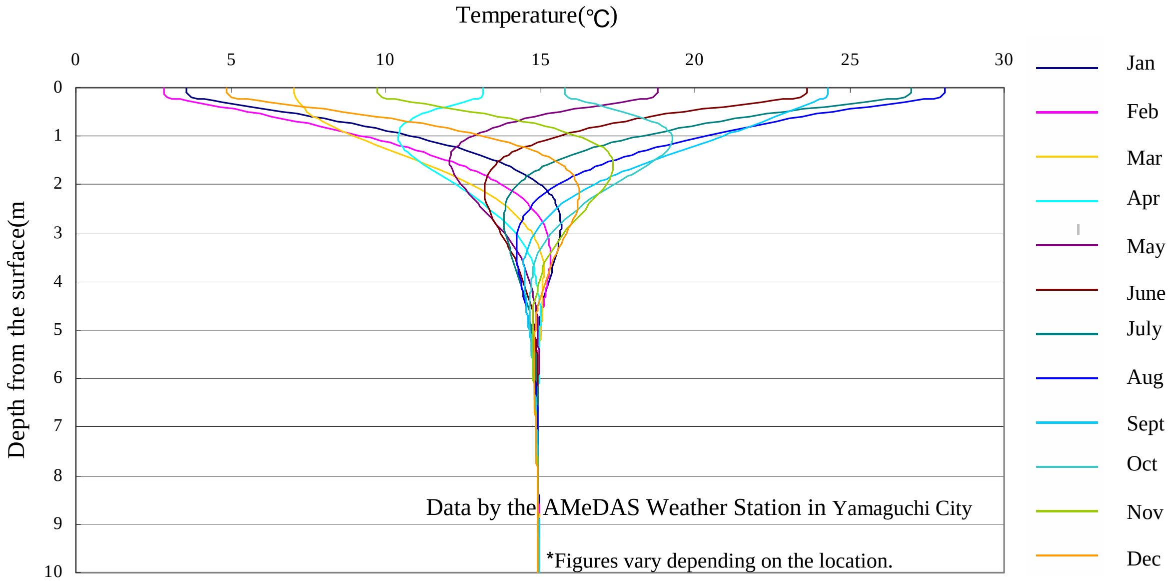

Fig.1 Underground Temperature Profile (all year).

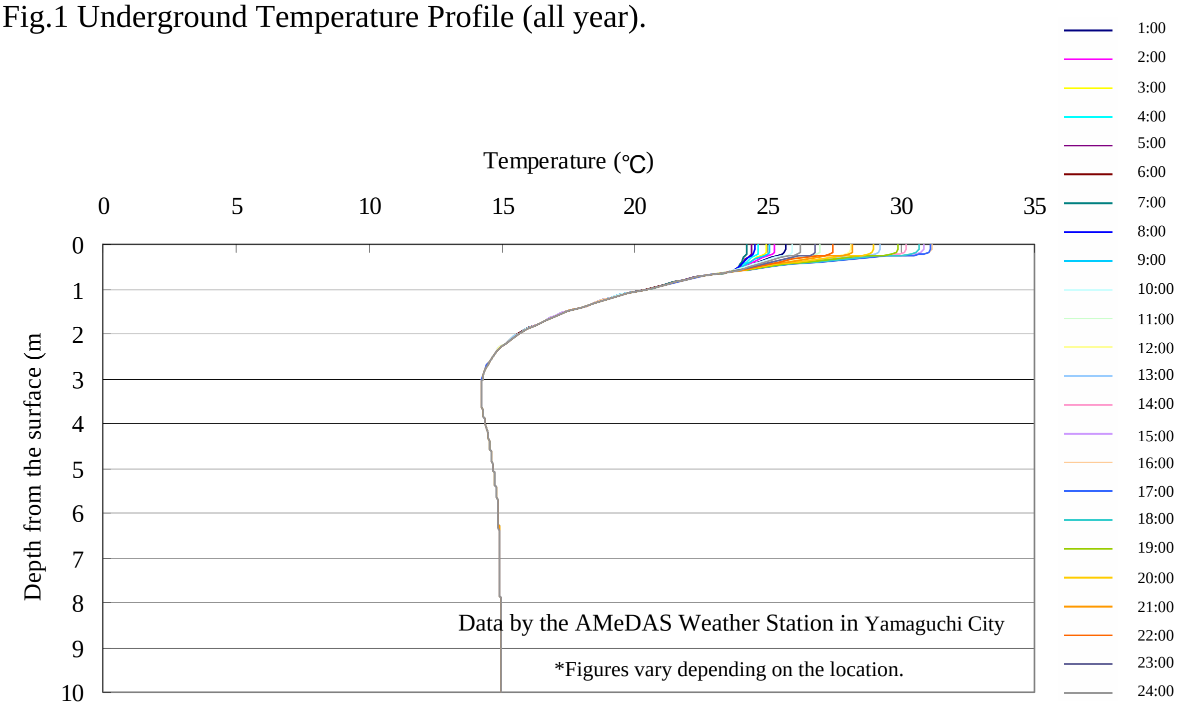

Fig.2 Underground temperature Profile(one day in summer).

METHODS

Description of the system

The system is shown in Fig.3 System Flow.

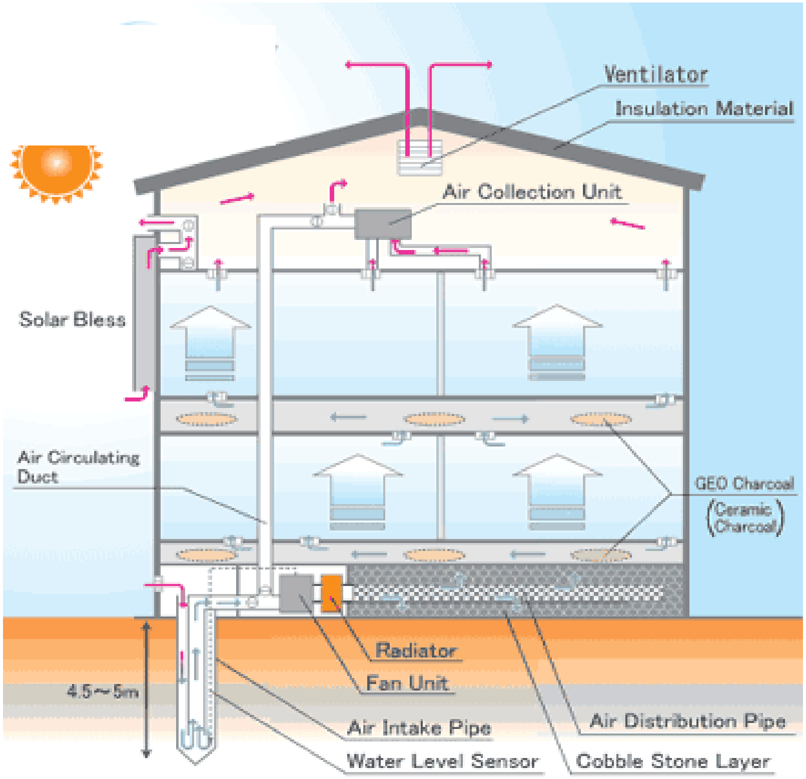

Fig.3 System Flow.

The system is a three in one system combining the effectiveness of three factors namely, Geothermal Energy + Air Circulation + Ventilation. The ground heat collecting double pipe is vertically installed down to 5m at the perimeter of the building and the cobble stone layer is arranged under the first floor board for heat sink. The outer pipe is good for heat transfer and the inner pipe is for insulation by plastic material. When the well water is available at site, radiator (fin-tube heat exchanger) is installed for further effective use of thermal heat.

For typical domestic houses in Japan, 250 mm diameter and 5 m length of aluminum pipe is used for outer pipe and the 150 mm polyethylene for inner pipe. The room air can be circulated or vent depending on the temperature and the circumstance.

Demonstration Tests in Yamaguchi, Japan

The temperature profile of the show room in Yamaguchi city, Yamaguchi Pref. Japan was taken by Yamaguchi University in summer 2003 and the efficiency was confirmed even 32°C air can be cooled to 22°C by this system. The graph is shown in Fig.4 Temperature profile at demonstration house in Yamaguchi Japan.

Though this is not so cool as the air conditioner, it is good for health reducing energy consumption and avoid over cooling.

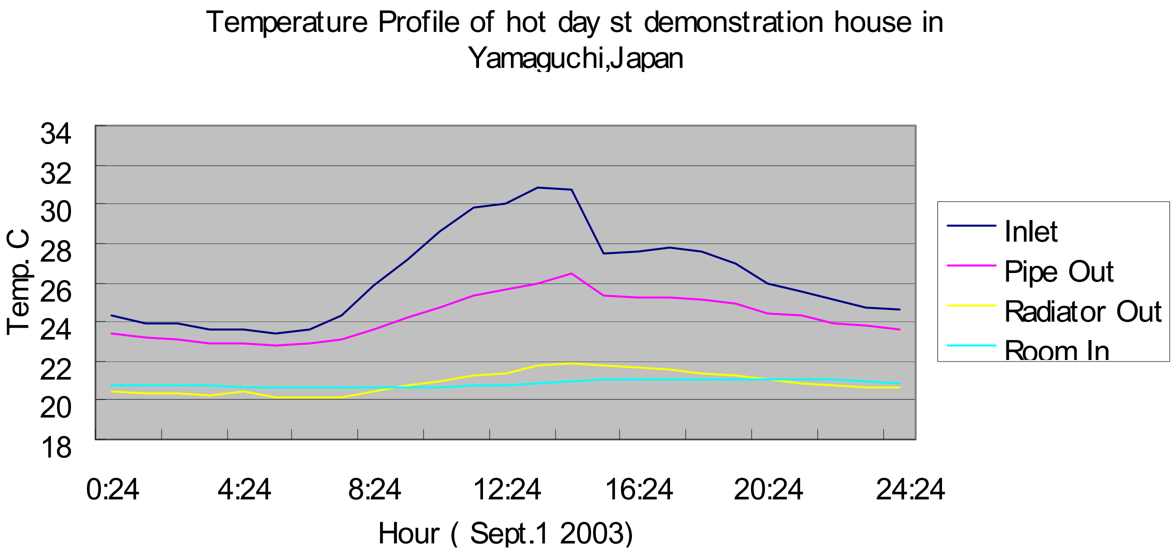

Fig.4 Temperature profile at Demonstration house in Yamaguchi, Japan.

RESULTS

Tests in Ottawa, Canada

In order to have an overseas performance of very hot summer and very cold winter, the agreement was made to comprehensive tests in Ottawa Canada with Natural Resources Canada (NRCan) for one year. The effect to the ground soil was also tested.

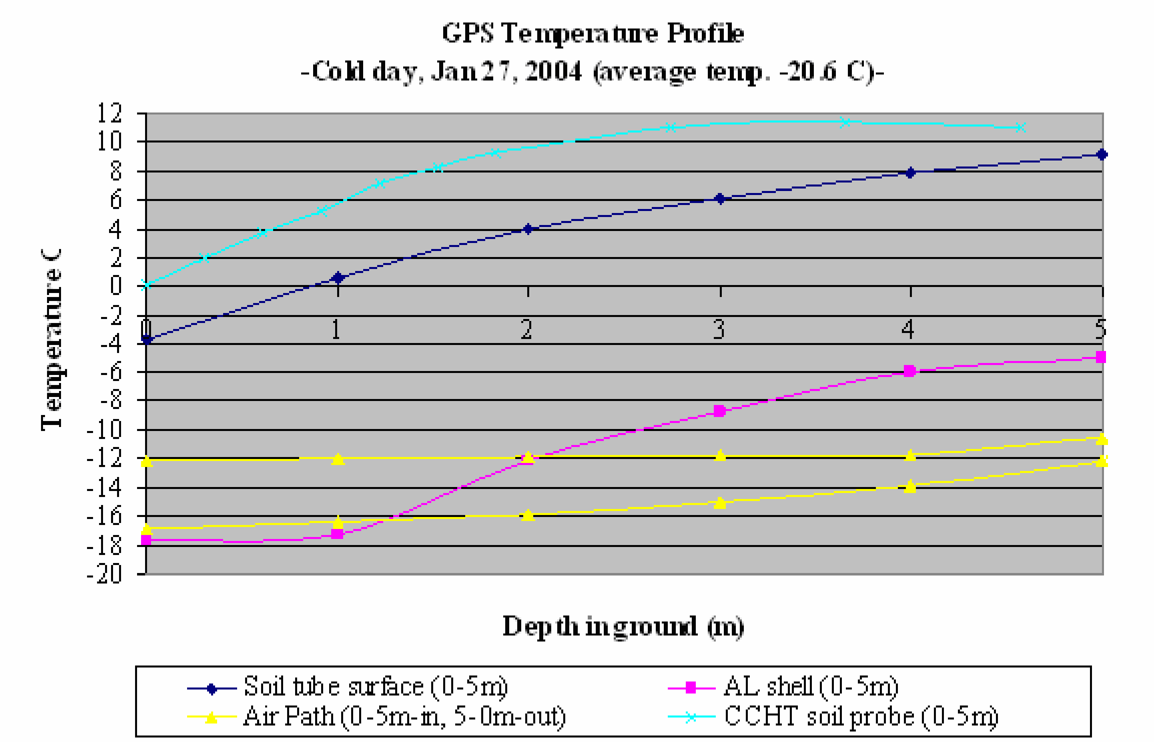

Fig.5 Cold winter day in Ottawa.

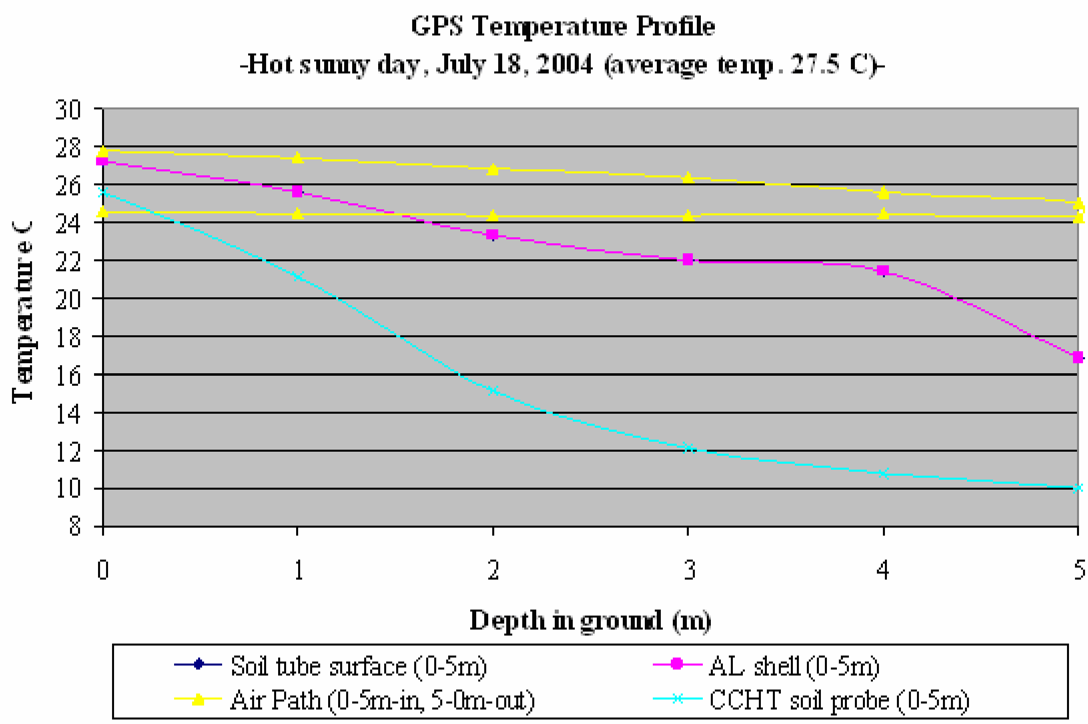

Fig.6 Hot sunny day in Ottawa.

The performance at a kindergarten in Shiga, Japan

With a subsidy of the Ministry of Environment, the kindergarten in Shiga, Japan installed 44 pipes and 11 radiators for their rooms of 2637 m2 by July 2005. Fig.7 and 8 show the temperature profile of one day in winter /summer. They enjoyed the energy saving consequently reduction of CO_2 emission and running cost saving as shown in Fib.9 ,10 and 11.

In a cold in winter of 0 to 1°C of outdoor temperature, Room-In Temperature of air is almost constant and Room Temperature varied in a day from 12 to 15C.

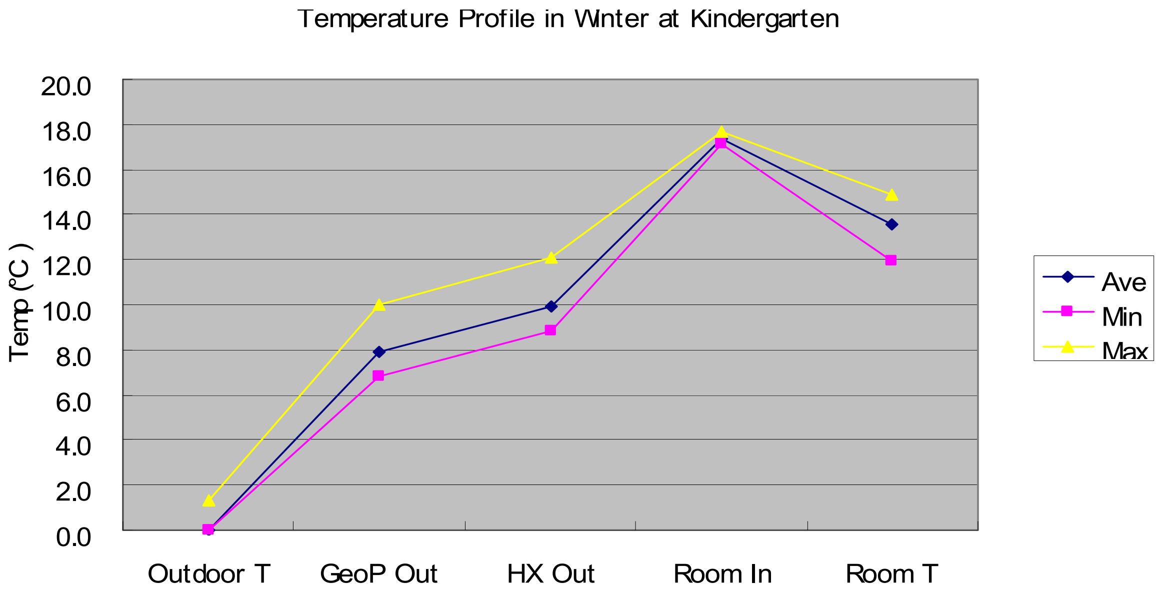

Fig.7 Temperature profile in winter (Feb.4 2006) at a kindergarten in Shiga, Japan.

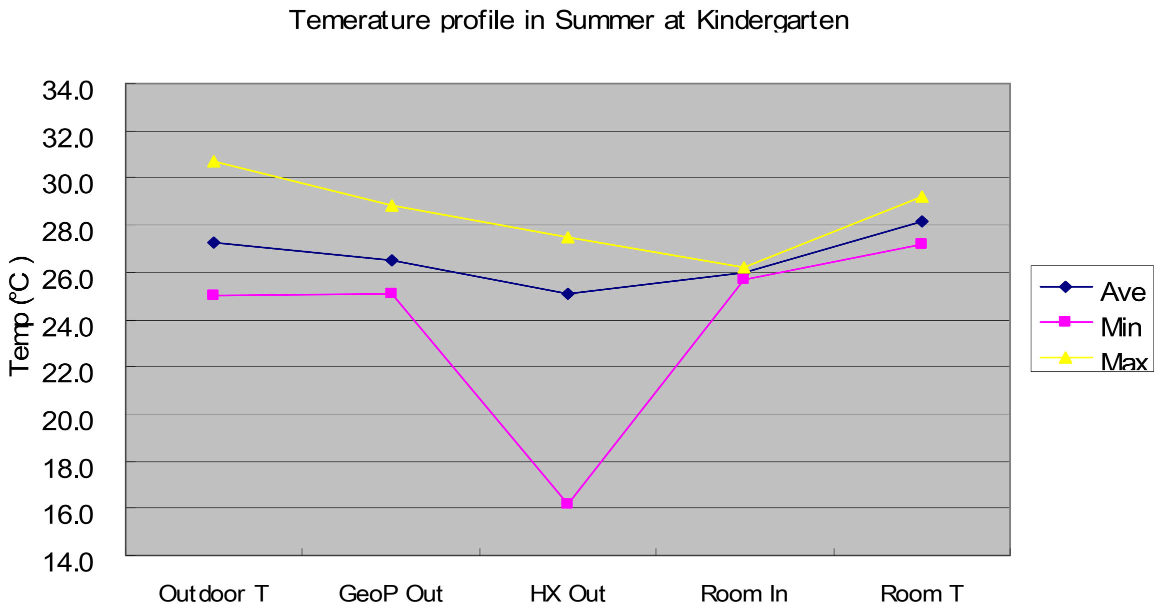

Fig.8 Temperature profile in summer (Aug.21 2006) at a kindergarten in Shiga, Japan.

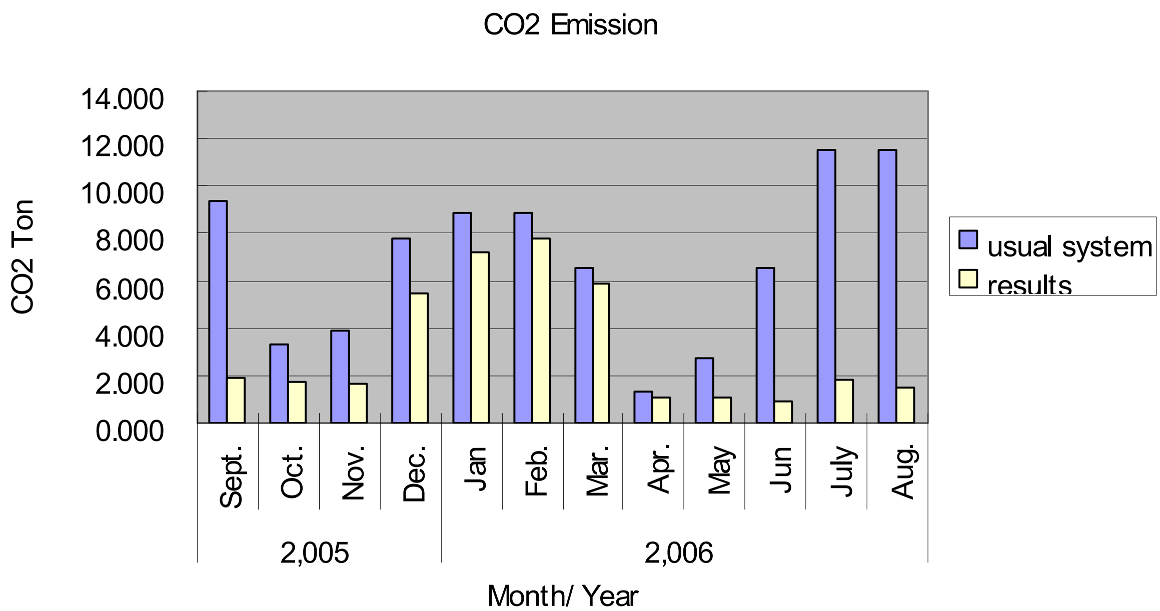

Fig.9 Comparison of CO_2 emission at a kindergarten in Shiga, Japan.

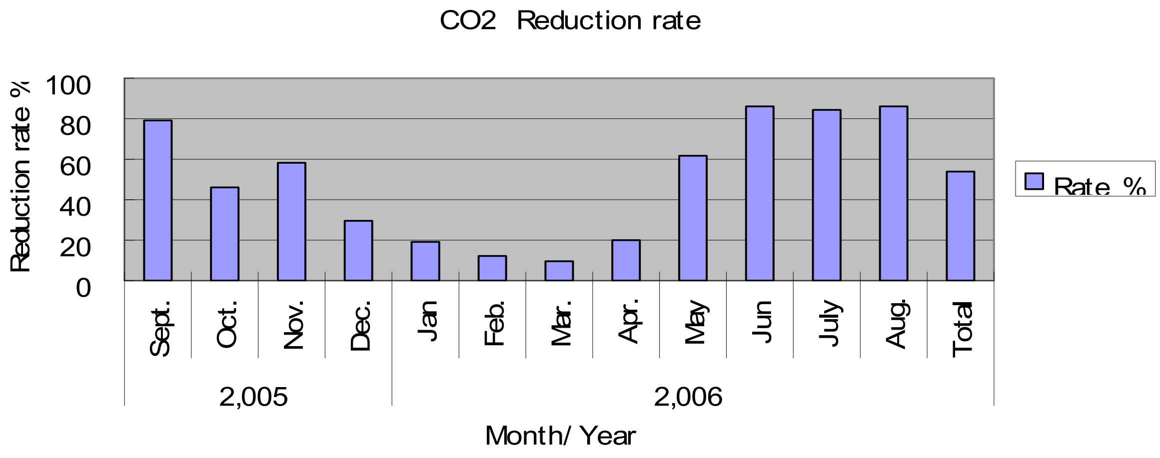

Fig.10 CO_2 reduction rate at a kindergarten in Shiga, Japan.

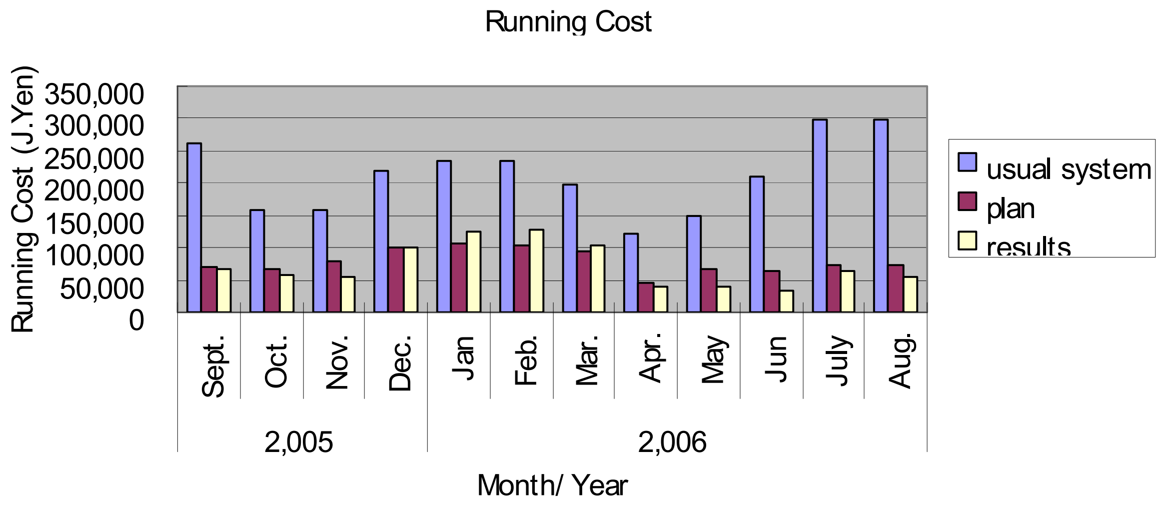

Fig.11 Running cost at a kindergarten in Shiga, Japan.

DISCUSSION

Comfortable life is not synonymous with healthy life. Symbiotic society of healthy environmentally friendly society helps to create genial personalities.

The wastes of electricity due to the excess use of air-conditioners and overuse of fossil fuels have accelerated of CO_2 in the atmosphere, worsening global warming.

More and more children have autonomic nerve dysfunctions, causing them to be unable to sweat, because of the excess use of air-conditioners. Deterioration of autonomic nerve functions can impair the natural functioning of human body temperature regulation and cause other function disorders.

The kindergarten which installed this system enjoyed the absent rate decrease of the gardeners.

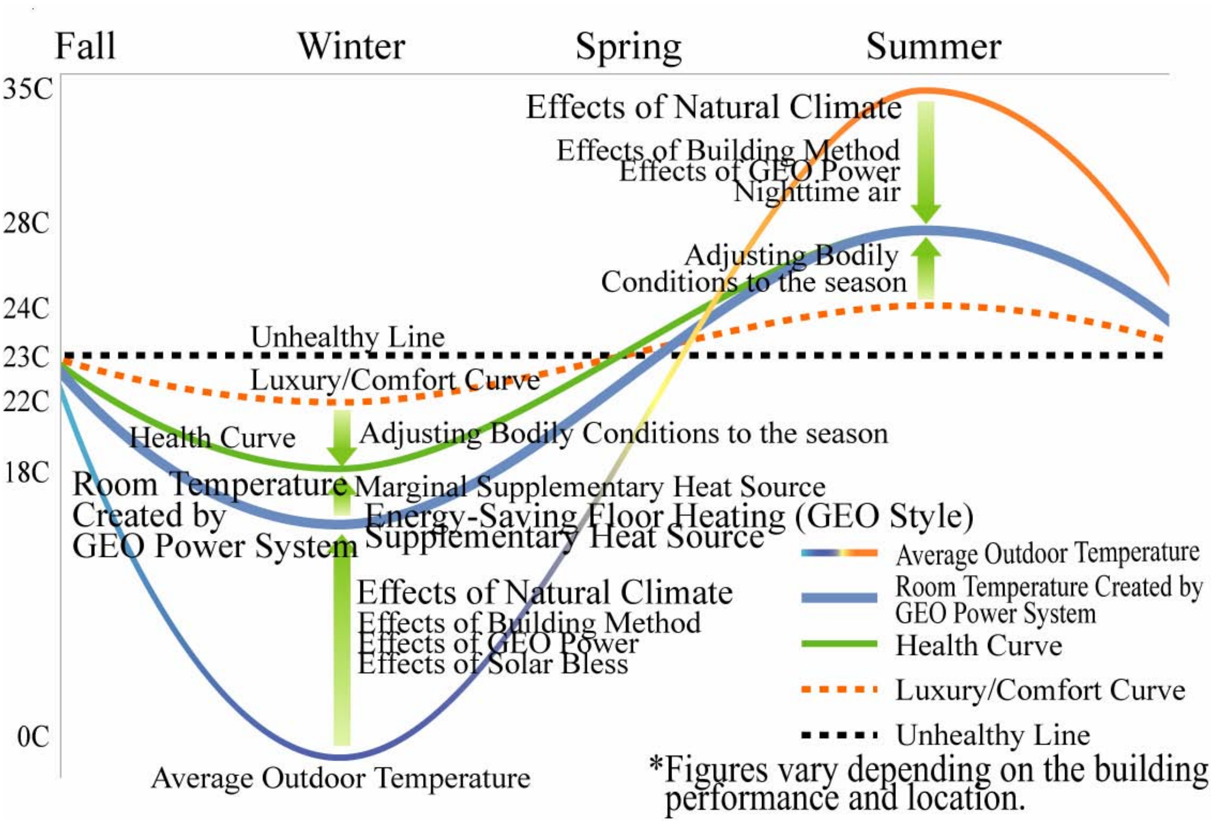

Fig.12 Health Curve shows the temperature profile for a year.

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Study on a New Ventilation System Using Ground Heat Collector of Vertical Double Pipe (Part 2 : Prediction of Heat Collector Performance and Methods to Improve Performance)

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SUMMARY

A ventilation and air conditioning system using the ground heat was developed as one of techniques using renewable energy. In the system, vertical double pipe type of heat collector is used. We developed a code, which is able to solve unsteady three dimensional heat conduction equation, heat balance equation at the ground surface and heat transfer equation from the heat collector to the ground simultaneously, to predict the performance of the heat collector. Main results are as follows: the outlet air temperature of the heat collector becomes higher /lower at most 7^{o}C than the inlet air; the performance of the heat collector increases with the length of the heat collector and increases with the heat conductivity of the soil; continuous operation of the system over a year contributes to improvement of the performance of the heat collector both in winter and in summer.

INTRODUCTION

Every country including Japan needs to reduce CO_2 discharge. To achieve it, many techniques using renewable energy are being developed now. As one of such techniques, have been developed ventilation and air conditioning systems using the ground heat. Sekine et al. [1,2] carried out feasibility studies of ground source heat pump system and they tried to reduce initial cost by using cast-in-place concrete pile foundations of building as heat exchangers. Hamada et al. [3] proposed a heat pump system with improved underground heat exchanger which is set up by no-dig method. They showed that the COP of the system was higher than ordinary air source heat pump. Tsukidate et al. [4] investigated the heat collection performance of single U tube of 75 meters and Martin et al. [5] discussed maintenance cost of geothermal heat pump. Hashimoto [6] developed a new ventilation system using geothermal heat. The heat collector of Hashimoto's system is made of double pipe with inner polyvinyl pipe of 0.15 meter diameter, the outer aluminum pipe of 0.25 meter diameter and 5 meter length. It is buried vertically up to five meter depth in the ground. The purpose of our study is to develop a code predicting the performance of the heat collector of the ventilation and air conditioning system developed by Hashimoto and to examine the effect of several factors on the heat collector performance.

To predict the performance of the heat collector, we need to know the outlet air temperature of the heat collector, so we develop a model to predict the outlet air temperature. The model should include heat balance equation at the ground surface, unsteady three dimensional heat conduction equation for the ground and equation for the temperature change of the air flowing in the heat collector. The three equations must be solved simultaneously. We also discuss the effect of several factors, such as the outer diameter of double pipe, the length of the heat collector, the heat conductivity of the soil etc., on the heat collector performance.

VENTILATION AND AIR CONDITIONING SYSTEM USING GROUND HEAT

The ventilation and air conditioning system using ground heat is described in detail in Part 1, so we mention the system briefly. The outside air is taken into the heat collector and is supplied into the room through heat storage room filled with crushed stones as shown in Figure 1 [7]. The outside air flows into the heat collector from the upper annular space and flows into the inner pipe through many holes on the wall of the inner pipe near the bottom and flows out of its upper end. The outside air is heated in winter and cooled in summer by the ground while flowing down in the annular space of the heat collector. Heated air or cooled air contributes to reducing the energy for heating and air conditioning.

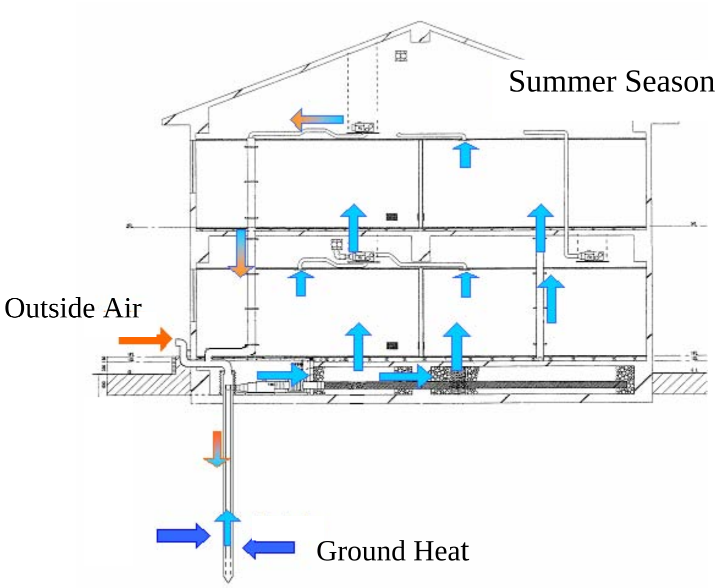


Figure 1. Ventilation, heating and air conditioning system using ground heat.

PREDICTION MODEL OF OUTLET AIR TEMPERATURE OF THE HEAT COLLECTOR

Unsteady heat conduction equation

Unsteady three dimensional heat conduction equation is expressed by equation (1),

$$\frac{\partial T}{\partial t} + a\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right) - S = 0\,, \quad (1)$$

where T is the soil temperature, t is time, a is thermal diffusibility of the soil, S is the source term and x, y and z are the coordinates.

Heat balance equation at the ground surface

Influence factors on the heat balance at the ground surface are solar radiation R_S, effective radiation R_N, convection heat transfer H, latent heat transfer E and heat transfer from the ground surface to soil Q_G. The heat balance equation is expressed by equation (2) using these factors (see Figure 2),

$$Q_G = R_S - R_N - H - E \,. \quad (2)$$

When calculating Q_G from equation (2), we used measured values of R_S and R_N and predicted values of H and E which are calculated using atmospheric stability scale of Monin-Obukhov.

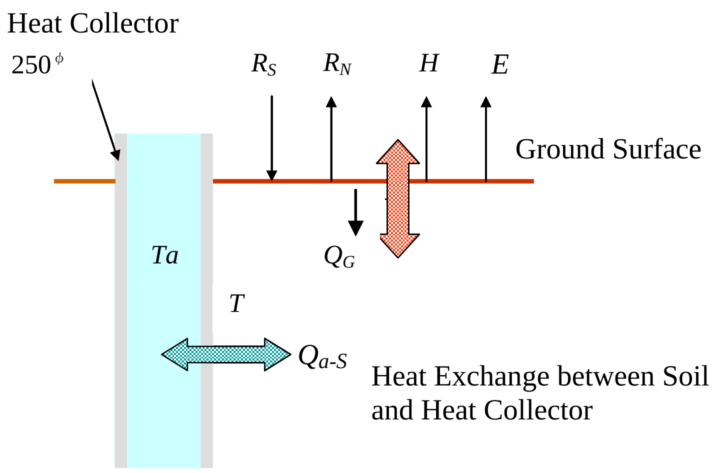


Figure 2. Heat balance at the ground surface and heat exchange between soil and heat collector.

Equation of temperature change of the air flowing in the heat collector

An equation for the temperature change of the air in the heat collector is expressed by equation (3)

$$\rho_a C_{pa} V_a \frac{\partial T_a}{\partial z} = -Q_{a-S} \,, \qquad (3)$$

where ρ_a is the density, C_{pa} is the specific heat, V_a is the flow rate, T_a is the air temperature, $Q_{a\text{-}S}$ is the heat transfer from the air in the heat collector to the ground and subscript a means the air.

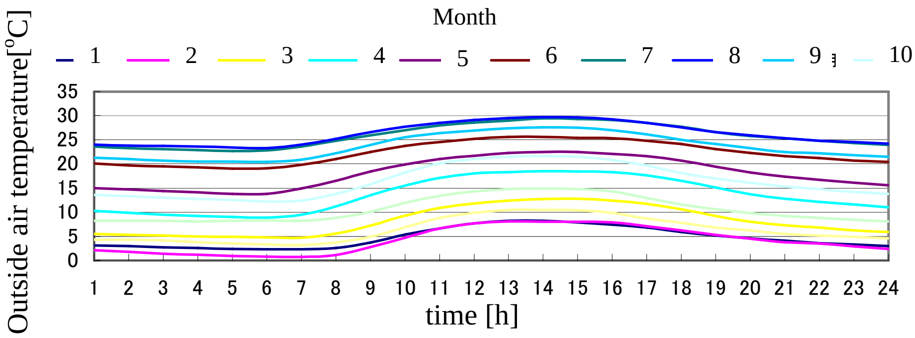
Computational method

The temperature in the ground, the downward heat transfer from the ground surface and the air temperature in the heat collector depend on each other, so the three equations (1), (2) and (3) must be solved simultaneously. Here, we calculate the temperature of the ground T, the downward heat transfer from the ground surface Q_G and the air temperature T_a by iterative computation method.

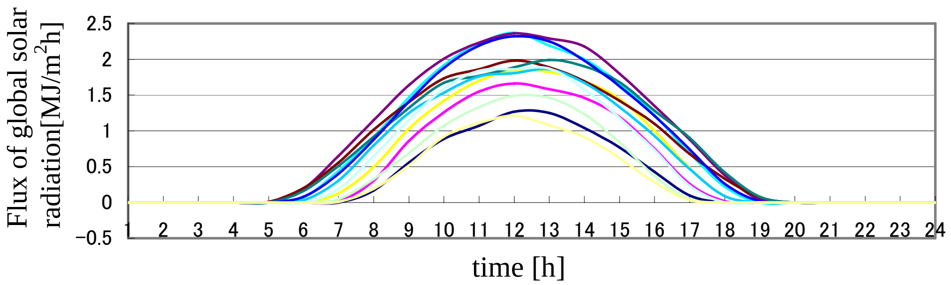
We applied free slip condition to the boundary condition of the temperature at the ground surface and the heat flow from the surface to the ground was taken into consideration as heat source of the control volume just under the ground surface. The same method was applied to the boundary between the outer surface of heat collector and the soil.

Computational conditions

The weather conditions are necessary for the above calculation. Here, we used measured values of the outside air temperature, solar radiation, effective radiation, absolute humidity and the outside air velocity. As an example, we show standard data of the outside air temperature and the solar radiation in Figure 3.



(a) The outside air temperatures.



(b) Solar radiation.

Figure 3. Weather conditions.

Heat transfer coefficient on the inner wall of the outer pipe is calculated using an empirical equation applied to well developed turbulent flow in pipe and Reynolds number in the annular space was calculated with hydraulic mean radius. We assumed that the air in the inner pipe is thermally insulated from the air in the annular space.

RESULTS

Annual change of ground temperature

Annual change of ground temperature from January to December is shown in Figure 4. The result is when the heat collector does not exist. The ground temperature near the ground surface changes widely with month but the temperature in the depth more than five meters hardly changes and is kept at nearly constant temperature. It is well known that the temperature of the deep ground is nearly equal to the local average air temperature, that is, the temperature of the deep ground is higher than the outside air temperature in winter and lower in summer. This is the reason why energy saving is possible by using ground heat to ventilation, heating and air conditioning system.

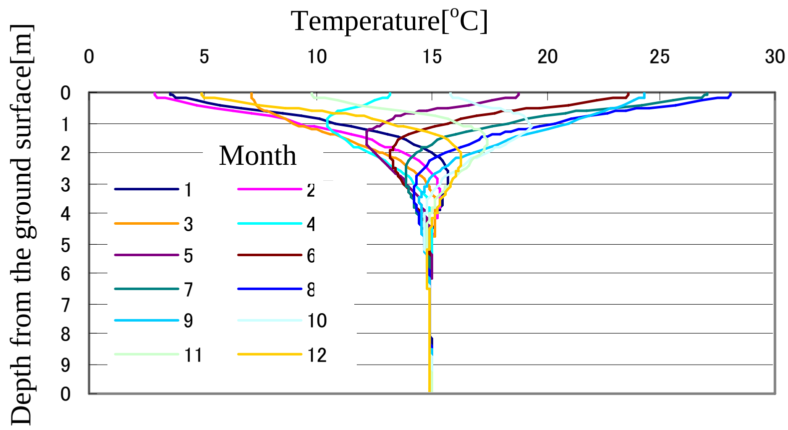


Figure 4. Annual temperature change in the ground.

Outlet air temperature of the heat collector

Calculated outlet air temperature was compared with measured temperature to examine the accuracy of the prediction model. The calculated temperature agreed well with the measured temperature as shown in Figure 5.

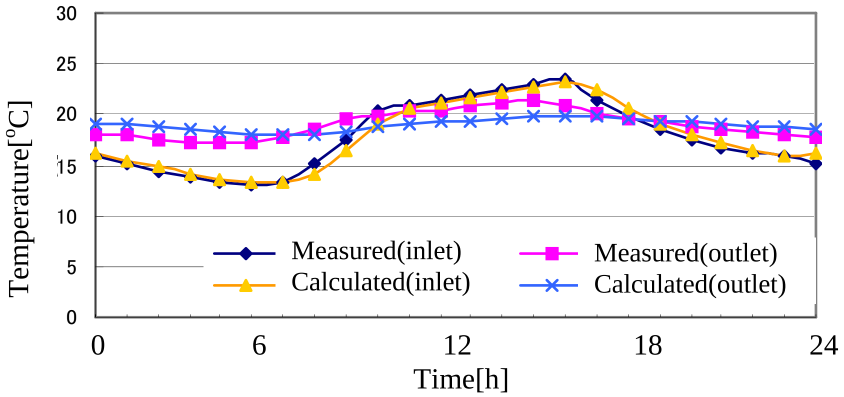


Figure 5. The inlet air temperature and the outlet air temperature of the heat collector.

Figure 6 shows calculated annual change of the outlet air temperature of the heat collector. The outlet air temperature changes with the outside air temperature and becomes at most about 7 ^{o}C higher than the inlet in winter and at most about 7 ^{o}C lower in summer. Here, the inlet air temperature is assumed to be equal to the outside air temperature.

The temperature of the soil in the surroundings of the heat collector is affected by the heat flow from the heat collector to the soil and it is expected that the soil temperature increases gradually with time in summer and decreases in winter. Figure 7 shows the change of the temperature distribution in the surroundings of the heat collector from July 10th to August 30th. We can see that the region of high temperature is expanded in August 10th. And the outlet air temperature of the heat collector increases gradually with time as shown in Figure 8 because of the raise of soil temperature.

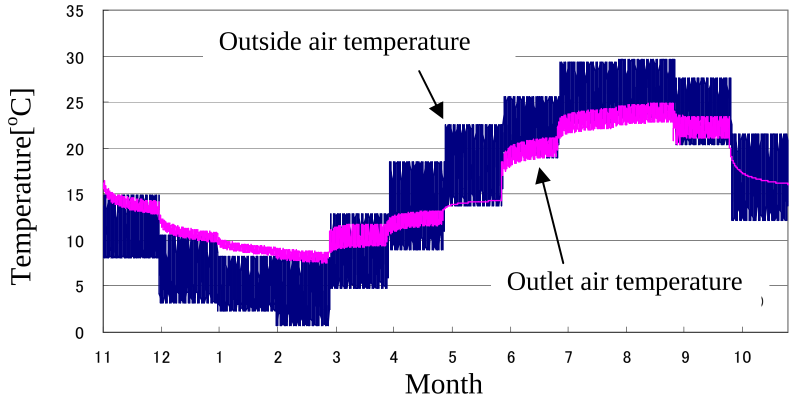
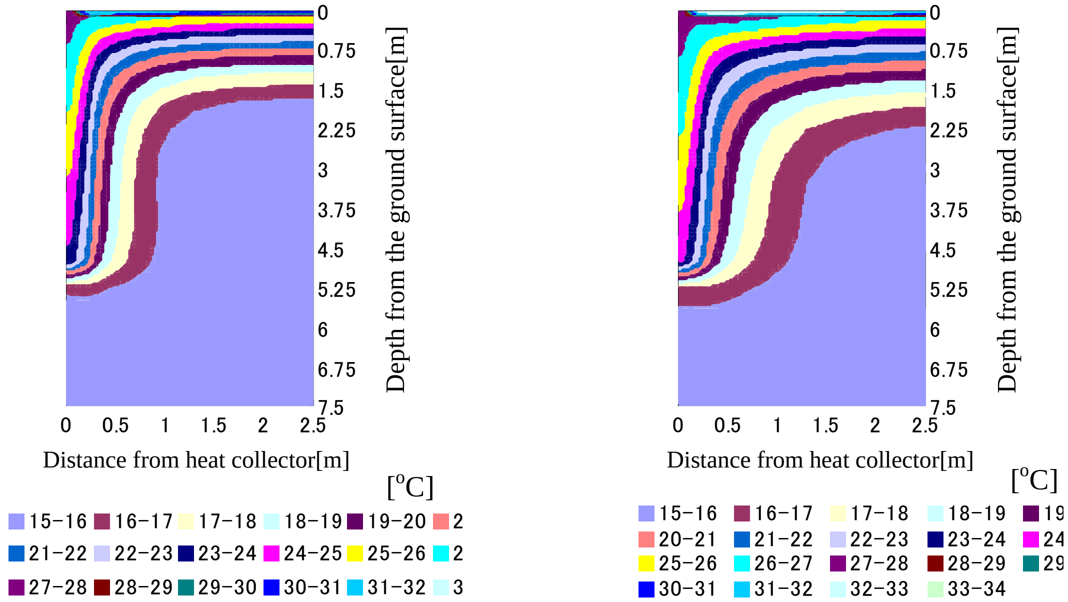


Figure 6. The outside air temperature and the outlet air temperature of the heat collector.



July 10th August 30th

Figure 7. The soil temperature in the surroundings of the heat collector.

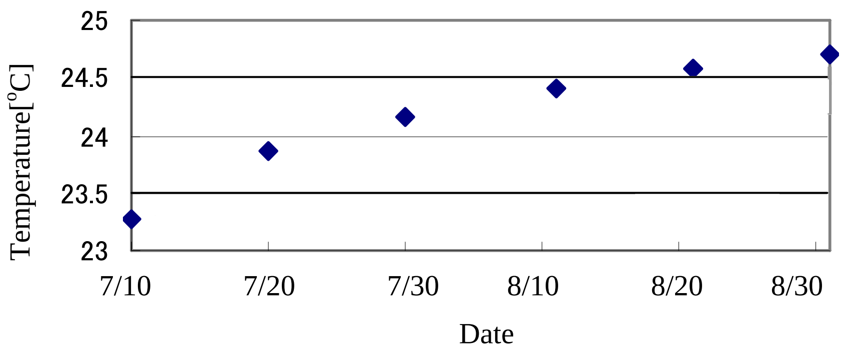


Figure 8. The change of the outlet air temperature of the heat collector.

Effect of pipe size on the performance of heat collector

It is expected that the performance of the heat collector increases with the diameter of the outer pipe and the length of the heat collector because heating/cooling surface area increases. The computation results showed the same inclination as expectation, that is, the quantity of collected heat increases with the outer pipe diameter and increases with the length of the heat collector as shown in Figures 9 and Figure 10. The heat collection performance of the heat collector of 400mm outer diameter increases by about 3% in winter and about 9% in summer comparing with the standard heat collector of 250mm outer diameter. The heat collection performance of the heat collector of 10 meter length increases by about 61% in winter and about 86% in summer comparing with the heat collector of 5 meter length.

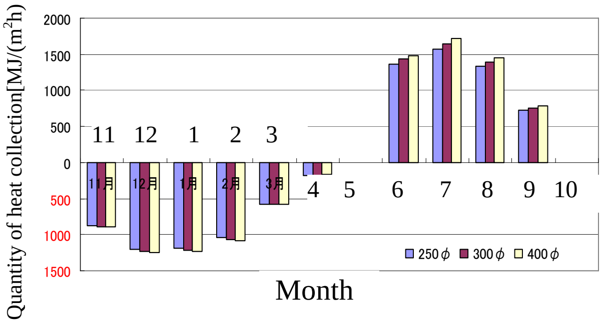


Figure 9. The effect of the pipe diameter.

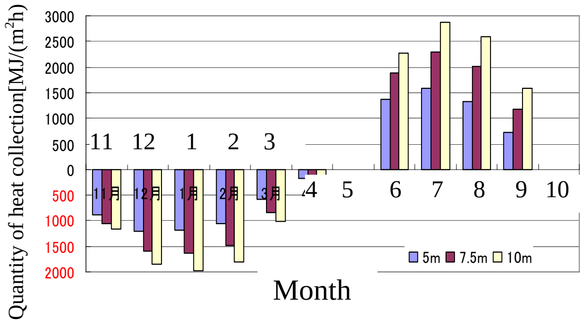


Figure 10. The effect of the pipe length.

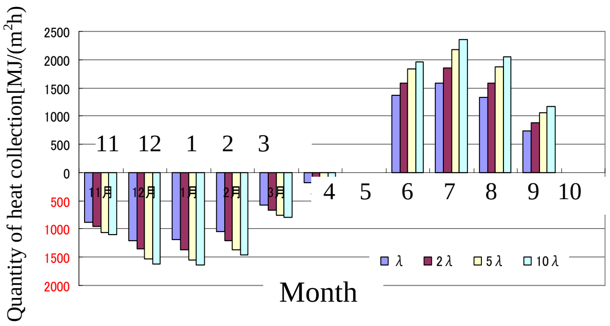


Figure 11. The effect of the heat conductivity.

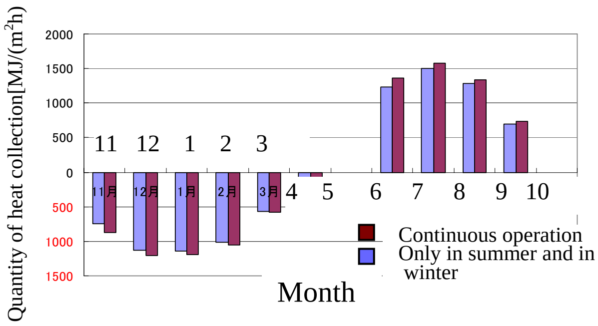


Figure 10. The effect of operation condition.

Effect of heat conductivity of soil

Here, we consider the case when the soil in the peripheral region of the heat collector, from 2 meter depth to 6 meter depth and within 0.5 meter distance from the center of the heat collector, is replaced by the material with the heat conductivity higher than the soil. The result is shown in Figure 12. The heat collection performance increases by about 36% in the case of 10 λ in winter and increases by about 51% in summer comparing with in the case of λ.

Effect of continuous operation of the system

The soil in the surroundings of the heat collector is heated in summer and is cooled in winter by operation of the system, so it is worth to examine the influence of continuous operation of the system on the heat collection performance. Here, we consider the two cases. One is when the heat collector is continuously operated all over a year and the other is when the operation of the system is limited to only in summer and in winter. As you can see from Figure 13, the quantity of collected heat is larger both in winter and in summer in the case of continuous operation over a year than in the case of the operation limited only in summer and in winter.

CONCLUSIONS

To predict the outlet air temperature of the double pipe type of heat collector buried vertically in the ground, we developed a code which can solve unsteady three dimensional heat conduction equation, heat balance equation at the ground surface and heat transfer equation from the heat collector to the ground simultaneously. We examined the heat collector performance and the following results are obtained: (1) the outlet air temperature becomes higher /cooler at most 7^{o}C than the inlet air; (2)the performance of the heat collector increases with the length of the heat collector; (3) the heat collector performance increases with the heat conductivity of the surroundings of the heat collector; (4) Continuous operation of the system over a year contributes to increase of the heat collector performance both in winter and in summer.

ACKNOWLEDGEMENT

The authors are grateful to Mr. Naoki Yamashita who is a general manager of oversea division of Geo Power System and has given us many appropriate suggestions.

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Cooling tower cooling of large HVAC-systems: case of the Flemish Institute for Biotechnology Building at the Ghent University

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SUMMARY

In 2003 a new advanced biotechnology laboratory building was opened at the Ghent University. Very high standards were demanded for the HVAC-installation.

The refrigeration system consists of two large electrical chillers cooled with two cooling towers and the possibility to use free cooling with a plate heat exchanger and the cooling towers if outside air temperatures allow it.

An energy survey was made of the whole building from 2003 to 2006, with special attention being paid to the refrigeration equipment. Measurements showed a strong rise in energy use of the campus, after start-up of the building. Several measures were introduced to reduce energy use and related heat dissipation.

The performance of the chillers was monitored. Measurements show that performance of the free cooling is disappointingly low. This is caused by bad design and suggestions are made for amelioration

Secondly, it was noted that the cooling towers used a lot of water. This was due to the way the towers were operated to give minimal energy use in the HVAC-system and ensure the performance all time of the year. An analysis of the water consumption is made.

INTRODUCTION

In November 2003 the Fiers-Schell-Van Montagu research building was inaugurated at the Ghent University campus Ardoyen. The building was named after the pioneers in biotechnical research at the Ghent University. The building houses the research departments of Molecular Biomedics and Plant System Biology, both being part of the Flemish Interuniversitary Institute for Biotechnological Research (VIB). Research in the building is done on cancer, virology, cell analysis, … The research groups use very advanced techniques and laboratory appliances. Very high requirements are set for the indoor air quality and conditions, to provide optimal growing conditions for the cells and plants studied.

The building consists of two laboratory wings (east and west wing) of rectangular shape, 36 m wide and 3 floors high. In front of the laboratory wings a main building is situated as shown on Figure 1. Each laboratory wing is divided in three sections. The central section contains the laboratory equipment with high power density (up to 80 W/m²), such as ultra freezer and laminar flow tables. The central part is surrounded by corridors. The left and right hand side of the wings consist of laboratory modules for about 10 people each, where manual laboratory operations are performed and small offices are available.

Figure 1 - Lay-out of the VIB-Building

There are 300 people working in the building, with a total ground surface of 13000 m², of which 8000 m² (60%) is used for laboratory facilities. Figure 2 shows the measured electricity use of the building during 2004. The average value is about 500000 kWh/month, resulting in a power density of about 53 W/m².

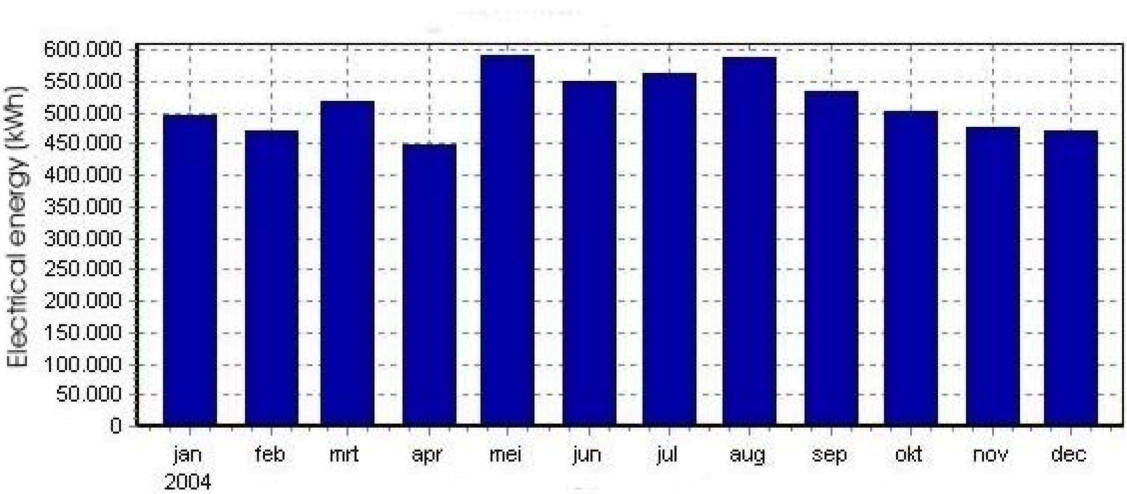


Figure 2 - Energy use of the VIB-Building (2004)

The building HVAC system consists of several pulsion/extraction groups for each wing and for the main building. Temperature and relative humidity control is set to 18°C/RH>30% in summer and 21.5°C/RH>30% in winter. Heating is provided by natural gas fired condensing boiler units and humidity is controlled by vapour injection, being produced by a small gas fired steam boiler. Cooling is done with an ice-water circuit, providing ice water at 7 °C.

Local units are used to provide high indoor air quality requirements for certain laboratory modules. They use fresh air coming from the central system, mix it with room air and condition it within very narrow margins.

MAIN COOLING SYSTEM – FREE COOLING OPTION

Figure 3 shows the layout of the main cooling system. The air conditioning units in the building use a 7/12 regime, delivering ice water at maximal value 12°C to the cooling plant. The cooling plant has two screw compressor cooling machines of 1139 kW and 600 kW cooling power. The cooling machines have a priority setting for the smaller machine. The bigger cooling machine only operates at high cooling load conditions. Both machines are coupled to cooling towers with 1425 kW and 710 kW thermal power respectively. Water flow rate to the towers is controlled by a three way valve. All pumps are constant flow rate pumps. The cooling tower fans are frequency controlled. Readings of the frequency controller set-point in the building management system can be translated into the electrical power use of the fans during operation. To this purpose controller settings and power use were logged over a certain period.

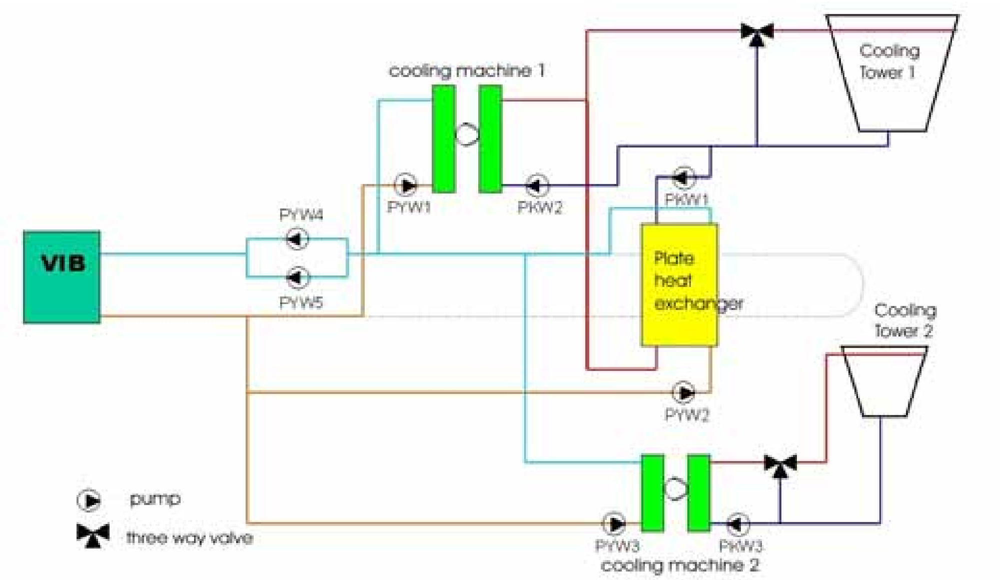


Figure 3 - Main Cooling system lay-out

Next to the cooling machines a plate heat exchanger was installed in order to be able to use the bigger cooling tower directly in free cooling mode during winter. The ice water is directed to one side of the heat exchanger while an extra pump circulates the cooling tower water over the other side. A part of the cooling water flow rate can be circulated over a by-pass in this case (dotted line). Free cooling is started by the building management system when the outside wet bulb temperature drops below 6°C.

In this paper the performance of the cooling system is studied both working in normal and free cooling mode. To this end data supplied by the building management system was used as well as measurements done with temperature loggers mounted on the piping and electrical power measurements on pumps and compressors.

MEASUREMENT RESULTS

Smaller cooling machine performance

Measurements were done during two periods: January 10 to 21 2005 and during April and May 2005. During winter only the smaller cooling machine is active. In Figure 4 measurement results are shown. Thermal power was determined by:

$$\dot{Q} = \dot{m}_{water} c_{p,water} \left(T_{evaporator,in} - T_{evaporator,out} \right)$$

Electrical power was directly measured on the compressor. On the right hand side axis the COP is shown for the cooling machine and for the installation including pumps and fans. The cooling machine is constantly turned on during the measurement period. During weekends less cooling is needed. During night time very little difference is seen in electrical power use. The small peaks in power use are caused by water sanitation twice a week during which the water is heated to an extreme value during a short period. After this the cooling machine has to run at full power to reach operational values again. COP of the cooling installation varies from 3 to 4, which is in fact lower than promised by the supplier. If pumps and fans are included COP drops to 2.

Figure 5 shows the temperatures of the water flows entering and leaving the condenser and evaporator of the smaller cooling machine. Day time variation of the ice-water temperature can again be clearly seen. Water coming from the cooling tower is at a constant temperature of 25°C. At the end of the measurement period displayed strange.

Oscillations can be observed in the temperature curves. As the outside air temperature during the end of the period climbed above 26.6 °C, the bigger cooling machine started to take over. In Figure 6 the energy use and performance of the cooling machine is again shown. Now a COP of 6 is reached, peaking to 8. (For the installation including fans and pumps COP drops to 5 and 4). If the cooling machine is running in part load, poorer performance is clearly seen.

Free cooling mode

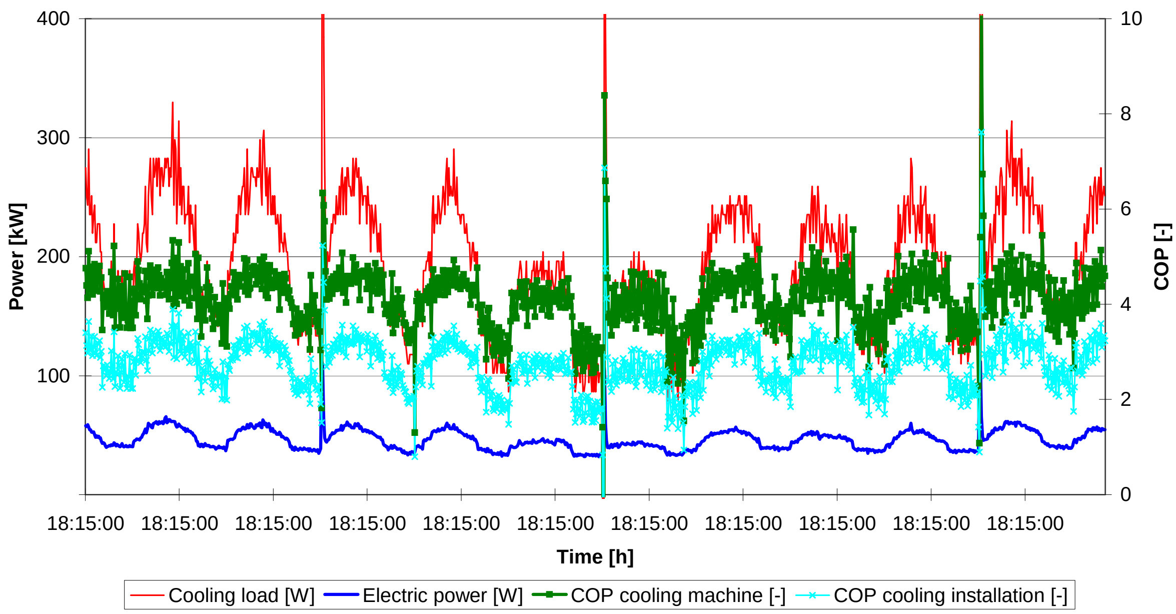


Figure 4 - Smaller cooling machine performance (January 2005)

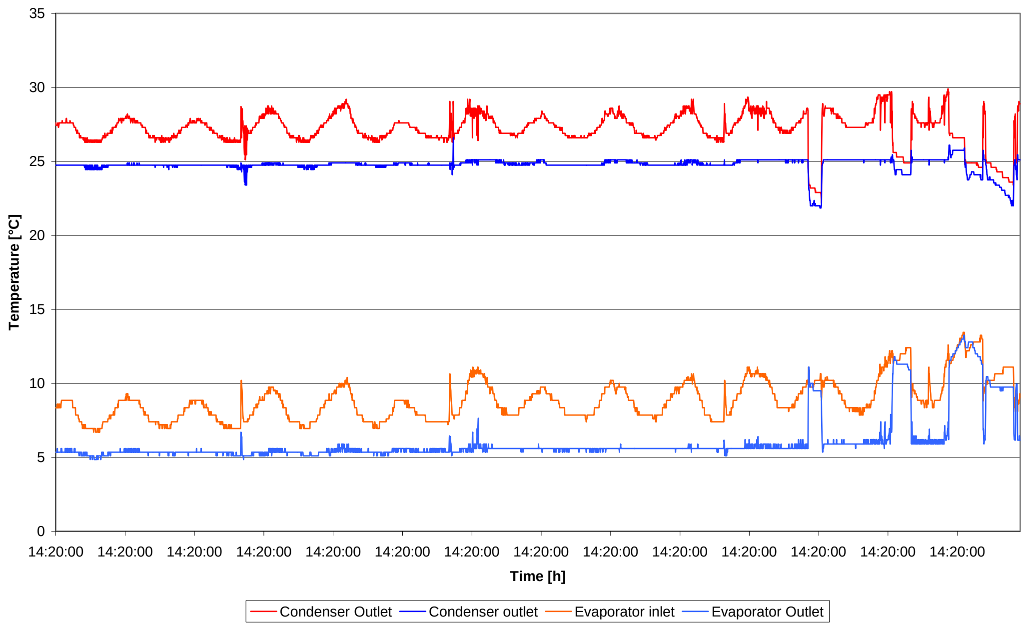


Figure 5 - Temperature measurements in smaller cooling machine (April-May 2005)

During the last week of February 2005 and the first week of March 2005 weather conditions were favourable for using free cooling. The inlet and outlet ice-water temperature are registered by the building management system. Because the pumps have a constant flow rate regime, the flow rate could be measured once, with an ultrasonic flow meter. The heat extracted from the building could thus be determined by:

$$\dot{Q}_{extracted} = \dot{m}_{ice-water} c_{p,water} \left(T_{ice-water,in} - T_{ice-water,out} \right)$$

The electricity used to operate the free cooling is coming from the pumps and the cooling tower fans. COP for free cooling is then given by:

$$COP = \frac{\dot{Q}}{P_{pumps} + P_{Fans}}$$

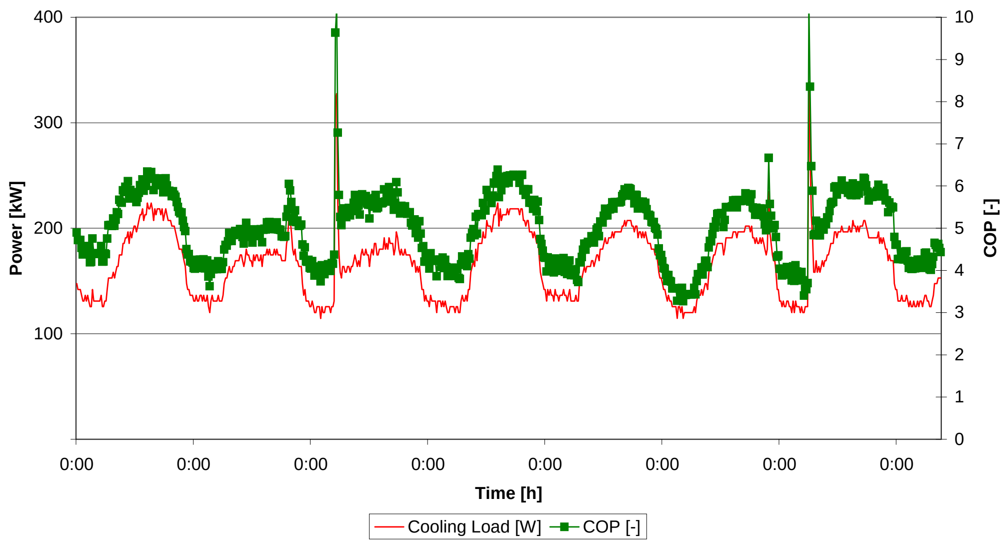


Figure 7 - Performance of the Free Cooling system

Figure 7 shows thermal capacity and COP of the free cooling mode. The COP of free cooling is approximately two times as large as for the cooling machine. It is striking to notice that the COP varies between 4 and 6, which is low compared to values found in literature [1]. The plate heat exchanger was designed to deliver 1307 kW of thermal power (building full load cooling conditions). During times when free cooling is possible, cooling load is a lot less, only 200 kW. The whole ice-water flow rate is nevertheless sent through the heat exchanger. This causes the temperature change of the ice water to be small (1°C or less, compared to 4°C in design) and the effectiveness of the heat exchanger under this condition to be only 0.37, as shown in Figure 8. Furthermore the cooling water for the heat exchanger is coming from the larger cooling tower. Again this causes the fan and pumping power to be too large, as the tower is oversized. Free cooling performance this way, does not reach the desired performance.

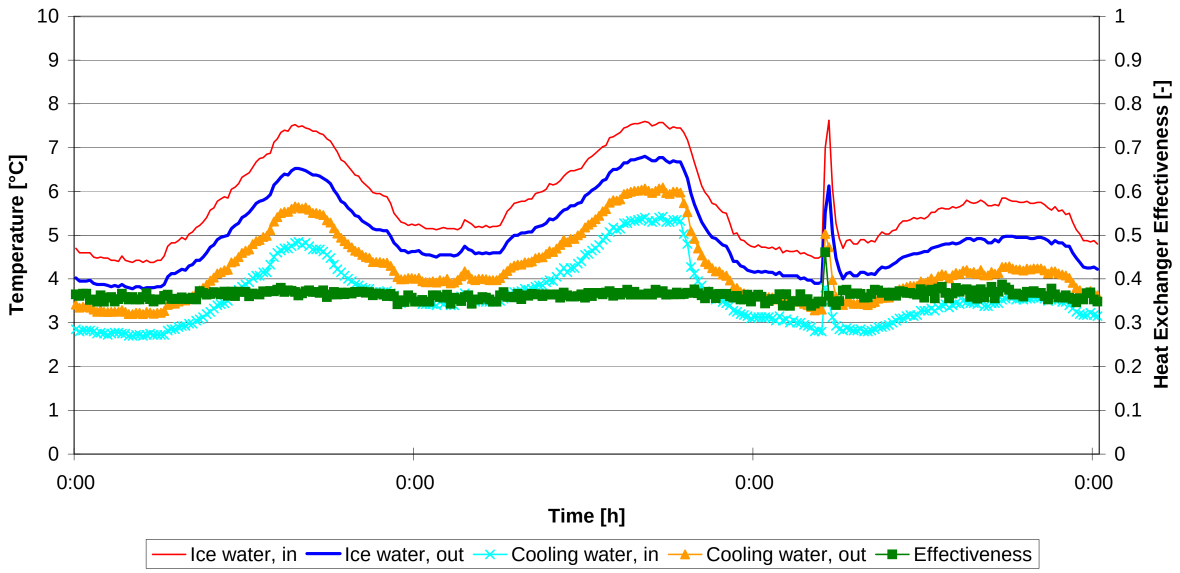


Figure 8 - Temperature measurements over the plate heat exchanger

Simulations and free cooling mode ameliorations

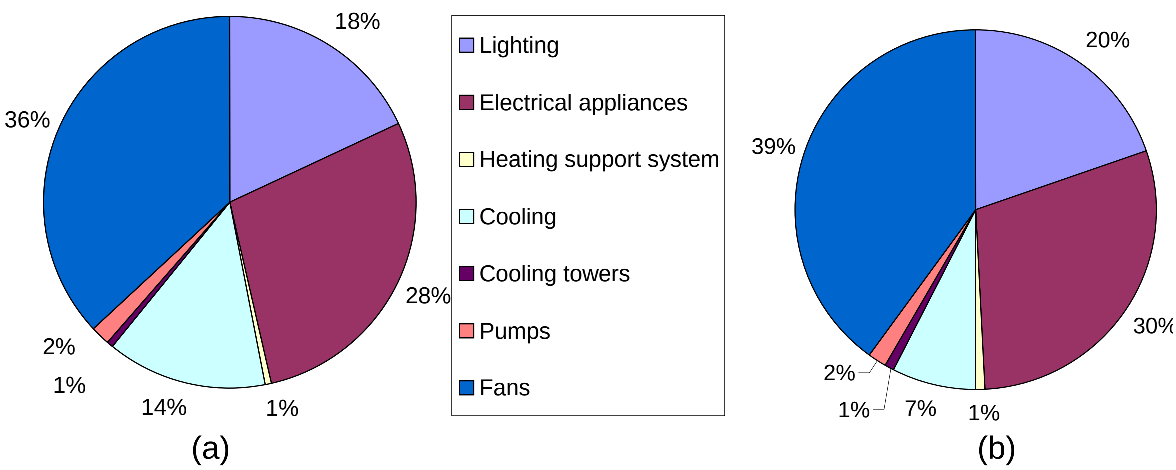


Figure 9 – Energy use in the VIB building (a) with free cooling (b) with free cooling (COP=10)

Using DOE_V6.2 a simulation model was constructed of the VIB-building. The model was calibrated with the data of the energy use during 2004. Total monthly electricity use could be predicted within a margin of 5%.

Figure 9 (a) shows the electrical energy use distribution of the building when no free cooling would be present. As expected, the fans of the HVAC-system are the main users of electricity. Second largest use for electricity goes to the electrical appliances, representing more than a quarter of the total use. This is more than for office buildings, because of the large laboratory facilities in the building. Finally cooling represents 14% of the total use.

If a three-way valve would be introduced in the ice-water system before the heat exchanger, only a part of the flow rate can be sent over the heat exchanger and mixed with an uncooled flow. This way the temperature drop of the cooled ice water would go up, resulting, after mixing, in the same ice-water temperature going to the building. Doubling the temperature change (using half the flow rate) would result in doubling the heat exchanger effectiveness.

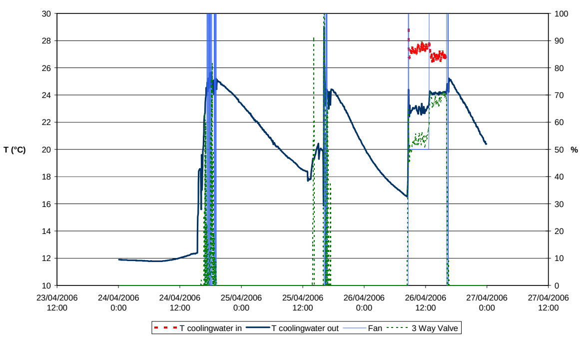
A resulting measure would be the reduction of flow rates at the cooling tower side. This could be done by installing frequency controller on the pumps. Finally the smaller cooling tower should be used. This will increase performance again. These measures will result in a COP of 10 for the free cooling system.

Simulations of the building showed that with a COP of 10 for free cooling the energy use for cooling could be strongly reduced (Figure 9 (b)). This could, also from simulations, result in a reduction of energy use of 80 000 kWh per year (about 1.25% of the total electricity use).

WATER USE

From April to August 2006 a second measurement campaign was conducted focusing on the water use of the cooling towers. Figures 10 and 11 show results for the smaller cooling tower.

In the month of April the cooling power is about 600 kW. In the morning of April 26 the cooling power is about 12% higher than in the afternoon. Water flow rates are also shown. Even with lower water flow rate a high cooling power is measured. The cooling towers are well sized. Evaporation loss lies between 0.5 and 0.9 m³/h. Note that evaporation loss is lower during the regime with higher water flow rates (afternoon of April 26). A lower cooler power is measured, meaning less evaporation occurs.



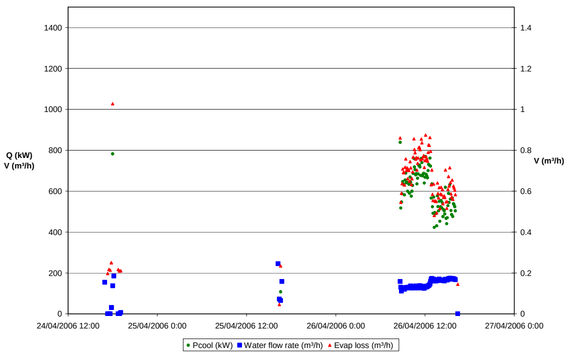


Figure 10: Cooling Tower 1, Cooling power, water flow rate and evaporation loss, April 2006

From 3 to 11 July 2006 a hot summer spell occurred. The cooling tower is thus running at full load most of the day. (Figure 11). July 4 at noon the water temperature after the chiller reaches a maximum value of 32 °C. The water is then cooled to 27 °C, as was given in the specifications of the constructor. From July 7 on the outside temperature goes down. Evaporation loss reaches 1.8 m^3/h at the maximal cooling load and drops to 1.2 m^3/h at lower loads. The water flow rate over the cooling tower is proportional to the power (10 drop in power results in 10% drop in flow rate), but the evaporation losses over the tower reduce a lot more. With lower air temperatures the water is less cooled by evaporation, and more by direct convective transfer.

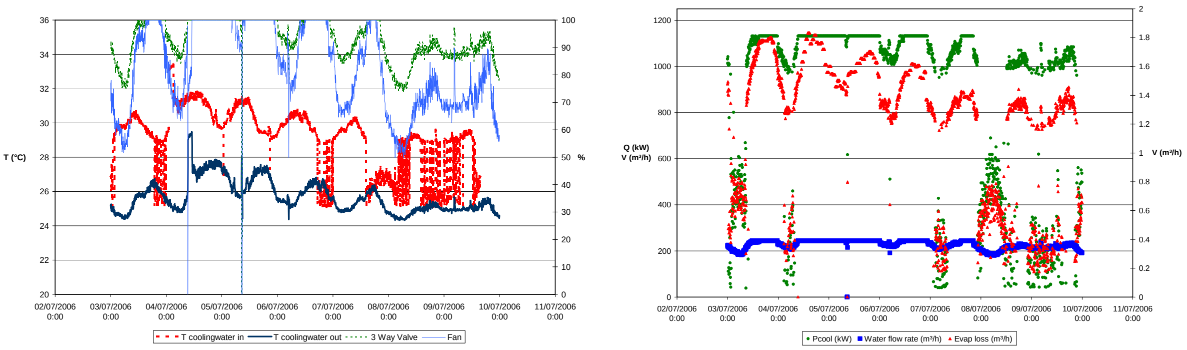


Figure 11: Cooling Tower 1, Cooling power, water flow rate and evaporation loss, July 2006

CONCLUSION

A measurement campaign was conducted on a high performance laboratory building. Special attention was paid to the free cooling system included.

Measurements showed that the free cooling system was badly sized during design. This causes the free cooling performance of the installation to be lower than expected.

By taking simple measures, the energy savings of Free Cooling could be easily doubled in the future.

Water consumption of the tower is not negligible. In general it is according to the specifications of the constructor.

NOMENCLATURE		***Subscripts***	
COP	Coefficient of Performance, -	in	inlet
$c_{p,water}$	specific heat capacity of water, J/kgK	out	outlet
$\dot{m}$	mass flow rate, kg/s		
P	electrical power, W		
$\dot{Q}$	cooling load (thermal), W		
T	temperature, K		

REFERENCES

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A precise rating method for energy performance of vapour compression chillers serving commercial buildings

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SUMMARY

This study aims to develop a new method to specify precisely the coefficient of performance (COP) of vapour compression chillers under part load conditions. Most chiller manufacturers currently specify a chiller COP based on standard rating conditions given in the (Air-conditioning & Refrigeration Institute) ARI standard 550/590. The COP calculated at such conditions is applicable for certain US conditions and is not appropriate for specifying the part load performance of chillers operating in parallel and working in any given climate zone. The ton-hour distribution of office buildings and hotels in Hong Kong was considered to indicate the unique operating conditions of chillers with specific number and capacity. Using the simulation program TRNSYS 15, a thermodynamic model was developed to determine the COP of air-cooled chillers under part load conditions. Results of this study highlight the need to use a set of part load performance curves with the working range of condensing temperature to describe precisely a chiller COP under various operating conditions.

INTRODUCTION

Vapour compression chillers are commonly used to provide cooling energy in air-conditioned buildings but with considerable electricity consumption. One possible way to reduce the electricity demand for commercial buildings is to select chillers with a high COP which is defined as the cooling capacity in kW over chiller power in kW. With regard to the performance rating of air-cooled chillers, most chiller manufacturers currently specify a chiller COP based on standard rating conditions given in the ARI standard 550/590 [1]. The full load conditions refer to an ambient temperature of 35^{o}C, a chilled water flow rate of 0.043 L/s per kW cooling capacity and a chilled water supply temperature of 6.7^{o}C. Considering that chillers in a chilled water system tend to operate at part load for most of the time, rating conditions were established for part load operation under which the part load ratios of a chiller ranging from 0.33 to 1 are related directly to outdoor temperatures ranging from 12.8 to 35^{o}C. Based on these part load rating conditions, the integrated part-load value (IPLV) was developed to express the part load performance of chillers. IPLV is defined as the sum of the chiller COPs at part load ratios of 0.25, 0.5, 0.75 and 1, multiplied by weighting factors of 0.12, 0.45, 0.42 and 0.01, respectively. The determination of the weighting factors is based on the average of buildings and climate conditions in 29 US cites.

Air-cooled chillers used in the EU countries may comply with the Eurovent certification programme for liquid chilling packages [2]. Regarding the Eurovent's part load ratings, chiller part load ratios ranging from 0.25 to 1 are related directly to outdoor temperatures ranging from 20 to 35^{o}C. A European seasonal energy efficiency ratio (ESEER), equivalent to

IPLV, was established for the universe of buildings and climates in the EU. In the calculation of ESEER, the weighting factors multiplying the chiller COPs at part load ratios of 0.25, 0.5, 0.75 and 1 are 0.23, 0.41, 0.33 and 0.03, respectively. It is worth ascertaining whether an IPLV or ESEER can present the part load performance of chillers working in any climate zone and running at various combinations to meet the changing building cooling load. The ARI standard 550/590-98 White Paper reminded that for a specific building in a certain climate zone, using the rated COP and/or IPLV would not come up with optimal selection of chillers. A whole year energy simulation is the more accurate way to compare chiller performance under all possible load conditions.

At present there are no alternative rating methods to deal with diverse operating conditions under the multiple chiller arrangement and to gauge the COP of chillers working in any given climate zone. In connection with this, standards for chiller performance requirements used in eight places [2,3-9] specify only the minimum allowable levels for the full load COP which vary from 2.7 to 3.2. Yet the COP of modern air-cooled chiller products at the full load conditions is in a range of 2.8 to 4.2 which, in most cases, exceeds the minimum COP levels. This suggests that the current standards are incapable of encouraging the use of highly efficient chillers. Only the ASHRAE and Canadian Standards Association standards [3,5] consider the minimum IPLV levels for chillers at part load operation. Yet the IPLV may fail to represent the actual part load performance of chillers. Indeed, the IPLV given by chiller manufacturers is higher than the full load COP, which contradicts the part load COP data monitored in existing chiller plants [10]. The discrepancy between the IPLV and actual part load COP is mainly due to a situation where the ARI rating conditions assume the drop of ambient temperature at each % load reduction and do not consider at what level the evaporating temperature and condensing temperature are set. As far as air-cooled chillers are concerned, the condensing temperature can hover by various degrees above the outdoor temperatures ranging between 13 and 35^{o}C. For any given combination of chiller loads and outdoor temperatures, the chiller COP can vary with the condensing temperature floating at different values based on head pressure control. It remains to be ascertained how the variation of the condensing temperature influences the specification of chiller COP.

In view of the pitfalls of the current performance rating methods, the aim of this paper is to illustrate a more precise rating method to specify the COP of air-cooled chillers under part load conditions. The ton-hour distributions of an office building and a hotel were used to explain the diverse operating conditions of chillers when working in parallel to meet the building cooling load. Drawing on a sophisticated thermodynamic model for air-cooled centrifugal chillers [11], the interaction between the chiller COP, part load ratio, condensing temperature and outdoor temperature was investigated. Results of this study will provide guidance on how to formulate a set of part load performance curves with the working range of condensing temperature to describe precisely a COP under various operating conditions. Engineers, manufacturers and researchers will better understand how the chiller COP curves help determine a chiller performance benchmark for buildings in any climate zone.

METHODS

Evaluation of hourly building cooling loads and chiller loads

The simulation program TRNSYS 15 [12] was used to calculate the hourly cooling loads of a reference office building and an existing hotel in Hong Kong throughout the year. The office building is widely employed to study the average energy performance of local office buildings. The hotel is the representative of the trade and has average characteristics in terms of the number of guestrooms, total floor area and annual electricity consumption in kWh per unit floor area of the building in m^2. Table 1 summarizes key information about the buildings and their air-conditioning systems. Detailed features of the buildings and their simulation procedure were described elsewhere [11]. It was assumed that every piece of air side equipment is capable of delivering the cooling energy to meet the cooling demand for the thermal conditions specified in each air-conditioned area. The hourly local weather data in 1989 were used because 1989 was the test reference year which is considered representative of the prevailing weather conditions in Hong Kong in building energy analysis.

Table 1. Key information about the office building and hotel.

Building	Office	Hotel
Total air-conditioned area (m^2)	42,840	45,540
Building operation schedule	11 hrs/day, 5.5 days/wk	24 hrs/day, 7 days/wk
Total business hours per year	3131	8760
Total cooling hours per year	2834	8152
Annual cooling energy (kWh)	7,423,883	17,692,414
Peak cooling load (kW)	6,389	4,753
Chiller plant details Total capacity installed (kW) Number and type of chillers Nominal capacity per chiller (kW) Chilled water distribution system Number and type of chilled water pump Rated power of each pump (kW)	 6,744 6 air-cooled centrifugal chillers 1,124 Single-loop pumping system with a differential pressure by-pass pipe 6 constant speed 31.3	 5,000 5 air-cooled centrifugal chillers 1,000 Single-loop pumping system with a differential pressure by-pass pipe 5 constant speed 16.8
Air side system details Type of air handling units Chilled water flow control	 Variable air volume 2-way valve	 Variable air volume 2-way valve

As Table 1 illustrates, the total capacity of the chiller plants was designed to meet the peak cooling load and equally sized chillers were installed to operate in parallel to meet the changing building cooling load. Chiller sequencing was implemented so all the chillers were operating at the same part load ratio, and no additional chillers started to operate until each of the running chillers was carrying full load. Tables 2 and 3 show the schedules of staging chillers and ranges of chiller part load ratios for the office building and hotel. The part load ratio (PLR) of each operating chiller is given by the building cooling load divided by the number of chillers staged and their nominal capacity.

Table 2. Schedule of staging chillers and ranges of chiller part load ratios for the office building.

Building load ratio (BLR)	Number of operating chillers and their capacity (kW)	Total capacity of operating chillers (kW)	Step	Chiller part load ratio (PLR)
$0<BLR\leq 0.17$	1 x 1124	1124	1	$0.29\leq PLR\leq 1$
$0.17<BLR\leq 0.34$	2 x 1124	2248	2	$0.5\leq PLR\leq 1$
$0.34<BLR\leq 0.51$	3 x 1124	3372	3	$0.67\leq PLR\leq 1$
$0.51<BLR\leq 0.68$	4 x 1124	4496	4	$0.75\leq PLR\leq 1$
$0.68<BLR\leq 0.85$	5 x 1124	5620	5	$0.80\leq PLR\leq 1$
$0.85<BLR\leq 1$	6 x 1124	6744	6	$0.83\leq PLR\leq 0.95$

Table 3. Schedule of staging chillers and ranges of chiller part load ratios for the hotel.

Building load ratio (BLR)	Number of operating chillers and their capacity (kW)	Total capacity of operating chillers (kW)	Step	Chiller part load ratio (PLR)
0<BLR≤0.21	1 x 1000	1000	1	0.21≤PLR≤1
0.21<BLR≤0.42	2 x 1000	2000	2	0.5≤PLR≤1
0.42<BLR≤0.63	3 x 1000	3000	3	0.67≤PLR≤1
0.63<BLR≤0.84	4 x 1000	4000	4	0.75≤PLR≤1
0.84<BLR≤1	5 x 1000	5000	5	0.80≤PLR≤0.95

To avoid presenting a huge set of hourly building cooling loads and chiller loads, the temperature bin method used to determine IPLV was followed to compile the load data in the form of ton-hours. For each temperature bin, the ton-hours represent the total number of equivalent full-load hours, which was determined by adding all hourly loads expressed as ratios of their peak value. The interval of each bin was set at 2.8°C and there were ten bins with temperatures ranging from 7.2 to 35°C to cover all hourly outdoor temperatures found in 1989. With regard to the four bin groupings to evaluate the four weighting factors of IPLV, the 1st temperature bin of 32.2–35°C is for the design bin, 2nd to 5th bins from 21.1–32.2°C for the peak bin, 6th to 9th bins from 10.0–21.1°C for the low bin and 10th bins below 10.0°C for the minimum bin. For each bin group, the weighting factor, average load ratio and average entering condenser air temperature (i.e. outdoor temperature) were computed. These three parameters were used to judge if the IPLV formula is suitable for local applications.

Four building operation categories (Groups 1 to 4) are defined in the ARI standard 550/590 to generate four separate IPLV formulas (see Table 4, where A is COP at 100% load, B is COP at 75% load, C is COP at 50% load and D is COP at 25% load). The office building and hotel generally comply with the building operation categories of Group 3 and Group 1, respectively. Group 2 and Group 4 were considered here by applying airside economizers to the hotel and the office building, respectively, when the outdoor temperature is below 18.3°C, instead of 12.8°C shown in Table 4. The level of 18.3°C is adequate for deciding free cooling based on local weather conditions and is the base temperature for calculating cooling degree days.

Table 4. Building operation categories and IPLV formulas given in ARI standard 550/590.

Group	Building operation category	IPLV formula
1	24 hrs/day, 7 days/wk, -17.8°C and above	IPLV = 0.009 A + 0.309 B + 0.413 C + 0.269 D
2	24 hrs/day, 7 days/wk, 12.8°C and above	IPLV = 0.012 A + 0.423 B + 0.565 C + 0.0 D
3	12 hrs/day, 5 days/wk, -17.8°C and above	IPLV = 0.015 A + 0.409 B + 0.392 C + 0.184 D
4	12 hrs/day, 5 days/wk, 12.8°C and above	IPLV = 0.018 A + 0.501 B + 0.481 C + 0.0 D

Determination of the COP and operating data of air-cooled chillers

A thermodynamic model was developed using TRNSYS 15 to determine the operating variables of an air-cooled centrifugal chiller with characteristics shown in Table 5. The chiller capacity is comparable to that used in the chiller plants of the two buildings. Details about the development of the chiller model were given in Ref. [11]. The model was verified using a wide range of operating data and the performance data in chiller specifications. It considered the mass balance of refrigerant and energy balance at the evaporator, and the real process phenomena, including the capacity control of the compressor, variations in the overall heat transfer coefficients of the evaporator and condenser at part load. An algorithm was included to compute the number and speed of condenser fans staged based on a set point of condensing temperature. The model was sophisticated enough to study the steady-state behavior of chiller COP at various combinations of chiller loads and outdoor temperatures when different strategies for controlling the condensing temperature were present.

Table 5. Details of the chiller model.

Refrigerant type	HFC-134a
Nominal capacity (kW)	1266
Evaporator component	
Type	Shell-and-tube flooded
Evaporating temperature (^{o}C)	5
Temperature of supply chilled water (^{o}C)	7
Temperature rise of chilled water at full load (^{o}C)	5.5
Nominal chilled water flow rate (kg/s)	55.1
Compressor component	
Number and type	One hermetic centrifugal compressor; constant speed
Capacity control	Inlet guide vanes; controlled at 25–100% full capacity
Condenser component	
Designed condensing temperature (^{o}C)	50 at an outdoor temperature of 35^{o}C
Number and type of condenser fans	3 constant speed fans
Nominal heat rejection airflow (m^3/s)	118; each fan provides 39.3 m^3/s
Type of expansion valve	Orifice plate

RESULTS AND DISCUSSION

Ton-hour distribution of building cooling loads and chiller loads

Based on simulation results of hourly building cooling loads and chiller loads, the ton-hours distributions were constructed for the office building and hotel, as shown in Figure 1. The distributions generally follow an irregular "hill" shape. Based on local weather conditions, the number of total cooling hours is highest (604 hrs for office; 1890 hrs for hotel) at the bin of 26.7–29.4^{o}C with the average of 28.1^{o}C. This results in the highest ton-hours in all the operation categories, regardless of the building cooling load or chiller load. The building load means the chiller load when a single chiller is installed in a chiller plant. At a given temperature bin, higher ton-hours would mean a higher frequency of chiller operation at near full load. It is found that the multiple-chiller arrangement is a direct approach to prolonging the operation of the chillers at higher loads. The ton-hours distributions could vary a lot, depending on the type of building and its operating hours, the use of free cooling, and the chiller plant design with respect to the number and capacity of chillers. Based on the vague relationship between the ton-hr and outdoor temperature, it is questionable to gauge the real part load COP by the IPLV formula with a specific set of part load rating conditions.

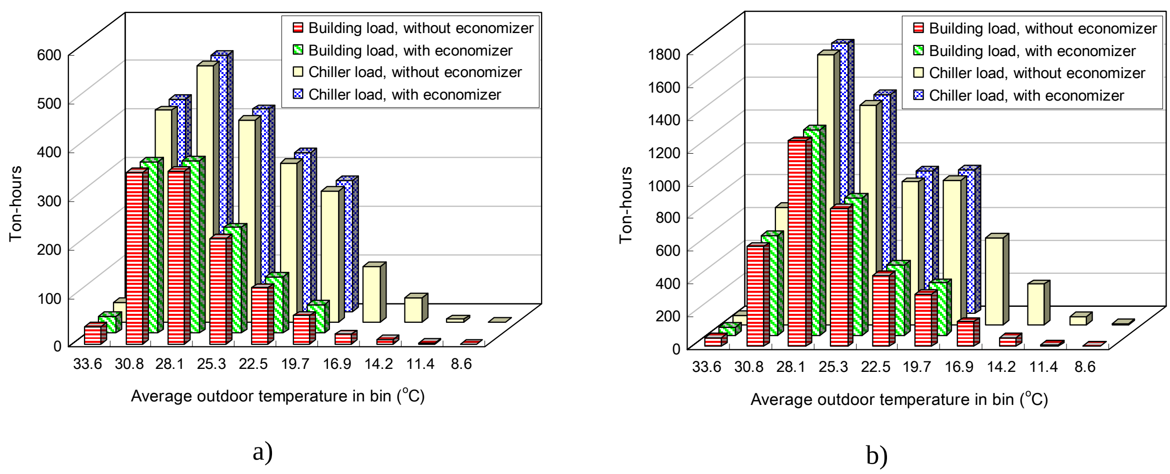


Figure 1. Ton-hour distribution of building cooling loads and chiller load. a) office building, b) hotel.

Critique of using the IPLV formula

The parameters used to create the IPLV formula were evaluated for the buildings with different operating characteristics and chiller plant designs. The results of the parameters are summarized in Table 6. At each bin group, the average outdoor temperature varies slightly at different categories of load and operation and the average load ratio changes widely (0.78–0.91 for design bin; 0.51–0.85 for peak bin; 0.12–0.70 for low bin; 0.09–0.41 for minimum bin). This confirms the inadequacy of using a specific set of part load ratings (stated in the ARI and Eurovent standards) to present the diverse combination of outdoor temperatures and chiller loads. Furthermore, it is insufficient to consider only four loading conditions (25%, 50%, 75% and 100% of full load) to present an IPLV. It is important to ascertain COPs at the peak bin with part load ratios of 0.51–0.85 because this loading range accounted for over 72% of the total ton-hours based on the weighting factors calculated. The ton-hrs at above 80% full load accounted for only 0.9–3% of the total ton-hours. This weakens the importance of using a full load COP to specify minimum chiller performance requirements. The accuracy of using IPLV to present the actual part load performance of chillers relies heavily on assessing the prominent ambient and load conditions and adjusting the weighting factors from time to time in response to various building features and chiller plant designs. It is worth considering other performance rating methods to account for diverse operating conditions of chillers.

Table 6. Parameters of IPLV formulas for the office building and hotel

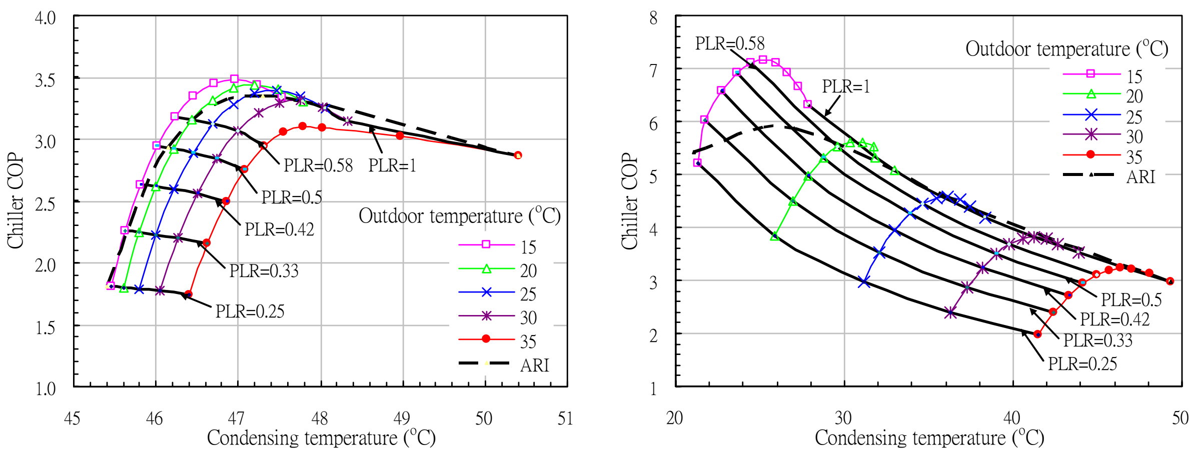
Parameter	Building load, without economizer		Building load, with economizer		Chiller load, without economizer		Chiller load, with economizer	
	Office	Hotel	Office	Hotel	Office	Hotel	Office	Hotel
Total hours at all 4 bin groups	2834	8152	2480	6713	2834	8152	2480	6713
Total hours at each bin group:								
Design bin	45	64	45	64	45	64	45	64
Peak bin	2023	5383	2023	5383	2023	5383	2023	5383
Low bin	766	2692	412	1266	766	2692	412	1266
Minimum bin	0	13	0	0	0	13	0	0
Total ton-hrs at all 4 bin groups	1162	3723	1132	3508	2186	6342	2014	5515
Total ton-hrs at each bin group:								
Design bin	35	52	35	52	40	58	40	58
Peak bin	1039	3135	1039	3135	1704	4575	1704	4575
Low bin	88	534	58	320	411	1703	270	882
Minimum bin	0	1	0	0	0	5	0	0
Weighting factor at each bin group:								
Design bin (at 100% load)	0.030	0.014	0.031	0.015	0.019	0.009	0.020	0.011
Peak bin (at 75% load)	0.894	0.842	0.918	0.894	0.780	0.721	0.846	0.830
Low bin (at 50% load)	0.076	0.144	0.051	0.091	0.202	0.269	0.134	0.160
Minimum bin (at 25% load)	0	0.0003	0	0	0	0.001	0	0
Average outdoor temperature at each bin group (^{o}C)								
Design bin	33.6	33.6	33.6	33.6	33.6	33.6	33.6	33.6
Peak bin	26.8	26.5	26.8	26.5	26.8	26.5	26.8	26.5
Low bin	17.9	17.6	19.7	19.7	17.9	17.6	19.7	19.7
Minimum bin	-	8.6	-	-	-	8.6	-	-
Average load ratio at each bin group								
Design bin	0.78	0.81	0.78	0.81	0.90	0.91	0.90	0.91
Peak bin	0.51	0.58	0.51	0.58	0.84	0.85	0.84	0.85
Low bin	0.12	0.20	0.14	0.25	0.58	0.63	0.65	0.70
Minimum bin	-	0.09	-	-	-	0.41	-	-

Significance of using a set of part-load performance curves

Based on a verified model for air-cooled centrifugal chillers, the interaction between the COP and operating variables was analyzed. Figure 2 shows how the COP at various operating conditions can be described precisely by a set of part load performance curves when the chiller operated under head pressure control (HPC) or condensing temperature control (CTC). HPC means that the number of condenser fans staged is kept minimal to control the condensing temperature by various degrees above 45°C. CTC, on the other hand, means that the condensing temperature hovers closely above the outdoor temperature by staging all condenser fans as much as possible to minimize the sum of compressor power and condenser fan power. Existing rating methods consider only the correlation between the COP, chiller part load ratio (PLR) and outdoor temperature. The part load COP curves show the importance of specifying the working range of condensing temperature in order to reflect the possible range of chiller COP at any given operating condition or combination of outdoor temperatures and part load ratios. It is possible to judge the superiority of part load performance with the reduced condensing temperature. As no specific link between the outdoor temperature and chiller load is considered, such curves are generic to compare the part load performance of chillers working in any given climatic zone and chiller plant design.

A set of part load COP curves with the working range of condensing temperature not only helps engineers to better understand the chiller COP at various operating conditions, but also enables plant operators to verify whether their chillers can operate with the specified performance. The condensing temperature is generally monitored. The part load ratio of the chillers will be ascertained if the flow and temperature difference of chilled water across each chiller are measured. The chiller power is typically monitored in terms of % full load ampere. Any decline in the COP of the chillers due to deficient operation can be assessed by comparing the measured COP with the nominal COP given from the part load COP curves.

Given the ARI's part load ratings, ARI performance curves were constructed for HPC and CTC. Each of these curves fails to explain how the chiller COP can rise with a decrease in the outdoor temperature or condensing temperature at a given part load ratio. The ARI performance curves also give imprecise information about the uniqueness of maximum COP at the PLR of 0.75 for HPC and 0.4 for CTC for all ambient conditions. The optimum loading points for maximum COP, in fact, are different, depending on the outdoor temperatures.



a) b)

Figure 2. Part load performance curves of the chiller. a) head pressure control, b) condensing temperature control.

CONCLUSIONS

This paper demonstrates how to specify precisely the COP of air-cooled chillers under various operating conditions. An analysis on the ton-hour distribution of typical local chiller operation categories showed that ARI's part load rating conditions are incapable of presenting diverse operating conditions of chillers working in parallel to meet the changing building cooling load. A set of part load performance curves should be developed to show how the chiller COP varies at different condensing temperatures for any given combination of outdoor temperatures and part load ratios. It is expected that such curves are applicable to presenting the part load performance of water-cooled chillers when the outdoor temperature is replaced by the entering condenser water temperature which depends on the ambient wet bulb temperature and the working approach of cooling towers. It remains to be seen how to foster chiller manufacturers to provide their products with specific sets of part load performance curves. It also remains to be investigated how to specify the upper limit of condensing temperature to state the minimum chiller COP at any given part load condition and hence to boost the part load performance of chillers. If minimum allowable COPs at part load conditions are established, a pragmatic chiller performance benchmark can be formulated for buildings in any climate zone.

ACKNOWLEDGEMENT

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Modelling of a heating and cooling plant coupled to a hotel using HFC or CO_2 as a working fluid. Comparison of the performances.

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SUMMARY

Since ozone depleting substances were banned, HFCs have become the most common working fluids in refrigeration devices. However, these greenhouse gases are responsible for global warming and are likely to be banned by European directives. Interest is now being turned towards natural fluids such as CO_2. The Heating and Cooling Plant (HCP) is intended to produce domestic hot water (DHW) and to heat and cool residential and smaller office buildings. The HCP presents a special design of the refrigeration circuit and an innovative defrosting system. Two HCPs (CO_2 and HFC) coupled to a hotel have been modelled using TRNSYS software. Simulations show that CO_2 is less efficient. Nevertheless, this working fluid seems to be well suited to the operation of the HCP, because it enables to produce DHW at 55°C and because the high amount of energy recoverable by subcooling increases the efficiency of the defrosting system.

INTRODUCTION

Many buildings have simultaneous needs in heating and cooling. Typically, during summer, they face needs in cooling and in domestic water heating. During in-between seasons, from March to May and from October to November, rooms facing north have to be heated and rooms facing south have to be cooled, especially in buildings with significant glazed surfaces. In winter, only heating is required. The Heating and Cooling Plant (HCP) [1] is intended to satisfy the needs for domestic hot water production, space heating and cooling all year round and uses a special defrosting technique.

This article describes an innovative defrosting system for a Heating and Cooling Plant. This machine is coupled to a hotel whose needs in domestic water heating are quite substantial. It is assumed that the system uses either a hydrofluorocarbon (HFC) or carbon dioxide as a working fluid. As the two machines are quite similar, only the carbon dioxide type is described in this paper. Then annual simulations are run using TRNSYS software [2] for both plants and the results are compared and analyzed.

THE HEATING AND COOLING PLANT

Three heat exchangers

The CO_2 HCP (figure 1) has three heat exchangers: a condenser and an evaporator that heat and cool two water tanks for space heating and cooling and an exchanger that works either as a condenser or as an evaporator on ambient air to adapt production to needs. This last exchanger evacuates any excess cold produced in a heating mode and any excess heat produced in a cooling mode.

Three operating modes

The HCP works in three modes: a heating mode using the water condenser and the air exchanger as an evaporator; a dual mode preparing hot and cold water using the water heat exchangers; and a cooling mode using the water evaporator and the air heat exchanger as a secondary condenser. The cooling mode also integrates the production of domestic hot water by using the top part of the water condenser as a primary condenser.

An innovative defrosting technique

The water condenser is divided into three parts. The top part is used for domestic water preparation, the middle part is dedicated to the production of hot water for space heating and the lower part is connected to the cold water tank. During winter, this feature enables the storage of heat in the cold tank. The HCP alternates between heating mode and dual mode. In heating mode, the heat exchangers involved are the air evaporator and the three parts of the water condenser. The temperature of the cold tank rises to a maximum of 15°C. The fins of the air evaporator become frosted. The HCP then changes mode to operate in the dual mode. In this mode the fan of the air heat exchanger stops. The water evaporator and the top and middle parts of the water condenser are now involved. This allows for better COP by working on water as a source on both sides. A second advantage of the circuit design is that the defrosting system is automatic: after passing through the water condenser for domestic hot water production and space heating, the working fluid enters the air heat exchanger at a temperature around 35°C and is subcooled by exchange with the frosted fins. The cycle finishes on dual mode until the cold tank temperature decreases back to 5°C. Then the HCP switches back to heating mode and so on.

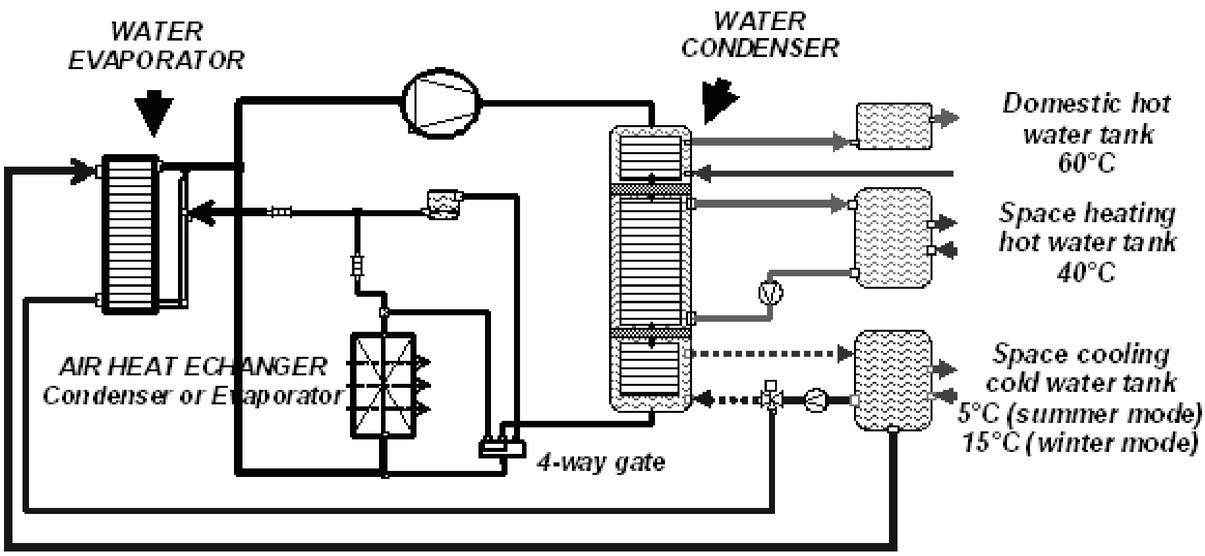

Figure 1. The Heating and Cooling Plant working with CO_2 as a working fluid

THE REFRIGERANTS

R407C

The choice of the refrigerant is linked to three criteria: performance, cost and security and environmental regulations.

Table 1. Characteristics and properties of some refrigerants

Fluid	CFC-12	HCFC-22	HFC-134a	NH_3 R-717	C_3H_8 R-290	CO_2 R-744
Natural substance	No	No	No	Yes	Yes	Yes
Molar mass (g/mol)	120,92	86,48	102,03	17,03	44,10	44,01
Volumic refrigerating capacity at 0°C (kJ/m^3)	2740	4344	2860	4360	3870	22600
ODP	1	0.05	0	0	0	0
GWP (100 years)	7100	1500	1200	0	0	1
Flammable?	No	No	No	Feebly	Yes	No
Toxic?	Feebly	Feebly	Feebly	Yes	No	No

Table 1 shows the most common fluids used until now in refrigeration. R12 is a chlorofluorocarbon (CFC) and R22 is a hydrochlorofluorocarbon (HCFC). Because of their impact on the ozone layer (Ozone Depletion Potential ODP $\neq 0$), the sale of devices using these substances was banned in Europe in 2000 for CFCs and in 2004 for HCFCs [3] [4]. Widely used in mobile air-conditionning systems, R134a is one of the best performing HFCs however it has a high Global Warming Potential (GWP = 1200). Therefore these refrigerants are also likely to be banned in future by the European directive on F-gases. R407C is a HFC mostly used in heat pumps and air-conditionning for residential applications. Its GWP is 1503.

Carbon dioxide

According to Gustav Lorentzen, author of "Revival of carbon dioxide as a refrigerant" [5], CO_2 is today's best natural refrigerant as a substitute for HFCs. CO_2 is quite performant when used in transcritical cycles (Fig. 2). In such cycles the major problem is dealing with high pressures. Indeed for CO_2, low pressures are around 40 bars and high pressures can reach 120 bars. During heat rejection, the pressure of the refrigerant is above its critical value (73.8 bar for CO_2). This implies that there is no drastic change of phase at heat rejection. When the temperature decreases, supercritical gas progressively turns to supercritical liquid. The temperature of the fluid at the compressor outlet is between 70 and 100°C. The gliding temperature obtained at the heat rejection side enables the production of hot water at different temperatures as required for the HCP (Dosmestic hot water at 60°C and space heating water at 40°C). Producing DHW with a HCP R-134a would have required an auxiliary heater. The CO_2 transcritical cycle seems to suit residential and office applications selected for the HCP. Besides, the defrosting technique can also benefit from the good thermodynamic properties of supercritical carbon dioxide because of the high amount of energy created by subcooling.

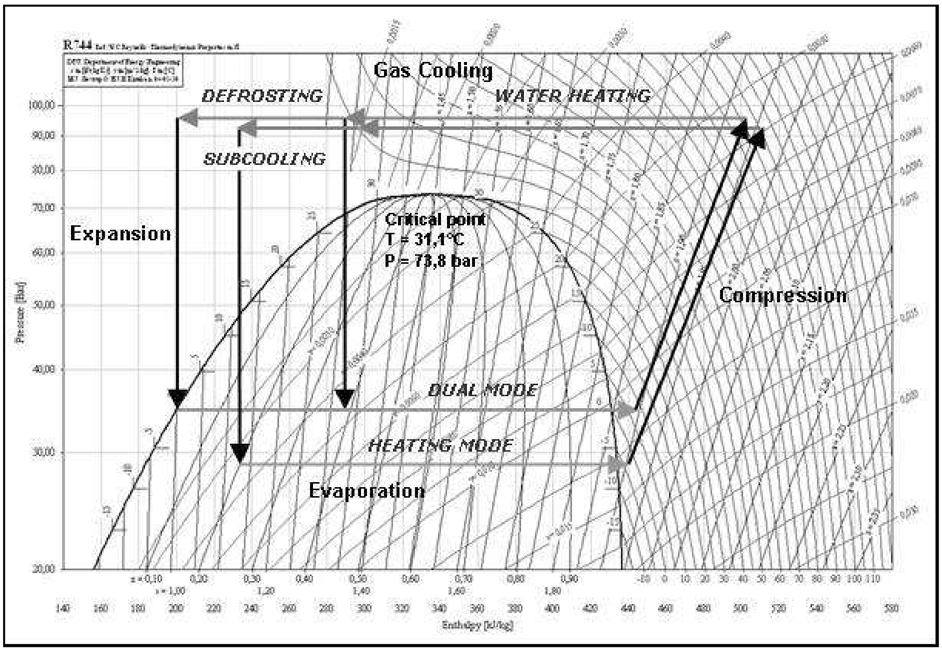

Figure 2. Transcritical cycles of carbon dioxide on the Mollier diagram

MODELLING HYPOTHESES

Building model

The building model is a hotel of 45 rooms where domestic hot water consumption is quite significant. Thermal zones, wall compositions, glazings, solar protections, ventilation et occupancy and light schedules are defined using the building module of TRNSYS so as to calculate the hotel's daily needs in heating and cooling. The simulations are run using a weather data file from Paris.

Heating and cooling plant model

The components of the system are also modelled using TRNSYS software. Simulations calculate the heating and cooling capacities and the coefficients of performance for the three operating modes (heating, cooling and dual mode) according to the ambiant temperature. The HCP is supposed to have a heating capacity of 100 kW in the heating mode for the base temperature of Rennes (-6°C). This capacity corresponds to a maximum operating time of 10 hours during the coldest day the year.

Operating strategy

The HCP is designed to work mainly in the dual mode as this is the most efficient. Then it works in the heating or the cooling mode depending on the needs of the building. When heating, the HCP alternates between the heating and dual mode. The cold tank temperature increases in the heating mode and decreases in the dual mode between 5 and 15°C. Time ratios were previously calculated in order to take into account the alternation of the two modes in the heating sequence with and without frosting (Equations 1 and 2). These ratios were supposed constant and are reported in Table 2. The ratios are different with and without frosting because when appearing the defrosting capacity is added to the heating capacity. To balance the higher heating capacity, the cooling capacity increases as well. Therefore the cold tank temperature decreases more quickly from 15 to 5°C.

$$r_{DM-noF} = \frac{dual\ mode\ time\ without\ frosting}{time\ of\ the\ heating\ sequence} \tag{1}$$

$$r_{DM-F} = \frac{dual\ mode\ time\ with\ frosting}{time\ of\ the\ heating\ sequence} \tag{2}$$

Table 2. Time ratios in heating sequences

Time ratio	CO_2	R407C
r_{DM-noF}	0.28	0.15
r_{DM-F}	0.25	0.12

A third time ratio was calculated to estimate the time of frosting conditions during each day of the year (Equation 3).t is assumed that frost appears when the ambient temperature is beyond 7°C.

$$r_{T<7°C} = \frac{Number\ of\ hours\ with\ T_{amb} < 7°C}{24} \tag{3}$$

Operating times per mode

Dual mode is the first mode until the minimum need is satisfied (Equation 4). The right-hand member is due to the alternation between dual and heating mode during a heating functioning.

$$t_{DM} = \min\left(\frac{q_h}{Q_{hDM}}, \frac{q_c}{Q_{cDM}}\right) + \left[r_{T<7°C}.r_{DM-F} + \left(1 - r_{T<7°C}\right).r_{DM-noF}\right] \times \left[\frac{q_h}{Q_{hHM}} - \min\left(\frac{q_h}{Q_{hDM}}, \frac{q_c}{Q_{cDM}}\right)\right] \tag{4}$$

where q_h is the daily need in heating, q_c, the daily need in cooling, Q_{hDM}, the heating capacity in dual mode and $_{HM}$ stands for heating mode.

In heating mode, the real operating time is the time needed to cover all the heating needs minus the dual mode time (Equation 5).

$$t_{HM} = \frac{q_h}{Q_{hHM}} - t_{DM} \tag{5}$$

In cooling mode, the real operating time is the time needed to cover all the cooling needs minus the time dedicated to cooling of the dual mode (Equation 6).

$$t_{CM} = \frac{q_c}{Q_{cCM}} - \min\left(\frac{q_h}{Q_{hDM}}, \frac{q_c}{Q_{cDM}}\right) \tag{6}$$

where $_{CM}$ stands for cooling mode.

Coefficients of performance

The daily and annual coefficients of performance are calculated in proportion to the operating times.

RESULTS AND DISCUSSION

Building simulation

The annual simulation of the 45-room hotel (Figure 3) shows the evolution of heating and cooling needs all year round. Annual needs amount to 25322 kWh in DHW production, 68640 kWh in space heating and 30146 kWh in cooling. Paris is situated in a temperate climate where heating needs are higher than cooling needs. It can also be pointed out that during in-between seasons heating and cooling needs appear simultaneously. During summer the plateau in the heating curve corresponds to the hot water production that is nearly constant. From the end of April to the end of October hot and cold productions are simultaneous during at least a few hours day.

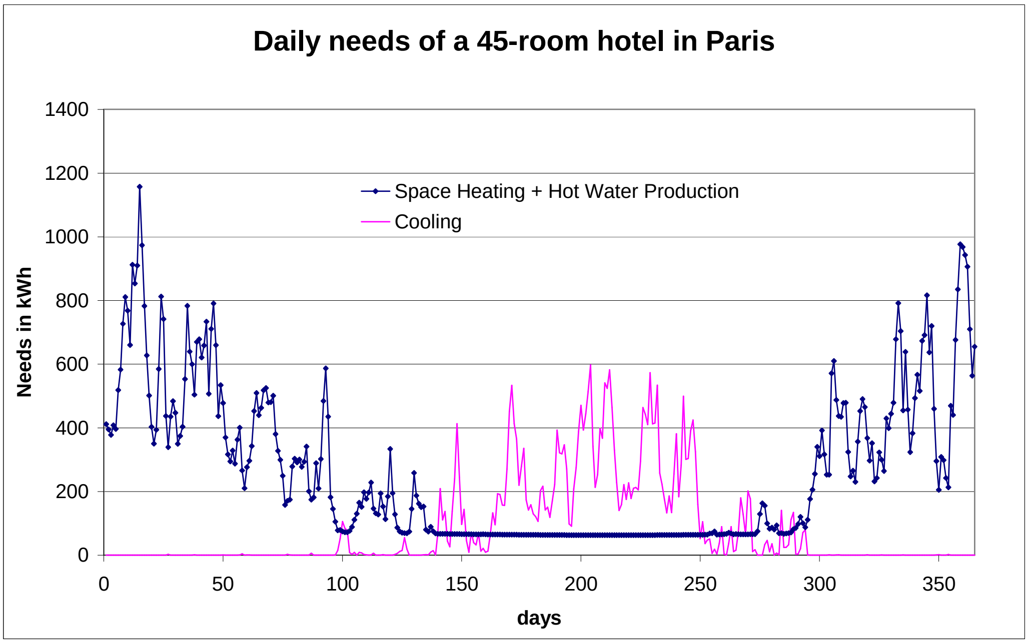

Figure 3. Evolution of daily needs in heating and cooling of a 45-room hotel in Paris

Operation times

Operation times per mode to satisfy the needs of the building are calculated and reported in table 3. Performances of carbon dioxide being lower the total operation time is greater. The weaker cooling capacity of CO_2 implies an increase in the cooling mode time whereas in heating mode the weaker performances are balanced by an increase in the dual mode. This is due to the better time ratio between dual and heating mode during winter. These values show that in spite of its poorer performance CO_2 as a working fluid better suits the modal operation technique of the heating and cooling plant.

Table 3. Comparison of the operation times per mode

Operation times	R407C	CO2
Dual mode	134.7 h	244.4 h
Heating mode	543.5 h	513.0 h
Cooling mode	127.2 h	291.1 h
Total	805.4 h	1048.6 h

Coefficients of performance

These graphs on figure 4 show clearly that CO_2 has a lower COP for space heating and cooling than any HFC. The curves record a serious increase during in-between seasons, and summer especially for R407C. For carbon dioxide the performances in cooling are not as good as in heating. So when cooling demands are important, the average COP goes down. Annual COPs are near their value in winter because the operation times are higher in the heating period for the climate of Paris.

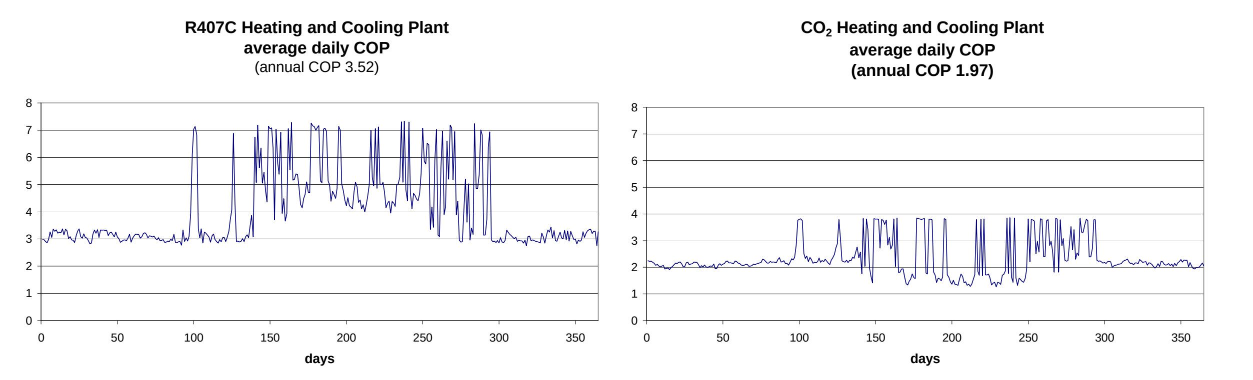

Figure 4. Comparison of daily coefficients of performance between R407C and CO_2

In addition the results of the HCP can be compared to the ones of a classic reversible heat pump (Table 4). The decrease of the performances due to carbon dioxide is less for the HCP than for the reversible heat pump. By another way, replacement of a standard heat pump by a HCP leads a better increase with carbon dioxide than with a HFC.

Table 4. Comparison of annual COPs of HCPs and reversible heat pumps

Fluid	R407C	CO2	Decrease
Reversible Heat Pump	3.18	1.58	-50.4%
Heating and Cooling Plant	3.52	1.97	-43.9%
Increase	10.7%	25.1%	

CONCLUSIONS

Today, HFCs are the most efficient “secure” fluids on the market. Unlike these greenhouse gases, carbon dioxide is environmentally friendly but is about 50% less efficient. CO_2 technology needs more development to be competitive in space heating and cooling applications. Nevertheless its thermodynamical properties allow domestic hot water production. It is an option to explore for the CO_2 HCP. The new defrosting technique for heat pumps noticeably increases average performance in heating mode, and increases more with high subcooling capacity.

The model presented here show how this Heating and Cooling Plant can satisfy the needs in heating, cooling and domestic hot water production of a hotel of 45 rooms situated in Paris. The HCP records better annual performances by between 10 and 25% and could even be improved with carbon dioxide for lower cooling needs obtainable in Northern countries such as Finland.

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Theoretical Studies And Experimental Research Regarding The Evaluation Of Energetic Consumptions During The Production Of Menial Refrigeration

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SUMMARY

The present paper brings out to knowledge the results of the studies and the experimental research, carried out during a project belonging to the National Program of Excellency Research, regarding the evaluation and analysis of energy consumptions during the production of menial refrigeration in Romania.

The authors analyze the refrigerating equipment categories and present the graphic evolution in time of afferent energy consumptions (time versus energetic consumption).

Also, specific consumption indexes, for the manager refrigerating equipment used in Romania, are being defined and determined, both theoretically and experimentally. The equipment mentioned previously is either used for producing alimentary refrigeration or for producing ventilation refrigeration.

In order to define the specific consumption index, one must keep in mind: the type if the refrigeration equipment, the type of the refrigerating agent which was used, the effect of the exploitation conditions and the manufacturer.

Furthermore, we present studies and analysis regarding the improvement of energy consumptions during the production of ventilation refrigeration, by using heating pumps and the heating pump on the whole-refrigerating equipment.

INTRODUCTION

Within a national research project of Energsys Excellence, along with the energetic rehabilitation analysis of the menial consumers’ charging systems, with either thermal or electrical energy, in Bucharest, the energetic consumption for the production of menial refrigeration and the refrigeration used in the comfort air-conditioning, were theoretically and experimentally analyzed, as well.

Lately, a continuous rise of the electric energy’s consumption which insures the alimentary menial refrigeration, has been noticed. We are witnessing a continuous change in Terra’s climate, Romania being affected, as well, by the atmosphere warming, and with this, the home air-conditioning becoming more of a necessity.

THE ENERGETIC CONSUMPTION'S EVOLUTION FOR THE MENIAL REFRIGERATION AND FOR THE AIR CONDITIONING IN ROMANIA

The energetic consumptions' for menial refrigeration

Using the statistic data, given by specialized institutions, we distinguished graphically the evolution in time of the equipments' categories used in the menial refrigeration (refrigerators, freezers, combines) represented by the cooled volume encountered with a frequency of more than 95% among the menial consumers. We must add the fact that in 1975, refrigerators were represented by one door refrigerators, in 1985, refrigerators with freezer and combines, made in Romania, begin to be used (Arctic and R12), and, by 1990 numerous major equipments in producing menial refrigeration start to be commercialized (Zanussi, Electrolux, Indesit, Gorenje, etc), with thermodynamical performances and with various effects on the medium. If, in the beginning, the commercialization was accessible enough, allowing also equipments using refrigerants forbidden by the Montreal Protocol, after it's ratification by Romania (1993) only refrigerating equipments with ecological working fluids have been imported (R134a, R600).

In figure 1, we can observe a continuous volume rise of the cooling volume for the menial consumption. Simultaneously, because of the increased comfort and the need of a more reduced specific energy's consumption, the population turns to more advanced equipments (with larger volumes, but more efficient), a contribution to this being brought by the energetic labeling of these equipments. More and more used are the refrigerators with more efficient equipments (compressors, heat exchangers), in search of placing them within the energetic classes A and A+.

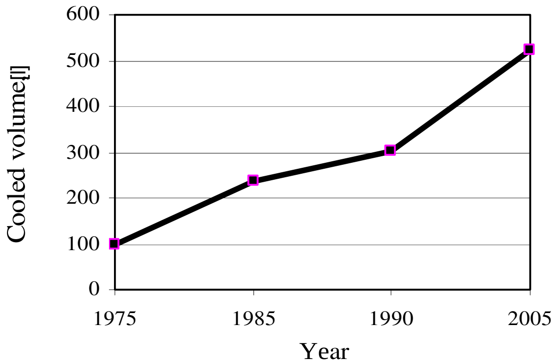

Figure 1: The required cooled volume vs. year.

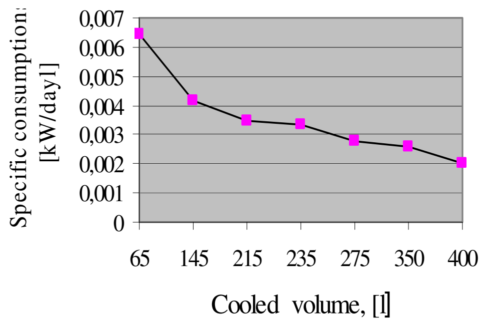

Figure 2: The variation of the energy specific consumption vs. cooled volume.

In figure 2 one can notice a drop of the specific energy's consumption (W/l). The highest specific consumptions correspond to the more reduced volumes (65….1251), a characteristic of the less advanced Romanian equipments (ARCTIC-the old generation). Along with the rise of the cooled volume (the ones used lately being the equipments with higher volume), but also with the use of advanced equipments, one can observe a continuous diminish of the specific energy consumption.

Tables 1, 2 and 3 present equipments for domestic cold used in Romania, remarking different technical characteristics, due to the used frigorific fluid (The best being those with R 600), as well as to the construction (refrigerators with one door, with two, as well as combines).

Table 1: Refrigerators with one door.

Type	Total volume, [l]	Refrigerator volume, [l]	Freezer volume, [l]	Refrigerant	Electric power, kWh/24h	Energetic class
Arctic FB 14	123	109	14	R 134A	0.60	B
Arctic FB 18	156	137	19	R 134A	0.75	C
Arctic FB 24	228	208	20	R 134A	0.71	B
Zanussi ZT 75	62	57	5	R 134A	0.53	A
Zanussi ZRX 7 JB	70	65	5	R 134A	0.38	A
Zanussi ZERC 0750	67	62	5	R 134A	0.33	A+
Zanussi ZRT 15 JB	146	128	18	R 134A	0.64	A
Zanussi ZRT 16 JB	160	153	7	R 134A	0.42	A
Zanussi ZRC 19 JB	190	172	18	R 134A	0.69	A
Zanussi ZC 244 AO	231	211	20	R 134A	0.84	A
Electrolux ERE 3000	290	290	0	R 600a	0.37	A+
Gorenje R63391W	388	388	0	R 134a	0.50	A

Table 2: Refrigerators with two doors.

Type	Total volume, [l]	Refrigerator volume, [l]	Freezer volume, [l]	Refrigerant	Electric power, kWh/24h	Energetic class
Platinium PR 23	180	132	48	R 134a	0.62	A
Platinium PR 26 A	235	187	48	R 134a	0.81	A
Platinium PR 35 A	312	242	70	R 134a	1.08	A
Technolux REF 400 B	337	267	70	R 134a	1.32	C
Technolux REF 455 B	425	326	99	R 134a	1.23	B
Arctic D276+	263	190	73	R 134a	0.65	A+
Arctic D285	261	210	51	R 134a	0.82	A
Arctic RD 29 N	293	222	71	R 134a	0.90	A
Arctic D306+	288	215	73	R 134a	0.67	A+
Indesit TAN 2 W	251	195	56	R 134a	0.82	B
Indesit TAN 2	259	199	60	R 134a	0.79	A
Indesit TAAN 3	303	233	70	R 134a	0.72	A+
Indesit BAN 13	330	217	113	R 134a	0.92	A
Whirlpool ARC 2220	240	187	48	R 600a	0.91	A
Whirlpool ARC 3130	277	219	58	R 600a	1.05	A
Zanussi ZD22/5 AO	270	220	50	R 134a	0.82	A
Zanussi ZRD 27 JC	270	220	50	R 134a	0.82	A
Electrolux ENL 6298X	522	357	165	R 134a	1.43	A

Table 3: Combines .						
Type	Total volume, [l]	Refrigerator volume, [l]	Freezer volume, [l]	Refrigerant	Electric power, kWh/24h	Energetic class
Arctic K 245	207	145	62	R 134a	0.78	A
Arctic	237	175	62	R 134a	0.81	A
Arctic K 346	340	253	87	R 134a	0.91	A
Arctic KS 32 NC	301	200	101	R 134a	1.27	B
Zanussi ZK 21/10 B	268	200	68	R 134a	0.74	B
Zanussi ZRB 34 N	335	226	109	R 134a	0.92	A
Indesit BAN 13	330	217	113	R 134a	0.92	A
Platinium PR 36 A/2C	318	228	90	R 134a	0.91	A
Beko CDP 4501	287	200	87	R 134a	0.7	A+
Beko CDP 4701	331	244	87	R 134a	0.75	A+
Whirlpool	320	212	108	R 134a	0.93	A+
Whirlpool ARC 5570	350	242	108	R 600a	0.96	A
Whirlpool ARC 7630	332	239	93	R 600a	1.01	A
Whirlpool ARC 6700	359	236	123	R 600a	0.78	A+
Whirlpool ARC 7632	332	237	95	R 600a	1.01	A
Ariston MBT 1912 FI	396	280	116	R 134a	1.10	A
Ariston MBL 1923 CV	415	283	132	R 134a	0.85	A+
Electrolux ERB 4052	400	291	109	R 600a	0.81	A
Electrolux ENB 3451 X	340	249	91	R 600a	0.97	A
Electrolux ERN 2921	290	214	76	R 600a	1.15	B
Electrolux ERN 2925	290	214	76	R 600a	0.87	A
Electrolux ERD 3420X	333	263	70	R 600a	0.92	A
Gorenje RK6334E	308	205	103	R 134a	1.30	B
Gorenje K317CLA	310	242	68	R 134a	0.91	A
Gorenje RK6355W	322	204	118	R 134a	0.97	A
Gorenje RK65365A	331	245	86	R 134a	0.87	A+
Gorenje RK61391W	364	278	86	R 134a	0.99	A

Energetic consumptions for air conditioning

For home air conditioning, the corresponding rise of the electrical energy consumption is even more spectacular, being almost absent before 1990 and reaching today to a point with air conditioning installation in almost 50% of homes. This phenomenon is due to the continuous rise of the air’s temperature in summer, on one hand, and to the comfort conditions inside the houses, which are higher by day, on the other hand. Considering the importance of electric energy consumption for building air conditioning out of the total of menial consumptions, we consider adequate the analysis of these individual consumptions, as well as the necessity of their reduction by adopting new design solutions and new exploitation techniques. Two systems for comfort ventilation inside buildings are being used: systems with direct evaporation (split, multisplit) and systems with indirect cooling with secondary refrigerant (chiller and fan coils).

Systems with direct evaporation

For small cooling capacity (for a surface of 25÷120m2), the system which imposed itself is the one with direct evaporation, with one or more inner units. Studies and experimental researches regarding the performance of the systems with direct evaporation (Table 4) show their electric energy consumptions (Pc), in various working conditions: ambient air temperatures, evaporation temperatures (t0) and condensation temperatures (tC), of the common refrigerating agents used in installations (R134a, R404A, R410A).

Table 4: COP for air conditioning systems with direct evaporation .

$t_{amb.}$, [°C]	t_{int} [°C]	t_0, [°C]	t_C, [°C]	Q, [kW]	R 134a		R404A		R410A	
					Pc, [kW]	COP, [-]	Pc, [kW]	COP, [-]	Pc, [kW]	COP, [-]
30.00	20.00	8.00	45.00	9.60	2.24	4.29	2.52	3.81	2.43	3.95
32.50	20.00	8.10	47.50	9.60	2.42	3.96	2.76	3.47	2.65	3.61
32,50	20.00	8.30	50.00	9.60	2.61	3.68	3.02	3.17	2.89	3.32
37.50	20.00	8.40	52.50	9.60	2.81	3.42	3.31	2.90	3.15	3.04
40.00	20.00	8.60	55.00	9.60	3.01	3.19	3.62	2.65	3.43	2.80

Romania, as a country in progress, has benefited up until this year of facilities regarding the Montreal Protocol exigency, and so, in the present study the only refrigerant taken into account is R134a, as it is the most often used at this moment. The average electric consumption for homes' air conditioning was calculated for a functioning period of 10 hours a day, having as a result the data in table 5.

Table 5: The performance of air-conditioning systems with direct evaporation.

Living space	Living surface	Technical features of cooling plants		Average consumption of electrical power
	[m²]	Cooling capacity, [kW]	COP, [-]	[kWh/24h]
Studio flat	25	1.5	4,61	3,25
Apartment 2 rooms	40	2.4	4,61	5,2
Apartment 3 rooms	60	3.6	4,61	7,81
Apartment 4 -5 rooms	80-120	4.8 – 7.2	4,61	10,4…15,61

Indirect cooling systems (with secondary refrigerant).

In conditions identical to the ones of the direct cooling (the same spaces, the same inner and outer temperatures and the same functioning period), the cooling capacity and the driving electrical power are presented in table 6.

Table 6. The performances of air conditioning systems with secondary refrigerant.

No.	Living space	Living surface	Technical features of cooling plants		Average consumption of electrical power
		[m²]	Cooling capacity, [kW]	COP, [-]	[kWh/24h]
1.	Studio flat	25	1.5	3,77	3,98
2.	Apartment 2 rooms	40	2.4	3,77	6,37
3.	Apartment 3 rooms	60	3.6	3,77	9,55
4.	Apartment 4 -5 rooms	80-120	4.8 – 7.2	3,77	12,73…19,1

It is clearly seen that the systems with a direct cooling are more economical (the electric consumption is reduces with 18%). Nevertheless, the systems with indirect cooling are being used due to the advantages given by: the possibility of using ammonia, the most advanced, the cheapest and most ecological refrigerant; it uses a low quantity of refrigerant with a favorable influence over the ambient; the possibility of being used in district cooling systems.

Both systems have evolved towards variants with reversible functioning (insuring both the air conditioning and the building heating) or with heat recovery, which improve more and more their performances.

SPECIFIC INDICATORS

Further, we present the experimental data for the specific indicators, for the production of menial refrigeration and air conditioning, using the most customary refrigerant (R134a).

Menial refrigeration

Table 7. Energy consumption for the production of menial refrigeration.

No.	Living space	Cooling volume of refrigerator, [l]		Average consumption of electrical power, [kWh/24h]	Specific consumption [10^{-3} kW/(day,l)]
		Refrigeration	Freezing		
1.	Studio flat	228	20	0,71	2,87
2.	Apartment 2 rooms	322	70	0,89	2,37
3.	Apartment 3 rooms	425	99	1,17	2,22
4.	Apartment 4 5 rooms	522	165	1,43	2,08

The average energy consumption was calculated for certain equipments presented in tables 1, 2, 3 in accordance with the size of the menial consumer.

Air conditioning energy consumption (working time 10 hours/day)

Direct evaporation systems

Table 8. Specific air conditioning indicators with direct evaporation systems.

No.	Living space	Type of air conditioning unit	Cooling capacity, [kW]	Electrical power, [kW]	Average consumption of electrical power [kWh/24h]	Specific consumption [kW/(m2,zi)]
1.	Studio flat	Monosplit	1,5	0,325	3,25	0,13
2.	Apartment 2 rooms	Bisplit (2 UI)	2,4	0,47	4,7	0,1175
3.	Apartment 3 rooms	Multisplit (3 UI)	3,6	0,72	7,2	0,12
4.	Apartment 4-5 rooms	Multisplit (4-5 UI)	4,8 – 7,2	0,94….1,37	9,4…13,7	0,1175.... 0,114

Indirect cooling systems with secondary refrigerant

Table 9. Specific air conditioning indicators with indirect cooling systems.

No.	Living space	Cooling capacity, [kW]	Electrical power, [kWh/24h]	Average consumption of electrical power [kWh/24h	Specific consumption kW/(m2,zi)]
1.	Studio flat	1,5	0,398	3,98	0,159
2.	Apartment 2 rooms	2,4	0,572	5,72	0,143
3.	Apartment 3 rooms	3,6	0,855	8,55	0,1425
4.	Apartment 4 -5 rooms	4,8 – 7,2	1,12….1,68	11,2…16,8	0,14

From the data in tables 8 and 9, one can observe that the specific indicator has higher values in the case of the indirect evaporation with cca 21%, in rapport with the direct evaporation; the same dropping tendency can be observed in the case of the specific indicator's value the same as the decrease of the cooled volume.

DISCUSSION

The data regarding the energy consumption for the production of menial refrigeration and air conditioning are specific to Romania today.

Between the years 1975 and 2005, it has been canned a. ascending evolution of the cooled volume for menial refrigeration, determined by the continuous rise of the lifestyle. The specific consumptions for the production of menial refrigeration have canned a continuous drop determined by the increase of equipment's performances.

The air conditioning has also canned major increases and determined, at the same time, the electric energy's menial consumption rise.

The average energy consumption necessary to the air conditioning depends on the volume of the cooled space and on the cooling system type.

Nevertheless, in spite of their reduced electric consumption of the air conditioning system with direct evaporation, they are not recommended.

NOMENCLATURE

- UI – interior unit;
- COP -coefficient of performance.

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A 95% More Efficient Telecom Cooling System

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SUMMARY

Reduction in investment of over 100 million Euros reduction and more than 5 million Euros per annum in operating costs for cooling telecom systems, combined with a higher level of cooling availability and in particular of the telecom technology; these are the consequences of a very special paradigm change in the cooling of telecom systems. The use of the new cooling system will enable Swisscom alone to make savings of around 70,000 MWh in electricity throughout Switzerland, with investments being reduced by more than 100 million Euros. The sustainability can also be seen in the drastic reduction in refrigerant quantities and generally lower material flows. However, the solution strategies are not only attractive to other telecom providers but also offer valuable input for future computer centre cooling systems.

INTRODUCTION

Because of ageing, the existing cooling systems used in Swisscon telecommunications centres and remote units are having to be replaced. Replacing the air-conditioning systems in the more than 1,000 centres will give rise to costs of € 180 million for the whole of Switzerland. In addition, the current high operating costs will be reduced by a not-inconsiderable amount. The rapid development in the field of telecommunications technology demands more flexible and more low-cost cooling systems.

These are the driving forces behind the start-up of the Mistral projects which are aimed at the development and implementation of new telecom cooling systems. The requirements were as follows:

- adherence to the required air-conditioning levels
- simplicity of the system
- value for money
- flexibility in changeover
- modularity, expandability
- redundancy in the case of higher thermal loads and high availability
- energy savings
- minimal environmental pollution
- easy operation and maintenance
- adherence to the Swisscom safety specifications

The Project Team has already accumulated three years experience from the design, pilot and test phase and implementation of the innovative indoor temperature control systems. To-date, implementation has been carried out in around 150 control rooms and most of these have been operating for several years.

BASES

The ETSI guidelines (European Telecommunications Standard Institute) form the basis for cooling telecom system control rooms throughout Europe. These define normal and extreme ranges. A distinction is drawn between two profiles of requirements depending upon the importance of the telecom equipment. These are the **ambient classes 3.1** and **3.6** in accordance with the ETIS standard 300 019-1-3 [1]. In practice the 3.1 requirements apply almost exclusively for all telecommunications installations at Swisscom.

Control rooms for the **ambient class 3.1** (see Figure 1) are defined in the ETIS standard.

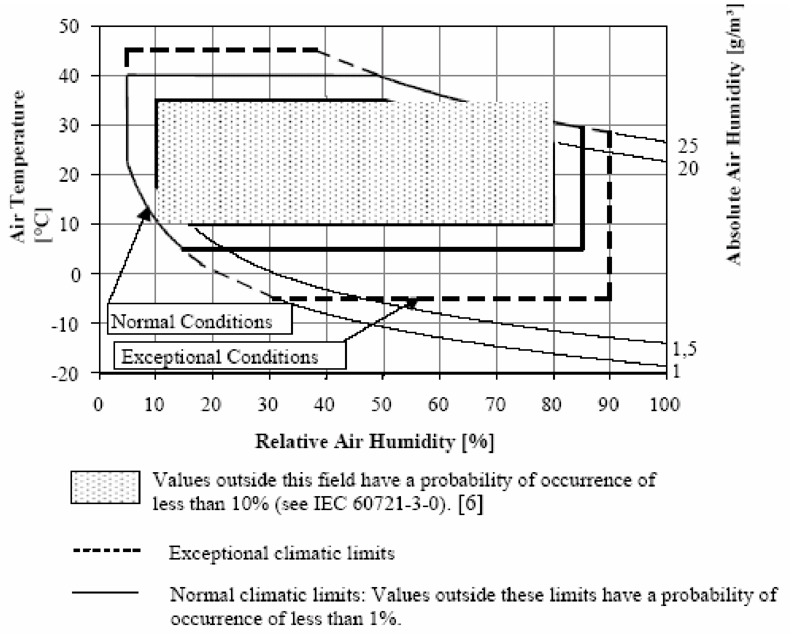

Figure 1. Climatogram for Class 3.1: Temperature-controlled locations.

Swisscom aims to achieve an average annual temperature of < 26°C in order to extend the service life of the telecom systems and to maintain the comfort for the employees at a high level. However, higher levels over a short period (a couple of hours per year) have no affect on the service life and are readily accepted by the users.

In order to provide a full picture at this point we show Figure 2 for rooms in the **ambient class 3.6**. It defines the standards for control rooms which have high temperature requirements.

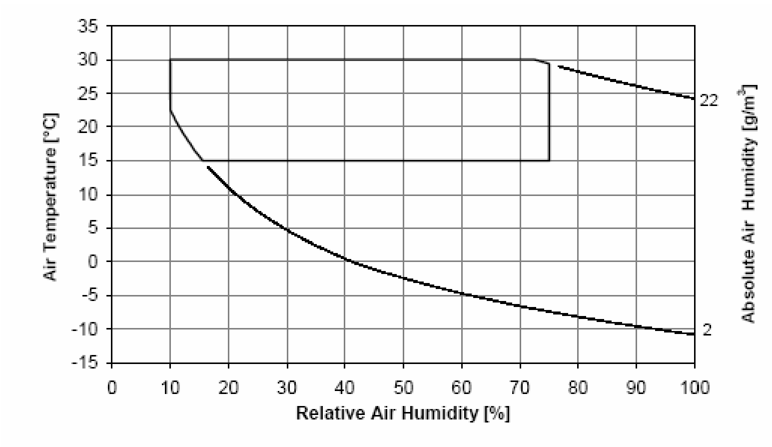

Figure 6: Climatogram for Class 3.6: Telecommunication Control Rooms

Figure 2. Climatogram for Class 3.6: Telecommunication Control Rooms.

The relationship between availability and room air temperature is shown in the graphic below. The warmer the average room air temperature the higher the relative error rate with the telecom installations. One of the specifications was consistent or even improved availability. The current average room temperature of 26°C was therefore taken as a basis.

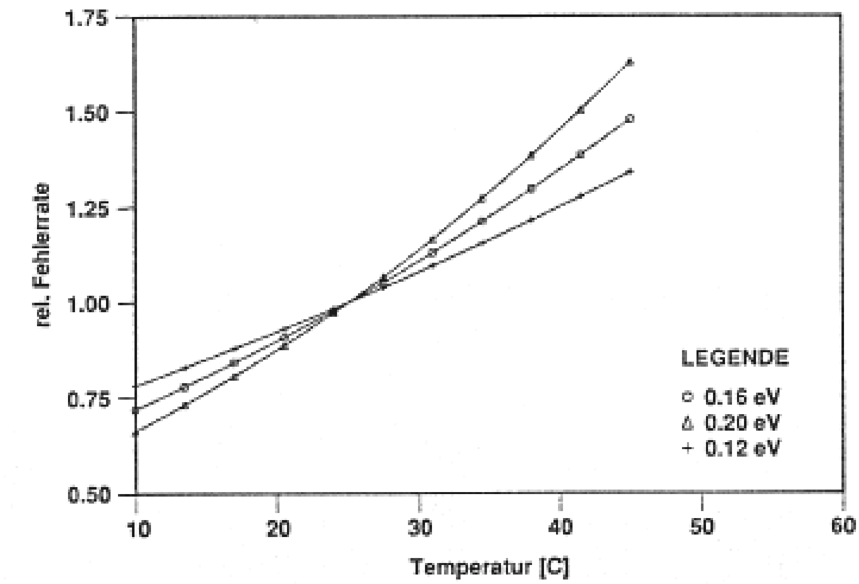

Figure 3 Relative failure rate of exchange depending on room air temperature for error activation energy of 0.12, 0.16 and 0.20 eV respectively [2]

FUNCTION / PRINCIPLE

The approach for an efficient solution is to tackle the root cause of the problem.
The heat that escapes above the rack is extracted as close as possible to the source and evacuated directly out of the room. This hotspot extraction prevents the hot air from the racks having any effect on the room at all.

Thanks to the direct evacuation of the heat only a minimal portion becomes active in the room through thermal radiation and so this can be ignored. This means in turn that at most we will achieve the external temperature in the room by supplying non-cooled outside air for the fresh air supply. There is however the added factor of the influence exerted by the building mass. At night, as the outside air cools, the ventilation reduces the temperature in the room and of the building mass and the storage effect of the mass has a positive effect on the development of the room temperature.

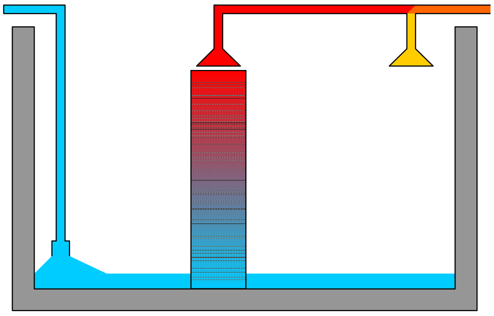

Diagram 1 This sketch of the principle shows the simple function method, the hotspot extraction is fitted above the rack. The fresh air supply using the outside air is carried out via screens with filters.

Depending upon the specific thermal load and type of application, there are a number of different systems from which to choose. The most frequent solutions typically selected are those with two exhaust fans.

Table 1. Description of ventilation principle

Ventilation principle	**Thermal load[W/m^2]**	**Description**
Mistral	Up to 680	2 outlet air fans each with 50% of the air volume flow with hotspot extraction and fresh air supply above ground via screens and filters. 50% redundancy

Photo 2 Mistral outlet 2V

Photo 3. Hotspot extraction

RESULTS

Readings clearly demonstrate that the systems are functioning superbly. The results from last Summer, a genuine 'hot' Summer in Switzerland with extreme climatic conditions, are shown below.

Meteorological data

The daily mean temperatures reached in July 2006 are the highest since the Swiss Meteorological Institute (SMA) began taking systematic records in 1864. If we look at the annual external temperature graph 2005/2006 from the SMA in Bern-Liebefeld we can see an exceptional density of high external air temperatures in July 2006. The highest reading was 33.7°C. The annual mean temperature of the external air is 8.9°C, this reflects quite a hot year for the Swiss Mittelland region.

Readings in a solid room with thermal load of 300 W/m²

Diagram 4 shows a Mistral system in a Swisscom telecommunications centre near Bern. The system is able to maintain the mean room air temperature over the course of the year at 24.8°C. In Summer, peak levels of 33°C were recorded over a period of 21 hours, 12 K below the permitted maximum in accordance with the ETIS ambient class 3.1 (Figure 1). The target, low average annual temperature is therefore maintained.

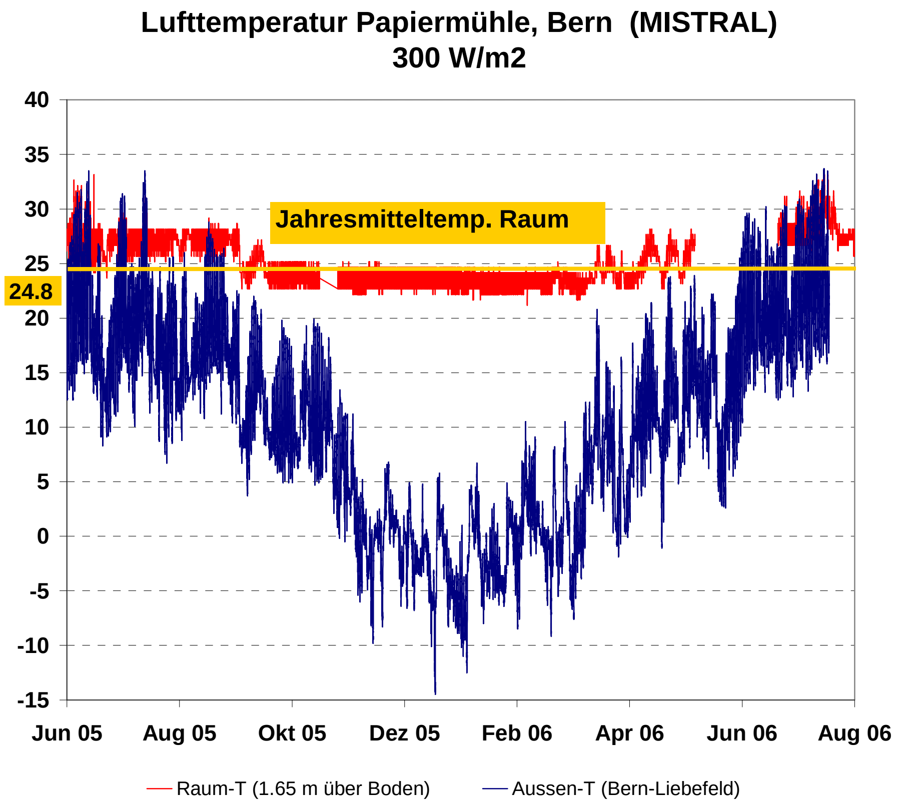

Figure 4. Graph of annual temperature at the Bern plant.

Table 2. Sum frequency temp.

t_{Room} [°C]	Number of hours [h]	Percent [%]
>34	0	0
33	21	0.3
32	37	0.5
31	67	0.9
30	113	1.5
29	151	2.0
28	869	11.6
27	744	10.0
26	457	6.1
25	701	9.4
24	1800	24.1
23	2359	31.6
22	150	2.0
<21	0	0

The sum frequency table for the room temperature using the example of the paper mill in Bern illustrates the distribution throughout the whole year. Over 55% of the hours in the year are at a temperature level of 23-24°C. Temperatures above 30°C account for just 3.2%. Therefore when we talk about “temperature peaks” we mean just a few hours per year. The room air temperature hardly, if ever, exceeds the outside air temperature. It is therefore clear that the optimum thermal load extraction was not achieved but the building mass helps to keep the levels very low.

Readings in a lightweight construction cabin with 108 W/m²

Diagram 4 below shows a Swisscom Mobile outside unit in Lavigny (Vaadt). The room air mean temperature in the Summer months is 27.1°C, the figure throughout the whole year is approx. 23.5°C.

The mobile rooms differ from the Fixnet rooms in terms of their construction. The mobile cabins have virtually no building mass. The result of this is that there is almost no activation effect by the building mass and so the temperatures are slightly higher than at the solid construction locations. However, since the hotspot extraction was able to be implemented to better effect here the levels achieved are once again very satisfactory.

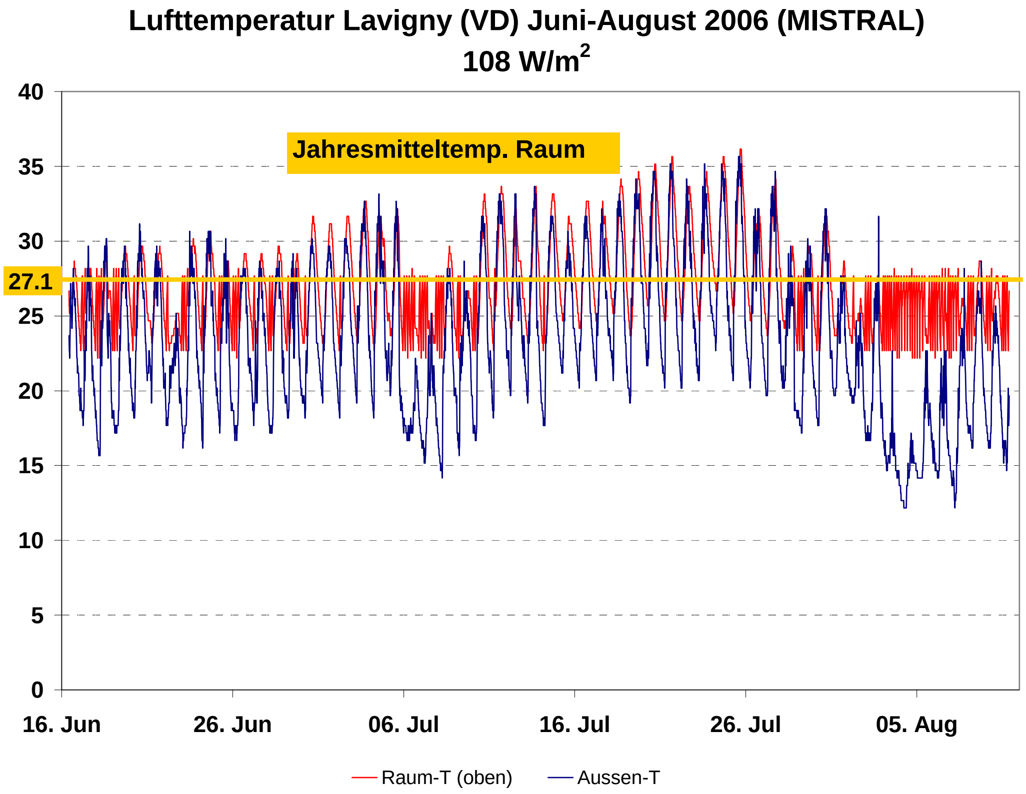

Figure 4. Graph of 3 months temperature
Mistral mobile

Operating experiences in Summer 2006

An independent temperature monitoring system is fitted in all Swisscom Fixnet's operating control rooms. In the hot Summer of 2006 the incoming temperature error messages were recorded and statistically evaluated over a period of three months. The task was to determine the relationships and frequency between the new and original systems on the basis of these error messages. 144 alarms were received, 10 of which were triggered by new systems. These 10 alarms occurred in a total of 4 rooms and were able to be traced back to incorrect thermal load data or operator error. There was therefore clear evidence that the system reliability is very high and is functioning properly.

Rooms with thermal loads up to 680 W/m2 had been previously implemented successfully using this system.

Humidity

Under the guideline (see Figure 1) the room humidity requirement of 5-85% relative h. must be met. The upper figure is quite safe since the air in the room is warmed up (except when outside temperatures are high) and so no relative humidity can be achieved in this area. On hot days the outside air humidity in Switzerland is never above 70% so here again, as the readings demonstrate, the upper figure can never be achieved.

What appears more critical is not to fall below the 10% relative humidity. The Mistral Team tested this at one of the highest telecommunications centre in Switzerland on the Säntis, 2503 metres above sea level. The readings in the room and all others which have been measured show figures of min. 10% relative humidity and have therefore never fallen below the lower limit.

Summary of the results

The field readings and experiences prove that the telecommunication control rooms can be successfully cooled to a functional temperature level without any problems using mechanical ventilation.

At annual mean temperatures of 24-25°C this can be easily maintained, as can the specifications in accordance with ETSI 3.1. The peak temperatures are limited to a minimal period and are clearly below the limits set by the ETIS standard. The requirements in terms of room humidity are met without exception.

There were no climate-related failures of the telecom systems in the 2006 hot Summer.

KEY FIGURES

A number of key figures should illustrate the potential that lies in this cooling system for telecommunication control rooms:

The imminent replacement investment for the outdated cooling systems for the whole of Switzerland, without Mistral, is € 180 million. This also means savings on maintenance in the sum of € 2 million and energy costs in the sum of € 4 million.

- 80% lower investment costs than conventional cooling systems (for new Systems)
- 80% lower maintenance costs
- 95% lower electricity costs

The total energy consumption per annum will be reduced by 70GWh after full implementation at existing and new planned sites at Swisscom.

OUTLOOK

Telecom operators in other European countries have already expressed significant interest in the new system. Belgacom from Belgium is already in the process of planning pilot projects. This publication is intended to draw the attention of specialists in the field of air-conditioning to this subject. The market potential in Europe alone is enormous and the potential gain for the environment more than considerable.

DISCUSSION

The impressive features of the system are not only its simplicity but its economic and ecological advantages. Combined with the improved availability and low maintenance operation, it is a showpiece of simulation-aided modern planning results.

The publication of this solution is however also intended to initiate a process of discussion and review. Many experts are still insufficiently familiar with the opportunities offered by natural cooling, many are unable to envisage the functionality of the simple solution.

However, the range of uses for this simple solution is not limited purely to cooling telecommunication installations. Depending upon the outside climatic conditions we can envisage being able to achieve the same levels as those in a computer centre. Faced with the knowledge gained from these results the IT equipment manufacturers should consider how much is needed that their items of equipment also can be cooled by using this system. Computer centre operators and the environment will feel the result of this change.

ACKNOWLEDGEMENT

We would like to take this opportunity to express our thanks to Swisscom for their active support and providing the resources for the realisation of this project.

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Development & Analyses of Prototype Thermoacoustic Refrigeration System

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ABSTRACT

Understanding the nature and thermodynamic behavior of sound waves and the attempt to harness the effect as a useful heat engine is called Thermoacoustic's. Thermoacoustic refrigerators utilize no environmentally hazardous gases. They use inert gases which are both readily available, inexhaustible, and completely environmentally benign! And, because they can utilize waste heat as an energy source, they are extremely environmentally friendly. In addition, since thermoacoustic refrigerators do not use the compressors, lubricants, sliding seals and after gismos present in vapor compression refrigerators, the thermoacoustic refrigerator should more reliable.

An inexpensive demonstration model of thermoacoustic Refrigeration system is built from a loudspeaker unit (Ahuja) of 90W with glass tubing and aluminum plug. The digital thermometers were used for recording the temperature and its variation along the resonator tube. The results were achieved after running the system for 40-50 minutes and were analysed.

INTRODUCTION

Understanding the nature and thermodynamic behavior of sound waves and the attempt to harness the effect as a useful heat engine is called Thermoacoustic's. A thermoacoustic refrigerator uses the thermoacoustic effect to remove heat with sound. A thermoacoustic prime mover uses heat to create sound. Thermoacoustic effect is the conversion of heat energy to sound energy or vice-versa. Utilizing the thermoacoustic effect, refrigerator can be developed that uses Acoustic energy as a source to remove heat with no moving parts.

In the mid 1990's the production of CFC's refrigerants (Freon mainly) was banned by the Montréal protocol. The CFC's are the major player in the depletion of ozone layer. In addition, CFC's are nasty green house gases. While the recent HFC and HCFC replacements are less harmful, they will still be major contributors to the greenhouse effect and there is concern over possible health hazards for many of the newer chemicals. In addition, since both HFC's & HCFC's are expected to be banned early in the 21st century, an entirely new refrigeration technology needs to be developed.

Thermoacoustic refrigerators utilize no environmentally hazardous gases. They use inert gases which are both readily available, inexhaustible, and completely environmentally benign! And, because they can utilize waste heat as an energy source, they are extremely environmentally friendly. In addition, since thermoacoustic refrigerators do not use the compressors, lubricants, sliding seals and after gismos present in vapor compression refrigerators, the thermoacoustic refrigerator should more reliable.

An inexpensive demonstration model of thermoacoustic refrigeration system is built from a loudspeaker unit (Ahuja) of 90W with glass tubing and aluminum plug. The digital thermometers were used for recording the temperature and its variation along the resonator tube.

Principle of working

The thermoacoustic refrigeration cycle is illustrated in Fig. 1 & 2. As the motion of the sound wave causes a gas parcel in the stack to move left towards the closed end of the tube. The pressure increases and the gas is compressed. The compressed gas parcel is now hotter than the nearby stack wall so it dumps heat to the cooler stack, thus shrinking in volume. As the standing wave continues through its cycle the parcel is pulled back to the right where the pressure is lower. The rarefied parcel is now cooler than the nearby stack wall so it absorbs heat from the warmer stack wall and expands. The cycle repeats with the net effect of a small amount of heat being moved a short distance along the stack from the colder towards the hotter end. Thus significant amount of heat is moved from one end of the stack to the other.

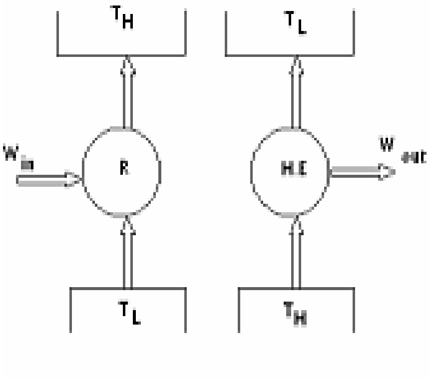

Fig: 1 Schematic showing basic actions of
(A) Heat Pump or Refrigerator,
(B) Heat Engine or PrimeMover.

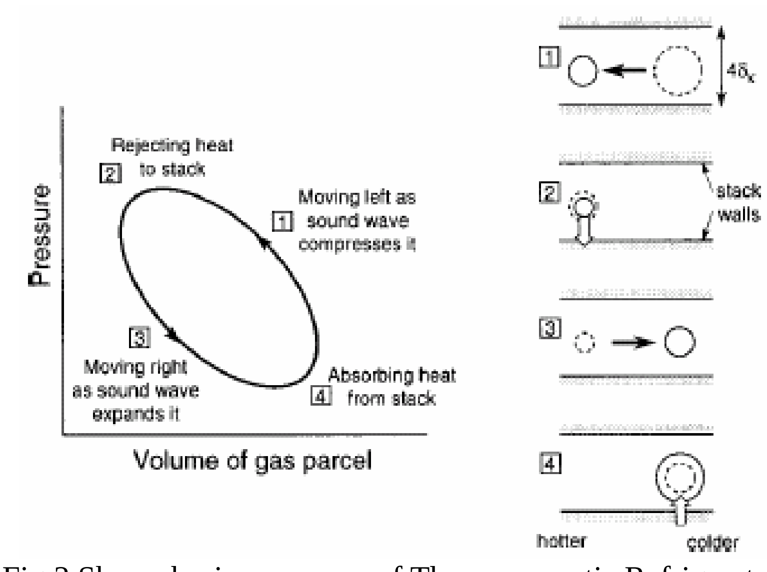

Fig:2 Shows basic processes of Thermoacoustic Refrigerator.

Literature Review

Advances in the development of thermoacoustic engines and acoustical compressors require an understanding of extremely intense sound fields (acoustic Mach numbers of order one) in resonators that are not cylindrical. Nonlinear effects such as harmonic generation, waveform distortion, and nonlinear frequency shift are to be investigated with a one-dimensional model[5]. All properties of the flow field throughout the resonator are to be calculated numerically. To estimate energy losses in the resonator, we have to consider the gas flow in the boundary layer along the walls of the resonator in the linear approximation and match it to the nonlinear solution in the volume. Energy loss due to turbulence is to be evaluated by introducing a model for eddy viscosity.

An analytical study is to be conducted of the thermoacoustic effects induced by the interaction of a strong acoustic field with a rigid boundary such as that in a thermoacoustic engine. With the sphere as a representative object, it has been found that the acoustic field can create a spatially periodic heating and cooling pattern on its surface just as in the stack of a thermoacoustic engine[10]. The thermoacoustic effects are generated primarily in the narrow Stokes boundary layer region on the sphere and are diffused and convected over the remaining part of the fluid domain.

Pressure oscillations in a sound wave is accompanied by temperature oscillations. In the presence of a solid boundary, the heat transfer from the oscillating gas to the solid boundary causes dissipation of the acoustic energy. This results in the attenuation of the sound wave[3]. This thermal-relaxation dissipation process has a negative effect on the performance of thermoacoustic heat pumps and engines. A simple analytical model describing the interaction between an acoustic wave and a solid boundary is presented. The effect of the solid material and gas type on thermal-relaxation dissipation is analysed. The main result of this model is that the choice of a solid material with the smallest possible heat capacity per unit area in combination with a gas with the largest possible heat capacity per unit area minimises the thermal-relaxation dissipation. From the different combinations solid-gas used in the calculations, the combination cork-helium leads to the lowest thermal attenuation of the sound wave. In this case, the heat transfer from the gas to the wall less damps the temperature oscillations. However, because of the porosity of cork that may cause some problems, it is suggested that the combination of polyester-helium can be used in practice to minimize the thermal-relaxation losses.

Thermoacoustically driven thermoacoustic refrigerator powered by solar thermal energy has been successfully built and tested by Jay A Adeff[14]. A 0.457 m diameter Fresnel lens focuses sunlight onto the hot end of a 0.0254 m diameter reticulated vitreous carbon prime mover stack, heating it to 475°C, thereby eliminating the need for the most troublesome component in a heat driven prime mover, the hot heat exchanger. The high intensity sound waves produced by the prime mover drive a thermoacoustic refrigerator to produce 2.5 watts of cooling power at a cold temperature of 5°C and a temperature span of 18°C.

Experimental set-up

The thermoacoustic refrigerator demonstration model described in this paper is of the standing wave variety, and consists of a half/quarter-wavelength resonator with an open-closed tube, driven by a loudspeaker unit of 90W. This is the easiest resonator shape to build, Since the primary purpose of this apparatus is to demonstrate the action of an acoustic refrigerator.

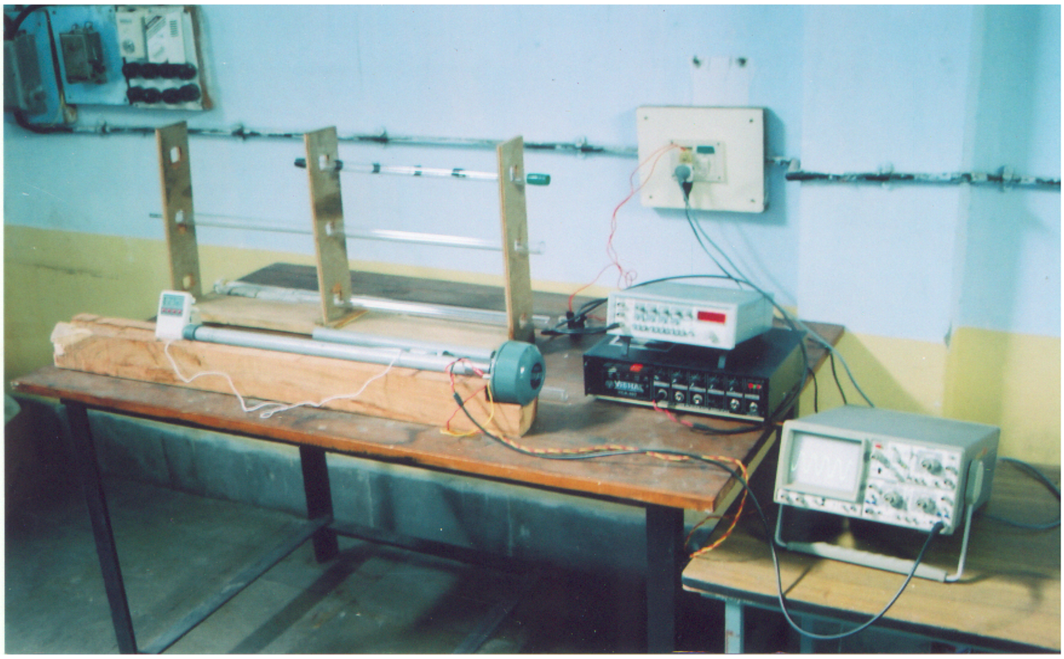

Fig:3 Experimental Set-up. Acoustic Driver, Amplifier, Function Generator, Cathode Ray Oscilloscope, Resonance Tube and digital thermometers etc.

The resonator for this refrigerator was made up of a 1m length of glass tubing with an inner diameter of 20 mm. The length defines the resonance frequency of the system. The loudspeaker (Ahuja Unit) was driven by a frequency generator producing a sine wave through a 90W audio amplifier and a 1.2-inch diameter o-ring was used to provide a seal around the edge of the speaker. An aluminum plug was milled to fit snugly into the end of the tube, forming the closed end. The resonator for this refrigerator was made up of a 1m length of glass tubing with an inner diameter of 20 mm. The length defines the resonance frequency of the system. For recording the frequency and amplitude of the wave, Cathode Ray Oscilloscope was used. Digital thermometers were used for recording the temperature at each node – antinodes points. The pressure amplitude inside the resonator tube was not measured but was obtained following the ideal gas equations. The power to the speaker was varied and increased until a second harmonic became audible indicating that the system was becoming non linear. The most important part of an acoustic refrigerator is the stack, which consists of a large number of closely spaced surfaces aligned parallel to the length of the resonator tube. The stack for this apparatus was constructed from photographic film around a central spindle so that adjacent layers of the spirally wound film provide the stack surfaces, so that air could move between the layers along the length of the stack parallel to the length of the resonator tube. If stack layers are too far apart the gas cannot effectively transfer heat to and from the stack walls. If the layers are too close together viscous effects hamper the motion of the gas particles.

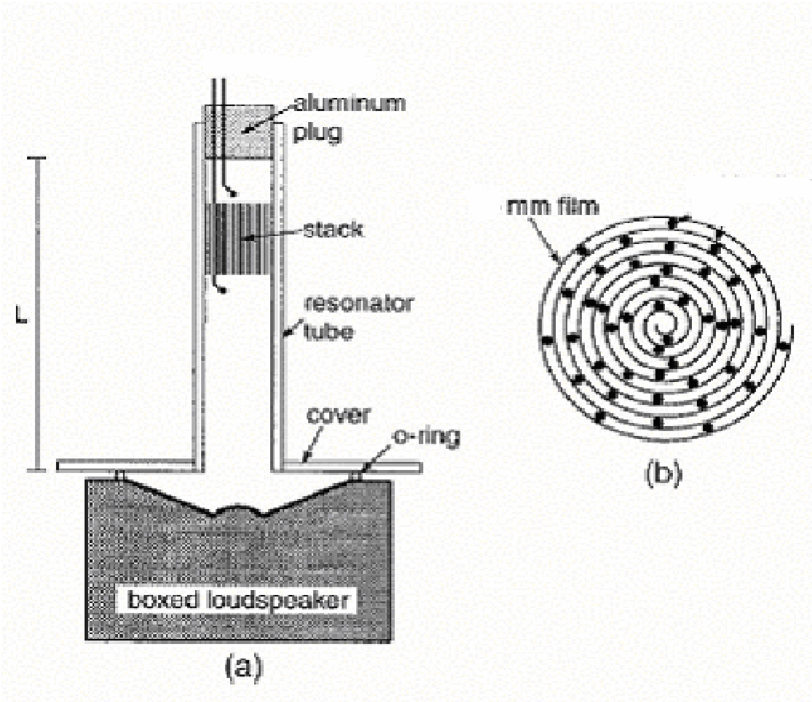

Fig:4
(a) Schematic diagram of the prototype of Thermoacoustic Refrigerator model,
(b) Cross section of the stack showing how the film layers were separated.

Experimentation

Initially water was poured in the tube and the loudspeaker was turned on by setting the frequency of the acoustic wave. The working substance used for this setup is the inert gas called air, which is alternatively compressed and expanded in the tube due to generation of the sound waves. The acoustic waves which are longitudinal waves, standing waves are set in the resonator tube. Standing waves causes the gas particle to oscillate back and fourth parallel to the walls of the resonator tube. The alternating compression and rarefaction of the gas causes the local temperature of the gas to oscillate due to adiabatic nature of sound waves. As there is alternatively compressions and rarefactions existing in the resonator tube the temperature difference is obtained due to the pressure amplitude of the of the acoustic wave. As there is 90degree phase lag between the oscillations of the pressure and velocity inside the tube, the water poured in the tube swirls at the junction of acoustic pressure amplitude (antinodes).

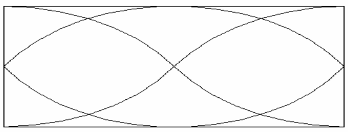

Fig:5 Pressure & Velocity, variation in the resonance tube.

Thus heat is transferred to the water medium and the gas gets cooler causing refrigeration effect. The temperatures were recorded before and after the points of water swirl i.e. at the nodes and antinodes of the acoustic waves. It was observed during experimentation the maximum temperature difference of 22^{o} was achieved and the lowest temperature achieved with the demonstration model was found to be 14^{o}.

The results obtained while experimentation, were plotted and the nature of the curve has been studied.

The Fig.6 shows the comparative study of the temperature variation in the resonance tube along the length at various frequencies. It is observed that maximum temperature is attained at the low frequencies, since the wavelength of the acoustic wave is longer and the occurrence of the nodes and antinodes is clearly defined.

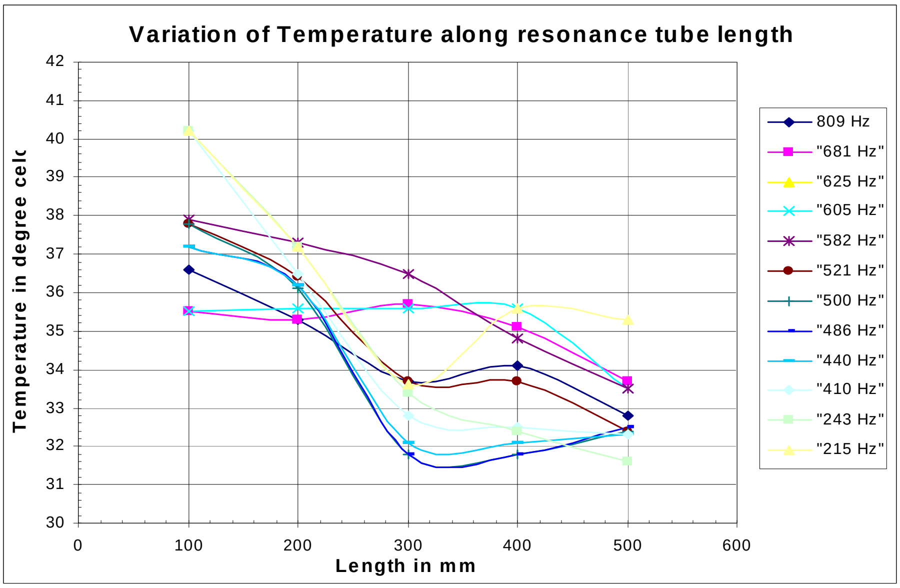

Fig: 6.

The Fig.7 gives the comparative study of the maximum & minimum temperature variation in the tube indicating room for the refrigeration effect. The maximum & minimum temperature in the tube occurs in the frequency range of 200-400Hz and this is the range one has to concentrate his studies for the achievement of the fruitful results for causing refrigeration effect.

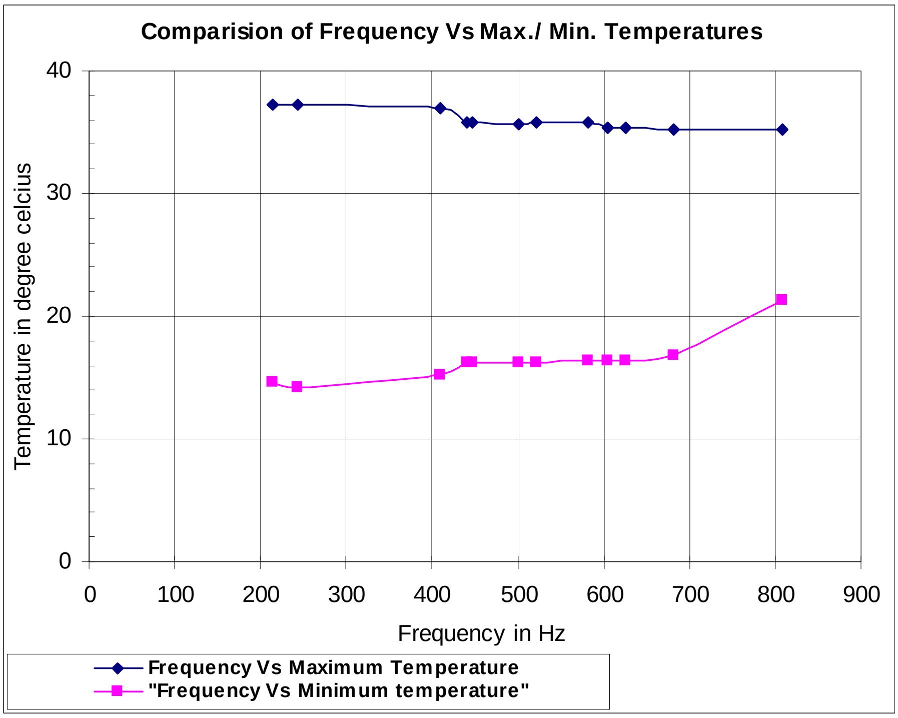

Fig:7.

CONCLUSION

The Experimental data implies that the frequency of acoustic wave is of great concern in the performance of the thermoacoustic refrigerator. The frequency range of 200 – 400 Hz. Could be used to achieve the commercially viable unit. The results obtained in this paper with the experimental setup developed can be used as a guideline for further development of the system in this field.

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Thermal Flux Sampler for Onsite Performance Evaluation of VRV System

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SUMMARY

This paper reports an alternative technique to evaluate onsite performance of Variable refrigerant volume (VRV) multiple indoor units connected air conditioning systems. The newly-devised Thermal Flux Sampler gathers and merges the blowing flow at multiple measuring points on the exhaust grille of outdoor unit into a single typical point using a sampling tube. The system coefficiency of performance is calculated from the amount of heat discharge and the electric power consumption of compressor and whole system. Because enthalpy and airflow velocity are greatly different in each part of exhaust grille face, sampling air is collected by tubes from twenty points on the grille face, and representative enthalpy and airflow velocity are measured at the merging point. The practical operating conditions of the VRV system installed in a certain building were investigated by using this technique. Though the investigation period was one week of summer, there was much time when the VRV system was operating by the very low load overwhelmingly.

INTRODUCTION

VRV systems of the multi-unit type especially are now showing remarkable popularity. VRV systems used to replace central air-conditioning systems in small-scale buildings and be added to compensate for inadequate central air-conditioning systems. Recently in Japan, however, VRV systems have been applied even to main air-conditioning systems in office buildings having total floor areas of the 100,000 m^2 class and are replacing central air-conditioning systems. Central air-conditioning systems were improved by research and development to realize an indoor environment of higher accuracy. Meanwhile, VRV systems have gradually achieved performance equivalent to central air-conditioning systems while featuring easy design and construction, individual operability, and individual controllability. By gaining complicated functions through this process, VRV systems became a “black box” beyond the understandings of facility designers and users.

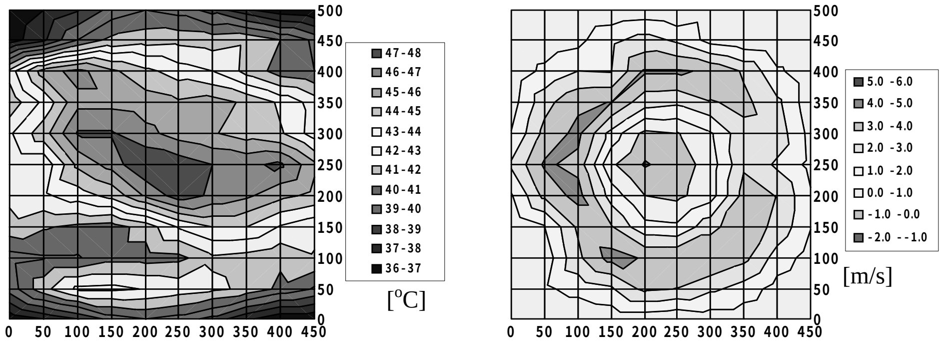
Also, in evaluating the energy performance during actual operation, it is very difficult to understand VRV behaviors because each indoor unit shows complicated operation. A bench test in a laboratory is also difficult. Therefore, we cannot avoid saying that VRV is a “black box” in terms of not only functionality but also performance. To solve this problem, the authors developed a technique of easily evaluating VRV behaviors in actual operation as an outdoor unit that doesn’t require entering a room. This paper introduces the basic concept of this evaluation technique and the results of an experimental application to a VRV system in operation.

In recent Japan, where VRV systems are spreading remarkably, we have been seeing many studies to verify various performance factors. The authors compared the evaluation technique of measuring the blowing flow from an indoor and outdoor unit by transverse equipment with a heat balance technique using a calorie box [1].

METHODS

Properties of blowing flow from outdoor unit

Figure 1 shows the temperature distribution and the airflow velocity measurement of the blowing flow from the outdoor unit on the exhaust grille face. Since the blow-off air is the swirling flow appearing immediately after the axial flow fan, the scalar wind velocity, vector wind velocity, and air temperature are not uniform. Therefore, the conventional technique of measuring typical points by inserting a rectification duct into the exhaust grille face has low accuracy of measuring the amount of heat discharge. To accurately calculate the amount of heat discharge from the outdoor unit, therefore, it may be necessary to measure the exhaust at multiple points of the exhaust grille simultaneously.



a) b)

Figure 1. a) Temperature distribution and b) airflow velocity on exhaust grille face.

Configuration of thermal flux sampler

It is very difficult to maintain many anemometers outdoors for a long time because of the economic burden and equipment weather resistance. To solve these problems, the authors attempted to realize simple and long-time continuous measurement by devising a thermal flux sampler that gathers and merges the blowing flow at multiple measuring points on the exhaust grille into a single typical point using a sampling tube.

Figure 2 shows the configuration of the thermal flux sampler. The thermal flux sampler consists of 20 sampling tubes, an interval adjustment plate, and a merging section. By moving the interval adjustment plate, the sampling points can be rearranged according to the size of the exhaust grille. The choke valve is installed at each sampling tube to adjust the sampling air volume.

Heat discharge from outdoor unit

When the thermal flux sampler is used, the amount of heat discharge from the outdoor unit to the atmosphere can be calculated by Equation 1.

$$Q = \gamma \times C_p \times v'' \times S'' \times (t'' - t_0) \times \alpha \times \beta \times \sin \overline{\theta} \times \frac{x_t}{x''}, \qquad (1)$$

Where Q is the quantity of exhaust heat from outdoor unit [kW], γ is the specific weight [kg/m^3], C_p is the specific heat [kJ/kg K], v'' is the airflow velocity of merging section [m/s],

S'' is the area of merging section [m^2], t'' is the air temperature of merging section [K], t_0 is the suction air temperature of outdoor unit [K], α is the constant of airflow velocity ratio tuned by the choke valve, β is the constant ratio of representative area of grille face for cross section of sampling tube, $\overline{\theta}$ is the average composite angle of grille face and airflow [^{o}] (approximately 50^o with the following case), x_t is the total airflow volume of all sampling tubes [m^3/s], and x'' is the airflow volume of measuring point [m^3/s].
These details and the verification of the measurement accuracy were mentioned in another paper [2] of the poster session.

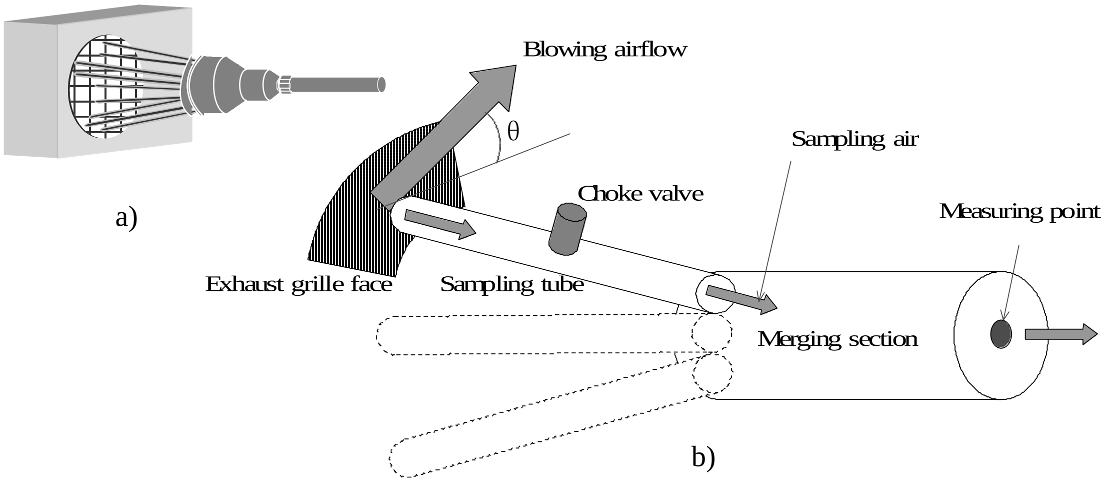


Figure 2. a) Configuration and b) Concept of Thermal Flux Sampler.

EXPERIMENTAL APPLICATION TO VRV SYSTEM IN OPERATION

Outline of object system

Using this thermal flux sampler, the authors evaluated a VRV system in operation. From August 31 until September 5, 2005, an actual VRV system was measured at a laboratory building of a university in the suburbs of Tokyo. Table 1 lists the system facilities. The VRV system is a heat pump for both cooling and heating and the compressor of the outdoor unit is controlled by an inverter. Seven indoor units are connected to the single outdoor unit. Since the experiment was conducted in summer, the VRV system was in cooling mode.

Table 1. System facilities.

Designation		Specifications
Electric Source		3Φ 200V
Refrigerant		R-22
Cooling capacity		56.0 W
Heating capacity		63.0W
Rated COP	Cooling	2.64
	Heating	3.44
Indoor unit		4-way ceiling mounted cassette unit

Measurement items

This VRV system had four fans in the outdoor unit. The thermal flux sampler was installed at the exhaust grille of each fan and the wind velocity and air temperature were measured at the

merging section (Figure 3). The power consumption was measured for the outdoor unit system and the indoor unit system. For the outdoor unit system, the power consumption by the compressor was also measured. The intake air temperature and the amount of solar radiation were also measured.
To calculate the amount of heat produced at the outdoor unit, the power consumption by the compressor was subtracted from the amount of heat discharge measured by the thermal flux sampler. To calculate the system COP, the amount of heat produced by the outdoor unit was divided by the power consumption of both the outdoor and indoor units.

Figure 3. Thermal flux samplers mounted on VRV system.

RESULTS

Figure 4 shows the outdoor air temperature and the amount of solar radiation in the experimental period. The weather was generally fair throughout the experimental period. From the evening of September 4 until September 5, a typhoon caused torrential rain.
Figure 5 shows the load factor appearance frequency. The load factor was calculated with the rated cooling performance of the outdoor unit as 100%. Despite the great cooling load in summer, the load factor rarely became 50%, and usually was up to 20%.
Figure 6 shows a change of the system COP at every five minutes of system operation during the experimental period. Figure 7 shows the system COP by day. The poor system COP of this VRV system is partially attributable to the power consumption of about 300 W in the standby state.
Figure 8 shows the load factor appearance frequency and the system COP in the experimental period. Where the load factor is low, the system COP is also low.

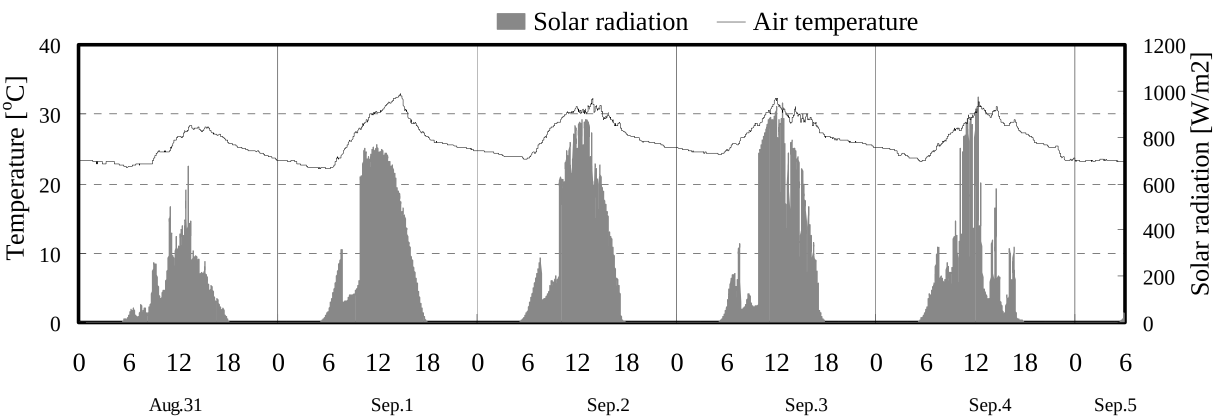


Figure 4. Outdoor air temperature and amount of solar radiation in experimental period.

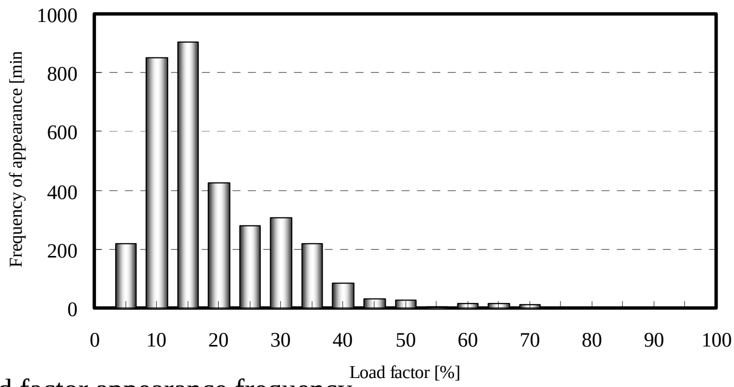


Figure 5. Load factor appearance frequency.

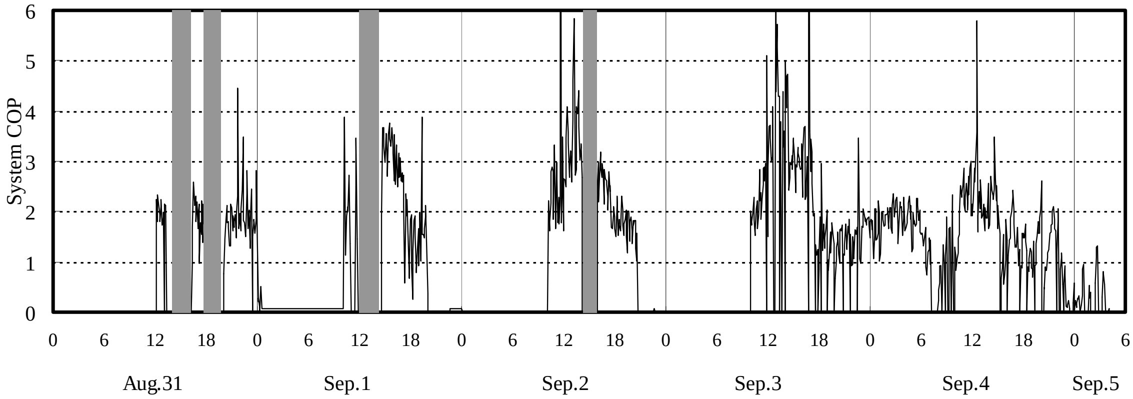


Figure 6. System COP at every five minutes.

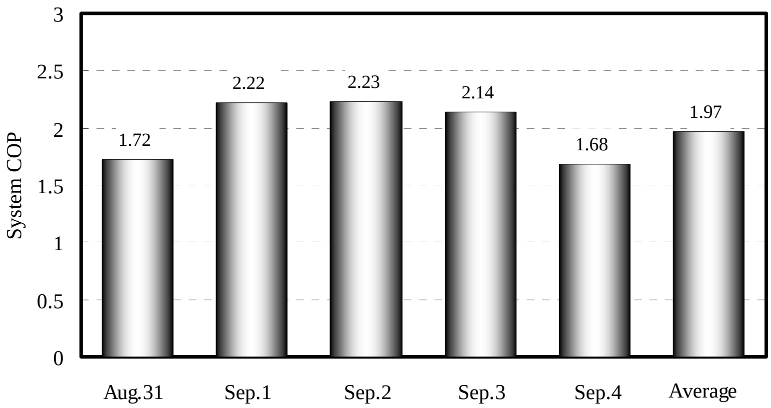


Figure 7. System COP by day.

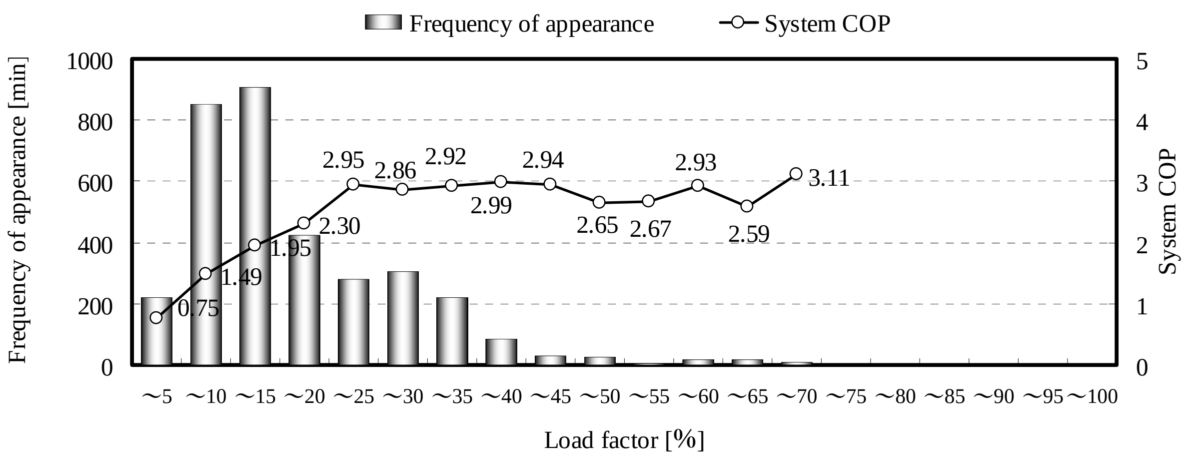


Figure 8. Load factor appearance frequency and system COP.

DISCUSSION

The authors developed a technique of continuously measuring VRV system operation status easily by measuring the amount of heat discharge from the outdoor unit with the thermal flux sampler. The results of an experiment using the sampler, however, clarified several points.

One result is that the sampler is susceptible to external wind velocity. At the end of the sampler merging section, a spherical windshield is attached to reduce the influence of dynamic pressure by the outdoor wind velocity. For stable measurement, however, further improvement is necessary.

The second problem is about the power consumption measurement. To measure the power consumption by the compressor of the outdoor unit, a precise watt-hour meter is necessary on the secondary side of the inverter. We need to consider the logic of estimating the power consumption by the compressor from that by the entire outdoor unit.

The VRV system evaluated using the thermal flux sampler was found to have a very small load factor despite the cooling peak in summer. Compared with central systems, unitary systems tend to have excess facility capacities. Therefore, VRV systems are expected to show no COP decrease even under extremely small loads.

ACKNOWLEGEMENT

The authors gratefully acknowledge Prof. S. Kametani of Tokyo University of Marine Science and Technology for his support.
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Accuracy Verification of Thermal Flux Sampler for Onsite Performance Evaluation of VRV System

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SUMMARY

This paper reports the accuracy of a measurement device for evaluating the actual performance of a variable refrigerant volume (VRV) system. A newly devised thermal flux sampler can be installed on an outdoor unit easily and measure the exhaust heat quantitatively. An experiment was conducted in a room-type calorimeter chamber to verify the accuracy of the thermal flux sampler. The accuracy was verified by measuring the energy efficiency of the test unit by three measuring methods: the heat balanced method, the improved duct inserted method and the sampler method. In a cooling condition, the average mean error of the sampler method and the improved duct inserted method was 12% and 19%, respectively, in comparison to the results of the heat balanced method.

INTRODUCTION

It is important to understand the actual performance of VRV (variable refrigerant volume) multiple indoor units connected to an air-conditioning system (herein referred to as the “VRV system”) to optimize its practical operation. The VRV system in air-conditioning systems is noted for its easy installation and convenience. However, the on-site performance of the VRV system is not clear, because the performance is evaluated in a static condition according to Japan Industrial of Standard (JIS) rules. Thus, it is necessary to investigate a practical evaluation method of the VRV system in actual operating conditions. Therefore, a technique to evaluate on-site performance was examined and a sampling system for the quantity of exhaust heat from an outdoor unit was developed. In this paper, an experiment was conducted in an environmental chamber to verify the accuracy of the newly devised thermal flux sampler.

CONCEPT OF THERMAL FLUX SAMPLER

Air distribution of outdoor unit

Figure 1 shows the air distribution on the air outlet of an outdoor unit. The air distribution was investigated in detail and yielded the following results [4], [7]:

1. The exhaust air temperature and airflow are greatly different in each part of the air outlet of the outdoor unit.
2. The airflow distribution of the outdoor unit is concentric circular.
3. The airflow is exhausted as rotational flow. Thus, it is necessary to consider the vertical angle of the airflow.

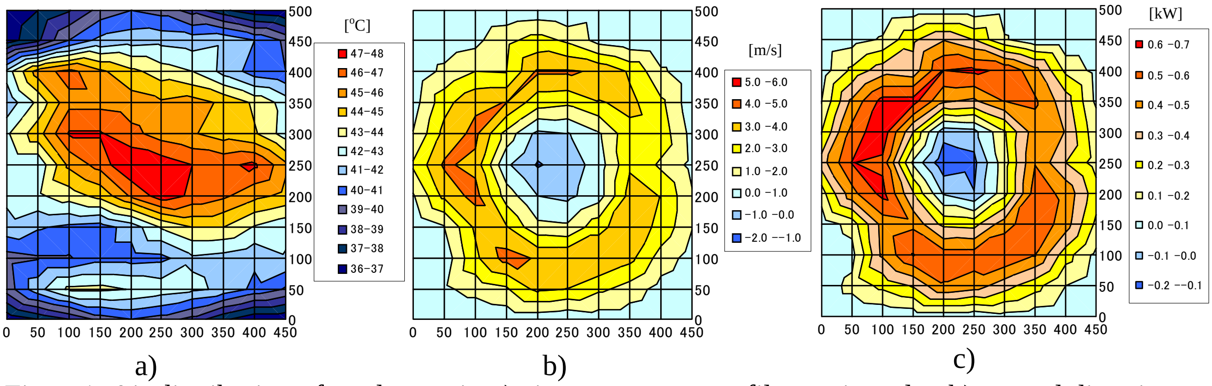


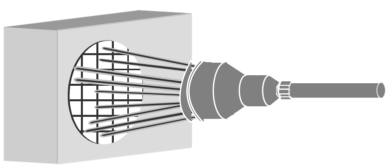
Figure1. Air distribution of outdoor unit: a) air temperature profile on air outlet, b) normal direction profile airflow on air outlet, c) exhaust heat profile on air outlet

Characteristics of thermal flux sampler

The thermal flux sampler is a sampling device that samples the exhaust heat from an outdoor unit. An illustration of the thermal flux sampler is shown in Figure 2. The quantity of exhaust heat is identified from the thermal flux of the measurement point in the merging section of the thermal flux sampler. Since the air distribution is greatly different in each part of the air outlet, the exhaust airflow is measured in twenty parts. The sampling air is collected by conduits and the representative air temperature and airflow velocity are measured in the merging section.

Calculation formula for the thermal flux sampler

The exhaust heat calculation formula for the thermal flux sampler is shown in Equation (1). The thermal flux of the representative area was calculated in Equation (2).The quantity of exhaust heat from outdoor unit is the total of the thermal flux of representative area (Equation (3)).The thermal flux of each conduit is shown in Equation (4).Thus, Equation (5), the quantity of exhaust heat from outdoor unit is deduced from Equation (2), (3) and (4). Furthermore, v_n / v_n' becomes a constant by adjusting the ratio of airflow velocity from the representative area to outlet of each conduit. Moreover, S_n / S_n' becomes a constant by dividing the measurement area equally. If the heat loss of airflow that passes conduits is negligible quantity, t_n and t_n' are equal and the ratio of $(t_n - t_o)$ and $(t_n' - t_o)$ is a constant. Furthermore, v_n / v_n' and S_n / S_n' are replaced with α and β. Thus Equation (5) is substitute for in Equation (6). Where $\bar{\theta}$ is the average of composite angle grille face and airflow. The thermal flux of the measurement point in the merging section is calculated in Equation (7).It is the total of thermal flux of all conduits. This is shown in Equation (8).The quantity of exhaust heat from outdoor unit is deduced from Equation (6), and (8). Moreover, Equation (1), the quantity of exhaust heat by using this method is deduced to compensate the air volume reduction in process of merging section from inflow entrance of conduits.



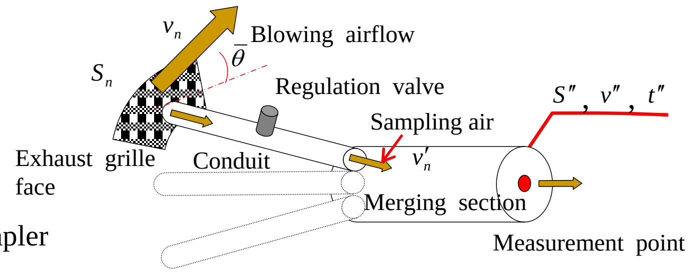


Figure2. Illustration of the thermal flux sampler

$$Q = \gamma \times C_p \times v'' \times S'' \times (t'' - t_o) \times \alpha \times \beta \times \sin\bar{\theta} \times x_t / x'' \quad (1)$$

$$q_n = \gamma \times C_p \times v_n \sin\theta_n \times S_n \times (t_n - t_o) \quad (2)$$

$$\sum_{n=1}^{20} q_n \fallingdotseq Q \quad (3)$$

$$q'_n = \gamma \times C_p \times v'_n \times S'_n \times (t'_n - t_o) \quad (4)$$

$$Q = \sum_{n=1}^{20} q_n = \sum_{n=1}^{20} q'_n \times \frac{v_n \sin\theta_n \times S_n \times (t_n - t_o)}{v'_n \times S'_n \times (t'_n - t_o)} \quad (5)$$

$$Q = \alpha \times \beta \times \sin\bar{\theta} \times \sum_{n=1}^{20} q'_n \quad (6)$$

$$q'' = \gamma \times C_p \times v'' \times S'' \times (t'' - t_o) \quad (7)$$

$$q'' = \sum_{n=1}^{20} q'_n \quad (8)$$

γ : Specific weight [kg/m3]

C_p : Specific heat [kJ/kg K]

v'' : Airflow velocity of merging section [m/s]

S'' : Area of merging section [m2]

t'' : Air temperature of merging section [K]

θ_n : Composite angle of grille face and airflow [°]

v_n : Airflow velocity of measurement area [m/s]

v'_n : Airflow velocity of each conduit outlet [m/s]

S_n : Representative area for measurement point [m2]

S' : Cross section of each conduit [m2]

v'' : Airflow velocity of merging section [m/s]

x_t : Total air volume of all conduits [m3/s]

x'' : Air volume of merging section [m3/s]

t_0 : Suction air temperature of outdoor unit [K]

Q : Quantity of exhaust heat from outdoor unit [kW]

q_n : Thermal flux of representative area [kW/m2]

q_n : Thermal flux of each conduit [kW/m2]

q'' : Thermal flux of merging section [kW/m2]

Components of thermal flux sampler

Figure 3 shows the components of the thermal flux sampler. The thermal flux sampler is composed of conduits, two supporting plates of conduits and a merging section. The each conduit is fitted into a regulating valve to adjust the pipe flow velocity. The measurements points of the airflow and air temperature are shown in Figure 4. The representative area reflects the vertical airflow profile and was divided into concentrical and equal circles. The supporting plates are equipped to keep the position of these measurement points. Moreover, it enables to correspond to various fan sizes of outdoor units by adjusting the mutual distance.

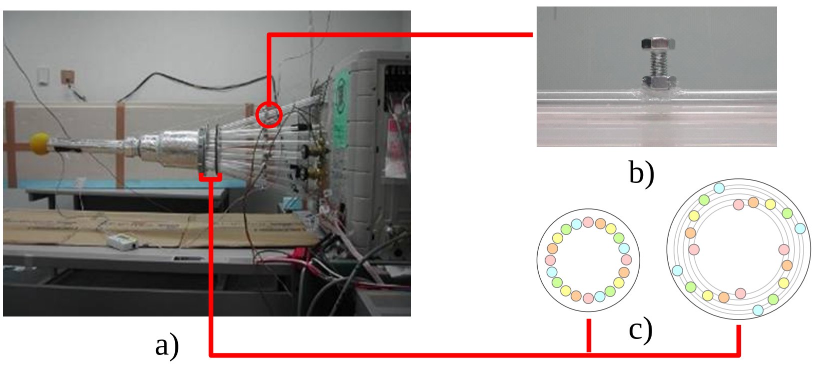


Figure3. Components of the thermal flux sampler: a) overall view, b) regulation valve of the conduits, c) supporting plates of the conduits.]

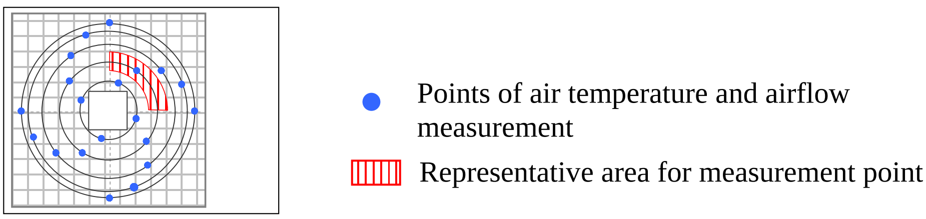


Figure4. Points of air temperature and airflow measurement

Experiment apparatus

Figure 5 shows the calorie meter chamber and position of the test unit. The experiment was conducted in a room-type calorimeter chamber owned by Kogakuin University. Table 1 shows the specification of the calorie meter chamber. It has a pair of chambers (room A and room B) which has glass windows on the south side. There is a thermal buffer zone that does not include the glass windows. The dimensions of the chamber are length 5.5 m, width 4.1 m, and height 5.9 m. Each chamber is equipped with an air-conditioning system to conduct a comparative experiment under different thermal conditions, and the thermal buffer zone is also equipped with six air-conditioners. Thus this chamber is able to supply a thermal load quantitatively by controlling the air-conditioning. In this experiment, the glass window on the south side was covered with outside shutters to exclude the influence of disturbance.
Table 2 shows the specification of the test unit. The test unit is a packaged air-conditioner, which is split type. In this experiment, it was experimented by alternately using the indoor unit: a 4-way ceiling-mounted cassette unit and a ceiling-suspended cassette unit. Thus, we installed the test unit in each chamber and used room A as the outdoor side and room B as the indoor side.

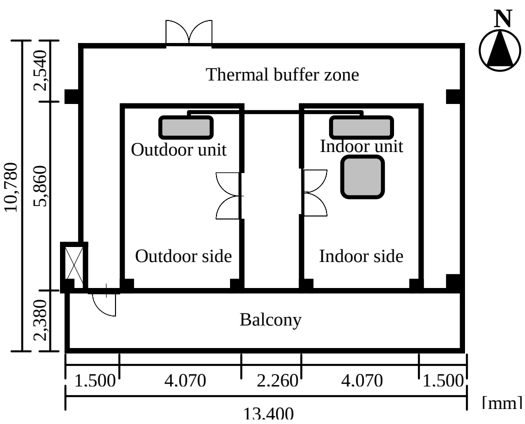


Table1. Specification of the calorie meter chamber

Designation	Specifications
Refrigerator	Air cooling chiller (45 kW)
Heater	Electric heater (40 kW)
Cool bath	FRP sandwich panel (3 cubic meter
Heat bath	Monocopue steel (2 cubic meter)
Outdoor air conditioning unit	Air volume (1000 cubic meter/h)
Air conditioning system for chambers	Air volume (1000 cubic meter/h)
Packaged air conditioner for thermal buffer zone	Multi type, 8 indoor units(5Hp)
Controllable temp.	-5 to 50 (^{o}C)
Controllable humidity value	10 to 100 (%)
Chamber size	5.9(L)×4.1(W)×5.5(H)
Wall heat transfer coefficient	Under 0.2 W/m2K
Thermal insulation	Polystyrene foam (200mm)

Figure5. Plan of the chamber and the position of the test unit

Table2. Specification of test unit

Designation		Specifications
Indoor unit		4-way ceiling mounted cassette unit /Ceiling suspended cassette unit
Electric Source		3 Φ 200 V
Refrigerant		R-410 A
Cooling capacity		3.60 kW
Heating capacity		4.00 kW
Rated COP	Cooling	3.99/4.73
	Heating	4.02/5.26

METHODS

Experimental conditions

The experimental conditions are shown in Table 3. There are fourteen experimental conditions which are based on the measurement conditions set by the JIS [1], [2] and [3] Table 4 shows the conditions of the supply air to the indoor side. It was decided to become the indoor side conditions of Table3 with the indoor unit operated. Thus, these values were calculated in consideration of the sensible heat factor (SHF) and the supplied thermal load to the indoor unit. The standard of the supplied thermal load to the indoor unit was decided based on the cooling and heating capacity of the outdoor unit.

Table3. Experimental conditions

type 1: 4-way ceiling mounted cassette unit
type 2: Ceiling suspended cassette unit

	Indoor side			Outdoor side			Operating condition	test unit
	DB(°C)	WB(°C)	RH(%)	DB(°C)	WB(°C)	RH(%)		
no.1	24.0	17.0	49.6	35.0	24.0	40.3	Cooling	type1
no.2	24.0	17.0	49.6	35.0	24.0	40.3		
no.3	24.0	17.0	49.6	20.0	12.0	37.6		
no.4	24.0	17.0	49.6	20.0	12.0	37.6		
no.5	27.0	19.0	47.0	35.0	24.0	40.3		
no.6	27.0	19.0	47.0	35.0	24.0	40.3		
no.7	24.0	17.0	49.6	35.0	24.0	40.3		
no.8	24.0	17.0	49.6	20.0	12.0	37.6		
no.9	27.0	19.0	47.0	35.0	24.0	40.3		type2
no.10	27.0	19.0	47.0	35.0	24.0	40.3		
no.11	24.0	17.0	49.6	35.0	24.0	40.3		
no.12	24.0	17.0	49.6	20.0	12.0	37.6		
no.13	22.0	17.0	60.9	7.0	6.0	86.8	Heating	type1
no.14	22.0	17.0	60.9	7.0	6.0	86.8		type2

Table4. Conditions of supply air to indoor side

	DB [°C]	WB [°C]	RH [%]	Air volume [m3/h]	Specific volume	Supply thermal load[kW]	SHF
no.1	34.0	20.6	29.2	1003	0.884	3.6	0.9
no.2	33.0	20.6	32.1	997	0.881	3.6	0.8
no.3	45.0	20.6	29.2	1003	0.884	3.6	0.9
no.4	33.0	20.6	32.1	997	0.881	3.6	0.8
no.5	33.7	21.4	33.4	1423	0.884	3.6	0.9
no.6	32.9	21.4	35.9	1416	0.883	3.6	0.8
no.7	30.3	19.4	35.7	1521	0.873	3.6	0.9
no.8	30.3	19.4	35.7	1521	0.873	3.6	0.9
no.9	33.7	21.4	33.4	1423	0.884	3.6	0.9
no.10	32.9	21.4	35.9	1416	0.883	3.6	0.8
no.11	30.3	19.4	35.7	1521	0.873	3.6	0.9
no.12	30.3	19.4	35.7	1521	0.873	3.6	0.9
no.13	15.2	14.2	90.0	1507	0.829	4.0	
no.14	15.2	14.2	90.0	1507	0.829	4.0	

Heat balanced method

The quantity of generated heat from the indoor unit was determined by the heat-balanced method. The heat balanced method is the way to determine the capacity of the test unit from the supplied thermal load when the generated heat from the indoor unit and the supplied thermal load to the calorimeter chamber are balanced. Thus, this method can identify and measure the generated heat from the indoor unit as the supplied thermal load to the calorimeter chamber. The supplied thermal load to the indoor side was calculated from the mean value of the supplied thermal load before the test unit operated and the supplied thermal load after the test unit operated. The supplied thermal load before the test unit operated was determined as the mean value of one hour before the test unit operates [6].

Improved duct inserted method

The quantity of exhaust heat from the outdoor unit was measured by the improved duct inserted method. We installed the rectangular duct on the air outlet of the outdoor unit and measured the quantity of exhaust heat. The quantity of exhaust heat was calculated from the airflow velocity and the difference of the exhaust air temperature and suction air temperature in the cooling condition. In the heating condition, the quantity of exhaust heat was calculated from the airflow velocity and difference of the exhaust air enthalpy and suction air enthalpy because a latent load is generated. Figure 6 shows the points of measurement of the outdoor unit. The exhaust air temperature was measured at three points in the rectangular duct, and the suction air temperature was measured at five points on the back side of the outdoor unit. The point of the airflow velocity measurement was determined by examining the airflow distribution on the air outlet of the rectangular duct [5].

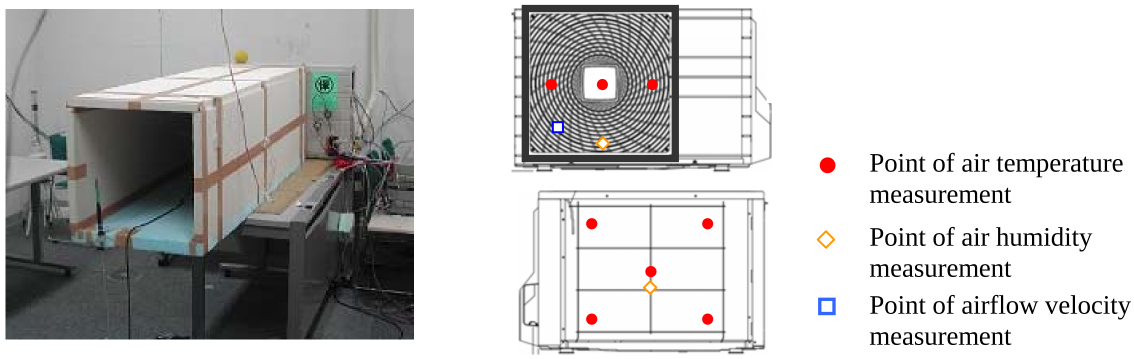


Figure6. The improved duct inserted method: a) experimental situation, b) points of measurement

Sampler method

Figure 7 shows the experimental setup of the Sampler method. The quantity of exhaust heat from the outdoor unit was measured by the Sampler method. We installed the thermal flux sampler on the air outlet of the outdoor unit and measured the quantity of exhaust heat. In the cooling condition, the quantity of exhaust heat was calculated from the airflow velocity, the exhaust air temperature and the suction air temperature. In the heating condition, the humidity of the exhaust air was measured and the quantity of exhaust heat was calculated from the airflow velocity, the exhaust air enthalpy and the suction air enthalpy.

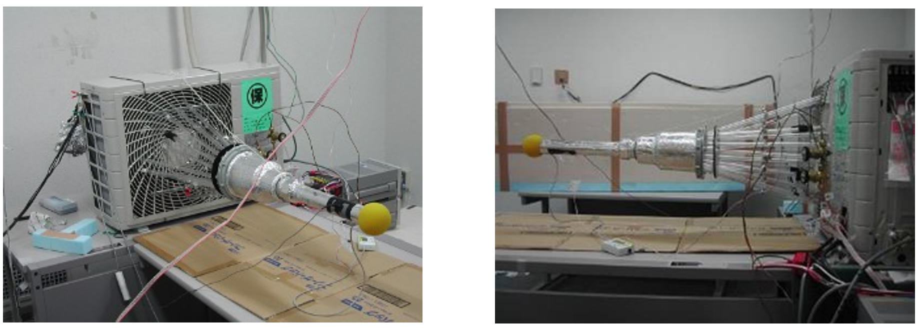


Figure7. Experimental setup

EXPERIMENTAL RESULTS

The accuracy of the thermal flux sampler was verified by comparison with the system coefficient of performance (herein called the “system COP”) that was measured by the three measuring methods, including the Sampler method. The experimental results of the three methods become approximately the same theoretically. The system COP is calculated from the quantity of generated heat from the indoor unit and the electricity consumption of the entire system. Thus, for the case of the Sampler method and improved duct inserted method,

the quantity of generated heat from the indoor unit is identified from the quantity of the exhaust heat and the electricity consumption of the compressor.
Figure 8 shows the system COP ratio based on the experimental results of the heat-balanced method. The system COP ratio is calculated with the experimental result of the heat balanced method as 100%. Figure 9 shows the average mean error of system COP ratio. The experimental results of the three kinds of methods were approximate. However, compared to the results of the heat balanced method, the average mean error of the Sampler method was 12% and that of the improved duct inserted method was 19% in the cooling condition.
In the heating condition, the experimental results are shown in Figure 10 as a reference, because the experiments were conducted in just one thermal condition.

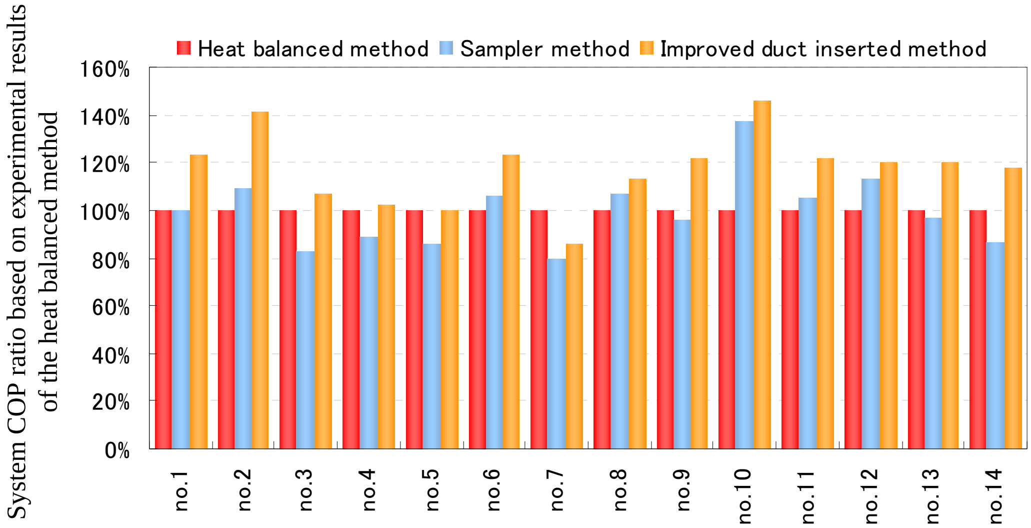


Figure8. System COP ratio based on experimental results of the heat balanced method

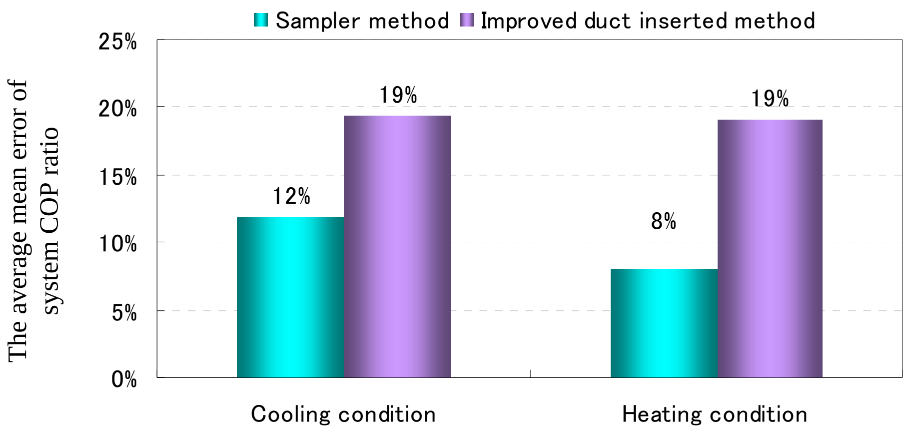


Figure9. The average means error of the measurement results based on the heat balanced method

DISCUSSION

In the cooling condition, in comparison to the results of the heat-balanced method, the average mean error of the Sampler method was 12% and the improved duct inserted method was 19%. Though the improved duct inserted method determined the point of airflow measurement by investigating the air distribution on the air outlet of the rectangular duct, the average mean error was 19%. This shows that it is difficult to measure the quantity of exhaust heat at a point of the representative airflow measurement. On the other hand, the average mean error of the Sampler method was 12% and it was shown that this method is effective.

ACKNOWLEDGEMENT
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Optimized Refrigeration Vapor Compression System for Power Microelectronics Cooling

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SUMMARY

An analytical model is developed of an optimized small scale refrigeration system using vapor compression, with application to the cooling of electronic components populating a Printed Circuit Board (PCB). The author's previous study [1] evaluated a vapor compression system using an off-the-shelf compressor, focusing on the thermal feasibility of non-optimized mechanical refrigeration system and on-chip system-level incorporation. Investigation focuses on the miniaturization of evaporator micro-channels, fitting smaller packages. The design takes into account the miniaturized system components, optimized to address the appropriate power dissipation ranges. R-134s refrigerant is used, providing the best COP/feasibility ratio, suitable for microelectronics applications [2].

Various micro-channel designs are considered. The efficiency is evaluated for various structures; the optimal design is compared to designs using mechanical compression cooling. The advantages of the optimized design are highlighted, establishing a performance vs. size relationship for vapor-compression refrigerators, to serve as the basis for enhanced cooling of future miniaturized refrigeration applications.

INTRODUCTION

One of the main concerns of the microelectronics industry is the increased heat flux reached by the individual components, leading to failure or faulty operation. It is widely recognized that the current cooling techniques using fans, baffles, heat sinks, heat spreaders, will cease to satisfy the projected thermal requirements of the power systems. Several alternative active cooling methods incorporating liquid cooling, Thermoelectric Coolers (TEC's) or phase change techniques, have been extensively explored before, yet the application of classic refrigeration has not been considered to a significant extent, due to the large size constraints.

The present work fills this gap, proposing a miniaturized refrigeration system, designed to fit the existing microelectronics, with efficient cooling and minimal associated costs. The advantage of the proposed method is the sub-ambient cooling of the junction, leading to increased operating speeds of the processors (maintained thus at fairly constant temperatures), also to the increased overall reliability and component feasibility.

Studies over the past several decades acknowledged that vapor-compression refrigerators have been the leading refrigeration technology at the macroscale [3]. Additional studies [4 – 6] provided an enhanced understanding of the heat transfer and multi-phase related flows and boiling processes in smaller scale systems with micro-channels.

The performance of *miniscale* vapor compression refrigerators, however, has not yet been experimentally demonstrated, although recent theoretical studies [2, 7 and 8] have presented an overview of refrigeration alternatives for microelectronics cooling, and indicated that at temperatures near room temperature and above, a vapor compression refrigerator may compete successfully with a high efficiency thermoelectric cooler.

The objective of the study is threefold: 1) To evaluate a baseline vapor compression refrigeration design with a given overall size and cooling power; 2) To provide a comprehensive comparison between various miniaturized evaporator designs incorporating micro-channels, and 3) To extract meaningful performance factors and identify the evaporator design with the best performance.

FUNCTIONAL DESIGN FOR A VAPOR COMPRESSION REFRIGERATION SYSTEM

The author's previous study [1] has focused on evaluating the overall thermal performance of a miniaturized vapor compression system (Fig. 1) for a given overall size and cooling power, without the individual performance optimization of the system components (evaporator, condenser, compressor). This study focuses on the optimization of the compression system elements, starting with one of the most critical components, the evaporator.

The evaporator makes direct contact with the package to be cooled, and enhancing its thermal performance will eventually lead to lower junction temperatures, with associated reliability and operating benefits for the microelectronic component. The evaporator uses micro-channel heat exchangers. The functional schematic with the evaporators (E1, E2) placed on top of each package/component, is presented in Figure 1.

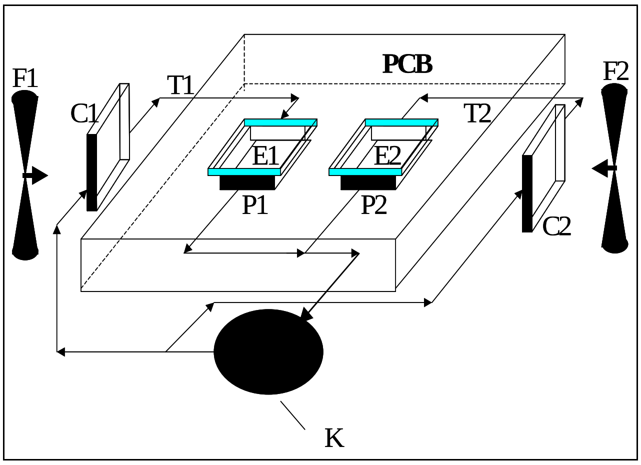

Figure 1: Overall Refrigeration System (w/PCB and packages)

COMPONENT SIZE OPTIMIZATION

The Evaporator

As seen in Figure 1, the evaporator is in direct physical contact on one side with the heat dissipating component (package), and is thermally insulated (in order to minimize heat losses) on the other side. The evaporator can be made relatively small, due to the good thermal coupling between itself and the heat load (i.e., the package). Aluminum is the material of choice for the evaporator in this study, due to its high thermal conductivity and manufacturing/processing convenience.

As it is, the heat transfer rate is limited only by the conductive coupling, which can be made very efficient using a high-conductivity heat spreader (e.g. an inverted exposed pad package) and thermal interface material at any solid/solid junctions. The conceptual schematic is shown in Figure 2, with the evaporator consisting of a micro-channel structure in contact with the package.

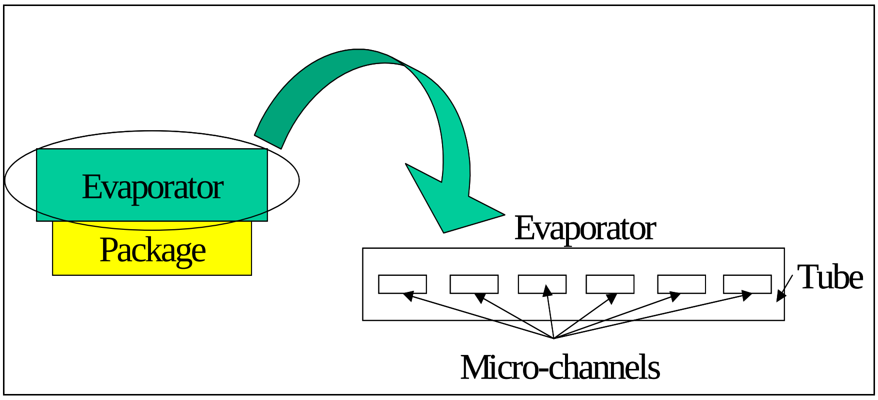

Figure 2: Evaporator/Package Stack-up

It is desired to optimize the evaporator structure/geometry, the number of micro-channels and the effective surface area available for the heat transfer. The inputs are: the evaporator rate of heat transfer (Q_o), T_o, T_{pi} (package initial temperature), T_{pf} (package final temperature), also V_o (volumetric flow rate of vapor at state 1 – compressor entrance).

<u>Evaporator Micro-Channels Structure/Array</u>

For a micro-channel with a hydraulic diameter D_h, the volumetric flow rate is described by:

$$V_o = N\, u\, (\pi\, D_h^2)/4, \qquad (1)$$

Where: N = no. of micro-channels, u = vapor velocity. The number of micro-channels can be easily determined:

$$N = 4\, V_o/(\pi\, u\, D_h^2), \qquad (2)$$

<u>Evaporator Heat Transfer Surface Area</u>

The basic relationship used for the calculation of the surface area is derived from the global heat exchanger governing equation:

$$Q_o = K\, A_o\, \Delta Tmo, \qquad (3)$$

With: Q_o = evaporator heat power, K = overall heat transfer coefficient, A_o = evaporator surface area, ΔT_{mo} = average temperature difference between the package and evaporator refrigerant.

$$\Delta T_{mo} = (T_{pi} - T_{pf})/ln[(T_{pi} - T_o)/(T_{pf} - T_o)], \qquad (4)$$

The overall heat transfer coefficient K can be expressed as:

$$K = [1/\alpha_o + R_{tc} + R_{tw}]^{-1}, \qquad (5)$$

Where: α_o = the vaporizing process s heat transfer coefficient, R_{tc} = thermal contact resistance and R_{tw} = evaporator's wall thermal conductive resistance.

The vaporizing process heat transfer coefficient α_o is further calculated using one of the classical flow boiling correlations, with total vaporizing of the refrigerant at evaporators' exit, followed by vapor super-heating.
The analytical formulation was suggested by Schrock and Grossmann, further modified by Chaddock and Noerager [9]:

$$\alpha_o = \alpha_{mf} F, \qquad (6)$$

where α_{mf} = convective heat transfer coefficient for single-phase flow, and

$$F = 1.85\ [B_o\ 10^4\ (1/X_{tt})^{0.67}]^{0.62}, \qquad (7)$$

$$B_o = q_o/(rG), \qquad (8)$$

$$X_{tt} = [(1\text{-}x)/x]^{0.9}\ (\rho_v/\rho_l)^{0.5}\ (\mu_l/\mu_v)^{0.1}, \qquad (9)$$

With: q_o = heat flux (W/m^2), r = latent heat (J/kg), G = mass flow rate (kg/m^{2}s); x = vapor quality (value of 0.8).

The single-phase heat transfer coefficient α_{mf} may be calculated with the convective heat transfer correlation inside the tubes [10].

RESULTS

A total of 6 evaporator (with micro-channel heat exchangers) structures have been evaluated, with the final purpose to identify the smallest available design with the highest efficiency, able to fit existing microelectronic components. The first case (Design 1) evaluated below is connected to the baseline overall mechanical compression refrigerator design presented in detail in the previous paper [1].

REFRIGERATION SYSTEM DESIGN FOR PRACTICAL MICROELECTRONICS APPLICATIONS

Evaporator - Design 1

The simulation was based on a practical microelectronics application dissipating a total power of ~ 100W. This could apply to a single or an array of power microelectronic packages. In order to design the evaporator, the following inputs are needed (Table 1):

Table 1: Inputs for Evaporator Design

Q_o (W)	T_o (°C)	T_{pi} (°C)	T_{pf} (°C)	V_o (m^3/hr)	D_h (m)	U_o (m/s)
100	10	120	20	0.14	1.54E -06	3.5 - 5

Using the analytical correlations from the previous paragraph, the optimal dimensions of the evaporator are extracted, leading to N = 6 micro-channels. The micro-channels are arranged inside the tube as described in Figures 3 and 4.

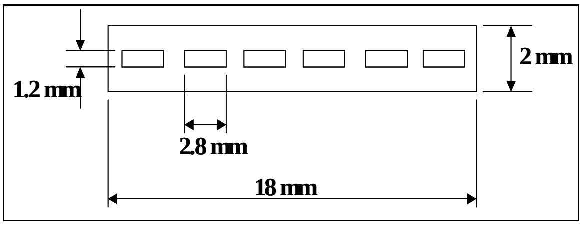

Figure 3: Evaporator Cross-Section.

The other dimensional values are: $\Delta T_{mo} = (T_{pi} - T_{pf})/\ln[(T_{pi} - T_o)/(T_{pf} - T_o)] = 41.7K$; K = 8260 W/m^{2}K; A_o = 2.9E –04m^2; A_o = L l (where L is the tube length and l is the tube width); hence, L = 1.61E-02m. A summary of the main results is provided in Table 2.

Table 2: Results – Design 1

Overall K	So	Tm	Qo
w/m2K	mm2	k	w/m2
8262.96	289.8	41.7188	99.90

The design details are consistent with the specifications provided by evaporator manufacturing companies. The final evaporator configuration is presented in Figure 4.

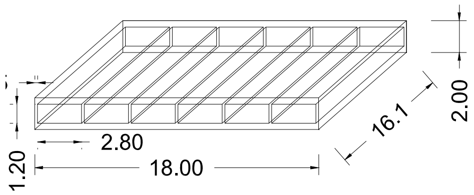

Figure 4: Evaporator – Design 1.

It is the intent to further miniaturize the system and identify the evaporator design with the smallest specific size which provides the best cooling performance.

Ideally, the evaporator should fit the exposed pad of the power package, since being slightly larger than the package could lead to significant reduction in the cooling efficiency. Several designs are further evaluated to reduce the overall size and yet maintain the efficiency at high levels.

Evaporator - Design 2

A similar procedure is used for all the following designs, to calculate the optimal size of the micro-channels for a specific hydraulic diameter and power dissipation.

In the second design, the hydraulic diameter is reduced to 1 mm, and the overall system size is maintained as before at 18 x 16.1 x 2 mm. The individual size of the micro-channels has reduced (Fig. 5). The units in Figs 5 – 8 are millimeters.

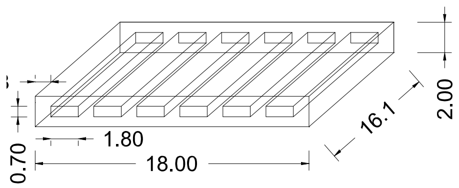

Figure 5: Evaporator – Design 2.

Equations 1 – 9 are used again to calculate the equivalent overall heat transfer coefficient. The value in this case is about twice compared to the first case, thus the increased cooling power (~ 190W) for this case. A summary of the heat transfer coefficient, area available for cooling, and cooling power are summarized in Table 3.

Table 3: Results – Design 2

Overall K	So	Tm	Qo
w/m2K	mm2	k	w/m2
15796.05	289.8	41.72	190.98

Evaporator - Design 3

In the third case, the hydraulic diameter is further reduced to 0.5 mm, while maintaining the overall size as in the previous cases. Keeping the same number of micro-channels (a total of six), will lead to smaller micro-channels dimensions (Fig. 6).

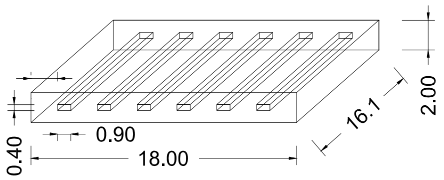

Figure 6: Evaporator – Design 3

The results of the study are summarized in Table 4 and indicate that the overall heat transfer coefficient is even higher than for the first two cases, at ~ 44700 W/m2K. For the same overall size of the evaporator, when using smaller micro-channels, the effective overall heat transfer coefficient (K) increases. In this case, the cooling power reaches almost 540W.

Table 4: Results – Design 3

Overall K	So	Tm	Qo
w/m2K	mm2	k	w/m2
44706.88	289.8	41.72	540.51

The last two cases focus on two evaporator designs with reduced overall dimensions.

Evaporator - Design 4

Compared to the previous designs, the overall size of the system for design 4 has changed to 6.8 x 6.8 mm, and maintained the smaller 0.5 mm hydraulic diameter.

By reducing significantly the size, the evaporator will fit better on top of the inverted exposed pad power package, providing also a better thermal contact. However, by maintaining the same hydraulic diameter as in case 3, the overall heat transfer coefficient will remain unchanged.

Following the same calculations as before, the cooling power in this case is ~ 85W, and this is satisfactory for power packages operating at 100% duty cycle. The new evaporator schematic is presented in Figure 7, while the new results are included in Table 5.

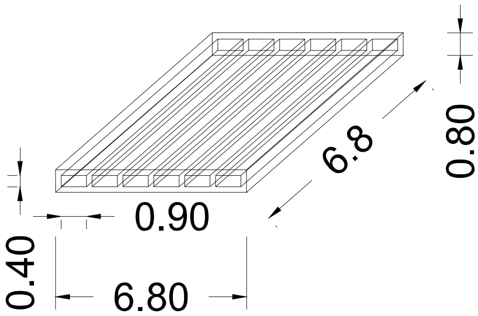

Figure 7: Evaporator – Design 4

Table 5: Results – Design 4

Overall K	So	Tm	Qo
w/m2K	mm2	k	w/m2
44706.88	46.24	41.72	86.24

Evaporator - Design 5

In the last case, the size is further reduced to 6.1 x 6.1 mm, with the hydraulic diameter still at 0.5 mm. The micro-channels will be more compact in this design, with the schematic presented in Figure 8.

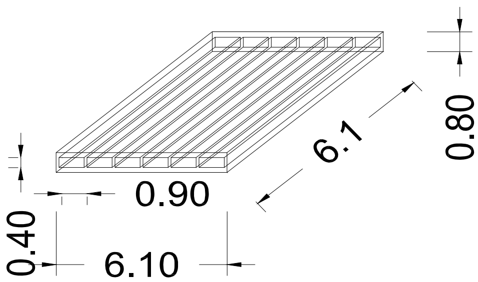

Figure 8: Evaporator – Design 5

Again, as the hydraulic diameter is the same as for the previous two cases, the overall heat transfer coefficient (K) is the same. However, due to the reduced overall size of the evaporator compared to previous cases, the cooling power is much smaller and reaches ~ 69.4 W. Results summarized below.

Table 6: Results – Design 5

Overall K	So	Tm	Qo
w/m2K	mm2	k	w/m2
44706.88256	37.21	41.7188	69.4010219

EVALUATION STUDY

The results are summarized in Table 7. The size, overall evaporator surface and volume, the cooling power as well as the micro-channel overall surface (S_{mch}) are reported for each case. The micro-channel overall surface area is decreasing with the decrease in the overall evaporator size.

Table 7: Summary - Various Designs

Design	A (mm)	B (mm)	H (mm)	L (mm)	l (mm)	D_h (mm)	S (mm^2)	V (mm^3)	Q_o (W)	S_{mch} (mm^2)
1	2.8	1.2	2	18	16.1	1.54	289.8	579.6	99.9	467.35
2	1.8	0.7	2	18	16.1	1	289.8	579.6	190.9	303.47
3	0.9	0.4	1.2	18	16.1	0.5	289.8	347.76	540.5	151.73
4	0.9	0.4	0.8	6.8	6.8	0.5	46.24	36.99	86.24	64.08
5	0.9	0.4	0.8	6.8	6.8	0.5	36	28.8	69.4	57.5

The impact of the hydraulic diameter on cooling efficiency for same size designs (Cases 1 –3) is presented in Figure 9. The cooling power decreases with the increase in hydraulic diameter, due to the decrease in the overall heat transfer coefficient.

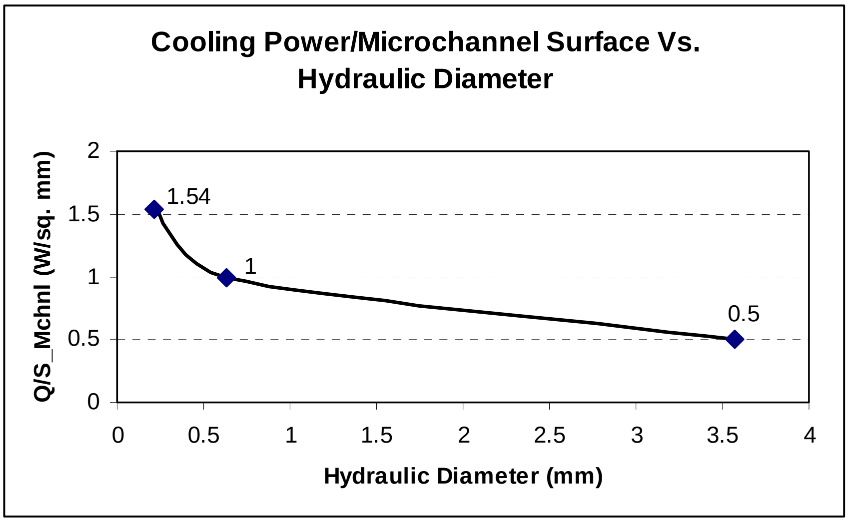

Figure 9: Cooling power variation with hydraulic diameter for same overall volume (Designs 1 - 3)

Furthermore, the cooling power varies almost linearly with the evaporator volume for the same hydraulic diameter (as seen in Figure 10). This is due to the fact that the heat transfer coefficient is the same, but the available surface area for heat transfer increases linearly with the increase in volume.

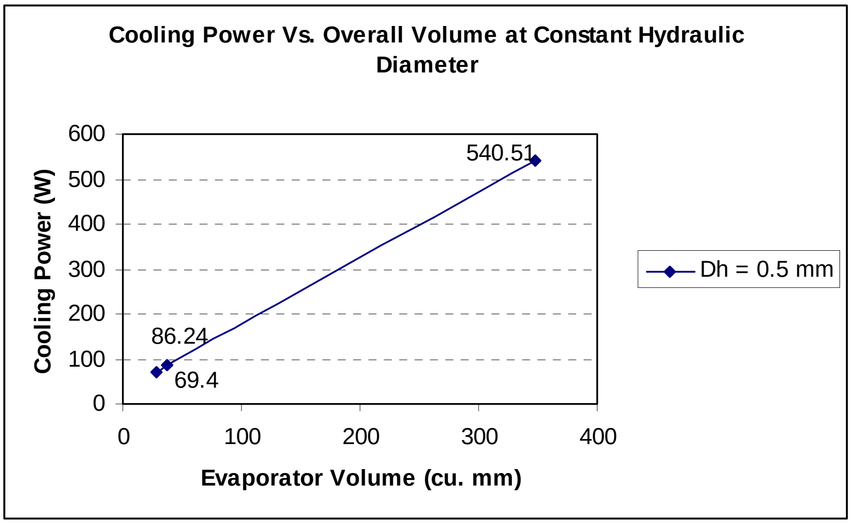

Figure 10: Cooling power variation with evaporator volume for same hydraulic diameter (Designs 3 – 5)

It is desired to have an optimal ratio between the overall heat transfer coefficient (closely related to the micro-channel hydraulic diameter) and the contact surface area between the evaporator and package. Figure 11 shows the ratio for cases 1 – 5.

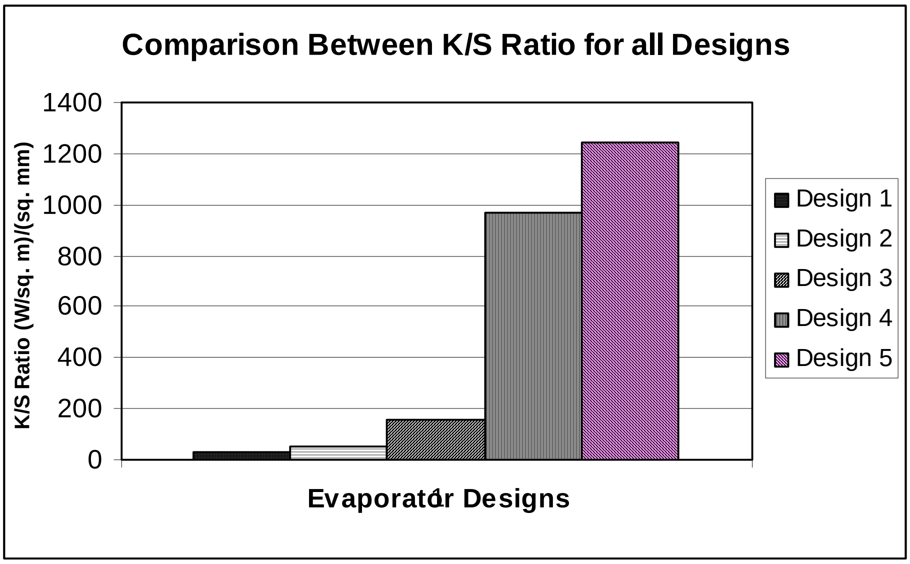

Figure 11: Comparison Between Evaluated Cases

Based on the values summarized in Table 7, the smallest design evaluated (Design 5) is the best of all, as the cooling power per micro-channel surface area is the highest.

Also, Figure 11 indicates that Designs 5 is the most efficient as the heat transfer coefficient divided by the contact surface area is the highest, thus validates the concept that the most compact evaporator will provide the most efficient mini-scale cooling.

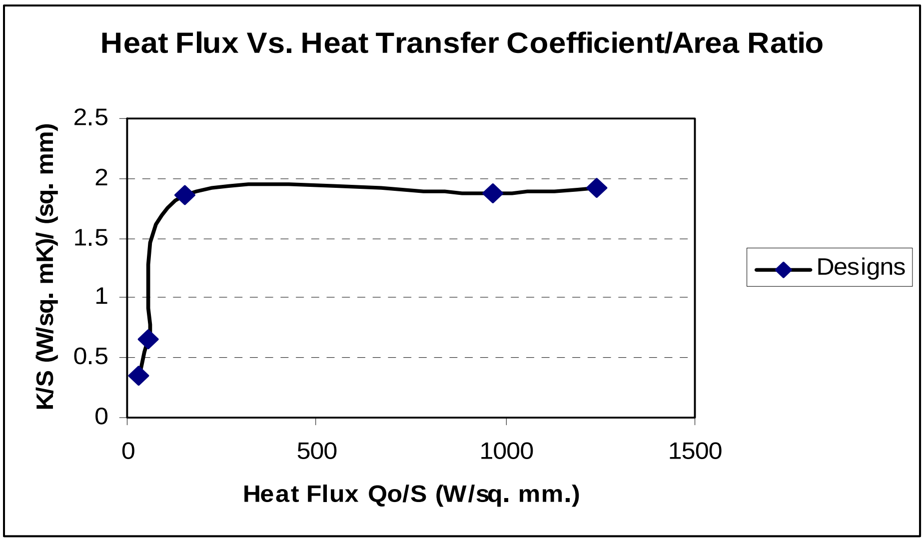

Figure 12: Heat Flux Vs. Heat Transfer Coefficient Ratio

Finally, Figure 12 shows the steep increase of the heat flux when increasing only slightly the heat transfer coefficient to area ratio, and then reaches asymptotically a constant value for the smaller hydraulic diameter micro-channel designs. This indicates that the system will reach a peak in its thermal performance that cannot be further improved by decreasing the hydraulic diameter, which controls both K and S.

NAMES AND UNITS

P Power (W)	Q Rate of Heat Transfer (W)	T Temperature (ºC)	u Refrigerant Velocity (m/s)
m Mass Flow Rate (kg/s)	V Volume Flow Rate (m^3/s)	p Pressure (N/m^2)	h Specific enthalpy (kJ/kg)

CONCLUSIONS

An analytical model of the evaporator design is developed using the R-134a refrigerant. Various micro-channel designs are considered while miniaturizing the evaporator to be incorporated in the vapor compression system. The efficiency of the proposed system is evaluated for various structures, and the optimal design is further identified and compared to previous designs using mechanical compression cooling. The advantages of the optimized refrigeration design are highlighted, establishing a performance vs. size relationship for vapor-compression refrigerators, to serve as the basis for the enhanced cooling of future miniaturized refrigeration applications.

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A Comparison Between Refrigerants Used In Air Conditioning

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SUMMARY

It's clearly known that growing dimensions of the hole on ozone layer has very important damages and effects on living creatures today. Investigations show that refrigerants that commonly used in air conditioning have great impacts on damage of ozone layer. For this reason, researches on alternative refrigerants to be used for air conditioning are still continuing. In this study, performance of different refrigerants like R600a, R134a, R290, R1270, R32, R22, and R152a, in vapor compression refrigeration cycles and heat pumps was investigated. Analyses was made on the ideal vapor compression refrigeration cycle and a second cycle which was created by adding subcool and superheat regions on the first one. While evaporation temperature is changing between -40 °C and – 10 °C, condensation temperature Tcond = 40 °C is constant for 2,2kW of cooling load. Also a comparison between refrigeration performance and variation on compressor power was carried out at different evaporation temperatures of different refrigerants.

INTRODUCTION

Natural elements such as carbon dioxide, air, water and ammonia have been used as refrigerants in cooling systems which started being developed at the second half of 19th century. During the following century, chlorofluorocarbons and hydro chlorofluorocarbons have replaced some of these elements and have seen heavy use. [1] However, as a result of the dispersal of these elements into the atmosphere, several environmental problems such as damages on the atmosphere and the greenhouse effect have occurred. Especially the ozone layer which protects living beings from the harmful rays of the sun, is being damaged as the chlorine atoms are released from the structure of cooling elements and start damaging the weak ozone molecules.[4] The problem that is the destruction of the ozone layer has gained worldwide significance and countries have united to find a way to prevent it. With the Montreal Protocol that has been signed by 43 countries in the year 1987, the production and usage of CFC class refrigerants which also include R12 have been gradually constrained. Aside from the countries that have signed the protocol, other countries have set up new law regulations regarding the import, export and production of these refrigerants. It was expressed by medical authorities that there was an increase in the number of several clinical diseases with the growing of the holes in the ozone layer. As a result of the studies that were carried out about this issue which is highly important for environment and human life, it was found out that the CFC group refrigerants have got a great potential of damaging the ozone layer. Many alterative refrigerants were invented to be used instead of these harmful refrigerants and the researches for developing new are still taking place. The refrigerants that are used as replacement for the HFC's are R134a, R404A, R407A, R410A, R22, and hydrocarbons (propane and butane) and ammonia. Propane could also be used, with an acceptable degree of risk, for car air conditioning, again provided that appropriate precautions were taken. Some tens of thousands of car air conditioning systems have been ‘‘unofficially’’ converted to operate on propane by private individuals in USA and Australia. [5]

In this study, the thermodynamic analysis of the R22 refrigerant, the pure refrigerants R134a, R32 and R152a as well as the easy-to-obtain, cost efficient, environment-friendly natural refrigerants R290, R1270 and R600a have been carried out in a single phase steam compression cooling cycle. The mass flow rate, compressor power and COP changes of the aforementioned refrigerants have been shown as graphs. While evaporation temperature is changing between -40 °C and – 10 °C, condensation temperature Tcond = 40 °C is constant for 2,2kW of cooling load. Analyses was made on the ideal vapor compression refrigeration cycle and a second cycle which was created by adding subcool and superheat regions

THE COMPARISON OF THE THERMODYNAMIC STATISTICS OF COOLING LIQUIDS

While designing air conditioning and cooling systems, refrigerants that do not damage the ozone layer or cause global warming should be preferred instead of those that have the same physical features but have harmful effects.

Table 1. Refrigerants of thermodynamics property

REFRIGERANTS	MOLAR MASS (KG/KMOL)	SATURATED TEMPERATURE(^{0}C)	T_C (^{0}C)	P_C (BAR)	TLV (PPM)	LFL (%)	DELTA h_{comb} (MJ/Kg)	ODP	GWP (100 YEARS)	ATMOSPHERIC LIFETIMES (YEARS)
R600a	58,112	-11,670	134,67	36,4	800	1,8	49,4	0	~20	-
R134a	102,03	-26,074	101,06	40,593	1000	0	4,2	0	1300	14,6
R290	44,096	-42,090	96,675	42,471	2500	2,3	50,3	0	~20	-
R1270	42,080	-47,690	92,420	46,646	375	2	-	0	2	-
R32	52,024	-51,651	78,105	57,820	1000	13,3	9,4	0	650	5,6
R22	86,468	-40,810	96,145	49,900	1000	0	2,2	0,04	1500	12,1
R152a	66,051	-24,023	113,26	45,168	1000	3,1	17,4	0	140	1,5

The R22 refrigerant which is used as an alternative to CFC-included refrigerants, has got a low ODP value and thus, its use is permitted until the year 2030. As a result, a refrigerant that has similar thermodynamic properties could be an alternative to R22. As Table 1 shows, taking into account the Global Warming Effect (GWE), the Ozone Depletion Coefficient and the other statistics, the R134a could be an alternative to R22. However, the fact that R134a has a relatively high global warming effect requires an alternative to be found for this refrigerant too. The hydrocarbons like R600a, R290 and R1270 are good alternatives to R12, R22 and R502 since they do not damage the ozone layer much and they have a very low greenhouse effect. Even though their high rate of flammability prevents them from being used commonly, the propane-butane and propane-isobutene mixtures are good alternatives to R12 in house-type coolers. Most of the refrigerants that are to be compared against one another in a simple coolant cycle are chosen from hydrocarbons in this study too.

THEORETICAL ANALYSIS

This study has been carried out on a compressed vapor cooling cycle which runs on -30 °C evaporation temperature and 40 °C condensation temperature and which has been designed for a no-frost refrigerator. The saturated and boiling steam properties of refrigerants R22, R134a, R32, R152a, R290, R1270 and R600a to be analyzed have been taken from the Refprop 7.0 software.

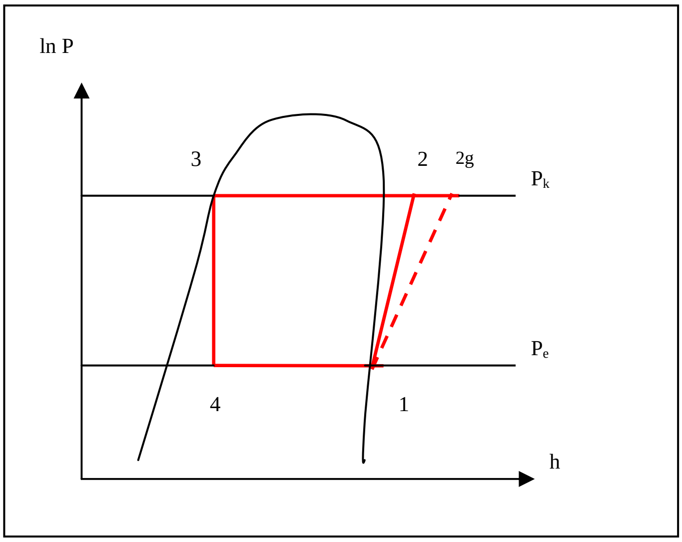

Figure 1. lnP-h diagram of ideal vapor compression cycle

Table 2 Refrigerants enthalpy value

Refrigerants / Enthalpy	R600a	R134a	R290	R1270	R32	R22	R152a
h_1 (kj/kg)	515,21	380,32	540,22	548,17	506,27	392,69	485,55
h_2 (kj/kg)	605,19	432,33	639,23	650,83	603,17	450,38	569,1
h_{2g} (kj/kg)	627,68	445,332	663,982	676,495	627,395	464,802	589,98
h_3 (kj/kg)	297,03	256,41	307,82	305,01	275,61	249,65	271,35
h_4 (kj/kg)	297,03	256,41	307,82	305,01	275,61	249,65	271,35

Taking the cooling capacity of the evaporator as 2.2 kW's, the various mass flow rates, compressor works and condenser capacities have been calculated for different refrigerants and the COP values have been acquired. Thus,

Q_l= 2,2 (kW)

the mass flow rate of the liquid to travel through the cooling cycle is calculated as:

$$m = Q_l / (h_1 - h_4)\ (kg/s), \qquad (1)$$

The work done by the compressor has been calculated at Equation 3, by taking the isentropic efficiency as (0,80).

$$0{,}8 = (h_1 - h_2) / (h_1 - h_{2g}), \qquad (2)$$

$$W_{comp} = m\ (h_{2g} - h_1)\ \ (kj/s), \qquad (3)$$

The condenser capacity has been calculated according to Equation 4.

$$Q_k = m(h_{2g} - h_3), \qquad (4)$$

Cooling performance COP;

$$COP = Q_l / W_{comp}, \qquad (5)$$

has been calculated at Equation 5.

The change in the compressor power has been compared in Figure 2 by changing the evaporation temperature of the refrigerants in use between -10 °C and – 40 °C. The increasing compressor power is a parameter that decreases the COP. It is understood from the figure that for all the cooling liquids, the compressor power decreases linearly with an increase in the evaporation temperature. It is shown that R290 requires more compressor power than the other refrigerants at evaporation temperatures lower than -20 °C, while R152a requires less. It is also shown that R 32 requires more compressor power than the other refrigerants at evaporation temperatures higher than -20 °C, while R 22 requires less.

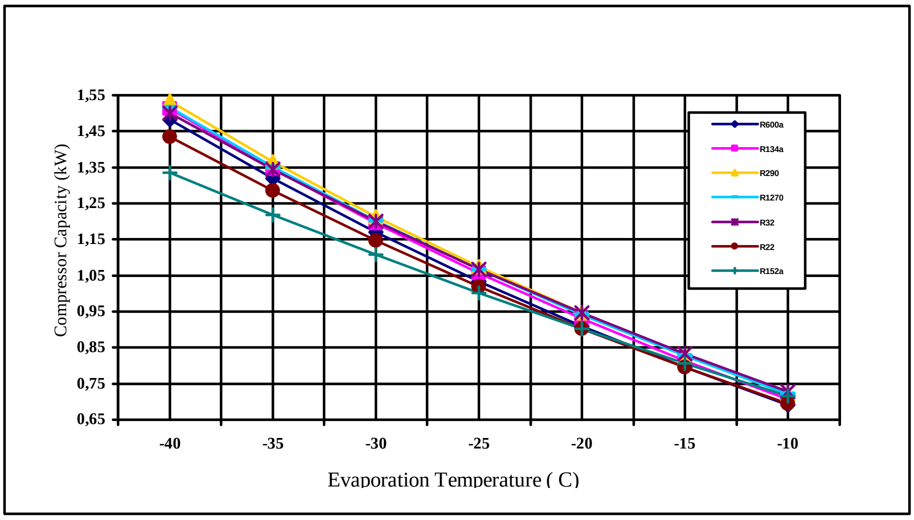

Figure 2. Effect of evaporation temperature changes of refrigerants on compressor capacity

Figure 3 shows the change of the mass flow rate of refrigerants along with the changing evaporation temperatures, for 2.2 kW's of cooling capacity. And for all the refrigerants, mass flow rate decreases linearly with increasing evaporation temperature. The highest mass flow rate between the temperature range of -40 °C and -10 °C is R134a liquid's, while the lowest is the R1270's. Having a low mass flow rate with in the system is a parameter that reduces the cost of operating.

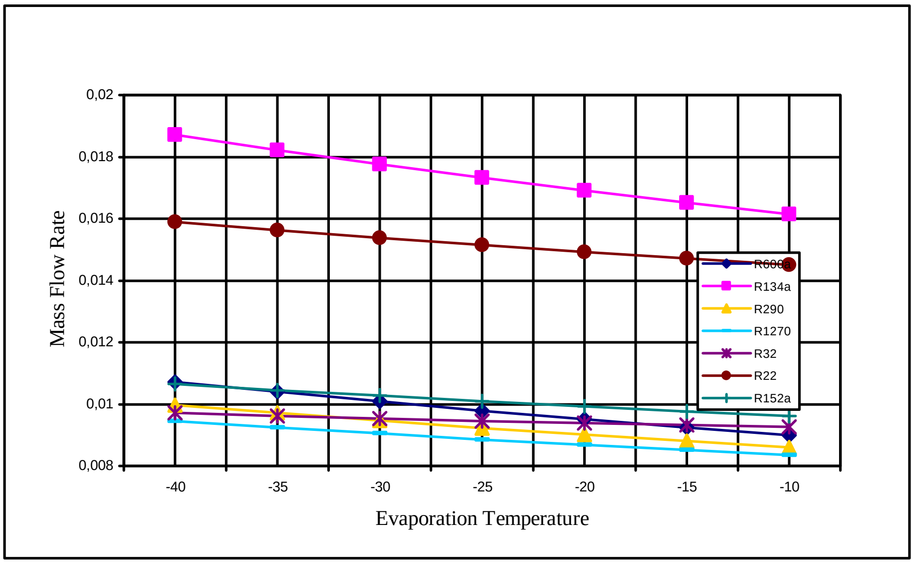

Figure 3. The mass flow rate changes of refrigerants with the changing evaporation temperatures, for 2.2 kW's of cooling capacity

Figure 4 shows the change of the cooling performance (COP) of refrigerants for a cooling cycle with a cooling load of 2,2 kW and with a -30 °C evaporation temperature and 40 °C condensation temperature. According to the figure, it is seen that the COP of R 152 a is the highest while the COP of R 290 is the lowest.

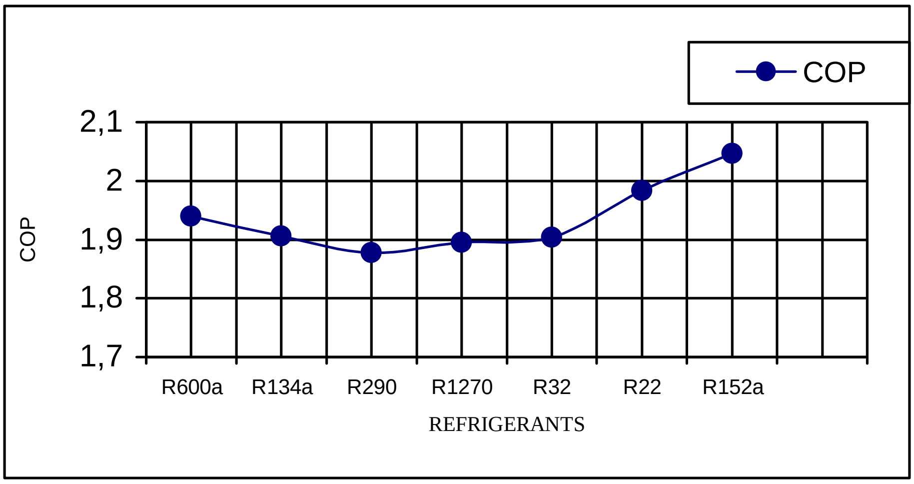

Figure 4. The change of the COP of refrigerants for a cooling cycle

The COP of R 22 has been considered to be 100% in Figure 5 and the comparison for the COP's of the rest of the alternative refrigerants has been made. It was seen that the performance of R 152 a is better than R 22, while the others are worse.

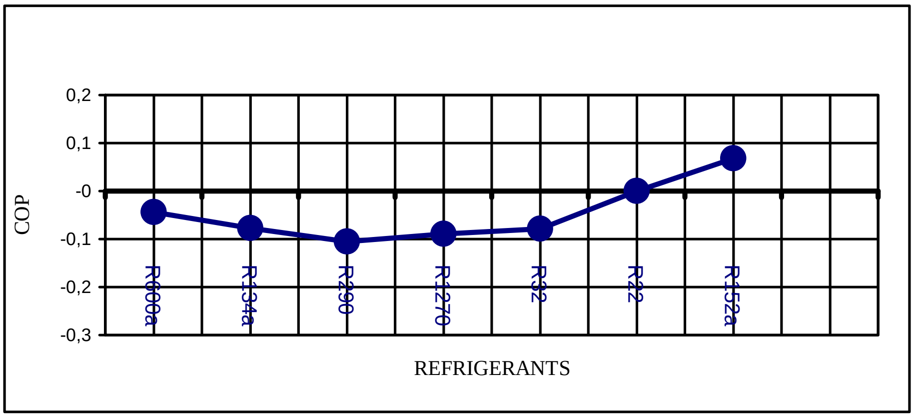

Figure 5. The comparison between R 22 COP and alternative refrigerant's COP's

Figure 6 shows the COP changes of the refrigerants chosen for this work between 40 °C condensation temperature and -40 °C to -10 °C evaporation temperature. In the refrigerants R600a, R134a, R290, R1270, R32, R22 and R152a, the COP increases with increasing evaporation temperature. But this increase is varying upon the refrigerant used.

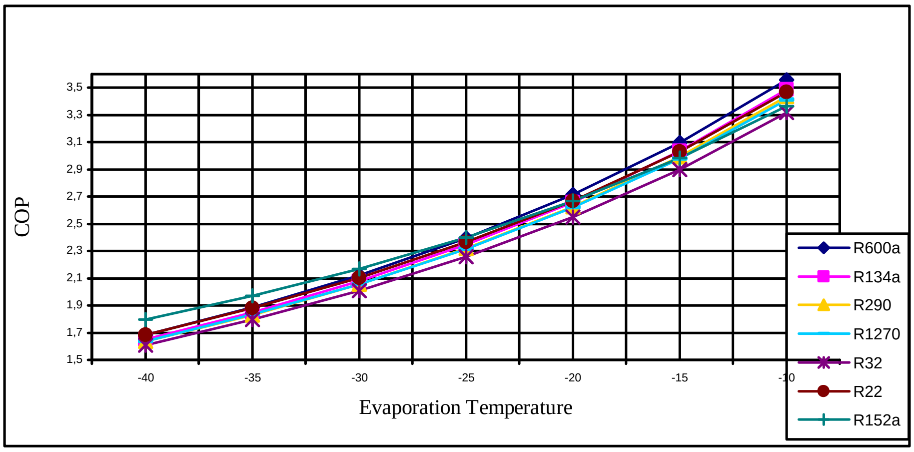

Figure 6. The change of the COP of refrigerants with the changing evaporation temperatures

From this graph it is possible to conclude that the COP value of R 152a is high for evaporation temperatures below -25 °C and the COP value of R 600a is higher than the rest of the refrigerants between -25 °C to -10 °C evaporation temperatures and that it can be preferred to the other refrigerants within this temperature range.

The cooling performances of refrigerants have been tested by making 7 °C of subcooling is made within the condenser and 7 °C of superheating in the evaporator.

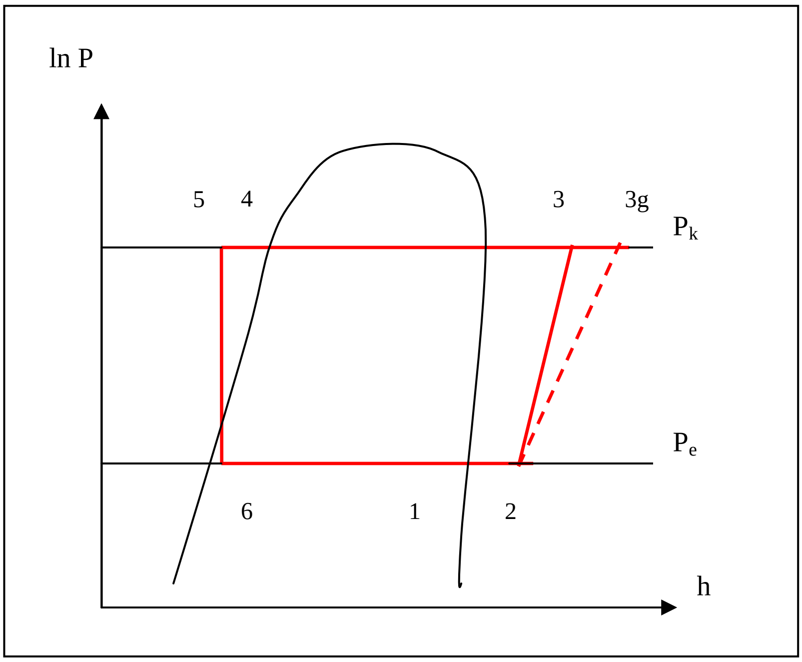

Figure 7. lnP-h diagram of ideal vapor compression cycle by making subcooling and superheating

Table 3: Refrigerant's enthalpy value

Refrigerants / Enthalpy	R600a	R134a	R290	R1270	R32	R22	R152a
h_1 (kj/kg)	515,21	380,32	540,22	548,17	506,27	392,69	485,55
h_2 (kj/kg)	525,45	385,79	551,03	558,17	513,1	397,78	492,24
h_3 (kj/kg)	618,27	439,61	653,57	664,49	613,78	457,7	578,49
h_{3g} (kj/kg)	641,475	453,065	679,205	691,07	638,95	472,68	600,0525
h_4 (kj/kg)	297,03	256,41	307,82	305,01	275,61	249,65	271,35
h_5 (kj/kg)	279,34	246,07	287,71	285,3	261,01	240,45	258,32
h_6 (kj/kg)	279,34	246,07	287,71	285,3	261,01	240,45	258,32

For the situations that 7 °C of subcooling and 7 °C of superheating has been made, the amount of heat that the evaporator is going to absorb is found out by equation 6.

$$Q_{l2} = m\,(h_2 - h_6), \qquad (6)$$

Taking into account the isentropic efficiency of the compressor, the work expanded by the compressor has been calculated for various refrigerants for the 2nd cooling cycle that has sub cooling and superheating.

$$0{,}8 = (h_2 - h_3)\,/\,(h_2 - h_{3g}), \qquad (7)$$

$$W_{comp} = m\,(h_{3g}-h_2)\ \ (kj/s), \qquad (8)$$

Condenser capacity has been calculated at equation 9

$$Q_k = m\,(h_{3g}-h_5), \qquad (9)$$

COP has been calculated at equation 10

$$COP = Q_{l2} / W_{comp}, \qquad (10)$$

Figure 8 shows the COP values for the refrigerants that work within the ideal vapor compression cooling cycle and the comparison of these COP's for the 7 °C subcooling and 7 °C superheating cooling cycle situations.

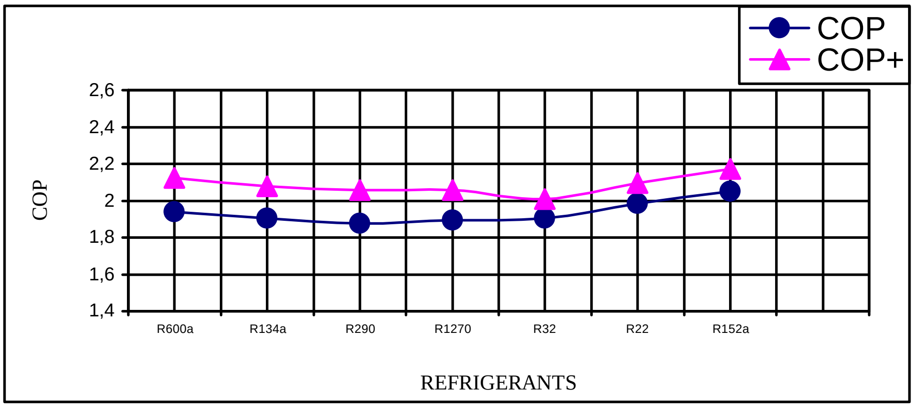

Figure 8 Comparison between COP and COP+.

The increase in the COP has been calculated by applying subcooling and superheating to the vapor compression cooling cycle. The increase rate of the COP's of the refrigerants have been shown to be different. Figure 9 shows the COP increase rates for different refrigerants. According to this, R600 has the greatest increase and R152a has the least.

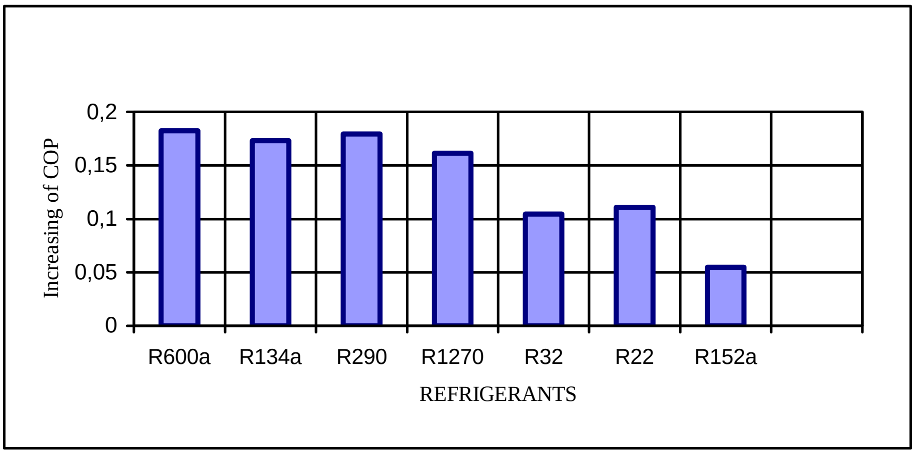

Figure 9. Comparison of COP increase rates of refrigerants by applying subcooling and superheating

RESULTS

Within this study, the different thermodynamic properties of refrigerants R600a, R134a, R290, R1270, R32, R22 and R152a have been analyzed. And in the modern day world which ozone destruction is very important, it is very interesting to find out that the R152a that belongs to the hydro fluorocarbon (HCF) group has an ozone damage rate of zero-to-none. In addition to this very important feature of it, its effect on the global warming is only 2% of the effect of R12. R152a is an alternative at heating pumps for R12 and R500. It also works well with mineral oils. R152a is not flammable and it is odorless. It shows no poisonous properties. Thus, there is no need for the modification of the compressor in the cycle of the cooling system that this refrigerant is used in. The hydrocarbon based refrigerants used in the study have high flammability and this prevents their fields of use to be limited. Although they can be used on their own, they can also be used as mixtures of each other. Propane-isobutene and propane-butane mixtures can be used as alternatives to R12 in house type cooling systems. They can be used together with oils that could be used together with R12. In addition to the fact that among the hydrocarbons, Propane is cheap and easy-to-obtain, its cooling capacity per volume is around 35-50& higher than R12's, and these facts make this coolant a possible alternative for R12.

Further in this study, the condensation temperature (T_c = 40 °C) has been taken as a constant for 2.2 kW of cooling load and the evaporation temperature (T_e) has been changed between -40 °C and -10 °C. It was seen that for all the refrigerants, the mass flow rate and the compressor power increase while the COP values decrease, with increasing evaporation temperatures. It can be concluded that the COP value of R 152a is high for evaporation temperatures below -25 °C and it is advisable to use this refrigerant rather than the others at these temperatures while the COP value of R 600a is high for evaporation temperatures between -25 °C and -10 °C and it is advisable to use this cooling liquid rather than the others at these temperatures.

As a conclusion of the study, the comparison of COP's for the vapor compression cooling cycle situations and the situations that superheating and subcooling have been carried out. It was found out that superheating and subcooling are elements that increase the COP but this increase rate varies depending on the type of refrigerant used. The COP increase for the R600a, R290 and R1270 type refrigerants which belong to the hydrocarbon family is higher. The increase of COP means energy savings. This shows that the hydrocarbons might be the choice for alternative refrigerants in the future.

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Energy efficient heating systems

Integrated Design Of Thermally Activated Building Systems And Of Their Control

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SUMMARY

A new promising integrated design method for thermally activated building systems (TABS) and their control is presented. In contrast to the conventional iterative design method of TABS, which is based on iterative dynamic model simulation studies, the new approach is a more straight-forward calculation process. The new design method is less time-consuming and enables more insight into the potential of TABS. For example, it shows the limitation of TABS application and points out when additional auxiliary heating or cooling devices have to be installed. It is based on lower and upper bounds for the heat gains that have to be estimated within the design process.

KEYWORDS: control, thermally activated building systems (TABS), concrete core conditioning, integrated design, unknown-but-bounded

INTRODUCTION

A research project on the control and design of thermally activated building systems (TABS) [1], [2] has been started in May 2004 [3], [4]. This paper presents one selected result: A TABS design method for building automation and HVAC design engineers.
It is well known among TABS designers that TABS cannot be designed and sized like other HVAC systems. The design and sizing of an HVAC system and its control is usually done in two steps. In a 1st step the HVAC system is designed and sized using a static model for full load operation. The 2nd step is the design of its control, where sometimes dynamic models are used. If the HVAC system is a TABS this approach is not appropriate. This has two reasons. The temperatures in a TABS usually never achieve steady-state conditions. The dynamic behavior has therefore to be considered. This requires dynamic instead of static models. The second reason is that the dynamic behavior of the temperatures depends also on the control. Therefore the control should be taken into account when designing and sizing the TABS. Thus it should not be designed afterwards in a second step. That’s why a TABS and its control should be designed and sized in an integrated way. This is well known among designers of TABS. A frequently heard argument is that simulations have to be performed to design and size TABS. And for such simulations it is necessary to assume some kind of control.
This paper presents a new promising integrated design method for TABS and their control based on [3], [4]. In contrast to the conventional integrated design method of TABS, which is based on iterative dynamic model simulation studies, the new approach is a more straight-forward calculation process. It is based on lower and upper bounds for the heat gains that have to be estimated within the design process.
The first section of the paper will describe how the new integrated design method has been developed. The second section gives an overview of the resulting new integrated design method. A design example will illustrate the method in the third section.

DEVELOPMENT OF THE INTEGRATED DESIGN METHOD

In this section it is outlined how the integrated design method has been developed. It has been designed together with the control algorithms.

The following questions Q1 to Q4 have guided the design process essentially:

Q1 How to switch automatically between heating and cooling a TABS using the outside air temperature information?
This question has been motivated by often heard statements as: The switching between heating and cooling of TABS must be done by hand. Automatic solutions switch too frequently which causes waste of energy. Or automatic solutions based on outside air temperature dependent switching lead to comfort problems.

Q2 How to control rooms with different specific heating and cooling loads?

Q3 What are the advantages and disadvantages of water flow on/off control and how should it be done?

Q4 Is it possible to use room temperature feedback control, and if yes, how should it work and what is the benefit?
This question has been motivated also by often heard statements as: The self-regulating effect is sufficient to control the room temperature, no conventional room temperature feedback controller is necessary. Conventional room temperature feedback controllers do not work because the system to be controlled does react very slowly.

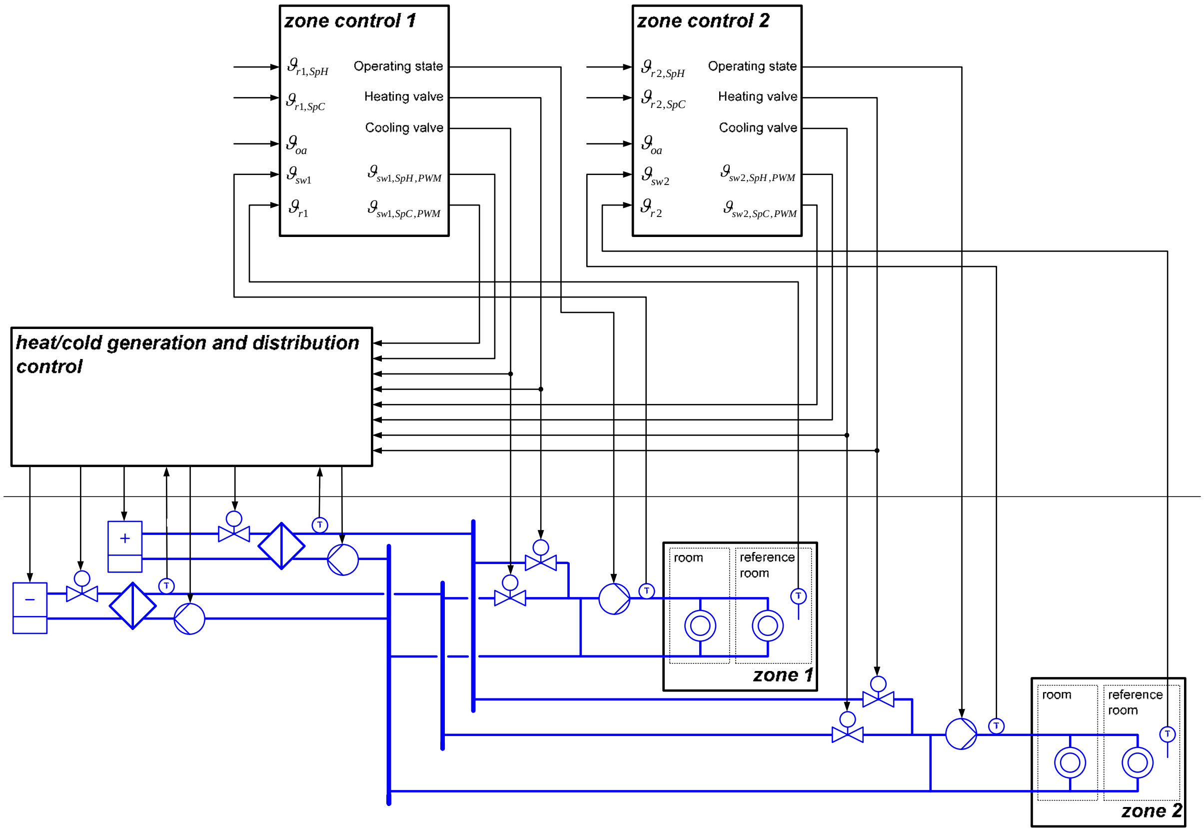


Figure 1: The overall TABS control concept, including heat/cold generation and distribution

Development step 1:

Question Q1 has been addressed first. It led also to an answer of question Q2. In order to find an answer to Q1 only one zone has been considered, e.g. zone 1 in Figure 1. It has been assumed that no room temperature sensor in a reference room is available. Thus the zone controller (Zone control 1 in Figure 1) has only two sensor inputs, the supply water temperature ϑ_{sw} (also called flow temperature) and the outside air temperature ϑ_{oa}. The zone controller acts on the zone heating and the zone cooling valve and it includes a supply water temperature feedback controller with the set-points $\vartheta_{sw,SpH}$ for heating and $\vartheta_{sw,SpC}$ for cooling. The zone controller has also the room temperature heating set-point $\vartheta_{r,SpH}$ and the room temperature cooling set-point $\vartheta_{r,SpC}$ as inputs, which define together the room temperature comfort range.

Development step 2:

In order to find a solution to the problem expressed in question Q1 it has been started with a related problem which is easier to solve, as it is recommended by G. Polya in his famous little book on how to solve problems [8]. The related problem is a special case: the outside air temperature dependent switching between heating and cooling for a system where the *outside air temperature and the heat gains vary only slowly*. This allows us to use a static model to solve the problem. It has additionally been assumed that all rooms of the zone are identical and have the same heat gain $\dot{q}_g$, which allows us to assume that all rooms have the same room temperature ϑ_r. This problem can be formulated mathematically as follows (**problem 1**):

Static model (cf. Figure 2):

$$\dot{q}_w = \frac{1}{R_{l,f}} \cdot \left(\vartheta_r - \vartheta_{oa}\right) - \dot{q}_g \tag{1}$$

where $\dot{q}_w$ is the TABS heating or cooling power and $R_{l,f}$ the thermal resistance of the façade.

$$\dot{q}_w = \begin{cases} 0 & \textit{if no heating and no cooling is required} \\ \dfrac{1}{R_t + \tilde{R}} \cdot \left(\vartheta_{sw} - \vartheta_r\right) & \textit{if there is heating or cooling demand} \end{cases} \tag{2}$$

where $R_t + \tilde{R}$ is a thermal resistance characterizing the TABS between the supply water temperature and the room temperature.

Known:

$R_{l,f}$, $R_t + \tilde{R}$, $\vartheta_{r,SpH}$, $\vartheta_{r,SpC}$, ϑ_{oa} and $\dot{q}_g$.

Determine:

When is it required to heat and when is it required to cool, and if either of them is required, what is the required supply water temperature ϑ_{sw}, such that $\vartheta_{r,SpH} \le \vartheta_r \le \vartheta_{r,SpC}$ and $\left|\dot{q}_w\right|$ is minimal.

Result:

Heating is required if $\vartheta_{oa} \le \vartheta_{oa,LmH}$ where $\vartheta_{oa,LmH}$ is the heating limit and

$$\vartheta_{oa,LmH} = \vartheta_{r,SpH} - R_{l,f} \cdot \dot{q}_g \tag{3}$$

$$\vartheta_{sw} = \vartheta_{sw,SpH} = \vartheta_{r,SpH} + \frac{R_t + \tilde{R}}{R_{l,f}} \cdot \left(\vartheta_{r,SpH} - \vartheta_{oa}\right) - \left(R_t + \tilde{R}\right) \cdot \dot{q}_g \tag{4}$$

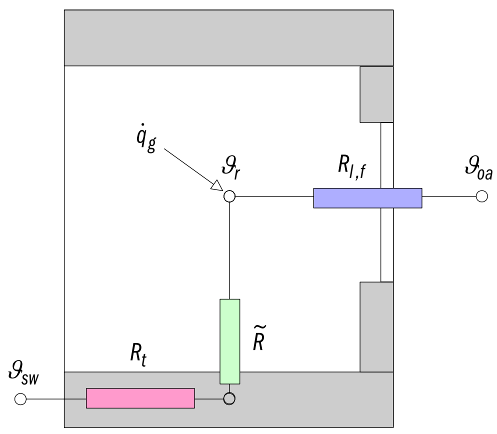
Cooling is required if $\vartheta_{oa} \ge \vartheta_{oa,LmC}$ where $\vartheta_{oa,LmC}$ is the cooling limit and

$$\vartheta_{oa,LmC} = \vartheta_{r,SpC} - R_{l,f} \cdot \dot{q}_g \tag{5}$$

$$\vartheta_{sw} = \vartheta_{sw,SpC} = \vartheta_{r,SpC} + \frac{R_t + \tilde{R}}{R_{l,f}} \cdot \left(\vartheta_{r,SpC} - \vartheta_{oa}\right) - \left(R_t + \tilde{R}\right) \cdot \dot{q}_g \tag{6}$$

Neither heating nor cooling is required if $\vartheta_{oa,LmH} \le \vartheta_{oa} \le \vartheta_{oa,LmC}$.

The result is illustrated by Figure 3. HC is the heating curve and CC the cooling curve.



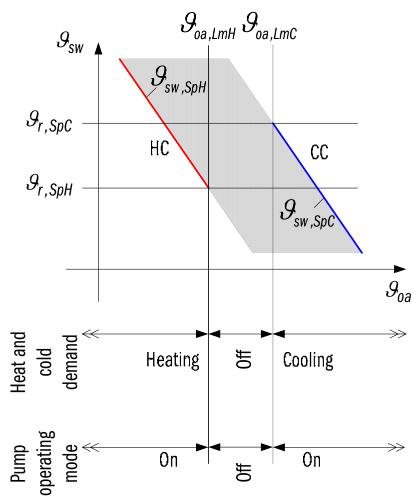


Figure 2 (left): Static building and TABS model

Figure 3 (right): Heating curve (HC) and cooling curve (CC) for known heat gain $\dot{q}_g$

Development step 3:
A next step was to solve another related problem which is a little more difficult. It has been assumed that the heat gain $\dot{q}_g$ in the room is not known but only a lower bound $\dot{q}_{g,lb}$ and an upper bound $\dot{q}_{g,ub}$ of it are known. That means $\dot{q}_{g,lb} \le \dot{q}_g \le \dot{q}_{g,ub}$. This reflects one of the following situations:

- the designer of a controller is uncertain on the value of the heat gain,
- the heat gain in a room is varying very slowly, in a non predictable way and cannot be measured
- the rooms of the zone have different specific heat gains

This more difficult problem (**problem 2**) can also be solved with a static model.
Depending on $\Delta\dot{q}_g = \dot{q}_{g,ub} - \dot{q}_{g,lb}$, which we call the heat gain span, three cases can be distinguished, as can be seen in Figure 4. Case (a): if the heat gain span $\Delta\dot{q}_g$ is small, the solution is similar to that of problem 1. There exists an outside air temperature interval in which certainly neither TABS cooling nor TABS heating power is required. Case (b): for medium heat gain span $\Delta\dot{q}_g$, the heating and the cooling curve overlap, i.e. there exists an outside air temperature interval in which – with knowing only the outside air temperature – it is not possible to determine whether there is heating or cooling demand or no power demand at all. Nevertheless, the supply water temperature feedback controller is able to automatically do the correct action (lowering ϑ_{sw} if it lies above the cooling curve, rising ϑ_{sw} if it lies below the heating curve or water circulation only if ϑ_{sw} lies between heating and cooling curve). Case (c): for large heat gain span $\Delta\dot{q}_g$, it is not possible anymore to maintain comfort for all heat gain situations as the heating curve lies above the cooling curve! In certain cases, the integration of a control loop (with room temperature sensors) may solve the problem. But in most cases, this situation can only be handled by using auxiliary heating or cooling systems and/or additional sensors (not outlined here).
The solution of the problem 2 supplies us with formulas which can be used to calculate the parameters of the heating and cooling curves. The solution leads also to a switching algorithm which can be used in the zone controller to switch between heating and cooling.
The model used so far, where the uncertainty of the heat gains is characterized by bounds, in system theory is called an unknown-but-bounded model. That's why the resulting method to design the TABS and its control has been called **unknown-but-bounded method**.

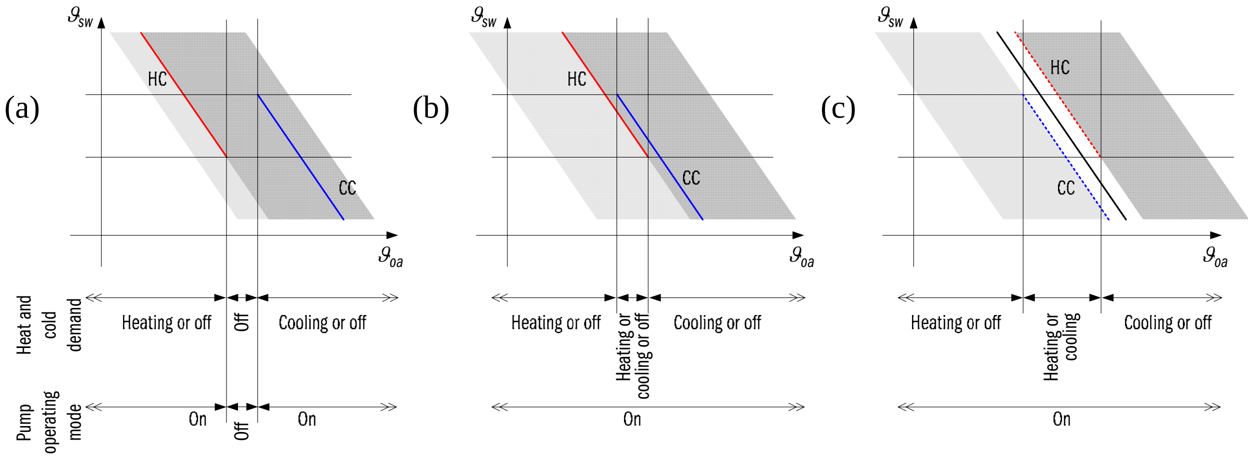


Figure 4: Heating curve (HC) and cooling curve (CC) for different heat gain spans $\Delta\dot{q}_g$

Development step 4:
The problem solved so far differs from the real situation mainly in two points:

- Daily variation of the outside air temperature
- Daily variation of the heat gains. For example considerable internal heat gains during the occupancy time from 08:00 until 18:00 in an office building at weekdays. This usually leads to daily variation of the lower and the upper bound of the heat gains.

It is easy to cope with the first point. Because of the large thermal inertia of TABS it is reasonable that the zone controller takes the average $\overline{\vartheta}_{oa}$ of the past 24 hours of the outside air temperature instead of the actual value ϑ_{oa} to determine the supply water temperature set-point. This averaging of the outside air temperature leads to a slowly varying variable.
To cope with the second point – the daily variation of the heat gain bounds – is more difficult. It has been solved by calculating an equivalent lower bound $\dot{q}_{g,elb}$ and an equivalent upper bound $\dot{q}_{g,eub}$ of the heat gains, which replace $\dot{q}_{g,lb}$ and $\dot{q}_{g,ub}$ in step 3 and which can be used instead of $\dot{q}_g$ in the formulas (4) and (6) to calculate the heating and the cooling curves. To calculate these equivalent bounds a linear dynamic model of TABS and room is used, which represents the dynamic thermal behavior . Further details are given in [5].
What has been done so far in the four steps is the derivation of a first base control strategy for a zone controller – called in analogy to EN 12098-1 “outside temperature compensated flow temperature control” – and of a method to calculate its parameters. Additionally a solution of how to group and assign rooms to zones has been found. Also a method to tune the heating and cooling curves during the first phase of operation has been developed, based on [9].
Development step 5:
Looking for an answer to question Q3 has lead to an option of the base control strategy – called “pulse width modulation” – and looking for an answer to question Q4 has lead to a second option – called “room temperature feedback control”. Finally other base control strategies – e.g. the “outside temperature compensated return temperature control” – have been derived, again with the two options mentioned above. The base control strategies together with the two options altogether lead to a set of control strategies among which a designer may choose [6]. Additional control strategies have been derived for the case where additional auxiliary heating or cooling devices are used.

OVERVIEW OF THE INTEGRATED DESIGN METHOD

The integrated design method is represented by the flow chart in Figure 5. Steps 1 to 6 in this chart represent the straight forward design part. Step 7 is recommended to be performed, when the result of step 6 leaves doubts whether the resulting comfort will be accepted (see example in the next section). A prototype of a design tool which supports to design a TABS and its control according to this method has been developed.

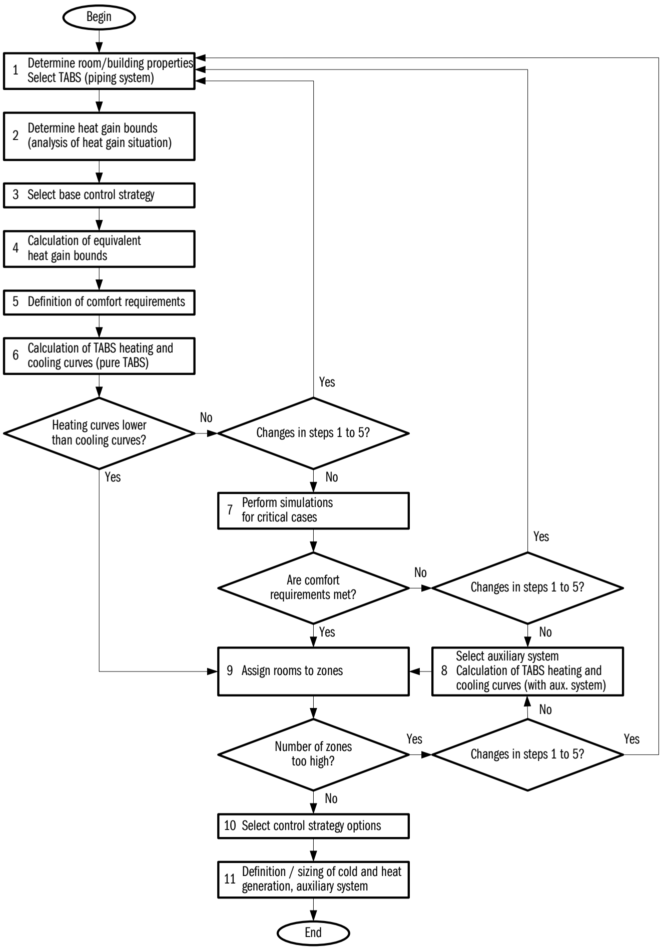


Figure 5: Flow chart of the integrated TABS design method

AN EXAMPLE (Design steps according flow chart in Figure 5)

Design step 1: "Determine room/building properties, select TABS (piping system)"

Table 1. data of the rooms

		Room 1 (normal office)	Room 2 (corner office)
Space length, width, height		6 m x 6 m x 3 m	6 m x 6 m x 3 m
Façade	Orientation	West	West / South
	Area	18.0 m^2	36.0 m^2
	Overall U-Value	0.65 W/m^2K	0.65 W/m^2K
	Glazing fraction façade	42 %	42 %
Additional internal Wall (light)		36 m^2	36 m^2
Natural air change		0.1 h^{-1}	0.2 h^{-1}
Ventilation according to indoor air quality requirements (no cooling/heating by ventilation assumed)			

Table 2. data of the TABS configuration (valid for room 1 and 2)

TABS covering fraction (floor area)	80 %
Thickness concrete slab	250 mm
Pipe spacing	200 mm
External/internal pipe diameter	20/15 mm
Specific supply water mass flow rate [a)]	15 $kg/(h\ m^2)$
Fictitious TABS thermal resistance (R_t) [a)]	0.08 $(m^2\ K)/W$
Thermal resistance of flooring (carpet)	0.125 $(m^2\ K)/W$

[a)] in terms of floor area covered by tabs

Design step 2: “Determine heat gain bounds (analysis of heat gain situation)”

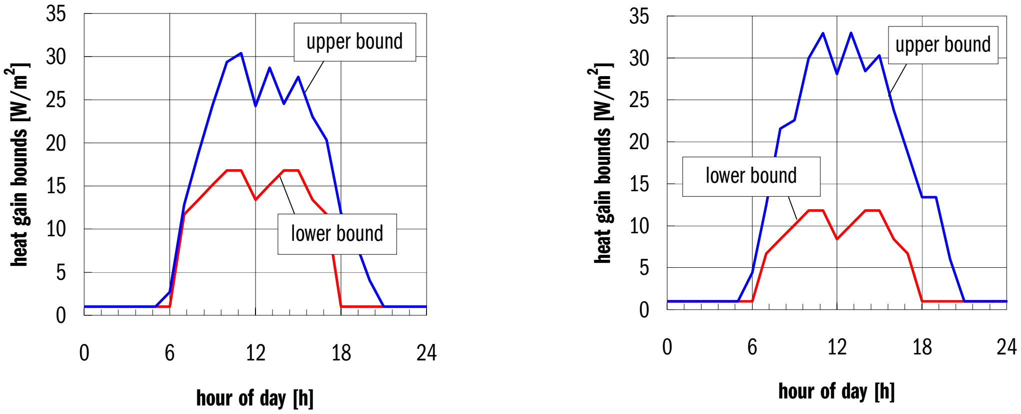


Figure 6: Heat gain bounds for room 1 (left) and room 2 (right), valid for week days

Design step 3: “Select base control strategy”

Here, “outside temperature compensated flow temperature control” is selected.

Design step Step 4: “Calculation of equivalent heat gain bounds”

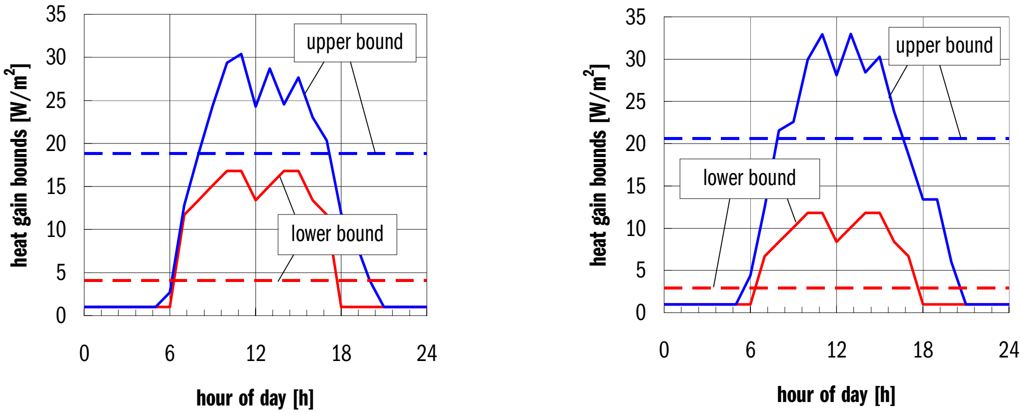


Figure 7: Equivalent heat gain bounds for room 1 (left, dotted lines) and room 2 (right, dotted lines), valid for week days

Design step 5: “Definition of comfort requirements”

Here, a comfort according to [7] is defined (see comfort range in Figure 9).

Design step 6: “Calculation of TABS heating and cooling curves (pure TABS)”

Results (Figure 8) show that room 2 is just at the limit to the critical case (c) in Figure 4.

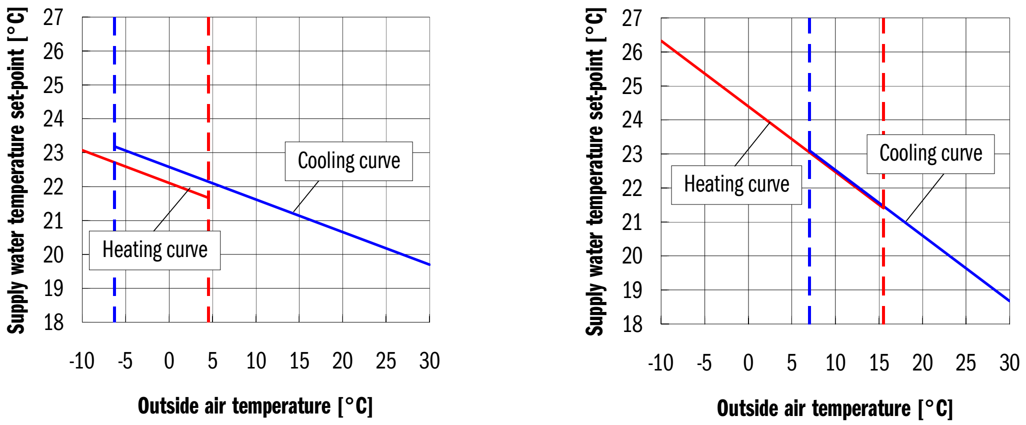


Figure 8: Heating and cooling curves for room 1 (left) and room 2 (right), valid for week days

Design step 7: "Perform simulations for critical cases"

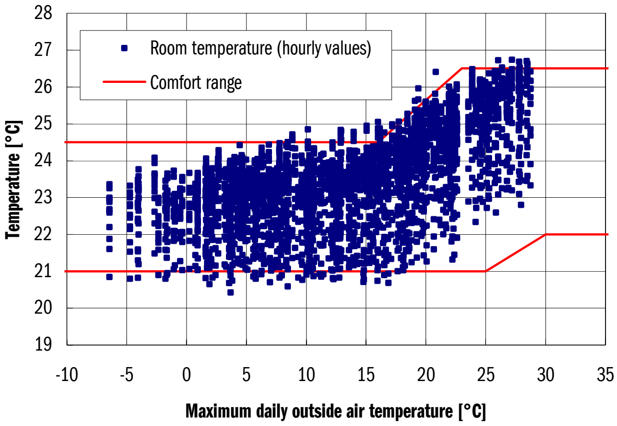


Figure 9: Simulated room temperatures for room 2 (whole-year simulation)

CONCLUSIONS

The new design method is less time-consuming and enables more insight into the potential of TABS. For example, it shows the limitation of TABS application and points out when to install additional auxiliary heating or cooling devices. Additional to the conventional iterative TABS design, the new design process also includes the direct calculation of the parameter values for the underlying base control strategy.
Further details of the method can be found in the design and commissioning handbook which will be published in the frame of our work.

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The Active Utilisation of Thermal Mass of Hollow-Core Slabs

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SUMMARY

The hollow-core slabs can be used actively as a cold and heat storage by conducting supply air through the cores to the space. TERMA project (2004-2006) developed several concepts to integrate thermal mass of hollow-core slabs and air-conditioning systems to meet the requirements of energy efficiency and good indoor conditions. The development was done by using advanced modelling and simulation methods. Thermal behaviour of the hollow-core slab integrated to ventilation was also measured in laboratory. Simulation models were validated against these measurements. The basic strategy for thermal mass utilisation during cooling season is cooling down the slab during unoccupied time by night ventilation to decrease the day time cooling need in the room. By active use of thermal mass of hollow-core slabs the indoor temperature can be reduced at least 2ºC. The active use of thermal mass showed 4..10% savings in heating energy, 46..47% savings in cooling energy and 12..23% savings in fan energy compared to systems traditional systems with passive use of thermal mass.

INTRODUCTION

The thermal mass of the building has a big influence on the energy consumption of the building and indoor temperatures. The present European trend is that the heating and cooling energy consumption is more and more emphasised as a basis for the standards and design work .

Passive use of the building mass has been studied widely. The passive use of thermal mass showed energy savings potential of 4..7% in heating energy and 42..52% in cooling energy /1/. Total energy saving potential can be increased by active utilisation of the building mass. Recently in active utilisation it has been focused on the night ventilation and air circulation trough hollow cores in hollow core slabs. According to Hietamäki et al. /2/ in general night cooling decreases cooling power 40% and cooling energy 20%. With active night ventilation trough hollow core slabs has been decreased 15% cooling energy need.

The hollow-core slabs can be used actively as a cold and heat storage by conducting supply air through the cores to the space. TERMA project (2004-2006) developed several concepts to integrate thermal mass of hollow-core slabs and air-conditioning systems to meet the requirements of energy efficiency and good indoor conditions. The main basis was in Finnish office building. The objective was to achieve 10 % reduction in heating and at least 20% reduction in cooling energy consumption compared to recent calculated level of passive effect of thermal mass.

The development was done by using advanced modelling and simulation methods: spatial level dynamic energy simulations for a whole building, detailed 3D thermal modelling of the hollow-core slab and computational fluid dynamic simulations for indoor conditions. Thermal behaviour of the hollow-core slab integrated to ventilation was also measured in laboratory. Simulation models were validated against these measurements. In the end the LCC survey and design guide were done for developed concepts.

The main objectives of this paper are to present modelling and simulation methods and measurements which have been used to find out the potential concepts. The most potential concepts are presented as well as theirs' influences to the indoor comfort and energy consumption.

METHODS

In the beginning of the TERMA project state-of-the-art study of existing systems was done /3,4/. The project continued with study of possible integrated concepts /5/. According to the preliminary studies the development was focused on the integrated hollow-core slabs and air-conditioning system. Figure 1 shows the development of the TERMA concept.

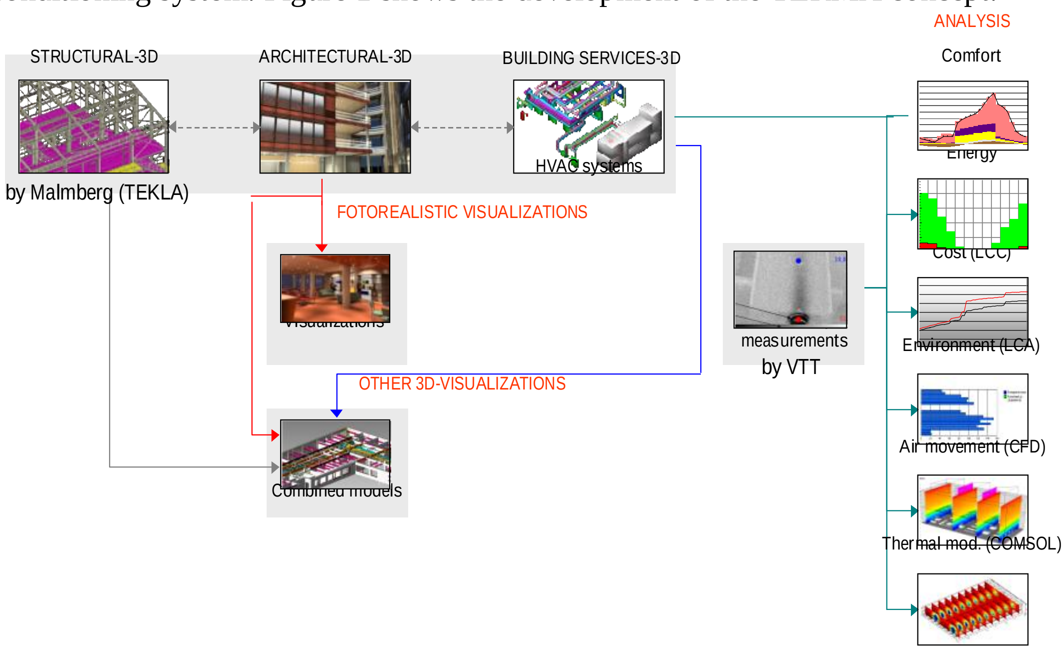


Figure 1. The development of TERMA concepts of utilising advanced modelling, analysis and measurements in TERMA project.

The analysis of the integrated hollow-core slabs and air-conditioning system concepts were done with multiphysic program /6 /. The purpose of the calculations was to determine dynamically in one day period the effect of the air circulated in hollow-core slab to the indoor temperature in south and north faced office rooms in summer conditions. The changing variables were the routing of the air in the slab cores, the temperature of the air conducted to the cores and the supply air temperature from the cores to the room.

The simulations results were indoor temperature, supply air temperature to the room space after core, the temperature distribution of the hollow-core slab and heating and cooling effect of the hollow-core slab to the room space. Figure 2a presents simulated hollow-core slab and 2b the connection between room model and hollow-core slab.

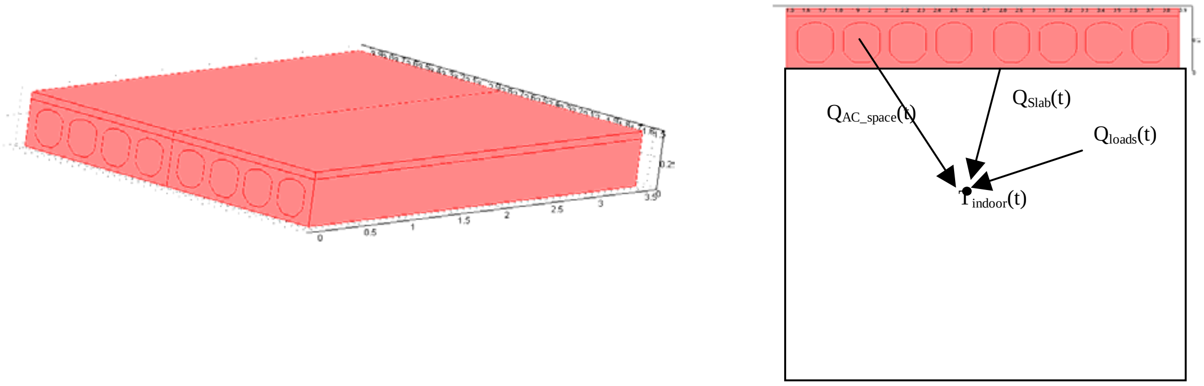


Figure 2. a) Simulated hollow-core slab. b) Connections between hollow-core slab and room model in the multiphysics model.

The energy consumption and heating and cooling needs for office building with different integrated concepts were defined with dynamic energy calculation program /7/ and multiphysics simulation program (Figure 3) /8/.

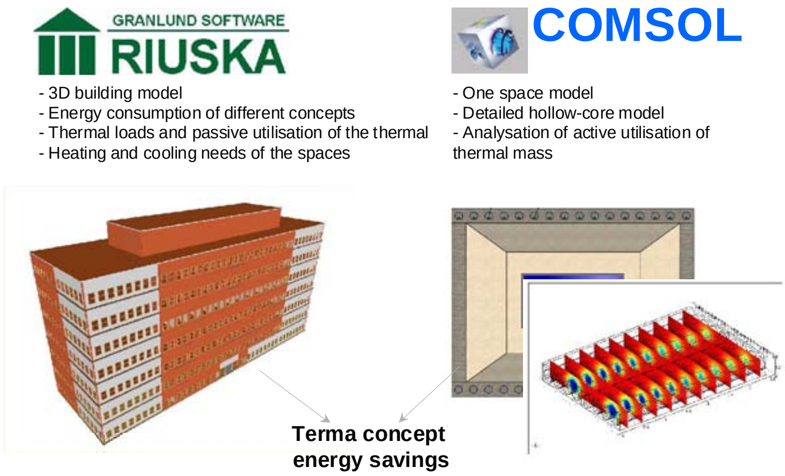


Figure 3. Calculation of the energy consumption and heating and cooling needs.

The energy consumption of the different concepts were compared to typical variable volume air conditioning system (VAV system) and cooled beam system (CB system).

The laboratory tests of the hollow-core slab were performed. The objectives of the tests were to measure and estimate the thermal behaviour of hollow-core slab. One hollow of the hollow-core slab was used as supply air duct of the ventilation system. The inlet supply air temperature was varied like outdoor air temperature in hot summer day (24 h sinus wave model, Tmin=12°C, Tmax=25°C) /9/. The ambient temperature was 20°C. The size of the hollow-core slab was 1200 mm(width) x 400 mm(height) x 10000 mm (length). The measurements were done with two nominal air flows 20 dm³/s and 40 dm³/s in one core. Figure 3a presents the location of the measurement points and figure 3b measurements at 8 m distance.

Measurements of the Terma-slab at VTT Research Hall, Espoo — **Line, x** (distance in meters from the beginning of the slab)

Code	No	Declaration	0	0.5	1	2	3	4	5	6	7	8	9	10	Total
TIx	001	**Air flow temperature inside the main mid-hollow**	1	1	1	1	1	1	1	1	1	1	1	1	12
TIx	002	Air temperature inside the empty right edge-hollow			1				1						2
TIx	003	Air temperature inside the empty mid-hollow			1				1						2
TIx	004	Air temperature inside the empty left edge-hollow			1				1						2
TIx	005	Ambient air temperature above the slab			1				1						2
TIx	006	Ambient air temperature under the slab			1				1						2
TPx	007	**Upper surface temperature inside the main hollow**		1	1		1		1			1			5
TPx	008	**Right surface temperature inside the main hollow**			1		1		1			1			4
TPx	009	**Lower surface temperature inside the main hollow**		1	1		1		1			1			5
TPx	010	**Left surface temperature inside the main hollow**			1		1		1			1			4
TPx	011	Upper surface temperature of the slab, right neck			1		1		1			1			4
TPx	012	Upper surface temperature of the slab, hollow			1		1		1			1			4
TPx	013	Upper surface temperature of the slab left neck			1				1						2
TPx	014	Upper surface temperature of the slab, left (empty) neck			1				1						2
TPx	015	Left edge surface temperature of the slab (insulation)			1				1						2
TPx	016	Lower surface temperature of the slab, left (empty) neck			1				1						2
TPx	017	Lower surface temperature of the slab left neck			1				1						2
TPx	018	Lower surface temperature of the slab, hollow			1				1						2
TPx	019	Lower surface temperature of the slab, right neck			1				1						2
TPx	020	Right edge surface temperature of the slab (insulation)			1				1						2
TPx	021	Floor surface temperature under the slab			1				1						2
TBx	022	Concrete mass temperature above the slab (left neck)			1		1		1			1			4
TBx	023	Concrete mass temperature under the slab (left neck)			1		1		1			1			4
															74
		Total number of measuring points	1	3	23	1	9	1	23	1	1	9	1	1	**74**

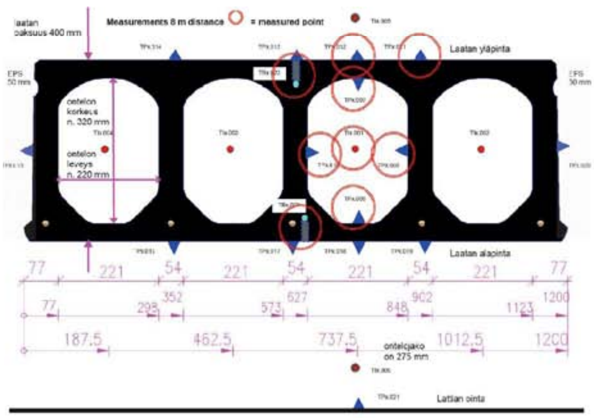


Figure 4.a) The location of the measurement points. b)The measurement at 8 m distance.

The multiphysics model was validated against the laboratory tests.

CFD simulations were performed to ensure indoor comfort in different integrated concepts /10/. The aim of the CFD simulations were to estimate office indoor air flow field on summer design day conditions in case the hollow-core slab were used.

RESULTS

The multiphysic model was validated against laboratory tests. Figure 5 shows as an example the temperature comparison of the supply air conducted trough the hollow-core slab in simulations and tests.

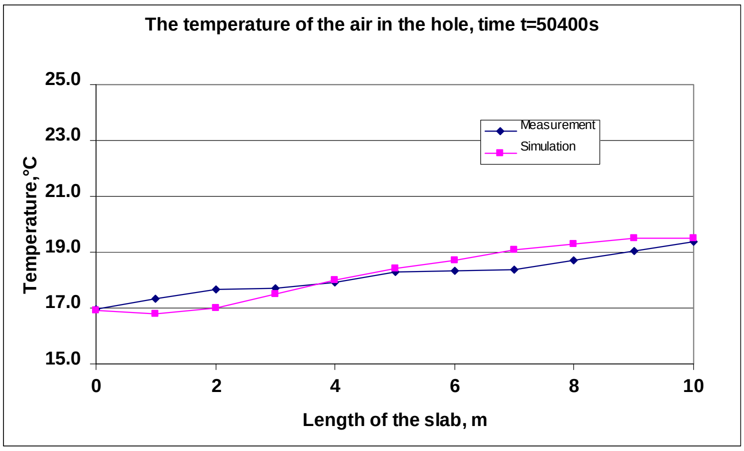


Figure 5. Simulated and measured supply air temperature.

After the multiphysics model was validated, the analysis of the integrated hollow-core slabs and air-conditioning system concepts were performed for 24 different concepts /6/. Figure 6 shows an example of temperature distribution of the hollow-core slab in case where supply air was conducted trough two holes to the space.

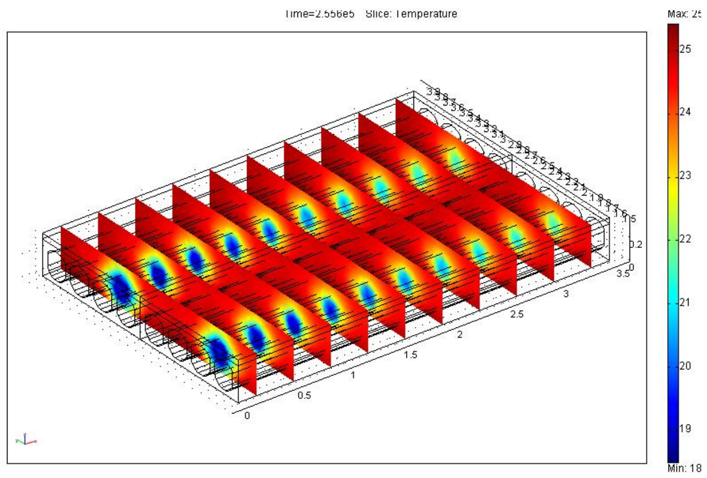


Figure 6. The temperature of the hollow-core structure at midnight.

The results showed that the best indicator for the efficiency of the hollow-core slab as a heat storage is indoor temperature. Even, if the supply air is brought trough one hollow at outdoor temperature the indoor maximum temperature can be dropped 2.2°C. If the air flow is doubled and conducted trough one hollow, the drop in indoor temperature is 6°C in south faced room. By increasing air flow and number of the cores as air channels, indoor temperature can be dropped even 8°C. In north faced room, the indoor temperature can drop from 29.8°C to 24.1°C by using hollow-core slab and increasing air flow. All in all, the excess cooling power is needed at least south faced rooms to achieve indoor temperature 24°C. However, the required cooling power is smaller than in case the hollow-core is not used.

The laboratory tests showed that during the summer night hollow-core slab preheats the incoming cool supply air from +9 °C to +14 °C. Hollow-core slab stores 40 - 70 % of the total cooling energy of supply air. Average storing capacity is 25 W/m² (supply air flow is 40 dm³/s) and 14 W/m² (20 dm³/s) (slab area 12 m²). Less supply air preheating is needed in air handling unit. Most of the time the slab preheats the supply air enough. During the hot summer day hollow-core slab cools down the incoming hot day time supply air from +29 °C to +23 °C. Hollow-core slab stores 60 - 95 % of the excess heat load of hot supply air. Average storing capacity is 20 W/m² (40 dm³/s) and 9 W/m² (20 dm³/s).

Minimum surface temperature of the hollow-core slab in cooling case was +16 ... +19 °C. Lowest temperatures were measured near the supply air inlet of the hollow-core and straight above the hollow-core used as air duct. Surface temperatures do not create any condensation problems or uncomfortable thermal conditions. Even more uniform temperature distribution was measured when extra surface slab was used.

For the full utilisation of the thermal mass it is recommended to use at least two hollow-cores per slab as supply air ducts.

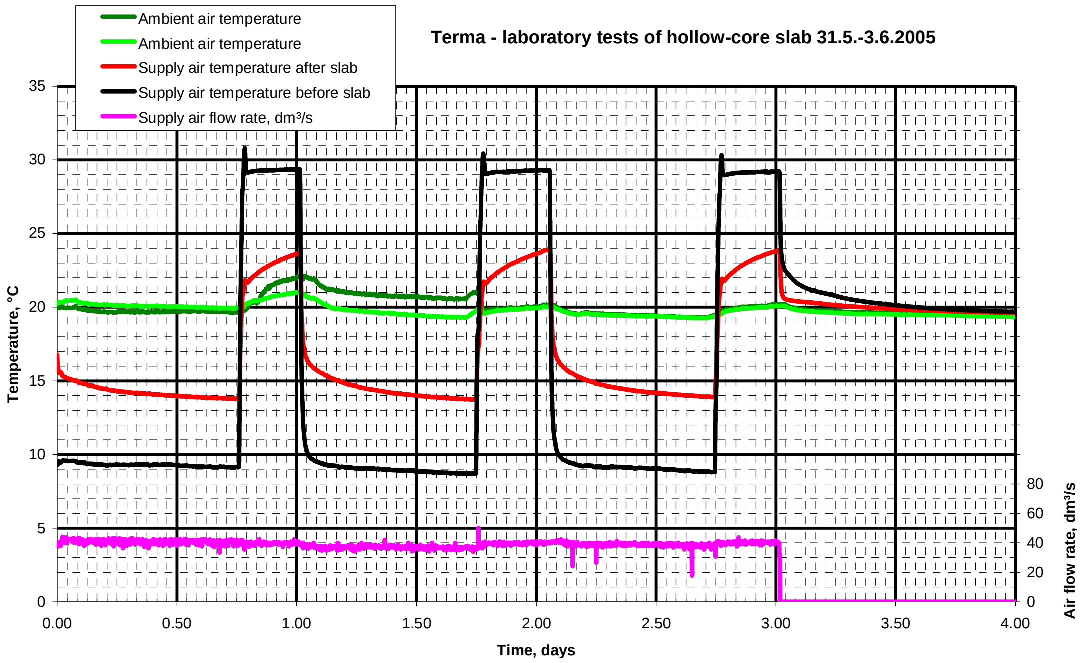


Figure 7. The air temperatures of the hollow-core slab, air flow rate 40 dm³/s, one core as air duct.

After multiphysics simulations and laboratory tests the main TERMA concepts were fixed /11/. TERMA concept utilises thermal mass by air circulation. Both supply and exhaust air distribution and additional night time forced air circulation can be integrated to utilise hollow-core concrete slab (Figure 8). The basic strategy for thermal mass utilisation is:

- *Cooling season*: the slab is cooled down during unoccupied time by night ventilation or forced air circulation to decrease the day time cooling need in the room.
- *Heating Season*: the ventilation air need to be heated less mechanically when the supply air is reheated in the slab by day time heat gains.

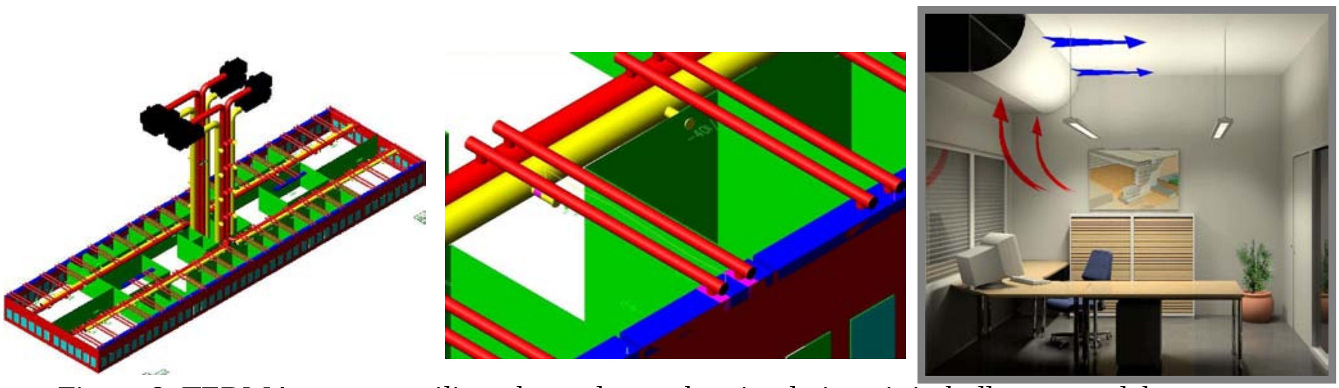


Figure 8. TERMA concept utilises thermal mass by circulating air in hollow-core slabs.

The concepts are divided in four main concepts (TERMA Alfa, TERMA Beta, TERMA Gamma and TERMA Future) which are presented in Figure 9. The characteristics of the different concepts are also presented as well as potential level of energy savings.

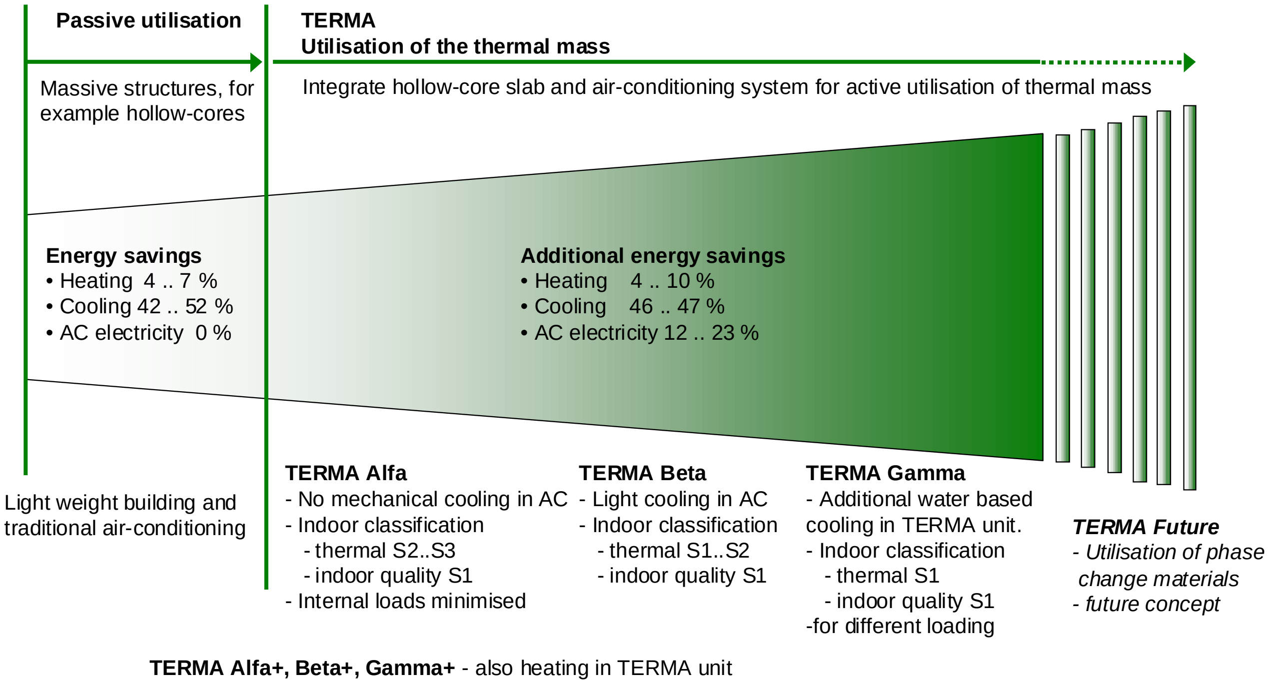


Figure 9. TERMA concepts to fulfil different requirements of building use and indoor comfort.

CFD simulations were used to confirm good thermal comfort for critical parts of occupied zone by using different TERMA concepts. Figure 10 shows the indoor conditions in TERMA Gamma system /10/.

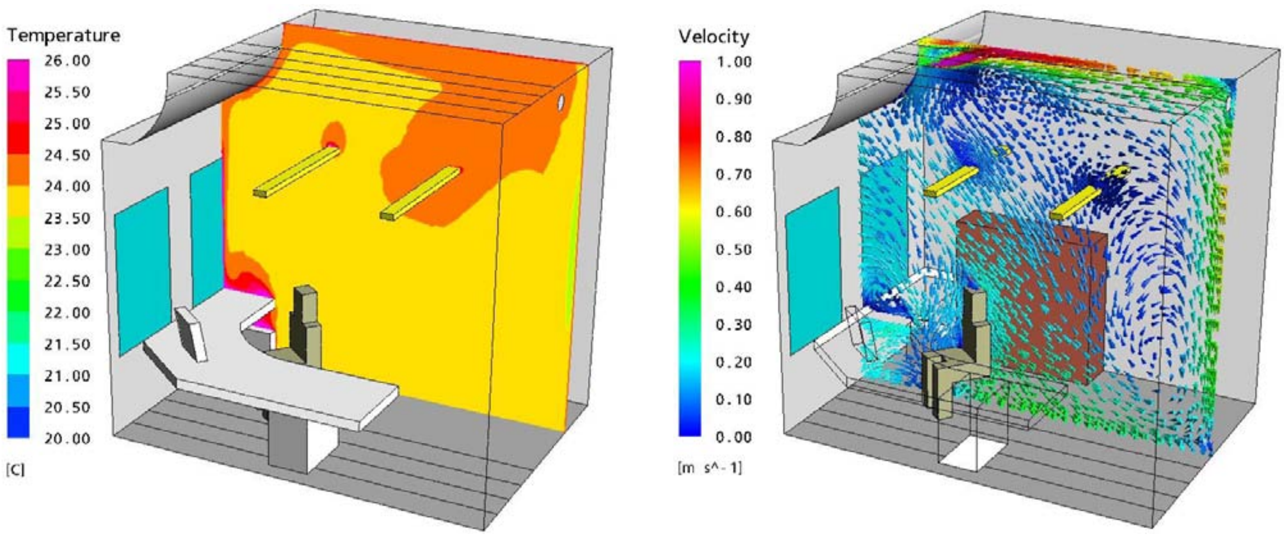
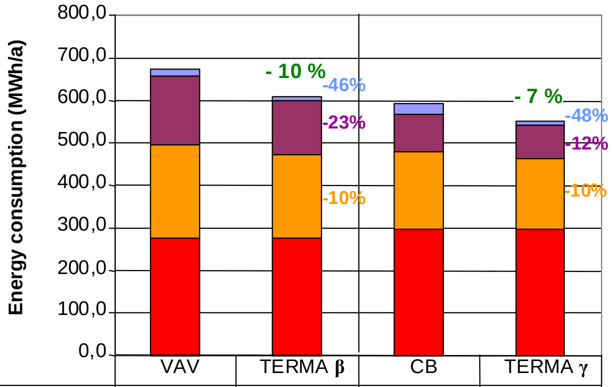


Figure 10. Indoor conditions in TERMA Gamma concept.

TERMA concepts save cooling energy almost 50% (46-47%) compared to traditional variable air volume (VAV) system or cooled beam (CB) system, Figure 11. The saving potential in fan electricity is 12-23% which is indirectly related on cooling, because of smaller air flows in TERMA concepts than in other systems. The heating of the supply air is reduced appr. 10%

and total heating energy of the building appr. 4% compared to traditional AC systems, VAV and CB system /8/.



	VAV	TERMA β	CB	TERMA γ
Cooling electricity	16,5	8,9	24,0	12,6
Fan electricity	164,4	127,4	89,2	78,8
Air ventilation heating	218,6	197,0	181,7	163,8
Space heating	276,3	276,3	297,7	297,7

Figure 11. The energy consumption of the office building.

DISCUSSION

The basis of the concept development in TERMA was in advanced modelling and simulation analysis and laboratory measurements. The simulation models were validated against laboratory measurements. Therefore, the reliability of the hollow-core slab multiphysics simulation was improved considerably. The modelling and simulation gave a good background for development of the new TERMA concepts. The results of the project were good. Four different concepts which integrate thermal mass of hollow-core slabs and air-conditioning system were developed and almost 50% of cooling and 4..10% of heating energy savings potential were recorded.

ACKNOWLEDGEMENT

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Increased Energy Efficiency And Improved Comfort

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SUMMARY

This paper summarises the work of the LowEx co-operation /1/. The aim was to promote rational use of energy by encouraging the use of low temperature heating systems and high temperature cooling systems of buildings. These systems can use a variety of fuels and renewable energy sources. Energy is used efficiently while providing a comfortable indoor climate. Exergy defines the quality of energy and is a concept for designing and assessing different heating and cooling systems. Application of exergy analysis into buildings has not been common before. Low exergy systems are defined as heating or cooling systems that allow the use of low valued energy as the energy source. In practice, this means systems that provide heating or cooling energy at a temperature close to room temperature.

Low exergy systems successfully combine both traditional and innovative new approaches. Research shows that people living in houses with low exergy systems are very satisfied with indoor environment quality. The demonstration projects, in turn, show the wide variety of possibilities to apply low exergy heating and cooling systems in buildings. The Guidebook, which was the final product of LowEx cooperation, is meant to help engineering offices, consultants and architects in their search for energy efficient heating and cooling systems that can provide the occupants with comfortable, clean and healthy environment. In addition, some background information is offered for real estate builders, building maintenance managers, political decision makers and the public at large.

INTRODUCTION

The necessity for an increase in the efficiency of energy utilisation in buildings is obvious and indisputable. Heating, cooling and lighting appliances in buildings cause more than one third of the world’s primary energy demand. Thus, the building stock contributes as a major actor to the energy related environmental problems. “Energy saving” and emission reduction are both affected by the energy efficiency of the built environment and the quality of the energy carrier in relation to the required quality of the energy. Taking into account qualitative aspects of energy leads to introduction of the exergy concept in comparison of systems, which was the key idea of LowEx research programme. LowEx, the international low temperature heating systems research programme (IEA ECBCS Annex 37), was part of the International Energy Agency’s (IEA) Energy Conservation in Buildings and Community Systems programme (ECBCS). The aim of the programme was to promote rational use of energy by encouraging the use of low temperature heating systems and high temperature cooling systems of buildings. In Annex 37 these systems were called low exergy (or LowEx) systems.

Exergy is energy, which is entirely convertible into other types of energy. High valued energy such as electricity and mechanical workload consists of pure exergy. Energy, which has a very limited convertibility potential, such as heat close to room air temperature, is low valued energy. Low exergy heating and cooling systems allow the use of low valued energy, which is delivered by sustainable energy sources (e.g. by using heat pumps, solar collectors, either separate or linked to waste heat, energy storage etc.).

Future buildings should be planned to use or to be or to be suited to use sustainable energy sources for heating and cooling. One characteristic of these energy sources is that only a relatively moderate temperature level can be reached, if reasonably efficient systems are desired. The development of low temperature heating and high temperature cooling systems is a necessary prerequisite for the usage of alternative energy sources. The basis for the needed energy supply is to provide occupants with a comfortable, clean and healthy environment.

EXERGY CONCEPT

What we really should start talking about, because this is what we actually mean, is “saving exergy” and “exergy consumption”. One reason for using the exergy approach and not just settling for the primary energy approach is that it is physically more correct to talk about exergy than energy consumption and production. It is a fact that the term exergy is an unknown term even for scientists not to mention “people on the street”. In order to make this term less strange we need to first of all start talking more about it and stop using the term energy when we actually mean exergy.

In Figure 1 are shown a large enclosure with adiabatic boundaries containing a lot of air at the initial temperature of Ti and a small container of fuel. It is furthermore supposed that the fuel burns in air, heating the surrounding air and environment so that there is a slightly warm mixture of combustion products and air in the final state. It is obvious that the total quantity of energy in the enclosure is the same as in the initial state. But the combination of fuel and air in the initial state has a greater potential to be useful than the warm mixture in the final state. The fuel can be used in a device to generate electricity, do work or heat rooms. But the uses for the slightly warm combustion products are much more limited. It can be stated that the initial potential has been destroyed to a large extent.

The same fact, that there is an energy quality, can be illustrated by another example evident for us from our experience in daily life (Figure 2). It is obvious that 100 kJ electricity stored in a 12 V / 2.3 Ah car-battery is more useful, easier to transform into something useful for us, than the same amount of energy stored in 1 kg water at a temperature of 43 °C in an ambient temperature of 20 °C. The electricity is suitable for running a machine, like a computer, operating a light bulb of 40 W for 42 min or at least heating 1 kg of water with 23 °C. The 100 kJ heat contained in the 1 kg water is only suitable for washing our hands or doing the dishes. It becomes clear that there is a difference between the types of energy. By introducing the term exergy we appreciate the fact that energy manifests itself by its quantity and its quality.

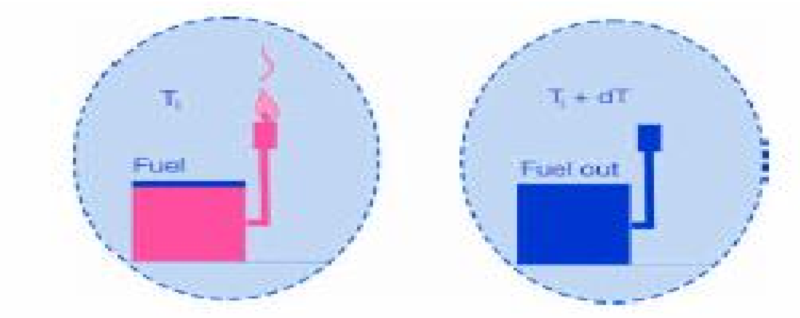


Figure 1. Combustion of fuel in air as an example to show the difference between energy and exergy analysis.

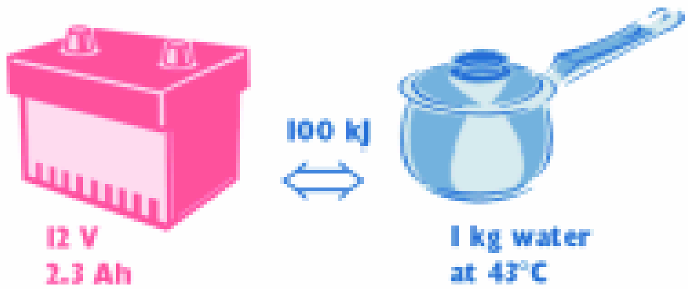


Figure 2. Both systems contain the same amount of energy but not the same amount of exergy.

Despite the efforts made to improve energy efficiency in buildings, the issue of gaining an overall assessment, and comparing different energy sources still exists. Today's analysis and optimisation methods do not distinguish between different qualities of energy flows during the analysis. An assessment of energy flows from different sources is first done at the end of the analysis by weighting them with the primary energy factors. The primary energy factors necessary for the calculation are not based on analytical ground or thermodynamic process analyses, yet they have been derived from statistical material and political discussion. In the theory of thermodynamics, the concept of exergy is stated to be the maximum work that can be obtained from an energy flow or a change of a system. The exergy content expresses the quality of an energy source or flow. This concept can be used to combine and compare all flows of energy according to their quantity and quality. Exergy analysis is commonly used in, for example, the optimisation processes of power stations. This method can be applied to buildings, as well. Most of the energy is used to maintain room temperatures at around 20 °C. In this sense, because of the low temperature level, the exergy demand for applications in room conditioning is naturally low. In most cases, however, this demand is satisfied with high quality sources, such as fossil fuels or using electricity. Exergy analysis provides us with additional information on where and when the losses occur. It helps us to see in which part of the energy chain the biggest savings can be achieved. The need for energy saving is based on the will to reduce the effects of our actions to the environment.

Exergy consumption within the human body becomes higher in a cold environment due to larger difference in temperature between the human body and its surrounding space and also becomes higher in a hot environment mainly due to sweating. It is interesting that the thermally comfortable condition is provided with the lowest exergy consumption rate within the human body. This suggests that rational heating and cooling systems in buildings would work well with low exergy consumption under a condition in which we humans consume as low amount of exergy as possible. That is, we may be able to establish both thermal comfort and low-exergy consuming systems at the same time.

The lowest human body exergy consumption occurs at thermally neutral condition. Figure 3 shows a new relationship between the human body exergy consumption, thermal comfort (PMV*=0), room air temperature and mean radiant temperature. The lowest exergy consumption rate emerges at the point where the room air temperature equals 18 °C and mean radiant temperature 25 °C. This suggests that the use of radiant warm exergy is more effective than the use of convective warm exergy for a heating purpose to realise both thermal comfort and as low exergy consumption within the human body as possible. Such a built environment can be provided by a moderate radiant heating system combined with passive heating strategies, for example, good thermal insulation and suitable thermal exergy storage capacity of building envelopes, solar thermal exergy gain through properly insulated window glazing and others. It is interesting to see that, from the exergetic point of view, there is an optimal combination of room air temperature and mean radiant temperature which results in thermally neutral conditions, namely PMV*=0, although, from the conventional energetic point of view, there are many combinations of room air temperature and mean radiant temperature.

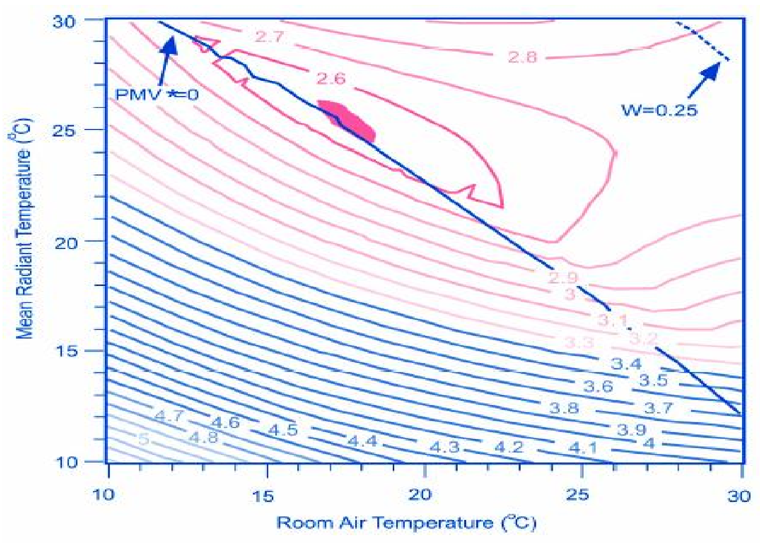
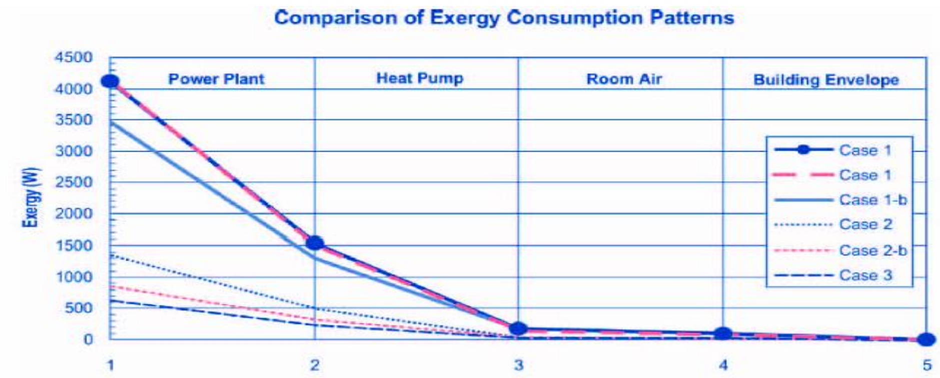


Figure 3. The relationship between exergy consumption within the human body (W/m^2), room air temperature, and mean radiant temperature. The solid line descending from the upper left corner to the lower right corner indicates thermally neutral conditions (PMV=0). The broken line in the upper right corner is skin wetness up to the amount which most people find tolerable (W=0.25). There is an optimal combination of room air and mean radiant temperatures which results in the lowest exergy consumption and thermal comfort.*

EXERGY ANALYSIS TOOLS

To increase the understanding of exergy flows in buildings and to be able to find possibilities for further improvements in energy utilisation in buildings, pre-design analysis tools were produced during the Annex 37 work. It was recognised the importance of developing a simplified tools to visualise why low exergy systems would be advantageous in some energy chains compared to high exergy systems. These tools should be easy to use and show the exergy flow through a system or energy chain (see Figure 4).



Case 1	Base Case - Moderate Thermal Insulation of Building Envelope, Internal Shading Devices
Case 1-b	Improved COP Coefficient
Case 2	Improved Thermal Insulation, External Shading Devices
Case 2-b	Improved Thermal Insulation, External Shading Devices, Improved Electric Lighting
Case 3	Improved Thermal Insulation, COP and Electric Lighting, External Shading Deivices

Figure 4. Example of the output of pre-design tools.

LOW EXERGY TECHNOLOGIES AND SYSTEM CONCEPTS

For future buildings, a minimum of energy at a very low level of temperature difference between the system and the room should be used for thermal conditioning. In this way a maximum of high quality energy (exergy) could be saved. The big efforts made in the field of energy saving in buildings by constructing well-insulated and tight envelopes, sufficient window shading and the use of thermal storage result to a much better usage of the energy. But there is still a big saving potential left. To make the energy use in buildings even more efficient, new low temperature heating and cooling systems are required. The components and systems presented show a step further in this direction. The Guidebook, which was the final product of LowEx cooperation, consists of sixtytwo information sheets, which describe the technologies; their basic principles, technical risks and benefits, advantages, limitations and state-of-art (commercially available, prototype or innovative concept). The idea was to give a quick overview of the possibilities and limitations of the technologies. Some system concepts, which are compiled with these components are also presented.

In recent years system solutions have appeared where heating and cooling is carried out in a holistic system solution where the energy use is planned in a wider and more general perspective. An example of this is the Sensus® building services system (Figure 5). The exergy consumption of the Sensus ® system is lower than in comparable high-standard systems, which also decreases environmental impact during use. Office ventilation employs a Sensus® ventilation unit connected to the Sensus panels with a three-pipe network. The ventilation units utilise surplus heat collected from the rooms with the cooling water system for the heating of intake air whenever heating is needed for the intake air. This conserves heating exergy. The ventilation machine also has an efficient rotating heat collector for the exhaust air (over 70 % heat efficiency). The Sensus® ventilation unit utilises outdoor air for cooling the cooling water for the rooms when outdoor temperature is sufficiently low (under +12–14 °C). This free cooling carried out with ventilation units operates alongside mechanical cooling when necessary. It has a considerably longer annual period of utilisation (over half of the year' s working hours) than conventional free cooling. This lowers the electricity consumption of cooling unit in the Sensus® system in comparison with conventional solutions.

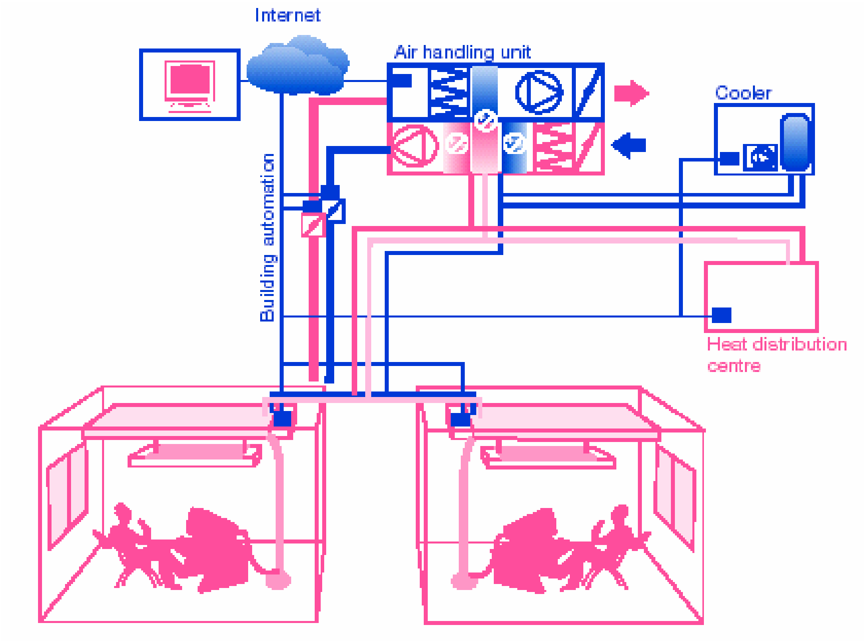


Figure 5. The Sensus system solution.

EXPERIENCES FROM CASE STUDIES

The case examples of the LowEx programme show the wide variety of system applications. They also demonstrate the flexibility of the systems with regard to the energy source. There are 30 examples of LowEx systems in dwellings and offices, but also in a museum, a church and a concert hall. Findings from cases show that the application of LowEx systems provides many additional benefits besides energy supply, such as improved thermal comfort, improved indoor air quality and reduced energy consumption.

One of the critical success factors for the implementation of, for example, low temperature heating systems in residential buildings is the way these systems are viewed and accepted by the occupants. An occupant survey was conducted in the Netherlands in October 1999 among 409 households with these systems. Occupants were asked about their opinion of the indoor climate in terms of whether or not it had improved in relation to the indoor climate in their previous dwelling. Notable was the very positive score for floor and wall heating. Although low temperature heating systems are very well accepted and appreciated, there are some negative aspects and disadvantages that should be taken into account and solved. These are, for example, system controllability per room (floor and wall heating) and the size, design and installation of low temperature radiators. The study shows that lowering the temperatures for heat distribution systems, besides the possibilities of savings in energy supply, gives additional benefits such as:

- thermal comfort increases in many respects (greater share of radiant heat transfer, less temperature gradients, better floor contact temperature, less draught and air turbulence);
- the indoor air quality is also positively influenced (less dust singe and house dust mites, less stuffiness and odours through lower air temperature, less suspended particles);
- in addition to the ability to use low valued energy, savings are gained from better performances of boilers and heat pumps, less piping heat loss and less ventilation losses.

Other benefits might occur, like avoidance of burning risk, extra space due to the absence of radiators, avoidance of mould growth, etc. It can therefore be concluded that LowEx systems are not only preferable from an exergy point of view, people also seem to appreciate the “softer” heat and coolness of the LowEx systems much more than the traditional heating and cooling solutions.

CONCLUSIONS

Exergy defines the quality of energy and is an important tool for designing and assessing different heating and cooling systems. Low exergy (or LowEx) systems are defined as heating or cooling systems that allow the use of low valued energy as the energy source. In practice, this means systems that provide heating or cooling energy at a temperature close to room temperature. Low temperature heating systems or high temperature cooling systems that are suitable for office buildings, service buildings and residential buildings, can use a variety of fuels and renewable energy sources. These systems use energy efficiently while providing a comfortable indoor climate. They should be widely implemented now in order to create possibility to use sustainable energy sources in the near future.

The classical exergy analysis enables to pinpoint the location, to understand the cause, and to establish the true magnitude of waste and loss. Exergy analysis is therefore an important tool for the design of thermal systems since it provides the designer with additional information on where and why the losses occur. The designer can then proceed forward and work on how to improve the thermal system. Application of exergy analysis into buildings has not been usual before. Exergy analysis can also be applied to human body to find optimal thermal conditions. Studies show that the lowest human body exergy consumption occurs at thermally neutral condition. Exergy consumption within the human body becomes higher in a cold environment due to larger difference in temperature between the human body and its surrounding space and also becomes higher in a hot environment mainly due to sweating. These findings suggest that heating and cooling systems may also work well in such conditions where the lowest amount of exergy is consumed by those systems. That is, we may be able to establish both thermal comfort and low-exergy consuming systems at the same time. The human body exergy analyses have now just started to articulate why LowEx systems are essential for creating rational and comfortable built environment.

There are currently many LowEx technologies available. Low temperature systems successfully combine both traditional and innovative new approaches to heating. Usually the heat is transferred into the room through air or liquid circulation systems and the same system can often be used for both heating and cooling. Research shows that people living in houses with low temperature heating systems are very satisfied with ambient indoor air quality. In particular, thermal comfort levels are considered to be higher than in houses with a traditional heating system. Residents also experienced a reduction in draughts and dust, and reported fresher air in houses with low temperature heating systems. By using low temperature heating systems the room temperature can be decreased by a few degrees, which is more energy efficient and healthier for occupants. Low temperature heating systems are sustainable because they are flexible. These systems are not bound to any one energy source and fuel switching does not entail excessive cost. Low temperature systems can utilise a variety of sources of heat including district heat, biofuel, solar energy, gas, oil or electricity, and so the user is not constrained by choices made in the planning phase.

Thorough planning and expert implementation are prerequisites for an appropriate and functional system. System flexibility will be dependent on the choice of appliances and overall system design, which can be difficult and expensive to change after installation. Low temperature heat distribution systems have an operating life of at least 30-40 years during which time the user benefits from the economic advantages offered by flexibility of fuel choice. The life cycle costs of a low temperature heating system are about the same as of a traditional system. Although the initial investment might be slightly higher, the system offers increased flexibility in terms of fuel choice and increased energy efficiency. For example the efficiency of solar heating is considerably higher in a low temperature heating system than in a traditional one.

The demonstration projects show the wide variety of possibilities to apply low exergy heating and cooling systems in buildings. The application of LowEx systems provides many additional benefits besides energy supply such as: improved thermal comfort, improved indoor air quality and reduced energy consumption. These aspects should be further promoted to increase the application of LowEx systems for heating and cooling of buildings. The building regulations and energy strategies should take the quality of energy into account more than today. Wide application of LowEx heating and cooling systems in buildings will create a building stock, which will be able to adapt to use of sustainable energy sources, when desired. Without this ability, the transfer towards an energy-wise sustainable world will be delayed for decades.

The Guidebook, which was the final product of LowEx co-operation, is meant to help engineering offices, consultants and architects in their search for energy efficient heating and cooling systems that can provide the occupants with comfortable, clean and healthy environment. In addition, some background information is offered for real estate builders, building maintenance managers, political decision makers and the public at large. The Guidebook is available as a CD-rom version and also freely available on the internet (http:// www.lowex.net/).

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A Combined Low Temperature Water Radiator and Floor Heating System

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SUMMARY

Space heating load is decreasing in modern Finnish apartments due to better construction of building envelopes (lower U-values for the construction and more tight envelopes) and use of heat recovery in the ventilation. The decreasing load for space heating makes it possible to develop a new combined low temperature water heating system. Such a system includes both radiators in rooms and floor heating in bathrooms. The nominal supply and return water temperatures are 45 ºC and 35 ºC, respectively.

In this study, the performance of an apartment building connected to a combined low temperature water heating system is determined by using dynamic building performance simulation. The results are compared with those for two conventional radiator and floor heating systems. The simulation results show that the combined low temperature water heating system performs well and it is comparable to the other conventional heating systems. It supplies the studied zones by the needed space heating energy and is able to maintain the zones in the required temperature levels showing significantly less temperature fluctuation compared to the floor heating system. This system appears to have the benefits of the two combined systems. The required primary energy for the combined low temperature system is low due to the use of district water heating. The system is also suitable for other types of low temperature heat sources, such as heat pump solutions.

Keywords: low temperature, water radiator, floor heating, heating energy

INTRODUCTION

A common heating system in Finnish apartment buildings is a water radiator system which operates by district heating water. The supply and return water temperatures are 70 ºC and 40 ºC, respectively, at an outdoor air temperature of –26 ºC, which are the design temperatures for southern parts of Finland. The supply water temperature is outdoor air temperature compensated; i. e. the supply water temperature increases as the outdoor air temperature decreases. In typical Finnish buildings, floor heating is expected in bathrooms and toilets. As the supply water temperature is too high for direct use in floor heating, a secondary circuit with a mixing valve is one option to lower the operating temperature. However, this arrangement is expensive and is not practically used. Another option is to use electrical floor heating. Major disadvantages of this method are high consumption of the primary energy and its On/Off switching.

Heat losses are becoming low in modern Finnish apartments due to lower U-values for the construction and more tight envelopes. Besides, heat recovery units, which recover heat form exhaust air, are increasingly used nowadays. All these factors result in a decreasing load for space heating. This situation leads to thinking about developing a new combined low temperature water heating system which includes both radiators in rooms and floor heating in bathrooms. The nominal supply and return water temperatures are 45 ºC and 35 ºC, respectively, at an outdoor air temperature of –26 ºC for southern parts of Finland. This system is simple, easy to install and expected to perform well compared with conventional systems. The application of such a system is mainly in apartment buildings and also possible in detached houses.

The objective of this study is to determine the performance of an apartment building connected to a combined low temperature water heating system by using dynamic simulation. Indoor air temperatures and required heating energy for the building will be compared with simulation results for two conventional radiator and floor heating systems.

Description of the Building

In this study, three flats in a third floor of a four storey residential building is investigated. Fig. 1 shows the building floor plan. The flats have concrete construction with medium heaviness. The U-values (W/m^2K) of the external wall and windows are 0.25 and 1.4, respectively. These values are in accordance with the Finnish National Building Code C3 of 2003 [1]. Good air tightness of the building is considered: the air leakage value n_{50} is 1.0 l/h.

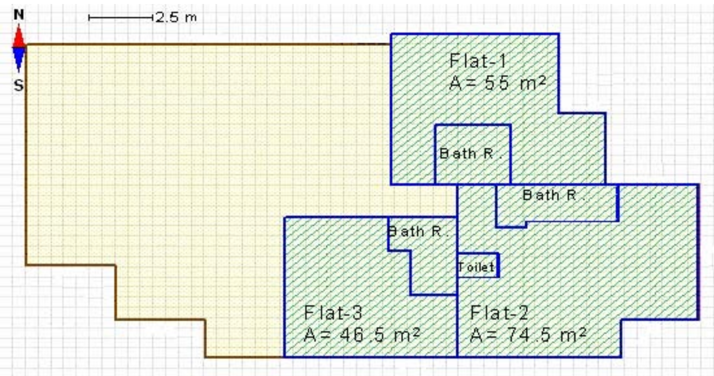

Fig. 1. Floor plan of the studied flats.

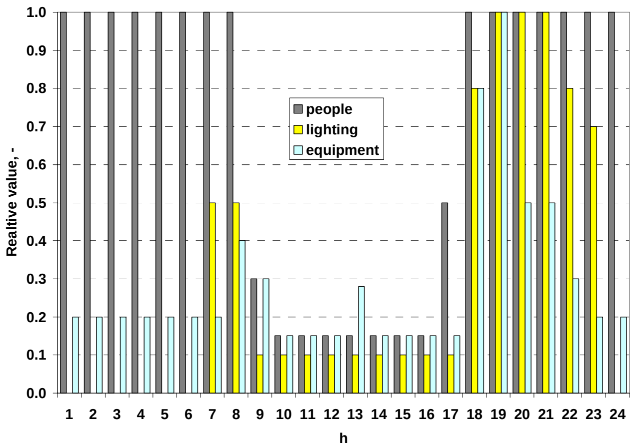

Fig. 2. Relative profile of the internal heat gains.

An hourly profile of internal heat gains (from people, lighting and equipment) for one day is assumed, which is based on averaging more detailed ones for working days and weekends. The relative profile is shown in Fig. 2. The number of occupants in each flat is: 2 in Flat-1, 3 in Flat-2 and 2 in Flat-3. The mean internal heat gain for Flats 1, 2 and 3 is 6.76, 7.80 and 6.07 W/m^2, respectively.

HVAC Systems

Three heating systems are investigated in this study and as indicated in table 1. For System 1, water radiators with conventional water temperature levels are used in the rooms and electric floor heating is used in the bathrooms. For System 2, water floor heating with conventional water temperature levels is used in all zones (rooms and bathrooms). The combined low temperature water system is used in System 3 which comprises water radiators in the rooms and water floor heating in the bathrooms. It is to note that bathrooms in this study refer to toilet rooms as well.

Table 1. Heating systems.

	Rooms	**Bathrooms**
System 1	Water radiators (sizing temperatures: supply/return = 70/40 ^{o}C)	Electric floor heating (Sizing power = 100 W/m^2)
System 2*	Water floor heating (max. water supply temperature is 40 ^{o}C)	Water floor heating (max. water supply temperature is 40 ^{o}C)
System 3	Water radiators with combined low water temperature levels (sizing temperatures: supply/return = 45/35^{o}C)	Water floor heating with combined low water temperature levels (max. water supply temperature is 45 ^{o}C)

* According to the temperature control in the rooms: System 2A is on the indoor air temperature at 21 ^{o}C and System 2B on the floor surface temperature at 24 ^{o}C.

The selected water radiators in the rooms for Systems 1 and 3 is PURMO-Compact type 11 (height = 400 mm and width = 1400 mm) [2]. The number of water radiators in each flat is: 3 in Flat-1, 4 in Flat-2 and 2 in Flat-3. The sizing of the water floor heating system is based on a maximum water temperature drop of 10 ^{o}C and a maximum heating power of 50 W/m^2. The supply water temperature, to the radiators and/or the floor heating, is outdoor air temperature compensated and as shown by Fig. 3.

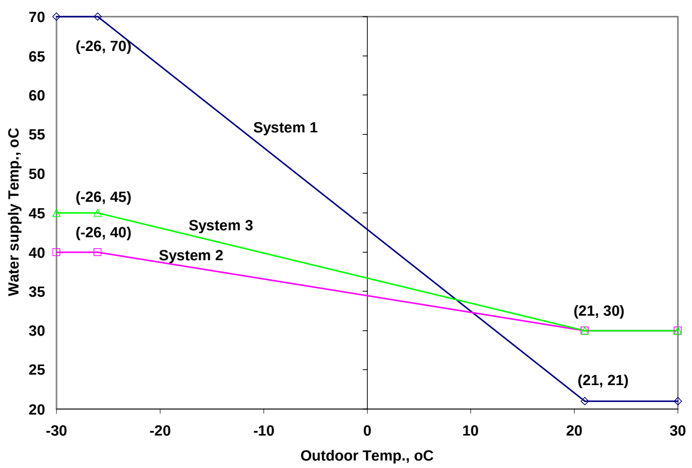

Fig. 3. Hot water supply temperature as a function of outdoor air temperature.

The temperature control inside the rooms is on the indoor air temperature at a set point of 21 ^{o}C. In addition to this control, the all-flat water floor heating investigates another possibility of control which is according to the floor surface temperature at a set point of 24 ^{o}C (System 2B). For the bathrooms, the temperature control is on the floor surface temperature at a set point of 27 ^oC. This temperature keeps the floor surface warm and produces drying effect on the bathroom's floor surface. To reach this temperature, the minimum water supply temperature to the floor heating in Systems 2 and 3 is set at 30 ^oC and as indicated by Fig. 3.

Each flat is connected to one air handling unit (AHU) which comprises two fans (supply and exhaust). The supply fan brings in fresh air from the outdoor. There is a heating coil in each AHU which heats air to a supply temperature of 17 ^oC. The exhaust fan drives the exhaust air through a heat recovery exchanger before it is discharged outside the building. The heat recovery exchanger heats up the air before it passes across the heating coil. The efficiency of the heat recovery is assumed equal to 0.6. The hourly air change rates (ACH) for Flats 1, 2 and 3 are 0.75, 0.69 and 0.89 1/h, respectively.

Normally, there is no cooling system in Finnish apartment buildings (neither inside the rooms nor in the AHU). Instead a window opening action in each room is implemented when the indoor air temperature exceeds 26 ^oC. The building location is in Helsinki. The hourly weather file for 1979 is considered.

METHOD

The performance of the building and the HVAC system is evaluated using the IDA Indoor Climate and Energy 3.0 program (IDA ICE 3.0) [3]. It is a whole-building simulation program with climate processor and simultaneous airflow and heat transfer dynamic simulation. The program can be used for the simultaneous performance assessments of all issues fundamental to a building design: shape, envelop, glazing, HVAC systems, controls, light, indoor air quality, comfort, energy consumption etc. The accuracy of the IDA ICE 3.0 was assessed through the IEA solar heating and cooling programme, Task 22, sub Task C [4]. In addition to heat transfer between adjacent zones, IDA ICE 3.0 calculates air flow through vertical openings and leaks between the zones.

Simulations are carried out to find the indoor air and floor surface temperatures of the rooms and the required energy for space heating for the studied flats. For this purpose the flats are divided into zones. Due to the different heating systems implemented for each zone, the bathroom and the toilet room are considered as separate zones than the rest of the flat which is taken as another zone.

RESULTS

Fig. 4 shows the hourly simulation results for the temperatures of Flat-2 zone (excluding the bathroom and toilet zones). Figs. 4A- 4D display the air temperature which show that it is affected by the characteristics of the implemented heating system. Fig. 4C is the air temperature duration curves and Fig. 4D is the room air temperature in the first week of January. From these figures, it is noticed that using the floor heating system (Systems 2A and 2B) results in higher fluctuations of the indoor air temperature (Fig. 4B) compared with the water radiator system (Systems 1 and 3) (Fig. 4A) which are more stable. This is due to the slow action of the floor heating system because it heats up the floor mass first, and from which heat transfers to the room air. The air temperature with System 2B is higher than with System 2A because the control in the former case is on the floor surface temperature at 24 ^oC, while for the latter case it is on the air temperature at 21 ^oC. For the two systems which use water radiators: the air temperature with the combined low temperature system (System 3) acts in a similar way to that with the conventional high temperature system (System 1). There is only a slight temperature drop between the two cases which is due to the lower inlet water temperature to the radiators for System 3. However, the air temperature inside the rooms is still in the acceptable temperature range (> 20 ^{o}C), which is even better than that for the water floor heating system (Systems 2A and 2B) as the duration curves inform.

Figs. 4E and 4F show the floor surface temperature of the Flat-2 zone. The temperature fluctuations with the floor heating system (Systems 2A and 2B) can be clearly noticed. The higher temperature levels are for System 2B. The floor surface temperature control for System 2B is set at 24 ^{o}C with ± 0.5 K band and as shown by these figures. For the water radiator systems (Systems 1 and 3), the room floor surface temperature follows that for the air temperature and as can be noticed from the results for the first week of January (Fig. 4F).

Fig. 5 shows the air and floor surface temperatures of the bathroom of Flat-2. The floor surface temperature control is set at 27 ^{o}C (with ± 0.5 K band). The proposed combined low temperature systems (System 3), as well as the other systems, appear to operate well in the required temperature range. However, using the electric floor heating (System 1), the floor surface exhibits more temperatures >27.5 ^{o}C (Fig. 5A). The reason could be the higher power rating of the electric floor heater (100 W/m^2) compared with that for the water floor heating (maximum is 50 W/m^2). The systems using water floor heating (Systems 2 and 3) work closely as indicated by the floor surface temperature duration curves (Fig. 5D). This can also be realised from the air temperature inside the bathroom (Figs. 5E and 5F), with higher values for System 2B as before.

The required energy for space heating of the three flats is indicated in Table 2. It is consisted of heating energy to the flats, which covers heating energy to the radiators and floor heating in the rooms and bathrooms, and energy to the heating unit in the AHU. To see the distribution of the heating energy, let us consider System1: the total required space heating energy is 8852 kWh. The heating energy to the AHUs is 2547 kWh (29% of total) and 6305 kWh to the flats (71% of total). This latter is consisted of 2889 kWh (32% of total) by the radiators in the rooms supplied by district heating water and 3416 kWh heating energy to the bathrooms (39% of total) supplied by the electric floor heaters. It is to note that the bathroom heating energy demand is a considerable percentage of the total energy, especially when we know that the bathrooms area is about 16% of the total flats area. This heating energy is used for bathroom heating and it also flows to other adjacent zones due to bathroom’s higher temperature. The conclusion is that the warm floor surface of the bathroom has its expense in terms of heating energy. This is valid for the other studied systems as well. For System 2B, the heating energy to the flats is 11% higher compared with that for System 1, while it is close for the other systems (1, 2A and 3). This is due to the implemented control on the room’s floor surface temperature in System 2B. The heating energy to the flats in Systems 2 and 3 (to the radiators and floor heating) is all supplied by district heating water. The AHU heating coils for all the systems work with district heating water. There is 10% reduction in the heating energy for the AHU in System 2B compared with that for System 1, which is due to higher amount of recovered heat from the exhaust air. As a resultant, the total required space heating energy is close for all the studied systems as there is a only a slight difference (-3% to +5%) form that for System 1.

Besides, table 2 indicates the required primary energy for the studied systems. Typical conversion factors for CHP are implemented to calculate the primary energy from the space heating energy (0.5 for district water heating and 2.0 for electric heating). The latter is applicable for the bathrooms floor heating in System 1. This produces a dramatic increase in the required primary energy for this system with respect to the other systems. The required primary energy for the combined low water temperature system (System 3) is at the lowest level of the investigated systems as appears from the results in table 2.

Table 2. Required energy for the three flats.

	Space Heating Energy (kWh)				Primary Energy (kWh)			
	Flats	AHUs	Total	Ratio to system 1	Flats	AHUs	Total	Ratio to system 1
System 1	6305	2547	8852	1.00	8277	1273	9550	1.00
System 2A	6118	2479	8597	0.97	3059	1240	4299	0.45
System 2B	7024	2291	9315	1.05	3512	1145	4657	0.49
System 3	6108	2460	8568	0.97	3054	1230	4284	0.45

CONCLUSIONS

The simulation results indicate that the combined low temperature water heating system can keep the three studied flats in an acceptable temperature range. Despite of its low water supply temperature (max. 45 ^{o}C), the water radiators can still fulfil the thermal requirements inside the rooms by keeping the air temperature > 20 ^{o}C. Besides, it is capable of maintaining the bathroom floor surface within the required temperature levels. Such a system gathers the benefits of the two combined systems.

The space heating energy provided by the combined low temperature system is comparable to those provided by the other two systems (water radiators with conventional water temperature in the rooms and electric floor heating in the bathrooms, and water floor heating with conventional water temperature in all zones). The required primary energy for the combined low temperature system is at the lowest level of the investigated systems. It is also revealed that bathroom floor heating energy is a significant amount of the total energy heating demand.

The combined low temperature water heating system could be used with decentralised low temperature water production units serving a certain number of buildings. It could also be connected with primary district water heating networks operating with conventional high supply temperatures. In this case, better use of the thermal potential of the primary district heating water could be made which allows lowering its return temperature. This would result in increase of the efficiency of the heat generation plant.

ACKNOWLEDGEMENT

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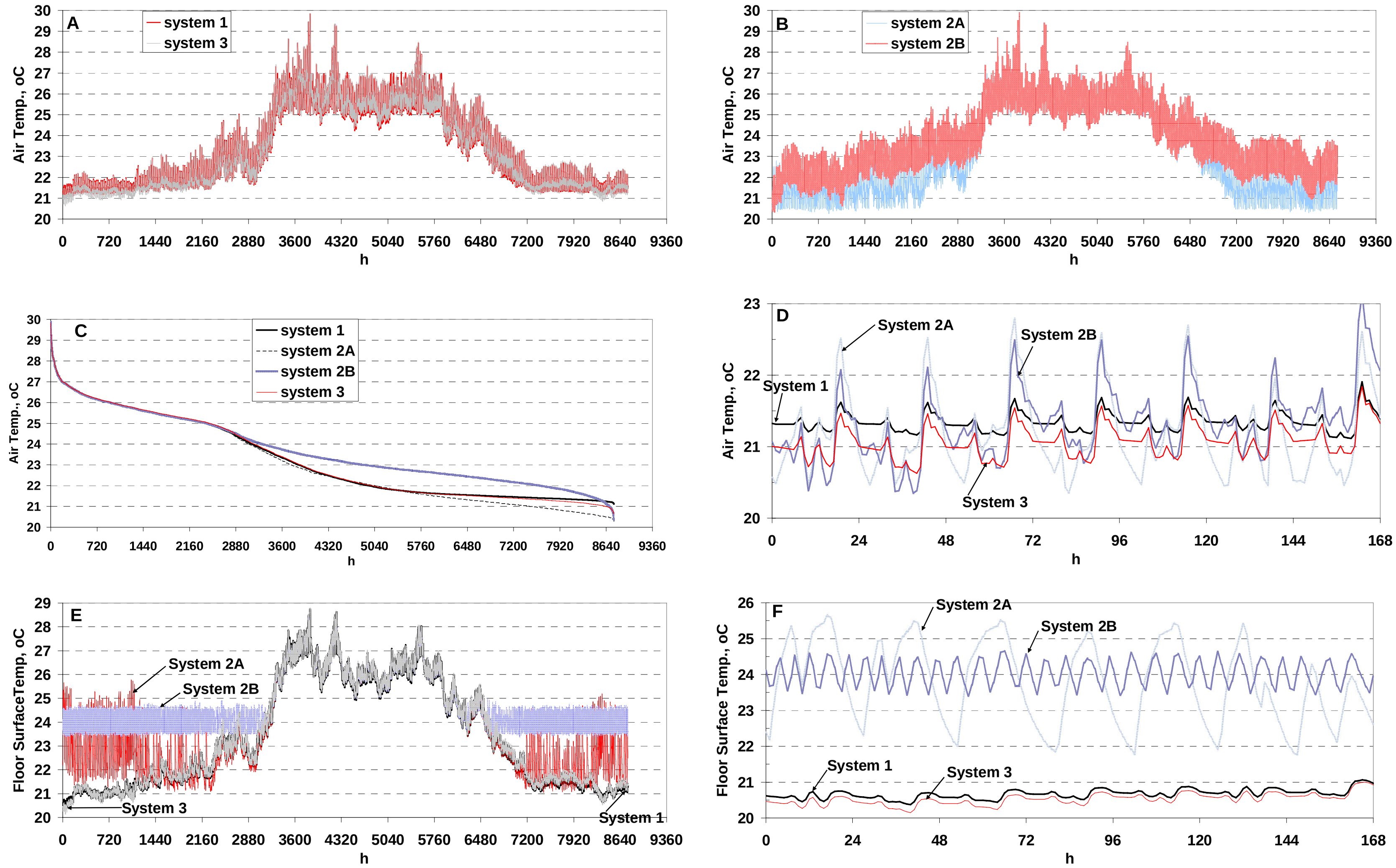

Fig. 4. Air and floor surface temperatures of Flat-2 (excluding bathroom and toilet).

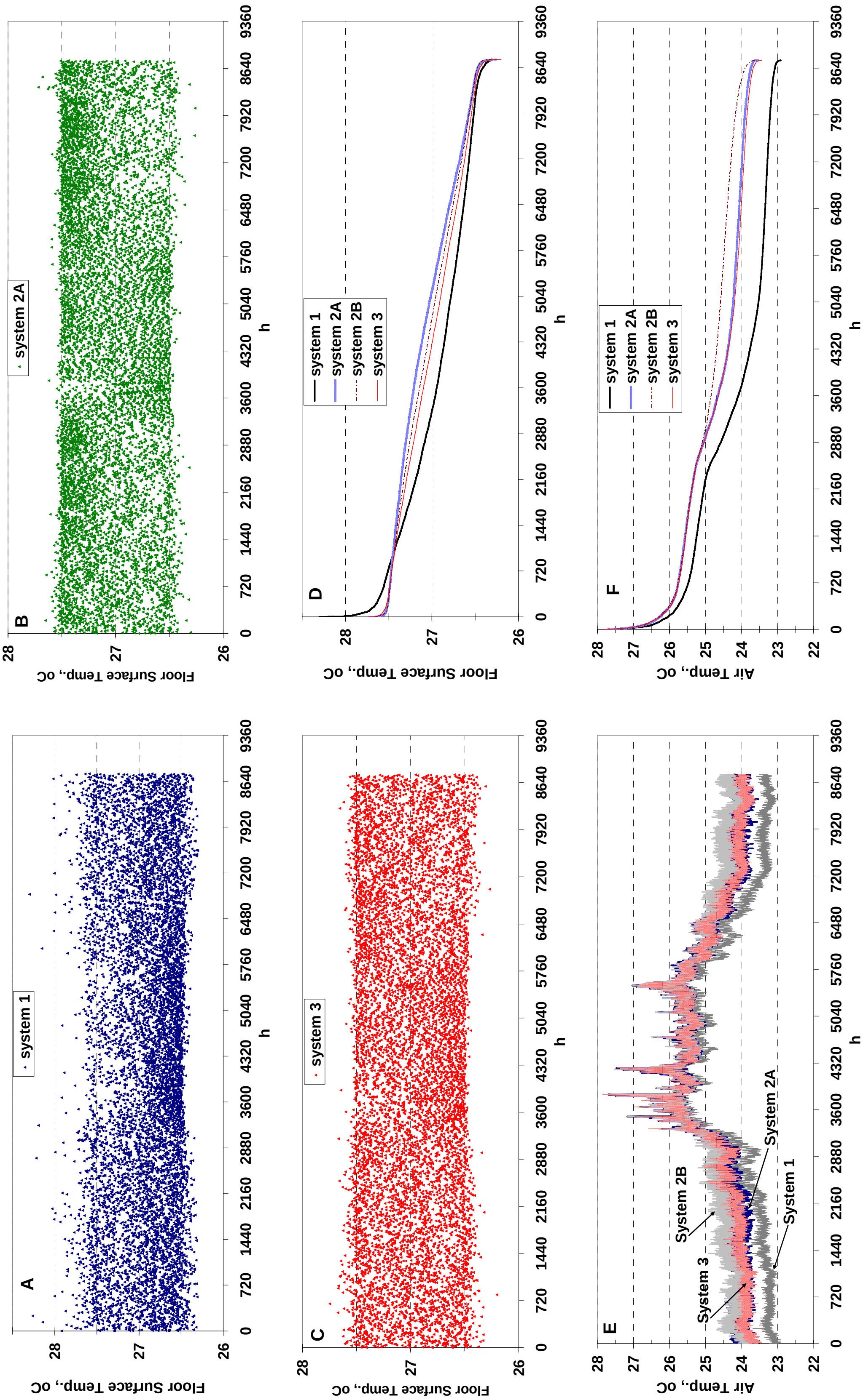

Fig. 5. Air and floor surface temperatures of bathroom of Flat-2.

New European standards for design, dimensioning and testing embedded radiant heating and cooling systems.

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SUMMARY

Due to the increasing use of embedded, hydronic systems for heating and now also cooling of buildings there has been a need to revise the existing European standard EN1264-part2 for floor heating. At the same time a new set of standards EN15377 for these systems has been developed in relation to the many CEN-standards developed for implementing the Energy performance of Buildings directive.
The EN1264 standard series [1,2,3,4,5] have been revised as testing and prove standards for floor heating. A new part is now included for other surfaces (ceiling, walls) and also for cooling. The standard presents a test method either by calculation or by experimental testing. In this way the heating-cooling capacity can be tested under standardized conditions and used for certification.
The new standard EN15377 [6,7,8] includes calculation method for design and dimensioning of embedded radiant heating and cooling systems. For some systems the same calculation methods as in EN1264 is used for design and dimensioning. For other system types not covered by EN1264 new calculation methods are included. A separate part is dealing with systems (TABS) embedded directly in the building mass (slabs). This part shows how to take into account the dynamic behavior of the system.

INTRODUCTION

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The new standard for embedded, water based surface heating and cooling systems, EN15377 consists of the following parts:

— Part 1: Determination of heating and cooling capacity

— Part 2: Design, dimensioning and installation

— Part 3: Optimizing for use of renewable energy sources and dynamic considerations

In Part 1 the steady state heating and cooling capacity is determined calculation in accordance with design documents and a model. The calculation models are listed in prEN1264 part 2 and 5 and in EN15377-1. In the case of special constructions and if necessary, the determination of thermal performance by calculation is combined with a test method according to EN1264-2.
The heating-cooling capacity is given as a function of the temperature difference between room and mean water temperature.
The surface temperature and the temperature uniformity of the heated/cooled surface, nominal heat flow intensity between water and space, the associated nominal medium differential temperature and the field of characteristic curves for the relationship between heat flow intensity and the determining variables are given as the result.
The standard include several methods like general Finite Difference or Finite Element methods, simplified calculation methods depending on position of pipes and type of building structure. The simplified calculation methods are specific for the type of system. The standard is for

systems which are calculable in accordance with EN1264 part 2 and part 5. The simplified methods include certain boundary conditions, which must be met before the given method is applied.

CONCEPT OF THE METHOD TO DETERMINE THE HEATING AND COOLING CAPACITY

A given type of surface (floor, wall, and ceiling) delivers, at a given average surface temperature and room temperature (operative temperature θ_i), the same heat flow intensity in any space independent of the type of embedded system. It is therefore possible to establish a basic formula or characteristic curve for cooling and a basic formula or characteristic curve for heating, for each of the type of surfaces (floor, wall, and ceiling), independent of the type of embedded system, which is applicable to all heating and cooling surfaces.
Two methods are included in this standard:
Different simplified calculation methods are included in for calculation of the surface temperature (average, maximum and minimum temperature) depending on the system construction (type of pipe, pipe diameter, pipe distance, mounting of pipe, heat conducting devices, distribution layer) and construction of the floor/wall/ceiling (covering, insulation layer, trapped air layer, etc). The simplified calculation methods are specific for the given type of system, and the boundary conditions listed in the standard must be met. In case a simplified calculation method is not available for a given type of system, either a basic calculation using two or three dimensional finite element or finite difference method can be applied or a laboratory testing in combination with a calculation may be applied according to EN1264
Based on the calculated average surface temperature at given combinations of medium (water) temperature and space temperature, it is possible to determine the steady state heating and cooling capacity. If proved certificated values for the specific thermal output shall be used, generally EN 1264 part 2 and/or Part 5 apply.

HEAT EXCHANGE COEFFICIENT BETWEEN SURFACE AND SPACE

The relationship between heat flow intensity and the temperature difference between room and average surface temperature (θ_i - $\theta_{S,m}$) is given by equations (1) to (4)) depending on the type of surface (floor, wall, ceiling) and whether the temperature of the surface is lower (cooling) or higher (heating) than the space temperature.

Table 1 - Total heat exchange coefficient (combined convection + radiation) between surface and space, recommended max/min surface temperatures and heating capacity by 20 °C room temperature and cooling capacity by 26 °C room temperature for cooling (EN15377-1, Olesen et. al. 2000 [10]).

		Total heat exchange coefficient W/m².K		Acceptable surface temperature °C		Maximum capacity W/m²	
		Heating	Cooling	Max. Heating	Min. Cooling	Heating	Cooling
Floor	Perimeter	11	7	35	19	165	42
	Occupied Zone	11	7	29	19	99	42
Wall		8	8	~40	17	160	72
Ceiling		6	11	~27	17	42	99

Floor Heating and Ceiling Cooling $\quad q = 8{,}92\ (\theta_i - \theta_{S,m})^{1{,}1}$ (1)

For other types of situations the following relations shall be used:

Wall heating and Wall cooling:	$q = 8\,(\lvert \theta_i - \theta_{S,m} \rvert)$	(2)
Ceiling Heating:	$q = 6\,(\lvert \theta_i - \theta_{S,m} \rvert)$	(3)
Floor cooling:	$q = 7\,(\lvert \theta_i - \theta_{S,m} \rvert)$	(4)

SIMPLIFIED CALCULATION METHODS FOR DETERMINING HEATING AND COOLING CAPACITY OR SURFACE TEMPERATURE

Two types of calculation methods can be applied according to the type of system:
One method is based on a single power function product of all relevant parameters developed from the finite element method (FEM). Another method is based on calculation of equivalent thermal resistance between the temperature of the heating or cooling medium and the surface temperature (or room temperature).
A given system construction can only be calculated with one of the simplified methods. The correct method to apply depends on the type of system, A to G (position of pipes, concrete or wooden construction) and the boundary conditions listed in the standards.

Universal single power function according to EN1264-2.

The heat flux between embedded pipes (temperature of heating or cooling medium) and the space is calculated by the general equation:

$$q = B \cdot \prod_i (a_i^{m_i}) \cdot \Delta\theta_H \qquad (\mathrm{W/m^2}) \qquad (5)$$

where:

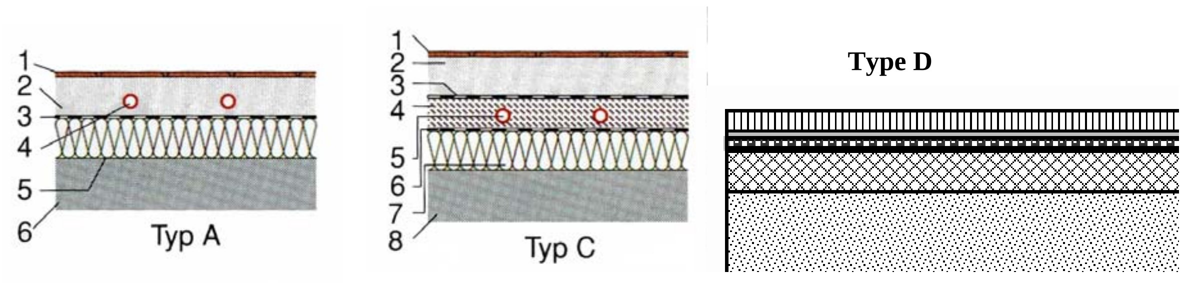
B a system-dependent coefficient in $W/(m^2 \cdot K)$. This depends on the type of system and on the heat exchange coefficient

$\prod_i (a_i^{m_i})$ the power product, which links the parameters of the structure (surface covering, pipe spacing, pipe diameter and pipe covering).

The heat flow density is proportional to $\Delta\theta_H$ where the heating /cooling medium differential temperature is:

$$\Delta\theta_H = \left| \frac{\theta_V - \theta_R}{\ln \dfrac{\theta_V - \theta_i}{\theta_R - \theta_i}} \right| \qquad °C \qquad (6)$$

Where: θ_i = Room operative temperature, °C
Θ_V = Supply water temperature, °C
θ_R = Return water temperature, °C



1-Floor covering; 2-Screed; 3-PE foil; 4-pipes; 5-Insulation; 6-Structure slab; -

1-Floor covering; 2-Screed; 3-PE foil; 4-Scred; 5-pipes; 6-vapour barrier; 7-insulation; 8-structure slab.

Floor covering, load distribution Plane floor section, insulation, Structure slab

Figure 1: System type A, C and D covered by the calculation method in EN1264-2 and 5.

This calculation method is given in EN1264 part 2 for the system types A, B, C, and D (see figures 1 and 2)

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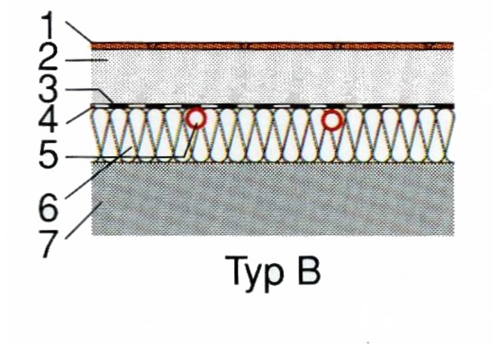
<td>
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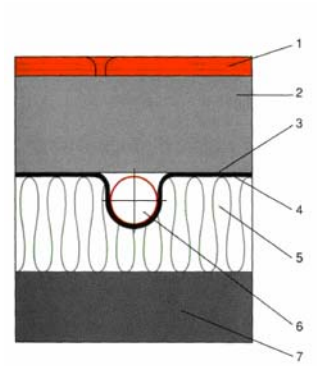
<td>
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<tr>
<td>1-Floor covering; 2-Screed; 3-PE foil
4-Heat conducting plate (device); 5-Pipes
6-Insulation; 7-Structure slab</td>
<td>1-Floor covering; 2-Screed; 3-PE foil
4-Heat conducting plate (device); 5-Pipes
6-Insulation; 7-Structure slab</td>
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Figure 2: System type B covered by the calculation method in EN1264-2 and 5.

Thermal resistance methods

The heat flux between embedded pipes (temperature of heating or cooling medium) and the space or surface is calculated using thermal resistances. The concept is shown in Figure 3.
An equivalent resistance, R_{HC}, between the heating or cooling medium to a fictive core (or heat conduction layer) at the position of the pipes is determined. This resistance includes the influence of type of pipe, pipe distance and method of pipe installation (in concrete, wooden construction, etc). In this way a fictive core temperature is calculated. The heat transfer between this fictive layer and the surfaces, R_i and R_e (or space and neighbor space) is calculated using linear resistances (adding of resistance of the layers above and below the heat conductive layer).
The equivalent resistance of the heat conductive layer is calculated in different ways depending on the type of system. This calculation method, using the general resistance concept, is given in for the following two types of systems shown in figure4 (Type E and F) and figure 5(Type G).
The equivalent resistance of the conductive layer may also be determined either by calculation using Finite Element Analysis (FEA) or Finite Difference Methods (FDM) or by laboratory testing according to prEN1264-2.
The heating and cooling capacity are in some of the described calculation methods determined directly (see EN1264 part 5).
In other calculation methods, the average surface temperature is determined and the heating and cooling capacity is calculated according to:

$$q_{des} = h_t (| \theta_{s,m} - \theta_i |)$$

For evaluation of the performance of the system – and when calculating the total heating and cooling power needed from the energy generation system (boiler, heat exchanger, chiller, etc.) – the heat transfer at the outward (back) side shall also be considered. This heat transfer shall be regarded as a loss if the outward side is facing the outside, an un-conditioned space or another building entity, and it depends on the temperature difference between the pipe-layer as well as the heat transfer resistance to and the temperature in the neighbor space or outside.

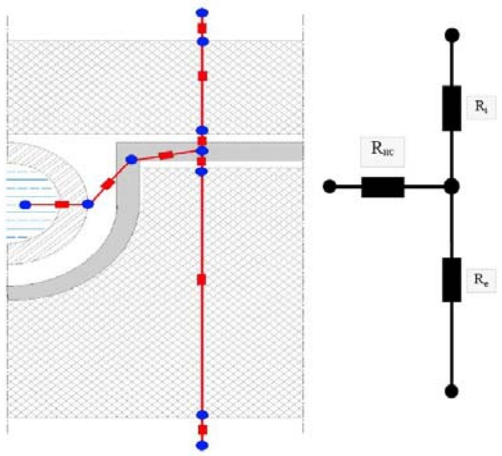

Figure 3 Basic networks of thermal resistances

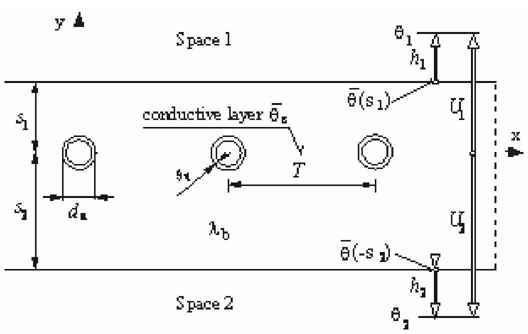

Figure 4a Pipes embedded in a massive concrete layer, Type E (EN15377-1)

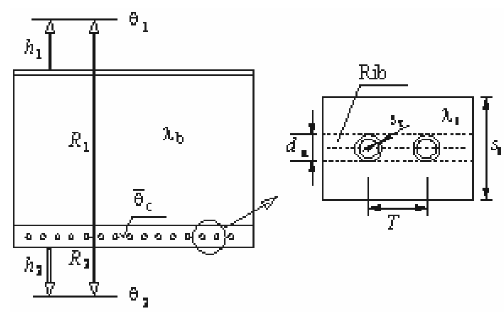

Figure 4b Capillary pipes embedded in a layer at the inner surface, Type F (EN15377-1)

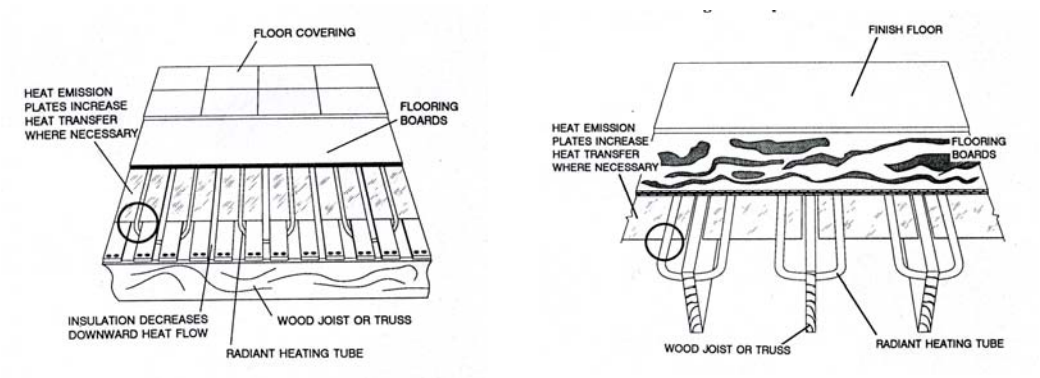

Figure 5 Pipes embedded in a wooden floor construction, Type G (15377-1, [11])

EN15377- PART 3: OPTIMIZING FOR USE OF RENEWABLE ENERGY SOURCES

The aim of this standard is to give a guide for the design of water based embedded heating and cooling systems to promote the use of renewable energy sources and to provide a method for actively integrating the building mass to reduce peak loads, transfer heating/cooling loads to off-peak times and to decrease systems size. A section in the standard describes how the design and dimensioning can be improved to facilitate renewable energy sources. Peak loads can be reduced by activating the building mass using pipes embedded in the main concrete structure of the building (**T**hermo-**A**ctive-**B**uilding-**S**ystems, TABS). For this type of systems, the steady state calculation of heating and cooling capacity (part 1 of this standard) is not sufficient. Thus, several sections of this standard describe methods for taken into account the dynamic behavior. The proposed methods are used to calculate and verify that the cooling capacity of the system is sufficient and to calculate the cooling requirements on the water side for sizing the cooling system.

Thermo Active Building Systems (TABS)

A Thermo-Active-Building-System (TABS) is a water based heating and cooling system, where the pipes are embedded in the central concrete core of a building construction. The heat transfer takes place between the water (pipes) and the concrete, between the concrete core and the surfaces to the room (ceiling, floor) and between the surfaces and the room.
The peak-shaving is the possibility to heat and cool the structures of the building during a period in which the occupants may be absent (during night time), reducing also the peak in the required power (Figure 6). In this way energy consumption may be reduced and lower night time electricity rate can be used. At the same time a reduction of the size of cooling system including chillier is possible

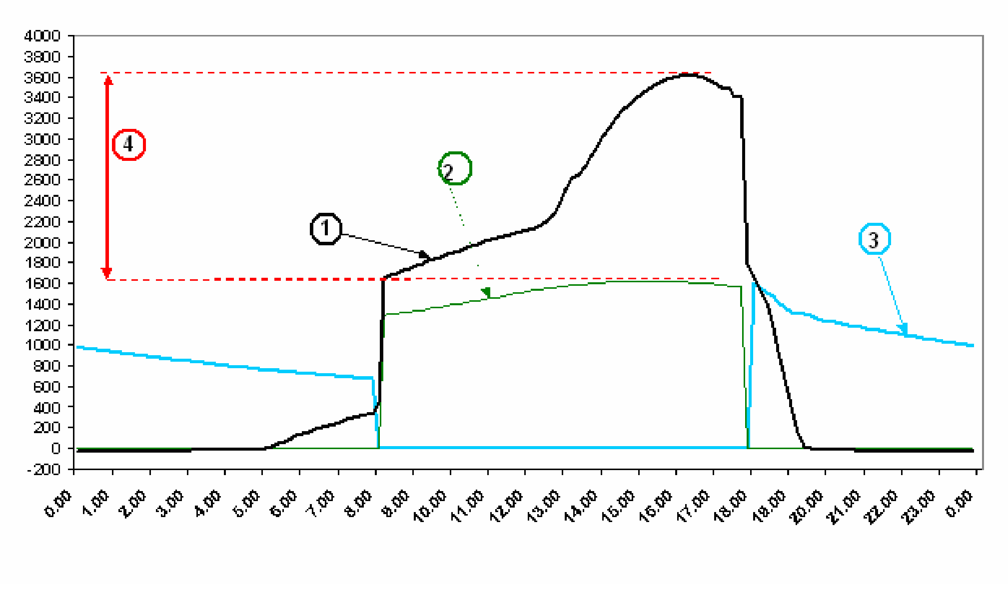

Figure 3 – Example of peak-shaving effect (X-axes: time; y-axes: cooling power W)
1) heat gain, 2) power needed for conditioning the ventilation air, 3) power needed on the water side, 4) peak of the required power reduction

The performance and dimensioning of TABS can be done by full dynamic building simulations with commercial programs including calculation models for embedded pipes. (Olesen and Dossi, [9]). The standard includes a more simplified calculation method. Besides the standard includes diagrams like the one shown in Figure 7 [12]. This simplified diagram give the relation between internal heat gains, water supply temperature, heat transfer on the room side, hours of operation and heat transfer on the water side. The diagrams correspond to a concrete slab with

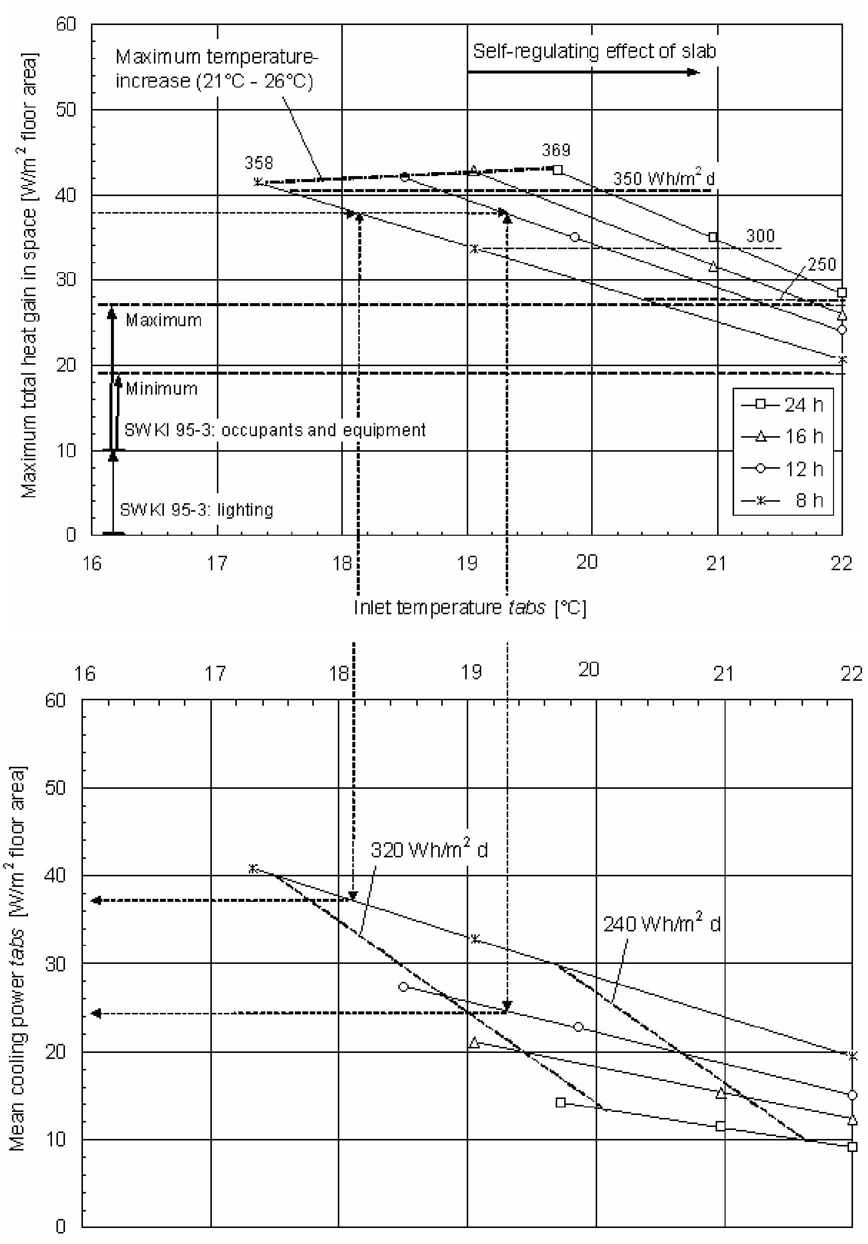

Figure 5 – Working principle of TABS (Koschenz and Lehmann [12]

raised floor (R=0.45 m^2K/W) and a permissible room temperature range of 21 °C to 26 °C. The upper diagram shows on the y-axis the maximum permissible total heat gain in space (internal gains plus solar gains) W/m^2 , and on the x-axis the required water supply temperature. The lines in the diagram correspond to different hours of operation (8h, 12h, 16h, and 24h) and different maximum amount of energy supplied per day Wh/m^2 d . The lower diagram shows the cooling power W/m^2 required on the water side (for dimensioning of chiller) for thermally activated slabs as a function of supply water temperature and operation time. Further, the amount of energy rejected per day is indicated Wh/(m^2 d). The example shows, that by a maximum internal heat gain of 38 W/m^2 and 8 hour operation, a supply water temperature of 18,2 °C is required. If, instead, the system is in operation for 12 hours, a supply water temperature of 19,3 °C is required. In total, the amount of energy rejected from the room is app. 335 Wh/m^2 per day. The required cooling power on the water side is by 8 hours operation 37 W/m^2 and by 12 hours operation only 25 W/m^2. Thus, by 12 hours operation, the chiller can be much smaller. The total heat rejection on the water side is app. 300 Wh/m^2 per day.

SUMMARY AND DISCUSSION

This paper presents a new and a revised European standard for the calculation of heating and cooling capacity for hydraulic, radiant surface heating and cooling systems. Different "simplified" calculation methods, depending on the type of construction, have been presented. In contrast to radiant heating and cooling panels, where the heating/cooling capacity must be determined by testing in a standardized test room, the determination for embedded systems is based on calculations. Besides the included "simplified calculation methods," the standard also allows the use of finite difference and finite element methods. The manufacturers of radiant heating and cooling systems can use the standardized calculation methods to develop diagrams relating water temperature and space temperature to the cooling-heating capacity. This will avoid unnecessary testing of systems.

Besides the new standard includes a part describing methods to take into account the dynamic effects of thermally activated building systems (TABS), where the pipes are embedded in the main building structure (concrete slabs or walls), to activate the building mass.

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Energy savings and thermal comfort with ventilation radiators – A dynamic heating and ventilation system

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SUMMARY

Studies indicate that a high ventilation rate with fresh air supply directly from outdoors gives better thermal comfort conditions, less SBS (Sick Building Syndrome) symptoms and increased work productivity. The drawbacks with a high ventilation rate in natural or exhaust ventilated buildings are normally increased energy use for heating and cold air draught. Such problems may be minimized with ventilation radiators, radiators where cold ventilation air is brought directly from outdoors through a wall channel into the radiator where it is heated before entering the room.

This paper discusses advantages with ventilation radiators in comparison to those of traditional heating systems. Focus has been on energy aspects and thermal comfort. The main conclusions are that ventilation radiators may give a stable and uniform thermal indoor climate. The high thermal gradient between cold ventilation air and the radiator surface inside the ventilation channel also makes the ventilation radiator more efficient than other systems. A method to vary indoor climate on a daily basis according to where people stay is proposed for additional energy savings with ventilation radiators. The deductions were based on results from CFD simulations in a well validated office model.

INTRODUCTION

A CFD (Computational Fluid Dynamics) model of an office room was made to simulate and investigate thermal climate and energy aspects with different heating systems. The goal of the project was to find ways to improve the thermal efficiency of water-based space heating, mainly radiators, and adapt them to low systems temperatures. Low-temperature heating, either by heat pumps or district heating systems, have several positive environmental and economical aspects. Some of them are more efficient energy use, decreased thermal losses in distribution and improved thermal comfort [1-3].

During the study interaction between heating and ventilation systems proved to be an important factor for controlling the thermal climate. Cold draught tends to be the largest hazard, especially in low-temperature systems. The reason is because low temperature systems such as floor and wall heating have weak buoyancy power to counteract cold down-flow from air inlets and windows. With ventilation radiators, on the other hand, the ventilation air is already heated to room air temperature when it enters the room after passing through the radiator. This secures stable indoor climate. The ventilation radiators even proved to perform better in terms of thermal efficiency compared to the other heating systems tested. That is why the focus of the project turned to the evaluation and development of ventilation radiators.

There are still great possibilities for improvement. This paper deals with experiences from the study and explains some advantages with ventilation radiators that can be utilized further in the future.

PRINCIPLE OF VENTILATION RADIATORS

A ventilation radiator is a combined ventilation and radiator system where cold ventilation air passes through the gap between the radiator panels before entering the room. The mean temperature gradient, $\Delta\theta_m$, between the radiator surface and the passing air is larger than in other low temperature radiator systems, just as the heat transfer coefficient, k. The radiator efficiency is improved as heat is more easily extracted from the radiator surface. Figure 1 shows the principle of a ventilation radiator.

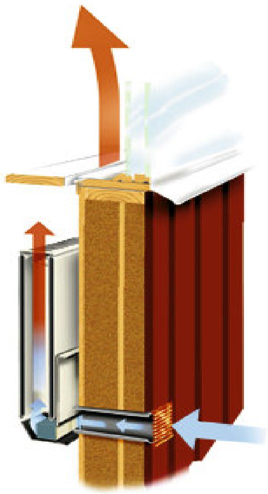


Figure 1. Illustration of a ventilation radiator

Cold ventilation air (blue arrows) enters a gap in the wall because of pressure differences between outdoors and indoors. The air stream is directed between the parallel radiator plates and rises as it is pre-heated to room air temperature.

Because of high efficiency with ventilation radiators (the product of parameters on the right hand side of Equation (1)) the temperature difference between incoming and outgoing water in the radiator circuit, $\Delta\theta$, automatically becomes larger. The water leaving the radiators, $\theta_{\text{water, out}}$, may theoretically achieve temperatures similar to that of the room air where the radiators are placed, or even lower, depending on the mass flow rate, $\dot{m}$, inside the circuit. This is impossible with conventional radiators and very interesting in terms of energy savings and environmental aspects.

$$\dot{m} \cdot c_p \cdot \Delta\theta = k \cdot A \cdot \Delta\theta_m , \qquad (1)$$

where c_p and $\Delta\theta$ are the specific heat capacity of water and temperature difference between water entering and leaving the radiator, $\theta_{\text{water, in}}$ and $\theta_{\text{water, out}}$. The terms on the right hand side, k and A, are the total heat transfer coefficient and the area of the radiator surface, respectively. The mean temperature difference, $\Delta\theta_m$, is given below.

$$\Delta\theta_m = \frac{\theta_{\text{water, in}} - \theta_{\text{water, out}}}{\ln \dfrac{\theta_{\text{water, in}} - \theta_{\text{air}}}{\theta_{\text{water, out}} - \theta_{\text{air}}}}, \qquad (2)$$

where $\theta_{\text{water, in}}$, $\theta_{\text{water, out}}$ and θ_{air} are the water temperature in and out of the radiator and the temperature of ambient air.

METHODS

A CFD model was made as a reproduction of a real life test lab where Olesen et al measured thermal comfort [4]. Even the characteristics of building materials and heating and ventilation systems were replicated. The room, which resembles an office, had an exhaust ventilation system and a window, but no furniture. It was exposed to an outdoor climate similar to that of a normal winter day in Stockholm, Sweden. A sketch of the office is shown in Figure 2 below. The same model was used in two previous studies at the KTH School of Technology and Health. These papers are referred to for more details of the model and simulations [5,6].

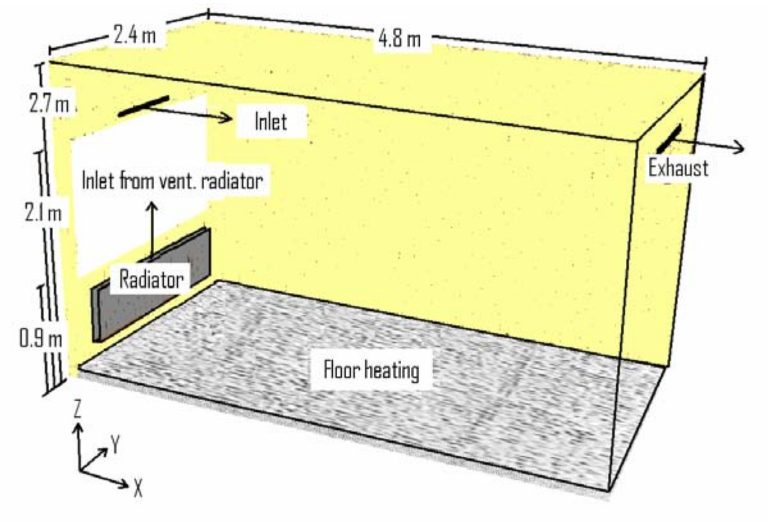


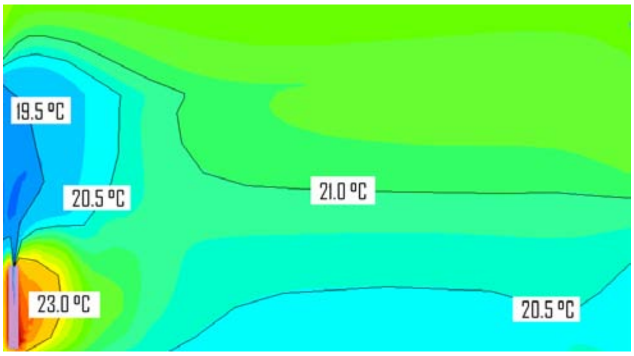
Figure 2. Sketch of the CFD model

About 450 W of heating power was needed to heat the room to a desired comfort temperature. Different heating systems were tested, among them ventilation radiators and a conventional radiator. All radiators had the same size. The heating systems were initially adjusted to give a comfort temperature of 21.0 °C at 1.1 m above floor level in the centre of the room. This did not necessarily mean that the thermal climate was similar for all cases. The aim of the study was to map differences in thermal climate and compare the performance of each heating system. The thermal comfort results were evaluated according to recommendations in ISO 7730:1994, an international standard that specifies conditions for thermal comfort [7].

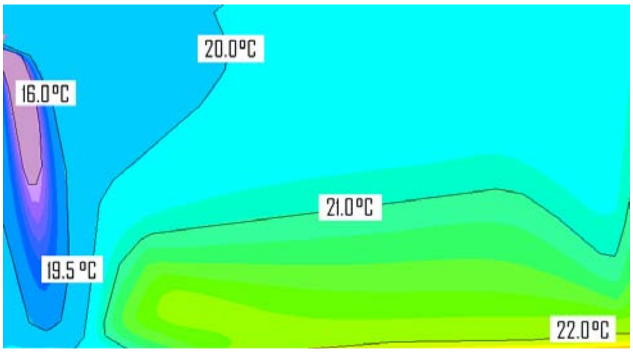
The water flow situation inside the heat emitters was not reproduced. Instead a fixed mean temperature was set for the whole heated surface. This simplification was made to make the CFD simulations less complicated even if a certain margin of error would occur according to theory. In reality the temperature variations over heated radiator surfaces are in linear relation to $\Delta\theta_m$, the mean temperature difference of water entering and leaving the radiator. The size of $\Delta\theta_m$ and the margin of error depend on the mass flow through the radiator. Higher mass flow means more uniform surface temperature.

RESULTS

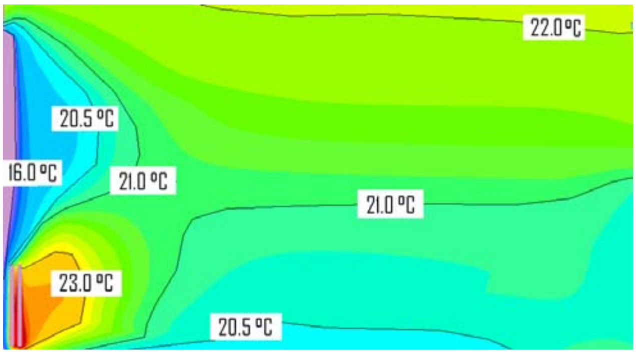
Figure 3 displays simulated thermal comfort results with different heating methods. Comfort temperature is, like the more commonly used operative temperature, a variable used to obtain an understanding of the perceived thermal climate. It considers the balance between radiant heating or cooling and the draught-induced air temperature effects on the perceived air temperature. Table 1 shows various data achieved during the simulations with three different ventilation radiators and a conventional radiator.



Ventilation radiator A, surface temp. 34.5 °C



Floor heating, surface temp. 26.0 °C



Conventional radiator, surface temp. 42.3 °C

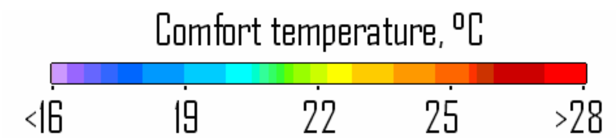


Figure 3 a-c. Comfort temperature with different heating methods

The illustrations are from the XZ plane at Y = L/2. Cold areas are represented by blue colours, while the warm areas are shown in red.

Table 1. Simulation results

	Vent. radiator Case A	Conventional radiator	Vent. Radiator Case B	Vent. Radiator Case C
Temperature of heated surface, °C	34.5	42.3	38.0	34.5
Comfort temp. at Z = 1.1 m and X = L/2, °C	21.0	21.0	21.0	23.2
Flow rate of ventilation air, l/s	7.0	7.0	7.0	3.0
Channel width in ventilation radiators, m	0.015	-	0.04	0.015
Air temp. at Z = 1.1 m and X = L/2, °C	21.3	20.8	21.4	23.8
Total heat output from radiator, W	435	483	445	335
Conv. heat output between panels, W	230	138	196	155
Total heat transfer coefficient, W/(m^2·°C)	9.6	6.7	7.8	8.8
Mean air speed in ventilation channel, m/s	0.93	-	0.35	0.4

DISCUSSION

The following discussion is based on simulation results from our low temperature heating project.

THERMAL COMFORT

Because the ventilation air had already been heated to about 18 °C before entering the room no areas inside the room had problems with cold air draught when using the ventilation radiator in case A. The temperature gradients between floor and ceiling, and between the walls, became smaller than with the other systems for the same reason. Small temperature differences in the room resulted in less air movements caused by buoyancy forces.

RADIATOR EFFICIENCY

The total heat transfer coefficient was 9.6 W/($m^2 \cdot$°C) for the ventilation radiator used in case A and 6.7 W/($m^2 \cdot$°C) for the conventional radiator. It is obvious that a large thermal gradient between radiator walls and cold air in contact with the panels boosts convection heat output. The convection heat emitted from between the radiator panels was 67 % higher for the ventilation radiator compared to the conventional radiator.

In case B and C adjustments were made to the ventilation radiator. The heat transfer coefficient decreased considerably when the channel width was enlarged or the air velocity was decreased in the ventilation channel. As expected from theory the degree of turbulence inside the ventilation radiator was crucial for the heat output. No radiator with convection fins were tested in the study.

THERMAL RESPONSE TIME

The heat output from a ventilation radiator is strongly dependent on the air temperature and air velocity of air passing in the ventilation channel. As a consequence a ventilation radiator has super-short response time and regulates its heat output automatically according to the ventilation rate in the room or changes in outdoor climate. In other water-based heating systems the need for manual adjustments of valves is much greater and the response time is much longer.

EXAMPLE:

When the outdoor temperature and the temperature of air entering the ventilation channel decrease the thermal efficiency of the ventilation radiator automatically increases simultaneously. The ventilation radiator emits more heat as the need for heat in the building increases.

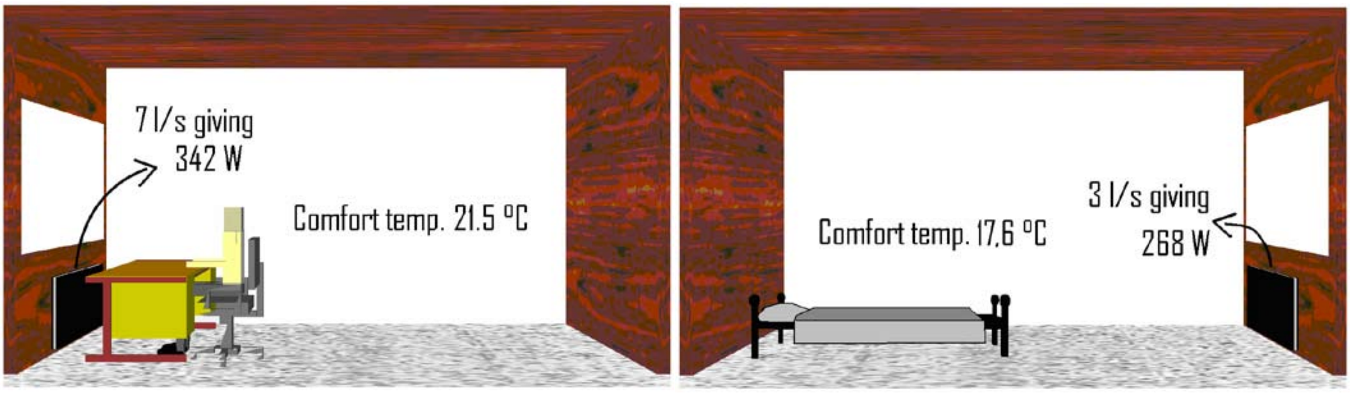
A WAY TO CREATE A DYNAMIC HEATING AND VENTILATION SYSTEM

The super-short response time of ventilation radiators way be used to create a dynamic heating and ventilation system where the radiator heat output is controlled by the ventilation rate in the room.

EXAMPLE:

The ventilation rate in different parts of a house is varied automatically on a daily basis depending on where people tend to stay, sleep and where fresh air and heat is needed. The heat output from the ventilation radiators increases and decreases in perfect synchrony with the variation in ventilation rate. All the time a desired thermal climate is kept where people stay. In the rooms where there are no people the ventilation rate is lower. Here the ventilation radiators do not emit as much heat, but still the room temperature is kept at an acceptable level. This renders energy savings. The example is illustrated in Figure 4.

Day



Night

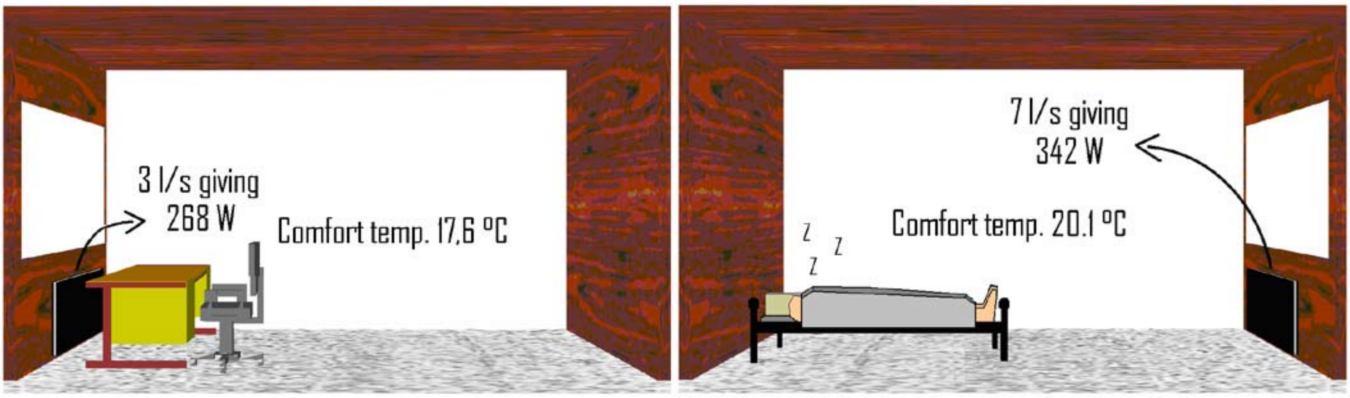


Figure 4. A dynamic heating and ventilation system with ventilation radiators is shown above

The ventilation rate was changed according to where a person stayed. Each illustration shows ventilation air flow, corresponding heat output from the radiator and comfort temperature in the centre of the room at Z = 1.1 m. The upper illustrations show the situation during daytime, while the lower illustrations show the same at night. Heat emitted by the person, 93 W when the person did office work and 70 W when sleeping, was included in the estimations. The radiator had a constant surface temperature of 31.7 °C.

CONCLUSIONS

The thermal climate proved to be more stable and uniform with ventilation radiators than with floor heating and the conventional radiator. Because the ventilation air was pre-heated before entering the room a high ventilation rate could be kept without any hazard of cold draught, even if the temperature of the supply air was below freezing. With conventional heating systems cold draught is often a major problem in wintertime, especially in naturally or exhaust ventilated buildings.

The ventilation radiators had higher thermal efficiency than the conventional radiator. In practice this means that a lower water temperature may be kept in the radiator circuit with the same heat output. This results in energy savings in heat production and distribution. Even more energy may be saved if the short reaction time of ventilation radiators is used to balance the need for heating and ventilation on a daily basis.

The ventilation radiator with the narrowest ventilation channel between the radiator panels had a larger total heat transfer coefficient and thus a better thermal performance than the other ventilation radiators. The reason was higher air velocity and a more turbulent flow inside the channel. This knowledge may lead to development of ventilation radiators with design differing from that of traditional radiators. It is likely that new types of slim double panel ventilation radiators attached close to the wall may be interesting both in terms of energy usage and visual aspects. To find the ideal geometries for increased convection heat output the degree of turbulent air flow in the ventilation channel should be optimized in relation to pressure losses and noise. This process should include calculations that show whether convection fins would be functional to have inside the ventilation channel.

ACKNOWLEDGEMENT

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Modeling of Heating Systems and Radiators in Combined Simulations

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SUMMARY

In this paper different approximation levels for the simulation of heating systems and radiators in a test chamber are presented and evaluated. It is shown that for more complex radiator types a detailed combined simulation including CFD should contain an appropriate model of the heating system. In addition, the level of approximation depends on the target of the calculation, e.g. thermal comfort or energy consumption, and on the geometrical complexity of the investigated radiator, respectively. Therefore a simulation tool was created which provides the ability to handle each desired level of approximation from simple tempered panels up to the fully coupled geometrical radiator models including internal heat conduction and water flow. Results of the calculations regarding the three different approximation levels are compared with each other as well as with experimental data in order to show the quality of the developed models.

INTRODUCTION

Combined simulations consisting of thermal building simulation, simulation of the technical equipment, and computational fluid dynamics (CFD) are very useful to get numerical data about thermal comfort, air quality, and energy consumption of rooms and buildings.
Although this kind of simulation is known to be already detailed there are still some options for advanced modelling. In case of technical equipment in rooms, it can be approximated on different levels. The main goal of our investigations was to generate a simulation tool which provides for the possibility to increase the level of approximation up to the fully modelled radiator configuration, including three dimensional heat conduction, geometrically highly resolved radiation, and in addition the internal water flow.
By means of the fully coupled simulation tool developed for this purpose one should be able to investigate geometrically very complex radiator models, both under standard conditions as well as under the dynamic conditions of real practical applications. The tool could especially be used to substitute the construction of different radiator prototypes and in this manner can yield enormous cost savings.

METHODS

Overview about the simulation tool

The designed simulation tool consists of three highly coupled program parts to assure that all relevant physical and technical aspects for the simulation of radiators are considered. In detail it consists of the thermal building simulation code TRNSYS® [1], the indoor air flow simulation code ParallelNS [2], and the commercial CFD code Fluent® [3]. Fluent® is applied for

simulation of the interior of the radiator, ParallelNS for the air flow calculation, and TRNSYS® for the radiation, the heat conduction of the surrounding faces, the controlling and the technical equipment.
Due to the coupling within the transient simulation process these three single codes together result in a very complex simulation tool which is able to satisfy the demands of practical applications, like radiator test chambers. In the following section the three modules and the coupling algorithm are described in more detail.

Modules

Thermal building simulation and control - TRNSYS®

TRNSYS® was developed at University of Wisconsin, Madison (USA). It is a commercially available simulation tool with modular structure, especially designed for the evaluation of the transient performance of thermal energy systems. The modular and to some enhancement open structure allows to extend the available modules, or to add new features, respectively. Based on version 14.2, the program was developed at TU Dresden as the building simulation unit for combined simulations, see [4], [5]. It can communicate with different CFD programs via PVM (Parallel Virtual Machine), in order to get detailed information about the convective heat transfer or about parameters of technical equipment.

Airflow and coupling - ParallelNS

The CFD code ParallelNS is used for the simulation of the indoor air flow, e.g. in the simulated test chamber. It is a research code developed at Göttingen University and Dresden University of Technology. Based on the Reynolds-averaged Navier-Stokes equations for incompressible non-isothermal fluids two turbulence models, a k-ε model and a φ-f-k-ε model are used for calculating the effects of turbulence. In the case of the k-ε model special care is devoted to the heat transfer at rigid walls by applying especially developed boundary layer approximations. This is of great importance for the investigations in terms of energy consumption. Moreover it is possible to consider a wide variety of additional transport equations, like the equation of age of the air or mass transport equations, respectively [6].
The discrete model was created by means of a stabilized Finite Element method, based on linear tetrahedron elements. Using this method one gets some advantages for specifying boundary conditions at openings, for details see e.g. [7] or [8]. The code itself can be parallelized very easily by a non overlapping domain decomposition method based on the public domain software package PVM. The same feature is applied for the interaction with the thermal building simulation program and also with the radiator simulation in Fluent®. Because ParallelNS has already applied PVM for domain decomposition, it now represents the master part of the whole coupled simulation tool.

Water flow and radiator - Fluent®

In case of fully coupled problems including the extended radiator model, the radiator itself is completely modelled and simulated by use of the commercial CFD code Fluent®. Fluent® is a finite volume code which allows for the calculation of a huge variety of flow types like non-isothermal unsteady compressible flows or flows with highly temperature dependent properties, as for instance the water flow in radiators. Moreover it is possible to simulate solid parts with thermal conduction within the flow field which are calculated together with the flow solution and are required to simulate the metal parts. Due to the ability to handle hybrid mesh types it is possible to model very complex three dimensional geometries, thus also the interior of multi-section radiators, see example I. The coupling with the building simulation was done by means of the implemented user programming interface, the so called “user defined functions” (UDF). Therefore the ability to exchange data via PVM was implemented in the UDF-code. For more information see the Fluent® user manual [3].

Coupling algorithm

The complete coupling algorithm works transient, according to the type of successive approximation. Because there are no inner iterations inside each time step between the different simulation tools the algorithm can bee seen as an explicit time integration procedure. Concerning this matter and in order to avoid divergence problems it is important to be careful when defining the size of the main time steps (in TRNSYS®).
The main program which is responsible for the entire data transfer between the three simulation codes is ParallelNS. At first, ParallelNS starts and initialises the thermal building simulation code. After a period of time of “stand alone” simulation within TRNSYS® the coupling with the air flow solver ParallelNS starts, in order to generate useful initial values. The coupling itself works across the transfer of boundary condition (BC) values at predefined BC interfaces. To ensure a realistic approximation of the interaction among the wall faces of the surrounding walls and the flow field, all the walls are split into a number of such BC interfaces. For the flow solver, the BC interfaces are additionally further refined to gain more stability due to the use of mean BC-values.

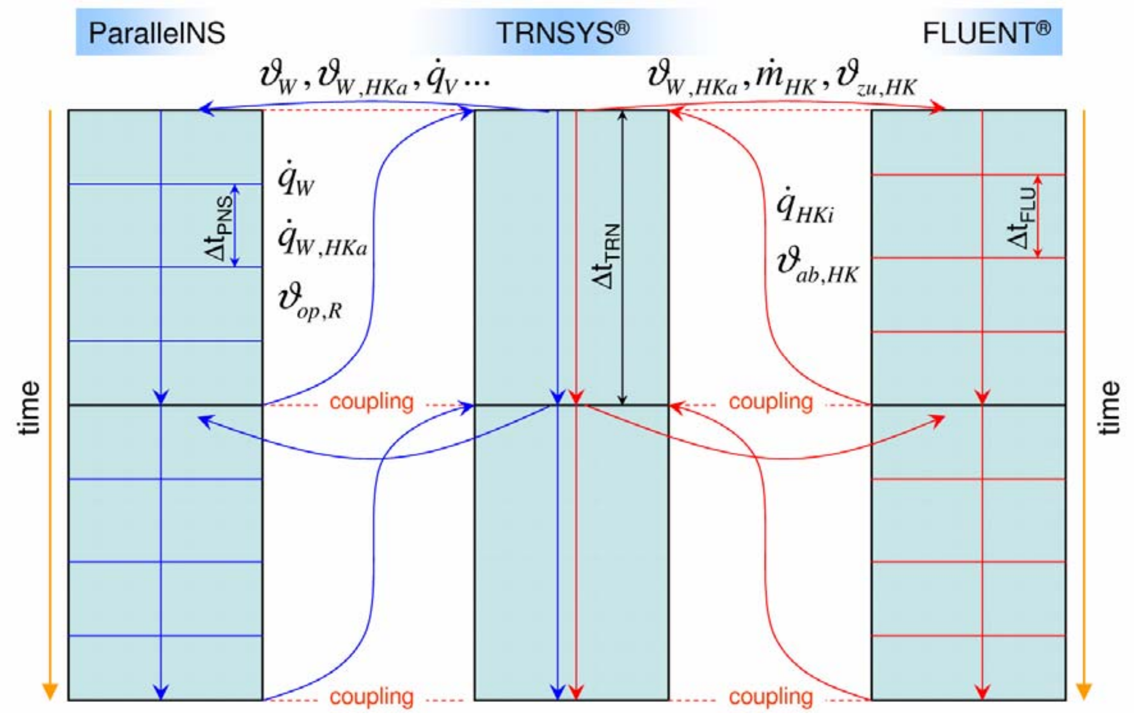

Figure 1. Overview of the coupling algorithm in the fully coupled case.

The chronological operating sequence of the BC-exchange now works as follows, see fig. 1. At first TRNSYS® sends an exchange vector of temperature- and other BC-values to ParallelNS. Using these values ParallelNS performs a transient flow field calculation, until it reaches

the next coupling time. When the coupling time is reached (next TRNSYS® time step), the flow solver calculates the desired heat fluxes at the BC-interfaces and sends them back to TRNSYS®. Now TRNSYS® starts its calculation (energy balances) for the current time step and sends the resulting temperature values for the next time cycle back to ParallelNS. More detailed information about the coupling algorithm can be found in [9].
If the fully coupled radiator model is enabled, ParallelNS simultaneously starts the radiator calculation based on the Fluent® model. Hereby the geometrical resolution of the radiator surfaces is much higher than the resolution of the surrounding walls. This is important in order to ensure the correct temperature distributions in the investigated radiator models. The temporal coupling for this level of approximation can also be seen in fig. 1. A special feature is that the whole radiator model is surrounded by a very thin layer of TRNSYS® BC-faces. The exchange of boundary conditions now takes places between Fluent® and TRNSYS® at these faces. The same TRNSYS® faces also exchange BC-values with the surrounding air flow model, so they get exchange vectors from both sides. The energetic balance for the radiator faces (including radiation) then is calculated in TRNSYS® taking into account both exchange vectors.

RESULTS

Introducing remarks

To illustrate the capabilities and the quality of our simulation tool we modelled two different radiator types, a multi-section radiator and a simple fluted radiator in a test chamber under standard conditions complying to EN 442 [10] in order to investigate their standard heat output. In addition, for the fluted radiator we used three different levels of approximation. The results are finally compared with each other and with measurement data from the radiator specifications, respectively.

Configuration of the simulated test chamber and the radiator models

The modelled test chamber has the overall dimensions of L x W x H = 4 m x 4 m x 3 m, see also fig. 2. The investigated radiator model is, as postulated in [10], located centered in front of the backside of the chamber, 0.1 m above the floor and 0.05 m away from the backside. All walls of the test chamber except the backside are controlled tempered walls. The materials and the wall constructions correspond to the guidelines in [10]. The sensor point for the temperature control (red point) is located in the middle of the test chamber at a height of 0.75 m above the floor.
Our model of the test chamber is equipped with two different control loops. One loop controls the tempered walls in order to keep the temperature at the measure point at a constant value of 20 °C. In the fully coupled case (I and II) the other loop controls the water influx against the radiator outflow temperature. In the simplified cases (IIA, IIB) the second loop controls the outflow temperature and mass flux of an internal radiator model in TRNSYS®. In our investigations we used constant inflow temperatures of 75 °C and required outflow temperatures of 65 °C.
The simulations were performed unsteady on a workstation with two processors until steady state conditions were reached.

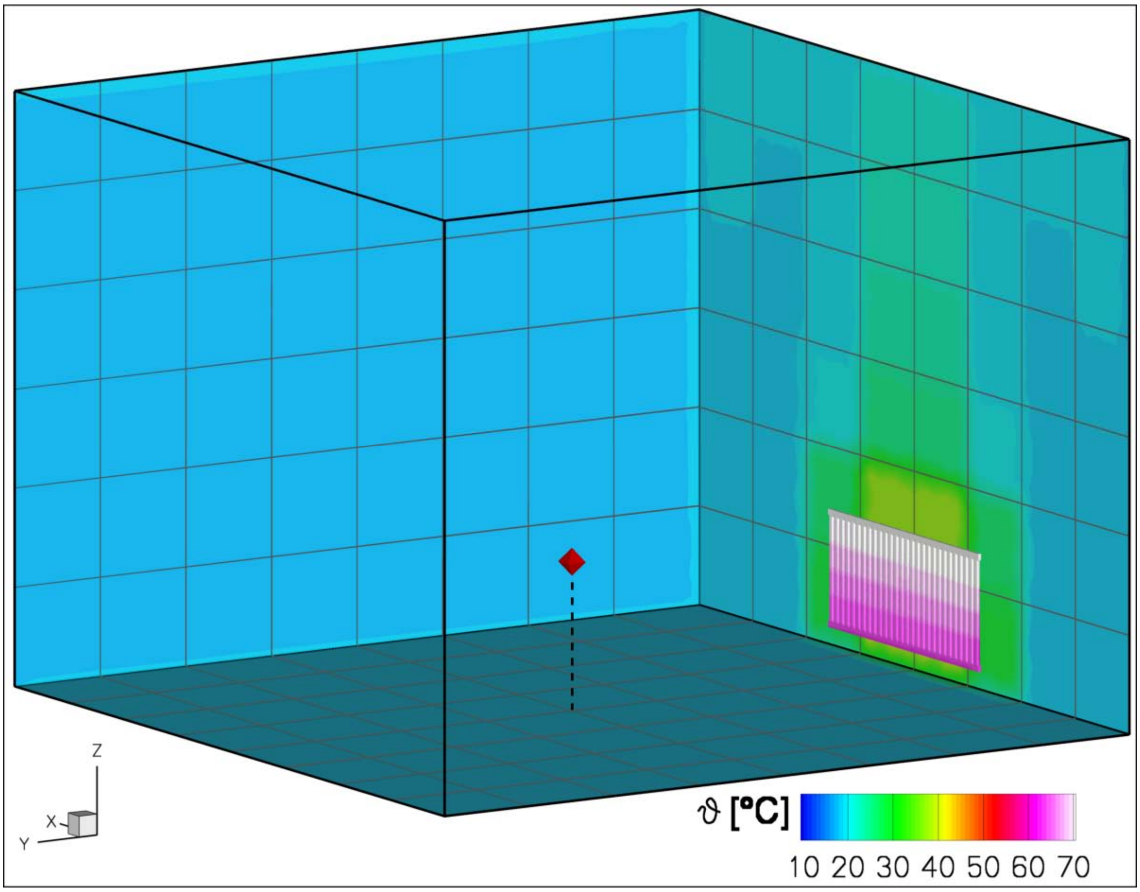

Figure 2. Geometry of the modelled test chamber with radiator and controller location.

For the discretization of the boundary condition exchange and the radiation exchange the test chamber walls are divided into small tiles (0.5 m x 0.5 m), see fig. 2 again.
In case of the fully coupled simulations the radiators are also divided into a large amount of very small tiles (up to 4000 BC faces are possible). This way their real geometry and their real physical behavior, especially in respect to the radiant and the convective heat transfer and the temperature distribution are reflected, see fig. 3.

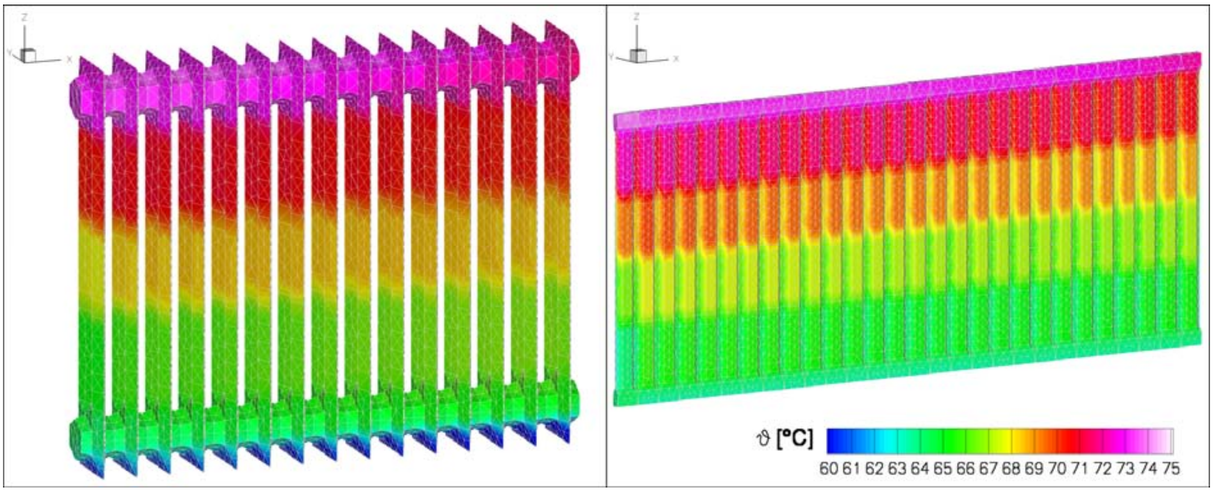

Figure 3. Wall temperatures and surface grids of the fully coupled radiator models.

Figure 3 shows the two investigated fully coupled radiator models (I and II) along with their temperature distributions in the steady state. The water inflow is located at the upper left side while the outflow is located at the same lower side.
In case of the simplified radiator models IIA and IIB, see fig. 4, the whole radiator shell has only two unique surface temperatures (front and back) which are results of the averaged radiant and convective heat transfer between the test chamber and the internal radiator model in TRNSYS®. The only difference between the two cases IIA (fig. 4 left) and IIB (fig. 4 right) is the higher geometrical resolution of IIB which leads to more realistic values of the averaged convective heat transfer. The calculation of the radiation transfer in both cases stays the same.

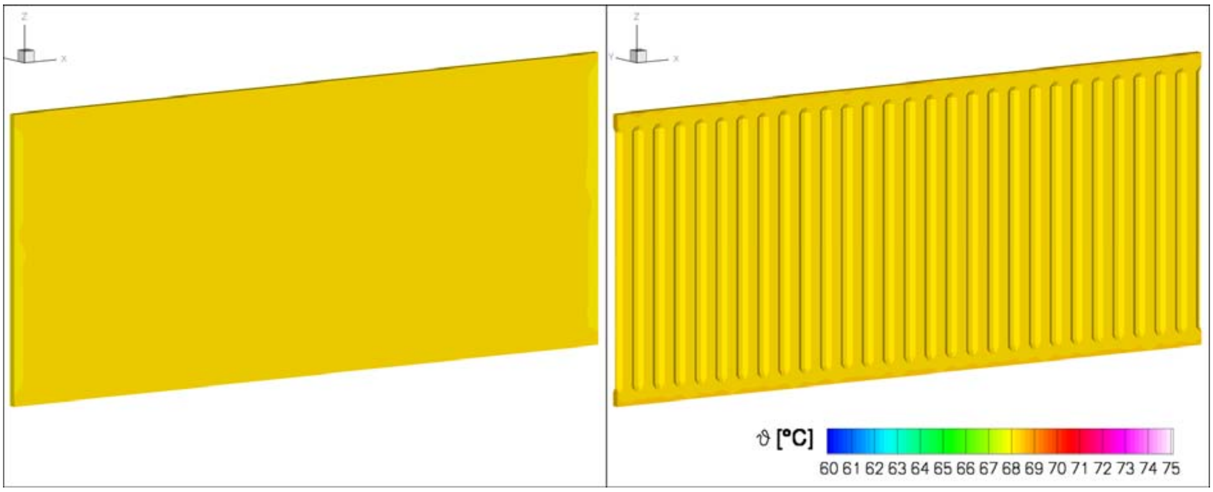

Figure 4. Geometry and wall temperatures and of the simplified radiator models IIA (left) and IIB (right).

Example I – Multi-section radiator model

The main results of the fully coupled simulation of radiator model I are resumed in table 1. Also shown in table 1 are measurement data of the radiator given by the manufacturer.

Table 1. Integral simulation results for radiator model I.

	ϑ_v [°C]	ϑ_r [°C]	Mass flux [kg/h]	Thermal output [W]
Simulation	75.00	65.00	70.02	815
Reference	75.00	65.00	69.68	810

Results regarding the thermal behaviour of the radiator, especially the value of the thermal output are in very good agreement with the standard values given by the manufacturer. The temperature distribution of the radiator, see fig. 3 left, meets the expectations. Due to the low main velocity and the high temperature differences between the fluid and the wall temperatures a nearly uniform vertical temperature stratification arises. Because of the high increase of the flow cross-section after the intake a very smooth distribution of the inflowing water across the radiator sections and hence the visible uniform temperature distribution arises.
The good results confirm the needs of such a complex geometrical model, especially in case of the radiation exchange. Using our simplified radiator model we were not able to predict the correct radiation exchange between the different sections of the radiator.

Example II - Fully coupled fluted radiator model

The simulation results for model II also very well match to the reference values from the manufacturer. The temperature distribution in fig. 3 right again shows the accuracy of the thermal behaviour of our model. In addition, the good results in both cases, I and II, illustrate that for such radiator models with moderate water flow velocities, like I and II, a vertical sectioning of 3 to 5 pieces is sufficient to obtain a very good agreement with the available reference data.

Table 2. Integral simulation results of radiator model II.

	ϑ_v [°C]	ϑ_r [°C]	Mass flux [kg/h]	Thermal output [W]
Simulation	75.00	65.00	81.10	945
Reference	75.00	65.00	80.64	938

Example IIA and IIB - Simplified and geometrical expanded fluted radiator model

Results of simulations of the simplified radiator models in terms of thermal output are resumed in table 3. The data are compared to the reference values from measurements and the values form the fully coupled model. It shows that the results of both detail levels also fit very well to the desired standard values. Here the model IIB with the more complex geometrical resolution even reproduces the standard thermal output as good as the fully coupled model.

Table 3. Comparison of the thermal output for different approximation levels, model II

	Model II	Model IIA	Model IIB	Reference
Thermal output [W]	945	971	932	938
Mass flux [kg/h]	81.10	83.40	80.06	80.64

The results allow the statement that the simplified models and the fully coupled model are on a par if the geometry of the radiator is simple enough, like it is in example II. This result is of great importance for the use of simplified radiator models because most of the investigations like studies of energy consumption of different building types etc. are done based on simplified radiator models.

DISCUSSION

The results of our investigations clearly show that the designed simulation tool is very well suited to reproduce the measured radiator parameters. The differences between measured and simulated thermal output in all investigated cases are less than 5 %. Considering the complexity of the model, these results seem to be very good. In case of standard radiators with simple geometry (model II) the investigations show that our radiator model implemented in TRNSYS® can be used to predict very reliable data of thermal output. Here the higher geometrical resolution (IIB) and the associated more realistic convective heat transfer yields increased quality of the results with comparable calculation effort.

Nevertheless the simplified radiator models are not able to predict the correct behavior of geometrically complex radiators, like the one in case II. Thus the fully coupled model is a very useful simulation tool for the design of complex and new radiator models.

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The Economy of Heat Cost Allocation and Temperature Control in Multiple Unit Dwellings

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SUMMARY

Heat cost allocation is primarily not a method to save energy. It is a way to give the households the temperature they are willing to pay for. The condition that gives the highest reduction of costs from heat cost allocation is at a high marginal cost of temperature. At a high marginal cost there will not be any reduction of the average indoor temperature at the introduction of heat cost allocation. The reduction of costs comes when the households who prefer a high indoor temperature gets the temperature they are willing to pay for. The reduced costs 114 SEK/yr dwelling in buildings built after 1980 and 262 SEK/yr dw in buildings from before 1975 in Sweden should be compared with the cost for meters, reading and administration 334 SEK/yr dw. (8 SEK = 1 USD)

INTRODUCTION

This is a theoretical analyse based on microeconomics. It compares MU-dwellings with heat cost allocation and temperature control that gives a collective indoor temperature, the same temperature in all dwellings, with systems that gives individual temperatures.

Heat cost allocation in homes with individual control and individual heat metering, individual temperature metering and collective allocation was described in [1] and [2]. The use of a fixed part and a variable part for heat cost allocation after heat quantity was not included. This gives a to low indoor temperature. A fixed part (grundkosten) and a variable part (verbrauchskosten) are used in Germany [3]. The fixed part there is between 30 –50 % of the sum of all costs for heating including cleaning and maintenance of the heating system. A paper about the fixed and the variable part is presented at this conference.

METHODS

The systems with individual temperatures are assumed to have the marginal cost of temperature that corresponds to the heat loss to the outdoor air and a cost for indoor temperature that is close to the cost for the heat loss to the outdoor air. This gives the same cost for an increase of the indoor temperature for both owner and household.

In figure 1 the supply water temperature, t_s is controlled after the outdoor temperature, and the heaters are chosen to give the same temperature in all dwellings. The cost of heat is distributed to the households after their share of the buildings dwelling area. Since all households get the same temperature the cost of cold will be high for the households who want a high indoor temperature. This normally gives a higher indoor temperature than if the households could chose individual indoor temperatures.

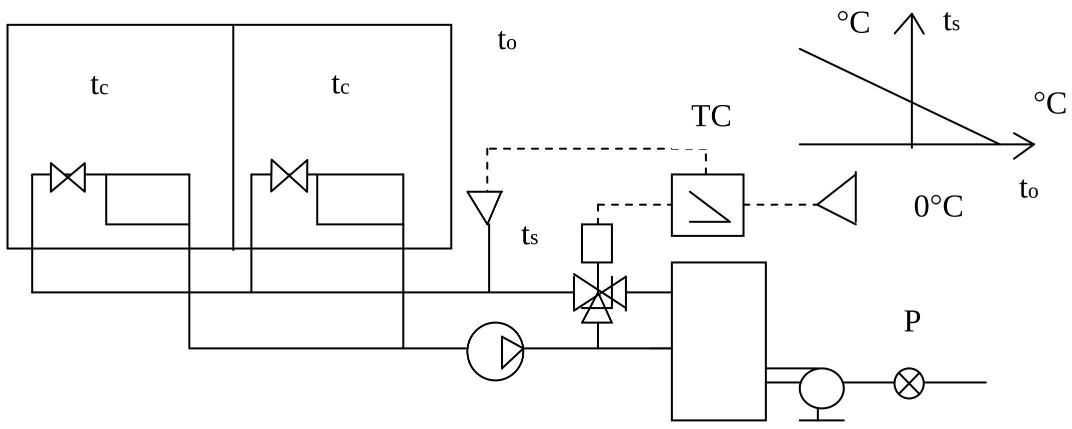

Figure 1. Heating system with a collective temperature, t_c.

The households in a building have different demand curves according to figure 2. The t* is normally distributed with t^*_m = 21.65°C, k = 0.004 (1/°C^2). DI' (SEK/h hh) (8 SEK = 1 USD) is the disposable income per hour and household [4] and [5]. The standard deviation s* = 1.3°C. s* is based on measurements from Swedish SU-dwellings in [6].

To get the demand curve for a collective temperature the individual demand curves in figure 2 are added vertically and divided by the number of households. This gives figure 3.

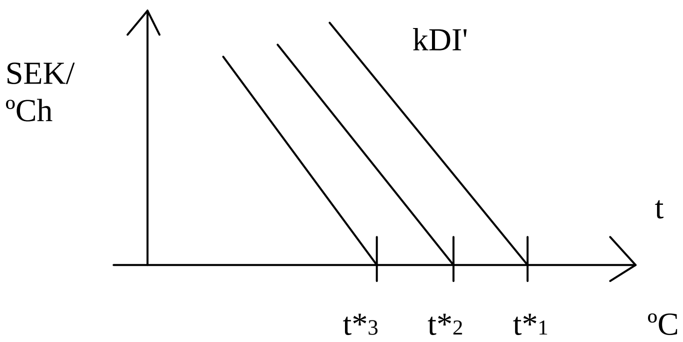

Figure 2. Demand curves for indoor temperature for different households.

The demand curve for a collective temperature in figure 3 goes up to the highest preferred indoor temperature among the households. At a low price of temperature the highest temperature demand will be satisfied. The horizontally marked area is the cost of heat and the diagonally marked area CC(t_c) is the average cost of cold for the households.

If individual heat cost allocation is introduced with the heating system in figure 1 then the indoor temperature is controlled with the valves on the heaters.

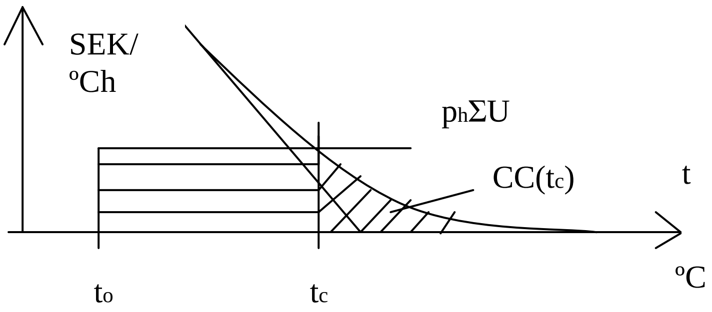

Figure 3. Demand and supply curve for a collective temperature, t_c.

Heat cost allocation systems that gives the marginal cost of temperature that corresponds to the heat loss to the outdoor air is heat metering with a suitable fixed and variable part and indoor temperature metering with the specific heat demand (W/°C) for the heat loss to the outdoor air.

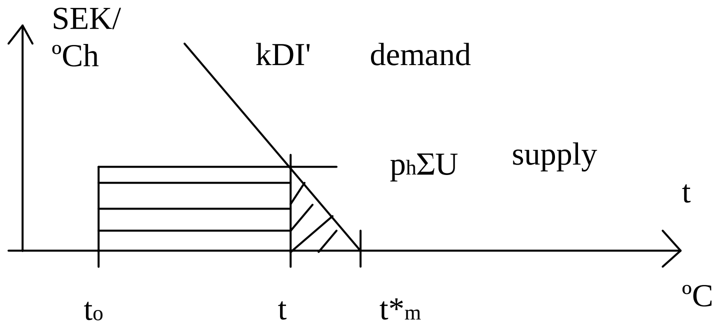

Figure 4. Demand and supply curve for the average of individual temperatures, t.

The supply curve in figure 3 and 4 is $p_h\Sigma U$ where p_h is the price of heat to the dwellings (SEK/kWh) and ΣU is the specific heat loss for a dwelling to the outdoor air (W/°C). t^*_m is the average of the highest demanded temperatures. The horizontally marked area is the average cost of heat and the diagonally marked area is the cost of cold for a household.

Before individual temperatures are introduced a collective temperature in figure 3 is used and after the average indoor temperature in figure 4 is used. The reduced costs RC are figure 4 subtracted from figure 3. The resulting area is shown in figure 5.

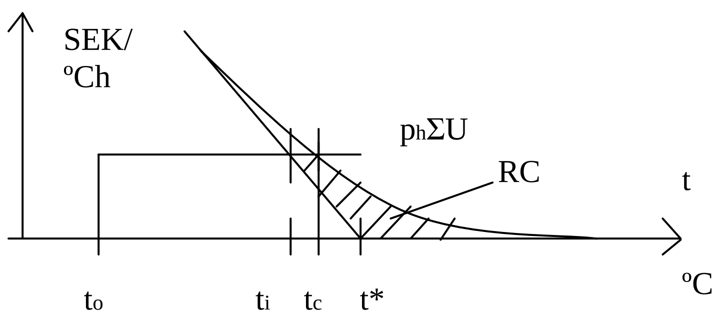

Figure 5. Area, RC representing both the reduced cost of cold and reduced cost of heat when individual temperatures, t_i are introduced. The collective temperature before is, t_c.

$$RC = CC(t_c) + p_h \Sigma U(t_c - t_i) - \frac{k \cdot DI'}{2}(t^* - t_i)^2 \qquad (1)$$

The diagonally marked area in figure 5 or equation 1 represents the reduced cost when the heat cost allocation system is changed from a collective temperature to individual temperatures. The average of the individual temperatures t_i is lower than the collective temperature t_c. The cost reduction is both reduced heat cost and reduced cost of cold. The area RC represents a change from a perfect collective indoor temperature to perfect individual temperatures meaning that they follow the theory. In reality the temperatures are slightly higher than both t_i and t_c. In order to find the perfect collective temperature the owner must know the demand curves of the households. If individual temperature control is used then some households will not control their indoor temperature witch will increase the average.

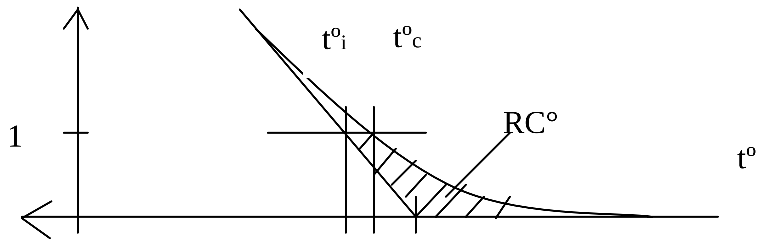

Figure 6. Dimensionless reduced costs, RC° when individual temperatures are introduced in a building with a collective temperature.

Dimensionless area RC° representing reduced costs when heat metering is introduced equation 2. The area CC in figure 3 must be calculated numerically so in order to make the calculations universally valid the axis are normalised. The dimensionless cost of cold for a collective temperature $CC(t_c)$ was shown in [7]. Dimensionless indoor temperature equation 3, individual indoor temperature $t°_i$ equation 4 and standard deviation s equation 5. The dimensionless indoor temperature and corresponding collective temperature, marginal cost, marginal cost of cold and the area that gives the cost reduction are given in table 1.

$$RC° = CC°(t°_c) + t°_i \cdot (t°_c - t°_i) - \frac{{t°_i}^2}{2} \qquad (2)$$

$$t° = \frac{t^* - t}{s} \qquad (3)$$

$$t°_i = \frac{p_h \cdot \Sigma U}{k \cdot DI' s} \qquad (4)$$

$$s = \sqrt{s^{*2} + {s_b}^2} \qquad (5)$$

The temperature in the dwellings in a building with the heating system in figure 1 follows a normal distribution with s_b = 0.8°C according to an estimate from measurements in [8]. s_b depends on the accuracy of the design and balancing of the heating system.

Reduced costs with heat metering equation 6 (SEK/h). To get the cost per year multiplie with the length of the heating season τ (h/year).

$$RC = RC^{\circ}\left(t^{\circ}_{i}\right) \cdot p_{h}\Sigma U \cdot s \qquad (6)$$

Table 1 Dimensionless average of individual, t^{o}_{i} and collective temperature, t^{o}_{c} marginal cost of cold, MC^{o} cost of cold, CC^{o} and cost reduction, RC^{o} at the introduction if individual temperatures.

t^{o}_{i}	t^{o}_{c}	$MC^{o}(t^{o}_{i})$	$CC^{o}(t^{o}_{i})$	$RC^{o}(t^{o}_{i})$
2.0	1.99	2.01	2.50	0.50
1.8	1.79	1.82	2.12	0.50
1.6	1.58	1.62	1.78	0.50
1.4	1.36	1.44	1.47	0.49
1.2	1.15	1.26	1.20	0.48
1.0	0.90	1.08	0.97	0.47
0.8	0.65	0.92	0.77	0.43
0.6	0.35	0.77	0.60	0.39
0.4	0.00	0.63	0.46	0.33
0.2	-0.50	0.51	0.35	0.23
0.0	-2.00	0.40	0.25	0

The highest cost reduction from individual heat cost allocation is at high t°_{i} or when the marginal cost of temperature is high in relation to the disposable income. These economic conditions means that the indoor temperature with collective temperature is low and that the indoor temperature will not be reduced at the introduction of individual heat cost allocation. The reduced costs is from the reduced cost of cold for those who want a high indoor temperature.

At low t°_{i} or at a low price of energy and a high disposable income the indoor temperature will be high with collective heat cost allocation and the temperature reduction at the introduction of individual heat cost allocation will be high. The value of the temperature reduction will be low since the heat has a low price.

RESULTS

The economic evaluation is for one building with MU-dwellings built before 1975 and for one built after 1980 in Sweden. The U-values and the ventilation were reduced after the energy crisis. The dwelling area is 70 m^2/dw.

The economic data for the household refers to 1992. The average disposable income for a house-hold in a multiple unit dwelling were 135.000 SEK/yr hh. The disposable income per hour is DI' = DI/8760 = 15 SEK/h, hh. The length of the heating season 6000 h/yr. The price of heat p_h = 0.5 SEK/kWh (including tax).

The standard deviation $s = (1.3^2 + 0.8^2)^{0.5} = 1.6$°C

The investment in equipment for temperature registration and heat metering is 1500 SEK/dw. It can be used for 10 years and the interest rate is 4 %. The factor of annuity is 0.123 /yr and the cost 184 SEK/yr. To read the heat meters costs 75 SEK/yr dw. The heat cost should be calculated and added to the rent. This is an additional 75 SEK/yr dw. The total cost for heat metering and cost allocation is 334 SEK/yr dw.

Table 2. Heat loss from MU-dwelling to outdoor air.

		Built after 1980		Built bef 1975	
	Area m^2	U $W/m^{2o}C$	ΣUA W/ºC	U $W/m^{2o}C$	ΣUA W/ºC
Wall	31	0.3	10	0.8	25
Roof	70	0.2	14	0.6	42
Window	10	2.0	20	3.0	30
Ventilation		25 l/s	30	30 l/s	36
Sum		ΣU W/ºC	74	ΣU W/ºC	133

Table 3. The average temperature with individual heat cost allocation, t_i and the collective temperature t_c with collective allocation in a MU-dwelling.

Built	$t^o{}_i$	t_i ºC	t_c ºC	RCº	RC SEK/yr
Before 1975	0.69	20.54	20.85	0.41	262
After 1980	0.38	21.03	21.73	0.32	114

The reduced costs 262 SEK/yr dw in buildings before 1975 and 114 SEK/yr dw in buildings after 1980 should be compared with the cost for meters, reading and administration 334 SEK/yr dw. Metering cost more than the cost reduction from metering. The cost of the control work from the households is not included.

DISCUSSION

A high marginal cost of temperature makes the households reduce their indoor temperature due to the cost. The reduction will be high at low disposable incomes. If a collective indoor temperature is used, the same temperature for all households, then the indoor temperature is highly reduced for those who preferes a high indoor temperature. Then the value of individual heat cost allocation will be high since everybody can have the temperature they are willing to pay for. The marginal cost of temperature is high if the price of heat is high and if the buildings are poorly insulated. Poorly insulated buildings are used in warm and mild climates. Individual temperatures can be expected were there are long mild winters.

Individual heat cost allocation benefits from badly designed and badly balanced heating systems that don't give the same temperature in all dwellings and from different indoor temperature preferences among the households.

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The Use of a Fixed Part and a Variable Part in Heat Cost Allocation after Heat Quantity in Swedish Multiple Unit Dwellings

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SUMMARY

This is a theoretical analyse based on microeconomics. It analyses heat cost allocation methods after measured heat quantity for MU-dwellings in a building with an owner who byes or produces the heat that is sold to the households.

A MU-dwelling loses heat to the outdoor air and exchanges heat with surrounding dwellings. The owner can use less heat to increace the indoor temperature than the household it self. The household must pay for heat transfer to other dwellings. This makes an increase in the indoor temperature more expensive to the household than to the owner. The households will have a to low indoor temperature. To compensate this only a part of the measured heat quantity is sold. The rest of the heat cost for the building is distributed to the households after dwelling area. In Swedish MU-dwellings 60-80 % the fixed part should be distributed after area and 20-40 % the variable part sold after measured quantiy.

INTRODUCTION

The influence of the marginal cost on the consumed quantity of a goods has been known from economics for hundreds of years, but it has seldom been used to analyse heating of buildings. The use of a fixed part is to reduce the marginal cost of indoor temperature or to make a temperature increase cheaper. This paper suggests how the fixed part should be calculated to give indoor temperature the marginal cost that corresponds to the heat loss to the outdoor air.

[1] called heating in MU-dwellings a semi-collective goods. Semi means that the acces to the indoor temperature is limited to the membes of the households. [2] used economics to analyse individual metering of hot water in MU-dwellings. [3] presented the economics of heating. The occupants sets their thermostats to the temperature where marginal cost equals marginal value of indoor temperature. He investigated the economic differences between a common temperature and individually chosen temperatures in two apartments and used the theory for public goods to find the best common indoor temperature in a building with two dwellings. [4] used economics to investigate low indoor temperatures in low income households.

The demand curve for indoor temperature in Swedish MU-dwellings was determined by [5] as a straight line and in SU-dwellings by [6] and [7]. Heat cost allocation in homes with individual control and individual heat metering, individual temperature metering and a collective temperature was described in [8]. The use of a fixed part and a variable part for heat cost allocation after heat quantity was not included in [8]. A fixed part (grundkosten) and a variable part (verbrauchskosten) is used in Germany [9]. The fixed part there is between

30-50 % of the sum of all costs for heating including cleaning and maintenance of the heating system.

METHODS

A household in a MU-dwelling produces indoor temperature whith heat and insulation. The household balances the cost of heat or cost of temperarure horizontally marked in figure 1 and the cost of cold diagonally marked against each other. The household choses the indoor temperature where the marginal cost of heat is equal to the marginal cost of cold. The marginal cost of heat is the cost of a temperature increase.

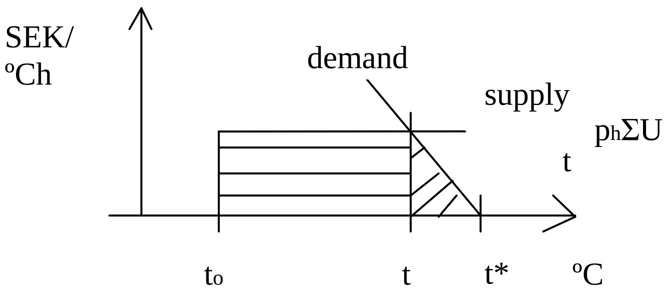

Figure 1. Demand and supply curves for indoor temperature. Indoor temperature, t. Outdoor temperature, t_o. Highest indoor temperature the household is willing to pay for, t*.

The MU-dwellings are in a building with an owner who byes or produces the heat that is sold to the households after meassured heat quantity. A MU-household in a building loses heat to the outdoor air and exchanges heat with surrounding dwellings. The owner can use less heat to increate the indoor temperaturre in a dwelling than the household it self. This makes an increase in the indoor temperature more expensive to the household than to the owner. The household must pay for heat transfer to other dwellings, but the owner only pays for the heat that goes out to the outdoor air. Heat to other dwellings repleaces heat from the heating system. The households will have a to low indoor temperature if the household pays for the heat after measured quantity.

In this case it is possible for the owner to reduce the price of heat to the household or to sell only a part of the measured heat quantity. The ideal is to make the cost of a temperature change the same as for the owner. This will increase the indoor temperature and well beeing for the households. To cover the cost of heat to the building the owner must increase the rent so his economy isn't affected. It is important that the cost of a goods or serivce is the same to all users in the society so the alternatives can be optimised against the same price. This is for the period during the year when heating is neccesary in all dwellings in the building. Three heat cost allocation methods are analysed.

The same cost for a temperature increase for both household and owner

Q is heat quantity to the building in figure 2 (W). Q_1 is the measured heat quantity (W) to dwelling 1. ΣQ_i is the sum of measured heat quantities to the n dwellings. ΣU_1 equation 1 is the specific heat demand of dwelling 1 (W/°C) for heat to outdoor air. q outdoor air rate for ventilation (m^3/s). ΣU_{12} is the specific heat exchange between dwelling 1 and dwelling 2 trough internal walls (W/°C) equation 2.

$$\Sigma U_1 = \Sigma UA + q \cdot \rho \cdot c_p \qquad (1)$$

$$\Sigma U_{12} = \Sigma UA_{int} \qquad (2)$$

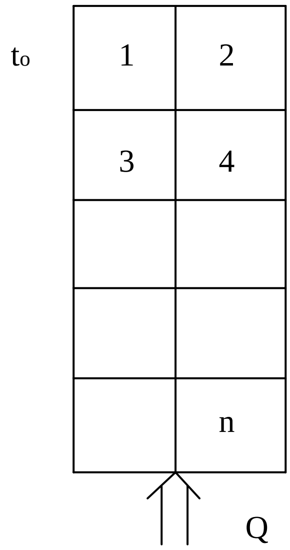

Figure 2. Building with n MU-dwellings. t_o outdoor temperature ºC.

The heat loss to outdoor air (W) from dwelling 1, equation 3. All rooms in a dwelling have the same temperature.

$$\Sigma U_1 \cdot (t_1 - t_o) \qquad (3)$$

The heat loss from dwelling 1 to outdoor air and to other dwellings Q_1, equation 4.

$$Q_1 = \Sigma U_1 \cdot (t_1 - t_o) + \Sigma\Sigma U_{1i} \cdot (t_1 - t_m) \qquad (4)$$

t_m average (weigthed) temperature in dwellings surrounding dwelling 1 (ºC). $\Sigma\Sigma U_{1i}$ is the specific heat exchange between dwelling 1 and surrounding dwellings trough internal walls (W/°C) equation 5.

$$\Sigma\Sigma U_{1i} = \Sigma U_{12} + \Sigma U_{13} + \Sigma U_{14} \qquad (5)$$

If equation 4 is differentiated with respect to t_1 and the variable share (1-Φ) times the heat demand for the household to increase the indoor temperatue is put equal to the differentiated heat demand for the owner equation 3. Then $(1-\Phi_1)$ can be solved equation 6. The fixed share Φ_1.

$$(1 - \Phi_1) = \frac{\Sigma U_1}{\Sigma U_1 + \Sigma\Sigma U_{1i}} \qquad (6)$$

If Q_1 $(1-\Phi_1)$ is sold to the houshold at the market price of heat p_h (SEK/kWh), (1USD = 8 SEK) or the price of heat is reduced to $(1-\Phi_1)$ p_h then it will be equally expensive for the household and the owner to increase the indoor temperatue in dwelling 1 but the owner will not cover his cost for heat. The measured heat quantity to dwelling 1 will still depend on the temperature in surrounding dwellings, t_m but the fixed part will reduce the influence.

Since this is a reduction in marginal cost for the household a higher indoor temperature will be used by the household. There is another variable share for dwelling 2.

The same cost for a temperature increase for both household and owner that covers the cost for the owner

If the price of heat or the sold quantity of heat is reduced then the owner must increase the rent to cover his costs.

The heat to dwelling i, Q_i is divided in two parts with the fixed share Φ_i for dwelling i in equation 7.

$$Q_i = \Phi_i \cdot Q_i + (1 - \Phi_i) \cdot Q_i \qquad (7)$$

All heat that the owner byes or produces Q will be sold to the households according to equation 7 and 8.

$$Q = \Sigma Q_i = \Sigma \Phi_i \cdot Q_i + \Sigma (1 - \Phi_i) \cdot Q_i \qquad (8)$$

Each household pays p_h (1 - Φ_i) Q_i as the variable part. The rest of the heat $\Sigma\Phi_i Q_i$ is equally divided between the n dwellings since they have the same area. This is called the fixed part and it is included in the rent.

The marginal cost of indoor temperature now depends on the number of dwellings n in the building so a new Φ_1 analog to equation 6 have to be derived. .

Household 1 pays for the heat quantity Q_{p1} according to equation 9.

$$Q_{p1} = \frac{\Sigma \Phi_i \cdot Q_i}{n} + (1 - \Phi_1) Q_1 \qquad (9)$$

The new Φ_1 should make the differential of Q_{p1} with regard to t_1 equal to ΣU_1 according to equation 3. This makes the marginal cost for the household in dwelling 1 equal to the marginal cost for the heat loss to the outdoor air. Q_1 from equation 4, Q_2 - Q_4 are analog to equation 4 and inserted in equation 9. Dwelling 2, 3 and 4 have heat exchange with dwelling 1.

If Φ_1 to Φ_4 are approximately equal then Φ_1 can be calculated from equation 10. This approximation is probably a better method than to use the same average Φ for all dwellings. Different Φ makes the marginal cost equal to the marginal cost for the heat loss to the outdoor air for every dwelling.

$$\Phi_1 = \frac{\Sigma\Sigma U_{1i}}{\Sigma U_1 \cdot \left(1 - \frac{1}{n}\right) + \Sigma\Sigma U_{1i}} \qquad (10)$$

Compared to the fixed share from equation 6, equation 10 gives a higher fixed share. It will be higher in buildings with a few dwellings.

The same cost for a temperature increase, the same cost for heating for both household and owner and that covers the cost for the owner

This is not possible with heat quantity measurement since the heat to other dwellings depends on the temperature in the surrounding dwellings and on the heat transfer properties to the other dwellings equation 4.

Heat quantity measurement will give a reduced cost for the household if heat from internal heating systems or free heat are used in the dwelling. But the variable part will reduce the part of the free heat that reduces the cost.

A cost allocation system that gives the same cost for a temperature increase, the same cost for heating for both household and owner that covers the cost for the owner is indoor temperature measurement and calculation of the heat to the outdoor air from the dwelling with equation 3. This system can not register the free heat.

Example

The heat to and from a MU-dwelling can be shown as function of the indoor temperature in a diagram figure 3. The dwelling get heat from heaters and depending on the indoor temperature exchanges heat with other dwellings. The dwelling also get heat from its occupants, from lighting, refrigerators etc. This is called free heat. If the heaters in the dwelling is shut of it will get the indoor temperature t_b. At higher temperatures the heaters are used. At the indoor temperature t_m there is no net heat exchange with other dwellings and at higher indoor temperatures heat is lost to other dwellings.
The line representing heat loss to the outdoor air starts in t_o. At t_b the line is thick. This part shows the heat power that a dwelling pays for if the marginal cost of temperature is for the heat loss to the outdoor air. The horizontal thick line shows the fixed part.

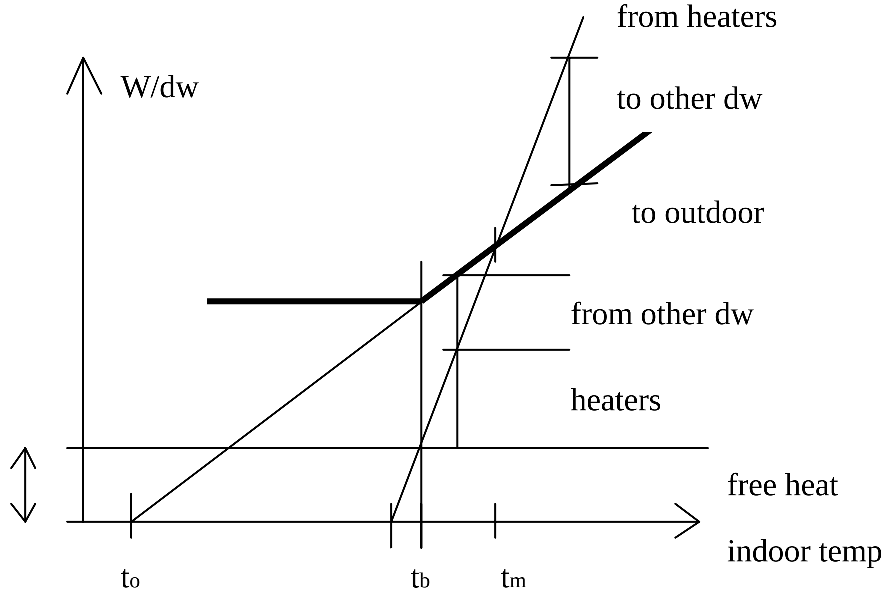

Figure 3. Heat to and from a MU-dwelling with free heat.

The heat from the heaters is more sensible to the indoor temperature than the heat loss to the outdoor air. If the heat from the heaters should be used for cost allocation then it must be corrected with Φ so the heating cost follows the cost for heat loss to the outdoor air, from t_b and up to higher temperatures. At the lowest possible temperature t_b the household pays the fixed part of the buildings heating cost.

RESULTS

Calcualtion of the fixed share Φ that gives the same cost for a temperature increase for both household and owner in Swedish MU-dwellings built before and after the energy crisis.

Table 1 Heat loss from a MU-dwelling to outdoor air trough different building elements.

Element		Built after 1980		Built bef 1975	
	Area m^2	U $W/m^2 ºC$	ΣUA W/ºC	U $W/m^2 ºC$	ΣUA W/ºC
Wall	31	0.3	10	0.8	25
Roof	70	0.2	14	0.6	42
Window	10	2.0	20	3.0	30
Ventilation		25 l/s	30	30 l/s	36
Sum		ΣU W/ºC	74	ΣU W/ºC	133

Table 2 Heat exchange between MU-dwellings $\Sigma\Sigma U_{1i}$

	Area m^2	U $W/m^2 ºC$	ΣUA W/ºC
Wall	41	2.5	102
Floor	70	1.5	105
Sum		$\Sigma\Sigma U_{1i}$ W/ºC	207

The ΣU_1 from table 1 and $\Sigma\Sigma U_{1i}$ from table 2 in equation 6 gives the Φ in table 3. If the building have preheated supply air then the heat demand for ventilation in table 1 is excluced.

Table 3 Fixed share Φ that fullfills the same marginal cost for both owner and household. In a MU-dwelling according to table 1 and 2. Variable share (1- Φ).

	Built after 1980	Built before 1975
Exhaust ventilation	74 %	61 %
Supply and exhaust vent	82 %	68 %

The Φ that gives the same cost for a temperature increase for both household and owner and that covers the cost for the owner to be used with equation 9 will be higher than in table 3 depending on how many dwellings there are in the building.

To calculate a Φ for all dwellings in the building that gives the same cost for a temperature increase for both household and owner and that covers the cost for the owner more data for the building must be known, but the fixed share will be of the same magnitude as in table 3. The fixed share used in Germany is between 30-50 %.

DISCUSSION

Well insulated buildings where the dwellings have a small heat loss to the outdoor air needs a high fixed part to reduce the marginal cost of temperature. The heat exchage with the outdoor air is a small part of the total heat exchange for a MU-dwelling. Small not so well insulated buildings needs a low fixed part since the heat exchage with the outdoor air is a big part of the total heat exchange for the dwelling.

More numeric calculations and data are necessary to find the fixed part for all equaly sized dwellings that cowers the cost for the owner. More research is necessary to find the best fixed part in a building with dwellings that differs in size.

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An innovative thermally activated light-weight steel deck system – numerical investigations and practical tests

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SUMMARY

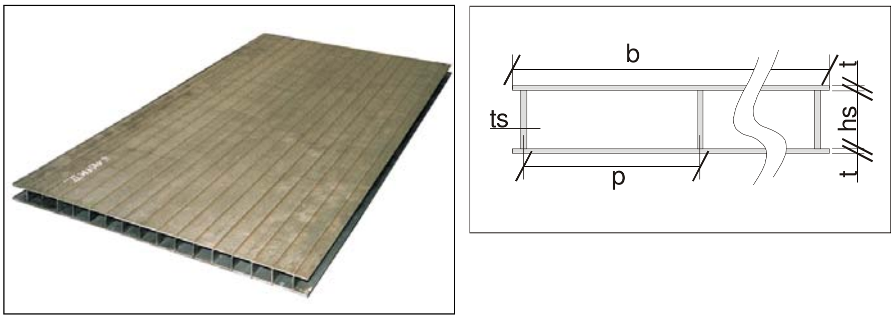
A prefabricated deck system made of laser-welded steel sandwich panels (LSP), up to now only used for decks in shipbuilding, shall be improved for the use in buildings. Beneath other items like fire safety, acoustics and vibrations here the thermal performance is of particular interest. Due to the very low weight the thermal inertia is very low. For compensation of this property this deck element was designed with an integrated piping system for simulating the thermal behaviour of a heavy construction, furthermore the characteristic of a heating and cooling ceiling could be reached, depending on the supply temperature.

Prototypes of these new deck panels were produced and implemented into the research building of the Institute of Steel and Light-weight Constructions, RWTH in Aachen. A detailed scientific programme was performed to work out the thermal capabilities of these elements using a detailed monitoring system, heat meters and infrared surveys. In parallel numerical investigations were made (FEM-calculations and TRNSYS-simulations) to compare the results and to show further steps for improvements.

The investigations show interesting results: The manufacturing must be properly performed to get sufficient thermal contact from pipes to the ceiling, the response time is low due to the low mass, and the benefit for the indoor climate of light-weight constructions is high with acceptable additional costs.

INTRODUCTION

The deck element in the scope of this investigation is a laser-welded steel sandwich panel (LSP) with vertical webs in a Vierendeel system. Figure 1 shows this element, which is originally used for decks in shipbuilding, and gives the range of dimensions, in which this panel is available, the maximum span amounts 10 m. The column indicated with “RWTH” shows the dimensions of the elements used for testing in the RWTH Modular Research Building (see below). The very low depth could only achieved by penetrating the laser-weld beam from the outside. These deck elements have a high stiffness at a very low mass. These properties make the panels interesting for highly prefabricated construction systems.



	from [mm]	to [mm]	RWTH [mm]
Width	500	3000	1800
Length	1000	10000	5400
t	2	10	3
p	120	120	120
hs	40	100	80
ts	3	4	4

Figure 1: Laser welded steel sandwich panel with upper deck plate, vertical webs and lower deck plate (left), dimensions of laser-welded panels (right)

One problem of having this very low mass is the missing thermal inertia, therefore a solution to compensate this disadvantage was developed and tested. A steel pipe meander was integrated in the panel to create a heating / cooling ceiling resp. a thermally activated building component (Figure 2). The denomination is not clear-cut, on one hand, the hydronic system is integrated in a structural component similar to a concrete core cooling, on the other hand it behaves like a cooling ceiling

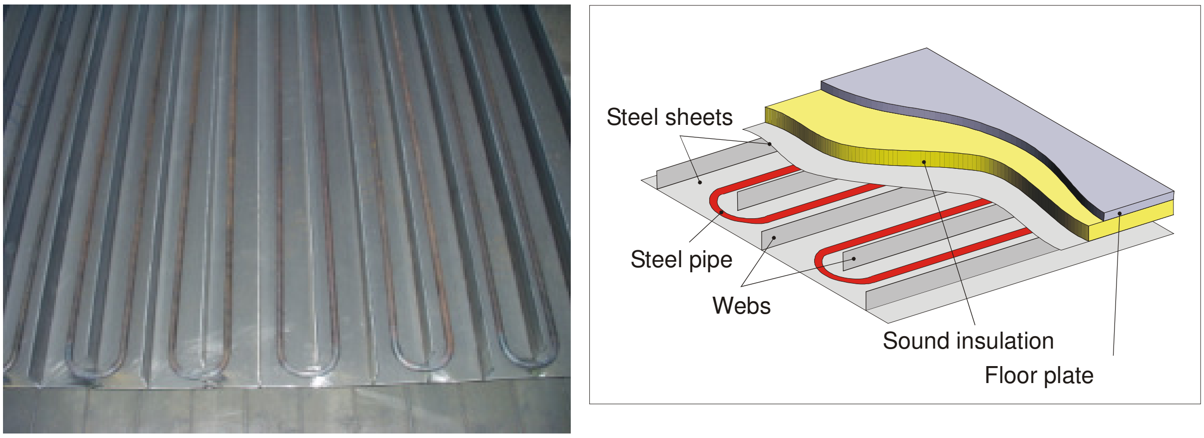


Figure 2: Integration of steel pipes in the steel deck system (left: steel pipes in the module before welding the upper steel sheet, right: principle of the completed deck system)

IMPLEMENTATION AND TESTING OF THE DECK SYSTEM IN A RESEARCH BUILDING

In 2005 the Institute of steel and light-weight construction, RWTH Aachen University, in collaboration with the german association IFBS (Industrieverband für Bausysteme im Metallleichtbau, Düsseldorf), erected a research building using light weight, prefabricated components made of steel. Drawings of the building are given in figure 3, figure 4 shows the erection and figure 5 the completed building. One objective of this research building is the measurement of the thermally activated LSP. The test room in the center of the ground floor is equipped with these elements.

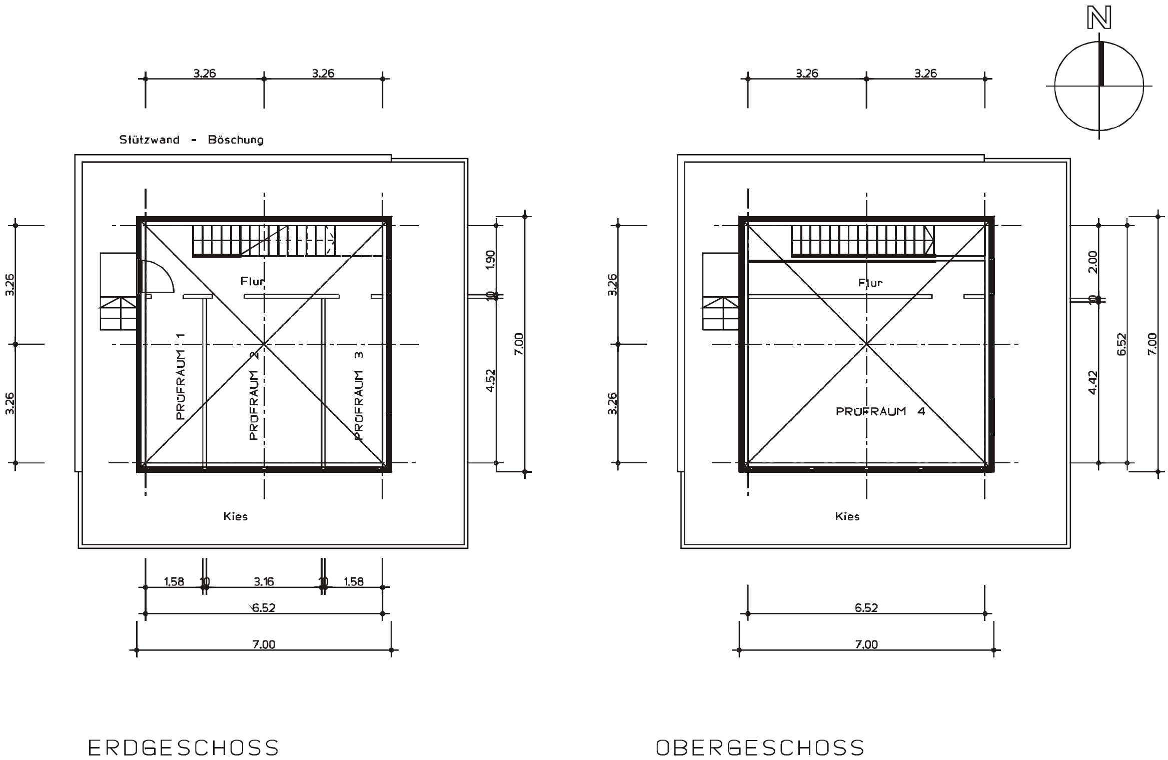


Figure 3: Ground plan of modular research building (left: ground floor, right: upper floor)

Figure 4: Erection of Modular Research Building

Figure 5: Modular Research Building, elevation North-West (left), South-East (right)

TESTING EQUIPMENT FOR DECK SYSTEM

The two active deck elements are feeded by a refrigerated circulator, providing pre-heated or pre-cooled water. Various valves are integrated in the circuit to allow different running modes (only one panel active or both in parallel or serial connection). Each panel has a heat meter for measuring volume flow and the consumption of heat resp. coolth (figure 6). Additionally heat flux plates and an infrared camera were used to get more reliable data by using different measurements at once.

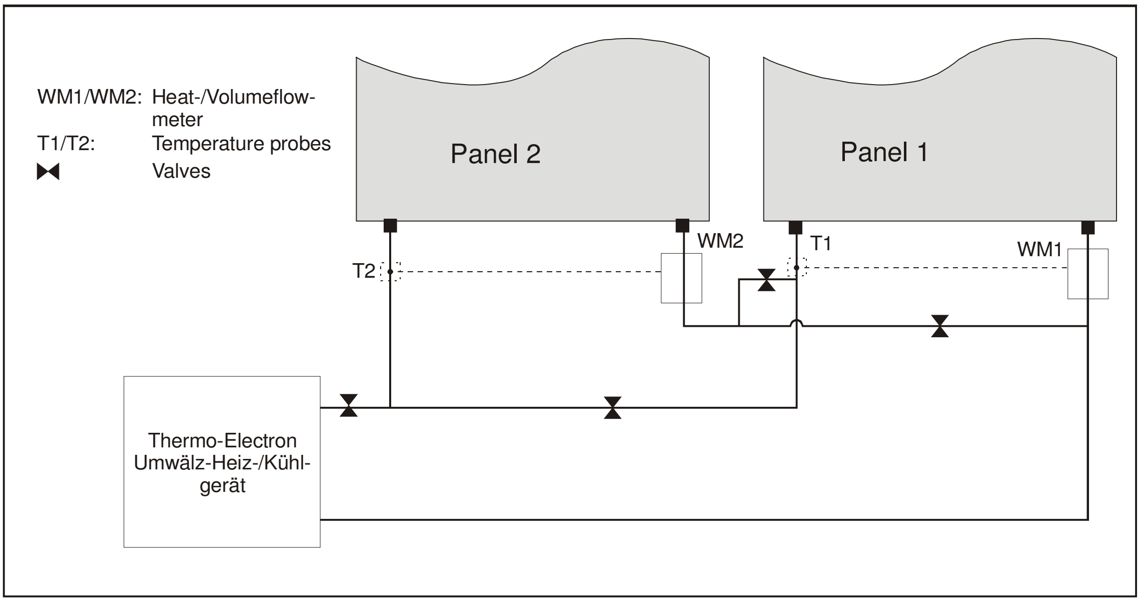


Figure 6: Sketch of testing set-up

Measurements were made for heating and cooling, due to the lack of space only the cooling results are presented (further results: see [4]). Figure 7 shows the specific cooling power depending on the mean temperature difference water – room. In the diagramm results taken from the heat meters and by heat flux meters in combination with infrared surveys are shown. Results between ca. 30 and more than 45 W/m² were measured at temperature differences between 6 and 8 K (room – mean water temperature).

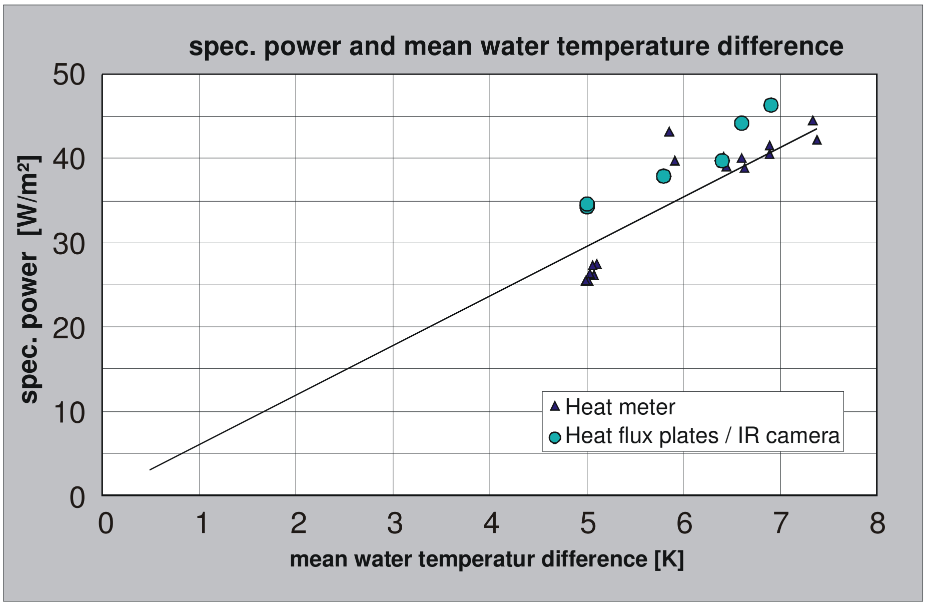


Figure 7: Measured cooling power of active deck element, cooling mode

An infrared survey from the ceiling were taken to get detailed information about the temperature distribution (figure 8). The view on the lower side of the panels shows some remarkable aspects: The steel pipes are sticked-welded to the lower deck plate of the section every 100 cm, these positions can be identified as “cold spots”, here is a good thermal contact guaranteed. In between the contact is more or less random – in some parts there is good contact also between the welded points, but in the majority part the temperature is higher – the thermal contact is not sufficient.

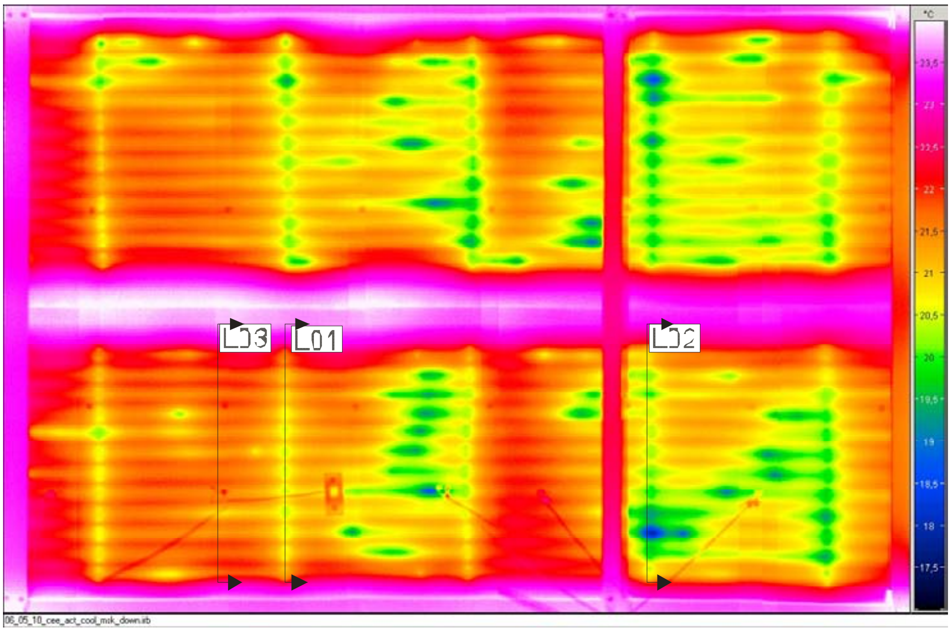


Figure 8: Infrared survey of active deck element, cooling mode (view underneath the steel slab)

FEM-CALCULATIONS

The tested deck element was also numerically investigated. A representative sectional model was used for a FEM-study. Two different cases were considered: The pure LSP with the pipe and a complete deck system including insulation and floor plate (figure 9). These two variations were tested in the research building, too.

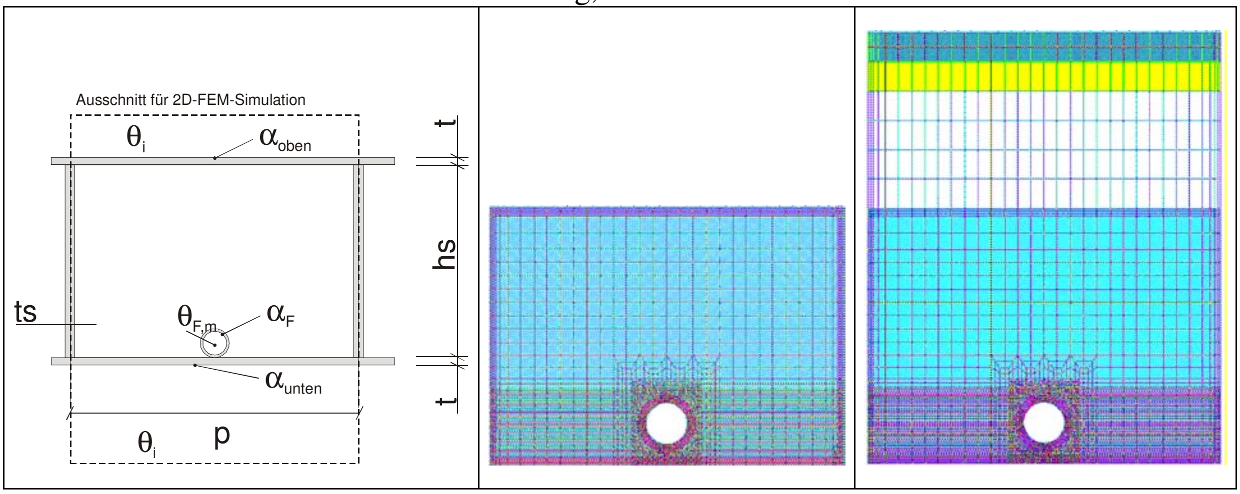


Figure 9: FEM-model of thermally activated LSP (left: sketch with boundary conditions, mid: FEM-model without flooring, right, FEM-model with insulation an floor plate)

The following boundary conditions were taken into account [2],[3]:

Mean water temperature $\theta_{F,m}$:	18 °C
Room temperature θ_i:	26 °C
Heat transfer coefficient α_{top}:	6.7 W/m²K
Heat transfer coefficient α_{unten}:	$8.92\ (\theta_i - \theta_{F,m})^{0.1}$ W/m²K
Heat transfer coefficient water/pipe α_F:	200 W/m²K

Figure 10 shows the results of the FEM-calculations. Without insulation a part of the coolth reaches the floor, the ceiling temperature is about 20 °C, the floor approx. 22 °C. If the insulation and floor plate is added, nearly all coolth is transferred by the ceiling (temperature 19 – 20 °C), the floor temperature is close to the room temperature.

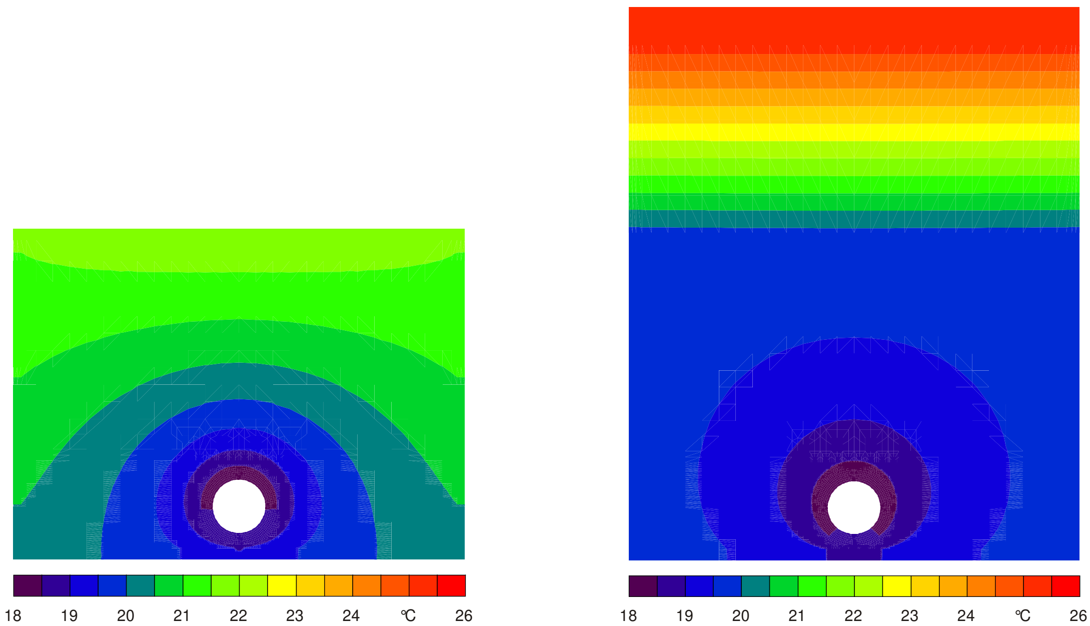


Figure 10: Results FEM-calculation (left: without flooring, right: with insulation and floor plate)

Based on these results the cooling power for various temperature differences was calculated (Figure 11); the diagram shows additionally curves for existing cooling ceilings, furthermore the measured results as shown in Figure 7 are marked. The FEM-results show a high performance of the thermally activated LSP, on the other hand, the measured results are significantly lower. This difference was studied in detail by an analysis of the IR-survey and a variation of the FEM-calculations (Figure 12).

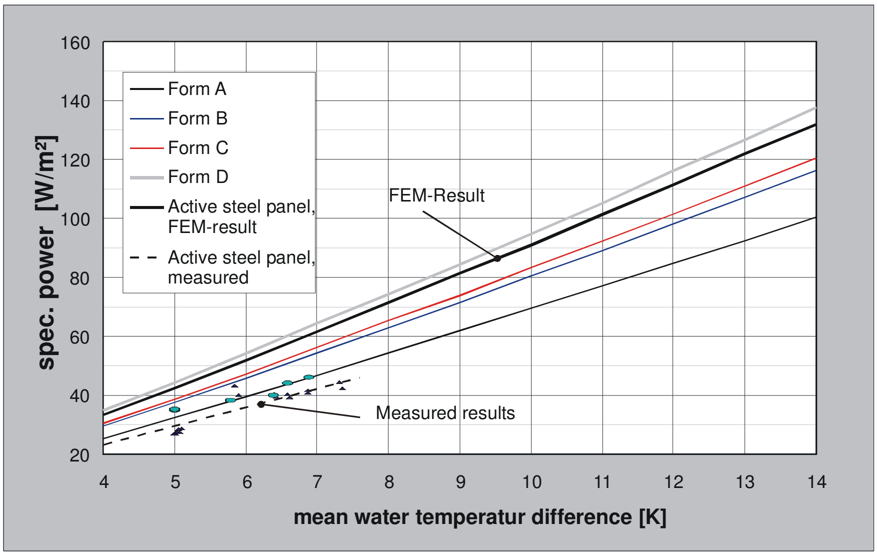


Figure 11: FEM-result (thermally activated LSP with insulation and floor plate), in comparison with conventional existing cooling ceilings (Form A to D, [1]) and the measured results

The comparison of Figure 12 shows, that very small air gaps between the pipe and the sheet lead to a significant loose of performance. The measured section L3 is close to the calculated result for a 1 mm air gap, on the other hand the measured cold spots (see L02) reach the temperature

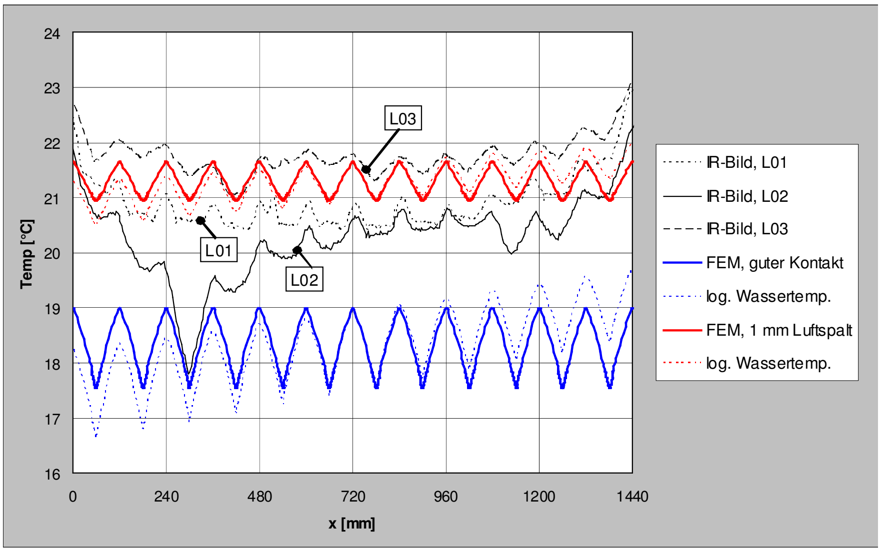


Figure 12: Comparison of FEM-results (good contact and 1 mm air gap) and sections of the IR-survey (as marked in Figure 8)

RESULTS

The thermal activation of a laser-welded steel sandwich panel is possible. The thermal performance was tested by measurements and in parallel investigated with numerical calculations. The theoretical performance is very good, a specific cooling power of 70 W/m² by a mean temperature difference water – room of 8 K was calculated, which is in the top range of existing cooling ceilings. The measured results are lower, a value of ca. 48 W/m² at the same temperature difference can be extrapolated from the measurements. The main reason for this deviations was identified: The steel pipes have good contact to the lower steel sheet only at the fixing points, in between frequently an air gap exist, whereas small gaps (1 mm) lead to a significant loss of power

DISCUSSION

Generally the deck system “LSP” presented in this paper has the potential for a high cooling performance, as the FE-calculations show. To reach this value in practice, the contact between pipe and sheet has to be worked out carefully. The punctual stitched welds lead to unwanted decrease of the cooling power. A continuous welding or other techniques (glue, thermal paste, e.g.) to improve the thermal contact will bring the practical results closer the calculated values.

ACKNOWLEDGEMENT

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A Strategy to Determine a Heating Curve for Outdoor Temperature Reset Control of a Radiant Floor Heating System

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SUMMARY

In many countries, a radiant heating system is gaining much popularity due to its high comfort level, energy efficiency, improved IAQ and so on. A lot of studies have suggested that the supply water temperature of a radiant heating system be modulated by a heating curve, which is a linear function of outdoor temperature. The heating curve, however, has been determined by an operator's experience or a tedious trial and error approach. This study aims at deriving a heating curve to determine supply water temperature based on heat loss of a building and a heat flow model of the floor structure. To evaluate the derived heating curve, TRNSYS simulation and a mock-up experiment were conducted. The results show that the developed heating curve provides optimum water temperature enough to keep the indoor environment comfortable. This study would be able to provide an algorithm for the outdoor reset controller of a radiant floor heating system.

INTRODUCTION

The goal of a radiant heating system is to supply heat at the same rate at which the building loses heat, thereby keeping the indoor environment stable and comfortable. To achieve this purpose, outdoor temperature reset control is widely applied to the radiant heating system. The main principle of the control strategy is to modulate the supply water temperature according to the heating curve or reset ratio, which means the relation between supply water and outdoor air temperature. Although a lot of studies have been conducted to evaluate and improve the performance of outdoor temperature reset control, there has been little research on the determination of heating curve or reset ratio.

Adelman proposed a heating curve based on supply water design temperature and supply water starting temperature, which corresponds to the coldest and mildest outdoor condition, respectively [1]. But he did not explain how water design temperature and water starting temperature should be calculated.

MacCluer used a reset ratio determined by U-value of the wall, thermal resistance of the slab and heat transfer coefficient of floor surface [2]. But the detailed floor structure was not considered in the calculation of a reset ratio. Leigh conducted several experiments on the outdoor temperature reset control and other control methods of a radiant floor heating system. In the experiment, a reset ratio was estimated based on design conditions and thermal characteristics of the building. But the reset ratio was adjusted by performing several pre-tests before the experiments [3].

Gibbs recommended that auxiliary control strategies be combined with outdoor reset control for the multi-zone control [4]. In the research, a reset ratio was set constant but the method to calculate the reset ratio was not described specifically.
Liao et al compared the energy consumption and control performance of the heating system with different heating curves. It was pointed out that the performance declines significantly if a heating curve is not determined properly [5]. The method of determining a heating curve, however, was not explained definitely. Zhang argued that outdoor reset control requires a good relation between zone load, outdoor temperature and supply water temperature. And it was emphasized that the absence of accurate knowledge of such a relation could result in either overcorrection or poor control [6].

In most residential buildings, however, a heating curve is determined by an operator's experience or time-consuming trial and error procedure. This makes supply water temperature determined incorrectly, with heating load, thermal characteristics of a building and heat flow from the radiant heating floor ignored.

The objective of this study is to derive a method to determine a heating curve which considers heating load of the building and heat flow from radiant heating floor. The result of this study could help to determine optimum supply water temperature for the enhancement of indoor thermal environment.

FACTORS OF A HEATING CURVE

Heating load is the first factor to influence the heating curve of a building. As it is general that the building is under steady-state when heating system is operated continuously, heating load can be expressed as follow:

$$Q_{load} = \sum_{exterior} U_i A_i (T_{set} - T_{out}) + \sum_{interior} U_j A_j (T_{set} - T_{ad,j}) + V_{air}(T_{set} - T_{out}) \quad \text{(W)}, \qquad (1)$$

where first two terms in the right side of the equation are heat loss through exterior walls and interior walls respectively. And the last term is heat loss by infiltration or ventilation. Solar gains and internal heat sources were not included so that the gains could act as a safety factor.

The second factor of a heating curve is heat flow from the radiant heating floor. Heat flow is determined by various parameters of floor structure such as pipe spacing, thermal conductivity and thickness of the screed, thermal resistance of surface covering and so on. According to EN 1264, heat flow between embedded pipes and the space is calculated by the general equation as follows [7]:

$$Q_{panel} = A_f B \prod_i (a_i^{m_i}) \Delta\theta_H \quad \text{(W)}, \qquad (2)$$

where A_f is an area of heating surface, and B is a system coefficient dependent on system type and the heat exchange coefficient (for floor heating, B=6.5W/m^{2}K). $\prod_i (a_i^{m_i})$ is the power product which links the parameters of the floor structure (surface covering, pipe spacing, pipe diameter and pipe covering) . $\Delta\theta_H$ is heating medium differential temperature determined by following equation.

$$\Delta\theta_H = \frac{T_{sup} - T_{ret}}{\ln \dfrac{T_{sup} - T_{set}}{T_{ret} - T_{set}}} \quad \text{(K)}, \qquad (3)$$

where T_{set}, T_{sup} and T_{ret} are room air temperature, supply water temperature and return water temperature, respectively.
In other words, heat flow from the radiant heating floor is proportional to $\Delta\theta_H$ and $B\prod_i (a_i^{m_i})$ can be considered as a heat transfer coefficient for the surface of a radiant heating floor. Figure 1. shows the factors influencing the heating curve of a building.

DERIVATION OF A HEATING CURVE

If a building is maintained steady-state with continuous heating, heat output from the floor is equivalent to heating load. And if downward heat flow is restricted with insulation and the lower space is heated, as in a multi-family house, heat output from the floor is also equivalent to heat discharged by hot water in the embedded pipe, which is calculated as follows:

$$Q_{water} = m_w \cdot c_{p,w} \cdot (T_{sup} - T_{ret}) = m_w \cdot c_{p,w} \cdot \Delta T \quad \text{(W)}, \qquad (4)$$

where m_w is a mass flow rate of hot water (kg/s), $c_{p,w}$ is a specific heat capacity of hot water (J/kgK), ΔT is a temperature drop of hot water (K).
Combining equation (2), (3) and (4), supply water temperature T_{sup} can be calculated as

$$T_{sup} = T_{set} + \alpha \Delta T \quad \text{(K)}, \qquad (5)$$

where α is a coefficient dependent on the geometry and thermal property of the radiant floor, expressed as following equation.

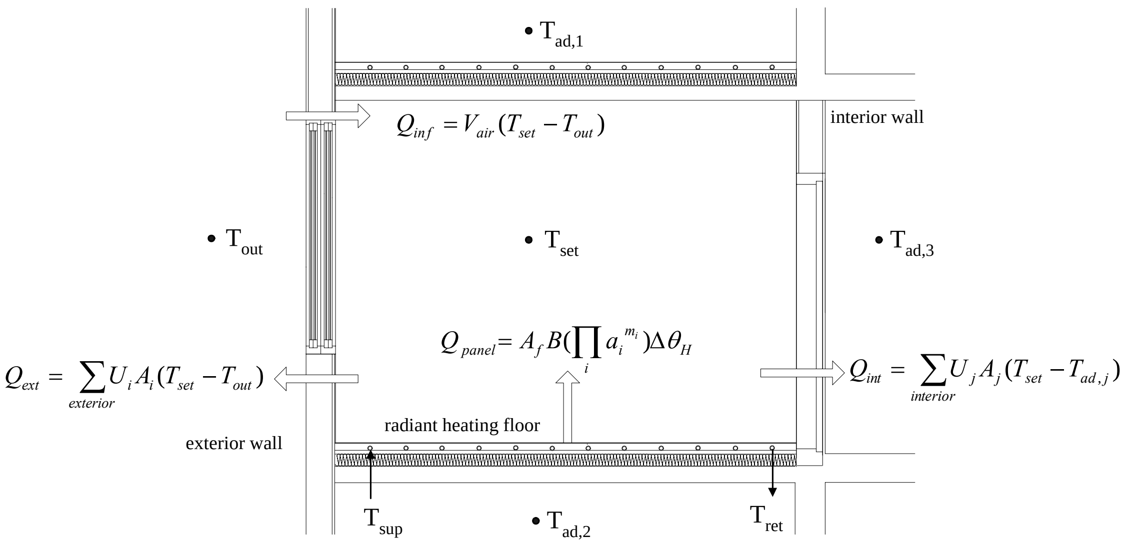


Figure 1. The factors related with the heating curve of a building

$$\alpha = \frac{1}{1 - \exp\left(- \frac{A_f B \prod_i (a_i^{m_i})}{m_w \cdot c_{p,w}} \right)} \qquad (6)$$

On the other hand, ΔT in equation (5) can be obtained by combining equation (1) and (4). Thus ΔT can be replaced with following equation (7).

$$\Delta T = \frac{(\sum_{exterior} U_i A_i + V_{air})(T_{set} - T_{out}) + \sum_{interior} U_j A_j (T_{set} - T_{ad,j})}{m_w \cdot c_{p,w}} \text{(K)} \qquad (7)$$

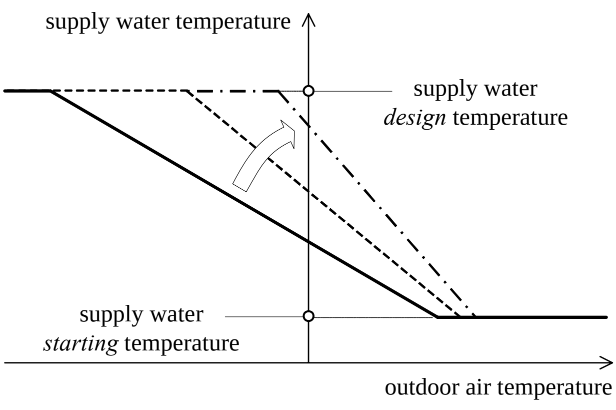
Putting equation (7) into equation (5) and arranging the equation with regard to T_{out} gives a final result, that is, a heating curve expressed as a linear function of outdoor air temperature.

$$T_{sup} = -\left(\frac{\alpha(\sum_{exterior} U_i A_i + V_{air})}{m_w c_{p,w}} \right) T_{out} + \left(1 + \frac{\alpha(\sum_{all} U_i A_i + V_{air})}{m_w c_{p,w}} \right) T_{set} - \frac{\alpha}{m_w c_{p,w}} \sum_{interior} U_j A_j T_{ad,j} \quad (8)$$

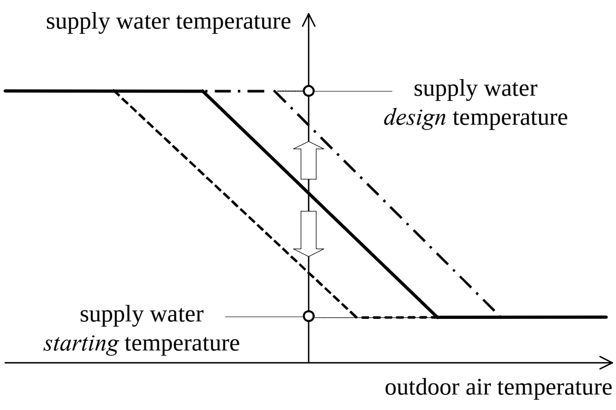
According to equation (8), the slope of heating curve gets steeper as the heat loss of a building becomes larger as described in Figure 2. a). On the other hand, the change of set temperature and adjacent room temperature tends to shift up or down without changing the slope of the heating curve, as illustrated in Figure 2. b).

EVALUATION OF THE DERIVED HEATING CURVE

In order to examine whether the derived heating curve can assure optimum supply water temperature, a simulation and a mock-up experiment were conducted. The analyzed building, located in the vicinity of Seoul, Korea, consists of two test chambers equipped with radiant floor heating system as illustrated in Figure 3. A radiant floor structure conforms to Type A in EN 1264, which has pipes inside the screed.

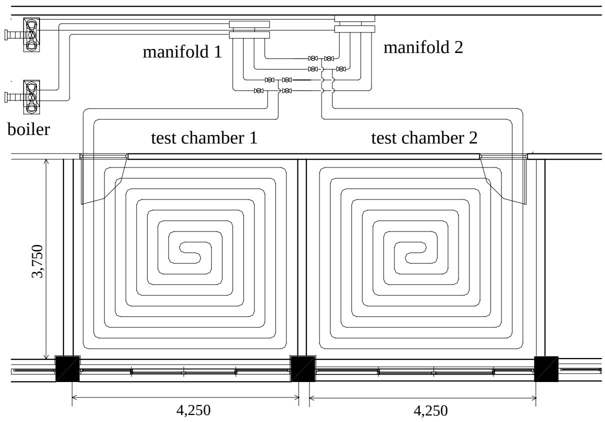


a) the change of heating curve slope

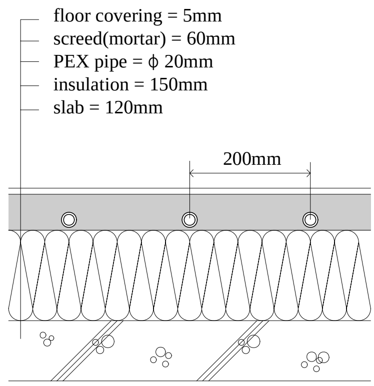


b) the shift up or down of heating curve

Figure 2. The characteristics of a heating curve



a) The plan of test chambers



b) floor structure

Figure 3. Test chambers for the mock-up experiments

Simulation Analysis

Based on the thermal properties and design condition of the chambers, heating curve was estimated as T_{sup} = -0.887T_{out}+43. In order to analyze the performance of the chamber with the estimated heating curve, a TRNSYS type which can modulate supply water temperature according to a heating curve was developed. The type receives an outdoor air temperature, calculates water temperature with the heating curve and transfers water temperature to the active layers of the analyzed thermal zone. But it does not receive the feedback from the room because outdoor temperature reset control is basically open-loop control algorithm.

The simulation was performed for the whole winter season using standard weather data of Seoul. The effect of internal heat sources was not taken account into the simulation. Figure 4 and Figure 5 describe the simulation result for the whole season and coldest 3 days, respectively. Although the heating system did not receive a feedback from the room, room air temperature was kept at set temperature, 23℃, by just modulating supply water temperature according to outdoor air temperature. Figure 5 shows the performance of the heating system more clearly. Room air temperature is maintained stable without much fluctuation, as in a conventional on/off control heating system. Thus it can be said that the derived heating curve can assure the optimum supply water temperature to keep the heated space comfortable.

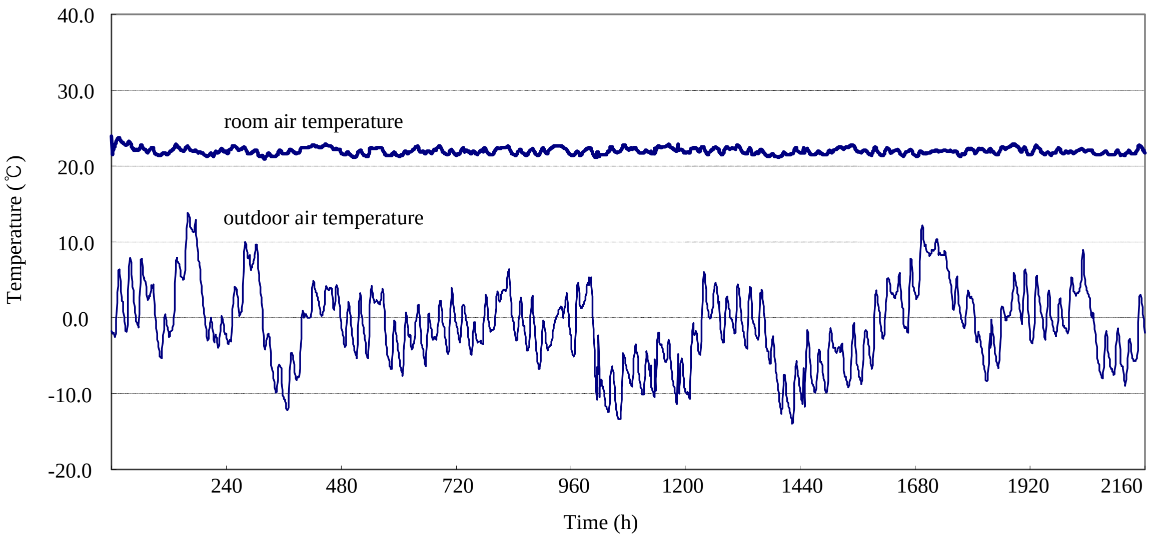


Figure 4. Simulation result for the whole winter season

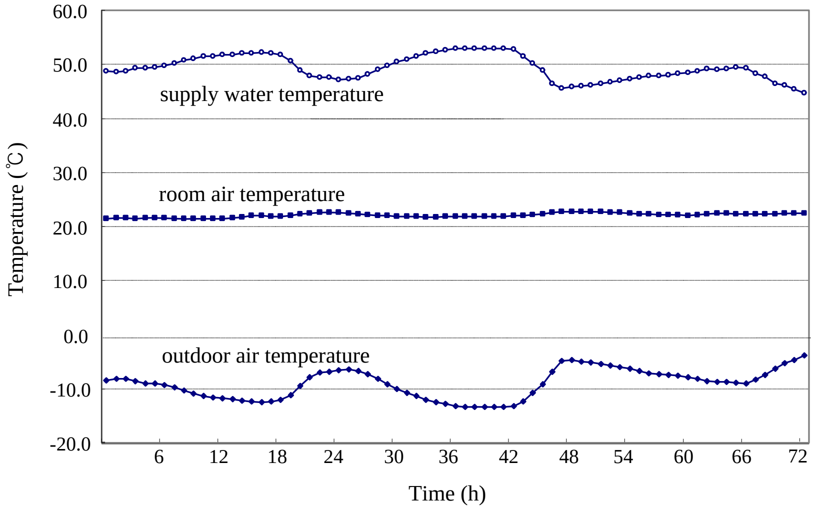


Figure 5. Simulation result for the coldest 3 days

Mock-up Experiment

For the purpose of more accurate evaluation, a mock-up experiment was performed. The derived heating curve was applied to the aforementioned test chambers. The chamber, water distribution system and control instruments used in the experiment are presented in Figure 6. As it is the main purpose to analyze the relation between outdoor air temperature and supply water temperature, a sun shade device was installed on the windows of a chamber.

a) a test chamber　b) gas-fired boiler　c) distribution system　d) control devices

Figure 6. Equipments for the mock-up experiments

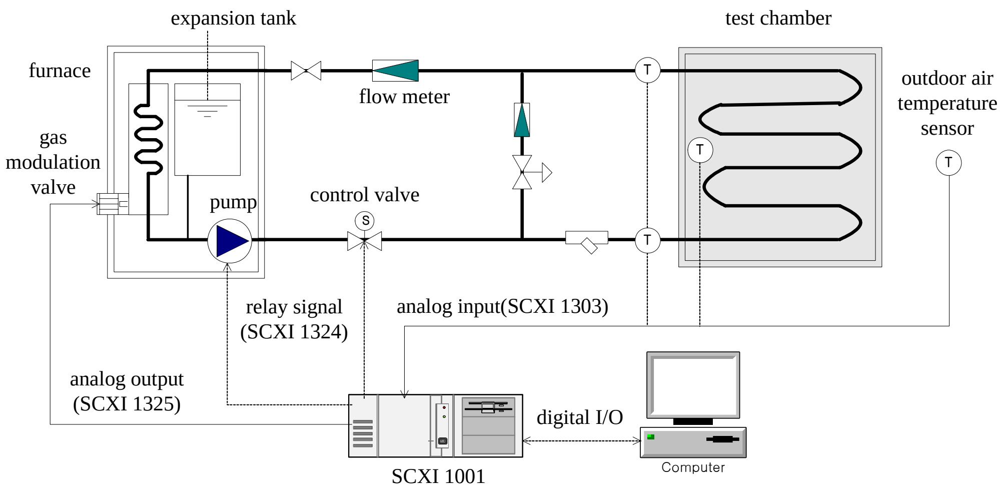


Figure 7. The schematic diagram of the instruments

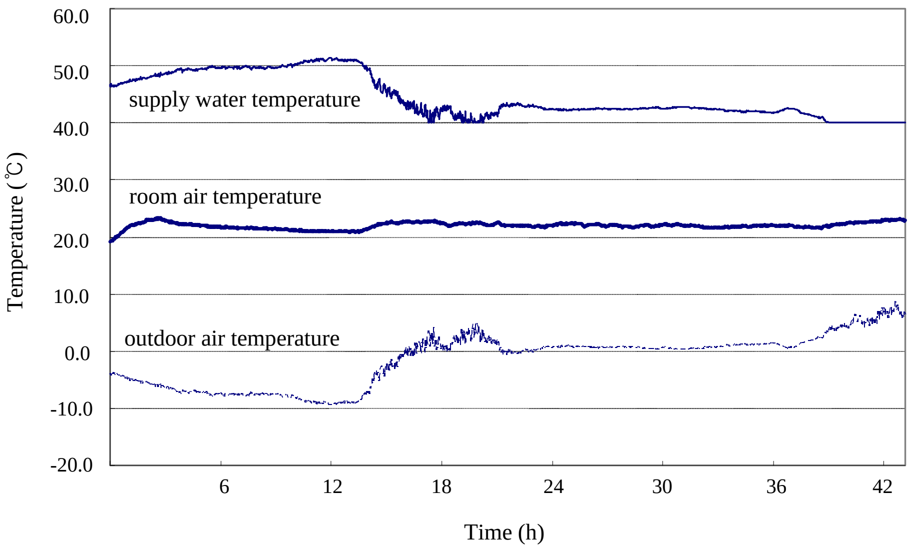


Figure 8. The result of a mock-up experiment

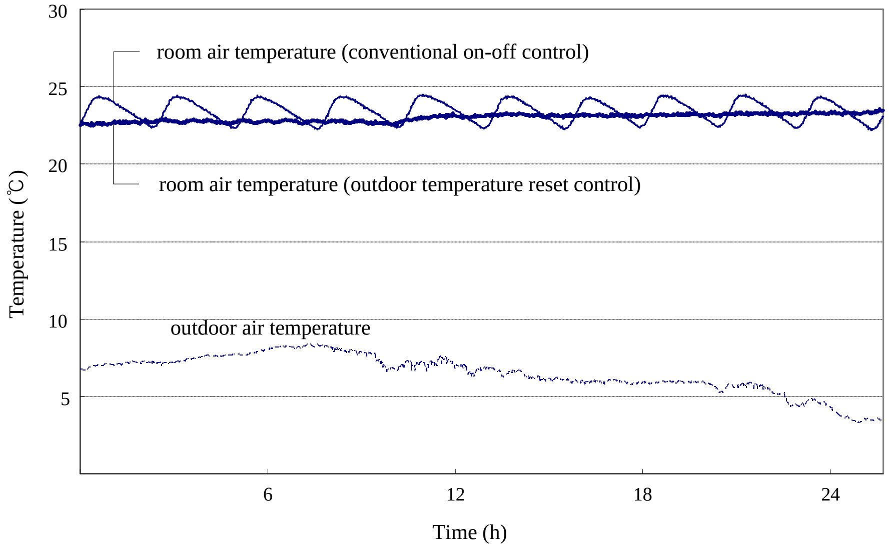


Figure 9. The comparison with a conventional on-off control

To modulate the supply water temperature, the firing rate of a gas-fired boiler was proportionally modulated with the analog input and output devices (NI SCXI-1001). Figure 7 describes the schematic of the equipments for the mock-up experiment. Supply water temperature was calculated by a heating curve, but maximum and minimum temperature were constrained to 80℃and 40℃ respectively for the purpose of preventing an overheating or flue gas condensation in a boiler.

Figure 8 shows the results of the experiment to examine whether the derived heating curve results in a good control of room air temperature. With the modulation of supply water temperature, room air temperature was maintained at averagely 23.1℃, with slight difference with set temperature, 23℃. Moreover, overshoot and/or undershoot, which occur frequently in conventional on/off control, were not observed in the experiment as shown in Figure 9. In conclusion, the derived heating curve can assure the optimum supply water temperature which can supply heat at the same rate at which the building loses heat.

CONCLUSIONS

In this study, a method to determine heating curve of a radiant heating system was proposed and evaluated. The heating curve could be expressed as a linear function of outdoor air temperature. It was also found that the slope of a heating curve varies with the heat loss of exterior walls and air change rate of the room. And set temperature of the room and adjacent room temperature would shift up or down of a heating curve.

The results of a simulation and a mock-up experiment show that the derived heating curve yields an optimum supply water temperature enough to keep the room stable and comfortable. The room air temperature was maintained closely at designed set temperature, without any feedback from the room. It can be said that the modulation of water temperature according to the derived heating curve provides as much heat as the building loses.

This study, however, assumes no solar gain and internal heat source. Those gains might result in slight increase of the room air temperature because they provide additional heat to the room. Thus it is proper to use outdoor reset control in combination with an auxiliary control to prevent additional heat supply. Outdoor temperature reset plus on-off control can be a good alternative. In addition, this study assumes that the heating system operates continuously and the room is under steady-state. Thus a heating curve should be modified so that it can cope with the dynamic condition such as an initial boost-up operation, a recovery from deep setback and so on.

Despite of those limitations, this study can be said to be significant because it provides a basic strategy to determine optimum supply water temperature, which considers heating load and the floor structure of a building. The results can be applied to the outdoor reset control algorithm of a heat exchanger or a boiler for a radiant heating system.

ACKNOWLEDGEMENT

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The Mine Water Project Heerlen, the Netherlands - low exergy in practice

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SUMMARY

In Heerlen, the Netherlands, warm and cold water volumes from abandoned mines is used for heating and cooling of buildings, based on a low exergy energy infrastructure. The combination of low temperature heating and cooling emission systems, advanced ventilation technologies and integrated design of buildings and building services provide an excellent thermal comfort and improved indoor air quality during 365 days/year, combined with a CO2 reduction of 50% in comparison with a traditional solution.

INTRODUCTION

Abandoned and flooded mines have a high potential for geothermal utilization as well as heat cold storage of water volumes in remaining underground spaces. The use of heat and cold from mine water is one of the important aspects of rational and sustainable utilization of post mining infrastructure and may bring positive socio-economic results, social rehabilitation and improved health for communities living in European areas with (former) mining activity. In Heerlen, the Netherlands, the redevelopment of a former mining area, including a large scale new building plan, is being realised with a low exergy infrastructure for heating and cooling of buildings, using mine water of different temperature levels as sustainable source. Mines have large water volumes with different temperature levels. In Heerlen the deeper layers (700 – 800 m) have temperatures of 30 – 35^0 C; shallow layers (200 m) of 15- 20^0 C. These water volumes can be considered as heat/cold storage as well as geothermal sources. Most crucial however is that these sources provide low valued energy (low exergy). As on the demand side heating and cooling for buildings also require low valued energy the intended design strategy is to realise the climatisation of the buildings in this pilot directly by mine water. The combination of low temperature emission systems with advanced ventilation technologies and integrated design of buildings and building services provide an excellent thermal comfort for 365 days a year, including sustainable heating and cooling and improved indoor air quality. This sustainable energy concept gives a reduction of primary energy and CO_2 of 50% in comparison with a traditional concept (level 2005).The project is funded by EC Interreg IIIb, the UKR program of the Dutch ministry of Economic Affairs and the EC FP6.

THE ENERGY CONCEPT

The mine water energy concept in Heerlen is in principle as follows. Mine water is extracted from different wells with different temperature levels. In the concession of the former ON III mine (location 1 Heerlerheide) mining took place to a level of 800m. In this concession the warm wells (30^{0}C) can be found. In the former ON I mine (location 2 Heerlen SON) mining took place to a level of 400m and here the cold wells are situated. The extracted mine water is transported by a primary energy grid to local energy stations. In these energy stations heat exchange takes place between the primary grid (wells to energy station) and the secondary grid (energy station to buildings). The secondary energy grid provides low temperature heating (~ $35^0 - 40^0$ C) and high temperature cooling (16^0 - 18^0 C) supply and one combined return (20^0 - 23^0 C).

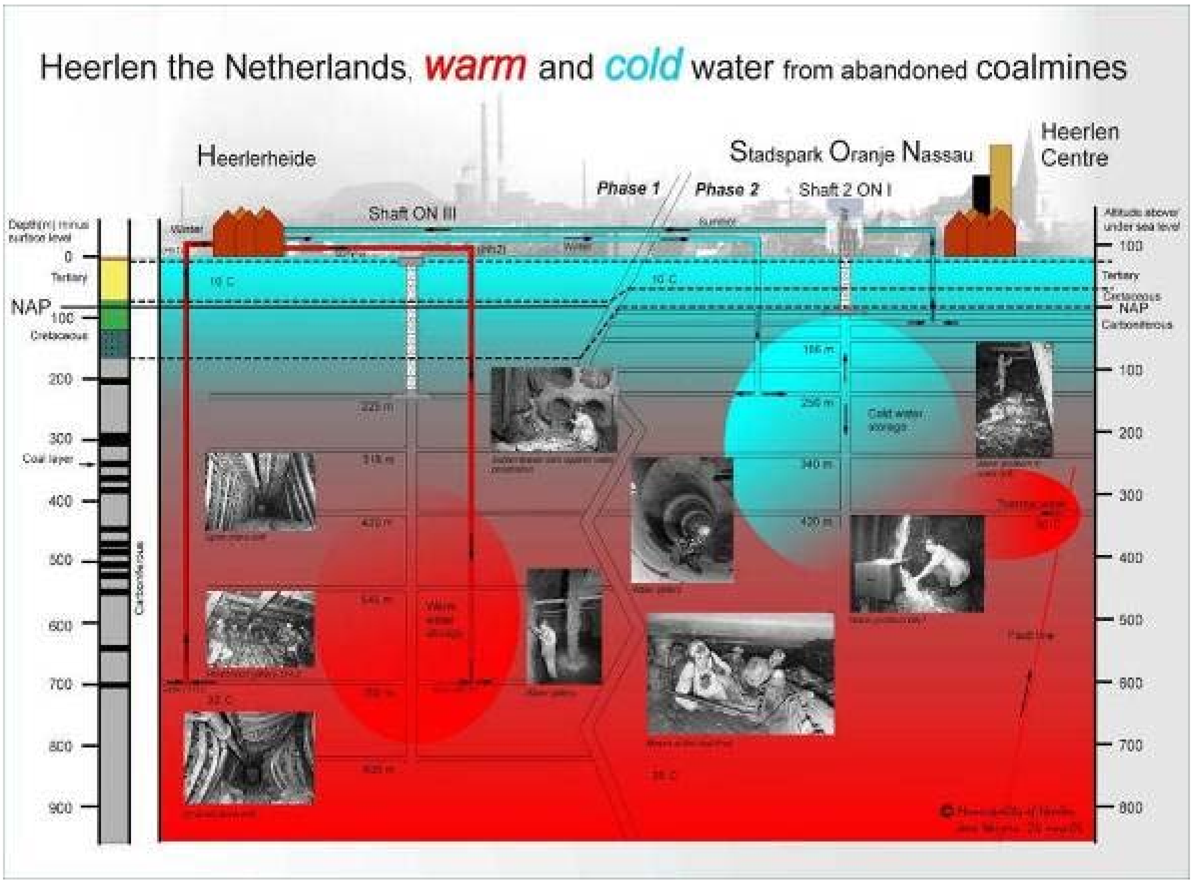


Figure 1. Schematic cross section of the underground conditions of the ON I and ON III mines

The two locations 1 and 2 are connected by a pipeline. Warm water is partly transported from location 1 to 2 and, visa versa, cold water is partly transported from location 2 to 1. Return water of 22 to 23^0C is transported to an intermediate well (400m). The temperature levels of the heating and cooling supply are “guarded” in the local energy stations by a polygeneration concept existing of heat pumps in combination with gas fired boilers and CHP on (preferably) on biomass in combination with local storage tanks. The (biomass fired) CHP provides the electricity to power the heat pumps but also the higher temperature levels (65 – 70^0C) for domestic hot water and peaks during extreme conditions. As, by this integral approach, the demand profile of DHW is almost equal to electricity, the CHP can be designed in the most economic and energy efficient way. The surplus of heat in buildings (for example, in summer, cooling, process heat) which can not used directly in the local energy stations can be lead back to the mine water volumes for storage. If necessary local sub-energy stations in buildings for preparation of DHW by heat pumps, small scale CHP or condensing gas boiler, depending on type of building and specific energy profile. The total system will be controlled by an Intelligent Energy Management System including telemetering of the energy uses/flows at the end-users. A scheme of the total concept is given in figure 2.

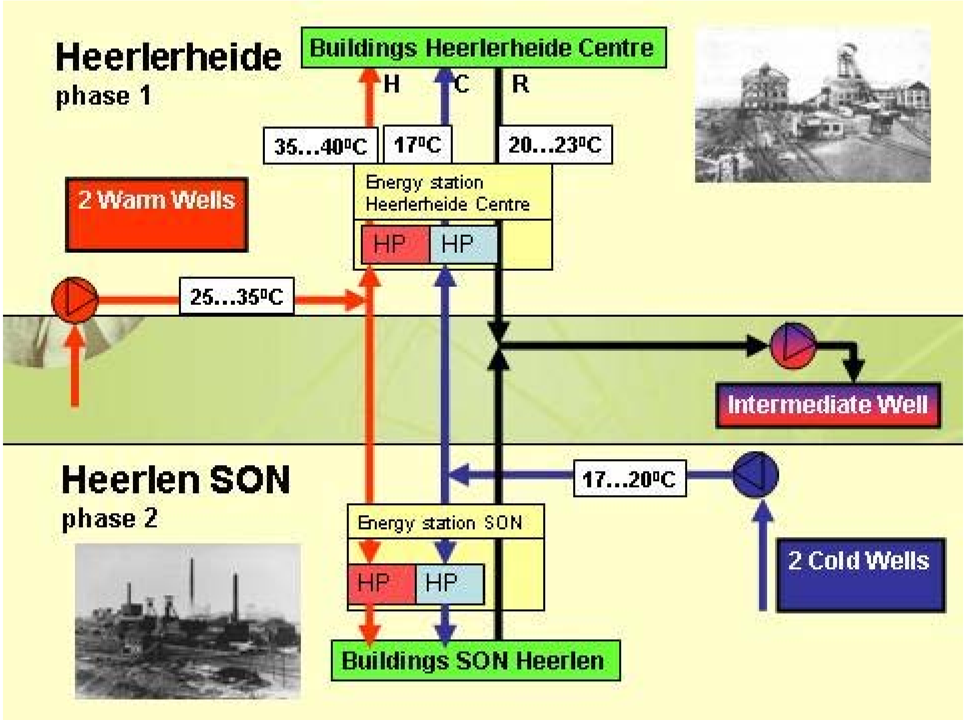


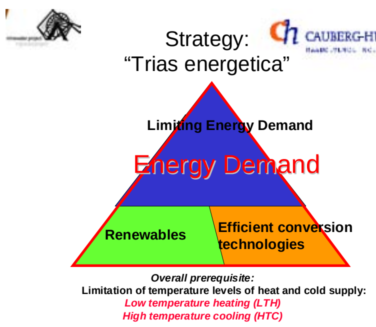
Figure 2. Schematic view of the energy concept in Heerlen, connection of the wells and energy stations

INTEGRATED DESIGN APPROACH VERSUS TRADITIONAL APPROACH

The present development of energy efficient buildings in an increasing way requires an integral design approach. A couple of decades ago energy efficient design and building mostly focussed on improving a certain technique or apparatus. Nowadays an energy efficient building, supported by an energy efficient installation, has to be combined into one integrated energy efficient concept with an optimal performance in terms of indoor climate, thermal comfort, user's satisfaction etc. This asks for an integral design approach in which well balanced choices are being made. This means that in sustainable building projects it is crucial to consider the design and realization of the sources, the heat generation (especially with non-traditional solutions such as heat pumps, cogeneration, heat/cold storage) distribution and emission together including all possible interactions with the building, building properties and building users. Only this approach can lead to a set of well defined performance criteria concerning energy performance, sustainability, indoor air quality, thermal comfort (365 days/year, winter and summer conditions), and health. Next to it is necessary to have specific emphasis on investments and energy exploitation, as well as communication to the end-users. A traditional approach is often based on partial optimization of the different disciplines. An integrated approach will achieve a total optimization, taken into account all disciplines and their interaction. Basis is a set of unambiguous well defined performance criteria. The design strategy applied in this approach is the so called Trias Energetica. It is a three step approach that gives a strategy to establish priorities for realising an optimal sustainable energy solution. The approach is introduced in 1996 by Novem the Netherlands and has been further worked out by the Technical University of Delft the Netherlands, containing the following steps:

Step 1: Limitation of energy demand
Step 2: Maximising share of renewables
Step 3: Maximising efficiency of using fossil fuels for remaining energy demand

With as overall prerequisite: limit the temperature levels of heat and cold supply (conform 2nd law of thermo dynamics)



In general the heating and cooling of buildings can be realized with very low valued energy, with medium temperatures close to required room temperatures. The better the building properties (extreme high thermal insulation, suitable emission systems) the closer the temperatures of heat an cold supply can be to room temperatures. In order to utilise these extreme moderated temperatures for heating and cooling the buildings must comply to a number of boundary conditions such as:

- Limitation of heat losses ($U_{envelope} < 0.25$, $U_{windows} < 1.5$)
- Limitation of ventilation losses and peaks by air tight building (n50 < 1.0), mechanical ventilation with heat recovery or state of art demand controlled hybrid ventilation systems
- Limitation of solar and interal gains to limit cooling loads, integrating shading and sun blinds in architectural design
- Application of combined low temperature heating and high temperature cooling emission systems, (thermally activated building components, floor and wall heating).

Technologies for low temperature heating and high temperature cooling are available on the market an are described for example in the IEA Annex 37 Guidebook (www.lowex.net)
For some functions higher temperatures will be necessary such as domestic hot water. Also lower temperatures can be necessary for certain functions (high cooling loads for some types of buildings, some processes, etc.). Another aspect to be taken into account is that the use of geothermal energy and heat/cold storage as such does not cover electricity use/sustainable electricity generation. Therefore additional sustainable solutions have to be taken into account. Sustainable electricity generation can be realized by cogeneration (such as biomass CHP). This combination can also deliver higher temperatures for DHW.

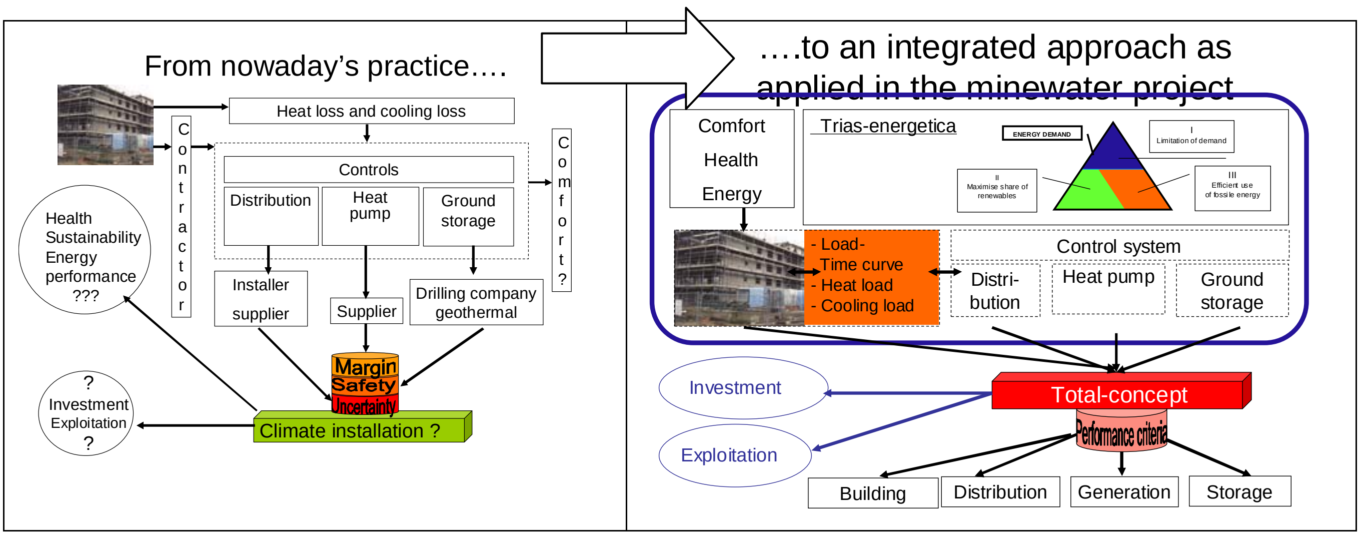


Figure 3. From a traditional to an integrated design approach

THE DEMONSTRATION LOCATIONS

1 Location Heerlerheide Centre:

This plan is situated on the concession of the ON III pit in a relatively deep mined area with warm wells (30 – 35^0). The plans include the following activities for ***new buildings:***

- 33.000 m^2 (330) dwellings (single family dwellings and residential buildings)
- 3800 m^2 commercial buildings
- 2500 m^2 public and cultural buildings
- 11500 m^2 health care buildings
- 2200 m^2 educational buildings

The first new building and construction activities in Heerlerheide Centre have started in 2006. The total plan will be realised between 2006 and 2008. All planned buildings will be connected to the energy supply (heating and cooling) from mine water. All these buildings are planned in a very compact area which is very favourable for energy distribution. The building location is situated between two potential wells. This means that the length of the transport lines between the wells and the energy station is limited. Next to it, the planned building functions require heating as well as cooling. The location of the wells has been determined as a result of geological research. The drilling of the wells took place from February to June 2006. The two warm wells and the primary net (i.e. the connection between the two wells) was completed in June 2006, followed by a successful testing in July. The energy supply includes the building of an energy station and a small scale distribution grid from this station to the buildings. In the energy station the mine water is brought to the necessary heating and cooling levels by heat pumps. In order to facilitate the process and to guarantee all real estate developers, involved in this building plan, the delivery of energy to the buildings the main investor, Housing Corporation Weller, is realising the exploitation of the energy supply, including the building and construction of the energy station and distribution grid. The liberalised energy market in the European Union makes it possible for housing corporations to exploit the energy supply for their buildings. It is important to realise, that with minor modifications this energy supply can also be functional and operational without the application of mine water. In that case the heat pumps will be connected to closed loop ground heat exchangers (separate bore holes). Although this option is less energy efficient in comparison with the use of mine water as source this will be still much more energy efficient then the traditional option with condensing gas-fired boilers and electrical cooling devices for air-conditioning.

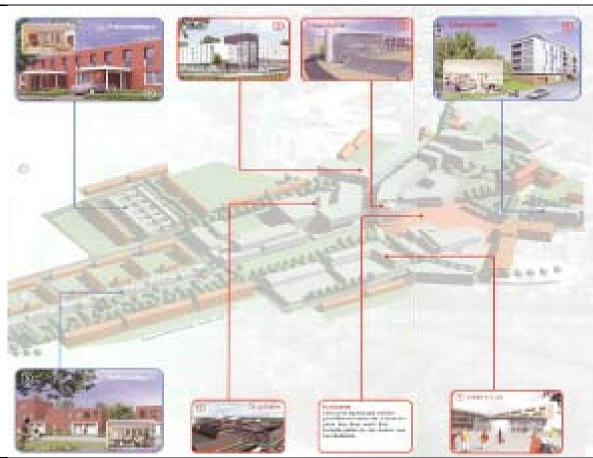
	
Old situation Heerlen, location Heerlerheide	New situation

Figure 3. Impression of location 1 Heerlerheide

2 Location SON (Stadspark Oranje Nassau)

This area is situated on the concessions of the ON I pit. This is a relatively shallow mined area (200 – 400 m) with cold and intermediate wells. There for this area is connected with a master connection to the warmer wells in the ON III area (Location Heerlerheide). Cold form the ON I area is transported to Location Heerlerheide Centre. The development of Stadspark Oranje Nassau has a strategic significance for the social and economical rehabilitation of Heerlen. This plan will offer an opportunity to improve the image of Heerlen, to develop new economical opportunities and to enhance and stimulate the existing (but poorly functioning) activities inner city of Heerlen. This plan will be realized in combination with sustainable mobility and accessibility. The total programme contains the realisation of approximately 100.000 m^2 of new buildings (offices, shops, residential, school and a hotel) and the renovation of a large existing office building (43.500 m^2) of the Dutch Central Office of Statistics.

3. Location Heerlen centre ABP head office

This location concerns the retrofitting of the ABP head office of 41.000 m^2. ABP is the pension fund for employers and employees in service of the Dutch government and the educational sector. The total building envelope is retrofitted to a level better then the current Dutch Building Decree values for new buildings. The minewater is used for comfort heating and cooling (i.e. low temperature heating and high temperature cooling in all offices). The ABP building will have a direct connection to the minewater wells and will have its own energy station to provide the required temperature levels for the distribution net. The energy station will have heat pumps. The emission systems in the offices are climate ceilings. Special glazing will be used to limit solar radiation in summer; this makes it possible to use high temperature cooling (in most of the time direct from minewater).

Maankwartier (Moon Quarter)

Renovation of the CBS office

Figure 4. Impressions of location 2 Stadspark Oranje Nassau (SON)

BALANCING SUPPLY AND DEMAND SIDE

For the elaboration of the final energy concepts following questions should be answered:
- total heating and cooling demand, how to control and limit this demand
- the target values for percentage of renewables in total energy demand
- what is the available amount of renewable energy form mine water (i.e. how much water can be extracted) and other renewables
- what is the most efficient conversion technology for the (not sustainable) back-up system.

This input is necessary for the integrated design process including building, sources and energy systems, distribution and emission systems. An important tool for the assessment of this process and balancing demand and supply side is the so called energy profile of a building, expressed in a so called load-duration curve, based on dynamic calculations (using TRNSYS) of the energy demands of the buildings. This curve is a profile representing the energy demand over a total year, including heating and cooling. This curve also provides a good indication of the maximal capacities for heating and cooling as well as the balance between heating and cooling demand. Important for balancing the supply and the demand side is the tuning and balancing between the cold and heat sources, in this case, the deep (warm) and shallow (cold) wells. This assessment takes place in relation to the required temperature levels, the yearly extracted volumes and the energy demands of buildings; this in relation to the available water volumes in the reservoirs. The load duration curves give important information about:
- the balance between cold and heat demands,
- the effect of optimisation (for example limiting heat losses by thermal insulation or heat recovery, etc.)
- the way how to limit the installed capacity of heat pumps, CHP and other heat generation, and , on the other hand, how to increase the number of operation hours, in combination with storage, to increase the efficiency and to decrease investment costs. Also thermal impact on the ground is made visible and can be assessed.

In order to establish a balance between the rational use of energy needs on the building side and the renewable energy supply a total annual heat-load duration curve of the total building plans in Heerlerheide Centre and SON is calculated by dynamic simulations with TRNSYS. In figure 5 the heat-load duration curve for Heerlerheide (location 1) is shown.

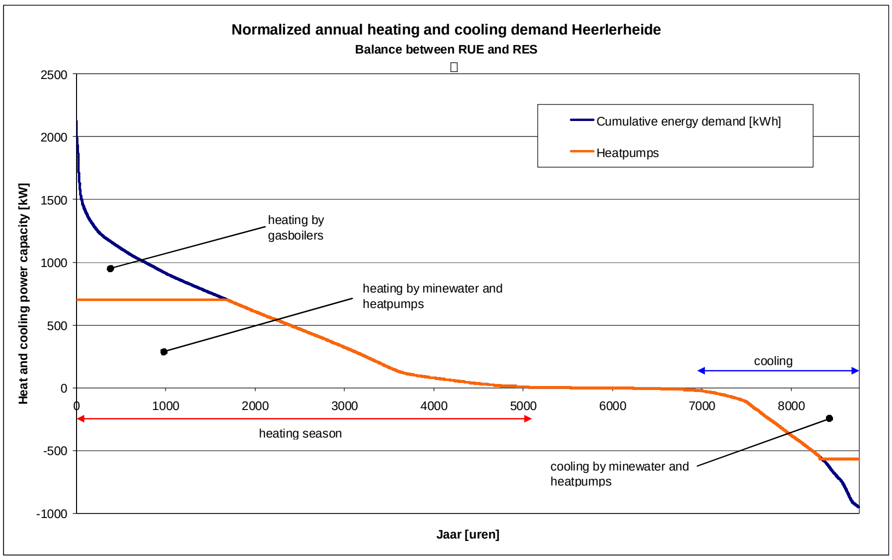


Figure 5. Load-Duration Curve for location 1 Heerlerheide

FEASIBILITY BY PRIVATE ORGANIZED ENERGY EXPLOITATION

Despite the rather high level of investments for the energy installations and buildings measures this concepts is economically feasible by private organized energy exploitation. In this case, the main investors will also organise the energy exploitation, i.e., in separate private owned Energy Exploitation constructions). These private organized companies can use lower internal interest rates, 6 to 8% instead of the usual 15% of utilities and district heating companies. The main reason is that profits from selling energy is not considered as core business. By establishing connection fees for heating and cooling and avoiding a gas infrastructure on building/dwelling level, as well as avoiding extra cooling installations, these constructions offer possibilities for economical sound energy exploitation. Economical benefits will also occur because of the integrated design and especially combining heating and cooling in the same emission system (i.e. floor heating and cooling, thermally activated building components etc.). Using these combined emission systems avoids the investment costs for a separate cooling system

CONCLUSIONS

Abandoned and flooded mines can be reutilized for a new sustainable energy supply for heating and cooling of buildings. The minewater project in Heerlen shows that temperatures of 28 – 30 ^{0}C can be found at 700 m and 16 -17 ^{0}C at 200 m. These temperatures can be used for heating and cooling of buildings if these buildings are very well insulated, have energy efficient ventilation systems and have emission systems suitable to operate with moderated temperatures like floor heating or concrete core activation. Despite the rather high investment costs such projects can be economical profitable avoiding additional cooling systems and by integrated design and if energy exploitation is organised by the investors.

ACKNOWLEDGEMENTS

The Minewater project is funded by the European Commission and the Dutch Ministry of Economic Affairs. These fundings are gratefully acknowledged.

Energy Performances Of A Radiant Floor Heating System Supplied By Solar Collectors With Ventilation Stream Heating By An Air To Air And An Air To Water Heat Exchanger

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SUMMARY

The energy analysis of a heated space with a radiant floor plant supplied by hot water provided by a storage system charged by a field of solar collectors, has been evaluated. A suitable control strategy has been evaluated to reduce thermal inertia effects. Given an external air-change ventilation flow rate must be provided, a system using two heat exchangers has been used: the first an air to air heat exchanger that uses the expulsion air from the heated space as hot fluid, and a second air to water heat exchanger that uses the outlet water from the radiant floor. The whole system has been studied by dynamic simulation code TRNSYS, to evaluate interaction between the devices and, overall, the contribution of the solar source on plant performance by the solar fraction, varying the collectors' surface and the storage system volume.

INTRODUCTION

Energy consumption in the building field is anything but ignorable considering that today, in the European Community, such a voice concurs a percentage higher than 40%. Some recent community directives [1], promote the energy consumption reduction in the winter heating of buildings. This result is obtainable by means of multiple solutions such as efficient building insulation of opaque surfaces, a wise choice of glass surfaces, the adoption of high efficiency thermal energy production, the use of low temperature energy distribution systems, powered by a supply coming from a field of solar collectors. The latter solution assumes greater value for houses situated in localities with cold climates, whose requirements for thermal energy for heating are considerable compared to other energy requirements of the building.
Among the various types of thermal energy distribution systems used for heating areas, radiant floor heating systems are those which permit the best direct use of solar radiation as a primary energy source. In fact, the floor surface temperature must be moderated so that a relatively low temperature fluid can be used which is obtainable from the solar source with elevated efficiency [2]. Another advantage deriving from the use of radiant floor heating systems lies in the reduction of winter energy requirements. The increase of average temperature radiating within the area permits the attainment of the same operative temperature with lower internal air temperatures, with a consequent reduction in transmission and ventilation losses through the building shell. On the other hand, radiant floor systems, compared to traditional ones, are characterised by notable thermal inertia, therefore the adoption of control systems which take into account this peculiarity is necessary [3]. In this article, thermal performance of radiant floor heating systems powered by an storage system

which receives energy from a field of solar collectors, with preheating of the ventilation airflow obtained by exploiting the building exiting airflow from an air to air exchanger and, successively, the exiting flow rate from the radiant floor in a second air to water exchanger are evaluated.

DESCRIPTION OF THE BUILDING USED IN THE SIMULATION

The building is considered as being for office use, it has a regular parallelepiped shape, with the longest side being 16m, and the shortest being 6m and a height of 2.75m. The walls of the largest surface area are North and South facing. On each external wall and triple glazed window is present having a global loss coefficient equal to 1.8 W/m²K and a solar gain coefficient of 70%. The glazed surfaces are respectively 8.8 m² for the south facing wall, and 3.3 m² each for the remaining three exposures. The total glazed surface area is therefore 22 m² while the total opaque dispersant is equal to 291 m². In Tab. 1 the thermal properties of the various opaque components of the building shell are reported. The thermal transmittance values are 0.198 W/m²K for the vertical walls, and 0.146 W/m²K for the covering floor and 0.207 W/m²K for the floor placed on the ground.

Table 1. Layers thermal characteristic of opaque walls.

	Thickness	Conductivity	Density	Thermal Resistance
	m	W/mK	Kg/m^3	m²K/W
Plasterboard	0.018	0.9	1800	0.02
Mineral wool with moisture stop	0.125	0.048	11	2.60
Mineral wool rigid support	0.063	0.035	100	1.80
Air gap	0.03	-	-	0.18
External surface bricks	0.075	-	650	0.28
Wooden beams and battens	0.045	-	850	0.25
Covering material in tile-concrete	0.055	-	1800	0.03
Floor covering material	0.01	1	1800	0.01
Light Concrete	0.09	1.2	1800	0.08
Heavy concrete	0.10	1.2	2000	0.08
Rigid panel insulating	0.10	0.035	37	2.85
Ground floor loose stone foundation	0.50	-	2000	0.28

In order to determine the internal energy gains it is supposed that the building is used as an office with a number of people equal to 12, present from 08.00 to 13.00 and from 14.00 to 18.00, from Monday to Friday, who work with the same number of computers. For each person a sensible load of 65W is estimated and a latent load of 55 W [4]; each computer transfers a sensible load of 80W. The lighting plant is active from 8.00 to 18.00 and causes a thermal flux equal to 5 W/m², for a sensible load of 480 W. Finally, with regards to ventilation, for each occupier an air-flow rate of 11 l/s was estimated [5].
The radiant floor properties are as follows [6]:

- pipes in polymeric material with k=0.33 W/mK, internal diameter of 0.01 m and a tube pitch of 0.15 m;
- light concrete layer with a thermal conductivity equal to 1.2 W/mK and a thickness of 0.1 m, covering material floor with k=1 W/mK and a thickness of 0.01 m.
- an inlet flow rate equal to 1191 kg/h;

DESCRIPTION OF THE HEATING SYSTEM

The plant of the system considered is reported in Fig. 1. The inlet flow rate of the radiant floor is taken from the storage tank which receives energy from the collectors field. In the case in which the temperature of the storage system is higher than that requested, a mixing by means of valves 1 and 2 is carried out to reach the desired temperature. In the case that the temperature is lower, an auxiliary system intervenes by means of the commutation of valves 3 and 4. The auxiliary system is situated parallel to the storage system, in such a way that the water flow feed is supplied either by the storage tank or by the auxiliary system. This planning configuration avoids the temperature of the water flow produced by the auxiliary system being conveyed to the storage system.

To reduce losses due to ventilation, a system which employs two heater exchangers is used. In the first exchanger, the external airflow is heated in counterflow of the expulsion airflow from the indoor environment; the second also in counterflow, uses the flow rate exiting from the radiant floor before this returns to the storage tank or auxiliary system. The sizing of the heat exchanger was carried out imposing a global thermal exchange coefficient so that for both 50% efficiency was obtained.

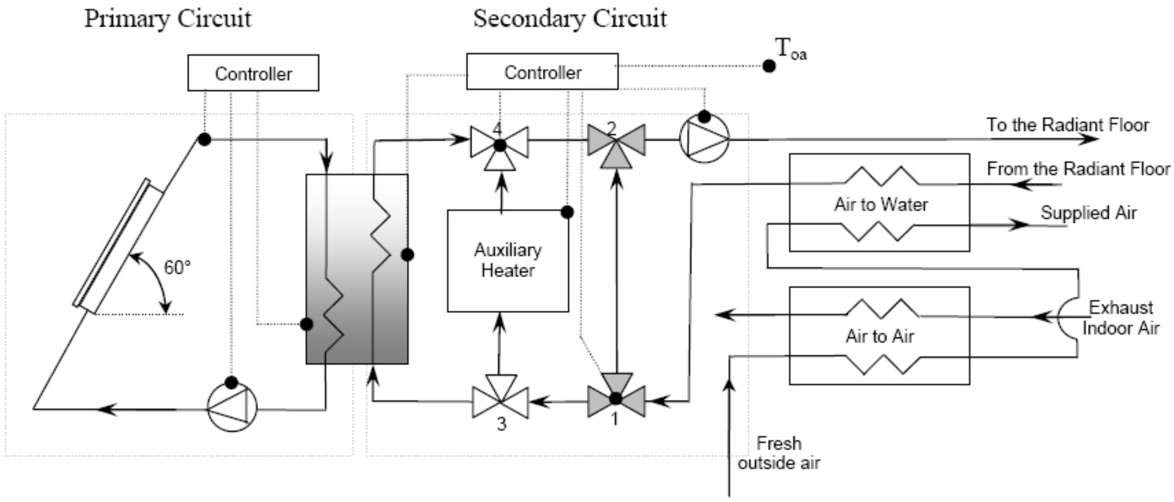

Figure 1. Plan of the solar plant for building heating.

SIMULATION CODES AND CONTROL STRATEGIES

The whole system was studied using the TRNSYS simulation code [7], which consents detailed simulation in a dynamic conditions to evaluate exchange mechanisms relating to the collector surface, storage tank, radiant floor and the indoor volume to be heated.

Moreover, the considered dynamic simulation environment has permitted the study of a adequate control strategy, with the aim of a rational use of the thermal level of the storage tank and the optimisation of the thermal performance of the radiant floor, otherwise penalised by excessive thermal inertia. In order to evaluate the energy performance of the entire system, the building was situated in three different localities in Europe having different climatic characteristic: Helsinki, Copenhagen and Munich.

The hourly values of solar radiation and external air temperature used in the simulations were generated by means of a special TRNSYS subroutine, starting from corresponding average monthly data of the locations considered, in such a way as to generate a metrological year type (TMY) [8]. For the average monthly temperature and horizontal solar radiation values for the considered locations, those contained in the European Solar Radiation Atlas (ESRA) [9] were used. The values of diffuse and beam radiation on the horizontal plane were obtained by Reindl relations [10], which use the solar height angle and index cloudiness. The projection of such components on the field of collectors and on the building walls was carried out using the isotropic sky model [11]. The average monthly values of external air temperature and solar radiation on the horizontal plane during the heating period relating to the three considered locations are listed in Tab. 2.

Table 2. Average monthly daily values of external air temperature (T_{OA}) and of solar radiation on the horizontal plane (HSR) .

		Sep	Oct	Nov	Dec	Jan	Feb	Mar	Apr	May
Helsinki	T_{OA} [°C]	10.0	5.9	-0.1	-4.2	-6.9	-6.1	-2.4	3.6	10.4
	HSR [kWh/m²]	2.16	1.00	0.03	0.12	0.24	0.89	1.96	3.68	5.27
Copenhagen	T_{OA} [°C]	14.0	9.4	5.3	2.5	0.0	0.2	2.1	7.0	11.8
	HSR [kWh/m²]	2.67	1.43	0.66	0.32	0.47	1.08	2.02	3.81	5.07
Munich	T_{OA} [°C]	14.4	9.7	3.3	0.8	-1.0	-0.9	4.2	7.7	12.6
	HSR [kWh/m²]	3.37	2.11	1.14	0.74	0.96	1.79	2.73	3.94	5.03

Given that the function of the solar collectors is prevalently in winter, the inclination angle of the collectors field is 60° for all three locations, to maximise the solar radiation incising on the collector field. The latter are characterised by the following efficiency expression:

$$\eta = 0.7847 - 4.4894 \cdot \frac{T_m - T_{oa}}{G} - 0.027 \cdot G \cdot \left(\frac{T_m - T_{oa}}{G}\right)^2 , \qquad (1)$$

with G (W/m^2) solar radiation, T_{oa} external air temperature, and T_m average water temperature in the collector field.

CONTROL STRATEGIES

In radiant floor heating systems, due to elevated thermal inertia, the type of control which must assure reduced plant response times and avoid thermal discomfort conditions assumes great importance. The methodology used is the control of the inlet temperature combined with a traditional ON/OFF type control, on the inlet flow rate in the case of overheating of the indoor environment (T_{ia}>22°C), due to eventual free energy gains, and its restoration when the temperature goes below a determined limit (T_{ia}<20°C).

The inlet temperature depends principally on the external air temperature, which in the heating period represents the prevalent external constraint on the heated area. Such a control takes into account the effects linked to floor thermal inertia, since the temporal delay with which the floor inlet temperature variation is carried out on the area is comparable with that with which a variation in external air temperature manifests itself within the building by means of dispersing walls. Therefore, no predictive estimate of the feed temperature is necessary, such a criteria is much truer as to the thermal inertia of the floor, and of the same entity of the external dispersing walls. The inlet temperature is that which guarantees an internal air temperature of 20°C in the least favourable conditions, in the absence of solar radiation, and is made to depend linearly on the external air temperature. Imposing the condition that the floor must not transfer thermal power when the external air temperature is 20°C , the law which determines the feed temperature assumes the following expression:

$$t_{INLET} = k(20 - t_{OA}) + 20, \qquad (2)$$

where t_{OA} is the external air temperature while k constant factor depends uniquely on the average global transmittance of the building's dispersing walls. For the structure taken into consideration, the value of variable "k" assumes the value -0.2183.

In order to improve the control function and minimise floor response times, the control relation (2) was further corrected multiplying it by a secondary function, linked this time to the internal air temperature. The logic is based upon the possibility of correcting the inlet temperature obtained from (2) with a minor unit coefficient when the internal air temperature passes a temperature of 20°C, thus making it unnecessary to increase the floor's thermal power, even when the external environment has a relatively harsh temperature, and of using a greater unit coefficient if the internal air temperature is lower than 20°C, in such a way as to render the plant response more rapid when the internal air temperature conditions are unfavourable. The relation used for evaluating C.C. corrective factor in function of the internal air temperature t_{IA} is the following:

$$C.C. = -0.11 \cdot t_{IA} + 3.2 \qquad (3)$$

Using relation (3), the feed temperature results as being 40% higher compared with that necessary determined by (2) if the internal air temperature is around 16°C, it is almost equal to that calculated with (2) if the internal air temperature is around 20°C, while the C.C. corrective factor is equal to about 0.8 if the internal air temperature is near 22°C. Combining relations (2) and (3), the real radiant floor feed temperatures assume the form:

$$t_{INLET} = [k(20 - t_{OA}) + 20] \cdot (-0.11 \cdot t_{IA} + 3.2) \qquad (4)$$

The use of relation (4), determines the raising of the inlet temperature of the radiant floor during the first hours of the day, in which the internal and external air temperatures assume relatively low values, limiting the plant's slow response at the moment of turning on. For such a reason, the implemented control strategy also foresees an attenuation operation at night and at the weekends, to avoid internal air temperatures going down too low. The attenuated operation was rendered necessary also to avoid too high feed temperatures which derive from the use of (4), whose field of application is effectively reduced to the internal air temperature range 16÷22°C. The attenuated operation starts when the internal air temperature goes below 16 °C. In such a case, the radiant floor feed temperature is 25°C, in order to exploit the thermal level of the tank and end at the reaching of 18°C.

RESULTS OBTAINED FROM THE SIMULATION

In Fig. 2, the hourly trends of the internal air temperature, of water in the storage tank, of the requested inlet temperature and of the inlet flow rate to the supply terminals for a period of 48 hours (7^{th} and 8^{th} of February) for Helsinki, are illustrated.

Feed temperature modulation, with values lower than 25°C avoids interruption of the plant feed thus permitting the exploitation of the storage tank even with low temperatures, is to be highlighted. Moreover, in some hours of the day, the inlet requested temperature is lower than the tank water temperature, and in the remaining hours the flow rate is provided entirely by the auxiliary system and the temperature foreseen by (4). Naturally, when the temperature in the tank is greater by less than 1°C compared to that requested, the recycling and extraction flow rate are nil because the airflow feed is prepared entirely by the auxiliary system with the aim of equipping the control system with minimum hysteresis.

In Fig. 3, for the same period, the temperature rise undergone by the ventilation airflow as an effect of heat recovery, is illustrated. The air to air exchanger is always in use, from 8.00a.m. to 6.00p.m., given that the exchange air flow must always be guaranteed during office opening hours. Heat recovery assumes a revealing role: for example at 8.00a.m. on the first day the external air temperature of -7.3°C was raised to +6.3°C. The second heat exchanger is only operative when the radiant floor plant is in use: in this specific case, since the return temperature from the radiant floor is slightly lower than 35°C, the air temperature undergoes an increase from +6.3°C to +17.2°C,. Such a recovery amply reduces load entity induced by ventilation airflow.

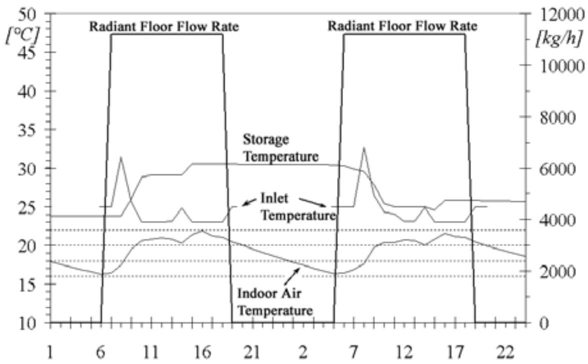

Figure 2. Trends of indoor air temperature, storage tank water, inlet temperature, flow rate to supply terminals (7th and 8th of February, Helsinki)

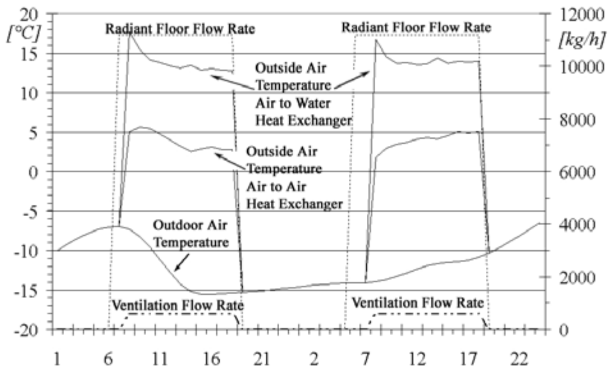

Figure 3. Ventilation flow rate temperature profiles (7th e 8th February, Helsinki).

In Tab. 3, for the three locations examined and for the entire heated period, which goes from September to May, the percentages of thermal energy requirements provided by the radiant floor (F_1), of the air to air exchanger (F_2) and of the air to water exchanger (F_3) are reported. It is possible observe how the percentage of thermal requirement recovered takes on important values, with a greater weight for the air to air exchanger which works continuously for 10 hours per day.

Table 3. Required seasonal energy for heating and seasonal fractions provided by the radiant floor (F_1), from the air to air exchanger (F_2), and the air to water exchanger (F_3).

	Heating Demand	F_1	F_2	F_3
	[GJ]	[%]	[%]	[%]
Helsinki	42.220	44.1	38.2	17.7
Copenhagen	29.053	39.1	44.9	16.0
Munich	27.255	36.4	48.7	14.9

In Fig. 4 seasonal energy provided by the storage tank, by the auxiliary, by the radiant floor, and by the two heat exchangers varying according to the collectors surface for an accumulated volume of 4 m^3 is reported for the city of Munich. It is possible to observe that both the energy provided by the radiant floor and that provided by the two heat exchangers are independent of the collectors surface, while energy provided by the fluid to the floor and to

the air to water exchanger is the sum of the energy provided by the storage tank and the auxiliary. The contribution in terms of energy provided by the tank is a high percentage.
In Fig. 5 trends of the average seasonal temperature of fluid present in the storage tank, and the efficiency of collectors, varying the collectors surface for an accumulated volume of 4 m^3 are reported. It is possible to observe that the average seasonal water tank temperatures increase with an increase in the collectors surface, with higher values for Munich due to a higher insolation level and lower for Helsinki which, in addition to having lower insolation, presents a greater thermal requirement. The average thermal efficiency of the collectors has, as a consequence, a decreasing pattern with the collectors surface, even if it does not alter very much for the three considered locations. Energy provided by the storage tank is reported in Fig. 6. Rising values are encountered with the collectors surface and strongly linked to incising solar energy. The best values are obtained for Munich which has greater insolation.
Finally, in Fig. 7, trends of the requirement fraction provided by the solar source which result as growing with the collectors surface and storage volume, are highlighted. In such a calculation, energy recovered in the air to air exchanger is not considered, in that it is not influenced by the feed system. The greater solar fraction values were obtained in Munich, reaching a value of 0.73 for a 40 m^2 collector surface and a storage volume of 4 m^3. For Copenhagen, the solar fraction varied from a minimum of 0.22 (S=10 m^2, V =1 m^3) to a maximum value of 0.52 (S=40 m^2 e V=3 m^3). Lower values were encountered in Helsinki, due not only to lower insolation, but also to a higher thermal requirement. For such a location the maximum solar fraction value resulted as being equal to 0.36.

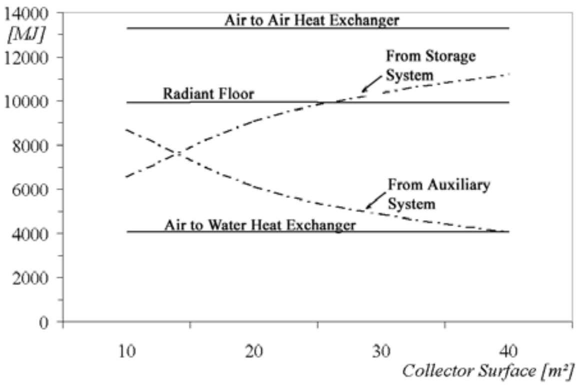

Figure 4. Seasonal energy provided by the storage tank, by the auxiliary, by the radiant floor, and by the two heat exchangers varying according to the collectors surface (Munich V=4 m^3).

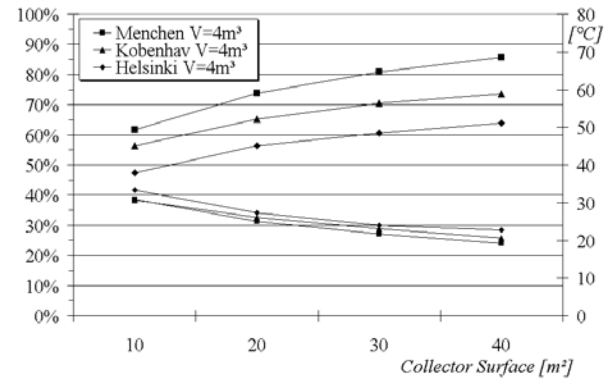

Figure 5. Storage tank average seasonal temperature trend and collectors' average efficiency.

CONCLUSIONS

The thermal performance of radiant floor heating systems supplied by a solar system were identified. The ventilation flow rate to the indoor environment is heated in two successive phases, the first using the air feed expelled from the indoor and the second with the feed exiting the radiant floor. The plants provide heating for areas used as offices in Munich, Copenhagen and Helsinki. This type of plant requires an efficient control system which assures the use of energy available from storage even when the temperature level is low, while thermal comfort conditions within the building are acceptable.
These plant configurations permit, by means of thermal recovery, the obtainment of significant and higher requirement fractions than those provided by the radiant floor.
The results obtained can be summarised as follows:

Winter requirements are delivered to the indoor environment by the radiant floor in the following percentages: 36.4% for Munich, 39.1% for Copenhagen and 44.1% for Helsinki, from the air to air heat exchanger in a percentage including between 38,2 for Helsinki and 48,7 for Munich, and finally from the second air to water exchanger a percentage slightly variable between 14,9% and 17,7%. The requirement fraction provided by the solar source principally depends upon available solar energy, the collectors surface and storage volume.
For collectors surfaces variable from 10 to 40 m^2 and storage volumes between 1 to 4 m^3 the solar fraction resulted as being variable between 0.32 to 0.72 for Munich; from 0.22 to 0.54 for Copenhagen and from 0.13 to 0.34 in Helsinki. Such results demonstrate that obtainable solar contribution with such plants, even in non Mediterranean climates, represents a non negligible fraction of the winter heat requirement.

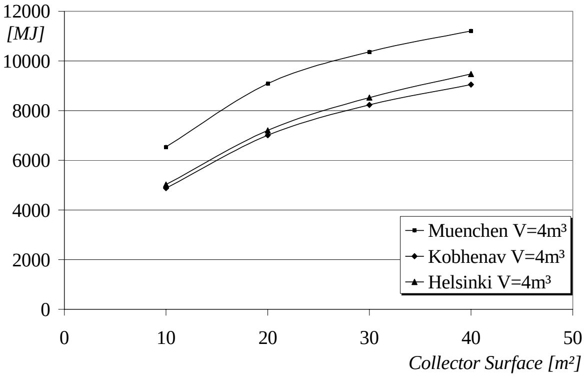

Figure 6. Energy supplied from the storage tank for a volume of 4 m^3 varying the collectors surface.

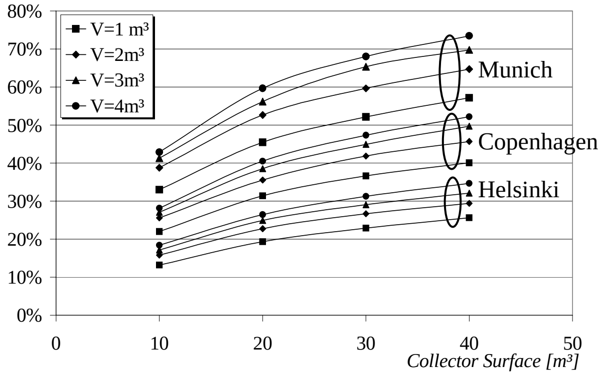

Figure 7. Fraction of the load supplied by the solar source compared to the total.

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Operative temperature control of radiant surface heating and cooling systems

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SUMMARY

The present study investigates the type of temperature sensor that best represents the operative temperature, i.e., the type of sensor that will integrate the influence of air and mean radiant temperature in the same way as a person. Size, shape and colour of the sensor will have an impact on the relative influence of air and mean radiant temperature in a space.
In an experimental chamber different combinations of air temperature and radiant heated or cooled surfaces were tested. Several types of sensor (flat, sphere, ellipsoid, half-sphere, grey, black, white) were used to measure the operative temperature. Besides comparing the type of sensor, the influence of the sensor position in the room was experimentally investigated. The results show that a grey sensor of 3-5 cm diameter is the best size. The best shape, however, depends on the position of the sensor in the space.

1. INTRODUCTION

Several indoor environmental parameters influence the thermal comfort condition for the occupants. In all existing standards for the indoor environment the requirements for the thermal environment and the room temperature are given by using the operative temperature as reference. (ISO EN 7730 [1], EN 15251 [2]).
The present paper analyses how the measured room temperature varies as a function of position and type of temperature sensor. The influence of sensor type and position in the room was studied in a test chamber with different combinations of heated and cooled surfaces.

The room temperature sensor (or thermostat) represents the first element of a heating/cooling system control loop and is therefore significant for the quality of the control. The operative temperature is defined as the uniform temperature of an enclosure where a person would exchange the same amount of heat by radiation plus convection as in the actual non-uniform environment [1]. Thus the sensor should represent the same ratio of heat exchange as the person. The operative temperature can be expressed as:

$$t_o = \frac{\left(h_c \cdot t_a + h_r \cdot t_r\right)}{\left(h_c + h_r\right)} \quad , \qquad (1)$$

where t_o is the operative temperature, t_a is the air temperature, t_r is the mean radiant temperature (°C), h_c is the convective heat exchange coefficient, and h_r is the radiant heat exchange coefficient for a person (W/m^{2}K).
In practice, in most cases where air velocity is small (<0.2 m/s), or where the difference between mean radiant temperature and air temperature is small (<4 °C), the operative temperature can be

calculated with sufficient approximation as the mean value of air and mean radiant temperature [3]:

$$t_o = \frac{(t_a + t_r)}{2}, \qquad (2)$$

This means that air temperature and mean radiant temperature are equally important for the level of thermal comfort in a space and therefore they have the same influence in providing acceptable thermal conditions.
In particular the mean radiant temperature, and consequently the operative temperature, are not evenly distributed and vary according to the location in the room. With radiant surface heating and cooling systems, the mean radiant temperature depends strongly on the surface temperatures of the heated/cooled surfaces, but also on the angle factors between the human body (position of occupant) and the room surfaces, as well as on the emissivity of the surfaces [4].
Therefore if the operative temperature is measured by a sensor, the result will be influenced not only by the position, but also by the shape, size and colour of the sensors.
Among commercially available sensors a grey coloured, ellipsoid-shaped (light grey, 160 mm long, 54 mm in diameter) has angle factors and projected area factors equivalent to the human body [4]. An approximation could be the use of spherical, half spherical and flat plane sensors.
By means of theoretical calculations and experimental testing, this project investigates how representative the temperature measured by the different shaped sensors and in different positions, is of the person's operative temperature in the centre of the room. The paper gives recommendations on positioning the sensor in a room and the type of sensor that is appropriate for radiant surface heating and cooling systems.

2. METHODS AND RESULTS

A theoretical calculation method was used to investigate the influence of sensor size, while an experimental set-up was used to test the influence of a sensor's colour, shape and position.

2.1 Influence of the sensor size

To measure the operative temperature directly, it is necessary to use a sensor with a diameter that will be influenced by the air temperature and mean radiant temperature equally as a person. By transformation of Equation (1) we obtain the following expression of the operative temperature, including the influence of the ratio of radiative/convective heat loss:

$$t_o = \frac{1}{1+\left(\frac{h_r}{h_c}\right)} \cdot t_a + \frac{1}{1+\left(\frac{h_c}{h_r}\right)} \cdot t_r, \qquad (3)$$

$$\left(\frac{h_r}{h_c}\right)_{sensor} = \left(\frac{h_r}{h_c}\right)_{person}, \qquad (4)$$

where h_c for a person was calculated based on ISO EN 7730 [1], Vogt et. al [7] and Mitchell (1974) [8], for a sphere according to ISO EN 7726 [4, 9] and for a plate according to [10]. The relative influence of the air temperature (a – weighting factor [1, 4, 6, 7, 8, 9]) can be calculated as a function of the temperature difference (Δt) by natural convection (see Figure 1) or air velocity (v_a) by forced convection (see Figure 2) and compared for sensors of different dimensions.

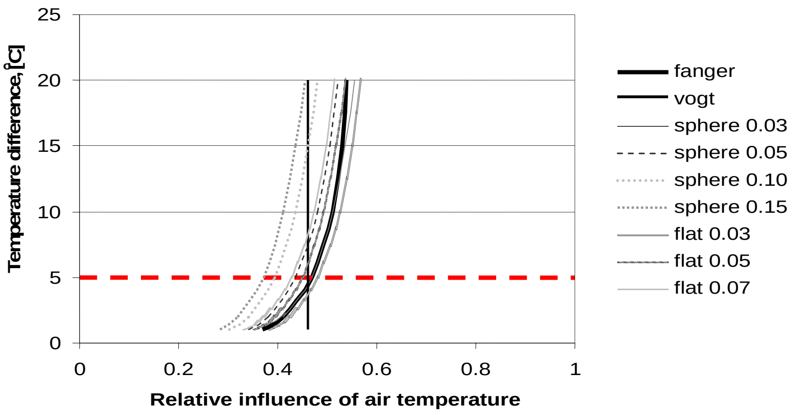

Figure 1. Comparison for natural convection.

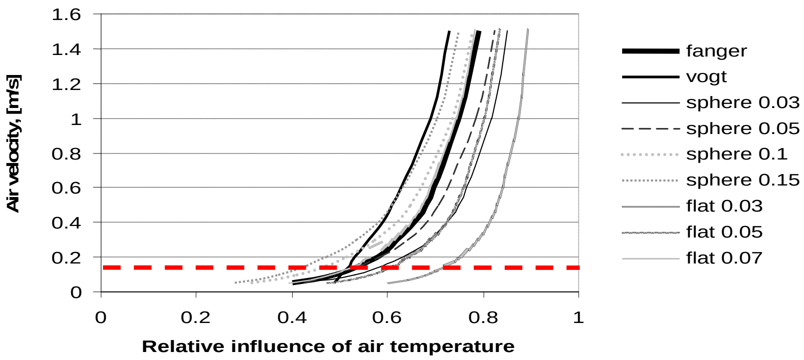

Figure 2. Comparison for forced convection.

The optimal size of the sensor depends on air velocity and Δt, and there is no general optimal size that represents a person's operative temperature. There is also disagreement concerning the convective heat loss for a person [5, 7]. In moderate indoor thermal environments the air velocity is normally lower than 0.15 m/s and the temperature difference between air- and mean radiant temperature is lower than 5 K. For such boundary conditions a diameter between 0.03 and 0.05 m for the sphere, and a length between 0.04 and 0.07 m for the flat shape may suffice (the one used in the present experiment had 3 cm). The lower the air velocity the smaller the required sensor.

2.2 Influence of colour

Three coloured sensors (black, grey, white) and one polished aluminium flat sensor were used in the experiment. The sensors were first exposed to a radiant ceiling (about 42 ^{o}C). The results in Figure 3 show that the colour of a sensor is not important when exposed to long-wave infrared radiation. A polished sensor, however, will reflect more of the radiation and show less influence. In the present test the polished sensor showed a 1 to 2 K lower temperature, depending on the position to the heated ceiling. Room sensors or thermostats may often be exposed to short-wave radiation from direct or diffuse sunlight. This was tested in another experiment. The sensors were exposed to short-wave radiation produced by a high temperature sun-spot lamp. The sensors' direction to the lamp was changed from a perpendicular (position 1 a), 45^{o} angle (position 1b) and a 90 ^{o} angle (position 1c). The results in 4 shows that the colour and finishing of the sensors had a significant influence on the temperature reading. The lowest effect was on a white and a polished sensor; the highest was on the black sensor. The highest differences (approximately 6 K between black and white) appeared when all sensors were placed 3.5 m from the lamp and faced front perpendicularly to the lamp. By turning around the horizontal axis from the perpendicularly facing position (0°) to parallel with the radiation from the lamp (90°), sensors receive less short-wave radiation and the influence decreases. In a real room the angle of the short-wave sun radiation varies during the day. Generally, a grey sensor gives intermediate results between black and white (see Figures 3 and 4), but closer to the black, and the polished sensor gives results very close to the white, especially when the sensors face the light source directly.

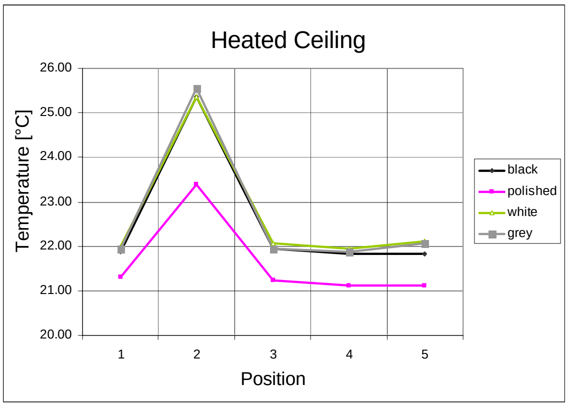

Figure 3. Sample of measured temperatures for flat sensors exposed to heated ceiling

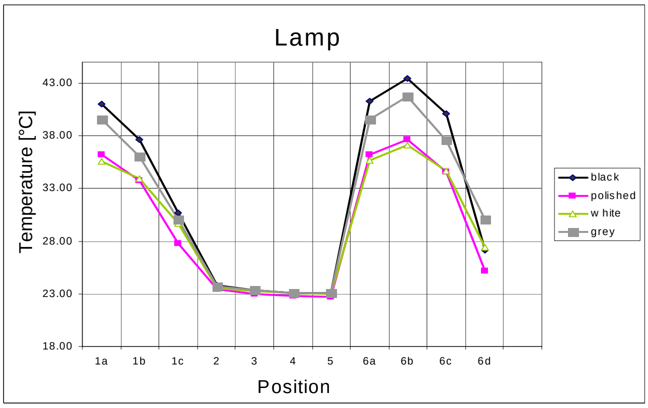

Figure 4. Influence of sensor colour when exposed to sun-lamp at different angles.

2.3 Experimental test of sensor shape and position

The experimental part was completed in a test room (see Figure 5). The tests included experiments with cooled and heated surfaces at different surface temperatures and sensors positioned at different room locations and directions (Figure 6). The experimental room was equipped with a heated/cooled window, heated/cooled ceiling, heated/cooled wall, heated floor, and combinations.

The operative temperatures were measured at 1.1 m height, to simulate the centre of the standing human body. This parameter was estimated by an ellipsoid-shaped sensor (assumed as reference, see Figure 7), a spherical sensor of 4 cm diameter (optimum diameter for use indoors [11] see Figure 8), a half spherical sensor of 4 cm diameter (see Figure 9) and four flat circular sensors of 3 cm diameter. They were made of aluminium plate and painted with black, grey and white colour and one was in polished material. The wall surface temperature was measured with surface temperature sensors.

All sensors were calibrated in a special chamber (with uniform and equal air t_a and wall temperature, placed inside an insulated polystyrene box of 3 cm in five steps (20, 25, 30, 34, 38 °C) and consequently the correction regression lines were developed.

Water-cooled panels (down to 10 °C, sizes of 0.8 m*2 m and 0.8 m*2 m) and electrical heating foils (up to 43 °C sizes of 0.6 m*1.25m and 0.6m*1.75 m) were installed in the test room in order to create significant differences between air- and mean radiant temperature and a high radiant asymmetry. The surface temperatures exceeded the standard values in order to be sure that the results will always be applicable for cases in the comfort range.

The air temperature in the test room was controlled by piston flow through a perforated floor in the range (19-21 °C).

2.4 Test results

The measurement was taken with the different sensors at different room locations at the same height (1.1 m). The results of the measurements with the different sensors are shown in Tables 1 and 2.

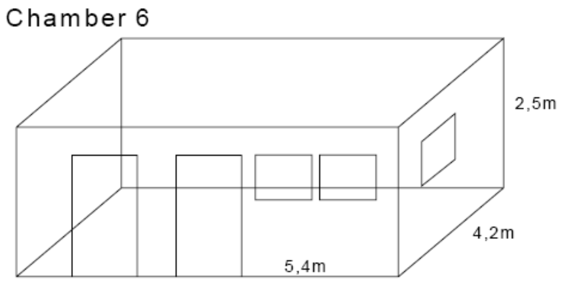

Figure 5. Test room, climate chamber 6 [12]

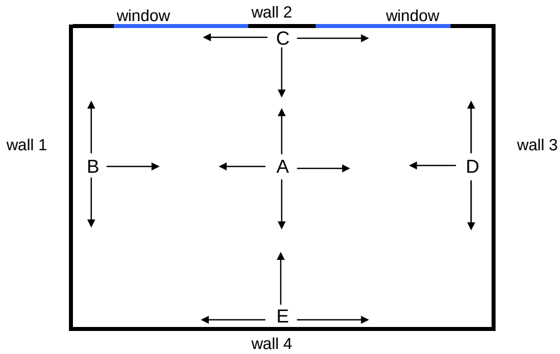

Figure 6. Position and direction of sensors in room test.

Figure 7. Ellipsoid-shaped sensor

Figure 8. Spherical sensor

Figure 9. Half-spherical sensor

Figure 10. Air temperature sensor

The radiant temperature asymmetry is used to describe the non-uniformity of the radiant environment by floor and ceiling heating. The difference between air and mean radiant temperature is also used to describe the non-uniformity. Table 1 shows the results for all sensors in the centre. This comparison is thus influenced only by the type of sensor and not the position. For typical conditions, the floor and the ceiling temperatures were respectively 29.2 °C and 42.4 °C, generating a maximum horizontal radiant temperature asymmetry of 18.8 °C for a heated ceiling. The maximum vertical asymmetry of 8.1 °C was found in the test with a heated wall combined with a cooled wall.
Even if in some cases the radiant temperature asymmetry was quite big the difference between operative and mean radiant temperature were in only two cases higher than 1K (heated ceiling, cooled wall/heated wall).
Table 2 shows the difference between the reference operative temperatures in the centre compared to the measured sensor temperatures in different locations. This is influenced by both the difference in sensor type and position. Table 2 also includes the measured air temperature at the different sensor locations.

Discussion

In terms of thermal comfort, it is preferable to control the room temperature as a function of the operative temperature in the area occupied by people. The positioning of the room temperature sensor in the occupied area may also save energy of 3 to 5% [16] as opposed to positioning the sensor on the wall, and the variations in the room temperature due to the control are smaller.

Other authors have shown the advantage of operative temperature control for maintenance of human comfort compared to air temperature control [13, 14].
In general, a high radiant asymmetry results also in a large difference between air and mean radiant temperature. The results show that using a sensor that is influenced by the radiation gives much better results than just using the air temperature.
Looking at Table 1 for the measurements in the centre, it can be seen that in all cases the sphere will give the best approximation. This difference is in all cases less than 0.5 K except for cases 8 (cooled wall/heated wall) and 2 (cooled windows), where it is respectively 1.7 K and 0.9 K. This is not surprising as the projected area factor for the sphere is closest to the ellipsoid. In the present tests the ellipsoid was in an upright position (standing person), which means a higher influence from the vertical radiation (walls) than from the horizontal radiation (floor-ceiling). This also explains why the greatest difference for the sphere is found for cases 2 and 9.
There is no clear difference between using a half-sphere and a flat sensor. In most cases the difference between the sensors and operative temperature (ellipsoid) were less than the difference between air- and operative temperature.
The results in Table 2 show also in most cases relatively small differences between the different sensors and the reference operative temperature in the centre. However, the sphere in general is not better than the flat and the half-sphere. In most cases the difference between local air temperature and operative temperature in the centre is bigger than the difference between the other sensors and the operative temperature. If positioned close to a heated or cooled wall, the sphere will over-evaluate the influence of the wall. The same is true if the half-sphere or flat sensor faces the heated-cooled surface.

Table 1. Comparison of measured data of all sensors to reference data of ellipsoid in the centre.

	$\Delta(t_{op}-t_{air})$	$\Delta t_{p,max}$			ΔT_{op} (ellipsoid-sensors)			
Set-up	Centre	horizontal	vertical	sensors	A1	A2	A3	A4
Case 1 Cooled Wall-3 Heated Ceiling	1.0	15.3	5.4	sphere	0.1	0.1	0.1	0.1
				flat	-0.2	-0.6	0.9	-0.6
				half sphere	-0.3	-0.3	0.8	-0.3
Case 2 Cooled Window	-1.0	0.7	1.3	sphere	-0.9	-0.9	-0.9	-0.9
				flat	-0.6	-0.1		
				half sphere	-0.7	-0.4		
Case 3 Cooled Window Heated Wall-1	0.7	1.5	5.3	sphere	0.5	0.5	0.5	0.5
				flat	-0.9	1.5	-0.9	-0.7
				half sphere	0.7	1.1	-0.7	0.1
Case 4 Cooled Wall-3	-0.2	0.5	1.9	sphere	-0.2	-0.2	-0.2	-0.2
				flat	-0.3		0.5	-0.3
				half sphere	-0.4		0.4	-0.2
Case 5 Cooled Window Heated Ceiling	0.8	15.6	1.6	sphere	0.1	0.1	0.1	0.1
				flat		0.9	-0.1	-0.5
				half sphere	0.0	0.6	-0.4	-0.3
Case 6 Heated Wall-1	0.9	3.1	5.4	sphere	0.3	0.3	0.3	0.3
				flat	-1.0	0.1		
				half sphere	-0.7	0.4		
Case 7 Heated Ceiling	1.6	**18.8**	0.2	sphere	0.2	0.2	0.2	0.2
				flat		-0.2		
				half sphere		-0.2		
Case 8 Cooled Wall-1 Heated Wall-3	**2.7**	1.0	**8.1**	sphere	1.7	1.7	1.7	1.7
				flat	0.4	2.8	1.6	
				half sphere	0.8	2.1	3.1	
Case 9 Heated floor	0.1	6.1	0.2	sphere	-0.1	-0.1	-0.1	-0.1
				flat	-0.2	-0.2	-0.2	
				half sphere	-0.2	-0.2	-0.2	

The room sensor should be placed either on an interior wall (half-sphere, flat) or preferably in the occupied zone (all wireless, sphere) of the room. Since the effects of the weather (solar radiation) often affect the area nearby windows, and often people are even positioned close to windows in rooms, it is recommended to position the room sensor approximately 1-2 m from a window at a height of 0.6 to 1.1 m since this height represents a sitting/standing person
Since solar radiation often causes an increase in the temperature in a room, it is beneficial if the room sensor (preferably grey coloured) is affected by solar radiation as soon as possible. A further advantage comes with wireless room sensors that can be installed (placed) in accordance with the room furnishings (furniture, shelves, pictures, etc.).

Table 2. Comparison of measured sensor temperature and air temperature in different locations with the reference operative temperature in the centre (ellipsoid).

	$\Delta(t_{op}-t_{air})$	$\Delta t_{p,max}$			ΔT_{op} (ellipsoid in the centre-sensors)					
Set-up	Centre	horizontal	vertical	sensors	B	C	D1	D2	D4	E
Case 1 Cooled Wall-3 Heated Ceiling	1.0	15.3	5.4	air	1.0		1.1	1.1	1.1	1.3
				sphere	0.4		1.1	1.1		0.3
				flat	-0.3		-0.6	0.3		-0.4
				half sphere	-0.1		-0.4	0.9		-0.1
Case 2 Cooled window	-1.0	0.7	1.3	air	-1.0	-0.7				-1.0
				sphere	-0.6	-1.1				-1.2
				flat	-0.8	-0.4				-0.9
				half sphere	-0.1	-0.5				-0.8
Case 3 Cooled Window Heated Wall-1	0.7	1.5	5.3	air	0.7		0.9	0.9	0.9	0.7
				sphere	-0.4		0.7			0.5
				flat	-2.9		0.0			0.5
				half sphere	0.4		0.4			0.2
Case 4 Cooled Wall-3	-0.2	0.5	1.9	air	-0.6		-0.1	-0.1	-0.1	-0.3
				sphere	-0.5		0.2		0.2	0.1
				flat	-0.4		-0.3		0.4	-0.2
				half sphere	-0.5		-0.2		0.2	0.0
Case 5 Cooled Window Heated Ceiling	0.8	15.6	1.6	air	0.9	1.4				1.0
				sphere	0.1	0.7				0.1
				flat	-0.8					-0.2
				half sphere	-0.4	0.6				-0.1
Case 6 Heated Wall-1	0.9	3.1	5.4	air	0.4		1.2	1.2	1.2	0.9
				sphere	-0.7		0.7			-0.4
				flat	0.0		0.3			-0.5
				half sphere	-0.9		0.2			-0.2
Case 7 Heated Ceiling	1.6	18.8	0.2	air	1.9					1.4
				sphere	0.4					0.1
				flat	-0.4					-0.2
				half sphere	-0.2					-0.2
Case 8 Cooled Wall-1 Heated Wall-3	2.7	1.0	8.1	air	1.7		3.1	3.1	3.1	2.5
				sphere	0.2		3.3			1.6
				flat	2.2		2.1			2.2
				half sphere	1.9		2.5			2.3
Case 9 Heated floor	0.1	6.1	0.2	air			0.1	0.1	0.1	0.4
				sphere			0.0			0.1
				flat			-0.2			0.1
				half sphere			-0.2			-0.1

CONCLUSION

The investigations show the advantage of taking into account the thermal radiation in a room, when controlling heating and/or cooling systems according to the operative temperature.

A sensor size (flat, sphere, half-sphere) about 3 to 5 cm will be influenced by radiant and convective heat exchange in a similar way as a person.
For long-wave radiation the colour of the sensor is not important.
For short-wave radiation such as sunshine, a black sensor will overestimate and a white or reflective sensor will underestimate the influence of the radiation. A flat grey or similar colour is recommended.
The influence of the sensor shape depends on the position of the sensor. If the sensor can be positioned in the centre of the room the best shape seems to be a sphere.
As a room thermostat is most often positioned on a wall, the sphere will overestimate the influence of the wall close to the sensor. In this case, a half-sphere is a better shape.
The half-sphere and the flat sensor give in many cases similar results; however, the half-sphere gives values closer to the operative temperature in the centre.
The influence of the sensor shape and position may be up to 2 K in environments where the difference between air and mean radiant temperature is less than 3 K.

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What is the effective thickness of a thermally activated concrete slab?

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SUMMARY

A Thermo-Active-Building-System is a modern sustainable efficient cooling system with pipes installed in the structural concrete slabs of multi-storey buildings. Due to the high thermal mass the accumulation or releasing of heat may occur at different times with respect to the heat load. The issue is how much of the concrete slab will actively be involved in the accumulation of energy. The paper shows the results of dynamic FEM analysis on how deep the room drifting temperatures of 6K penetrate into the concrete structure covered by various floor types (no covering, tiles, wood, acoustic insulation, raised floor). The fluctuations decrease exponentially with the depth while the attenuation ratio and time delay hardly depends on the floor type. Generally, the effective slab thickness is 15 cm, 10cm for wall. The vertical position of pipes should be considered approx. 7,5cm from ceiling surface according to operational mode (heating or cooling) and floor type.

INTRODUCTION

The modern architecture and building design follow the sustainable development by applying of energy efficient building systems. Requirements for insulation of building envelope increase in order to reduce the energy demand in winter however may thus increase cooling needs in summer especially by high internal load. Thermo-Active Building System (TABS) is a cooling system with pipes usually installed in the structural concrete slabs of multi-storey buildings. TABS performs as a heating or cooling elements fully integrated into the main building structure. Due to the high thermal mass the accumulation or releasing of heat may occur at different times with respect to the heat or cooling load (occupancy, sun radiation) [1]. Asynchronous operation of the conditioning system and thermal loads results in energy savings, shifts the fraction of loads to night time and reduces (shaves) the daytime peak load. The simulations [2] already showed how a TABS with 25 cm thick ceiling slab covered by the carpet dynamically behaves. The issue is how much of the concrete slab will actively be involved in the accumulation and release of energy during daily or weekly room temperature and load variations. Therefore it is important to investigate how deep the room drifting temperatures penetrate into the concrete structure, and the paper provides the analysis on this issue using computational method.

METHODS

The paper provides a dynamic software Finite Element Method (FEM) analysis [3] of the concrete structure exposed from the room side by drifting temperatures that follow the sinusoid curve (1) in range from 20 °C to 26 °C:

$$y = 23 + 3 \cdot \sin\left(\frac{2\pi \cdot x}{24}\right), \quad (1)$$

where x is the time [h] and y is the temperature [°C].

The results of the simulation will show the ability of penetration of drifting room temperature in various depths of a concrete structure. Heat transfer was modelled according to following two-dimensional thermal conductance equation:

$$div(a.gradT) = \frac{\partial T}{\partial t}\ , \qquad (2)$$

where dt is time and dT is temperature.

The thermal properties of the concrete structure without floor surface covering and with a thermal insulation was treated by mathematical model represented by heat transfer equation (2) in the investigated domain - the rectangle representative part of the slab 1m x 1m - given as the 2D section and with Newton boundary condition on the surface of a slab. The bulk temperature is considered as a periodic function (1).The simulation was carried out for four commonly used floor structures (Figure 1, Table 1) with different thermal resistance of surface covering ($R_{\lambda,B}$): no floor surface covering (ceramic tiles), with a light carpet of 1 cm thickness, with an acoustic/thermal insulation of 2 cm and 4 cm thickness was analyzed.

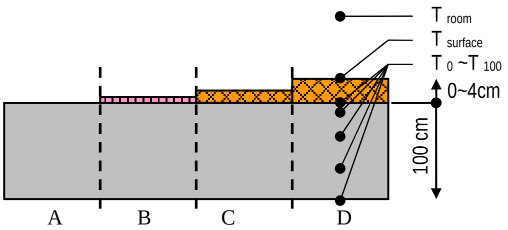


Figure 1. Simulated cases with a concrete and floor structures with reading points

Table 1. Parameters for calculated cases

Calculation case	A	B	C	D
Type of floor structure	No covering	Carpet (1 cm)	Rockwool (2cm)	Rockwool (4cm)
$R_{\lambda,B}$ [m^{2}K/W]	0	0,1	0,5	1,0
ρ [kg/m^3]	-	300	35	35
λ [W/m.K]	-	0,1	0,04	0,04
C [J/kg.K]	-	1000	1200	1200
Case corresponds to	No covering, Ceramic tiles	Light carpet, Wooden (floor) parquets	Floor with an acoustic insulation layer	Raised floor

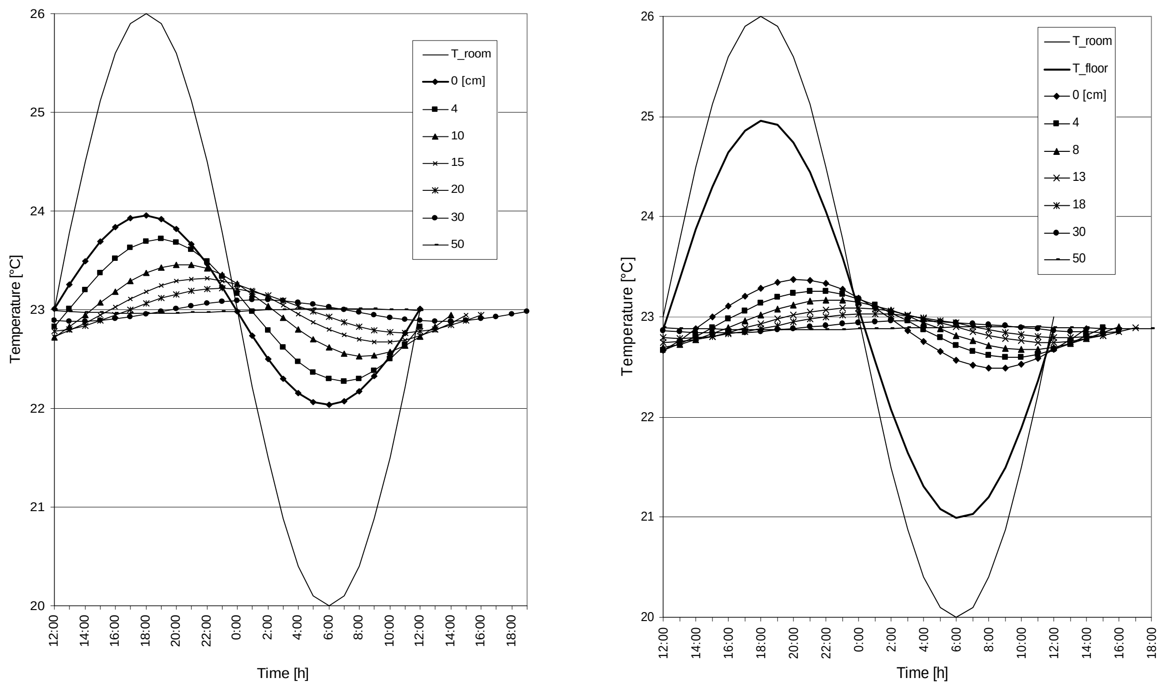
An issue was to select the constant value of total heat transfer coefficient (h_t, [W/m^{2}K]) as the software is not capable of dynamic recalculation of ht value per each time step separately.

Although the h_t may vary during the day, in our simulation h_t changes periodically according to sinusoid periodicity, in range of 6 - 11 depending on the position of the surface (wall/ceiling/floor) and temperature difference between surface and space (if there is heating or cooling).The value h_t of 8 belongs to wall heating and cooling, h_t 11 corresponds to floor heating and ceiling cooling, 7 to floor cooling and 6 to ceiling heating [4].

By periodic drifting room temperatures difference between surface and space is sometimes positive, and occasionally negative. It means that a surface performs partially as a cooling and partially as a heating element related to the room environment temperature. The h_t dynamically changes all the time in range approx. from 6-10. The h_t constant input value was finally chosen to 8.0 W/m^{2}K that covers with sufficient approximation all of available system types and boundary conditions.

RESULTS

The temperature fluctuations decrease exponentially with the depth of the concrete structure (Figure 1), while the attenuation ratio hardly depends from the $R_{\lambda,B}$. It is obvious that the accrued time delays of fluctuations can be observed by all of simulated cases but the most by raised floor structure (Figure 2). The conclusion will follow the assumption that the effective thickness of a concrete slab should be lower than such a concrete layer depth were the 10% of room temperature fluctuations is achieved.



a) b)

Figure 2. Daily temperature fluctuations inside a concrete structure. a) without surface covering, b) with acoustic insulation

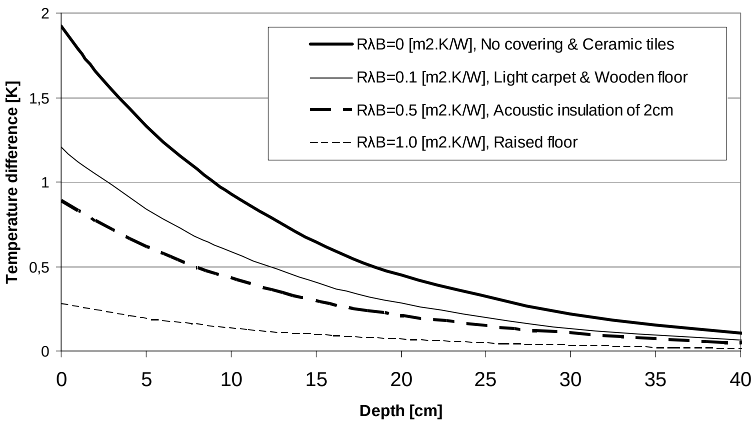


Figure 3. Temperature difference (fluctuation) between daily maximum and minimum temperature in horizontal concrete layers as a function of depth and floor surface covering.

DISCUSSION

However the floor surface coverings may differ in buildings and even from a room to another one. In residential buildings the acoustic insulation should be applied in order to keep the entire privacy. No surface covering floor structure is mostly used in non-residential, storage or industrial spaces. In domestic spaces occupied by persons (apartments and one-family houses) the wooden floor or ceramic tiles are usually used. Although the carpets may perform as a significant indoor pollution source they are still quite popular in some countries, though it might cover the room floor area just partly. The raised floor construction creates in office spaces creates a hollow trapped air layer that is available for leading the electric and IT cables, HVAC water distribution pipes or fresh air ducts, etc. Occasionally in some of the countries are room acoustic issues solved with a small additional acoustic insulation layer on ceiling of $R_{\lambda B}$=0,1. The higher is $R_{\lambda B}$ the lower is the effect of room temperature fluctuations on the mean slab temperature (Figure 1) and thinner part of a concrete layer is available for the heat exchange under the day cycle.

The results of the simulation may be concluded for three significant groups of cases with relatively low $R_{\lambda B}$=(0~0,1; case A B), relatively high $R_{\lambda B}$=(0,5~1,0; case C D) and the case with ceiling slab of thickness more than 150 mm. The ceiling slab with a low $R_{\lambda B}$ should not be of higher thickness then 10~15cm and the hydronic pipes should be located in mean slab level. A thicker slab might dispose higher thermal mass capacity, but it does not really matters, as the slab core would most likely be thermally passive. For cases with a high $R_{\lambda B}$ it is recommended to place the TABS pipes should be positioned closer 7,5~10 cm (or even less) to the ceiling surface in order to be included to heat transfer process following daytime cycle. The active slab thickness is approx. 10~15cm, and in any case should not exceed 20cm.

The height or extended column distance of a building may however sometimes affect sizing of load bearing structure. In this case the ceiling slab may be estimated to even more than 20 cm thickness. As such a slab will mostly not be fully activated it is reasonable to put hydronic pipes closer to the surface with lower $R_{\lambda B}$, usually ceiling surface up to 7,5 cm. Here the application of the raised floor structure comes suitable. This kind of construction is especially favourable by ceiling cooling system as the h_t achieves quite high value of 9-11. Even in case when a small acoustic ceiling insulation is applied.

The effective thickness of a thermally activated concrete slab is recommended to be 15 cm. The vertical position of pipes should be considered according to operational mode (heating or cooling), thermal resistance of floor covering and if a thin acoustic ceiling arrangement is present. The trend of last 10-15 years used to be of a build up the TABS with the slab from 20 to 30 cm thickness [1]. The recent simulations showed that there is no reason to dispose with so high building thermal mass as it is not effective from the point of daytime cycle. Naturally, it might be reasonable for long term heat storage (accumulation). But if for some reason the slab is too thick, the pipes should keep the position recommended above.

The paper describes an idea of an effective slab thickness from the point of view of the potential in all of its structure mass – even in the mean slab level - to be included into the heat exchange inside the room. It means that the whole slab is actively involved in the heat exchange under daily or weekly varying loads. On the other hand the study does not claim that the slab of a recommended thickness (10 - 15cm) is able to store certain amount of heat e.g. sufficient to cover high cooling loads just by using of night operation.

The accuracy of the results might be influenced by at least three facts. The h_t value was selected as a constant value because of the software imperfection; however, in reality h_t varies dynamically all the time. In a real room the fluctuations do not follow the same curve day after day but rather vary in the amplitude and the temperature drifts usually achieve in most of the days up to 4 K. Nevertheless the chosen sinusoid curve approximates the tendency quite well, because an extreme case was chosen for. By following this approach all of results should certainly be in range of those cases achievable under real - also extreme - boundary conditions.

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Control Of Thermally Activated Building Systems

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SUMMARY

A research project on the control and design of thermally activated building systems (TABS) has been started in May 2004. This paper presents one selected result after three years of work: A comprehensive TABS zone control strategy with a modular concept consisting of maximally four parts which allows creating four different control strategies.
The (mandatory) first part of the comprehensive control strategy is an outside air temperature compensated supply water temperature control based on an unknown-but bounded approach to cope with uncertainties and variations in the heat gains during operation.
The (optional) second part of the comprehensive control strategy is an algorithm for room temperature feedback control. With this part, room temperatures are controlled in an energy-efficient way. The algorithm also corrects wrong parameter settings of the other control parts.
The (optional) third part of the comprehensive control strategy is a pulse-width modulation (PWM) module for intermittent operation of the recirculation pump. The intermittent operation is used to save pump energy, but it is also applied to benefit from low-cost energy sources such as free-cooling during the night or electrical energy in low tariff phases.
The (mandatory) fourth part of the comprehensive control strategy is a standard sequence controller to control the zone supply water temperature within the according setpoint range.
In simulations, all four control strategies are applied to TABS and the simulation results are analyzed regarding comfort criteria and energy efficiency.

KEYWORDS: control, thermally activated building systems (TABS), concrete core conditioning, performance bound

INTRODUCTION

Thermally activated building systems (TABS) have emerged as an energy efficient and economical way for cooling and heating of buildings. They integrate the building structure in the overall energy strategy of the building as energy storage. The dynamic thermal behavior of building elements such as structural floors and slabs is exploited to provide either cooling by radiant and convective energy absorption from the space, or space heating by the release of stored energy. In contrast to radiant cooling by suspended ceiling panels, peaks in energy demand are flattened and the actual cooling is shifted in to the colder night time [1], [2].
So far, control is implemented downstream in the design process. The specification of control algorithms is difficult because of the large thermal inertia of the system and because of the challenge to comply with comfort requirements in different rooms with different heat gains within the same hydraulic zone. Various control approaches are presented (see [3]-[9]), but they often have disadvantages due to different approaches for cooling and heating, too frequent switching between heating and cooling, the need for manual switching between heating and cooling mode as well as the need for manual adjustment of parameters. In [10],

[11], an unknown-but-bounded approach is presented to cope with uncertainties and variations in the heat gains during the design phase. In [12], [13], an integrated design process for TABS and its control is given based on the unknown-but-bounded approach.
This paper presents possible additions to the base control concept described in [12] which leads to a modular control concept. This concept can be adapted to different TABS plants and thereby increase the energy efficiency, the comfort and/or the commissioning and operation effort. The concept has been developed in the frame of a research project on which was reported the first time in [10].

OVERALL CONTROL CONCEPT

The presented control concept is one part of an integrated process that additionally consists of TABS design (planning of HVAC system and its control), commissioning of TABS and optimization during operation. In the TABS design, the base TABS zone control strategy is used (mandatory parts, see [12]). This base control strategy can be extended by optional control parts to improve comfort and/or energy efficiency depending on given requirements and installations. The result is a modular control concept for a TABS zone which – by configuration – can be adapted to different TABS plants. Besides the control of the zones, also the heat and cold generation and the distribution have to be controlled. In this paper, only the TABS zone control is described in detail.
In Figure 1, the overall control concept is shown for a TABS plant with two zones. Two zone controllers (see next chapter) act on the zone circulation pumps and the zone heating and cooling valves. Together with the zone supply water temperature setpoints $\vartheta_{sw,SpH,PWM}$ and $\vartheta_{sw,SpC,PWM}$, these signals can also be used to set up a demand dependent control of the heat and cold generation and distribution as indicated in Figure 1.

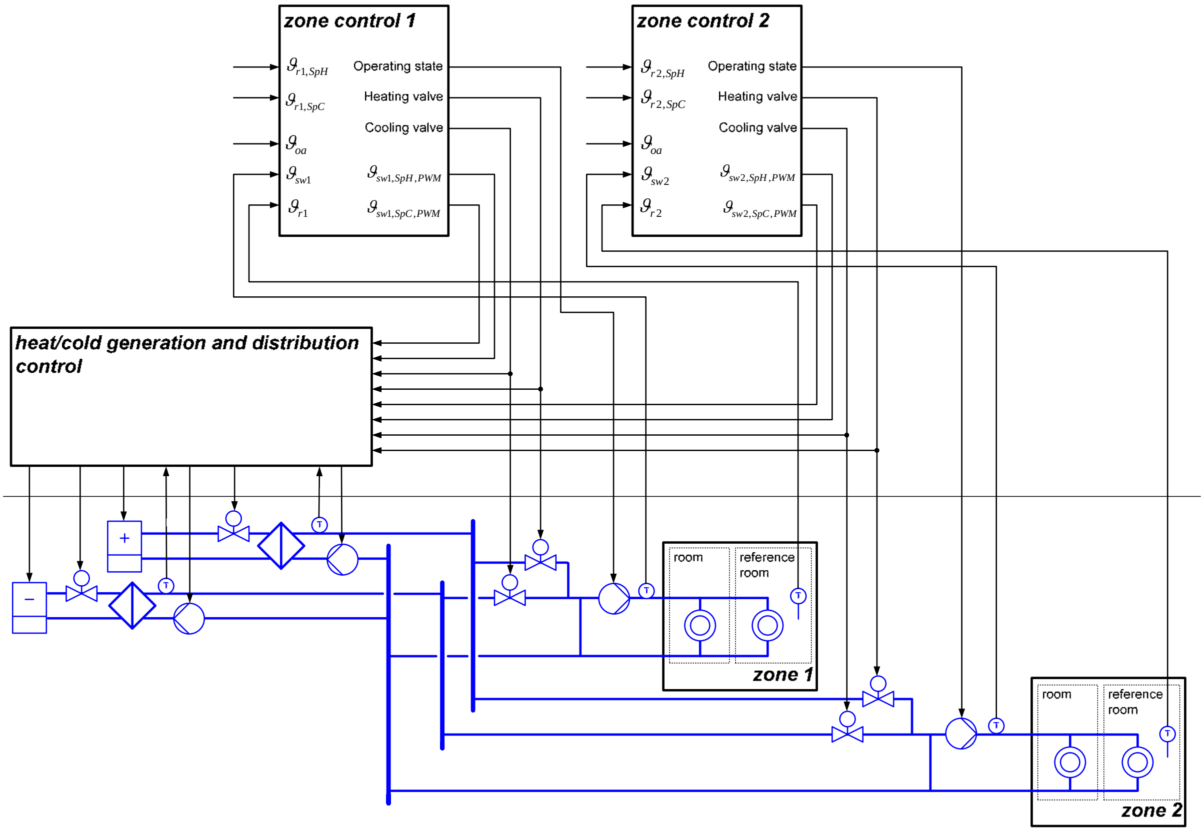

Figure 1: The overall TABS control concept, including heat/cold generation and distribution

ZONE CONTROL

In Figure 2, the modular TABS zone control is given in more detail. If an optional part is not needed or required, only the feed-through indicated by dotted arrows in the corresponding control part function block has to be realized. The functionalities of the four control parts are explained below.

1. Outside temperature compensated supply water temperature control (mandatory)

This core control part determines a supply water temperature setpoint range [$\vartheta_{sw,SpH}$, $\vartheta_{sw,SpC}$] as a function of the mean outside air temperature of the last 24 hours $\overline{\vartheta}_{oa}$ and the current room temperature setpoint range [$\vartheta_{r,SpH,FB}$, $\vartheta_{r,SpC,FB}$]. For that purpose, the so called heating and cooling curves are used. The heating and cooling limits serve to identify the operating mode depending on $\overline{\vartheta}_{oa}$ (see e.g. Figure 2). In order to define initial parameters of this control part, an unknown-but-bounded design process can be used [12].

2. Room temperature feedback control (optional)

If one or several room temperatures ϑ_r are measured in the controlled zone, a room temperature feedback control part can be added to the zone control. Room temperature feedback control has the following advantages (+) and disadvantages (–):

+ Comfort can be improved if heating and cooling curves are wrong placed or if the intermittent zone pump operation is inaccurate due to modeling errors (see part 3).
+ Energy efficiency can be increased when room temperatures are controlled making full use of the room temperature comfort range. This also leads to less frequent switching between heating and cooling demand of the zone.
+ Commissioning and tuning effort can be reduced since the feedback control corrects settings that are wrong (to a certain extent).
– Since TABS react slowly, only a day-to-day compensation is promising, an instant correction can not be achieved through the TABS.
– The placement of the room temperature sensor is critical: The measured temperature should be meaningful during the whole operation.
– If the zone consists of more than one room, the measured temperature has to be a reference temperature for the whole zone or several sensors in different rooms are used.

Since TABS react slowly, only day-to-day room temperature compensation is promising, an instant correction can not be achieved with TABS. A possible implementation of room temperature feedback control by two standard PID controllers is outlined here: The task of controller 1 is to keep the room temperature ϑ_r above the lower room temperature setpoint $\vartheta_{r,SpH}$ and the task of controller 2 is to keep the room temperature below the upper room

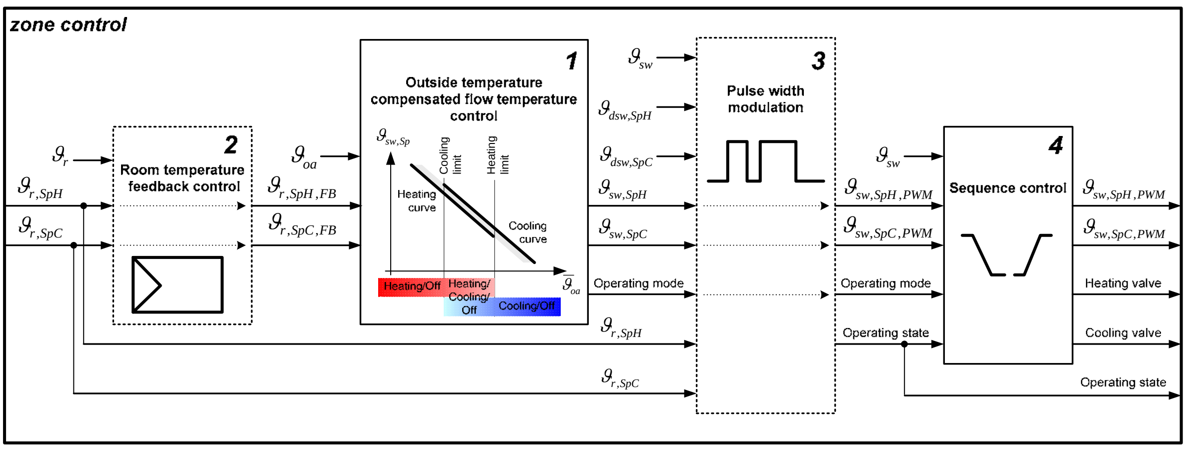

Figure 2: Scheme of the modular zone control (mandatory parts with solid frame, optional parts with dotted frame)

temperature setpoint $\vartheta_{r,SpC}$. The controlled variable for the P- and I-part of the controllers are the minimal (for controller 1) and maximal (for controller 2) room temperature of the last 24 hours. The controlled variable for the D-part of the controllers is the average room temperature of the last 24 hours. The controller outputs are limited (e.g. [-5K, 5K]) and added to $\vartheta_{r,SpH}$ and $\vartheta_{r,SpC}$ so that a corrected room temperature setpoint range [$\vartheta_{r,SpH,FB}$, $\vartheta_{r,SpC,FB}$] results. Controller 1 operates with $\vartheta_{r,SpH}$ while the setpoint of controller 2 is modified to cope with exceptional days when less internal heat gains are present (e.g. weekend).

3. Intermittent operation of zone water circulation pump (optional)

The control parts presented so far are based on continuous operation of the zone water circulation pumps. An additional third control part (see Figure 2) can be added to operate the zone water circulation pump intermittently which has the following advantages (+) and disadvantages (–):

+ Energy efficiency can be increased by reducing the operation time of the zone circulation pumps.
+ Energy efficiency can be increased by moving zone circulation pump on-times to time intervals where heat and cold can be delivered efficiently, e.g. free-cooling during nights.
+ Energy costs can be reduced by moving the on-time of the zone circulation pumps to time intervals where energy can be delivered at lower cost, e.g. using low tariff of the electrical energy to operate chillers and heat pumps.
+ Investment costs can be reduced by smaller dimensioning of the chiller when using it during the day for the air conditioning system and during the night for the TABS.
− The achievable comfort level generally decreases with intermittent operation of the zone circulation pump. During off-times, no heat is exchanged to the water circuit. This has to be compensated by lower or higher supply water temperatures during on-times, respectively, which reduces the self-regulating effect.
− The control of zone supply water temperatures for intermittent operation is more critical since turning the zone pumps on and off is heavily disturbing the control system.
− Intermittent operation can lead to more switching between heating and cooling demand than necessary.

A possible implementation of the intermittent operation control part is presented here: A pulse width modulation (PWM) approach is used to determine pump on- and off-times depending on the current operating mode. The pump operating mode (on or off) and the according supply water temperature setpoint range [$\vartheta_{sw,SpH,PWM}$, $\vartheta_{sw,SpC,PWM}$] are calculated based on a physical TABS model (see Figure 3). The same model then can be used to shorten or prolong pump on-times whenever the supply water temperature ϑ_{sw} is not kept within its setpoint range due to control errors or limitations of the heating/cooling system. Before starting an active heating or cooling process, an idle running operation is applied (the water circulates in the zone with zone valves closed). If after a specified idle operation time, the measured supply water temperature is within the range [$\vartheta_{sw,SpH}$, $\vartheta_{sw,SpC}$], the model based calculation is redone so that – whenever possible – no active heating or cooling process is necessary. To increase energy efficiency (or to reduce energy costs), on-time intervals are placed where heat or cold is expected to be delivered more efficiently (or less expensive).

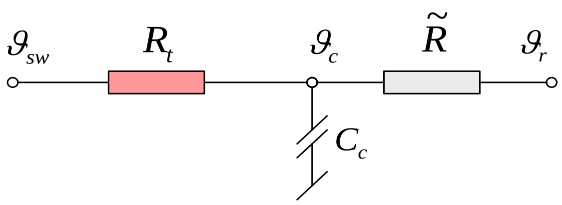

Figure 3: TABS first order model used for control of intermittent operation of zone water circulation pump

The values of the fictitious TABS thermal resistance R_t (between ϑ_{sw} and the concrete core temperature ϑ_c of the concrete core thermal capacity C_c) and the thermal resistance $\tilde{R}$ (between ϑ_c and ϑ_r) of the model in Figure 3 can be calculated according to [12]. The model then allows calculating turn-on ratios between on-time Δt_1 and the period length Δt for given supply water temperature setpoints. The simplified result for the turn-on ratio in the heating cases is given in (1), a similar equation is found for the cooling case. To calculate a non-simplified result, a calculation procedure as described in [15] can be applied.

$$\frac{\Delta t_1}{\Delta t} = \frac{1}{1 + \dfrac{\vartheta_{sw,SpH,PWM} - \vartheta_{sw,SpH}}{\dfrac{R_t}{R_t + \tilde{R}}\left(\vartheta_{sw,SpH} - \vartheta_{r,SpH}\right)}} \tag{1}$$

A simplified result for the turn-on ratio in the heating case for varying supply water temperatures is given in (2). This equation is used to shorten or prolong pump on-times whenever the supply water temperature is not controlled within its setpoint range.

$$\frac{\Delta t_1}{\Delta t} = \frac{\dfrac{1}{\Delta t}\displaystyle\int_{t=t_0}^{t_0+\Delta t_1} \vartheta_{sw}(t)dt - \dfrac{R_t}{R_t + \tilde{R}} \cdot \left(\vartheta_{sw,SpH} - \vartheta_{r,SpH}\right)}{\vartheta_{sw,SpH} - \dfrac{R_t}{R_t + \tilde{R}} \cdot \left(\vartheta_{sw,SpH} - \vartheta_{r,SpH}\right)} \tag{2}$$

4. Sequence control of zone supply water temperature (mandatory)
A standard sequence controller that controls the supply water temperature ϑ_{sw} (the according setpoint range is [$\vartheta_{sw,SpH,PWM}$, $\vartheta_{sw,SpC,PWM}$]) is acting on the heating and cooling valves. Depending on operating mode and operating state one or both sequences are disabled and the respective valve is closed.

RESULTS: CASE STUDY WITH DIFFERENT ZONE CONTROL TYPES

A simulation case study was made to compare four different TABS zone control strategies:
- control strategy 1 includes control parts 1, 4
- control strategy 2 includes control parts 1, 2, 4
- control strategy 3 includes control parts 1, 3, 4
- control strategy 4 includes control parts 1, 2, 3, 4

The simulated zone only contains one room; key figures of the room and the TABS configuration are given in Table 1 and Table 2 (see reference building in [12]).
Whole-year simulations for all four control strategies were carried out in a Matlab/Simulink simulation environment. Outside air temperature and solar heat gains were derived from local meteorological data for Zurich/Switzerland. The used individual heat gains of persons, equipment and lighting are based on the Swiss standard [14]. Room temperature setpoints $\vartheta_{r,SpH}$ = 21 °C and $\vartheta_{r,SpC}$ = 26 °C were used for the whole year. Other main control parameter values were determined in an unknown-but-bounded design process for the defined zone as described in [12], [13]. Here two major heat gain situations were differentiated: workdays and weekends; for both situations dynamic lower and upper heat gains were specified. This ultimately leads to the heating and cooling curve parameter values for weekdays and weekends. The design process also produces parameter values for the heating limit (13.1 °C) and the cooling limit (-1.2 °C).
In Figure 4, an example of the TABS zone control is shown for control strategy 4 for one week in summer: In the top diagram, it can be seen that $\vartheta_{r,SpC}$ is increased by control part 2

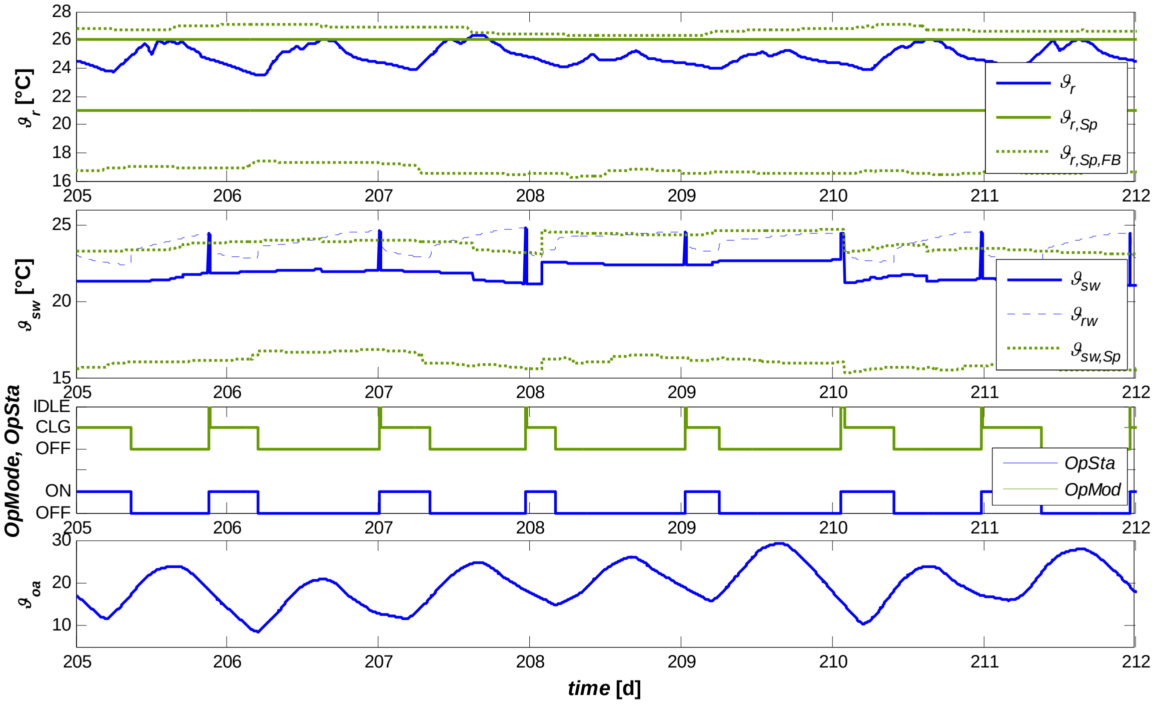

Figure 4: Example of zone control for control strategy 4 in summer

due to heat gains lower than the specified upper bound. The upper supply water temperature setpoint is then calculated by control part 1. Control part 3 lowers this setpoint and reduces pump on-time. The turn-on times are moved within the cooler nighttime. At the beginning of each cooling period, the zone is operated in idle operation to check if cooling is necessary. The resulting room temperatures for all control strategies are given in Figure 5. Figure 6 shows the heating and cooling energy demand, zone water circulation pump turn-on times as well as hours where the needed supply water temperature was lower than the outside air temperature. Figure 6 also contains performance bound energy demands which are calculated by optimal control with full knowledge of the controlled model and actual and future inputs.

Table 1. data of the simulated room

Space length, width, height		6 m x 6 m x 3 m
Façade	Orientation	West
	Area	18.0 m^2
	Overall U-Value	0.65 W/m^{2}K
	Glazing fraction façade	42 %
Additional internal Wall (light)		36 m^2
Natural air change		0.1 h^{-1}
Ventilation according to indoor air quality requirements (no cooling/heating by ventilation assumed)		

Table 2. data of the simulated TABS configuration

TABS covering fraction (floor area)	80 %
Thickness concrete slab	250 mm
Pipe spacing	200 mm
External/internal pipe diameter	20/15 mm
Specific supply water mass flow rate [a)]	15 kg/(h m^2)
Fictitious TABS thermal resistance (R_t) [a)]	0.08 (m^2 K)/W
Thermal resistance of flooring (carpet)	0.125 (m^2 K)/W

[a)] in terms of floor area covered by tabs

DISCUSSION

Since the parameter values of the control strategies are based on the exactly known simulated zone and since the boundaries of the heat gains have been specified reasonably, the open-loop room temperature control of strategy 1 is almost ideally in terms of comfort (see Figure 5). An added room temperature feedback control (strategy 2) is reducing the energy demand a little, but also – in a tolerable amount – the comfort. In this scenario, the main advantage of the feedback control is not apparent.

Application of PWM control (strategy 3) results in less pump turn-on times (47 %, see Figure 6), but increased heating and cooling energy demand. That is because the self-regulating effect is lower compared to a continuous operation (where supply water temperatures are closer to the room temperature), and therefore the control tends to compensate more than necessary (room temperatures stay within the comfort range for the whole year). Control strategy 4 with PWM and room temperature feedback control is able to avoid this: heating and cooling energy demand is similar to strategy 2 with similar comfort. Here, the PWM is done in a way so that during summer the on-times are moved to the colder night time in order to benefit more from free cooling (i.e. use of a dry cooling tower). To illustrate the resulting control behavior, Figure 6 shows the numbers of hours for which the outside air temperature was larger than the supply water temperature. During that time, the cooling demand cannot be satisfied by a dry cooling tower alone.

The presented control concept is one part of an integrated process consisting of TABS design (planning of HVAC system and its control), commissioning of TABS and optimization during operation. A modular control concept is used to handle different requirements and installations in a simple manner. Further details of the method can be found in the design and commissioning handbook which will be published in the frame of our work.

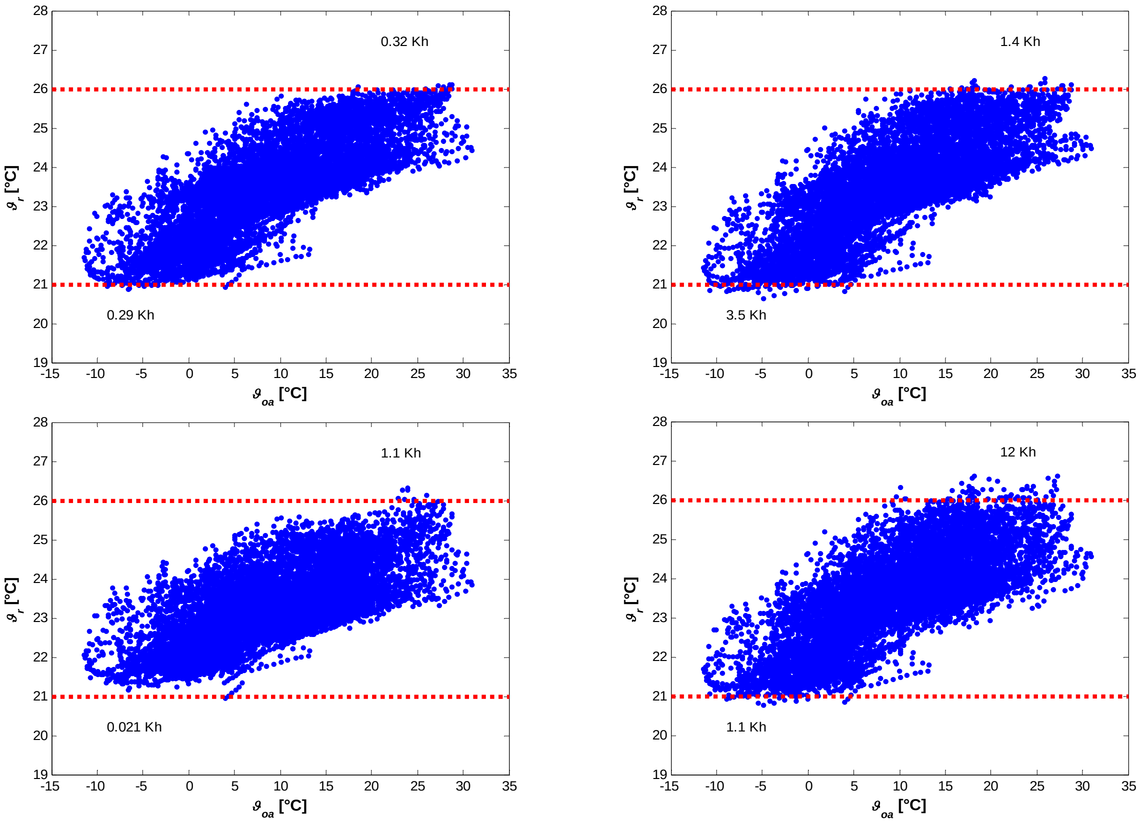

Figure 5: Whole-year simulation room temperatures (hourly values) for control strategies 1 (above left), 2 (above right), 3 (below left) and 4 (below right), room temperature setpoint violations are given within the figures

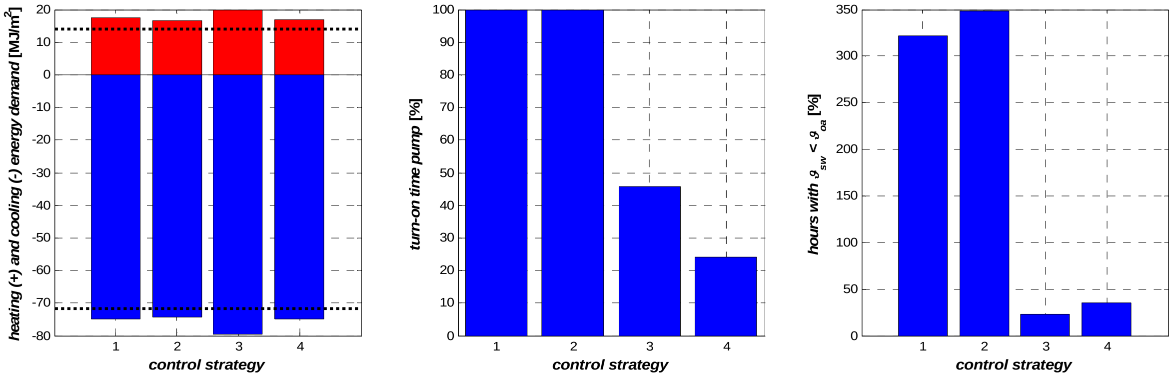

Figure 6: Whole-year simulation heating and cooling energy demand (left) with performance bound (dotted lines), pump operation time (middle) and hours where the supply water temperature was lower than the outside temperature (right)

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D05
Heating and piping systems

Sizing of boilers for residential buildings

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SUMMARY

The tendency towards better insulated and energy friendly ventilated buildings leads to an ever decreasing building heat load. On the other hand, the installed boiler size is not altered. Often this is due to requirements of the coupled domestic hot water production, or worse, due to the superstitious belief that larger means better.
Correct sizing of the boiler can not only reduce the heating system investment cost, but also lower the energy consumption, and maybe even improve the thermal comfort.
For this study, a simulation model has been set up in which the boiler's output capacity is varied and the resulting heating system evaluated for both thermal comfort and energy consumption. The simulations are performed for two natural gas boilers, i.e. a high efficiency and a modulating, condensing one. The performance of the boilers sized following the European standard design heat load calculation [1], is compared to the results of the broad range of thermal output capacities.
As expected, oversizing leads to increasing primary energy consumption, especially in the case of the High Efficiency boiler due to its inadequate control system and high skin losses. The heat load calculation from the European standard is not far from the optimum, but it overestimates the necessary reheating power after night set back.

INTRODUCTION

As the thermal resistance of a building enclosure increases, the impact of any additional thermal insulation on energy economy decreases. This physical law forces countries with tough insulation requirements to turn to Energy Performance Regulation (EPR). EPR not only evaluates thermal insulation, but also takes into account the energy efficiency of ventilation, lighting, hot water production and heating systems as well as the benefits of passive and active solar energy [2]. A great unknown in such a vast corpus of possibilities is the heating installation's efficiency. The efficiency should be determined on the building scale, so that interactions between heating system and building are included. By defining the overall efficiency as the ratio of the net heat demand of the building and the total energy consumption of the boiler, control losses are included, but also the partly recovering of distribution and boiler losses can be assessed.
Since the net heat demand is a theoretical consumption minimum, and can not be measured like the real energy consumption, TRNSYS 16.1 is used to simulate different types of boilers, different boiler controls as well as different emission controls in houses with varying insulation quality [3]. It is shown that the boiler size has an important influence on the efficiency of the heating system.

The estimation of the required thermal power is not evident. Standard calculation methods have been developed. However, as Corrado [4] and Wouters [5] state, standardization, and as a consequence simplification, of the heat load calculation must be done carefully. In this paper the boiler size is assessed to maximize the heating system efficiency while maintaining the indoor thermal comfort. This result is then compared to the boiler size as determined by the European standard procedure [1].

METHODS

The building is a compact terraced house with an outer volume of 446 m³ and an exterior surface of 226 m². The heated ground floor (level 0) has an area of 63 m². The day zone, which comprises the kitchen, hall and living room, has a floor area of 50 m². Its set temperature during the day is 21°C, the implemented night set back allows a temperature decrease down to 15°C. The first floor (level 1) includes a hall (14.3 m²), two bed rooms (16.3 m² and 19.3 m²) and the bathroom (6.6 m²). This last zone has a set temperature of 24°C during the day and 21°C during the night. The bed rooms are not heated. The upper floor (level 2), partly under the sloped roof, comprises a study (14.1 m²), a storage room (4.8 m²), the third bedroom (21.5 m²) and the upper part of the hall (16 m²). The study and hall are both heated as the day zone, while the other rooms on this floor are not heated. The boiler, however, is located in the storage room and thus indirectly heats this volume.
The average U-value of the building envelope is 0.8 W/m²K, which leads to a net annual energy demand for space heating equal to 11 573 kWh. The heat emission system consists of low temperature radiators and thermostatic radiator valves (TRVs). The ventilation is modelled as non-forced natural ventilation as described in the Belgian standard NBN D50-001 [6].
The building as described above is modelled in TRNSYS 16.1. A detailed description is available in [7]. The low temperature radiators are modelled on Type 72 from IEA Annex 10 and numerically optimized by Kummert [8]. They are dimensioned according to the technical reports of the Belgian Building Research Institute [9]. Each radiator is equipped with TRVs. In the framework of IEA Annex 10, a simplified TRV-model has been developed by Ast [10]. It is a lumped capacitance model of the temperature sensor including the (relatively small) thermal resistance of the casing and the (larger) thermal resistance between sensor and water. Further refinements and measured data have allowed to develop a representative model of a programmable TRV in TRNSYS.
The boiler models for both the condensing and the high efficiency gas boiler are elaborate refinements of the TESS-boiler model ([11] and [12]). The input data is based on test data and a static boiler EES model developed by Lebrun [13]. The TRNSYS boiler models can be regarded as a combination of an ideal heater and a lumped capacitance model. The ideal heater, the burner, is immediately turned on or off as requested by the boiler control logic. The heated water then ‘passes’ the capacitive element. This part also models the heat losses through the boiler’s envelope. The boiler’s water exhaust temperature, leaving the capacitance, is thus lower than the burner’s water outlet temperature.
The boiler’s control logic can be set to an on/off regime or to a modulating regime so that the Part Load Ratio (PLR) can vary between 25% and 100%. The high efficiency boiler is modelled as a non-modulating system. It is turned on if the TRVs allow water flow. If the water exhaust temperature exceeds the maximum water outlet

temperature, the boiler is turned off again. Obviously, this working regime leads to frequent cycling. The condensing boiler can modulate to a quarter of its maximum thermal power. The latter is equipped with an external temperature compensated control. If the requested PLR becomes lower than 25%, this boiler will have to cycle as well.
The boiler is switched off during night set back. However, during day time, the boiler is required to maintain its operation temperature, since it is coupled to TRVs which can always demand heat.
The simulations are performed with varying boiler sizes. Firstly the thermal power as calculated based on the European standard is determined for four reheat time periods, i.e. the heating system is turned on one, two, three or four hours before the set temperature has to be reached. This leads to four surplus powers for intermittent heating (Table 1) and thus four boiler sizes as shown in Table 2. Secondly, a range of power levels from 3 kW to 30 kW is simulated. Auxiliary power, thermal capacity, as well as the thermal transmittance of the boiler envelope varies proportionally. The reheat time is set to one hour for this range.
The European standard EN 12831 [1] specifies a method for calculation of the design heat load of either a heated space or an entire building. The focus in this paper is on dwellings, heated with a central heating system. The European standard calculation for the total building design heat load ϕ_{HL} is[1]:

$$\phi_{HL} = \sum \phi_{T,i} + \sum \phi_{V,i} + \sum \phi_{RH,i} \quad (1)$$

where $\phi_{T,i}$ indicates the transmission heat losses of the heated spaces, excluding the heat transferred towards other zones inside the building, $\phi_{V,i}$ indicates the ventilation losses of the heated spaces excluding heat transferred in the building and $\phi_{RH,i}$ indicates the reheat capacities of the heated spaces required to compensate for the effects of intermittent heating.
The building is considered as a whole, therefore, internal heat transmissions are not taken into account. The EN 12831-standard aggregates the transmission losses through this building envelope to be 4.2 kW. The ventilation losses are calculated based on the volume flow rates for natural non-forced ventilation [6]. This results in a design heat loss for ventilation equal to 3.0 kW. For the surplus due to intermittent heating, a cold winter day is considered: the simulated temperature drop during night set back amounts to 3°C, averaged over the building zones. The thermal power to compensate for intermittent heating, as a function of the reheat time for a 3°C temperature drop is given in Table 1.

Table 1: Surplus in thermal power due to intermittent heating after a 3°C temperature drop

Reheat time (h)	1	2	3	4
Surplus for intermittent heating (kW)	3.8	1.9	1.4	1.1

Summing up the thermal loads due to transmission, ventilation and intermittent heating, results in the design heat load. To determine the necessary thermal power of the boiler, the emission and control efficiency have to be taken into account [14]. That means a division by a factor 0.97. A safety margin is applied to calculate the thermal

[1] The simplified calculation method can be used as the restriction on the air exchange flow rate, n_{50} being lower than $3h^{-1}$, is fulfilled.

power of the boiler, i.e. the power is multiplied by 1.1. Table 2 shows the resulting thermal power levels.

Table 2: Total thermal power as a function of reheat time, according to the European standard [6].

Reheat time (h)	1	2	3	4
Thermal power to install(kW)	12.6	10.4	9.8	9.5

The thermal comfort is determined based on an adapted weighted temperature excess method. This method is developed to assess overheating and is based on the Predicted Mean Vote (PMV)-model as developed by Fanger [15]. The aim of this paper, however, is to evaluate the thermal comfort for varying boiler sizes. Therefore only 'underheating' periods during the heating season will be included, i.e. when the indoor temperature does not reach the set value of the heated room during day time. The Weighted Temperature Subceeding hours (WTS) method is defined as follows:

$$PMV_i > -0.5 \rightarrow WF_i = 0 \qquad (2)$$

$$PMV_i \leq -0.5 \rightarrow WF_i = \frac{\sum PPD_i}{10.16} \qquad (3)$$

Where i indicates the zone and WF the hourly weighting factor. When the PMV is higher than -0.5, or put otherwise, when is predicted that less than 10% of the people judge the room too cold, the weighting factor equals zero. In the other case, when more than 10% thinks it is too cold, the factor is directly proportional to the Predicted Percentage of Dissatisfied (PPD). Finally, the WTS-hours are calculated as the sum of these hourly weighting factors during the whole heating period.

The energy consumption of the whole heating installation will be assessed. The circulation pump's working regime is determined by the boiler's working load; therefore, pump's energy consumption should be included. The energy input of the heating installation consists of natural gas (gas_{cons}) and electricity ($elec_{cons}$). As the energy consumption is expressed in primary energy (PE_{cons}), a conversion factor for electricity (f_{conv}=2.5 [16]) is used:

$$PE_{cons} = gas_{cons} + f_{conv} \cdot elec_{cons} \qquad (4)$$

The overall efficiency is then defined as the ratio of the net energy demand of the building and the total primary energy consumption. The boiler efficiency can be defined as the ratio of the useful boiler output, i.e. the increase of energy of the water flowing through the boiler, and its primary energy consumption. The Higher Heating Value (HHV) is used throughout this paper.

RESULTS

As this paper focuses on both the indoor thermal comfort and the energy consumption, Figure 1 and Figure 2 show these variables as function of the boiler output capacity for both the high efficiency and the condensing gas boilers. Figure 1 shows that the indoor climate is acceptable for a heat output of at least 8 kW. The indoor climate is not influenced by oversizing the boiler. The effect of oversizing, however, can be seen in the increased energy consumption (Figure 2) and is mainly due to decreasing boiler efficiencies (Figure3). The effect is much more pronounced for non-modulating high efficiency boilers with a fixed water outlet temperature.

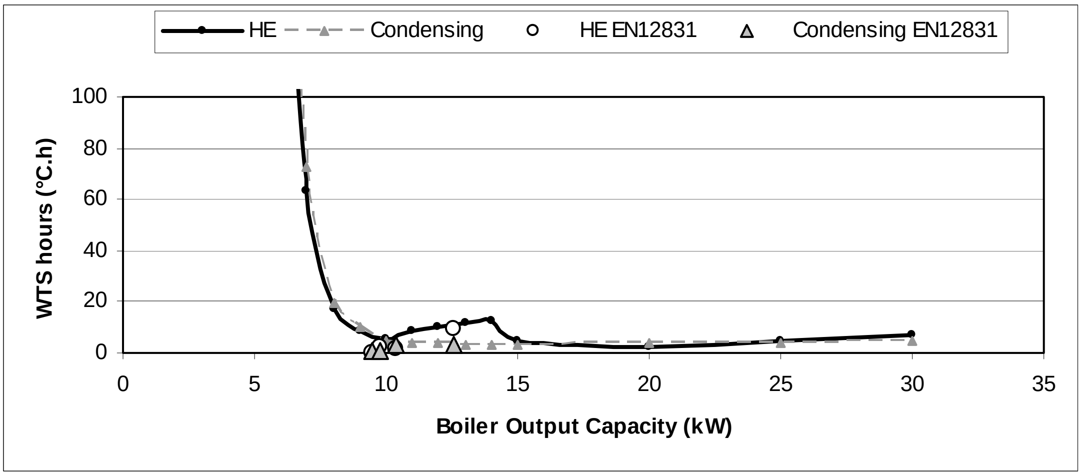

Figure 1: Weighted Temperature Subceeding hours versus the boiler output capacity for High Efficiency and Condensing boilers. The EN12831-cases are the boiler sizes determined by the European heat load calculation code with reheat times from one to four hours.

Figure 2 shows that the electricity use is low compared to the consumption of natural gas, so the effect on the primary energy use is limited. The higher electricity consumption for lower thermal power is due to increasing working time. For increasing boiler size, the cycling frequency increases (as shown in Figure 4), the number of working hours decreases and thus the auxiliary electricity consumption lowers as well. This effect counterbalances the growing ventilator power due to increasing boiler size. The modulating boiler can additionally profit from lower part load ratios and thus lower ventilator consumptions.

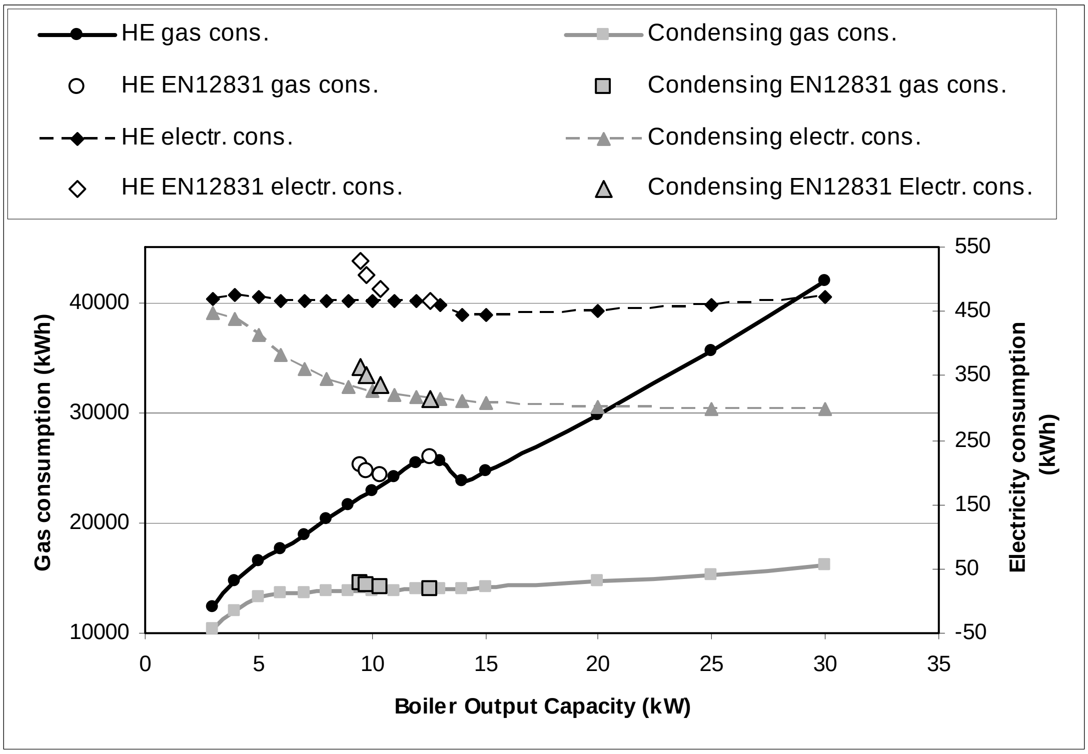

Figure 2: Gas and electricity consumption versus the boiler output capacity for both the High Efficiency on/off boiler and the Condensing modulating boiler.

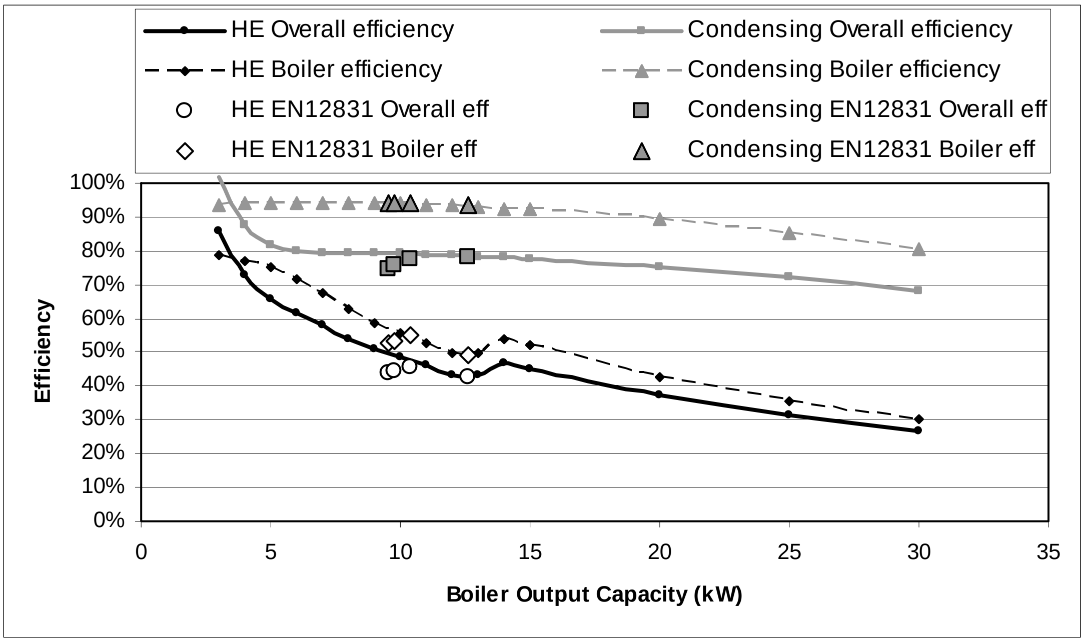

Figure 3: Overall efficiency (net heat demand / primary energy consumption) and boiler efficiency (net boiler output / primary boiler consumption at HHV) versus the boiler output capacity for both boiler types.

The decreasing boiler efficiency is correlated to the increasing cycling frequency. For boiler sizes under 6 kW, the cycling is limited. As Figure 4 shows, the boiler temperature stays below expectations despite almost continuously being switched on. The energy consumption and overall efficiency for these very low boiler output capacities are however not meaningful, since the boundary condition, a minimal indoor comfort, is not reached. The net heat demand is even bigger than the energy consumption in the case of a 3kW boiler. Therefore results below 8 kW should be neglected as a solution, but they are still visualised to show the trends.

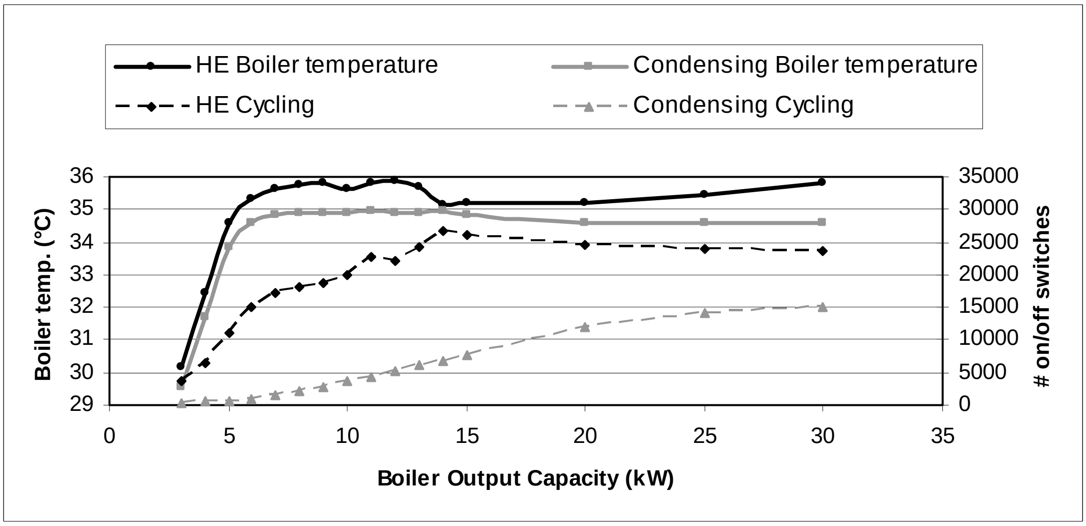

Figure 4: Boiler temperature and number of on/off switches versus the boiler output capacity for both boiler types.

Increasing the boiler size leads to more satisfying indoor conditions and a stable boiler temperature, but also implies a higher cycling frequency. The tendency of this

curve, though, changes for the different boiler types when further increasing the power. The high efficiency boiler, which is a non-modulating system, shows a maximum cycle frequency for thermal sizes slightly above the required heating power, around 15 kW. Modulating, condensing boilers with variable outlet temperature will smoothly tend towards the curve for non modulating systems, as extreme oversizing leads to a comparable working regime.
The results for the four boiler sizes as determined based on the European standard [1] diverge at the lowest boiler output capacities. This is due to the corresponding reheat times of one to four hours, while for the simulations of the range of boiler sizes one hour reheat time is used in all cases. The simulations show that this short reheat time is large enough during most of the heating season, except for a short period in winter. As a result, the increase in night set back time implies only a slightly better thermal comfort (Figure 1), but mainly a higher gas and electricity consumption (Figure 2). A temporary increase of the reheat time, or even adapted winter clothing could offer the same comfort but at a much lower energy consumption.
Deviant behaviour for boiler sizes of around 15 kW can be observed: lower thermal comfort (Figure 1), lower average boiler temperatures (Figure 4) and higher boiler efficiency (Figure 3). The main reason is that the oversizing results in frequent cycling, easily reached maximum temperatures and thus average boiler temperatures that are rather low (Figure 4). This effect is more pronounced during spring and fall. Lower temperatures allow more fume gas heat to be recovered and thus higher boiler efficiency, as can be seen in Figure 3. The modulating capacities cause much smoother curves for the condensing boiler.
Considering both thermal comfort and primary energy consumption, a thermal power from 9 kW to 11 kW leads to the best option, given the boundary conditions, i.e. in the case of the considered terraced house equipped with the specific heating system.

DISCUSSION

Oversizing does not influence the indoor thermal comfort, but leads to increase of primary energy consumption. However, there is a pronounced difference between boiler types. The condensing boiler can maintain its high efficiency quite well, due to low envelope losses, the ability to modulate its power down to 25%, and an outdoor temperature compensated boiler temperature which reduces the boiler temperature and thus skin losses even more.
The High Efficiency boiler with its simple on-off control, at the other hand, shows increasing difficulties to offer a constant water temperature to the TRVs with increasing boiler size. The boiler is required to cycle more and more, which leads to higher boiler temperatures and thus higher boiler losses. This is combined with a constant useful heat output and thus fast decreasing boiler efficiency. For these boilers a central room thermostat would offer a better solution, since this control allows the boiler to cool down between short intervals of (larger) heat demand.
The building heat loads calculated following the European standard only slightly overestimate the boiler size. However, the additional thermal power for intermittent heating seems difficult to predict. The corresponding reheat time is often overestimated, decreasing the overall efficiency. To put it otherwise, the heat load calculated for a reheat time of 4 hours suffices to heat up the building in slightly more than one hour.

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Heat loss in distribution system influence on energy requirement from heating station

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SUMMARY

The objective of the study was to verify the assumption that heat loss in the distribution system would increase the boilers intermittence, and hence increase the final energy requirement. To verify the assumption, a model for selecting boilers in an energy flexible heating system is used to calculate the final energy requirement. The study shows a rather considerable increase in the energy consumption due to increased intermittence and reduced system efficiency. The result of the study demonstrates the importance of locating the distribution system in the building so that as much as possible of the heat emission from the distribution system can be recovered.

INTRODUCTION

Predicting the energy consumption for building heating purposes is important for several reasons. For instance when selecting boiler based on lowest running cost in an energy flexible heating system or when energy budgets are calculated.

It was assumed that when taking into account the relative heat loss in the distribution system in central heating system, the boilers intermittence would increase and hence the boilers efficiency would decrease making the final energy requirement increase.

The objective of the study was to verify this assumption and increase the awareness of this effect on the final energy requirement and the total running cost.

METHODS

To investigate whether the heat loss in the distribution system have any influence on the final energy requirement a model for selecting boilers in an energy flexible heating system is used.

Relative heat loss, L_d

Parts of the heating system located in unheated spaces must be insulated to reduce undesirable heat losses. Suitable insulation classes can be selected from The European Standard EN 12828 [1]. When calculating which insulation class to be used for the distribution system three critical factors must be defined; the heating period for which the heating system is operating, the water- and surroundings temperatures and the fraction of heat emission considered wasted, f_a. When all these three factors are determined the operational parameter I can be calculated. Based on the value of I the corresponding recommended insulation class can be obtained from EN 12828, Annex C. The maximal thermal transmittance (W/m·K) corresponding each of the insulation classes from 0 to 6 can then be calculated for each heat pipe based on EN 12828. The total heat emission, L_e, for the entire distribution system can then be found by adding up all the heat emission from each pipe. L_e depends on temperature difference between water and ambient temperature, the pipe dimension and length of the pipe and the insulation of the pipe.

The relative heat loss from the distribution system, L_d, must not be confused with the relative heat emission from the distribution system, L_e. The relative heat emission is defined as the total heat emission from the adequate insulated distribution system, while the relative heat loss is defined as the fraction of the relative heat emission that is considered as wasted. The relative heat loss, L_d, will then be:

$$L_d = f_a \cdot L_e \quad (1)$$

Final energy requirement, Q_f

To demonstrate the effect of L_d on the final energy requirement a model building is constructed, and the final energy requirement is calculated. Riise, Jensen Sørensen [2] describes a method for calculating and selecting boilers in an energy flexible heating system based on lowest running cost.

The final energy requirement, Q_f, can be calculated when the total heat demand and the efficiencies of each boiler are known

$$Q_f = \frac{Q_d}{\eta_l} \text{ (kWh)} \qquad (2)$$

where:

Q_d total energy requirement to the heating system (kWh)
l the boiler in question
η_l efficiency of boiler l ( )

[2] presents a set of equations to find the efficiency as a function of the heat demand from the heating system for an intermittently controlled boiler. These equations have been modified to take into account relative heat loss due to inaccurate control, L_{con}. Heat loss due to inaccurate control will probably cause an increase in the final energy requirement.

$$\eta_l = \frac{Q_{out}}{Q_{inst}} \cdot (I_l + 1) \qquad (3)$$

The intermittence rate can be obtained by using equation (4)-(7) as follows:

$$I_l = \frac{N_l - 1}{\frac{N_l}{I_{l,\max}} + 1} \qquad (4)$$

$$N_l = \frac{Q_{inst} \cdot \eta_{l,\max}}{Q_{out}} = \frac{Q_{\max}}{Q_{out}} \qquad (5)$$

$$\begin{aligned} I_{o,\max} &= \frac{1 - L_{eg} - L_r - L_d - L_{con}}{L_r + L_g + L_d + L_{con}} \\ I_{e,\max} &= \frac{1 - L_r - L_d - L_{con}}{L_r + L_d + L_{con}} \end{aligned} \qquad (6)$$

$$\begin{aligned} \eta_{o,\max} &= 1 - L_{eg} - L_r - L_d - L_{con} \\ \eta_{e,\max} &= 1 - L_r - L_d - L_{con} \end{aligned} \qquad (7)$$

where

η_l	mean efficiency for the boiler in question ( )
Q_{out}	heat output from heat system (kWh/h)
Q_{inst}	gross installed boiler capacity (kWh/h)
Q_{max}	maximum heat output from heat system
I	intermittence rate ( )
N	help variable
I_{max}	maximum intermittence rate ($\eta=0$) ( )
η_{max}	maximum operating efficiency for unit ($I=0$) ( )
L_{eg}	relative exhaust gas heat loss ( )
L_r	relative radiation and convection heat loss for boiler surface ( )
L_d	relative distribution heat loss ( )
L_g	relative heat loss caused by air draught at boiler off time ( )
L_{con}	relative heat loss caused by inaccurate control ( )

The heat loss from each boiler must be obtained from calculations, measurements or given by the boiler manufacturer. The heat loss from distribution system must be calculated for each case. It is assumed that L_{con} will be affected by the size of f_a in the calculation of L_d. The effect of L_{con} on the total energy consumption is not a part of the study. Neither is the calculation of f_a.

When the efficiency for each boiler have been calculated for the time step in question, it is a simple matter to determine the final energy requirement, Q_f, Q_f ca be found by combining equations (2) and (3) for that time step:

$$Q_f = \frac{Q_d}{\eta_l} = \frac{Q_d}{\frac{Q_{out}}{Q_{inst}}\left(I_l + 1\right)} \text{ (kWh)} \qquad (8)$$

This approach is not widely used in cases where the selection of boilers is done manually, but it have become more and more common in automated heat stations where a control unit selects the optimal boiler based on lowest running cost.

Calculation / simulation model

A model building is used in a simulation case based on [2], with modified to account for L_{con}. Building data is shown in table 1, and necessary external data (outdoor temperature and energy prices) are displayed in figure 1 to figure 3.

Table 1. Building data for model building used in simulation case

Type of building	Office building
Indoor set-point temp	17°C
Area	3100 m^2
Volume	9900 m^3
Infiltration rate	0.3 h^{-1}
Total heat transfer incl. infiltration	3258 W/K
Heat demand domestic hot water	5 W/m^2
Heating period	365 days, due to production of domestic hot water.
Air specific heat capacity	0.35 $Wh/(m^3K)$
Airflow rate	20.000 m^3/h
Heat recovery efficiency, ventilation	0.75
Operating time ventilation	0600-1600 (Monday-Sunday)
Oil boiler gross installed capacity	400 kW
Electric boiler gross installed capacity	350 kW
External temperature	Daily maximum and minimum temperature given by Norwegian Meteorological Institute. Location Narvik. Year 2005.
Electricity prices	Hourly electricity prices given by Nord Pool ASA. Figure 2 shows average daily prices for 2005 including transmission charges, EURO/kWh, from local power grid owner, Narvik Energi AS.
Oil prices	Approximately marked price for fuel oil for 2005. For simulation purposes it is assumed different prices for each month of the year. Figure 3 shows prices for each month used in the simulation example.
Lower calorific value	10 kWh/litre
Specific oil boiler losses	$L_{eg} = 0.085$ $L_r = 0.035$ $L_g = 0.02$ $L_e = 0.045 - 0.063$ [1] $L_d = 0.000 - 0.045$ [2] $L_{con} = 0.05$ [3]
Specific electric boiler losses	$L_r = 0.025$ $L_e = 0.045 - 0.063$ [1] $L_d = 0.000 - 0.045$ [2] $L_{con} = 0.000$ [3]

1) The total heat emission from the distribution system depends on the insulation class defined in European Standard EN 12828. The insulation classes reflect the fraction of the heat emission considered wasted, f_a, from 0.0 to 1.0. A low fraction of the heat emission wasted allows a lower insulation class and hence a higher total heat emission from the distribution system since a larger share of the emission will benefit the building for heating purposes.

2) The total heat loss depends on the magnitude of f_a. Wigestad, Dokka, Pettersen and Myhre [3] suggest that all non direct-acting heating systems will have a relative distribution heat loss, $L_d = 0.05$ of all delivered heat from the heating station. In this study L_d is determined based on equation (1).

3) Wigestad, Dokka, Pettersen and Myhre [3] suggest $L_{con} = 0.05$ for oil fired boilers and $L_{con} = 0.0$ for electric boilers.

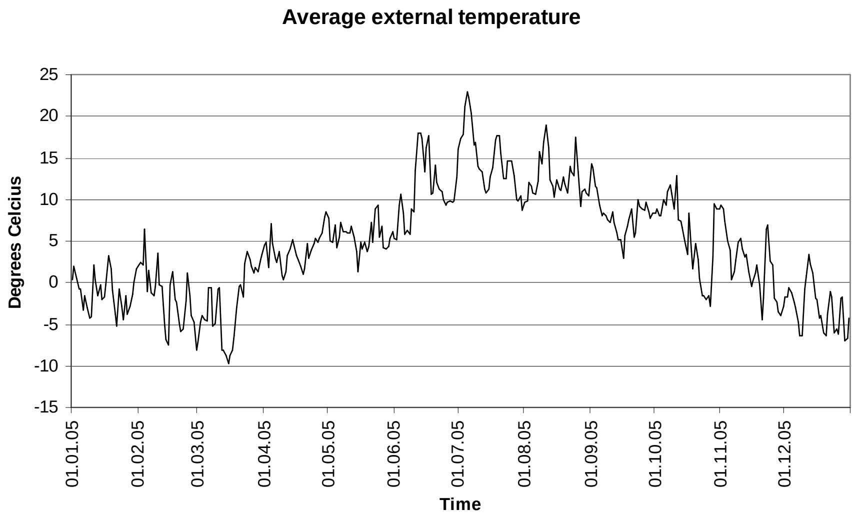

Figure 1. Average outdoor temperature in Narvik 2005 used for simulations.

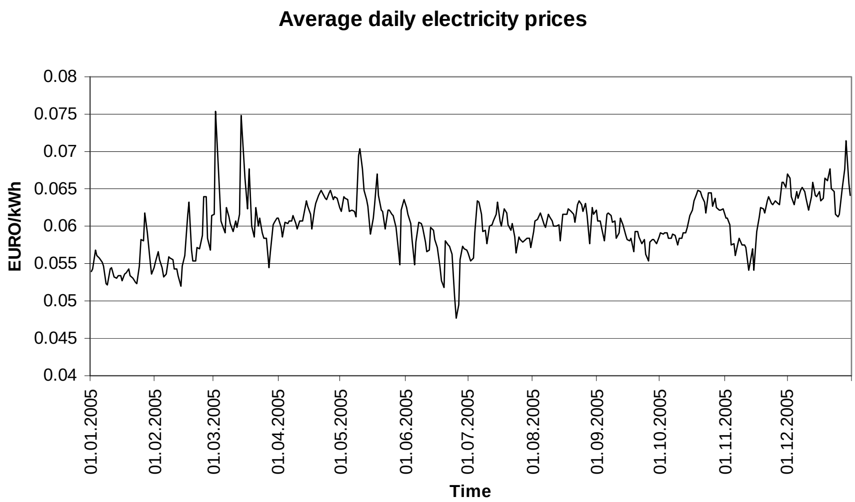

Figure 2. Average daily electricity prices including transmission cost used for simulations.

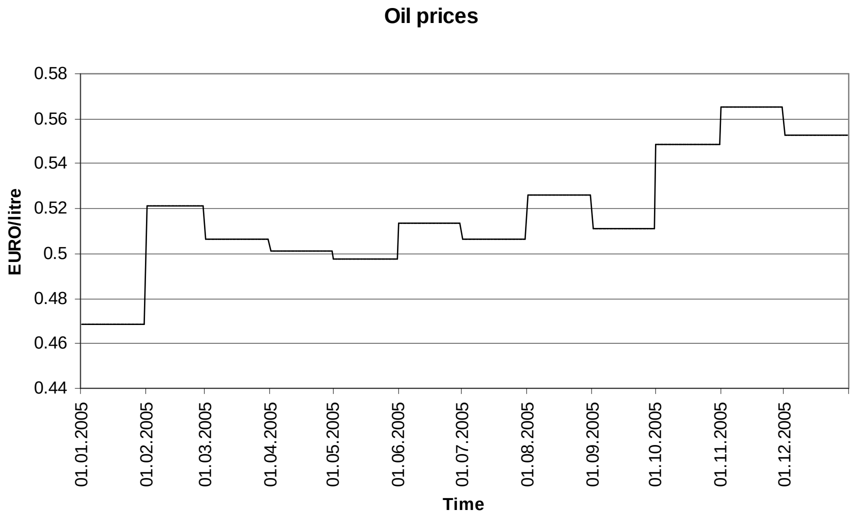

Figure 3. Average monthly oil prices used for simulations.

RESULTS

f_a, heat emission and heat loss

Total heat emission and heat loss depends on the fraction of heat emission from the distribution system considered wasted. Table 2 shows the connection between f_a and total relative heat loss from the distribution system and heat recovered from the heat emission for the model building together with the corresponding operational parameter, ***I***, and insulation classes.

Table 2. Relation between f_a and L_d

f_a	Operational parameter	Insulation class [1]	Relative total heat emission, L_e [2]	Relative heat loss (waste), L_d [3]	Relative heat recovered	Final energy requirement (kWh)
0.05	0.079	1	0.063	0.0032	0.0599	926 240
0.10	0.158	1	0.063	0.0063	0.0567	936 011
0.15	0.237	2	0.058	0.0087	0.0493	943 579
0.20	0.315	2	0.058	0.0116	0.0464	952 638
0.25	0.394	3	0.054	0.0135	0.0405	958 495
0.30	0.473	3	0.054	0.0162	0.0378	966 986
0.35	0.552	3	0.054	0.0189	0.0351	975 473
0.40	0.631	3	0.054	0.0216	0.0324	983 875
0.45	0.710	4	0.049	0.0221	0.0270	985 295
0.50	0.788	4	0.049	0.0245	0.0245	992 988
0.55	0.867	4	0.049	0.0270	0.0221	1 000 528
0.60	0.946	4	0.049	0.0294	0.0196	1 008 528
0.65	1.025	4	0.049	0.0319	0.0172	1 015 943
0.70	1.104	4	0.049	0.0343	0.0147	1 023 726
0.75	1.183	4	0.049	0.0368	0.0123	1 031 315
0.80	1.261	4	0.049	0.0392	0.0098	1 038 948
0.85	1.340	4	0.049	0.0417	0.0074	1 046 542
0.90	1.419	5	0.045	0.0405	0.0045	1 042 924
0.95	1.498	5	0.045	0.0428	0.0023	1 050 002
1.00	1.577	5	0.045	0.0450	0.0000	1 057 270

[1)] From EN 12828
[2)] Total heat emission from the distribution system relative the gross installed capacity.
[3)] Total heat loss for the distribution system relative gross installed capacity, $L_d = f_a \cdot L_e$

Table 2 demonstrates the effect of f_a on the total heat emission from the distribution system, and the fact that a higher share of the heat emission recovered for building heating purposes allows a higher total heat emission compared with a situation where all the heat emission is considered as waste.

Final energy requirement

Figure 4 shows the increase in final energy requirement from the heating station with different values of f_a, For this particular case the increase in energy requirement is 8.4 % with a f_a 0.5, and 15.4 % with a f_a 1.0.

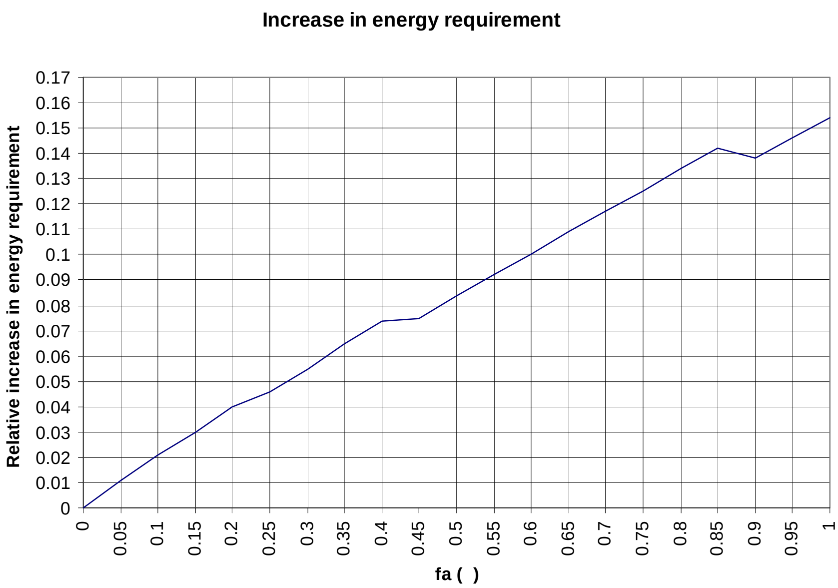

Figure 4. Relation between f_a and increase in energy requirement from the heating station.

It is also possible to investigate the effect on the final energy requirement if the considered value of f_a is incorrect. For instance, if f_a is assumed to be 0.10, but the actual f_a is 0.4. Then the final energy requirement will be approximately 6.3 % higher than calculated. In this case f_a 0.10 gives insulation class 1, and f_a 0.40 insulation class 3. If insulation class 1 is used instead of class 3, with f_a 0.4, the final energy requirement will be approximately 1.1 % higher than if insulation class 3 is used.

Intermittence rate

The reason for this relative high increase in the energy requirement can be identified by examining the intermittence, I, for the different cases. For $f_a = 0.05$; $I_{max,oil}$ is 5.15 with a boiler efficiency 0.27 and $I_{max,el}$ is 11.8 with a boiler efficiency 0.64. For $f_a = 1.00$; $I_{max,oil}$ is 3.8 with a boiler efficiency 0.23 and $I_{max,el}$ 7.3 with a boiler efficiency 0.42. A closer study of the result of the boiler selection shows that the electric boiler is the predominant under existing conditions.

System efficiency

The result of the increase in the energy requirement is also visualized by the mean system efficiency as shown in figure 5.

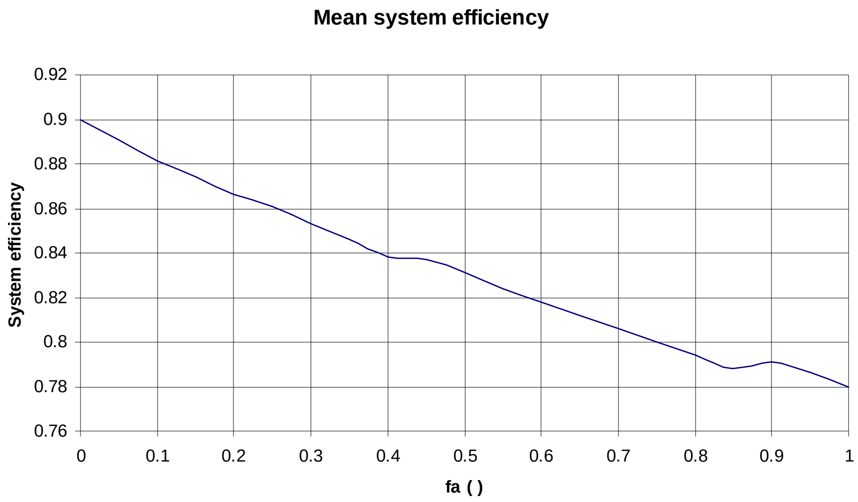

Figure 5. Mean system efficiency.

DISCUSSION

The results of the study show that the location of the distribution system in the building can be vital to the energy consumption, and hence the energy cost. Following the minimum insulation requirements, it is evident that to design the distribution system in such a way that as much as possible of the heat emission from the distribution system can be recovered for building heating purposes is preferable. If it is not possible to reduce f_a one should consider the possibility to exceed the minimum insulation requirement in EN 12828. The study has not considered the effect of an increase in the insulation thickness or quality.

Further it is not considered how the heat loss from the distribution system will affect the control of the heating system. It can be difficult to control the system correct if undesired heat loss from the distribution system is allowed to enter heated spaces in the building. EN 12828 recommends that an undesirable increase of room temperature of more than 2 K should be avoided.

To reduce undesirable heat loss, insulating the distribution systems is important. However, this study shows that it is not sufficient just to focus on the maximum thermal transmittance from the distribution system. The effect of heat losses from intermittent controlled boilers should also be considered.

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New energy efficient solution to control the flow in terminal units

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SUMMARY

This paper compares pressure independent balancing and control valves (PIBCV's) to traditional – pressure dependent – control valves.
Traditional valves are subject to changes in differential pressure that occur during normal operation. These pressure fluctuations cause deviations in the room temperature and change the control response. In practice this results in unpredictable control behavior and the inability to control small flows.
In contrast, pressure independent valves eliminate these fluctuations. As shown by calculations, laboratory tests and measurements in a real life installation, removing the differential pressure from the control equation results in significantly improved room temperature control. Additionally the PIBCV also automatically limits the flow to the designed flow rates at all load conditions, thus increasing the pump efficiency. Preventing overflows will also increase the performance of boilers and chillers because temperature differences over the terminal units will be in accordance with the design specifications.

INTRODUCTION

This paper is focused on control valves for fan-coil units (FCU), chilled ceilings & beams, air-handling units (AHU) and similar terminal units (TU). Most water based HVAC systems apply control valves to control the flow through heat exchangers. The flow is adjusted according to the difference between the set and the real temperature and normally this adjustment is achieved by de- or increasing the valve opening.
According to system demands, experience and personal preference the system designer has several choices with regards to valves like fast opening, linear or equal percentage, and control strategies like on/off, chrono-proportional or modulation.
Engineers worldwide use only a few different methods for sizing valves which are all derived from the same background. These methods range from assuming a fixed differential pressure (for example the 35kPa rule) or to limiting the minimum authority (for example a>0,25), while the selection of the valve characteristic is based on expectations of the performance (for example: Equal percentage are expected to be better then linear valves) or according to the characteristic of the terminal unit and the control authority of the valve.
The article's goal is to show the differences in performance between traditional control valves and pressure independent balancing and control valves (PIBCV's).

METHODS

In order to put a value to the performance of the control valve (provided there's adequate temperature control) we need to mention the study performed by Federspil et al [1] where the main findings after extensive complaint analysis were that 77% of all claims were of a thermal nature. Subsequently a model was developed to show the relation between temperature deviations and temperature change rates. According to this study any excessive temperature deviation is penalized by costly claim handling. The study goal didn't include Indoor Air Quality (IAQ) that could also negatively influence the working environment and add considerable costs. This study proves the benefits of adequate temperature control: The number of claims will double when the average temperature is 1K above ideal (23°C) and if we increase the standard temperature deviation from 0,5 to 2K. The cost of these claims could then be as high as 10%, compared to the total energy cost for the installation.

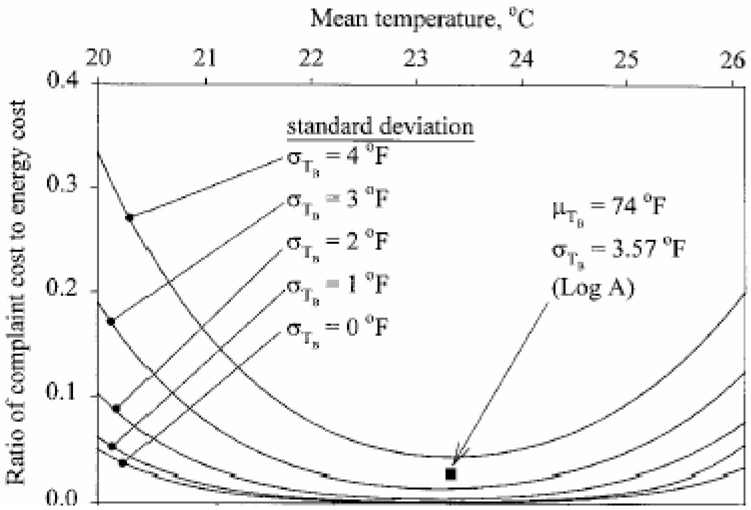

Figure 1: "Cost of complaints relative to cost of energy during summer in Huston" [1] in relation to room's mean temperature and its deviation .

One of the methods to select a control valve (CV) is described in Recknagel [2] where the main goal to design a CV is to compensate for:

- Un-linear characteristic of heat exchanger (terminal unit)
- The increase of the differential pressure when the valve closes (described by the valve authority)

The procedure of the valve selection is summoned in following graph (figure 2).

The last graph shows that the selection of a valve characteristic depends on the characteristic of the heat exchanger and the authority of the control valve. This means that for water-air heat exchangers (a typical factor a is between 0,15 to 0,3) an equal percentage valve should be selected, while for water-water heat exchangers (typically >0,5) a linear valve is preferable. However in between lines we need to accept a deviation from the ideal situation. Furthermore, the valve characteristic combined with that particular valve's authority will cause different gains at different loads. In order to provide a stable control of the temperature, the correct P-band needs to be defined. A higher deviation from the ideal selection would result in a bigger P-band, meaning more deviation from the required temperature.

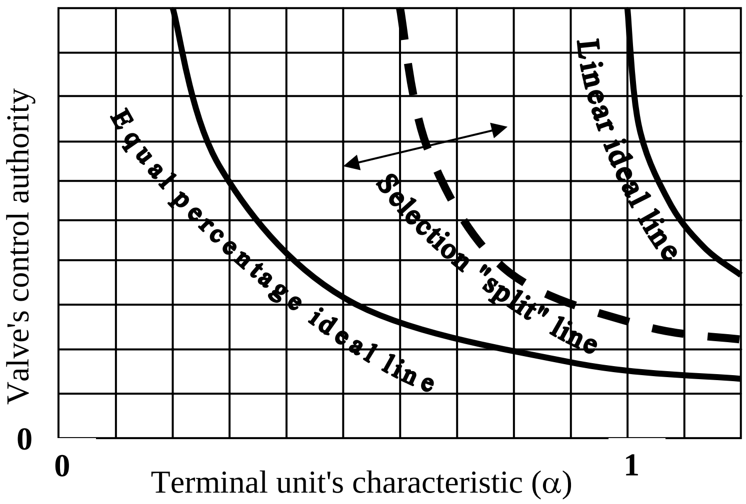

Figure 2: Selection criteria for control valve characteristic [2].

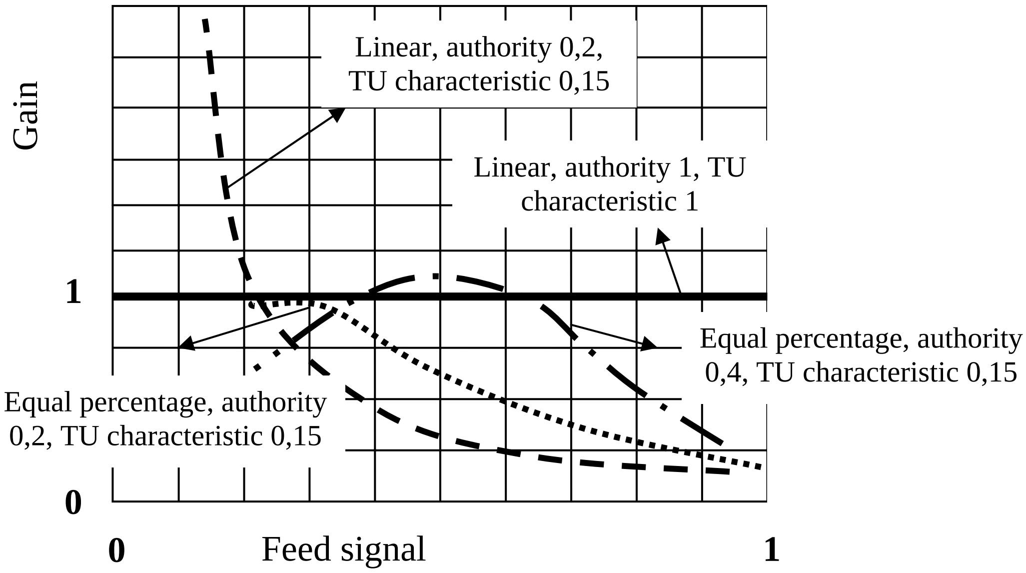

Figure 3: Gain for different valve types, TU characteristic and control authorities [2] .

Another issue is the ability of the control valve to control small flows. This is an issue especially with equal percentage valves where the ratio between the lowest kv that still complies with the equal percentage characteristic and the highest kv value (Kvs) is defined as the control ratio. This is typically between 25 to 100, meaning that stability can be obtained between 4% and 1% of the nominal kvs values respectively. On figure 3 we can see this phenomenon as the gain curve for equal percentage valve ends on 10% of the signal where valve characteristics change so gain increase in step. Detail definition of control range is defined by standard IEC 534-2-4 [3]. For example at a pressure change ratio (authority) of 0,1 and a control ratio of 1:50 we can only control emissions higher than 30%! For linear valves this is less of a problem because at lower valve positions their characteristic will stay linear or very close to linear, more or less indefinitely.

<u>The aforementioned methods assume that the valve under consideration is the only one in the system, meaning that the only change of the available pressure is caused by the valve itself.</u> When we consider control valves in a typical secondary system for FCU’s, chilled beams or AHU’s that is not the case. Since there will be many TU’s in a system, there will also be

many valves controlling the flows through the TU's. Changing the flow through one particular valve will have consequences for the available differential pressure of all other valves. This means that in a realistic situation we would need to consider all the possible gains (as depicted on figure 4) between minimal authority (a=design dp / pump head) to authority 1. The fact that any change of flow through one TU will change the flow through all the other TU's means there are constant interferences with the temperature control loop of the TU's. In effect this causes the valves to constantly adjust their positions to the ever changing available pressure, that results in changing flow – emission and at last temperature (figure 4). PIBCV's don't have these problems. Changes of flows in TU's will not result in a change of flow for any other TU. This means that by using PIBCV's the only interference to the temperature loop is a change in the demand (load). In most of the systems the load is changing slowly compared to the fast changes in available pressure. Systems with PIBCV's are therefore much easier to control than systems with traditional control valves.

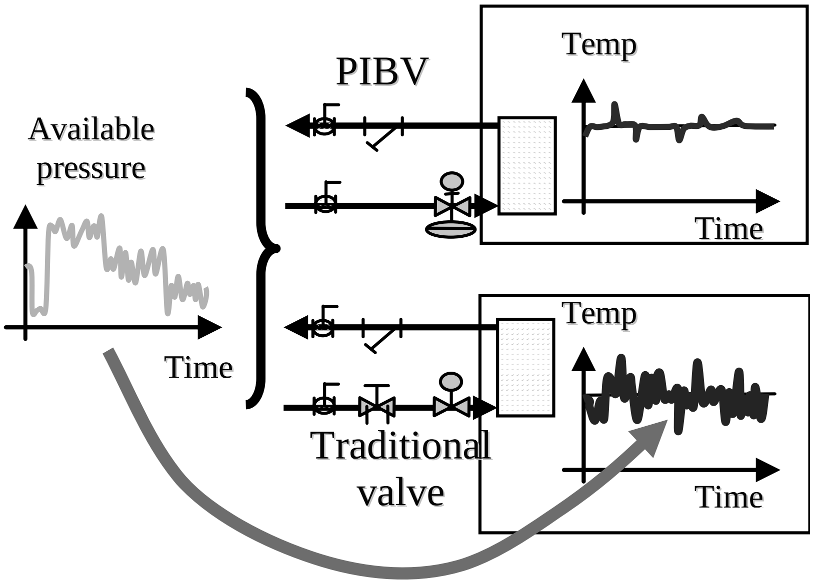

Figure 4: Changes of available pressure causes changes of flow – emission – temperature in case of traditional valve.

RESULTS

In order to research the ability to control small flows of different types of control valve a special test was performed where the control valve had to control the flow while the available pressure was changed. This is similar to conditions where the available pressure changes while the load remains constant. There were 3 valves exposed to the test: a linear valve, an equal percentage one (both kvs 1,6 m3/h) and a PIBCV on flow setting 450 l/h (the maximum for this size).

A comparative test was performed on the same test stand with the same actuator and with the same PI controller with the same settings. The PI-controller was set to realize 100 l/h and the pump was gradually increased and then decreased to zero available pressure. The results are depicted on figure 5. The linear valve is stable only at low pressures when it has a high authority. The equal percentage valve can control the flow up to 150 kPa of available differential pressure over the valve. Above 150 kPa the valve has to go below the minimum Kv-value as defined by the control ratio and this causes unstable control. The actuator on the pressure independent valve only needs to correct for the P-band deviation of the differential pressure controller, a mechanical controller inside the PIBCV.

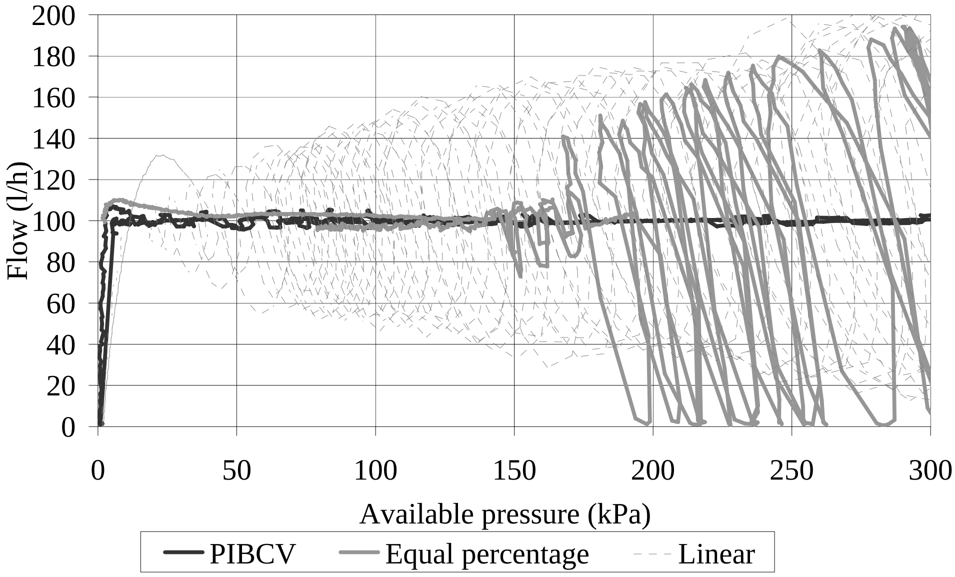

Figure 5: Performance comparison test.

In order to compare equal percentage CV’s and PIBVC’s measuring and monitoring of a real life installation was done. The installation was situated in an office-building in Amsterdam and for the purpose of the test two small offices were measured. The offices were next to each other, on the same floor and with the same orientation so the demand was comparable. The temperatures on the FCU inlet & outlet were measured. In one office an equal percentage valve was originally installed. In the other office we have exchanged the valves for PIBCV’s. The situation was monitored for 6 months and measurements stored on a temperature logger. Figure 6 depicts the temperature behavior in the morning hours when the load in both rooms was comparable (outlet temperature from FCU is similar). In the room with the equal percentage valve the outlet temperature from the FCU was changing considerably as opposed to the room with the PIBVC. At this time the valves were operating under low load conditions causing the equal percentage valve to operate below its range-ability as defined by its control ratio (1:100) and sizing (valve was sized for 10 kPa and max flow of 450l/h – resulting in kvs1,6). There are reasons to claim the sizing was not adequate. The sizing calculations assumed that balancing will provide the same available differential pressure for all CV’s in the system because the maximum flow was the same on all units. Above mentioned presumption is realistic if the balancing was perfect. However, balancing is done at the construction site with time consuming iterative and complicated methods, based on measurements of differential pressures across balancing valves or fixed orifices. Balancing valves ore often mounted on unsuitable places like behind elbows or other disturbances. Those are definitely not lab-conditions and significant deviations of the actual pressure and consequently flow can be expected. Manual balancing valves are used to achieve flows as close as possible to the design quantities, where pre-assumptions on CV sizing are long forgotten or probably unknown. It is noteworthy that manual balancing valves only provide proper flow limitations at design conditions, which means all the valves fully open. In any other hydronic condition (partial loads) they are not functional and only adding to the pump head or reducing the CV’s authority.

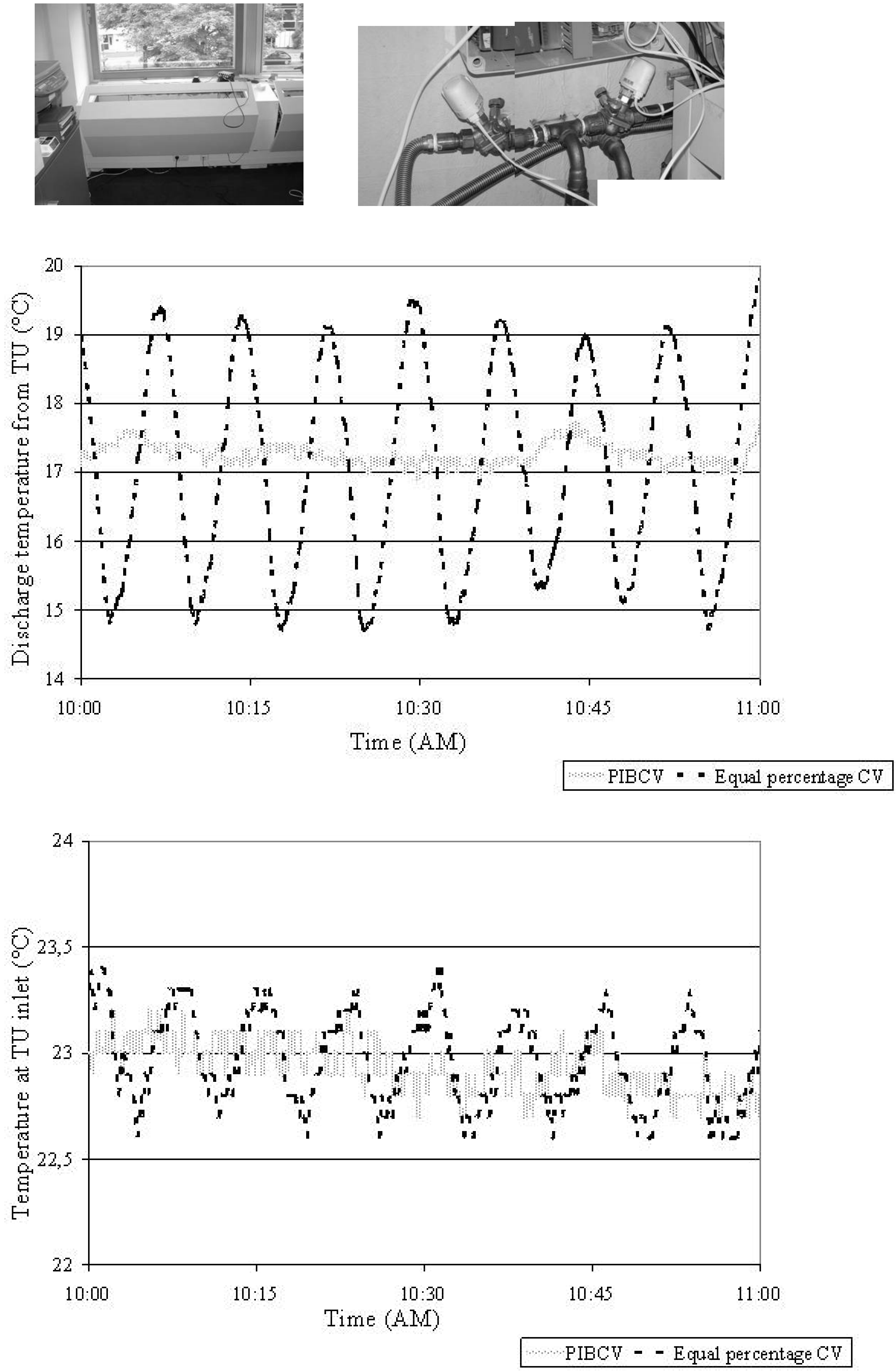

Figure 6: Real site comparison test – installation pictures and temperature on inlet and outletof TU.

DISCUSSION

It's impossible to compensate for a poor performing hydronic system with control software or by other electronic means. In order to provide well-being indoors and efficient transfer of energy between source and terminal units, control valves need to be sized with care and the balancing has to be performed correctly. An alternative to traditional control valves would be a pressure independent balancing and control valve that is prone to fewer disturbances and has a practically unlimited range-ability. Because PIBCV's also perform a flow limiting function there will be no overflows. This increases the efficiency of pumping and also benefits the chiller or boiler because the proper differential temperatures are maintained. Additionally, the PIBCV can be adapted to the characteristic of any heat exchanger because is has a control authority of 1 so the adaptation to the heat exchanger can be made by means of actuator/ controller signal transformation.

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Pressure Surges in Drinking-water Installations

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SUMMARY

The new guidelines, for example the new EN 806, and new perceptions of water hygiene force more better dimensioning of pipes for potable water. The results are often smaller diameters of the potable water tubes and smaller hot water boilers.

On the other hand pressure surges are more often observed. This report will try to declare these effects by different methods: First a test stand for the measurement of pressure surges and evaluation of drinking water armatures is presented. The investigation of the armature technology especially of fast closing armatures shall help to improve new realizations and optimizations for the building services and engineering supply.

At second the theory is introduced: A new technical guideline about this topic, the VDI 6006 „pressure surges in drinking-water installations, causes and preventions”[2], treats for the first time in Germany. It addresses itself to planners, but also consultants and manufacturers of sanitary-technical products and plants.

At last the connections are discussed. While new installation methods demand smaller pipe diameters, these demand increases the pressure surges. Thus a compromise between drinking water hygiene and technically necessary piping dimensions must be found. If the compromise should not be found, the search for technical solutions for the reduction of these effects has to be strengthened. So the closing time must be affected in such a way that a very short term closing is no longer possible. Perhaps also pressure impact absorbers are a solution, which must be examined before however also under hygienic aspects.

INTRODUCTION

The pressure surge is a phenomenon that can be observed more frequently. Its effects are not only the well-known noise problems, what disturbs particularly in house building, but also technical problems in the attachment of the pipes. These effects can be force incorrect and inadvertent responding of technical safety devices. In the last years theses phenomenons are observed more often as in further times.

Obviously there is a (new?) problem in drinking water installations of today. The question is why this problem is increased.

Sanitary engineers of today have only one theme to discuss: The water hygiene. New guidelines, for example the new EN 806, and new perceptions of water hygiene force more better dimensioning of pipes for potable water. This means in most cases smaller diameters of the potable water tubes and smaller hot water boilers.

Hygiene problems are no theme of today anymore in this well engineered installation systems. But now the pressure surge occurs as a new problem.

The pressure surge is a phenomenon which is not trivial. There are a lot of influences on it. One of the important facts is the diameter and length of the tube to the armatures. The armatures themselves are responsible too.

METHODS

To examine the problems of pressure surge – which are discussed briefly in the introduction - two proceedings are possible. On the one hand theoretical considerations should be cared about, on the other hand around practical examinations have to be made. Both methods are presented here briefly.

THEORATICAL BACKGROUNDS

The pressure surge in water installations depends on the medium itself, the velocity of the pressure surge and the variation of the drinking water velocity.

The highest value of the pressure surge (the so called Joukowski-surge) Δp_{max} is calculated:

$$\Delta p_{max} = \rho \cdot a \cdot \Delta v \qquad \text{in Pa} \qquad (1)$$

where is:

ρ density of water in kg/m^3 (about 1.000 kg/m^3)

a velocity of the pressure wave in m/s

Δv variation of the velocity in the drinking water pipe in m/s (at Δp_{max}: variation of velocity to 0).

The velocity of the pressure wave a depends on the elasticity of the medium. However in installation systems the pressure wave a is influenced by the elasticity of the medium and by the elasticity of the pipe material.

$$a = \frac{1}{\sqrt{\rho\left(\frac{d_i}{s\,E_R} + \frac{1}{E_W}\right)}} \qquad \text{in m/s} \qquad (2)$$

where is:

d_i inner diameter of the pipe in m

s thickness of the pipe in m

E_R elasticity modul of the pipe material in N/m^2

E_W elasticity modul of the drinking water in N/m^2

For example in copper pipes the following pressures are theoretically possible, when the final armature is closed suddenly. The values depend on the flowing volumes and the diameter of the pipes (fig.1)

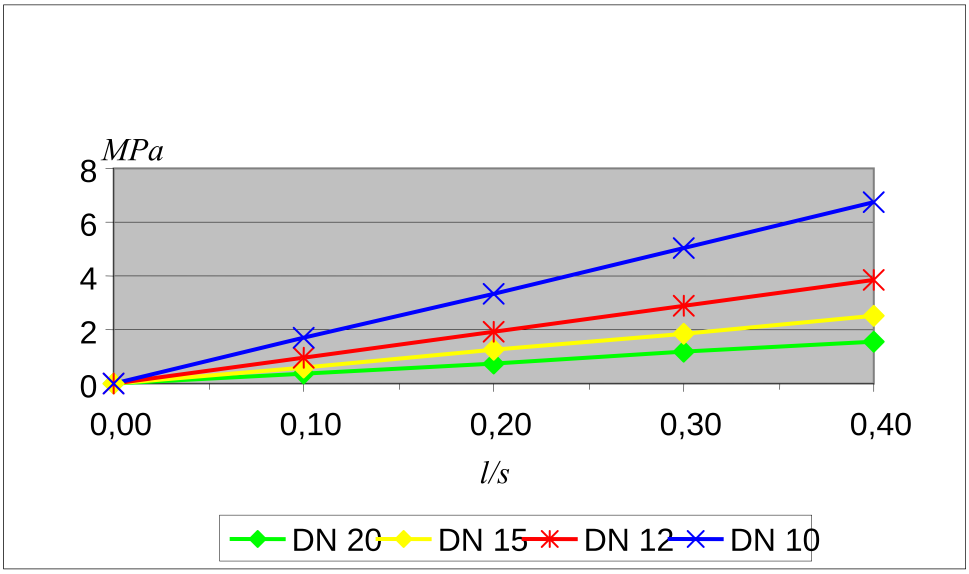

fig.1: maximum pressure surge in relation to the inner pipe diameter for copper pipes (remark: 1 MPa = 10 bar). [2]

There are a lot of other effects, which can not be declared mathematically. Some examples (Table 1):

table 1.: relevant parameters [2]

parameters	influence on pressure surge
volume velocity	++
water temperature	0
shut down time of the valve	++
length of the pipe	+
bent pipe, angle pipe	0
T-distribution	+
diameter	+
drain back valves	+
fixation at the building	0
flexible connection	+
material of pipe	++
++: heavy influence +: influence 0: nearly no influence	

MEASURING OF PRESSURE SURGE AND EXPERIMENTS

To compare reality with theoretical values the idea of experiments were born. in former times industries have made a lot of tests with their valves and armatures. You have to use oszillographes to measure peaks of pressure which was expensive und complicate. The results of these experiments were used to improve their products. Nobody was interested in the installations of the buildings.

In a first step at our university we started to build a multifunctional test plant. In a diploma work [3] this test plant was concepted, built and first experiments were done. The concept is that different armatures and valve and different pipes (concerning the length and material and diameter) can be tested and compared with each other.

Figure 2 shows the scheme and figure 3 a photo of the test plant. In the first step copper pipes are built in. Pipe 1 has a length of 5 metres; pipe 2 has a length of 50 metres. The installed copper pipes have a diameter of DN 15 or d_i = 13 mm.

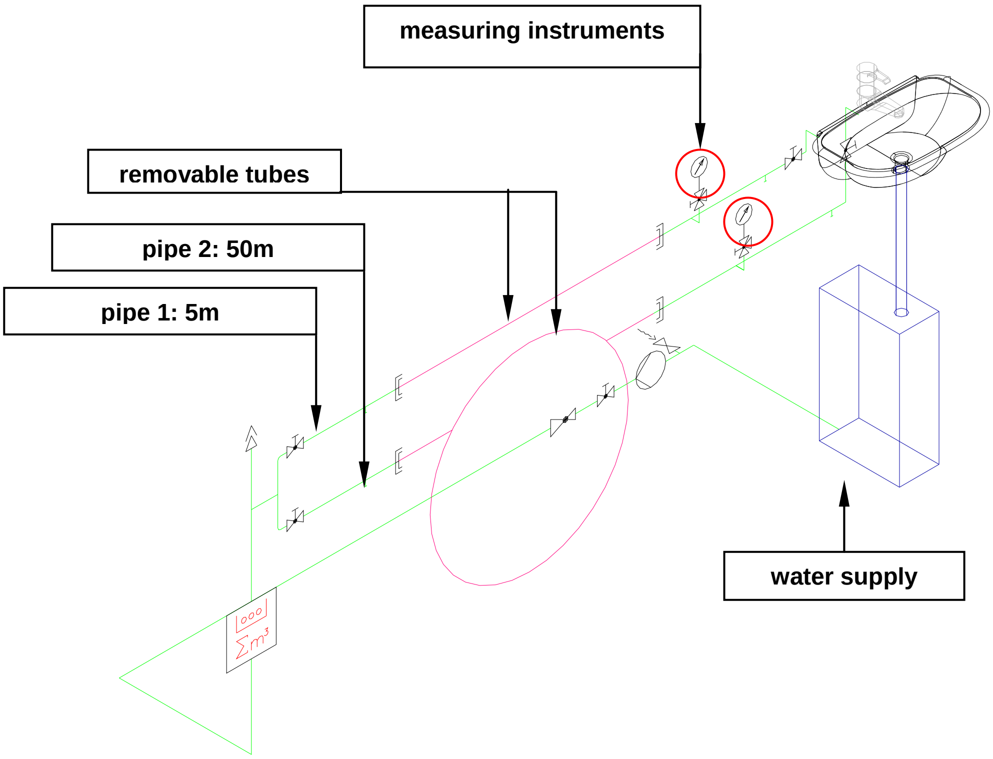

Fig.2: test plant to measure pressure surge [3]

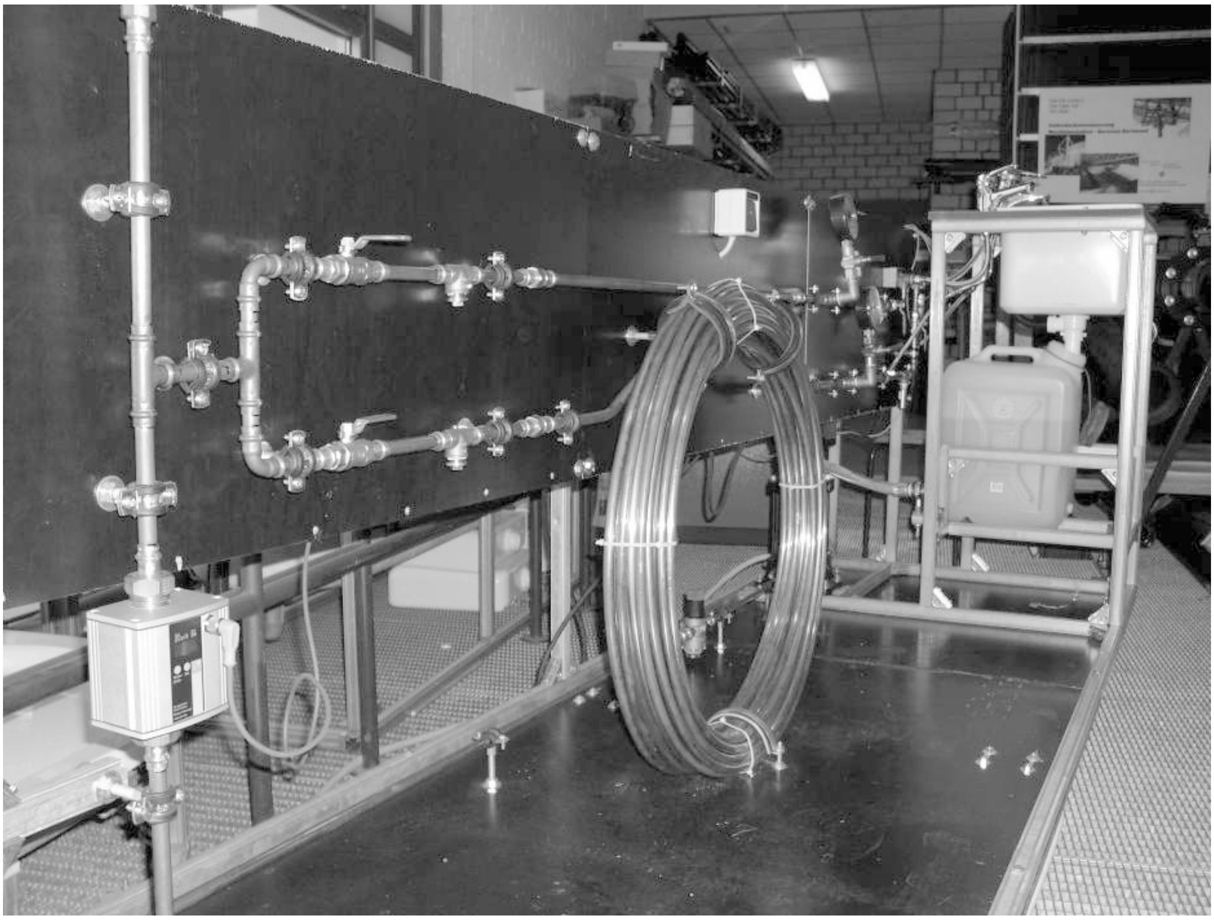

fig. 3: photo of the test plant to measure pressure surge [3]

There are a lot of details which had to be regarded: for example the measurement equipment or the measurement of the volume velocity. All these details are described in [3].

One problem, which is not solved until now, is the replayability of the shut down time. A second work at our university will deal with this theme and I hope that we will have results in about one year. Here in this report first results and the possibilities of this test plant shall be introduced.

RESULTS

First of all figure 4 shows the normal measurements while closing a normal armature (a so called one lever armature). In all following examples the armatures are closed fast and in an usual way. There were not unusual closings by beating on the armature or so on.

Hammering on the armature will cause very high pressures (up to 50 bar). The consequence will be a destroyed test plant. Our first results show, that pressure peaks about 20 bar are normal. Our technical rules, for example the EN 806, describe however that all components of drinking water installation system have to be built for a maximum of 10 bar.

A real test measurement is listed in figure 5. You see that we have a very high resolution in measuring pressure and that it is no problem to measure very short times.

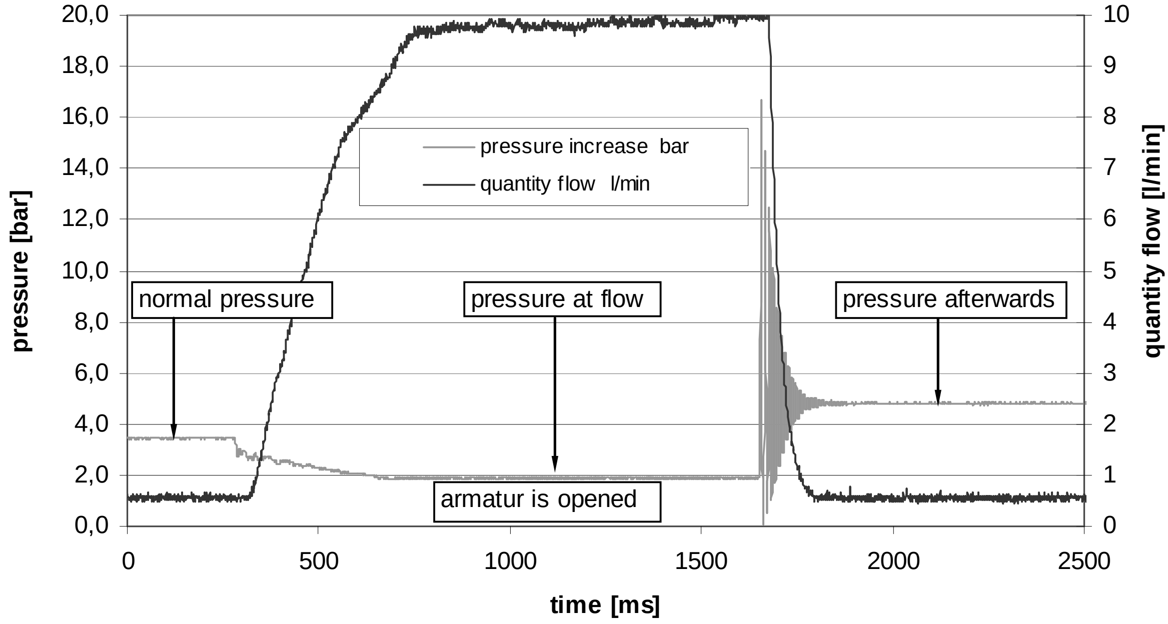

fig. 4: principle measurement result [3]

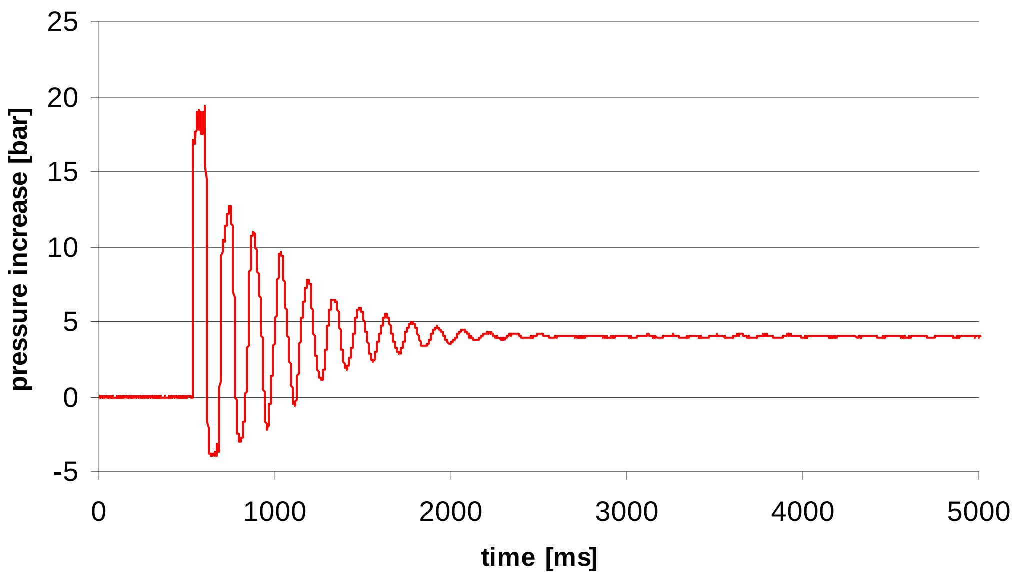

fig. 5: test measurement of a pressure surge [3]

There are some further results which could be observed:

- there are no big differences between two different armatures (different manufacturers)
- a very big difference makes the length of the pipes
- our installation elements had no problems with the pressure surges. The question is if there will rise up problems after longer times.

- longer pipes produce higher pressure surges
- higher normal pressure cause higher flow rates at the armatures. Higher peaks of pressure surge are the consequence.
- the pressure wave has times where the minus peak is near vacuum.

DISCUSSION

Considering the measurements it is not further astonishing that pressure surges cause some noise. Our materials and sanitary products are so good, that defects are seldom. Often pipes´ diameters are reduced to an absolute minimum to avoid hygienic problems. These small pipes with high water velocities force pressure surges.

On the other hand there is much to do in future. Is there the possibility to deaden the surges by constructing special armatures or valves? Is this theme only problem of loud noise, which can be insulated acoustically? Are there new problems by having vacuum at short times?

The thematic is very interesting and the VDI guideline 6006 [2] is a very good help for all civil engineers and experts because – at least in Germany – the occurring pressure surges are becoming more and more. Probably the phenomena will engage us for the next years.

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Effects of water flow regime on water quality in copper and plastic pipes

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SUMMARY

The effects of different flow regimes on water quality and formation of biofilms were studied in a pilot scale drinking water distribution system consisting of two parallel 100 m loops of copper and composite plastic (polyethylene, PE) pipes. The increase in water flow velocity from 0.03 m/s to 0.3 m/s increased significantly the formation of biofilms in both materials. Small changes in water flow disturbed the balance of sediments and biofilms in the pipes, which was seen as an increase in turbidity and the number of bacteria in water. Also, pressure shocks in water flow detached biofilms and sediments increasing the concentrations of metals, bacteria and particles in water. Water stagnation in pipelines increased the concentration of copper in outlet water of copper pipes and microbial numbers in outlet waters of both copper and plastic pipes.

INTRODUCTION

Drinking water distribution system affects microbiological and chemical quality of drinking water. For example, the concentrations of metals and bacteria in water may increase in distribution system. Typically most critical part of the system is household plumbing [1]. Most of the microbial biomass in drinking water distribution system is in biofilms growing on the inner surfaces of water pipes [2, 3]. Several factors, including disinfection, water microbiological quality, temperature, microbial nutrients and water hydraulic regime, affect the growth of biofilms.

With aging of pipes, soft deposits accumulate to the pipelines. These deposits consist of organic and inorganic matter and several metals [3, 4]. Deposits in pipelines originate mainly from water, corrosion products of pipelines, and particles and chemicals are derived from water treatment processes [5]. Changes in water flow can detach sediments and biofilms to drinking water and thus deteriorate the quality of water [4, 6]. This may be seen as an increase in water turbidity, concentrations of metals and bacteria [4, 6].

Typically household plumbing is constructed with copper or plastic pipes. In Kuopio, we have built a pilot scale drinking water distribution system with copper and plastic pipes. With this research environment several research projects have been realized to study the effects of pipeline materials and water flow regime on drinking water quality and formation of biofilms.

METHODS

Pilot distribution system

The pilot scale distribution system consist of two parallel 100 m loops of 10 mm (ID) copper and two parallel 100 m loops of 12 mm (ID) composite (polyethylene-aluminum-polyethylene) plastic tubings (PE). Distribution system was built in a year 2002. Drinking water can be produced in a pilot scale waterworks or taken from the munucipal water distribution system of Kuopio City. In the pilot waterworks lake water is chemically coagulated with ferric sulphate, and further purified using flotation and rapid sand filtration. Finally water hardness, alkalinity and pH were adjusted with lime and carbon dioxide and water was chlorinated with NaOCl (0.5 Cl_2 mg/l) and UV-disinfected (approximately 70 mWs/cm^2). However, when the water reached the pilot distribution system, concentration of active chlorine was decreased below 0.05 mg/l. Purified water was collected into a 4 m^3 stainless steel tank before pumping to the pilot distribution network, which explained the consumption of chlorine.

Effects of flow rate

The effects of water flow velocity on the formation of biofilms and water quality were studied by increasing the flow rate with water circulating pumps gradually from 0.2 l/min to 1.3 l/min corresponding the flow velocities of 0.04 m/s, 0.09 m/s, 0.17 m/s and 0.28 m/s in 10 mm copper pipes and 0.03 m/s, 0.06 m/s, 0.12 m/s and 0.19 m/s in 12 mm PE pipes [7]. With these flow rates the water flow was still laminar (Reynolds numbers were below 3000). Water samples were collected immediately after the increase the water flow rate, and water quality was then followed for three weeks. Biofilm samples were taken after each three week period [7].

Pressure shocks

The effects of pressure shocks on water quality were studied by producing the pressure shocks by compressed air valves placed at the ends of the pipes [8]. The valves halted the water flow for few seconds very rapidly. Before pressure shocks water flow in the pipes was 1 l/min (i.e. flow velocity in copper pipes was 0.21 m/s and in PE pipes 0.15 m/s). Water samples were collected before and after the pressure shocks. First pressure shock was done after 4 weeks of constant water flow (1l/min). Subsequently, pressure shocks were done 2 weeks, 1 week, 4 days, 3 days, 2 days and one day after previous pressure shock. Between the pressure shocks water flow was constant [8].

Water stagnation

The effect of water stagnation on the water quality was studied with experiment where the pilot distribution system was programmed to stagnate water for different times [9]. Stagnation times were 15h 40 min, 4 h, 2h, 1h, 40 min and 40 min. This program was run daily for four months. After each stagnation period, water was pumped for five minutes with the flow rate of 2.6-3.2 l/min (correspond the flow velocities of 0.55-0.68 m/s). Water samples were collected during the flushing and analysed form microbiological and chemical parameters. Water samples were collected nine times during the four month experiment.

Analyses

Concentration of iron was analysed with HACH DR/2010 spectrophotometer (Loveland, Colorado, USA) according to the manufacturers´ instructions. Total copper was analysed using the bicinchoninate method with HACH DR/2010 spectrophotometer. Turbidity was analysed with WTW Turb 555IR turbidimeter. Heterotrophic plate counts (HPC) were analysed with a spread plating method on R2A-agar (Difco). R2A-agar plates were incubated for 7 days at 22 °C before colony counting (cfu). Total numbers of bacteria were analysed with an acridine orange direct counting method. Stained bacteria were enumerated at 1000x magnification under blue excitation using an Olympus BX 51 epifluorescence microscope (Olympus Co. Ltd., Japan) equipped with an ocular grid. The number of particles was analysed with online particle counter (Pamas Water Viewer, WV-IMU, Pamas BmbH, Germany). The particle counter separated the number of particles per milliliter into 8 size classes ranging 1-1.5, 1.5-2, 2-5, 5-8, 8-15, 15-25, 25-50 µm and particles over 50 µm.

The growth of biofilms is analysed with biofilm collectors, which consists of 15 cm (copper) and 20 cm (PE) pieces of pipes. Biofilm collectors were installed in different parts of the pilot system. During sampling, the biofilm collector pipes were closed by ball valves before disconnecting and removed with the water inside the tube. Biofilms inside the collector pipes were removed by shaking with sterile 2 mm glass beads. Detached biofilms were analysed for heterotrophic plate counts, content of ATP and total number of bacteria, and results were normalized to the amount of surface area (cm^2). Total number of bacteria and heterotrophic plate counts were analysed as described above. The content of ATP was measured with a Bio Orbit 1251 luminometer (Turku, Finland) using ATP Biomass Kit HS (BioThema AB, Sweden).

RESULTS

We have found with the pilot distribution system that copper pipes decreased the number of bacteria in water for first six months, but thereafter copper had minor negative effect on the occurrence of bacteria [10]. These experiments also showed that microbial community structure was different in biofilms growing in copper and plastic pipes [10]. New plastic pipes had ability to release phosphorus to water for the first weeks after they were taken in use [10].

Flow rate

Increase in the flow rate of water increased the formation of biofilms in both copper and plastic pipes [7]. In copper and PE pipes the increase in flow velocity from 0.2 l/min to 0.8 l/min increased the total number of bacteria in biofilms on average by 13- and 15- fold, respectively [7]. Content of ATP in biofilms of copper pipes was on average 4 times higher with flow rate of 1.3 l/min than with the flow rate of 0.2 l/min. Corresponding difference with the PE pipes was 7-fold. Increase in the flow rate increased temporary turbidity and number of bacteria in water [7] by detaching biofilms and sediments from the inner walls of pipes. However, although the number of bacteria in biofilms increased with increasing flow rate, the microbial numbers in water did not increase [7].

Pressure shocks

Pressure shocks detached biofilms and sediments. This was seen as elevated concentrations of iron, copper, particles, bacteria and turbidity in water [8]. Immediately after pressure shocks the concentration of iron was on average 2.5 times higher than before pressure shocks in both pipeline materials (Fig. 1). In copper pipes the concentration of copper in water was on average two times higher immediately after the pressure shock than before pressure shock [8]. Time between the pressure shocks did not affect the amount of metals released to water. The numbers of HPC were in copper pipes 19 times higher and in PE pipes 15 times higher immediately after pressure shocks than before pressure shocks (Fig. 2).

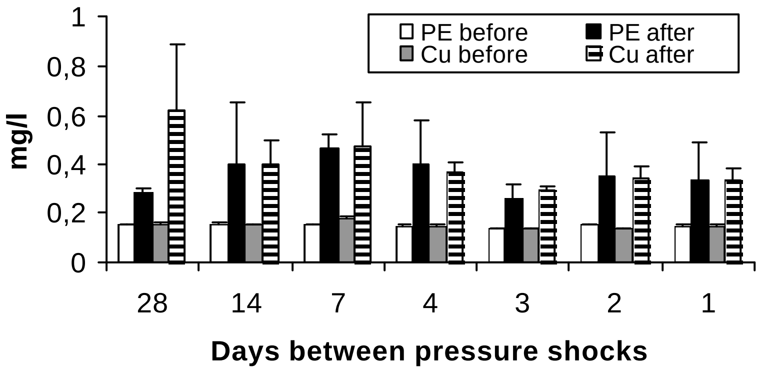

Figure 1. The effects of pressure shocks on the concentration of iron in drinking water in copper (Cu) and plastic (PE) pipes.

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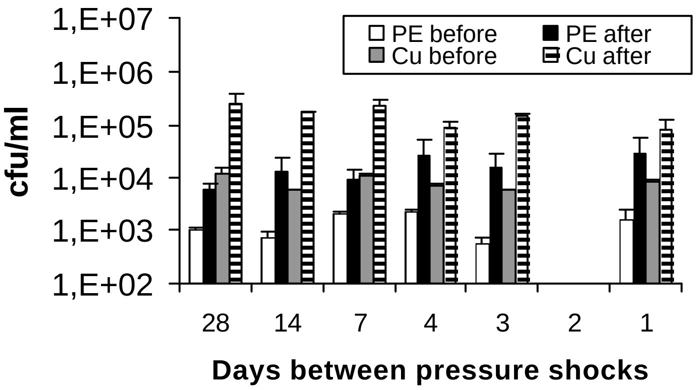

Figure 2. The effects of pressure shocks on the numbers of heterotrophic bacteria in drinking water in copper (Cu) and plastic (PE) pipes.

Water stagnation

Water stagnation in pilot distribution system increased the concentrations of copper and number of bacteria in water [9]. There was only minor increase in the copper concentrations after the stagnation time of four hours (Fig. 3). The number of bacteria (HPC) in water increased linearly up to the longest stagnation time of 15h 40 min (Fig. 4). A regression model showed that the number of bacteria in copper pipes increased on average 4400 cfu/ml/h and in plastic pipes on average 1400 cfu/ml/h [9].

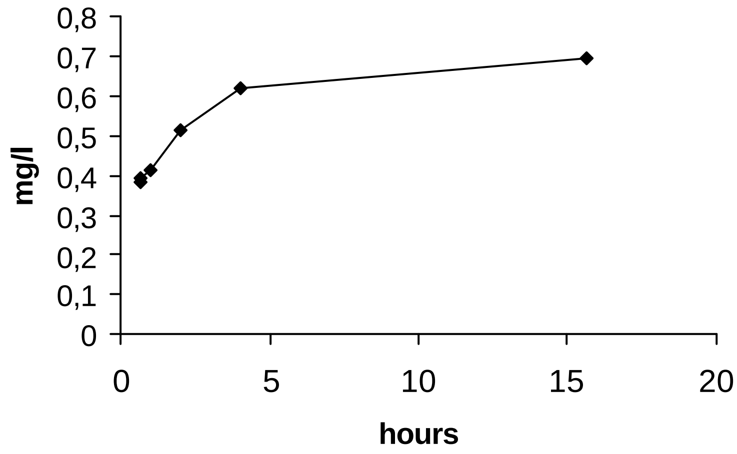

Figure 3. Concentration of copper in water after various stagnation times in copper pipes.

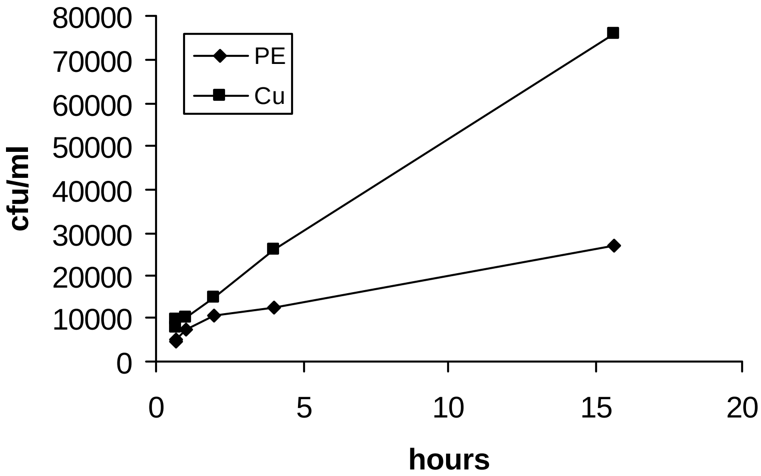

Figure 4. Numbers of heterotrophic bacteria in water after various stagnation times in copper (Cu) and plastic (PE) pipes.

DISCUSSION

The pilot scale drinking water distribution system has shown to be very valuable in studies for the interactions of piping materials, water flow characteristics, biofilms and occurrence of microbes in drinking water. The experiments with this system soon showed that the formation rate of biofilms and their microbial community structure differ between plastic and copper pipes [10].

Our results show that the increase in water flow rate increases number of bacteria in biofilms as found also in some previous studies [11, 12]. The increased biomass and activity in biofilms is also reflected in changes in the consumption of microbial nutrients in water [7]. Increase in the consumption of the nutrients in conjunction with the increased microbial numbers in biofilms are evidence of the enhanced mass transfer of the nutrients in biofilms at the higher flow rates, i.e. the growth of biofilms is mass-transfer limited, as has been proposed previously [12, 13]. Flow rates in our studies have been fairly low. It has been proposed that to remove biofilms and sediments from the pipeline, flow velocities should be higher, like 0.7 - 4 m/s [14, 15].

The balance of sediments and biofilms in pipelines is very sensitive for any disturbances in water flow. When water flow slightly increases, the balance is disturbed and a resuspension of sediments and biofilms occur. This can be seen as elevated concentrations of bacteria and turbidity in water [7]. When stronger pressure shocks are produced with compressed air valves, the number of bacteria increase over ten fold and the concentrations of metal also highly increase. So, biofilms and deposits in pipelines contain not only bacteria, but also copper, iron and microbial nutrients which can be released to water [4, 16].

Iron can be originated from the accumulated coagulation chemicals in the distribution system and copper from Cu-particles from the pipe surface, or be a result of microbially influenced cuprosolvency [17]. The accumulation time of sediments (i.e. time between pressure shocks) does not significantly affect the detached mass of metals or bacteria [8]. This indicates that the system recovers rapidly (less than one day) after the pressure shocks.

The concentration of copper in water is affected by stagnation of water in distribution system [9, 18, 19]. In our case even though stagnation increased the copper concentration in water, it still was clearly below the parametric value of 2 mg/l set by European Union drinking water directive [20], and did not increase significantly after four hours stagnation.

Stagnation of water increases the number of bacteria in water in both copper and plastic pipes [9]. Stagnation times of 16 hours can result to 6-10 times higher numbers of bacteria (HPC) in water than a shorter 40 minutes stagnation. In contrast to the copper concentration, the increase in the concentration of culturable bacteria has been linear [9]. The laboratory experiments have shown that there is only minor microbial growth in water during 16 hours incubation [9]. These results support the earlier conclusions that biofilms are the key sites for microbial growth in drinking water distribution systems and represent the critical source of bacterial biomass to water [2, 21].

To sum up, the studies with the pilot system show that the number of bacteria and metals in drinking water are strongly affected by disturbances in flow regime and stagnation of water in the pipelines. Thus effect of stagnation should be considered carefully when planning a sampling strategy for drinking water microbiological quality control in drinking water distribution systems or designing the piping systems.

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Effect Of The Hydraulic Piping Topology On Energy Demand And Comfort In Buildings With Tabs

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SUMMARY

A research project on the control and design of thermally activated building systems (TABS) has been started in May 2004. This paper presents one selected result after three years of work: Decision guidance and recommendations for HVAC design engineers regarding how to select an energy-efficient hydraulic piping topology for TABS.
The hydraulic piping system in TABS basically can be separated in a heating and cooling generation part, a distribution part and the consumer part which consists of several individually supplied zones. The heating and cooling demand of the zones is different from zone to zone and time-dependent. In this paper, two hydraulic piping topologies commonly used for TABS are analyzed in terms of energy consumption, resulting thermal comfort and understandability:
1) Injection circuits (in all zones) with one common return pipe for all zones
2) Injection circuits (in all zones) with separate return pipes for every zone
The paper presents a comparison between both topologies, using a realistic example, for which the piping topology has a considerable impact on the energy consumption. The example also shows the difference between the two piping topologies on the understandability and that it may have an impact on the comfort.
For the chosen realistic example, hydraulic piping topology 2) delivered better results.

KEYWORDS: HVAC design, hydraulic piping topology, thermally activated building systems (TABS), concrete core conditioning

INTRODUCTION

In this paper, two hydraulic piping topologies commonly used for TABS are analyzed in terms of energy consumption, resulting thermal comfort and understandability:
1) Injection circuits (in all zones) with one common return pipe for all zones (see Figure 1)
2) Injection circuits (in all zones) with separate return pipes for every zone (see Figure 9)
A common return pipe is generally selected for cost considerations.
Today, multiple flow pipes with various supply water temperature levels are often led to the TABS registers. The appropriate temperature levels can be selected (room or register assigned to a zone) with the help of a manual valve per zone during commissioning or changes in room use. In [1],[2] the corresponding implemented plants are described. The paper presents a comparison between the two topologies, using a realistic example, for which the choice of the piping topology has a considerable impact on the energy consumption. The example also shows the difference between the two piping topologies on the understandability and that it may have an impact on the comfort.

The dynamic whole-year simulation for this example will show that the topology with common zone return pipes (topology 1) has a considerable larger energy consumption than the other topology. The simulation will also indicate that the increased energy consumption comes from the fact that the circulated water must be cooled 360 days a year. The result is surprising at first glance and raises the following questions: Why? What is the behavior for a hydraulic topology with separate zone return (topology 2)?
We answer these questions and give the reasons in this paper by analyzing typical stationary load situations and whole-year simulations. We illustrate the different behavior of TABS with common and separate zone return as a matter of principle. Finally, we compare the energy demand of both investigated topologies.
The effect of the hydraulic piping topology on the energy demand and comfort with TABS was examined in the frame of a research project on which was reported for the first time in [3], [4].

1ST HYDRAULIC PIPING TOPOLOGY: COMMON ZONE RETURN

Figure 1 illustrates a typical hydraulic topology from a TABS, consisting of two zones with injection circuits with primary valves and a common zone return flow.

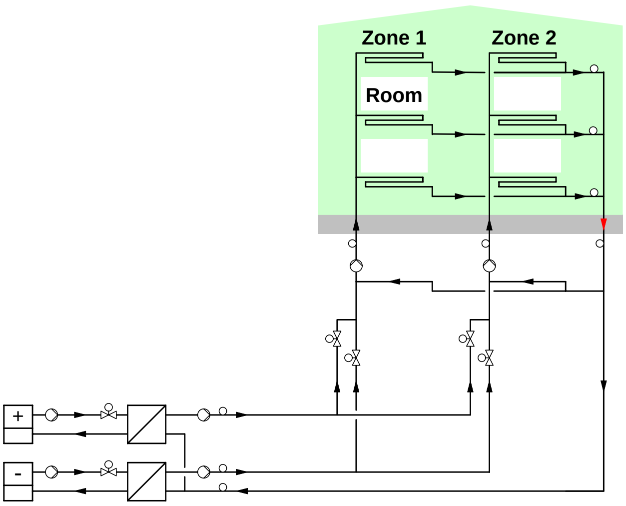

Figure 1: TABS hydraulic piping topology with common return pipe

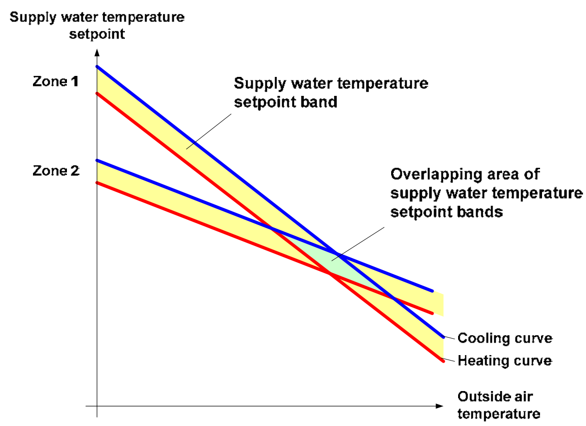

Figure 2: Supply water temperature setpoint bands for a two-zone plant, limited by heating and cooling curves

Supply water temperature setpoints are determined by using heating and cooling curves for each zone as outlined in Figure 2. In general, the curves or the resulting supply water temperature setpoint bands differ by their width and their slopes. Depending on the shape of these supply water temperature setpoint bands, there exists an outside air temperature range where the two supply water temperature setpoint bands overlap.
In [4], [5], [6], [7], an integrated design process for TABS and its control is given based on a so-called “unknown-but-bounded” approach. This approach allows – among other things – to determine adequate supply water setpoint bands.

Analysis of a stationary load situation

The following explains the behavior based on a typical stationary load situation in winter. Figure 3 and Figure 4 illustrate the supply and return water temperatures typically found during the winter: The supply water temperature for zone 1 must be at least 26°C and the temperature for zone 2 may be at most 23°C. The water cools by 0.5K in both zones, in other words heat is provided to the rooms.

At an equal volumetric water flow rate through zones 1 and 2, the common return water temperature corresponds to the mean of the zone return water temperatures, i.e. 24°C. The supply water temperature must be heated to 26°C for zone 1 and cooled to 23°C for zone 2 to achieve the desired supply water temperatures.

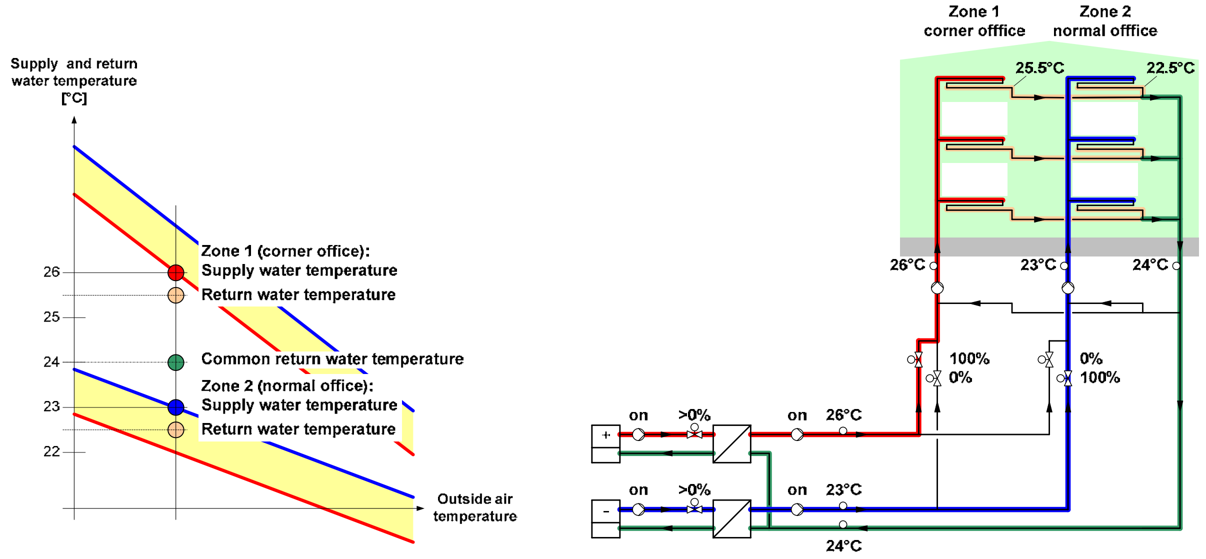

Figure 3 and 4: Temperature ratios and valve settings for common zone return during winter

The common return water temperature is higher than the supply water temperature setpoint for zone 2. The setpoint for zone 2 therefore cannot be achieved by admixing common return water to the supply.

- Although both zones have heat demands, in addition to heating also cooling is required to achieve the desired setpoints.
- If cold generation is blocked for this typical winter situation, the supply water temperature for zone 2 would be too warm and as a result, the room temperatures would exceed the room temperature cooling setpoint, i.e. thermal comfort cannot be achieved.

Analysis of a whole-year simulation

Two equal-sized offices, a corner and a normal office are assumed (see Table 1 and Table 2).

Table 1. data of the simulated zones (cf. [6])

		corner office	normal office
Space length, width, height		6 m x 6 m x 3 m	6 m x 6 m x 3 m
Façade	Orientation	West / South	West
	Area	36.0 m^2	18.0 m^2
	Overall U-Value	0.65 W/m^2K	0.65 W/m^2K
	Glazing fraction façade	42 %	42 %
Additional internal Wall (light)		36 m^2	36 m^2
Natural air change		0.2 h^{-1}	0.1 h^{-1}
Ventilation according to indoor air quality requirements (no cooling/heating by ventilation assumed)			

Table 2. data of the simulated TABS configuration, valid for both offices (cf. [6])

TABS covering fraction (floor area)	80 %
Thickness concrete slab	250 mm
Pipe spacing	200 mm
External/internal pipe diameter	20/15 mm
Specific supply water mass flow rate [a)]	15 kg/(h m^2)
Fictitious TABS thermal resistance (R_t) [a)]	0.08 (m^2 K)/W
Thermal resistance of flooring (carpet)	0.125 (m^2 K)/W

[a)] in terms of floor area covered by tabs

The corner office has twice the ratio of outside walls compared to the normal office. Construction, use and weather conditions correspond to a typical TABS building in the Zurich area (Switzerland). The same simulation set-up as described in [6] was used.

Refer to Figure 5 below, we assumed an outdoor temperature dependent comfort range according to SIA 382/1 [8]. During winter, the comfort range is between 21°C and 24.5°C, during summer between 22°C and 26.5°C. Pursuant to [6], [7], the supply water temperature setpoint band is influenced by the room temperature comfort range, among others: the wider the comfort range, the wider the supply water temperature setpoint band. As an example, refer to simulation results during the summer (Figure 7): the wide room comfort range during summer implies a wide supply water temperature setpoint band.

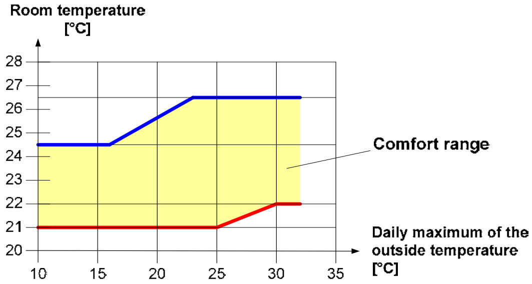

Figure 5: Room temperature comfort range according to Swiss national standard [8]

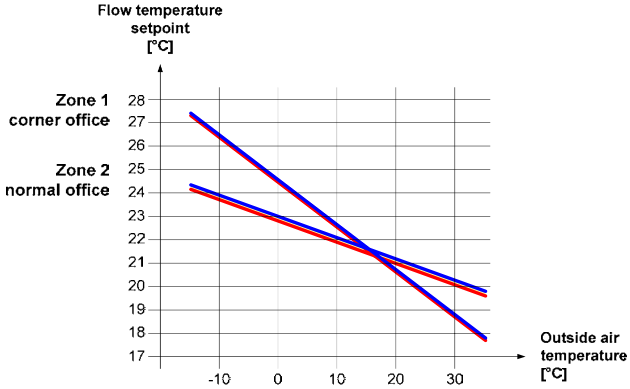

Figure 6: Heating and cooling curves for zone 1 (corner office) and zone 2 (normal office)

For the corner office and the normal office heating and cooling curves according to Figure 6 were determined. Due to the larger fraction of façade heating and cooling demand in corner offices is larger than in normal offices: This results in steeper heating and cooling curves than for normal offices. The heating and cooling curves were defined per the "unknown-but-bounded approach", which is described in [4], [5], [6], [7].
Since the heating and cooling curves are quite different for the two offices, separate zones have to be defined (see Figure 6).

The simulation results are recorded for a whole year, with days 180 to 270 representing summer. Simulation results are shown in Figure 7. Most of the time, the common zone return water temperature (green) is higher than the setpoint band for zone 2. As a result, cooling (magenta, negative value) is needed during most of the year to control the supply water temperature within the setpoint band.

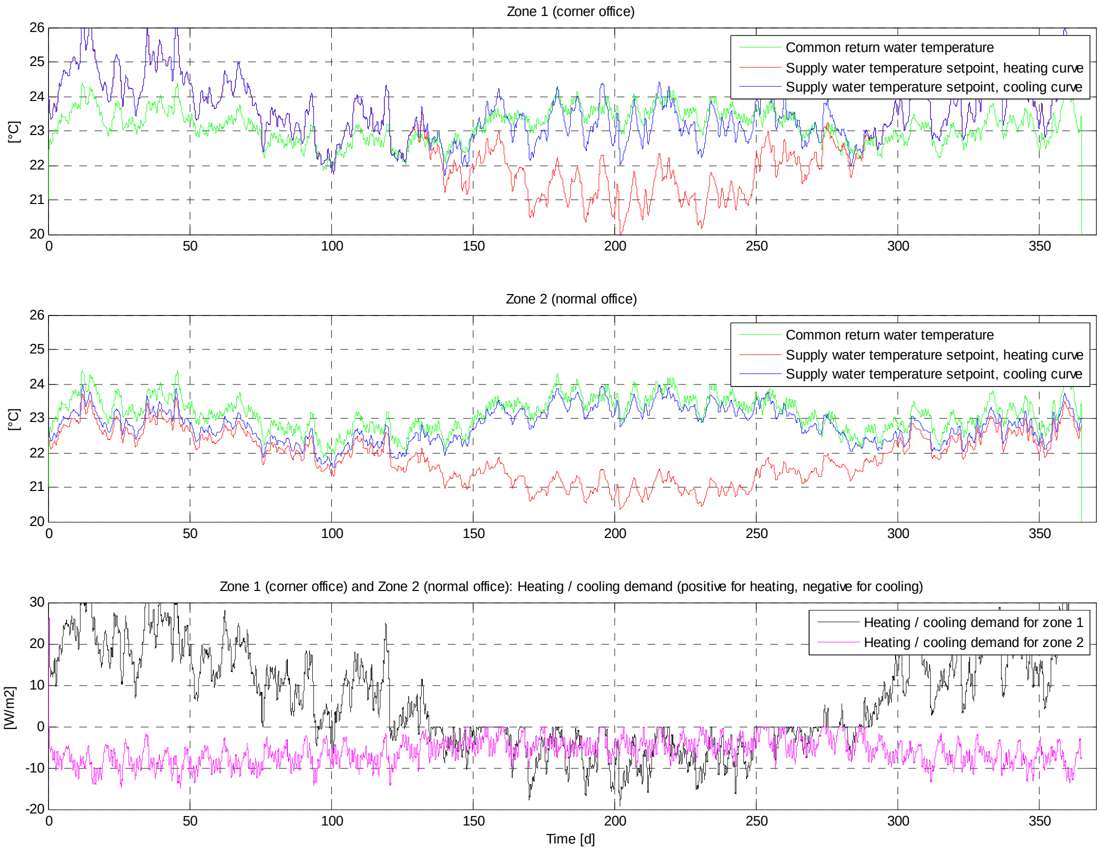

Figure 7: Whole-year simulation results for a TABS with common zone return

2ND HYDRAULIC PIPING TOPOLOGY: SEPARATE ZONE RETURNS

Analysis of a stationary load situation

Figures 8 and 9 illustrate supply and return water temperatures as typically found during the winter: The supply water temperature for zone 1 must be at least 26°C and the temperature for zone 2, 22°C. Water cools off by 0.5K in both zones, i.e. heat is provided to the rooms. The distribution supply water temperature is heated to the highest demanded supply water temperature (i.e. demand-controlled), or per zone 1 to 26°C. For zone 2, cooled off zone return water is mixed to achieve the setpoint (refer to Figure 9).

The setpoint for zone 2 can be achieved by mixing the distributed hot water with its own zone return water.

- Requires only heat to cover zone demand.
- Cold generation can be blocked for this typical winter situation.

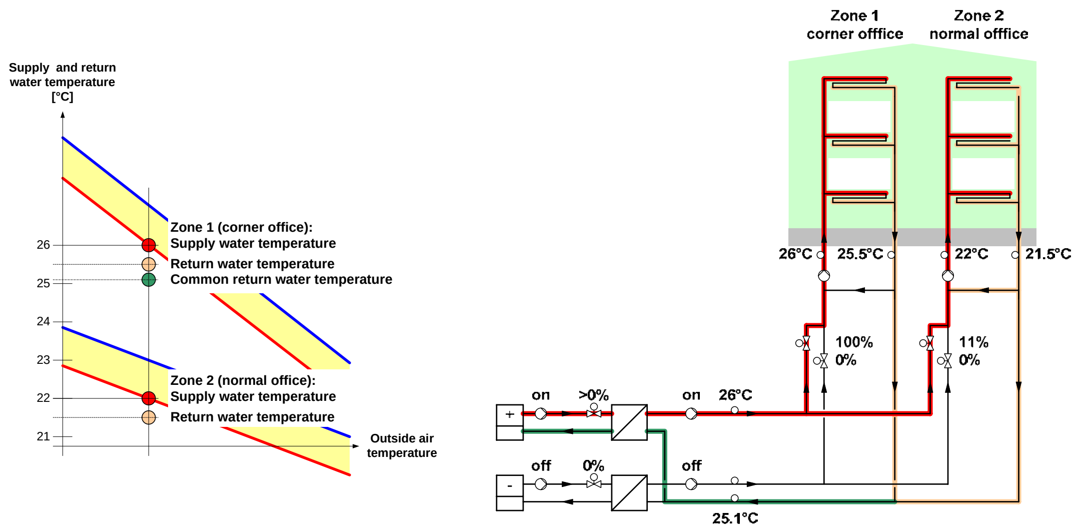

Figure 8 and 9: Temperature ratios and valve settings for separate zone returns during winter

Analysis of a whole-year simulation

In Figure 10, simulation results are presented when using the 2nd hydraulic piping topology. Here, the same simulation set-up was used as for the simulation with the 1st hydraulic piping topology (see Figure 7).

The simulation confirms the expectations: cooling is in winter only required during few days (magenta, positive value).

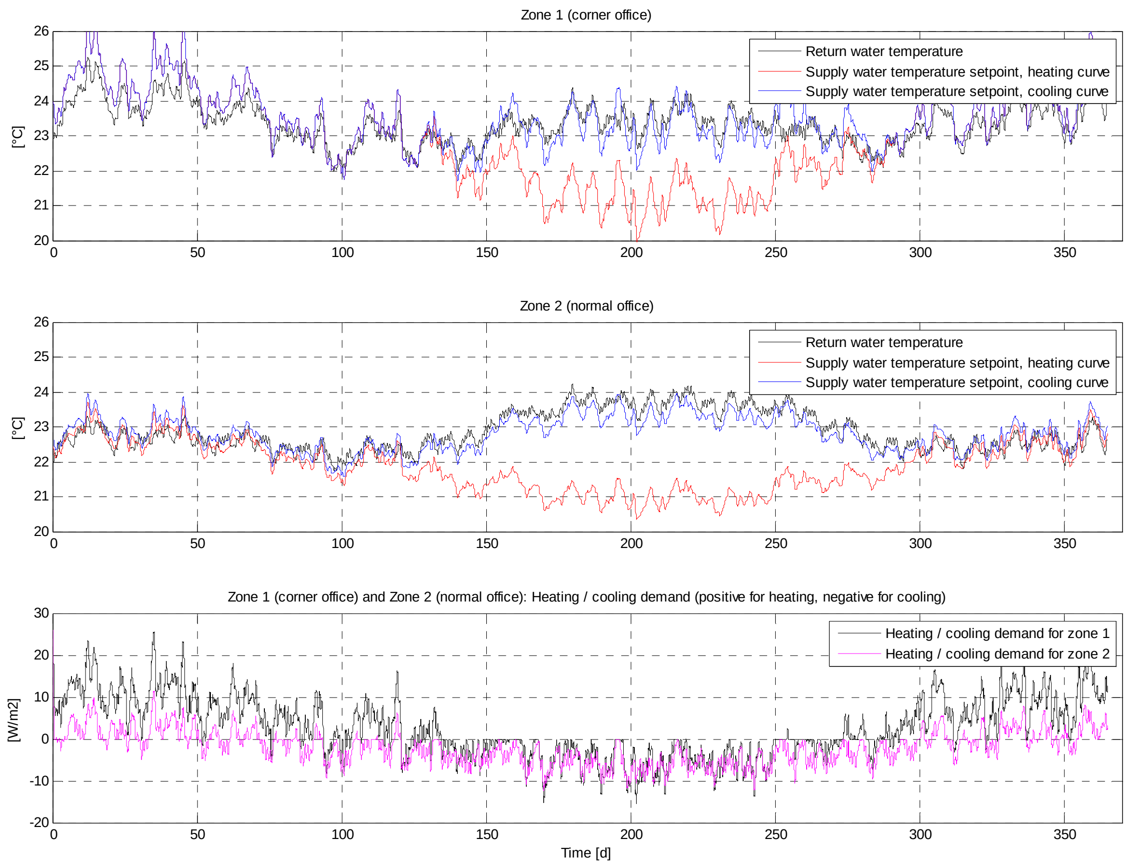

Figure 10: Whole-year simulation results for a TABS with separate zone returns

DISCUSSION

We found the following for hydraulic topologies with common zone return:
Decisive for current heating and cooling generation demand is not the TABS demand (what the room absorbs or is extracted from the room), but rather the actual position of the common zone return water temperature in relation to the supply water temperature setpoint bands of the zones:

1. If it lies between the zone supply water temperature setpoint bands pursuant to Figure 11 (cf. also Figure 2), cold water is required to achieve the zone supply water temperature setpoints below the common return water temperature and hot water above it. In other words, heating and cooling demand for the zones is not directly dependent on the TABS demand and therefore difficult to interpret. If, for example cold generation is incorrectly blocked for a typical winter situation (expected heat demand only), the supply water temperature setpoints below the common zone return water temperature can no longer be achieved. As a consequence, rooms in this zone may be too warm and the anticipated room comfort may no longer be maintained.
2. If it lies within the zone supply water temperature setpoint bands pursuant to Figure 11, a heat exchange may take place between the zones.

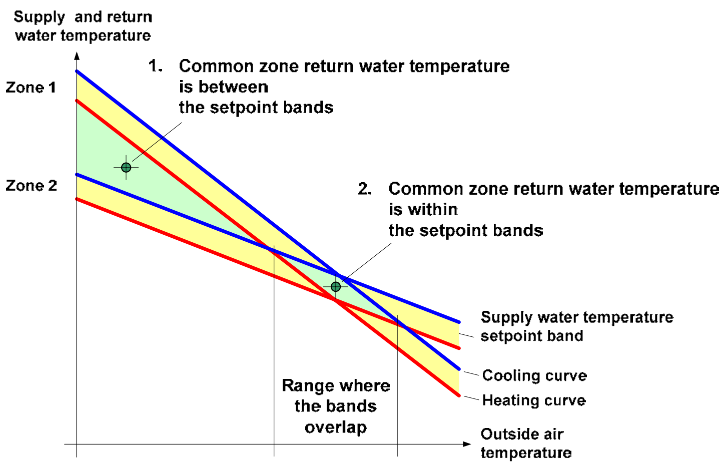

Figure 11: Common zone return water temperature between the supply water temperature setpoint bands (1.) and within the supply water temperature setpoint bands (2.)

The whole-year simulation resulted in the following for generation demand:

- 28 % less heat demand and 30% less cooling demand per year were required for the hydraulic piping topology with separate zone returns compared to the topology with a common zone return!

Depending on the shape of the supply water setpoint bands, an outside air temperature range results where the supply water setpoint bands overlap. The longer the real outside air temperature does stay outside of this outside air temperature range (over the year), the larger the potential energy savings by using a piping topology with separate zone return pipes compared to a piping topology with a common return pipe. If the potential is really exploited then depends on the load situations of the zones and the sizes (floor space) of the different zones (see [1]).
If the common zone return water temperature lies not within the zone supply water setpoint bands during a major part of time, we recommend separate zone return pipes based on current findings. This does result in higher investment costs, but lowers energy consumption,

increases understanding of operational behavior and improves room comfort. If separate zone return pips are installed and multiple flow pipes with various temperature levels are led to each register, (manual) valves for supply and return water have to be installed and they have to be switched so that supply and return are connected to the same zone.
As the present investigation is based on the specific assumptions concerning building characteristics, heat gain situation, proportion between normal and corner offices etc., the presented results strictly speaking are valid for the chosen example only. For situations with strongly deviating parameters results and conclusions may be different.
The presented results in this paper can be transferred to other systems than TABS and also to systems with different heat and cold transfer medium like air conditioning systems. Whether or not different hydraulic topologies provide additional benefits is the subject of further examinations for this research project.

ACKNOWLEDGEMENT

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The New Three Pipe System: Presentation And Evaluation of its Applications in HVAC Technology

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SUMMARY

The purpose of the present article is the presentation and evaluation of the new three pipe system which composes a new methodology and style of building hydronic networks in HVAC technology. The basic characteristics of the new system are as follow :

1. Applicable to all hydronic HVAC networks
2. Innovative and patent pending
3. Obtains rational energy use and saving
4. Simply, stable, easily planned and constructed
5. Experimentally and practically proved

The presentation and evaluation of the new system will take place through the presentation and the analysis of a 3 step procedure of building and optimizing a heat hydronic network. The standards of optimizing are set to a very high level, so that a network will be produced, characterized by simplicity, stability and rational energy use during its function. It will be proved that the new three pipe system is able to obtain such high energy and functional standards based only on its nature, form and hydraulic behavior.

INTRODUCTION

In the hydronic network building there are many styles and methodologies applied, each one with different results in positive and negative features imputed to the network.
The constant primary flow network obtains good authority and functionality of the control valves, negligible interference in the network but it involves energy waste and adaptation inability between produced, distributed and consumed heat power.
The variable primary network on the other hand, generally presents a good adaptation of produced and consumed heat power but it involves interactive phenomena which decrease the stability and functionality of the whole network and leads to energy waste and low efficiency. Furthermore the use of more than one production units with the aim to fit the production power to the variable consumed needs, is deterrent in efficiency and success due to the many instabilities which appear by the use of primary – secondary network, like reverse flows, interactivity and control complexity. The proposed new three pipe system is an effective combined form of the variable and constant primary flow network styles, keeping their benefits and minimizing their drawbacks.
The system in its primary form, presented in fig. 1, composes of two loops, the primary and the secondary. The primary loop is normally the heat production and distribution loop, where as the secondary is the consumption one. The terminal units are installed across the distribution pipes of the secondary loop, each one with a local circulator. A central circulator is also installed in the main pipes to compensate for friction losses in them.

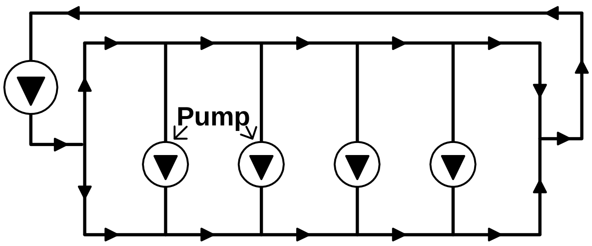

Fig. 1 New three pipe form

The new and innovative feature of the system is that the terminal units function between two central supplies. This has the effect that the initial pressure at the beginning and at the end of each terminal unit pipe is exactly the same so leading to negligible interference between them. The advantages of the proposed system are many and significant. The new three pipe system is applicable in every hydronic HVAC network from small and simple to large and complex, showing rationality in energy use, simplicity and stability in heat control, negligible interference and significant functional stability.

NETWORK ANALYZING & RESULTS

Initial data of the network :
The aim is the planning of a network with 7 consumers (terminal units) with a heat power overall of 248 KW.
The heat power choice of each terminal unit was chosen at random. A random choice was also the determination for the total resistance coefficient of each terminal pipe but in each case adapting to reality.

Table 1. Initial data

Consumer	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**Total**
Power (KW)	25	32	40	20	38	50	43	248
ΔT (K)	15	15	15	15	15	15	15	
Supply (m^3/h)	1.43	1.83	2.29	1.15	2.18	2.86	2.46	14.2
C (Resist. Coeff.)	10270	6870	4770	14370	5050	3300	4300	
ΔP (Pa)	21000	23000	25000	19000	24000	27000	26000	

A production unit of 260 KW heat power was chosen with a prospective ΔT=15.74 K.
The heat power in the terminal units is proportionally adjusted, though in the production units it is constant and the units function in on-off mode.
The mathematically solution of the system takes place based on the theory of the loops and branches, where $\Delta P = C * V^2$, V (m^3/h), H (Pa), C (Total resistance coefficient of the branch).
All the networks under investigation and comparison are at the same length for comparison clarity and efficiency.

The rules to be obtained by the optimized network are the follows :

Rule 1 : The distribution flow is adapted to the terminal units flow.
Rule 2 : The proper flow supply is always obtained by the minimum head of the circulators.
Rule 3 : The produced heat power is adapted to the consumed one.
Rule 4 : Maintenance of equality between production and consumption supply temperature and the proper direction of flows without reverse flow.
Rule 5 : Interactivity between the terminal units is less than 5 %.
Rule 6 : The production units are controlled on their activation by rationality and simplicity.
Rule 7 : Maintenance of constant flow supply in the production units.

First approximation of planning the network

The simplest form of the network applying the new three pipe form is shown in fig. 2.

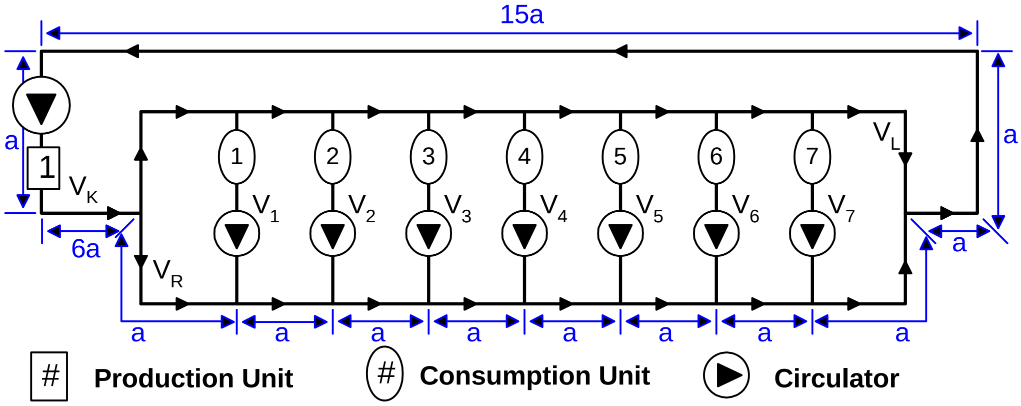

Fig. 2 First stage 3pipe network

Where a=4.5, the total resistance coefficient of the pipe parts. The heat power of the production unit <1> is assumed to be 260 KW.
The emitted heat power through the terminal units has been adjusted via the control of the speed of the local circulators and the totally distributed flow supply in the network is adjusted via the control of the speed of the central circulator.
The characteristic curves of the circulators are :

- Type A : $H=n^2*37708-n*8497*V-781*V^2$
- Type B : $H=n^2*49772-n*8604*V-755*V^2$
- Type C : $H=n^2*56309-n*3786*V-595*V^2$
- Type D : $H=n^2*59995-n*10252*V-543*V^2$
- Type E : $H=n^2*56351+n*670*V-172*V^2$

V (m^3/h), H (Pa), n (Speed coefficient 0 - 1)

Table 2. Determination of the circulator types

Consumer	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**Central**
Circulator Type	A	B	D	A	B	C	D	E

Table 3. Solving the system

	V_1	V_2	V_3	V_4	V_5	V_6	V_7	V_K	V_R	V_L
Flow (m^3/h)	1.43	1.83	2.29	1.15	2.18	2.86	2.46	14.94	0.74	0
	n_1	n_2	n_3	n_4	n_5	n_6	n_7	n_K		
RPM Coefficient	0.97	0.91	0.93	0.90	0.98	0.87	0.95	0.99		
	V_1	V_2	V_3	V_4	V_5	V_6	V_7	V_K	V_R	V_L
Flow (m^3/h)	0.47	0.61	0.76	0.38	2.18	2.86	2.46	10.50	0.01	0.77
	n_1	n_2	n_3	n_4	n_5	n_6	n_7	n_K		
RPM Coefficient	0.34	0.28	0.30	0.30	0.96	0.86	0.94	0.69		
	V_1	V_2	V_3	V_4	V_5	V_6	V_7	V_K	V_R	V_L
Flow (m^3/h)	0.47	0.61	0.76	0.38	0.72	0.95	2.46	6.70	0.03	0.32
	n_1	n_2	n_3	n_4	n_5	n_6	n_7	n_K		
RPM Coefficient	0.32	0.29	0.30	0.29	0.32	0.29	0.94	0.44		

Evaluation

The system and its equations is solved 3 times, see table 3. The first solution refers to full power emission through the terminal units. The second solution refers to partially power emission of the leading four terminal units and exactly at 1/3 of their nominal value. The third solution refers to partially power emission of the leading six terminal units and exactly at 1/3 of their nominal value. The criterion of adjusting the speed of the central circulator is the approach of zero flows in the passive bypasses, V_R, V_L, procedure which takes place via flow switches.
On the preset rules of optimization, the results are as follows :
(<**S**> : Satisfy, <**NS**> : Not satisfy)

Rule 1 : <S> The distribution flow is adapted to the terminal units flow. The surplus V_R, V_L is negligible.
Rule 2 : <S> The proper flow supply is always obtained by the minimum head of the circulators.
Rule 3 : <NS> The produced heat power is adapted to the consumed one.
Rule 4 : <S> Keeping maintenance of equality between production and consumption temperature and proper direction maintenance in flows, without reverse flows.
Rule 5 : <S> Interactivity between the terminal units is less than 5 %.
Rule 6 : <-> The production units are controlled on their activation by rationality and simplicity.
Rule 7 : <NS> Keeping maintenance of constant flow supply in the production units.

<u>Second approximation of planning the network</u>

In this form (fig. 3) an added separate three pipe cluster for the production is formed.

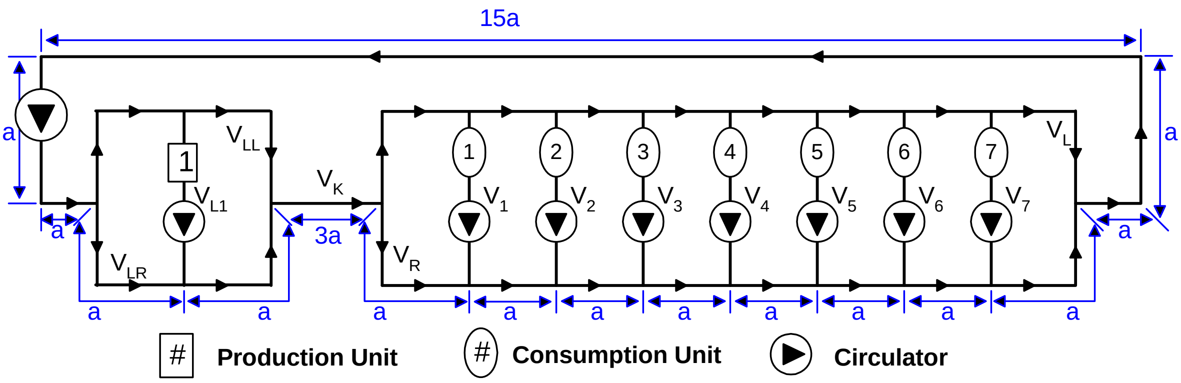

Fig. 3 **Second stage 3pipe network**

The local circulator of the production unit <1> is of type <E> and is determined to function with constant speed by a RPM coefficient of n=0.81. As concerning the rest of the circulators, table 2 is valid.
The criterion of adjusting the speed of the central circulator is the approach of zero flows in the passive bypasses in the consumption cluster, V_R, V_L.
Through this procedure of planning the network, a constant flow supply in the production unit is obtained, an essential fact for cooling applications. However a variable flow supply through the production unit, if allowed by the manufacturer, could be also obtained by the use of variable speed production unit circulator.

Table 4. Solving the system

	V_1	V_2	V_3	V_4	V_5	V_6	V_7	V_K	V_{L1}	V_{LR}	V_{LL}	V_R	V_L
Flow (m^3/h)	1.43	1.83	2.29	1.15	2.18	2.86	2.46	14.95	14.02	0.46	0.46	0.74	0.01
	n_1	n_2	n_3	n_4	n_5	n_6	n_7	n_K					
RPM Coefficient	0.97	0.91	0.93	0.90	0.98	0.87	0.95	0.99					
	V_1	V_2	V_3	V_4	V_5	V_6	V_7	V_K	V_{L1}	V_{LR}	V_{LL}	V_R	V_L
Flow (m^3/h)	0.47	0.61	0.76	0.38	2.18	2.86	2.46	10.62	14.07	-1.72	-1.73	0.04	0.86
	n_1	n_2	n_3	n_4	n_5	n_6	n_7	n_K					
RPM Coefficient	0.34	0.28	0.30	0.30	0.96	0.86	0.94	0.70					
	V_1	V_2	V_3	V_4	V_5	V_6	V_7	V_K	V_{L1}	V_{LR}	V_{LL}	V_R	V_L
Flow (m^3/h)	0.47	0.61	0.76	0.38	0.72	0.95	2.46	6.91	14.12	-3.60	-3.61	0.01	0.55
	n_1	n_2	n_3	n_4	n_5	n_6	n_7	n_K					
RPM Coefficient	0.32	0.29	0.30	0.29	0.32	0.29	0.94	0.46					

Evaluation

The system is solved again 3 times, see table 4. The first solution refers to full power emission through the terminal units. The second solution refers to partially power emission of the leading four terminal units and exactly at 1/3 of their nominal value. The third solution refers to partially power emission of the leading six terminal units and exactly at 1/3 of their nominal value.

On the preset rules of optimization, the out coming results are as follows :
Rule 1 : <S> The distribution flow is adapted to the terminal units flow. The surplus V_R, V_L is negligible.
Rule 2 : <S> The proper flow supply is always obtained by the minimum head of the circulators.
Rule 3 : <NS> The produced heat power is adapted to the consumed one.
Rule 4 : <S> Keeping maintenance of equality between production and consumption temperature and proper direction maintenance in flows, without reverse flow. The reverse flows in the production cluster (V_{LR} , V_{LL}) do not affect the supply temperature.
Rule 5 : <S> Interactivity between the terminal units is less than 5 %.
Rule 6 : <-> The production units are controlled on their activation by rationality and simplicity.
Rule 7 : <S> Keeping maintenance of constant flow supply in the production units.

Third approximation of planning the network

In this form (fig. 4) three production units are installed in the production cluster.

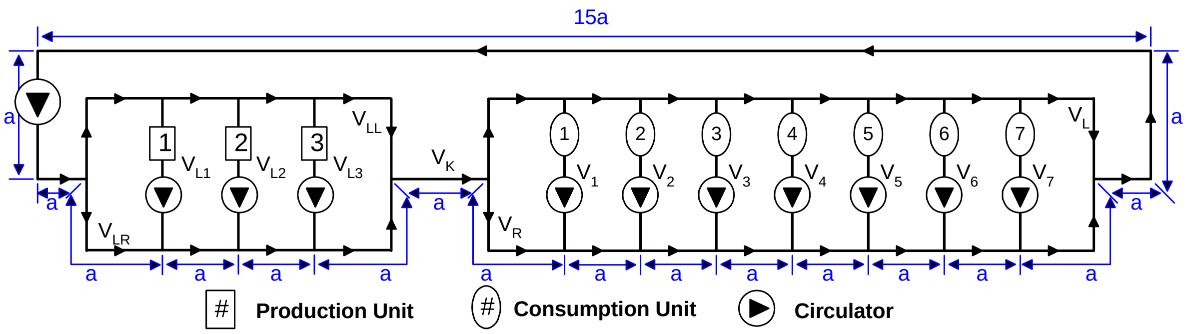

Fig. 4 Third stage 3pipe network

There will be used 3 production units, of total heat power 260 KW, in order to obtain adaptation between produced and consumed heat power in the network.

Table 5. Production unit data

Production Unit	**1**	**2**	**3**	**Total**
Heat power (KW)	60	100	100	260
ΔT (K)	15	15	15	
Flow supply (m^3/h)	3.44	5.73	5.73	14.90
C (resistance coefficient)	1000	400	400	
ΔP (Pa)	11833	13133	13133	
Circulator type	D	C	C	
RPM coefficient (n)	0.92	0.98	0.98	

All the production unit local circulators function with constant speed, determined by the relevant RPM coefficient as in table 5.
The emitted heat power through the terminal units has been adjusted via the control of the speed of the local circulators and the totally distributed flow supply in the network is adjusted via the control of the speed of the central circulator.
The criterion for adjusting the speed of the central circulator is the approach of zero flows in the passive bypasses in the consumption cluster, V_R, V_L, a procedure which takes place via two flow switches.
To continue, two points should be specified, firstly how the variable emitted heat power is determined and secondly by which procedure the production units will be activated.
Many and different procedures are proposed and applied for this purpose, as for instance through the measurements of flow supply temperature, temperature differences, return temperature etc, procedures with different results in success, accuracy, complexity and rational energy use. By the use of the new three pipe system, the proposed procedure of determining the emitted heat power is simple and new. It should be used only the already known RPM coefficient of the central circulator, which corresponds to the flow supply distributed in the network and which is equal to the flow supply through the terminal units, in fact analog the emitted heat power.

Table 6. Production unit activation rules

RPM Coefficient (%)	**0 – 20**	**20 – 35**	**35 – 60**	**60 – 75**	**75 – 100**
Production unit	1	2	1 + 2	2 + 3	1 + 2 + 3

Table 7. Solving the system

Active units (1+2+3)	**V_1**	**V_2**	**V_3**	**V_4**	**V_5**	**V_6**	**V_7**	**V_K**	**V_{L1}**	**V_{L2}**	**V_{L3}**	**V_{LR}**	**V_{LL}**	**V_R**	**V_L**
Flow (m^3/h)	1.43	1.83	2.29	1.15	2.18	2.86	2.46	15.13	3.40	5.65	5.67	0.78	-0.37	0.84	0.15
	n_1	**n_2**	**n_3**	**n_4**	**n_5**	**n_6**	**n_7**	**n_K**							
RPM Coefficient	0.97	0.91	0.93	0.90	0.98	0.87	0.95	0.99							
Active units (1+2+3)	**V_1**	**V_2**	**V_3**	**V_4**	**V_5**	**V_6**	**V_7**	**V_K**	**V_{L1}**	**V_{L2}**	**V_{L3}**	**V_{LR}**	**V_{LL}**	**V_R**	**V_L**
Flow (m^3/h)	0.47	0.61	0.76	0.38	2.18	2.86	2.46	10.42	3.41	5.68	5.69	-1.61	-2.75	0.01	0.69
	n_1	**n_2**	**n_3**	**n_4**	**n_5**	**n_6**	**n_7**	**n_K**							
RPM Coefficient	0.34	0.28	0.30	0.30	0.96	0.86	0.94	0.68							
Active units (2+3)	**V_1**	**V_2**	**V_3**	**V_4**	**V_5**	**V_6**	**V_7**	**V_K**	**V_{L1}**	**V_{L2}**	**V_{L3}**	**V_{LR}**	**V_{LL}**	**V_R**	**V_L**
Flow (m^3/h)	0.47	0.61	0.76	0.38	2.18	2.86	2.46	10.45	0	5.69	5.70	0.96	-1.90	0.03	0.70
	n_1	**n_2**	**n_3**	**n_4**	**n_5**	**N_6**	**n_7**	**n_K**							
RPM Coefficient	0.34	0.28	0.30	0.30	0.96	0.86	0.94	0.68							
Active units (1+2+3)	**V_1**	**V_2**	**V_3**	**V_4**	**V_5**	**V_6**	**V_7**	**V_K**	**V_{L1}**	**V_{L2}**	**V_{L3}**	**V_{LR}**	**V_{LL}**	**V_R**	**V_L**
Flow (m^3/h)	0.47	0.61	0.76	0.38	0.72	0.95	2.46	6.81	3.42	5.70	5.70	-3.44	-4.57	0.02	0.44
	n_1	**n_2**	**n_3**	**n_4**	**n_5**	**n_6**	**n_7**	**n_K**							
RPM Coefficient	0.32	0.29	0.30	0.29	0.32	0.29	0.94	0.45							
Active units (1+2)	**V_1**	**V_2**	**V_3**	**V_4**	**V_5**	**V_6**	**V_7**	**V_K**	**V_{L1}**	**V_{L2}**	**V_{L3}**	**V_{LR}**	**V_{LL}**	**V_R**	**V_L**
Flow (m^3/h)	0.47	0.61	0.76	0.38	0.72	0.95	2.46	6.89	3.43	5.71	0	-1.98	-0.27	0.08	0.46
	n_1	**n_2**	**n_3**	**n_4**	**n_5**	**n_6**	**n_7**	**n_K**							
RPM Coefficient	0.32	0.29	0.30	0.29	0.32	0.29	0.94	0.45							

Evaluation

The system is solved again 3 times, see table 7. The first solution refers to full power emission through the terminal units, where all 3 production units function. The second solution refers to partially power emission of the leading four terminal units and exactly at 1/3 of their nominal value. This solution procedure is made in two steps, firstly while all production units are functional, the RPM coefficient of the central circulator is determined and secondly the system is solved again maintaining only the proper production units functional, which in this case are the <2> & <3>, while n=0.68.
The third solution refers to partially power emission of the leading six terminal units and exactly at 1/3 of their nominal value. This solution procedure is made again in two steps, firstly all production units are functional and the RPM coefficient of the central circulator is determined and secondly the system is solved again maintaining only the proper production units functional, which in this case are the <1> & <2>, while n=0.45.
During the whole procedure of defining the active production units, it was determined in the background, that the relationship between the RPM coefficient and the distributed flow supply is linear proportional, a valid assumption in most of the cases. However this relationship could be set more precisely in each particular application.

On the preset optimization rules, the out coming results are as follows :
Rule 1 : <S> The distribution flow is adapted to the terminal units flow. The surplus V_R, V_L is negligible.
Rule 2 : <S> The proper flow supply is always obtained by the minimum head of the circulators.
Rule 3 : <S> The produced heat power is adapted to the consumed one.
Rule 4 : <S> Keeping maintenance of equality between production and consumption temperature and proper direction maintenance in flows, without reverse flow.
Rule 5 : <S> Interactivity between the terminal units is less than 5 %.
Rule 6 : <S> The production units are controlled on their activation by rationality and simplicity.
Rule 7 : <S> Keeping maintenance of constant flow supply in the production units.

CONCLUSIONS

The satisfaction of all rules of optimization by the new three pipe system, a normally dimensioned and easily planned and constructed network, opens new scientific and applicative ways in the hydronic HVAC technology.
Actually the simultaneous satisfaction of the above rules by using the known primary – secondary network style would not be feasible [1].
Furthermore the use of the new three pipe system comprises an invitation and challenge to the industry on producing more efficient variable speed circulators and proper flow switches.
For the fulfillment of the presentation, it should be focused on two more points.

1. The adjustment of the emitted power by the terminal units could take place by the use of two or three way control valves, though the local circulators would be of constant speed. In this case the satisfaction of the rule 2 would be partial, concerning only the central circulator.
2. The new three pipe system, by its negligible interference and rational energy use comprises a hydronic solution applicable mainly in constructions based on collectors, where the wide distances of the network are covered by the terminal unit pipes. By its use in wide spread networks, where the wide distances of the network are covered by the central pipes as for instance in district heating or cooling networks, a further financially and hydraulically investigation should be done.

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Assessing Boiler Efficiency

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SUMMARY

Because central heating (CH) boilers represent a considerable amount of the total energy consumption in the European Union (EU), initiatives have been taken by the European commission to phase out boilers with poor efficiency from the market. To fairly judge boilers, efficiency tests carried out by different laboratories have to be conform. Tests carried out in the past between different laboratories on the contrary have showed that this is not the case. In this paper the possible methods that are used to carry out the efficiency test on gas-fired boilers are analysed. Trough uncertainty and sensitivity analysis the factors that have the largest influence on the efficiency results are determined for the different cases.

INTRODUCTION

Even a few percent rise in efficiency of a boiler represent a considerable reduction in energy use due to the number of boilers sold on the EU-market. This knowledge led to a number of European initiatives to phase out boilers with poor efficiency from the market. As a result the efficiency requirements for central heating (CH) boilers have become more stringent. Efficiency requirements in connection with CE-marking of gas-fired boilers are mentioned in the directive 92/42/EEC. Nowadays there are also a few other European directives that are important for CH boilers and their performance. The energy performance of buildings (EPB) directive 2002/91/EEC resulted in many countries in new minimum requirements for boiler efficiency. The present version of directive 2005/32/EEC on the eco-design of the energy-using products (EuP) obliges the labeling of central heating (CH) boilers in eight categories. This situation has increased the importance of a high measurement-quality.

The European standards for gas-fired boilers [1] [2] [3] are frequently used by the laboratories which are certified to measure the efficiency of boilers e.g. when performing CE-label testing. But the standards permit different interpretations and give the laboratories a large degree of freedom. Looking at the requirements for the uncertainty of the efficiency measurement in the standards, only large limits for the individual measurement equipment of the input parameters and a maximum absolute uncertainty in the efficiency measurement of 2% is stated. So it is left to the laboratories to fix the measurement uncertainties in the measurement of the input so that the desired uncertainty on the efficiency is achieved. Also, too little details on the preferred measuring method and equations to be used to calculate the efficiency are recorded in the standards. Laboratories are free to choose which method and equations are used to determine the efficiency of the boiler. Also little is mentioned on the boiler stability during the measurements. This is nevertheless an indispensable requirement for a correct result.

A fair judgment of the CH boilers can only be achieved when the measured efficiency of different laboratories on the same boiler agrees and if the efficiency measurement is of a high quality. If not the risk of wrongful labeling of boilers is unacceptably high. This is a recent issue, seeing the only discrimination that was previously made was between boilers who did and did not satisfy the requirements for CE-marking. The new European directive divides the CH boilers into 8 label categories. This will lead to more discussions as each category covers a small range of the efficiency (typically 3 till 4%). Differences in the measured efficiency between different laboratories can lead to a different classification of the boiler.

In 1991 efficiency measurements were carried out on the same boiler by different laboratories [4]. This study led to the following conclusions: a reproducibility of 4% for the full load efficiency test and 6% for at the 30% part load test. The reproducibility (R) is the value below which the absolute difference between two single test results obtained with the same method on identical test material may be expected to lie within a specified probability of 95% under different conditions like apparatus, operators and laboratories. These observed differences are considerable when compared to the small ranges proposed for the labeling of boilers.

A number of laboratories have already joined forces in a network named LABNET. The aim of this collaboration is to investigate the problems that occur when testing CH boilers and in particular the non conformity of the obtained efficiency results. A document (Good Laboratory Practice or GLP) [5] detailing the appropriate testing rules, was established based on the investigations carried out by and the experience of the partners in the project. The most significant proposals to improve the quality of the efficiency measurements stated in the GLP are: stability criteria to verify the stability of the boiler, corrections for ambient conditions such as ambient temperature and atmospheric pressure and which equations to use for the determination of the total uncertainty of the efficiency. The impact of the GLP has been verified by again comparing the efficiency results of different laboratories [4] [6]. The results showed clear improvements: a reproducibility of 2.5% for the full load test and 5.9% for the 30% part load test.

At part load the results are still not satisfactory. The remaining difference may be attributed to differences in measuring instruments and the methods that can be used to carry out the efficiency test. The most difficult part in determining the total uncertainty of the efficiency measurement is determining the separate uncertainty factors of the measured input parameters. Research done by LABNET has shown that not all the laboratories have a clear idea about these uncertainties. A number of laboratories underestimate the measurement uncertainty due to insufficient knowledge of the different uncertainty sources. A first step to the improvement of the quality of measurement is to have the laboratories realize the importance of the uncertainty in their measurements. Sometimes the procedures are not sufficient, e.g. a calibration frequency that is too low. Taking into account that the standards and GLP are not obligatory, larger differences can be expected between the different testing laboratories, as not all participated in the comparisons.

BOILER EFFICIENCY

To execute the efficiency measurement the boiler is connected to the primary side of a heat exchanger (Figure 1). Some laboratories measure the output capacity of the boiler by measuring water mass flow rate (Q_{wp}) and water temperatures before and after the heat exchanger (*Tp1* and *Tp2*) in the primary water circuit of the heat exchanger. The efficiency (η) can then be expressed as:

$$\eta = \frac{Q_{wp}.(h_{p1} - h_{p2})}{Q_g.Hi}, \qquad (1)$$

where h_{p1} and h_{p2} are the water enthalpy before and after the heat exchanger, Q_g is the gas mass flow rate and Hi is the gross calorific value of the gas used. Other laboratories measure the output capacity of the boiler by measuring water mass flow rate (Q_{ws}) and water temperatures before and after the heat exchanger (*Ts1* and *Ts2*) in the secondary water circuit of the heat exchanger and taking into account the extra heat losses in the secondary circuit and heat exchanger (*L*). The efficiency is calculated with:

$$\eta = \frac{Q_{wp}.(h_{s2} - h_{s1}) + L}{Q_g.Hi}, \qquad (2)$$

where h_{s1} and h_{s2} are the water enthalpy before and after the heat exchanger. The goal of this paper is to examine the main differences between these two methods and to determine which method gives the highest quality of measurement. The main input factors that have an influence on the variation of the efficiency have to be determined.

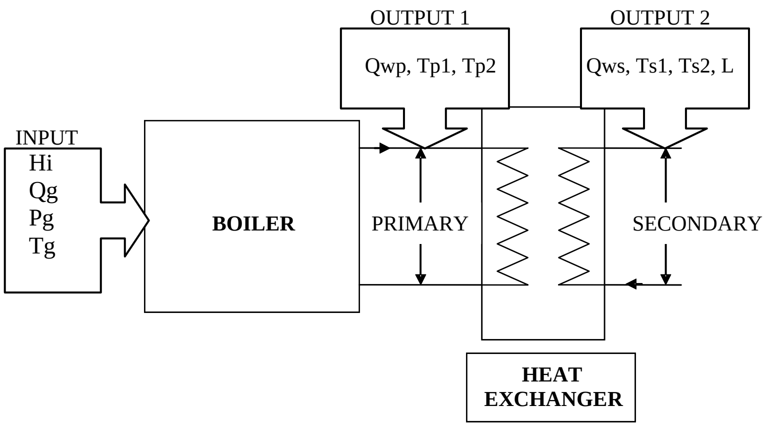

Figure 1. Different measuring methods

The main reason for laboratories to measure in the secondary water circuit is a reduced impact of the uncertainty of the water temperature measurement because of the larger temperature difference before and after the heat exchanger. As this uncertainty is an important factor in the total uncertainty, it leads to a reduced total uncertainty on the efficiency. This is especially important for the part load test because these tests have the smallest temperature difference (for part load typically in primary water circuit: 7°C, for the secondary water circuit: 20°C). This study is restricted to gas-fired CH boilers up to 70kW because firstly most of the tests in laboratories are carried out on such boilers and secondly because the biggest differences in efficiency measurements occur for relatively small boilers.

METHOD AND RESULTS

First of all the uncertainty analysis [4] of two different laboratories is compared to each other. Laboratory 1 performs measurements in the primary water circuit while laboratory 2 measures in the secondary water circuit. The different measuring instruments and procedures used are taken into account. The first laboratory calibrates the temperature difference between the two water temperature measuring probes and this calibration is carried out before and after the efficiency test on the boiler. The second laboratory calibrates the water temperature probes separately and with a smaller calibration frequency. For both laboratories four cases are

considered in order to investigate the possible differences between the laboratories and the proposed used methods:
1) measuring on the primary water circuit and separate calibration of the water temperature probes
2) measuring on the primary water circuit and calibration of the water temperature difference
3) measuring on the secondary water circuit and calibration of the water temperature probes separately
4) measuring on the secondary water circuit and calibration of the water temperature difference

During the analysis a normal distribution is assumed for the input parameters and a level of confidence of 95% is taken. The uncertainty analysis is carried out for both the laboratories for a measurement on a boiler of 28kW. The results are presented in Figure 2. Table 1 gives an idea of the part (in % of the total uncertainty on the output) of each important input parameter in the uncertainty of the efficiency.

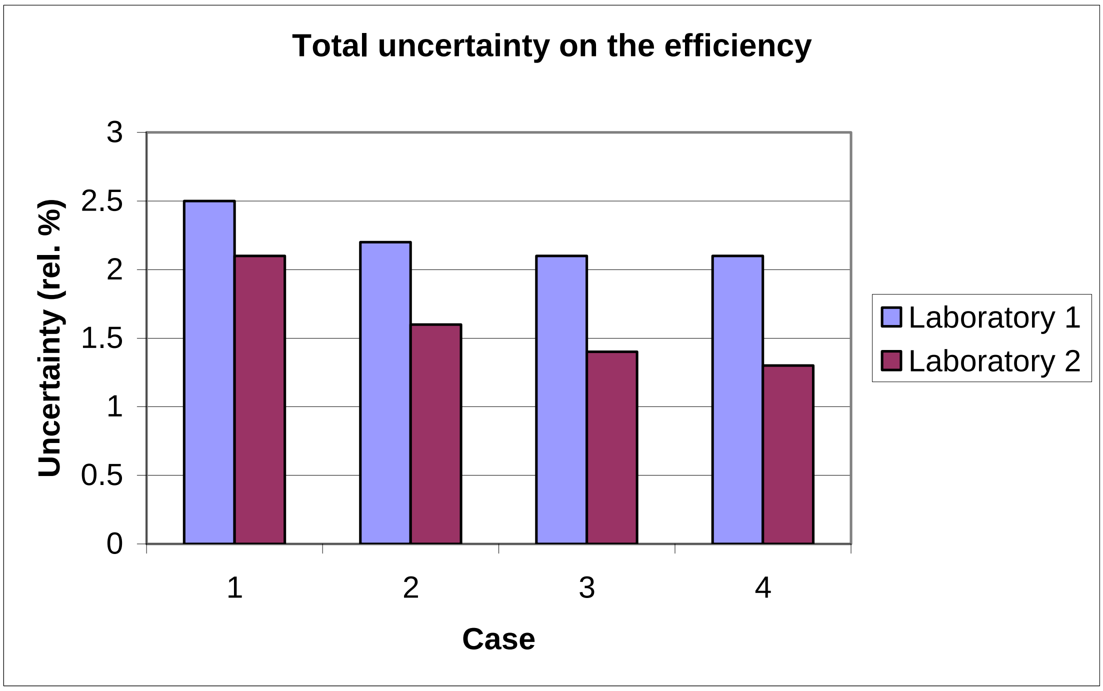

Figure 2. Results of the uncertainty analysis: total uncertainty on the efficiency

Looking at the same method (Figure 2) there is a clear difference in the total uncertainty results obtained by the two laboratories. The most important reasons for this are the uncertainty of the calorific value (Hi) and of the gas pressure (Pg) (as can be seen from Table 1). The uncertainty on the calorific value of the gas used is large for the first laboratory because of the use of natural gas for the efficiency test while the second laboratory uses gas from bottles (99.5% CH_4). In the first laboratory the calorific value of the natural gas is measured with a chromatograph a few times during the measurement. An uncertainty factor because of the unincorporated variation of the calorific value during the test has to be taken into account and also the error due to calibration of the chromatograph is considerable. This is not the case for the second laboratory which only has to take into account the uncertainty stated by the certificate of the bottle. The uncertainty on the gas pressure has to be taken into account for the first laboratory. The gas flow is measured with a volumetric flow meter so a correction has to be applied to convert the flow rate from measurement conditions (real gas temperature and gas pressure) to reference conditions (0°C and 1013.25 mbar). The most important reason for the large impact of the uncertainty of the gas pressure is the drift between calibrations. The second laboratory uses a thermal mass flow meter and no correction has to be applied. This results in a smaller total uncertainty.

Case	Input parameters	Laboratory 1	Laboratory 2
Case 1	Qg	14.55%	17.66%
	Hi	27.52%	2.08%
	Pg	17.31%	0.00%
	Qw	12.23%	9.33%
	Tp1	13.43%	36.11%
	Tp2	12.91%	34.22%
Case 2	Qg	17.75%	44.29%
	Hi	33.57%	6.95%
	Pg	14.92%	0.00%
	Qw	21.11%	23.41%
	ΔT	9.69%	20.83%
Case 3	Qg	19.67%	54.24%
	Hi	37.21%	6.39%
	Pg	23.41%	0.00%
	Qw	15.91%	27.58%
	Ts1	1.57%	5.19%
	Ts2	1.57%	5.19%
Case 4	Qg	20.11	59.02%
	Hi	38.04%	6.95%
	Pg	23.93%	0.00%
	Qw	16.27%	30.01%
	ΔT	0.98%	2.49%

Table 1. Part of the important input parameters in the total uncertainty of the efficiency

Looking at the results of all the cases for one laboratory (for example laboratory 2), it can be seen that the total uncertainty improves both by calibrating the temperature difference between the probes (case 2) or by measuring on the secondary water circuit (case 3) opposite to case 1. A large improvement compared to case 1 is obtained by both the laboratories when measuring in the secondary circuit using separately calibrated temperature probes (case 3). Calibrating the temperature difference between the probes (case 4) offers no significant improvement compared to case 3. The differences between the observed cases are smaller for the first laboratory because of the smaller contribution of the water temperatures in the total uncertainty on the efficiency (table 1).

Comparing the methods that are presently used by the laboratories, measuring in the secondary water circuit clearly offers a better result. This is the result of the differences mentioned in the sections above. It can be concluded that the best way to work is according to case 3. Due to the continuing improvement of boiler efficiency and better insulation of buildings, boilers with a smaller capacity will appear on the market. For these boilers the efficiency measurement at part load can not be done on the secondary circuit due to the small secondary water flow. Measuring on the primary side (case 2) then provides a solution to reduce the uncertainty value. Further reduction of the uncertainty of the efficiency is only possible through careful study of the input factors.

In the next part of the paper a sensitivity analysis is carried out for a condensing boiler of 24kW input capacity. The mathematical model is based on the general equations to calculate boiler efficiency taking into account the corrections for ambient conditions mentioned in the GLP. A distinction is made between measuring on the primary and the secondary water circuit. Also the cases of full load or part load are studied. For the sensitivity analysis we use a variance based method, the method of Sobol [8]. The sensitivity parameters used to investigate the model are:

$$S_i = \frac{V[E(Y \mid X_i)]}{V(Y)}$$

$$S_{Ti} = \frac{E[V(Y \mid X_{-i})]}{V(Y)} \qquad (3)$$

The first parameter S_i represents the first order sensitivity index. $V(E(Y \mid X_i))$ is the variance of the expected value Y, if we were able to learn the true value X_i. If S_i agrees with a high value, the corresponding input parameter X_i is very important for the total uncertainty of the efficiency. The second parameter S_{Ti} represents the total order sensitivity index. $E(V(Y \mid X_{-i}))$ is the expected output variance that would remain unexplained if only X_i were left free to vary over its uncertainty range. Orthogonal input parameters are supposed to simplify the determination of the conclusions. When the total index is zero, the input parameter has no influence on the total uncertainty.

The accepted variations of the input parameters are set to the allowed variations mentioned in the standards or to values based on user experience in testing of gas-fired boilers.

In figure 3 the five most important first order sensitivity parameters are represented for the different cases studied. In all the cases the water temperatures measured to calculate the corresponding enthalpies are by far the most important. When measuring on the secondary water circuit the importance of the water temperatures is decreased as expected. Also the gas pressure has an influence on the efficiency and this influence is higher when measuring at the secondary circuit. The importance of the parameters used in the corrections mentioned in the GLP is rather small. The large value of the sensitivity parameter for Tp2 when measuring on the primary circuit at part load can be explained by the knowledge that the possible variation of Tp1 allowed by the standards is smaller at part load than at full load while the variation of Tp2 is left at the larger value.

In figure 4 the most important second order sensitivity parameters are represented. It is clear that these parameters are much smaller than the sensitivity parameters of the first order. Comparing the appearing interactions, it can be seen that the second order parameters corresponding to the water temperatures appears more when measuring at part load, this because the influence of the water temperatures is larger at part load than at full load. The sensitivity parameters corresponding with the gas temperature and water temperatures are also important for most of the cases.

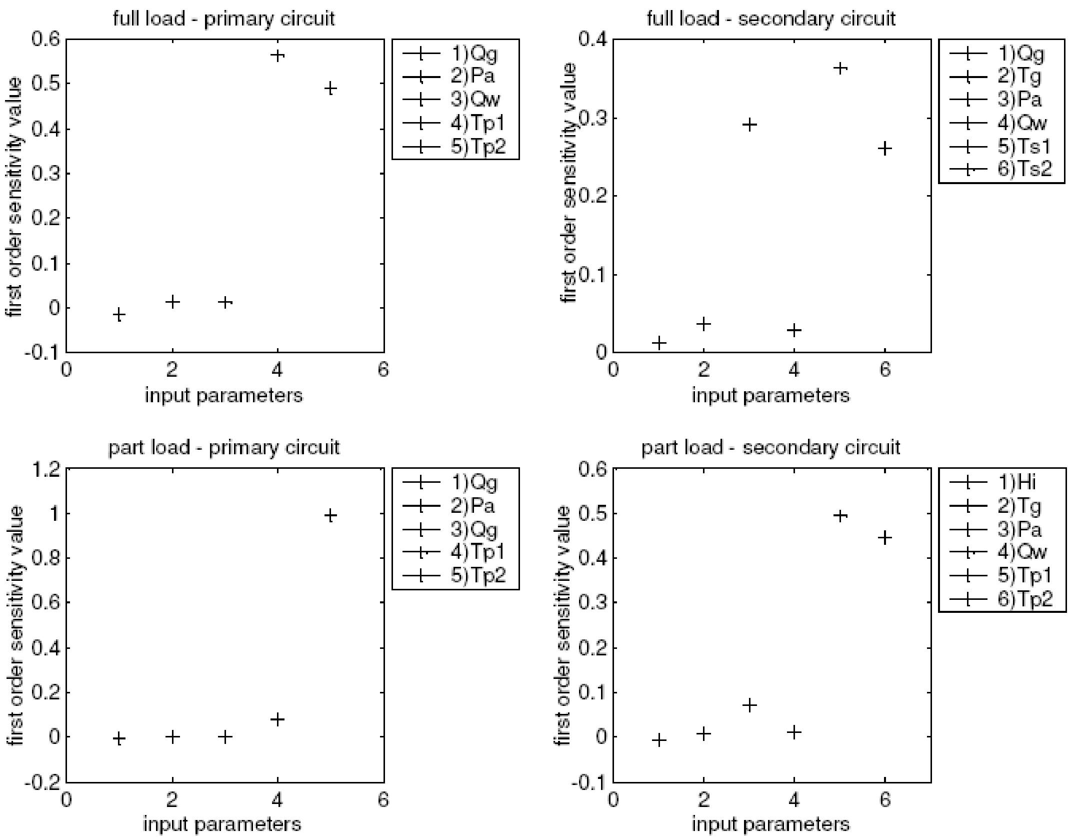

Figure 3. Most important first order sensitivity factors

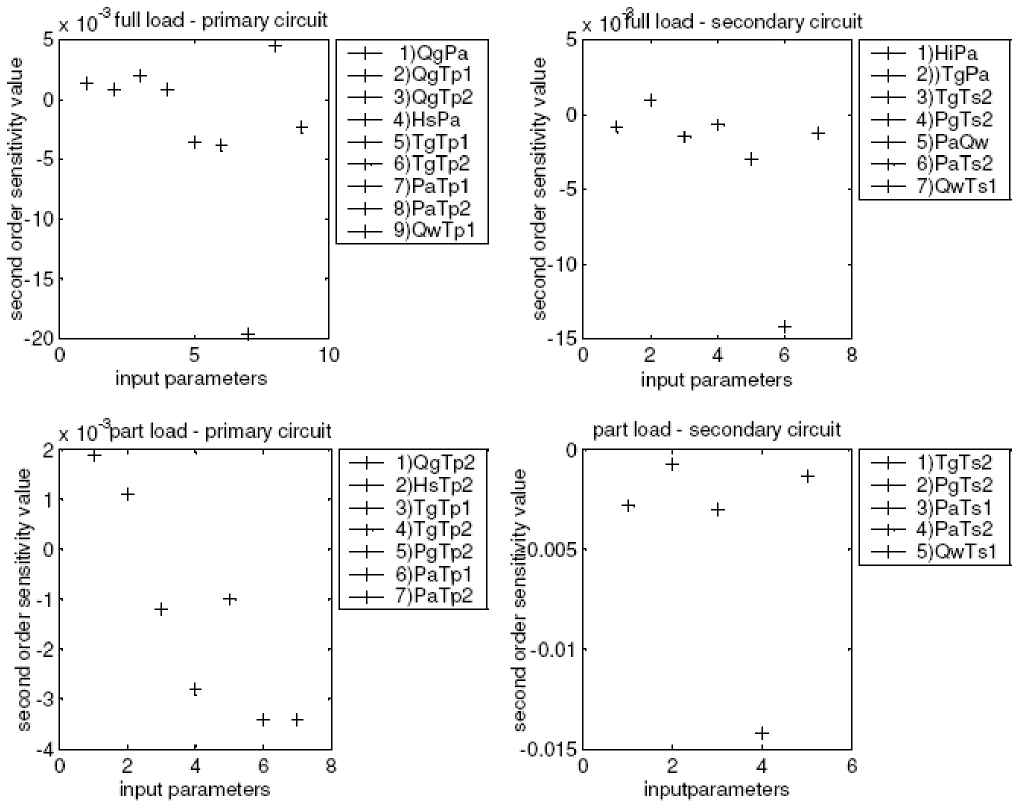

Figure 4. Most important second order sensitivity parameters

CONCLUSIONS

Out of the uncertainty analysis it can be concluded that five input parameters are important for the uncertainty in the efficiency measurement: water flow, calorific value, water temperatures and gas pressure. The measurement on the secondary water circuit with separate calibration of the water temperature probes seems to be the highest quality measurement. Measuring on the primary circuit with calibration of the temperature difference can be a good substitute when the boiler has a small capacity.

The sensitivity analysis of the efficiency measurement showed that especially the water temperatures are an important factor in the variation of the efficiency result. Also the atmospheric pressure and water flow have a non negligible effect on the efficiency.

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Interactions of Network Materials And Drinking Water

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SUMMARY

The aim of this project was to study 1) the materials in contact with drinking water 2) the quality of drinking water in Finland and 3) the interactions of materials and water. The study was based on scientific literature, reports and surveys. The main materials in distribution networks in Finland are polyethylenes, cement mortar, and polyvinylchloride. In water supply systems of buildings the main materials used are copper, brass, galvanised steel and polyethylenes. Drinking water quality in Finland is high in respect of the Drinking Water Directive (98/83/EC) and national legislation. However, especially groundwater in Finland, has low pH and are soft, so it is aggressive for materials.

INTRODUCTION

The quality of drinking water is dependent on many factors *i.e.* raw water (ground water vs. surface water), water purification technology and distribution systems. Drinking water flows days or even weeks in the distribution network before arriving to consumer's tap. Drinking water and materials in contact with it are in a complex interaction, which can result in deterioration of water quality and/or the deterioration of materials (Figure 1). The interactions between water and the materials in distribution systems in contact with drinking water are influenced *f.e.* by the quality of water, the characteristics of the materials, microbial activity in biofilms and corrosion. The main basic mechanisms in interaction of drinking water and materials are l) leaching of substances from materials 2) corrosion and 3) microbial actions in biofilms. Increasing temperature increases reaction rates and microbial activity. Also increasing flow rate can raise the corrosion rates or increase microbial activity.

Higher concentrations of substances leach from brand new materials, whereas the leaching gets slower when time passes by. All materials can leach out substances under unfavourable circumstances. Leached substances can affect the aesthetic or health related quality of water and microbial growth.

Corrosion is an electrochemical reaction, where the metal is oxidized and the metal ions migrate in to the water. In uniform corrosion, metal corrodes evenly, but in localized corrosion only in small areas. Amounts of metal ions that may have an effect on human health can only result in uniform corrosion. Biocorrosion is a form of corrosion where microbes themselves corrode materials or create circumstances where corrosion is likely to occur [1].

Biofilms are formed on surfaces of materials in distribution systems and water supply systems in buildings because of microbial activity. Most of the microbes in biofilms are not dangerous to human health. On the other hand, pathogenic bacteria are protected in the biofilms against disinfection chemicals. Biofilms can also cause technical and aesthetic problems in drinking water quality. [2]

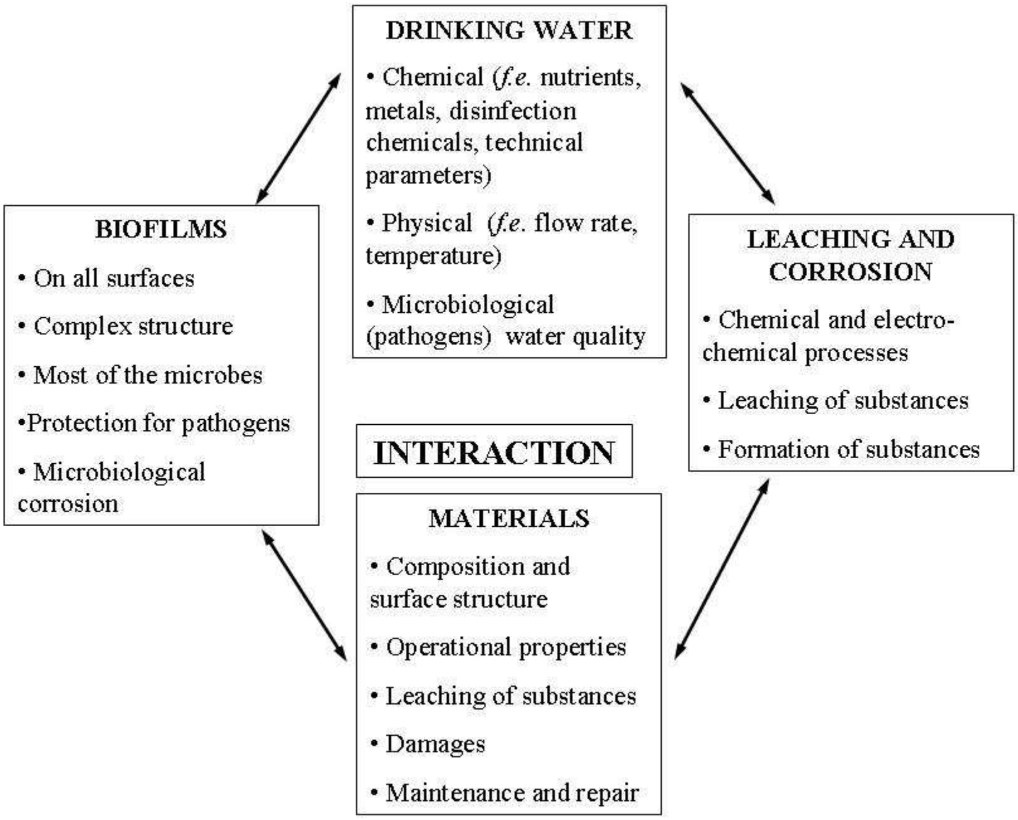

Figure 1. Interactions of drinking water and materials in contact with the water.

Information on materials and the quality of drinking water is needed in the preparatory work of the forthcoming mandatory European acceptance system for products that are in contact with drinking water, EAS (European Acceptance Scheme). The aim of EAS is to promote safer products for human health in drinking water distribution.

The aim of the studies was to gain information on a) the materials in contact with drinking water and b) the quality of drinking water in Finland. This information is needed in understanding the interactions between materials and drinking water.

METHODS

Materials in contact with drinking water in Finland

The information sources for the study on materials and their properties used in Finnish water distribution systems and water supply of buildings were scientific literature, surveys (10 companies and 48 waterworks), interviews and manufacturers'/importers' web pages. The study defines all the materials that are in contact with drinking water in Finland, their structure, composition, manufacture, use, fittings and the interaction phenomena between materials and water. Also materials in hot water systems in buildings were studied.

The waterworks that responded to the survey accounted 56 % of the water supply and 50 % of the population in Finland. Their networks consist 18 % (15 000 km) of the water distribution lines in Finland. [3]

Quality of drinking water in Finland in years 1984-2006

The drinking water quality in Finland was reviewed in the second part of the study. The study is based on published reports, surveys to waterworks (76 in number) and scientific publications containing data mostly from years 1984-2006. [4]

This study presents 72 parameters of which 48 are included in the Finnish legislation of drinking water and Drinking Water Directive (98/83/EC). The other 24 parameters are essential in estimating the technical quality of water and in the preparation work of EAS. The quality information of Finnish drinking water was compared with those obtained from Sweden and Norway.

RESULTS

Materials in contact with drinking water in Finland

The study showed that different products with different materials include pipes, fittings, valves, coatings, gaskets from the starting pipeline at the waterworks to the consumer's tap. Materials used are metallic, plastic, cementitious and rubber materials. The compositions of metallic materials are well known, since they contain metals in certain proportions. Plastics and rubbers compose of polymers and additives. In plastics, the principal group of additives is stabilisers. Also cementitious materials usually contain additives. Substances that can leach from materials altogether are mostly metals, cement or organic/inorganic additives. Nearly all of the materials that have been installed in the past are still being installed and in use. Products and their materials are in contact with drinking water for decades.

Mainly used pipe materials and their proportions (%) in our survey in distribution networks were high density polyethylene (HDPE) 28.6, ductile iron 22.5, polyvinylchloride (PVC) 17.2, grey cast iron 10.4, other plastics 9.6, low density polyethylene (LDPE) 3.2 and asbestos cement 2.8. Also steel, middle density polyethylene (MDPE) and stainless steel are in use. Materials that are no longer installed as pipe materials are grey cast iron and asbestos cement. Nowadays, practically all metallic pipes in new installations are coated, mainly with cement mortar. 15 materials that are being installed in contact with drinking water were identified as pipe, fitting or valve materials.

The main pipe materials in water supply systems in buildings in Finland are copper, galvanised steel and polyethylenes, but exact proportions are not known. The installation of galvanised steel pipes ended in the 1970's and the material was only used in cold water plumbing. Nowadays installed pipe materials are mostly copper, crosslinked polyethylene (PEX) and composites in which the material in contact with drinking water is either PEX or PE-RT (polyethylene of raised temperature resistance) (Table 1). Fittings are usually made out of brass or plastics (*e.g.* polyamide, polyacetal).

Table 1. Materials that are being installed in contact with drinking water in Finnish buildings. Coating and gasket materials are not included.

MATERIAL	**PRODUCT**			
	pipes	**fittings**	**valves**	**other**
Metals				
Galvanised steel				x
Stainless steel	x	x	x	x
Copper	x	x		
Brass		x	x	x
Aluminium				x
Ductile iron				x
Bronze			x	x
Red Brass		x	x	
Plastics				
Crosslinked polyethylene (PEX)	x			
PE-RT	x			
Glass reinforced plastic (GRP)				x
Polyamide (PA)				x
Polyacetal (POM)				x
Polyethene terephtalate (PET)				x
Polycarbonate (PC)				x
Polyphenylene oxide (PPO)				x
Polyphenylsulfone (PPSU)		x		x
Polypropylene (PP)	x	x		x
Styrene acrylonitrile (SAN)				x

There are 12 coating and gasket materials that are being installed altogether both in the distribution networks and in buildings. Usually plastic and always cementitious materials are used as coatings and rubbers mainly as gasket materials. The principal coating materials are cement-mortar (only in distribution networks), bitumen (only in distribution networks), epoxides, polyesters and polytetrafluoroethylene. Bitumen is no longer being installed into contact with drinking water. Coating is mainly used in the distribution networks. Principal gasket materials being installed in water supply systems in buildings are ethylene propylene diene monomer rubber (EPDM), fluoro rubbers, chlorobutyl rubber (CIIR), nitrile rubber (NR) and polytetrafluoroethylene (PTFE).

Quality of drinking water in Finland in years 1984-2006

The study on the quality of Finnish drinking water contained parameters requested in Drinking Water Directive and parameters important for EAS preparatory work. The parameters in the study were presented in main themes: 1) microbes, 2) nutrients, 3) metals, 4) chemicals and organic compounds, 5) non-metals, 6) parameters affecting technical quality of water and 7) radioactive compounds.

The study shows that microbes requested in the Drinking Water Directive have been adequately reported and drinking water is of high hygienic quality.

Results on microbial nutrients showed that concentrations of nitrogen compounds have been below the regulatory limit values. The amount of organic carbon is high in Finland, as the concentration of total organic carbon is generally 2 mg/l or more. The concentrations of phosphorus have not been widely reported, even though phosphorus is the microbial growth limiting factor in Finland.

The metals and non-metals requested in the Drinking Water Directive have been adequately reported and the concentrations have generally been below the regulatory limit values. However, high concentrations of iron and manganese have caused technical and aesthetical problems, which have been more obvious in well water than in water delivered by waterworks. In certain areas, fluoride concentration in water has also exceeded the limit value. Some metals such as molybdenum, zinc and tin are interesting in respect of EAS, but data available is very limited.

The technical variables of drinking water affect the corrosive properties of water and consequently the service life of distribution systems. Variables of this type are pH, alkalinity and hardness and concentrations of chloride and sulphate ions. The study showed that pH values in waterworks have been within regulatory limit values (6.5-9.5), whereas ground waters are more acidic (Figure 2). Concentrations of chloride and sulphate ions have been according to the Drinking Water Directive, and Finnish waters are soft (hardness 0.6 mmol/l). The capacity of water to resist the change of pH, alkalinity, has been low (1 mmol/l). The information available indicates that the quality of drinking water in Sweden and Norway is similar to that in Finland. According to the building regulations, the materials used in water supply systems in buildings should not lessen the quality of water for health. However, there are no specific instructions for water quality that may restrict the use of certain materials.

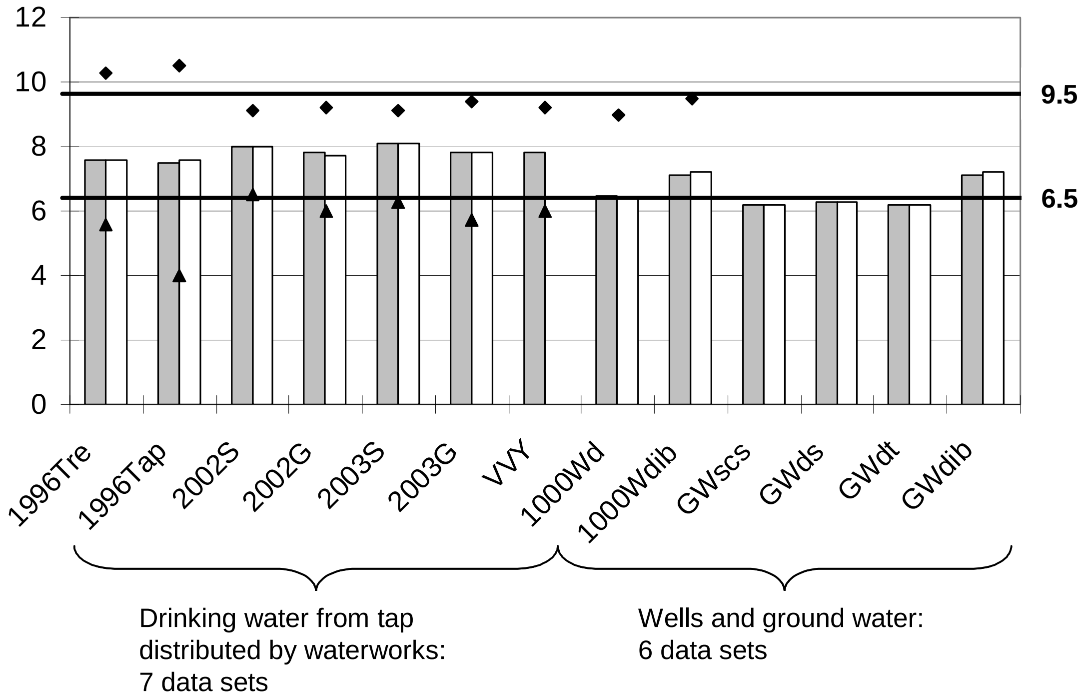

Figure 2. pH values in Finnish drinking waters from waterworks, wells and groundwater. Data originates mostly from years after year 1996. Average (grey column), medium (white column), maximum (♦ mark) and minimum are shown (▲mark). The acceptable regulatory limits are 6.5-9.5.

DISCUSSION

There are numerous different materials used in both distribution network and water supply systems in buildings in Finland. The interactions in the water supply systems in buildings are strongly dependent on the phenomena in the distribution networks. Materials used in Finnish distribution networks and water supply systems in buildings are not likely to have health effects except in some special cases. The area of the material in contact with drinking water is essential. Leaching from pipe materials can be slow, but the amount can be much higher because of the large contact area in pipes. The reactivity of the materials should also be considered. Materials with small area in contact with drinking water can leach out substances or diffuse harmful substances from outside of the system in much higher rates than pipe materials with large surface area. In buildings, the hot water systems can increase the reaction rates compared to cold water lines and the distribution network. Also the surface to volume ratio is considerably larger in the pipelines of buildings compared to distribution network.

Information on drinking water quality with health-related parameters is available and the water quality is high, whereas technical parameters are not taken into account as much. Water quality can differ dramatically within different geographical area of the country, which can influence the service life of materials used in pipes and accessories.

Still the knowledge of lifetime exposure to slow leaching of certain substances is weak. Also risks related to microbial growth and the effects of leaching substances on it need more research. Flushing of installed pipelines is essential not only from a technical aspect, but also from a health related point of view. It is necessary to remove the substances that might have leached from materials and impurities in water.

As a conclusion, the main interactions of materials and drinking water are: leaching of substances from materials, corrosion and microbial activities in biofilms. Further work within interactions of materials and drinking water will contain *e.g.* studies on a) damages of pipe materials and b) less studied water quality parameters. The information is needed whether the Finnish drinking water quality should be enhanced to be less aggressive for the materials used or should the use of materials in case of certain water quality be restricted to avoid the migration of harmful substances to water.

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Seismic Protection of Fire Sprinkler and Other Mechanical Systems: Best Practices from Turkey

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SUMMARY

Earthquakes are non-predictable natural disasters. Nobody can foresee a coming earthquake and escape from its consequences. What can be done is to observe happened earthquakes, collect empirical data and analyze these data for estimating effects of future earthquakes. This is the basis for seismic protection of our living spaces.

Today engineers can design buildings depending on expected seismic forces. However, an unacceptable mistake is to neglect the importance of seismic restraint for non-structural systems. This is crucial especially for fire sprinkler pipes, fuel lines, emergency and energy systems etc. Without having a functioning mechanical and electrical system there will be no fire protection, no energy supply, no communication and no health services.

This paper includes basic information about earthquakes and examples of seismic restraint design and applications for some selected buildings in Turkey; an office, a shopping center and a residence. Followed buildings codes are IBC 2003 and Turkish Code 1997.

INTRODUCTION

The purpose of this paper is to point out the importance of seismic protection for mechanical and electrical systems in buildings and facilities. These systems are the essential bases for a building and/or facility to be functional after and sometimes even during an earthquake. There is no scientific hypothesis in this work. Instead, this work first covers basic information about earthquakes and how they affect our buildings; then shows the empirical calculations of seismic forces and a comparison between the Turkish Local Code and the International Building Code; and finally demonstrates ways to keep our mechanical and electrical systems safe. The ultimate purpose of this work is to support the awareness for importance of seismic protection for non-structural systems in buildings.

Seismic forces can be calculated thru globally accepted building codes. The most widely used building code today is IBC (International Building Code) which has joined the previous ones (UBC, BOCA etc). Except Japan, almost the rest of the world is using IBC when dealing with non-structural seismic design. The reason is that many local codes (including Turkish Code) are technically weak in terms of seismic protection for mechanical and electrical systems. Once the seismic force assumed to act on a mechanical component (piping, equipments etc) is calculated, the next step is to make the necessary precaution. This can be done either by fixing it to the structure in an experienced and accepted way or by restraining it with help of specifically designed and manufactured seismic hardware (snubber, isolator, steel cable, bracket etc). In both cases, it is a must to calculate the forces and design protection system.

EARTHQUAKES AND DAMAGES ON MECHANICAL SYSTEMS

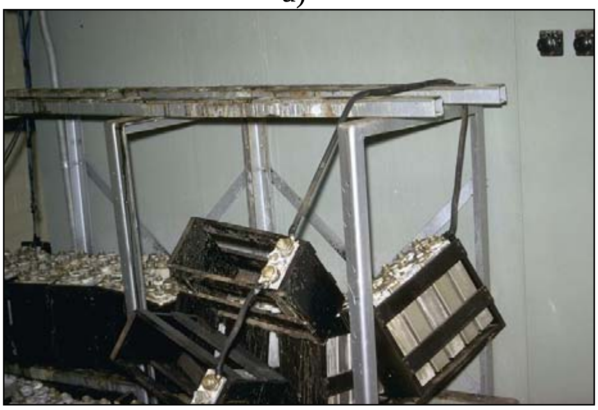
Earthquakes are non-predictable natural disasters caused by three reasons: volcanic action, collapse of an underground hole, or a ground layer motion which is called “tectonic action”. The first 2 are not as critical as the last one, since these happen seldom and affect rather smaller areas. Tectonic earthquakes, on the other hand, are more critical because of their strong and devastating effects on wide areas.

During an earthquake, a building can stand still if it has been constructed properly. But can we use that building after the earthquake? The answer of this question can only be “yes”, unless the building is still functional. This means, a building is useless without electricity, heating/air conditioning, water supply, etc. There is a more important question: How can we save lives during and right after an earthquake? Is it enough to make a building stand still after the seismic activity? What if a small fire has started? Will there be people to stop that fire? Or will we depend on our automatic fire sprinkler system to stop this little fire? What if our sprinkler piping has been damaged by the earthquake? A bunch of people will be trying to escape from the building and that small fire and/or smoke can easily cause deaths.

The above paragraph shows a very possible situation. But if we have a fire sprinkler system, strong enough to resist the seismic forces, we don’t need to be afraid of these kinds of situations. The same is for all mechanical and electrical systems. We have to consider the non-structural aspects when constructing our buildings and facilities. If we don’t, the followings can happen:

a)

b)



c)

d)

Figure 1. Examples of earthquake damages; a) A sheared pipe, b) A broken sprinkler head, c) Collapsed emergency battery racks, d) Ripped body of a chiller by the connected pipe.

BUILDING CODES AND SEISMIC FORCES

Just like the weather forecasts, seismic forces can not be known exactly. Instead, these can be calculated with empirical formulas, based on observations on past earthquakes. A very detailed formula can be found in IBC 2003 (Equation 1),

$$F_p = \frac{0{,}4\, a_p\, S_{DS}\, W_p}{\frac{R_p}{I_p}} \left(1 + 2\frac{z}{h}\right) \qquad (1)$$

where F_p is the design seismic lateral force, a_p is the component amplification factor, S_{DS} is the design spectral response acceleration at short periods, W_p is the component operating weight, R_p is the component response modification factor, I_p is the component importance factor, z is the height in structure at point of attachment of component, and h is the average roof height of structure.

The formula in the local Turkish Code 1997, on the other hand, is much less detailed, thus weak (Equation 2),

$$f_e = w_e A_0 I \left(1 + \frac{H_İ}{H_N}\right) \qquad (2)$$

where f_e is the design seismic lateral force, w_e is the component operating weight, A_0 is the ground acceleration, I is the building importance factor, $H_İ$ is the height in structure at point of attachment of component, and H_N is the average roof height of structure. In addition to this, there is a statement in the local Turkish Code to double the calculated force for fire suppression systems, emergency energy systems, and boilers of central heating systems.

Sample calculations

A sample seismic force calculation for a rigidly mounted transformer can be found below. The design lateral seismic force has been calculated both per IBC 2003 and Turkish Code 1997.

Physical characteristics of the sample equipment (same for both calculation methods):
W_p : Operating weight of the equipment (is equal w_e in Turkish Code);
taken as *5.000 kg*
z : Height in structure at point of attachment of component (is equal $H_İ$ in Turkish Code);
taken as *30 m* (assuming the *10*th floor of a building with *3 m* floor height)
h : Average roof height of structure (is equal H_N in Turkish Code)
taken as *45 m* (assuming the building has *15* floors with *3 m* floor height)

Parameters of IBC 2003 calculation (taken from tables in IBC 2003):
F_p : Design seismic lateral force; this will be the result of the calculation
a_p : Component amplification factor; taken as *1* for rigid mounted equipments
S_{DS} : Design spectral response acceleration at short periods; taken as *1,1* for Istanbul
R_p : Component response modification factor; taken as *2,5* as a moderate value
I_p : Component importance factor; taken as *1* for non-critical equipments

$$F_p = \frac{0{,}4\, a_p\, S_{DS}\, W_p}{\frac{R_p}{I_p}}\left(1+2\frac{z}{h}\right) = \frac{0{,}4\cdot 1\cdot 1{,}1\cdot 5.000}{\frac{2{,}5}{1}}\left(1+2\cdot\frac{30}{45}\right) = 2.053\, kgf$$

Parameters of Turkish Code 1997:
f_e : Design seismic lateral force; this will be the result of the calculation
A_0 : Ground acceleration; taken as *0,4* for Istanbul
I : Building importance factor; taken as *1* for non-critical buildings

$$f_e = w_e A_0 I\left(1+\frac{H_i}{H_N}\right) = 5.000\cdot 0{,}4\cdot 1\cdot\left(1+\frac{30}{45}\right) = 3.333\, kgf$$

Assuming a circulation pump with the same physical characteristics of the sample transformer is vibration isolated. In this case, the result of Turkish Code does not change, whereas the result of IBC multiplies by *2,5* since we have to take the a_p value as *2,5* for vibration isolated equipments.

Assuming a fire pump with the same physical characteristics of the sample transformer, which shouldn't be vibration isolated. This time we come up with *3.080 kgf* in IBC, since we have to take the I_p value as *1,5* for life safety equipments. On the other hand, the result is *6.666 kgf* in Turkish Code, since the Turkish Code says double the force for life safety equipments.

Table 1. Seismic force comparisons between IBC 2003 and Turkish Code 1997.

NON-STRUCTURAL EQUIPMENT	**Seismic Force (kgf) Per *IBC 2003***	**Seismic Force (kgf) per *Turkish Code 1997***
Transformer (non isolated)	2.053	3.333
Circulation pump (vibration isolated)	5.133	3.333
Fire pump (non isolated)	3.080	6.666

As seen in the above sample calculations, IBC 2003 gives more accurate results compared to Turkish Code 1997. The reason is that, there are more parameters in the IBC formula. It is obvious that a less detailed formula will give less detailed results. This, at the end will give either too low or too high results.

SEISMIC RESTRAINT DESIGN

Seismic restraint design for mechanical systems can be split into 2 groups:
1) Floor mounted, suspended or wall mounted equipments.
2) Suspended pipes, ducts, electrical cable trays etc.

Floor mounted equipments can either be rigidly mounted to the floor or to concrete pads or they can be mounted on vibration isolators with snubbers or directly on seismic isolators. If there is a need for vibration isolation (for a chiller for instance), rigid mounting cannot be done. If the equipment needs to be restrained, it is less expensive to use seismic isolators instead of using 2 different hardware, open springs and snubbers. Suspended equipments can also either be rigidly restrained if there is no need for vibration isolation or they can be hanged with vibration hangers and also restrained with seismic cables. Wall mounted equipments rarely need vibration isolation. Thus it is enough to rigidly mount them to the wall with anchors strong enough to resist the calculated seismic force.

Suspended pipes, ducts and electrical cable trays are critical, since those are subject to sway and damage both themselves and other systems close to them. Therefore, it is a must to brace these systems. The bracing (called seismic bracing, sway bracing or cable bracing) is a matter of design. The non-structural seismic engineer has to determine the points of bracing on a layout drawing. Many building codes (including IBC) and standards (including SMACNA) directs to put lateral bracing at each 12 meters and longitudinal bracing at each 24 meters. However, depending on the installation conditions, these distances can be less. In other words, seismic cable bracing points must be determined depending on seismic calculations.

Figure 2 shows a general concept of determining the cable bracing points on a pipe/duct/ electrical cable tray layout system. It also shows an alternate way of bracing (45° angles) instead of having lateral (transverse) and longitudinal braces separately. The installation details of seismic cables can be seen in Figure 3.

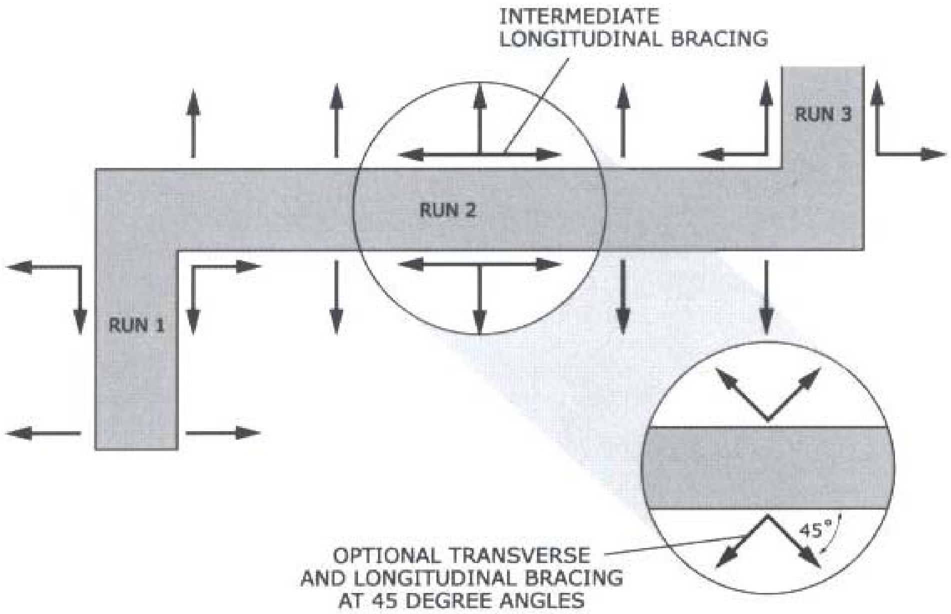


Figure 2. Seismic cable bracing for suspended pipe/duct/electrical cable tray lines.

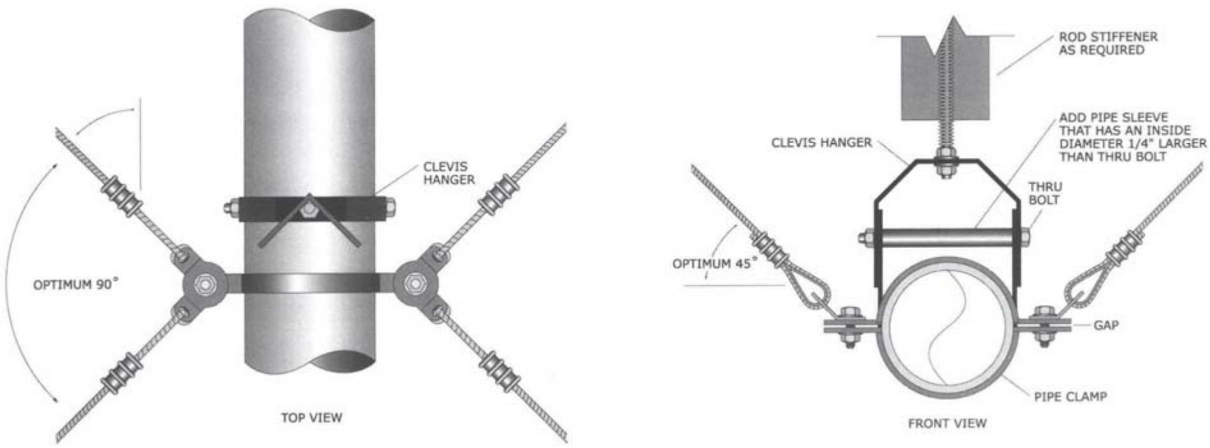


Figure 3. Seismic cable bracing details for suspended pipes.

SEISMIC RESTRAINT APPLICATIONS FROM TURKEY

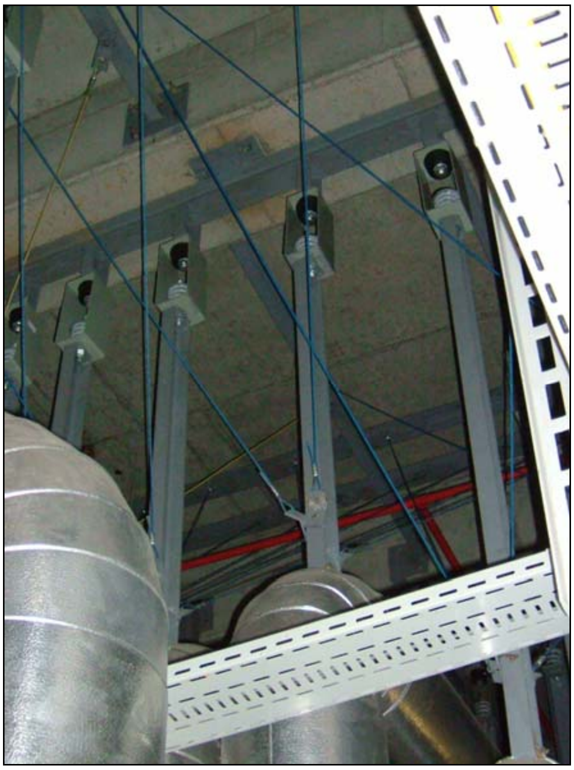
Tekfen Tower – Office Building: This is a 32 floor high class office building. The main HVAC equipments are located on top of the building right over a VIP office floor. This made it difficult to avoid vibration and noise. In order to achieve this, all equipments have been vibration isolated by using seismic isolators (Figure 4) and all pipe supports have been installed with vibration hangers and seismic cables (Figure 5).

a)

b)

Figure 4. Applications of seismic isolators in Tekfen Tower for; a) Pumps, b) Boilers.

a)



b)

Figure 5. Applications of seismic cables and vibration hangers in Tekfen Tower.

Kanyon Levent – Shopping Center: This is a building complex of shopping center, office tower and residence blocks. The main HVAC equipments have been vibration isolated with seismic isolators (Figures 6a) and pipes have been braced with seismic cables (Figure 6b).

a) b)

Figure 6. Applications of seismic isolators (a) and seismic cables (b) in Kanyon Levent

Sisli Plaza – Residence Tower: This is a premium class residence tower. All main HVAC equipments have been vibration isolated with seismic isolators and pipes have been braced with seismic cables. A seismic brace layout drawing of fire sprinkler system is shown in Figure 7.

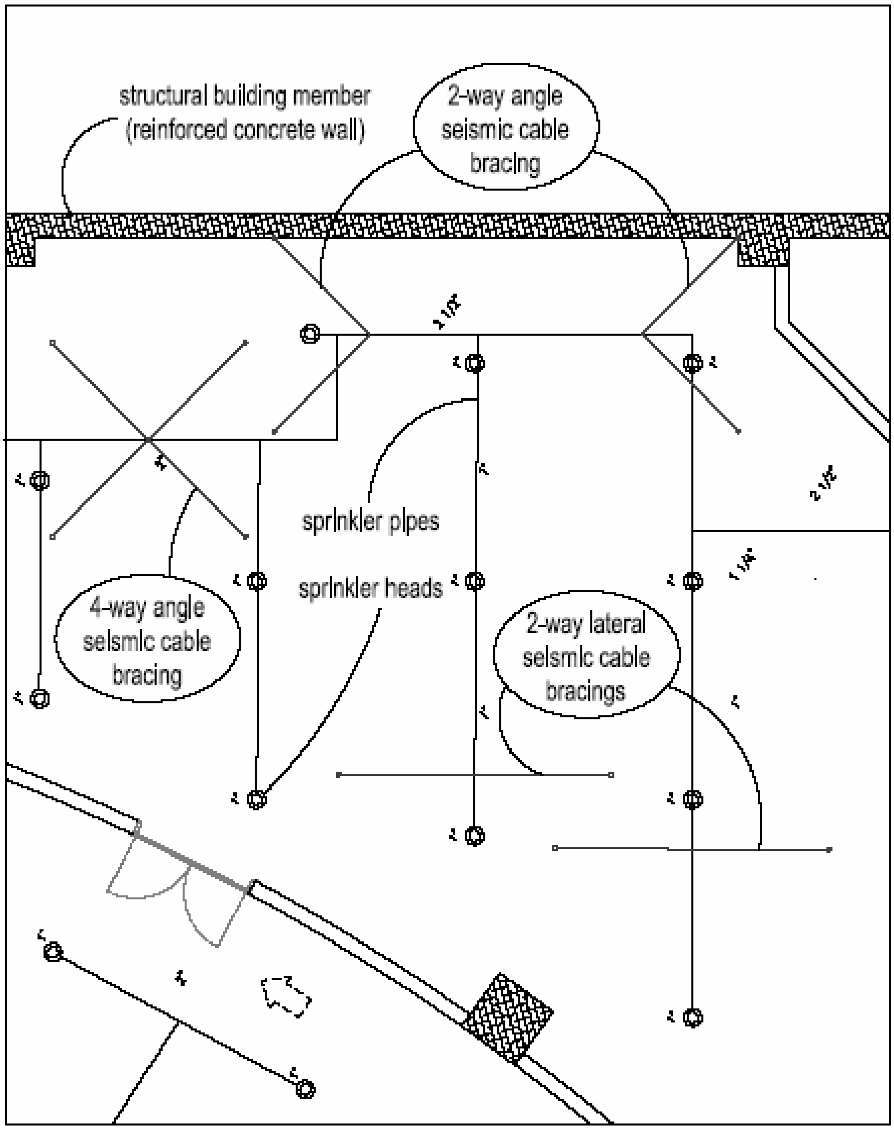


Figure 7. Seismic cable bracing layout drawing for Sisli Plaza Residence.

CONCLUSION

Seismic protection of mechanical and electrical systems in our buildings and facilities is crucial in terms of human life safety in the first place and also minimizing the costly damages. It is a matter of non-structural seismic engineering, which must be done by following the local and international building codes. It should also be done by professional non-structural seismic engineers. Finally, a necessity is to use certified seismic products only. These are the guidelines for the construction industry including the project owners, designers and consultants, contractors, control engineers and all other decision makers.

This issue is also very important for the code writers including government officials, university professors and managers of non-profit organizations. The weaknesses in local codes (for instance the Turkish Local Code) in terms of non-structural seismic protection must be taken into consideration carefully. Necessary precautions should be determined in the local codes and standards. As for Turkish code writers, the global level of awareness for seismic protection of non-structural systems should be reached.

ACKNOWLEDGEMENTS

Many thanks to my father Ceyhan Kalafat, PM of Tekfen Tower, for giving me my first job; my ex-boss Olgun Sonmez and Werner Stebner, VP of Amber/Booth, for opening me a door in seismic world; Robert E. Simmons for my training, to my colleagues Sarper Arun, Erkan Selek, Murat Kantarci, Okan Sever and others for being in my team; Prof. Mustafa Erdik, Prof. Abdurrahman Kilic and Prof. Ahmet Arisoy for their academic support; Celal Okutan, the founder of TTMD, and all other TTMD and MMO workers for their support.

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Heat recovery from raw sewage - An alternative for thermal energy supply in cities

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THE IDEA

- It is high time that we start recovering and re-using waste heat from industry and households. On a comprehensive view, waste heat could cover at least 30 % of our heating and cooling energy requirements. Today heat is being recovered from exhaust air and used on a very large scale. In many countries this is even governed by regulations. But what happens to waste water? Out of sight, out of mind.
- This was the starting point of the idea to utilize these resources, an idea resulting in the Rabtherm® Energy System. RABTHERM stands for heat recovery and utilization from untreated waste water. On leaving a house, the sewage has an average temperature of over 25°C and in the sewage system an annual mean of 15°C (summer 20°C, winter 10-12°C). Sewage is a continuously renewed source on a relatively high temperature level. With modern heat pumps one can transform this to a useful temperature of 65°C, high enough for hot water production and for the heating of newly constructed houses → heat pump COP's between 3.1 and 5.2, in special cases up to 6.2.
- The heat present in the waste water from residential buildings, trade and industry should therefore be utilized decentrally, i.e. locally, where it is generated, with a heat exchanger in the sewers and a heat pump. The temperature level is higher than that of most of the other natural renewable energy sources. The system can be used for heating and hot water in winter, and for cooling (air conditioning) and hot water in summer.
- The idea is not new but so far heat has been extracted practically only from purified waste water downstream of sewage works. However, since many sewage works lie at the periphery or outside residential areas, with large distances to the heat consumers, this strategy is uneconomic.
- Previous obstacles to the utilization of untreated water directly in the sewage system have been:
  - possible detriment to the biological purification stage in the sewage works, which are designed for a temperature of 8-13°C.
  - lack of a suitable heat exchangers which when installed in the drain channel, cannot lead to blockages.
  - → Today, these problems have been technically and economically solved, and the system is accepted by the authorities, i.e. Rabtherm®can now be marketed. The maximum cooling of the entire waste water on the way to the sewage works will in no case exceed 0.5 K.
- Contamination of the sewage channel and therefore the heat exchanger leads to a decrease in heat exchanger efficiency of up to 40%. (soiling with biomass film) Patented systems to solve this problem.

THE DEVELOPMENT

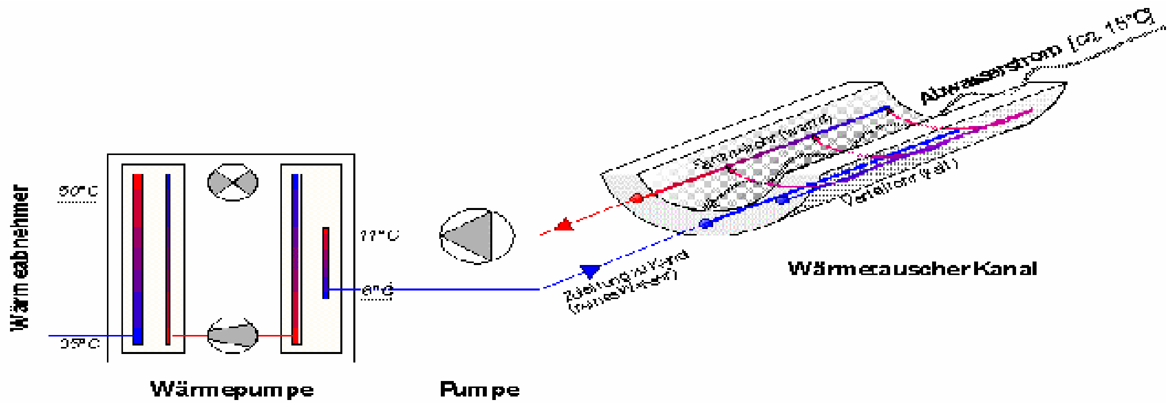
- The idea now had to be developed and brought to market in an economic manner.
- For this purpose, a trend analysis was required, e.g.
  - what is demanded by the energy market?
    → economical and/or ecological products
  - how are other technologies developing?
  - how fast is the political situation changing? (climatic changes, CO_2, fine dust)

  The trends were quite clear .
  - Electricity prices will drop .
  - Oil and gas prices have a tendency to rise .
  - Energy price surcharges or eco-taxes cannot be avoided sooner or later
- The expected result of the development process was that the local, decentralized utilization of the continuously available, renewable ambient energy by means of Rabtherm systems is
  - economic and
  - ecological / environmentally friendly.
- The goal was to develop a simple, robust and low-cost system. This required special efforts in the following areas:
  - Hydraulics, heat transfer
  - Materials
  - Joining technique
  - Design, installation

  The above objective has been achieved and led to patents. The know-how gained during this process represents a decisive lead over any competition.

THE PRODUCT (TECHNOLOGY)

- Working principle of the Rabtherm waste heat utilization system



Wärmeabnehmer .... user
Wärmepumpe .... heat pump
Pumpe .... circulation pump
Wärmetauscher Kanal .... heat exchanger sewage channel
Zuleitung zu Kanal (reines Wasser) .... connecting pipes (tap water)
Verteilrohr (kalt) .... distribution pipe (cold)
Sammelrohr (warm) .... collection pipe (hot)
Abwasserstrom [ca. 15°C] .... sewage

In the heat exchanger heat is extracted from the waste water and fed to the heat pump via the intermediate medium. The latter (pure water), circulating between heat pump (heat generation) and heat exchanger (heat utilization, heat extraction), is fed through the plastic pipe to the heat exchanger at the start of the cycle. The distributor pipe individually feeds each of the 1 to 3 m long heat exchangers. The intermediate medium warmed in the heat exchangers is then collected in the collector pipe and returned to the heat pump.

For summer cooling, the heat pump is hydraulically reversed, using the waste water as heat dump.

preassembled Rabtherm® - heat exchanger elements

- The specific extraction power of the heat exchanger is approx. 2-9 kW per metre heat exchanger (depending on the sewage flow rate, the flow speed or gradient and the degree of contamination). This rate can increase to 15 kW with pressure pipes. From 1 m3 of waste water (the contents of 5 bathtubs), the heat exchanger can extract 2-3 kWh of energy.
- Criteria for the application of Rabtherm systems.
  - sewage channel diameter > 400 to 500 mm
  - sewage flow average rate > 12 l/s
  - length of heat exchangers 9 m (min.) to 200 m (max.)
  - heating or cooling power output min. 80 kW
  - distance from sewage channel to user max. 150-300 m
  - heating temperature max. 70°C
- The heat exchanger is cemented into the sewage channel and is designed for a service life of at least 50 years.
  Corrosion, erosion / wear, leak tightness, channel maintenance and cleaning are some of the factors that because of the strict quality assurance criteria, had a decisive influence on choice of material, design, product (welding) and assembly / installation. Parallel to quality assurance, all possible damage repair scenarios have been established following the principles of "analysis-find-repair".
- The maintenance of the sewage channels with the integral heat exchanger requires no special effort. Blockage of the channels is impossible and they are cleaned with conventional equipment. The heat exchanger is dimensioned to withstand this treatment

Rabtherm Wülflingen (CH).

Trockenwetterrinnen Rabtherm Leverkusen (D) Dry weather channels.

THE MARKET

- After it had been recognized and proved that the system
  - makes good sense energetically, inasmuch as it has already been included in many urban energy plans
  - is economically feasible, in contrast to many other forms of alternative energy
  - brings ecological benefits with comparable or lower energy production costs than with energy from fossil fuels the question of markets and customers was considered.
- Who is interested in Rabtherm?
  - The heat consumer (user)
    . Rabtherm replaces the conventional heat energy supply with gas or oil by an environmental superior technology utilizing the waste heat from households that generates heat at economically comparable prices
    . The customer gets the same benefits (heat) at a comparable or better price, but in an ecological manner.
  - Who are the consumers?
    . Towns and cities with roughly 5'000 inhabitants upwards.
    . Communal co-operatives and consortia
    . Industrial enterprises with a significant fraction of consumption for space heating
    . Private building owners
    . The public sector
      . Rabtherm helps the public sector to
      . improve ecobalances
      . achieve energy policy goals
      . guarantee security of energy supply.
      → To supply 100 apartments with heat from waste water, the effluent from 300 apartments plus trade and infrastructure is required. To generate one kWh of heat, about 420 litres of waste water are needed.
      . Employment.
      . Rabtherm can generate around 30'000 man-years of work (up to 3'000 jobs).
- Who are our partners in the market?
  - Consulting engineers for the acquisition, study and planning of installations

   – Contractors for the installation, maintenance and operation of installations. Contractors are e.g.
      . Electricity utilities .
      . Municipal works .
      . Industrial enterprises

Contractors finance the construction of the installations and sell the energy generated. Electricity utilities and municipal works are highly interested in this lucrative second or additional source of income.

- What does Rabtherm cost?
   – Installations
      . CHF 1'800.00 - 2'500.00 per connected residential unit (€ 1170-1620)
      . CHF 500.00 -700.00 per kW of connected useful power (€ 325-455)
   – Heat exchangers
      . CHF 1'600.00 - 2'100.00 per m of heat exchanger (€1070-1360) .
      . The heat exchanger as an element, uninstalled, costs approx. 6-10% of the overall installation.
   – A sensitivity analysis predicts with high probability an improvement of profitability by over
   – 20%, influenced by
      . the price of electricity
      . the price of oil
      . investment costs
      . technological improvements (heat pump)

- How and where will Rabtherm® be employed?
   – The Rabtherm technology should be examined in the case of
      . renovation of large sewage channels .
      . renovation of large central heating or cooling works .
      . new sewage channels .
      . larger central heating or/and cooling works in the vicinity of larger sewage channels
- Market potential
   – Towns and cities
   – over 500'000 inhabitants 60 to 120
   – 200-500'000 inhabitants 27-60, equal to approx. 12'000 apartments
   – 100-200'000 inhabitants 20-27
   – 40-100'000 inhabitants 12-20
   – 15-40'000 inhabitants 5-12
   – 5-15'000 inhabitants 1- 5
      . Switzerland 2500
      . Germany 25000
      . Europe 120000
      . world 400000
- Existing and running installations 17 in Switzerland, Germany, Austria
- Installations in building or planning stadium over 100 in Switzerland, Germany, Austria, France, Ukraine, USA

ECOLOGY

- Rabtherm installations reduce the CO2 emissions over those from conventional plant by 50-80% (ecological benefit)Rabtherm installations reduce also the amount of primary / conventional energy.
- CO_2-output with the 17 running plants is reduced by 6000 tons.
- Rabtherm systems produce no fine dust like diesel engines and wood burning plants.

WHAT IS THE PRACTICAL SEQUENCE OF EVENTS IN THE PLANNING OF A HEAT UTILIZATION SYSTEM FROM UNTREATED SEWAGE?

- By means of a check list, a site can be roughly assessed for acceptance or rejection.
- Next the following procedure is scheduled .
  - Fundamental decision by the
    . sewage channel operator
    . sewage works operator
    . community
  - Analysis, site or feasibility study with data on the drainage system and heating furnaces. The studies contain the results of the local examination of sewage channel data (incl. condition) and furnace data, calculations of heating power and profitability, and cost estimates for the investments.
  - Clarification with the users or those to be connected .
  - Project with consulting engineers and contractor .
  - Execution and operation by contractors under long-term heat supply contracts.

Even with generally sinking heat demand, the Rabtherm technology, used in the district heating networks of city agglomerations, has an excellent chance of success, also in towns with an existing remote heating network.

ASSESSMENT / CONCLUSIONS

- The success of waste heat utilization from the public sewer system is based upon the following corner-stones

1. Profitability
2. Financing
3. Public interest
4. Ecology

- Profitability .
  - The heat production costs with a Rabtherm system are 5-20% lower than for conventionally generated heat.
  - The profitability increases when the installations are used in summer for cooling
  - Financing . by contracting
  - Public interest

  - Compliance with goals of energy and environmental policies
  - Creation of jobs
- Rabtherm Systems have an excellent outlook Future R&D can bring another 30% of improvement in quality, price and economy

RABTHERM PARTNERS

- engineers
- manufactoring companies
- concrete works
- contractors

have access to all the data, software and the latest research results.

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13 June 2007 at 13:30-15:00

D06
Low energy cooling

Simulation And Optimization of a Solar Absorption Cooling System Using Evacuated Tube Collectors

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SUMMARY

Summer air conditioning represents a growing market in commercial and residential buildings. Solar energy is a very interesting energy source because of its advantages. Instead of a compressor system, which uses electricity, an absorption cooling system needs only heat produced by a solar collector plant. Furthermore, Reunion Island has a high solar energy potential. The yearly average solar-radiation and is 5.4 kWh/m²/day. This paper presents a solar-powered, single stage, absorption cooling system, using a water–lithium bromide solution. The first part of this work deals with the dynamic modeling of an evacuated tube collector used for the simulation of heat production. In a second part, simulation and optimization of the system has been investigated in order to determine the optimum of solar collector plant surface, storage tank volume and nominal capacity of the absorption chiller. A new building code was added under TRNSYS16, to have dynamic coupling between building loads and cooling production.

INTRODUCTION

Reunion Island is a French overseas department located in the southern hemisphere characterized by a tropical humid climate. Conventional energy will not be enough to meet the continuously increasing need for energy in the future. In this case, renewable energy sources will become important. Solar energy is a very important energy source because of its advantages (insulation > 5 kWh/m²/day in Reunion Island). Of share our insularity an orientation towards renewable energies is today a perennial solution to obtain certain energy independence, cf. Figure 1.

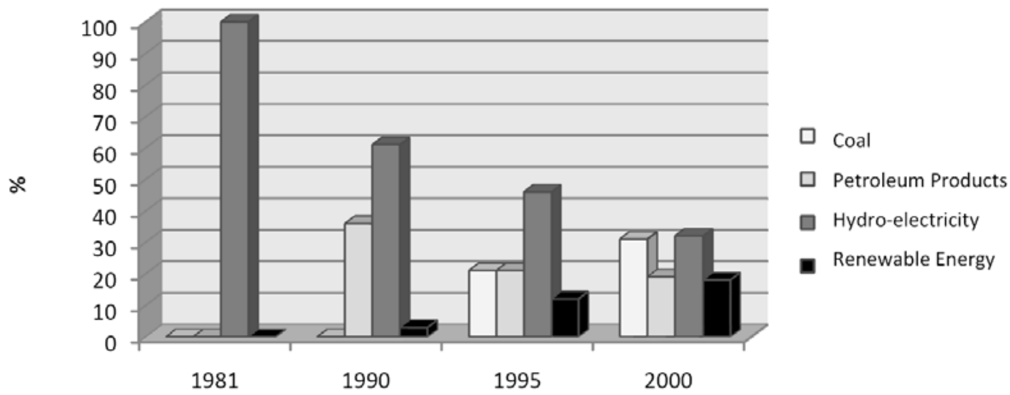

Figure 1. Evolution of the repartition of energy production in Reunion.

The building remains at the present time the most consuming sector in electricity. Under our latitudes, one of the most important demands for electricity relates to the air-conditioning of the buildings during the summer (period active of November 1 to April 30). Thus solar cooling system are particularly interesting as it is an application in which the demand for cooling energy closely matches the availability of solar energy, both in seasonal and the daily variations. One of the many categories of solar cooling systems is the solar absorption cooling. As no CFCs are used, absorption systems are friendlier to the environment. The possibility of cooling system using solar energy was initiated by the technological developments in the solar field, [1]. Absorption air-conditioning systems are similar to vapor compression air-conditioning systems, but differ in the pressurization stages. In general an absorbent in the low pressure side absorbs an evaporating refrigerant (H_2O). The most usual combinations of chemical fluids used include lithium bromide–water ($LiBr–H_2O$), where water vapor is the refrigerant, and ammonia–water ($NH_3–H_2O$) system where ammonia is the refrigerant, [2]. The electric quantity of power consumed by the pump is almost negligible. One needs nevertheless a contribution of heat, cf. Figure 2.

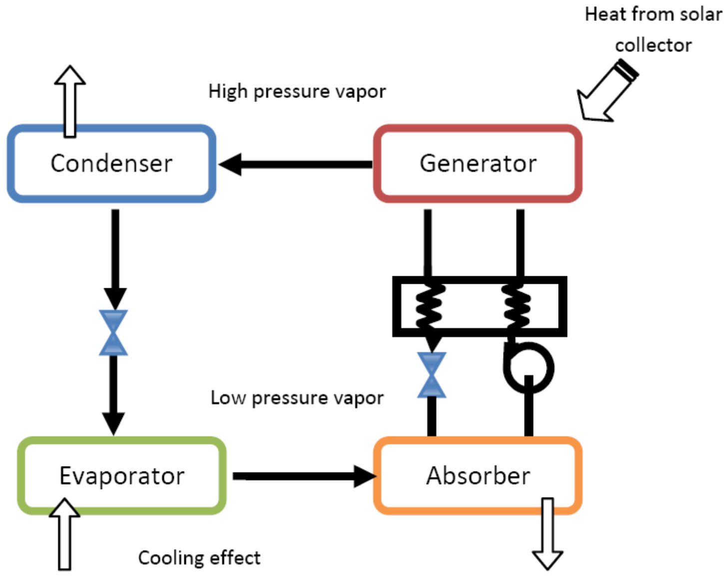

Figure 2. The basic principle of the absorption cooling system.

The single effect absorption chillers require for their operation a hot water temperature on the level of the generator of 80°C - 150 °C. Moreover, the temperature of cooling water must lie between 7°C and 43°C, [2]. The higher limit is founded in order to limit the differences in pressure between the generator and the absorber and the condenser and the evaporator. The lower limit makes it possible to avoid the crystallization of lithium bromide which is carried out at low temperature.

An European research program, IEA Task 25, was set up in 1998, in order to study these installations. Two approaches are classically used within the framework of this working group. The first consists of the establishment of real pilots, one counts at the present time an about sixty systems installed,[3]. The second consists to simulate and optimize existing or future systems via various codes. It is within the framework of second axis that this article is registered. The objective of this present work is to model the behaviour of the future installation at the Civil Engineering Department. Solar collectors represent the heart of the performances and the investments (~ 57%) of the installation. Thus, a good prediction of the performances of the solar loop is particularly judicious. The first part of this study was first interested in the dynamic modeling of the solar collectors. Then a coupling between the various components under TRNSYS has been investigate.

METHODS

There are three types of solar collectors which can be used within the framework of solar cooling: flat-plate, parabolic and vacuum collectors. Necessary heat for absorption chiller will be produced by evacuated tube collector. The literature contain numerous works on the modeling of solar collectors. These models developed have different levels of complexity. Usually, solar collectors are described by stationary models, considering the collector working under steady-state conditions. These approach are generally based on the work of Klein, [4].

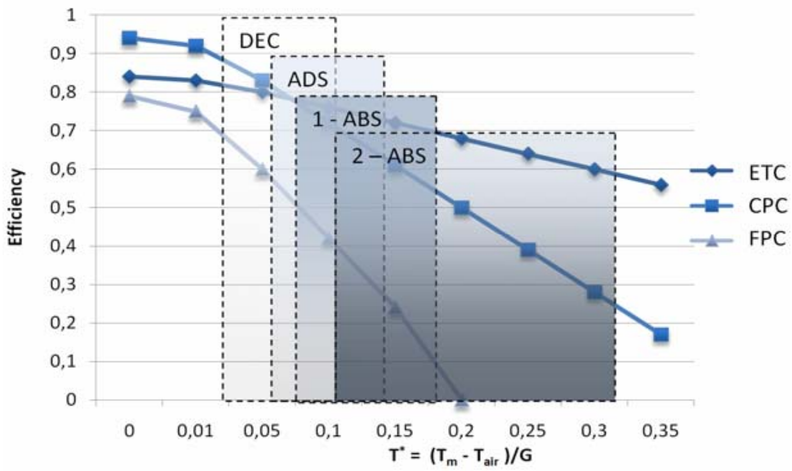

Figure 3. Comparison of Evacuated tube / compound parabolic / flat-plate collector for Desiccant / Adsorption / Absorption cooling systems.

A dynamic approach is more interesting in several cases: control strategies, dynamic testing procedures, coupling with others elements, particularly, in predicting the behaviour of collectors for short time step. The model developed corresponds to direct flow collector. The starting point of the model is a mathematical description proposed by Kamminga, [5]. The model consists of three nodes corresponding to the fluid, the absorber plate and the transparent glass cover. It is considered that the temperature of the fluid is a function of x. The fluid is moving in a single channel with the velocity u, along x-axis, see Figure 4.

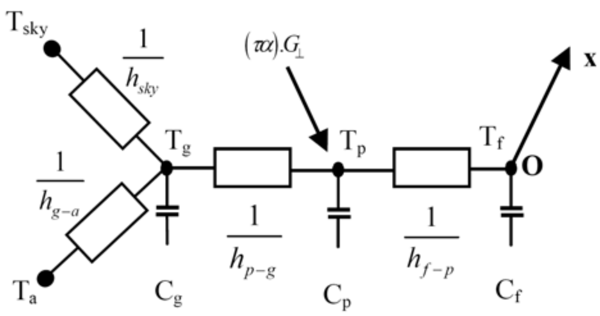

Figure 4. Thermal networks of the ETC model

The details of the model developed and elements of validation associated have been previously described by Praëne, [6]. A comparison between model forecast and measurements is presented in Figure 5. The simulation is carried out at the minute time step. A new type has been created under TRNSYS, in order to coupling with the rest of the solar cooling system.

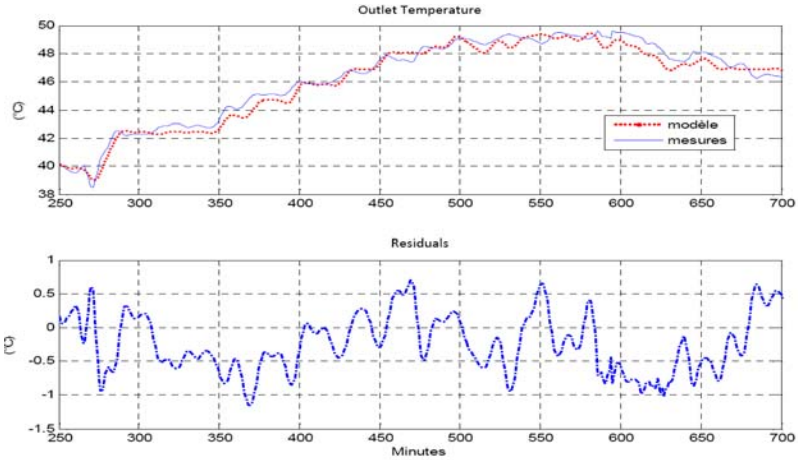

Figure 5. Simulation of the outlet temperature of the solar collector model.

The thermal behavior of the building was initially simulate over a whole year in order to observe cooling period. We used for that CODYRUN, [7], a thermal building code translated under TRNSYS to replace the TYPE56, this code has good accuracy for short time step. We evaluated over one year the level of comfort inside the classrooms and the building load to calibrate the absorption chiller refrigerating power.
We first simulate the evolution the inside classroom air temperature under natural ventilation during one year. The first condition of comfort is fixed for a temperature lower than 25°C. The graphic at Figure 6 shows that period spreading out May to October offers conditions of comfort. This period approximately corresponds to the southern winter.

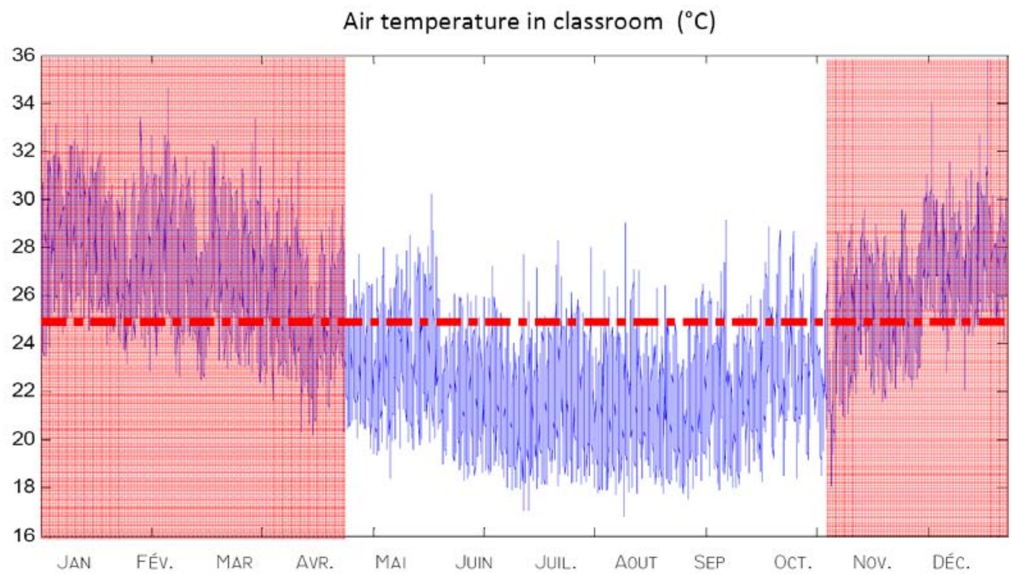

Figure 6. Yearly evolution of air temperature in classroom.

Many researches were carried out by the researchers in order to create simplified tools of indication of comfort, [8], [9]. We were thus based on the computation results of these indices of comfort within the framework of our simulations over the past year, in order to determine the building loads in relation to the indices of comfort. The starting point for the definition of the levels of comfort is based on the value recommended of PMV* (Predicted Mean Vote). That involves values of $-0.5 < PMV^* < +0.5$. In what concerns us, one is interested only in the positive part of this equation, which accounts for the requirements in air-conditioning. The evolution of the PMV*, according to Figure 7, comes well to confirm the zone of natural comfort which one has on the level of the building. This indices is particularly interesting because it considered the influence of air temperature and relative humidity.

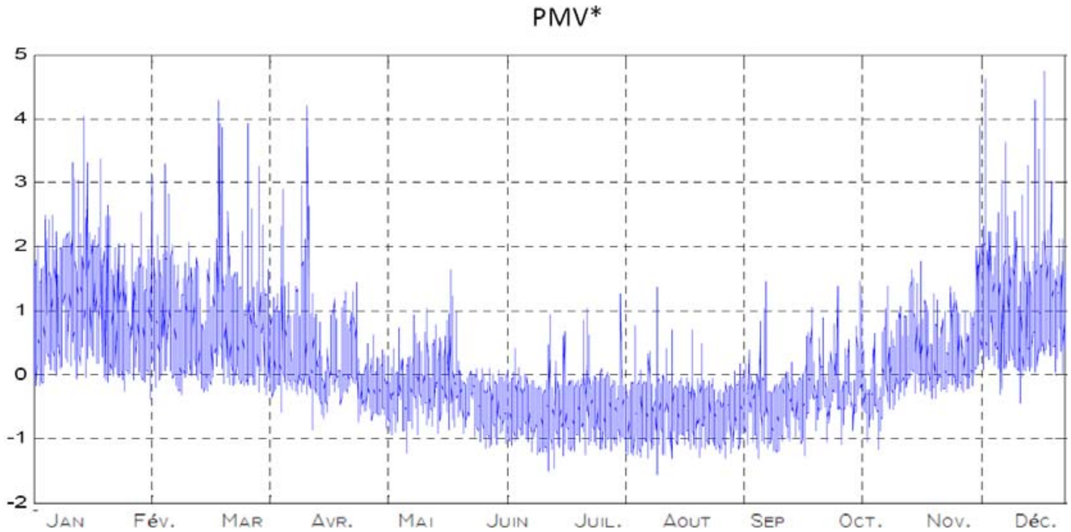

Figure 7. Yearly evolution of PMV* inside the classroom in natural ventilation condition.

The period from November to April corresponds to the period with maximum needs of solar cooling system. In order to evaluate the building, a sequence of January has been used. An air temperature of 25°C and humidity of 55% are fixed as comfort criteria to dimensioning the air-conditioning system, cf. Figure 8.

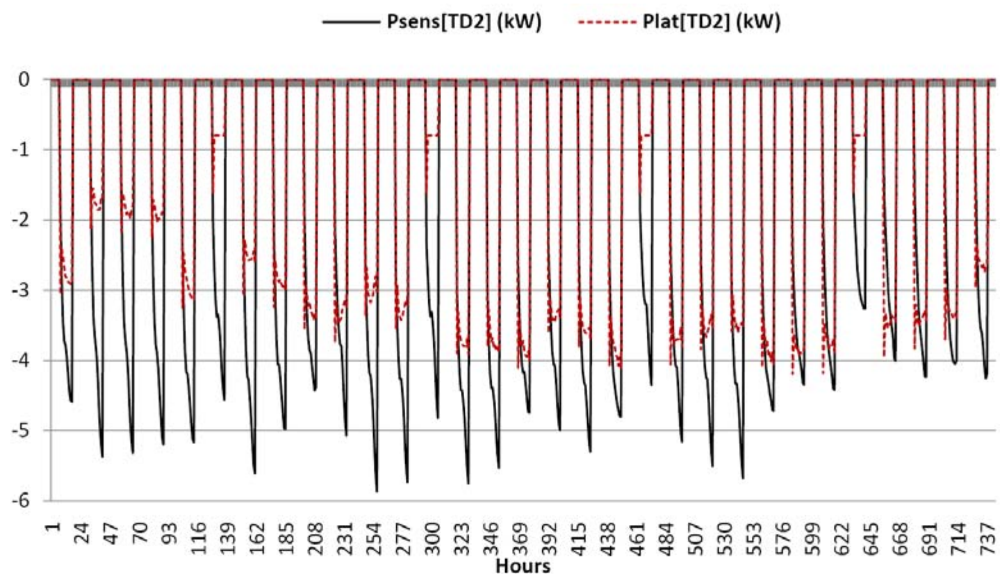

Figure 8. Sensible and latent power called by the classroom TD2 during January.

The maximum cold power called y the building is 9.5 kWf. This peak occurs after midday that is due to the accumulation of the walls inertia of the building and consequently restitution of this heat. Thus, for the four classrooms of the Civil Engineering Department, an absorption chiller of 35 kWf is necessary.

RESULTS AND DISCUSSION

This section presents the principal results obtain under TRNSYS16 for simulation of the global solar absorption cooling system. This program consists of the use of several "subroutines" which represents the components of the system described by ordinary or algebraic equations. Two types were added to the models existing under TRNSYS. It is about the model of simulation of the building and the model of vacuum collector. The useful surface area of collector used during simulation is 60 m ². The storage of hot water uses Type 38. That corresponds to a vertical roll of 800 L made up of copper insulated thermically with polyurethane. We consider a use of the machine with absorption without additional contribution of a contribution of auxiliary heat. We chose a sequence of January, to carry out our simulation, see Figure 9.

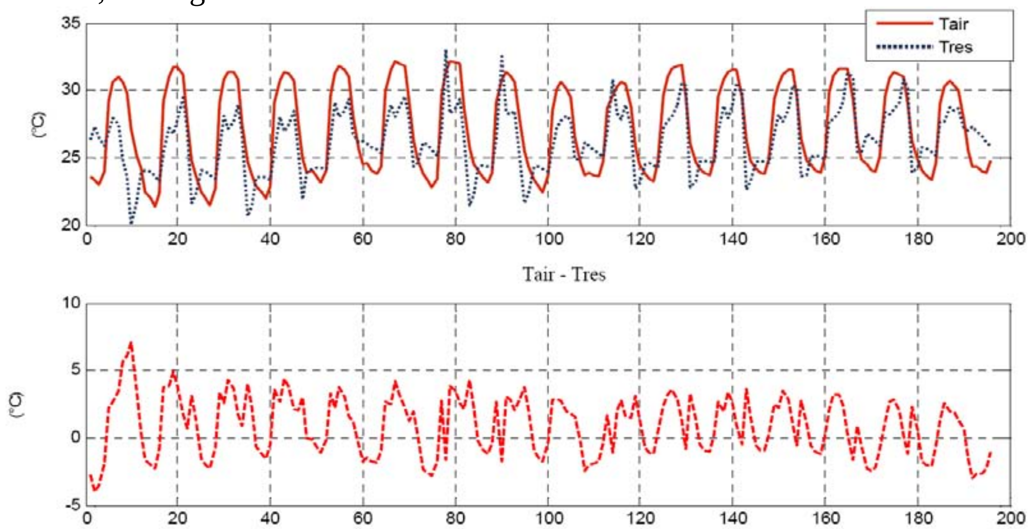

Figure 9. Comparison between resultant temperature (Tres) in classroom and outside temperature (Tair).

The average temperature in classroom is 27°C during the day. There is difference about 4°C between inside and outside air temperature.

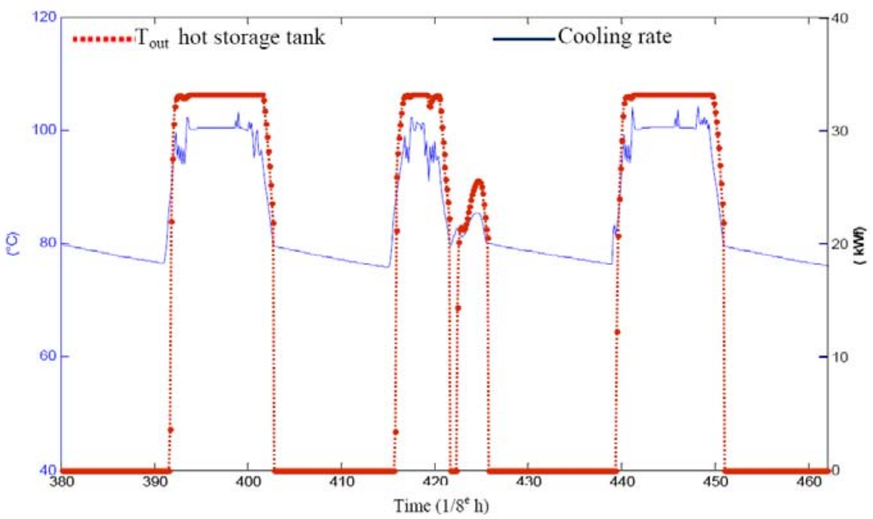

Figure 10. Evolution of cooling rate of the absorption chiller and outlet temperature of hot water storage tank.

The absorption chiller works without additional contribution of a contribution of auxiliary heat. As we can see on Figure 10, the 60 m² of ETC is enough to meet the minimum temperature of 80°C for the generator.
One of the most important points from an economic point of view of a solar cooling plant is the solar loop. The field of solar collectors accounts for approximately 60% of the total investment (in particular if it is vacuum collector). So it is important to dimension in a first place the needs for the building then the total surface of the field of solar collectors. Three points go in general to the decision of the field of solar collectors:
Ec onomic constraint: Budget dedicated to the project
Spa ce constraint: surface available for the solar field
Weather constraint: averag e numbers of good days during the hot period.
Also, it is important to quantify the influence of the elements of the solar loop, of as much we are in the case of an operation of solar cooling, without auxiliary contribution.

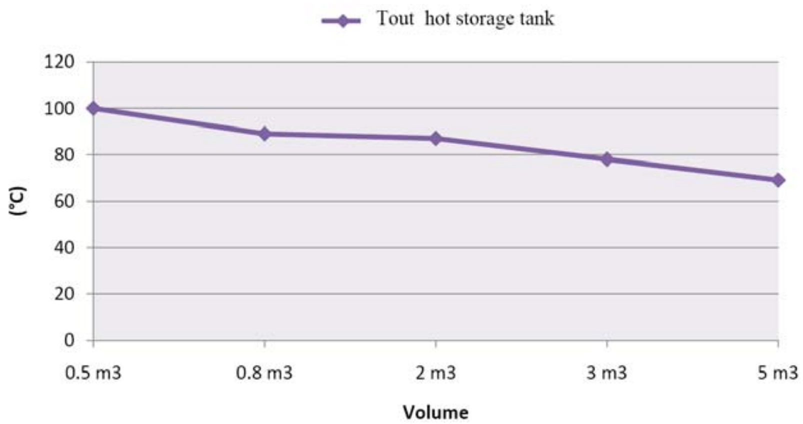

Figure 11. Effect of volume tank on the outlet of hot water temperature.

Many tests have been investigated to evaluate the sensitivity of the machine performance for different volume of storage tank, see Figure 11. For a volume superior than 2.5 m^3, the outlet temperature is lower than 80°C, thus the cold production will stop. Storage volume plays a dominating part because it has a buffering effect to the abrupt weather variations and makes possible to continue the hot water supply of absorption chiller The total solar collector area is the most important point (performance and economy). Several area of solar collectors have been tested (Figure 12.)

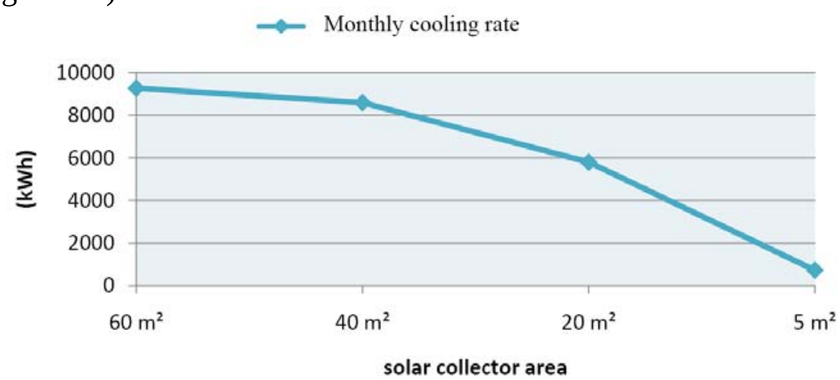

Figure 12. Energy production for different solar collector areas.

The effect of solar collector is evaluated against the energy from absorption chiller. The cooling rate from 60 m² to 40 m² is near. This production reaches to low value for a solar area less than 20 m². Thus, to have continuous condition for cold production a collector area of 60 m² is required.

CONCLUSION

The aim of the current work was to present a method of using solar energy for the air-conditioning. The system is modeled with the TRNSYS program. For the requirements in cooling for our classrooms, an optimum of the components consists of the use of 60m ² of vacuum solar collectors associated a storage tank of 0.8 m^3. We also fixed the nominal capacity of the absorption chiller with 33 kWf. The final objective of this work was to set up a simulation tool representing a solar cooling plant. This environment will be used as basic support with the simulation of various configurations. Theses simulations have been used to dimensioning an installation in our university. The cooling plant will be setup on may June 2007, in order to be in function at the beginning of summer in November 2007.

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Application Potential of Solar-assisted Desiccant Cooling System in Sub-tropical Hong Kong

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SUMMARY

Feasibility study of solar-assisted desiccant cooling system (SDCS) in Hong Kong has been conducted, by comparing the two major approaches – the outdoor air (OA) scheme and the mixed air (MA) scheme. In the former, desiccant cooling would fully make use of OA to satisfy the required cooling load. In the latter, MA from the pretreated OA and the return air would be mixed and supplied for cooling purpose. In principle, the latter may be better due to the enthalpy conservation from the return air. However from the year-round simulation study by using TRNSYS, it was found that the OA scheme would provide better cooling and energy performance under the typical meteorological year of Hong Kong. The system design of SDCS would depend on the local climatic conditions, and the extent of solar irradiation, air temperature and humidity should be considered thoroughly in the study.

INTRODUCTION

Hong Kong has the sub-tropical climate, featured with long hot and humid summer and temperate climatic conditions for half a year. Air-conditioning for comfortable indoor environment is therefore indispensable. Owing to the fact that air-conditioning and refrigeration becomes the biggest electricity "consumer" in Hong Kong [1], different measures of energy conservation and management have been implemented, in order to reduce energy consumption without sacrificing thermal comfort. Electrically driven compression refrigeration systems have been applied for a century, commonly found at homes, work places, industrial facilities and transportation. Due to the environmental impact of burning fossil fuel and non-stopping climbing of oil price, alternative energy sources other than fossil fuel generated electricity are being explored. Solar energy is definitely welcome if feasible technology for air-conditioning and refrigeration is available. In the recent years, there is encouragement from the government to seek for potential applications of renewable energy in Hong Kong, particularly to make use of solar energy [2]. Solar heating is a well-known technology, no matter using the flat plate collectors or evacuated tubes. However solar cooling has little mention, since it is a blooming technology even in Europe, USA and Japan, where research and development works are still ongoing [3,4]. The prominent advantage to utilize solar energy for building air-conditioning is the coincidence of solar irradiation availability and building cooling demand. In this study, the application potential of solar cooling in Hong Kong was perceived through suitable simulation methodology.

SOLAR COOLING IN CONTEXT

The basic principle of solar cooling is to apply the heat acquired from solar collectors for the thermally driven system, so that chilled water or conditioned air can be produced for air-conditioning purpose. From the latest research works [5-7], active solar cooling can be categorized in the following ways:

a. Solar-electric refrigeration: PV-operated compression cycle (compressor driven by a direct current motor)

b. Solar-thermal refrigeration
   i. Solar mechanical compression cycle (using Rankine cycle to drive a conventional compressor)
   ii. Absorption (note "ab") refrigeration
   iii. Adsorption (note "ad") refrigeration
   iv. Steam jet cycle (a novel cycle driven in 200°C)

c. Solar-thermal air-conditioning
   i. Solid desiccant cooling
   ii. Liquid desiccant cooling

In active solar cooling, the solar-electric refrigeration and solar-thermal refrigeration have the focus on the development of new types of chillers for refrigeration purpose. In solar-thermal air-conditioning, conditioned air with both design temperature and humidity is directly provided to indoor space. As incorporated with the desiccant adsorber, the humidity control can specifically handle the required latent load. Therefore its dehumidification performance would be commonly better than the conventional cooling coil.

SOLAR-ASSISTED DESICCANT COOLING SYSTEM

For the solar-assisted desiccant cooling system (SDCS), the core part is the sorbent component. Currently, both solid and liquid sorbents are available, like silica gel and lithium chloride respectively. Although the liquid desiccant cooling has feature of thermal storage in the regenerated liquid sorbent, the choice of the hygroscopic sorbent is limited, since the sorbent would be carried over into indoor space by the conditioned air. For the solid desiccant cooling, the processes of adsorption and desorption of moisture by solid sorbent are stable, that is more suitable to directly apply upon the conditioned air. Desiccant wheel is commonly used as the solid sorbent component. SDCS also includes the heat recovery unit, direct evaporative coolers, solar collectors and auxiliary heater, as shown in Figure 1. A typical psychrometric cycle in summer is also presented.

In principle, the adsorption or desorption process is adiabatic, but the wave propagation effect during the regeneration process of desiccant wheel would cause deviation from the theoretical process. Solar air collector is a common choice for SDCS. Since it can be directly coupled to SDCS for handling the regeneration air, and instantly make use of thermal gain without the need of thermal storage. Solar air collector is effective when the cooling load profile is in line with the availability of solar irradiation in day time. The heat recovery unit is used to conserve the sensible heat and pre-cool the outdoor air. The evaporative cooler EC1 would cool down the supply air up to the humidity level capable of tackling the space latent load.

Another evaporative cooler EC2 is installed to cool down the return air, for furnishing better sensible heat extraction at the heat recovery unit.

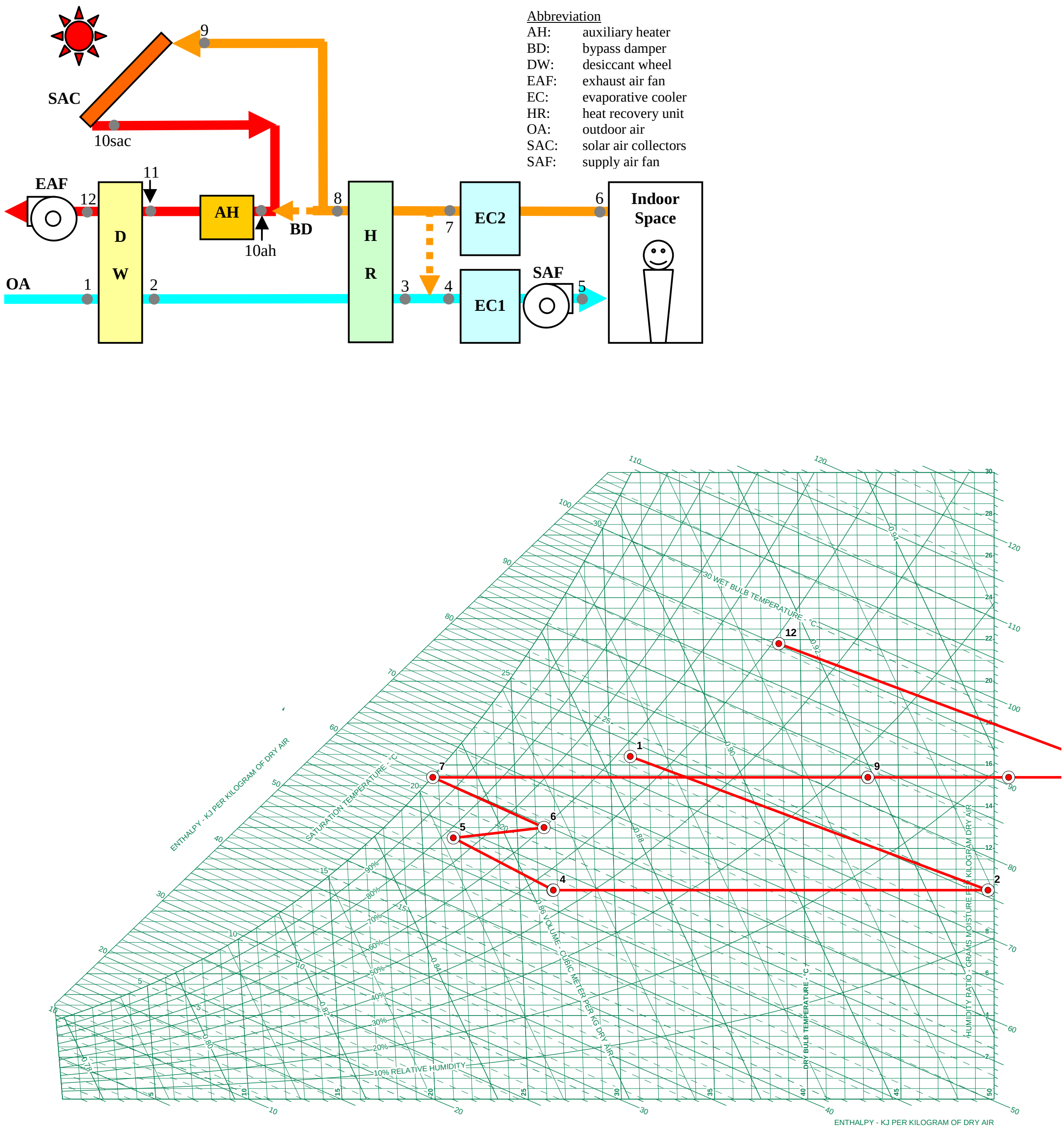
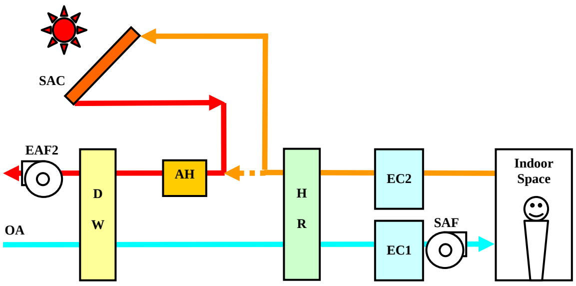


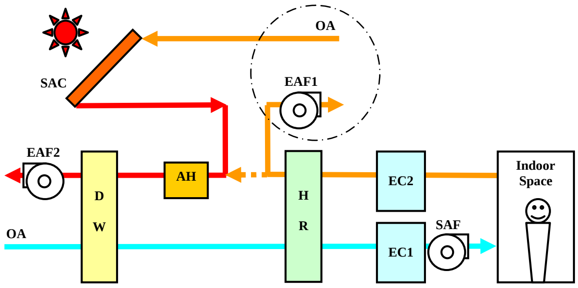
Figure 1. Schematic diagram and typical psychrometric cycle of solar-assisted desiccant cooling system in the summer.

The European countries have practical experiences in the system design and installations of SDCS [4]. Generally SDCS has the outdoor air (OA) scheme and mixed air (MA) scheme. In the former, desiccant cooling would fully make use of the OA to fulfill the required cooling load. In the latter, the MA from the pretreated outdoor air and the return air would be mixed for cooling purpose. Therefore the OA scheme would have larger sizes and higher capital costs of equipment components, since the amount of full OA is involved, while the MA approach would have smaller equipment sizes thus lower initial costs due to smaller amount of OA required. It seems that the latter can have the advantage of enthalpy conservation from the return air, but the cooling performance of the former may be better due to larger amount of conditioned air being produced. This would depend heavily on the local climatic conditions, particularly the solar irradiation, air temperature and humidity. As a result, simulation model was built for in-depth study between these two schemes. Moreover in each scheme, the regeneration air for the desiccant wheel can be either from the return air leaving the heat recovery unit, or a separate outdoor air stream. In summary, there are altogether four alternatives for the system design of SDCS as follows:

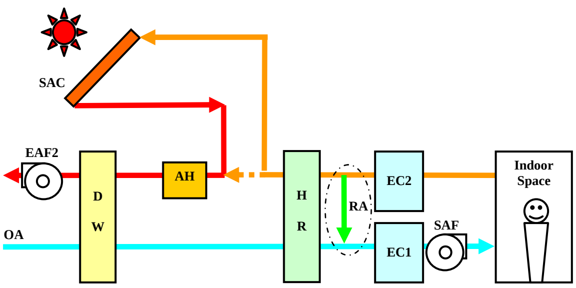
A. OA scheme with return air for regeneration (Figure 2a);
B. OA scheme with separate outdoor air for regeneration (Figure 2b);
C. MA scheme with return air for regeneration (Figure 2c); and
D. MA scheme with separate outdoor air for regeneration (Figure 2d).



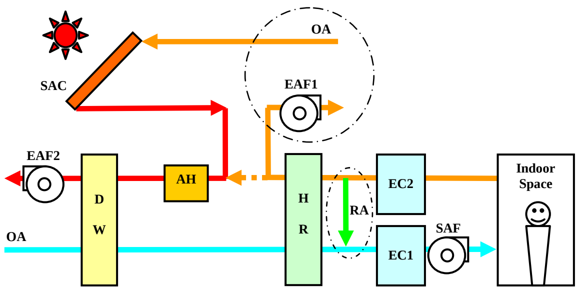
(a) Alternative A



(b) Alternative B



(c) Alternative C



(d) Alternative D

Figure 2. Four alternatives for system design of solar-assisted desiccant cooling system

DEVELOPMENT OF SIMULATION MODEL

In order to have an in-depth study and comparison for these four alternatives, simulation model of SDCS was built by using TRNSYS 16.01 [8], with the support of the component models from TESS [9]. The desiccant wheel model could determine the regeneration air inlet temperature based on the humidity ratio set point. The heat recovery unit model had the sensible effectiveness of 0.8. Each evaporative cooler model could vary its saturation efficiency between 0 – 100%. Each fan model was the variable speed type, and its power was adjusted from the rated power according to the flow change. The solar air collector model was an un-glazed type that passed the air behind the absorbing plate. It had the absorptance of 0.947 and emissivity of 0.85. The auxiliary heater model was electric type with efficiency of unity. In Hong Kong, the energy cost of electricity and that of town gas are comparable, so the electric heater was designed for system and operation simplicity. The Hong Kong Typical

Meteorological Year [10], which was in the EnergyPlus / ESP-r format (*.epw), was adopted in this simulation model.

In order to understand the effectiveness of the cooling performance of SDCS, it was designed to provide air-conditioning for function area with commonly high cooling intensity, therefore a seminar room located on the top floor was selected for simulation purpose. It had floor area of 213 m^2 and 3.82 m high. There were four external walls and a flat roof. The wall-fenestration ratio was 0.4 to 0.6. The internal and external shading factors of fenestration were 0.8 and 0.2 respectively. The internal heat gains included 120 persons seated at rest and 13 W/m^2 artificial lighting with 30% convective part. The daily air-conditioning period was from 08:00 to 18:00 to suit the solar availability.

According to the local design practice, the rated mass flow rate and rated power of the variable supply air fan were 3.47 m^3/s and 4.63 kW respectively, with the flow range from 30 to 100%. The rated flow rate of the variable exhaust air fans would depend on the alternatives, ranged from 1.04 to 3.47 m^3/s. The rated power of the exhaust air fans EAF1 and EAF2 were 2.89 kW and 5.79 kW respectively. The power of heat recovery unit and evaporative cooler were 0.19 kW and 0.034 kW respectively. The capacity of the electric auxiliary heater was 100 kW. The set point of the desiccant dehumidification was 0.01 kg/kg. 100 m^2 solar air collectors with tilt angle of 22°C were installed on the roof directly above the air-conditioned space.

Control scheme was included for year-round operation in response to different climatic conditions and changing heat gains. There were free cooling mode and desiccant cooling mode. Free cooling with adjustable fan speed of supply and exhaust air fans was implemented when the outdoor temperature was higher than 5°C. When the outdoor temperature was higher than the room temperature 24°C, desiccant cooling was called in and the supply air temperature was set at 13°C. Bypass control was applied whenever there was no thermal gain at the solar air collectors. The entire plant and energy simulation model was developed in a total energy approach, with the consideration at the equipment level, system level and operation level, in order to reflect the effectiveness of SDCS in different seasons in Hong Kong. The general cooling performance and year-round energy consumption of this simulation model was validated with a similar project in Freiberg of Germany [4].

RESULTS AND DISCUSSION

For the four alternatives, the simulation results for comparison are summarized in Table 1 and introduced below.

Table 1. Year-round simulation results of the four alternatives.

Alt	Annual electrical consumption (kWh)	Annual thermal gain (kWh)	Average solar fraction	Hour percent out of comfortable temperature	Average COP
A	154,417	17,166	0.15	15.0%	1.07
B	225,417	33,348	0.14	17.5%	0.51
C	79,167	13,207	0.20	79.4%	-0.45
D	103,599	28,181	0.25	78.8%	-0.41

The *annual electrical consumption* is the year-round electrical energy consumption of all the involved components of SDCS, including the fans, evaporative coolers, heat recovery unit, desiccant wheel, and electric auxiliary air heater. The *annual thermal gain* is the year-round thermal energy acquired by the solar air collectors during the operation of SDCS. The *average solar fraction* is to compare the solar fraction of the thermal gain from solar air collectors Q_{sac} to the heat generated from the auxiliary heater Q_{ah}. The monthly average solar fractions were determined first, then the year-round average value SF_{avg} was found as follows:

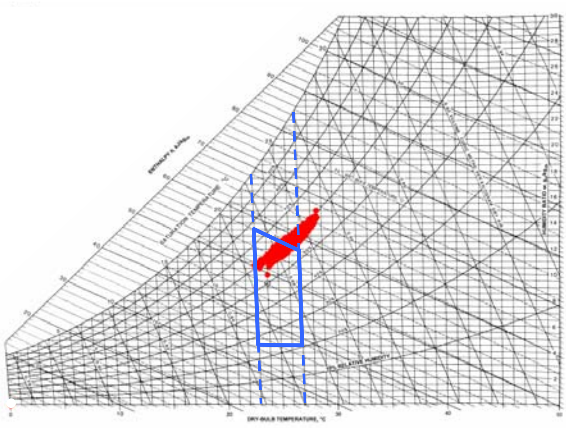
$$SF_{avg} = \frac{\sum_{i=1}^{12} Q_{sac,i} / Q_{ah,i}}{\sum_{j=1}^{12} j} \qquad (1)$$

The *hour percent out of comfortable temperature* is to consider the effect about thermal comfort. In this study, the upper limit of 27°C of the summer comfort zone from ASHRAE [11] is used to check against the room air temperature. The percentage was based on the annual air-conditioning hours. The hourly COP (coefficient of performance) of SDCS is defined as follows:

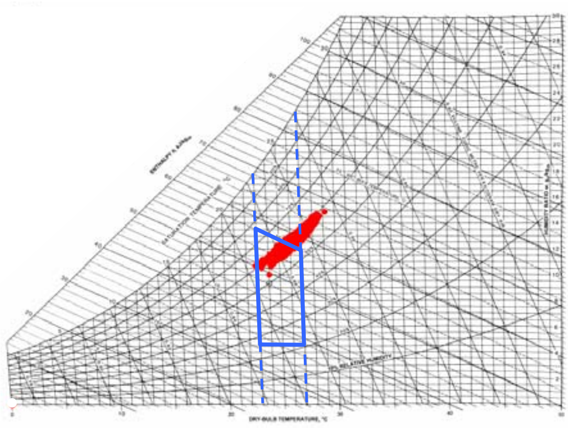
$$COP = \frac{\dot{m}_{\text{supply}}(h_{\text{outdoor}} - h_{\text{supply}})}{Q_{\text{in}}} = \frac{\dot{m}_{\text{supply}}(h_1 - h_5)}{Q_{\text{sac}} + Q_{ah}} \qquad (2)$$

where $\dot{m}_{\text{supply}}$ is the mass flow rate of supply air for the conditioned space, h_{outdoor} or h_1 is the outdoor air enthalpy, h_{supply} or h_5 is the supply air enthalpy, Q_{in} is the total thermal energy input that is the sum of Q_{sac} and Q_{ah}. The monthly average COP was determined, then the year-round average COP.

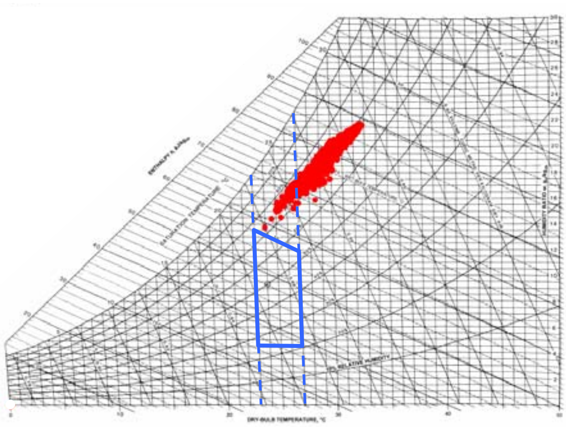
In addition to the results in Table 1, the room conditions during the air-conditioning hours are plotted in red spots on the psychrometric chart. The ASHRAE summer comfort zone is encompassed by blue solid lines.



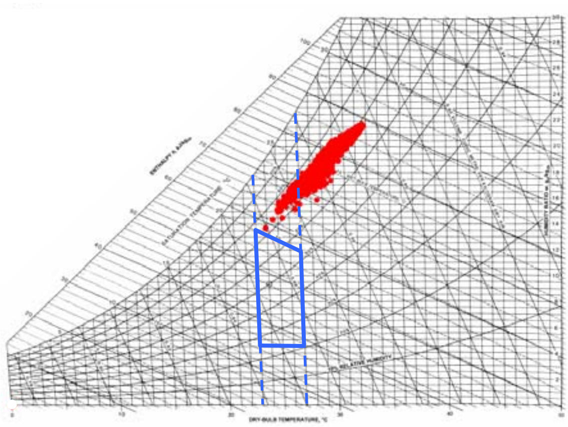
(a) Alternative A



(b) Alternative B



(c) Alternative C



(d) Alternative D

Figure 3. Room conditions during the air-conditioning hours throughout a year for the four alternatives.

From Table 1, it seems that Alternative C had the lowest annual electrical consumption among different system designs. But with regard to the other results, particularly Figure 3(c) and (d), both Alternatives C and D could not generally provide the required comfortable conditions for the air-conditioning area. It accounted to 79.4% and 78.8% of air-conditioning hours higher than the comfortable upper limit of 27°C. These two alternatives were the MA scheme, although they had relatively high average solar fraction due to better thermal gain from 30% OA passing the same area of solar air collectors. But such flow rates of process air and regeneration air were not enough to provide the necessary cooling effect for the space loads. Therefore only about 20% of the room temperature was satisfactory in a year. In addition, the average COP of both Alternatives C and D were negative in Table 1, implying that h_{outdoor} was lower than h_{supply} from Eq(2). This shows that the enthalpy of MA was even worse than that of the OA, and the room condition was not comfortable enough in most of the time.

As a result, only Alternatives A and B would be under consideration. Both of them were designed based on the OA scheme. From Table 1, it is obvious that Alternative A was advantageous over Alternative B due to lower annual electrical consumption, greater average solar fraction, smaller hour percent out of comfortable temperature and better average COP. The annual thermal gain of Alternative A was less than that of Alternative B, but it did not have contradiction to the result that Alternative A had a better solar fraction. It was because Alternative A applied the return air with higher temperature for regeneration in desiccant wheel (Figure 2(a)), while Alternative B used the OA with lower temperature for the same purpose (Figure 2(b)). Therefore the thermal gain in Alternative A would be less hence the demand of auxiliary heating for regeneration was also less, and the solar fraction was eventually higher. Consequently, Alternative A, the OA scheme of SDCS with return air for regeneration had the best overall performance in view of energy and cooling effect.

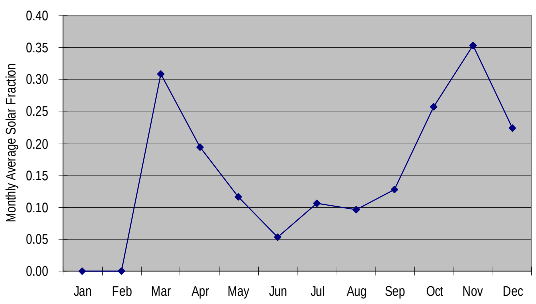


Figure 4. Profile of monthly average solar fraction in different months.

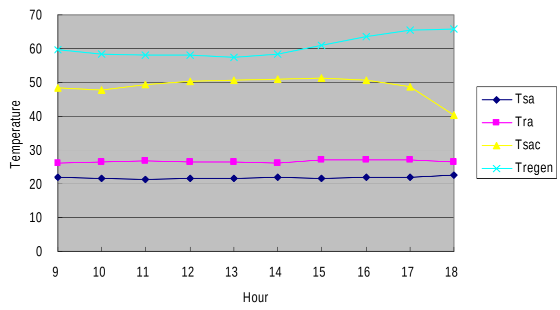


Figure 5. Temperature profiles of supply air T_{sa}, room air T_{ra} , solar air collector outlet T_{sac}, and regeneration air T_{regen} on a typical summer day.

For Alternative A, an in-depth performance appraisal was made. Figure 4 shows the changing profile of monthly average solar fraction. It is noted that the solar fraction was high around March and November, which are the spring and autumn in Hong Kong. The weather and cooling load of these two months were moderate, so the solar contribution could be more significant. For the local summer period from May to September, the solar fraction was low, between 0.05 to 0.13. This indicates that auxiliary heating was necessary in summer in order to achieve the design humidity set point for desiccant cooling purpose. Figure 5 illustrates the temperature profiles of supply air, room air, solar air collector outlet and regeneration air on a typical summer day. For both the supply and room air, their temperatures are fairly constant throughout a day, around 22°C and 27°C respectively. The outlet temperature of solar air collectors and regeneration temperature were around 50°C and 60°C. The temperature difference was accomplished by the auxiliary heater, this shows again the need of auxiliary heating in the summer time.

CONCLUSION

Although there are many solar cooling applications in the European and western countries, the system design should be carefully examined for local use based on the climatic conditions. In this study, the outdoor air scheme was found more suitable than the mixed air scheme for the solar desiccant cooling system (SDCS) in Hong Kong, and the system configuration of return air for regeneration was more effective. The average monthly solar fraction and COP were 0.15 and 1.07 respectively. Owing to the sub-tropical climate featured with hot and humid summer in Hong Kong, auxiliary heating was necessary in order to achieve the design performance of SDCS. Although the application potential of solar air collectors in Hong Kong was generally assured, there were 15% air-conditioning hours exceeding the comfortable temperature limit for the air-conditioning area with high cooling load intensity. This problem can be firstly handled by the other choices of solar collectors, like the flat plate collectors or evacuated tubes, and it would be studied in the next stage. Total solar cooling solution would be pursued to prevent from using auxiliary cooling.

ACKNOWLEDGEMENT

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Numerical Analysis of Heat and Moisture Transfer in Desiccant Wheel for Dehumidification

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SUMMARY

In the desiccant dehumidifier using adsorbent such as silica gel and zeolite, moist outdoor air is dehumidified by the adsorbent, and the damped adsorbent needs to be dried (regenerated) by giving heated air. Utilizing solar thermal energy as a heat source for the regeneration of the adsorbent leads to large energy conservation. We aim to develop the high efficiency desiccant dehumidification system using solar thermal energy. For this development, we investigated numerical analysis method of heat and moisture transfers in desiccant wheel in order to carry out the optimal design of this desiccant system. In this paper, the heat and water vapor transfer phenomena inside a desiccant wheel were formulated, and the algorithm of numerical calculation for the formulated equations was investigated. Its validity was checked by comparing the calculation results with the results of the repeated adsorption-desorption experiment for a fixed desiccant wheel (the wheel is not rotating), and with the experiment using an actual desiccant machine (the wheel is rotating). With regard to the numerical calculation targeted at the repeated adsorption-desorption experiment, the periodic change in moisture content was accurately reproduced by the calculation. However, with regard to the actual desiccant machine, this numerical analysis could not accurately predict the air temperature after passing through the desiccant wheel, although it could evaluate the amount of dehumidification fairly well.

INTRODUCTION

In Japan, solar heat systems have been introduced in mainly detached houses. In fiscal 2002, the quantity of these amounted to 740,000 kL in crude oil equivalent. The scheme for achieving the goals of the Kyoto Protocol [1], which was approved at a Cabinet meeting in April 2005, also includes the aim to further promote the introduction of solar heat systems. In order to achieve the goals, it is necessary to promote the use of solar systems for the public buildings, housing complex, and industrial fields, and so it is imperative to develop the technology for application in these fields. In order to spread the heating and the hot water supply systems that utilize solar heat, it is one key to find how to use the excessive solar heat during summer and spring/autumn seasons, and we started on a research project of “the floor heating & cooling, ventilation, and hot water supply system using solar heat” (Fig.1). For this system, we are developing a wheel-type solid desiccant air-conditioning system (Fig. 2) using this excessive solar heat. In the desiccant system that uses absorbents, such as silica gel and zeolite, outdoor moist air is dehumidified in absorbents, and heated up by adsorption heat, to become low-humidity high-temperature air, and then it passes sensible heat exchanger and evaporative cooling, to be supplied as cool and dry air. By utilizing solar heat as the heat source for regenerating (dehumidifying) the damped absorbents, it is possible to gain a great degree of the energy-saving effect. In addition, this is not dehumidification by the dew condensation method, and so it is possible to inhibit the growth of mold and fungi. As mentioned above, the desiccant system is an air-conditioning technology attracting people's attentions as an environmentally-friendly air-conditioning system from the perspectives of energy saving, air quality, and chlorofluorocarbon issues. But for the popularization of this system, we need to pursue higher efficiency, miniaturization, weight reduction of devices, the methods for utilizing solar heat effectively, and the technique for post-cooing at low energy costs.

In this study, in order to develop a high-efficiency desiccant system utilizing solar heat, we are examining the pre-cooling technique of the adsorbing part of the desiccant wheel, unique post-cooling methods, and the application of new adsorbing materials. We are also intended to optimize a variety of parameters that influence the performance of the desiccant system. There are a large number of parameters that determine the performance of a desiccant system—desiccant material's moisture adsorption/desorption characteristics, size of the wheel, air volume, process/regeneration splitting ratio, rotation speed, temperature of regeneration, void ratio, and the degree of pre-cooling. Therefore, it costs an enormous amount of time and labor to discover the desirable combination of parameters.

In this circumstance, we developed a technique for numerical analysis of heat and moisture transfers, considering the rotation of a desiccant wheel in order to carry out the optimal design of a desiccant system. The heat and moisture transfer phenomenon inside a desiccant element was formulated, and the algorism of numerical calculation for the formulated equations was investigated. First, we measured the equilibrium moisture content, which is used as the input value of the adsorption/desorption characteristics of a desiccant material for numerical calculation. Secondly, in order to confirm the validity of the numerical analysis method, we compared the results of the experiment of repeating the adsorption and desorption when a wheel is fixed with the results of the numerical calculation, and then compared the results of the measurement in the actual desiccant machine and the results of the numerical calculation that took the rotation of the wheel into account.

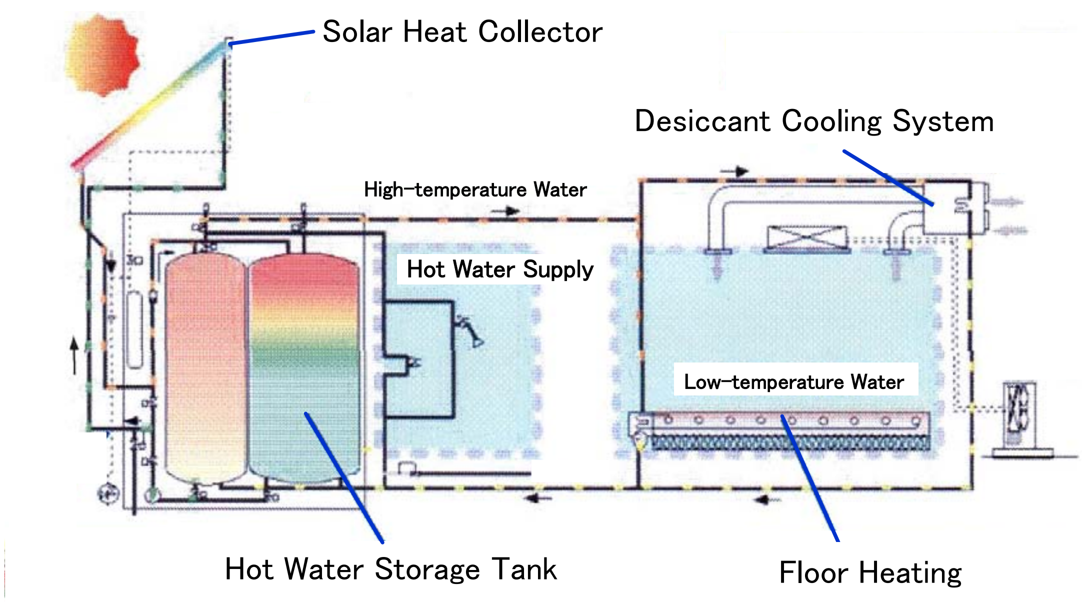

Figure 1. Schematic diagram of the floor heating & cooling, ventilation, and hot water supply system using solar heat.

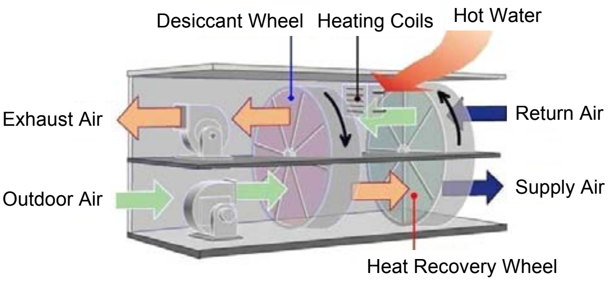

Figure 2. Desiccant cooling system

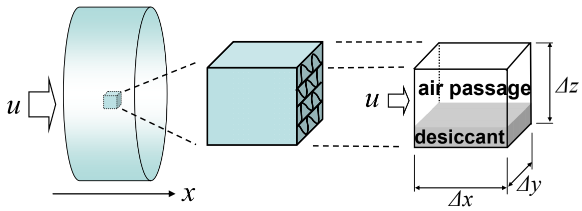

Figure 3. Modeling of desiccant wheel

TRANSPORT EQUATIONS OF HEAT AND WATER VAPOR INSIDE A DESICCANT ELEMENT

As shown in the left diagram of Fig. 3, a parallelepiped infinitesimal volume is taken from the wheel and modeled, as shown in the right diagram of Fig. 3. Considering the balances of water vapor and heat quantity inside this infinitesimal volume, the three transport equations (1), (4), and (5), as shown in Table 1, can be obtained. Equation (2) indicates that the moisture content of a desiccant material changes due to the water vapor transfer on the desiccant material's surface. Equation (3) is based on the assumption of local equilibrium in which the relation between the absolute temperature and moisture content of the desiccant material's surface (adsorption/desorption phase) instantaneously follows the adsorption isotherm. This formulation was conducted with reference to the theory of simultaneous transfer of heat and water vapor proposed by Matsumoto and Hokoi et al.[),)]. The adsorption isotherm of Equation (3) varies significantly according to materials, and so in this study, we measured this for the target desiccant wheel.

Table 1. Transport equations of heat and water vapor

Water vapor transport equation in air passage

$$\varepsilon\rho_a \frac{\partial X_a}{\partial t} = -\varepsilon u \rho_a \frac{\partial X_a}{\partial x} + \varepsilon \frac{\partial}{\partial x}\left(\lambda_a{}' \frac{\partial X_a}{\partial x}\right) - \gamma \frac{\partial w}{\partial t} \quad (\mathrm{kg/m^3 s}) \qquad \mathbf{(1)}$$

Equation of moisture content

$$\gamma \cdot \frac{\partial w}{\partial t} = \alpha' S\left(X_a - X_b\right) \quad (\mathrm{kg/m^3 s}) \quad \mathbf{(2)} \qquad w = f\left(X_a, \theta_a\right) = f\left(X_b, \theta_d\right) \quad (\mathrm{kg/kg_d}) \quad \mathbf{(3)}$$

Heat transport equation in air passage

$$\varepsilon C_a \rho_a \frac{\partial \theta_a}{\partial t} = -\varepsilon u \rho_a C_a \frac{\partial \theta_a}{\partial x} + \varepsilon \frac{\partial}{\partial x}\left(\lambda_a \frac{\partial \theta_a}{\partial x}\right) - \alpha S\left(\theta_a - \theta_d\right) \quad (\mathrm{J/m^3 s}) \qquad \mathbf{(4)}$$

Heat transport equation in desiccant material

$$\left(1-\varepsilon\right) C_d \rho_d \frac{\partial \theta_d}{\partial t} = \left(1-\varepsilon\right) \frac{\partial}{\partial x}\left(\lambda_d \frac{\partial \theta_d}{\partial x}\right) + L\alpha' S\left(X_a - X_b\right) + \alpha S\left(\theta_a - \theta_d\right) \quad (\mathrm{J/m^3 s}) \qquad \mathbf{(5)}$$

ε : porosity (-), ρ : air density (kg/m^3), X : absolute humidity (kg/kg'), t : time (s)
x : coordinate along air passage (m), u : air velocity (m/s), λ' : water vapor conductivity {kg/ms(kg/kg')}
γ : density of desiccant material (desiccant mass per volume including air passage) (kg$_d$/m^3)
w : moisture content (kg/kg$_d$), α' : water vapor transfer coefficient on desiccant surface {kg/m^{2}s(kg/kg')}
α : convective heat transfer coefficient on desiccant surface (J/sm^{2}K)
S : Surface Area of desiccant material per volume including air passage (m^2/m^3)
$f\left(X_a, \theta_a\right)$: adsorption isotherm (kg/kg$_d$), C : specific heat (J/kgK), θ : temperature (K)
λ : thermal conductivity (J/smK), L : latent heat of vaporization (J/kg)

Suffix $\quad _a$: air $\quad _d$: desiccant material $\quad _b$: desiccant material's surface (adsorption/desorption phase)

MEASUREMENT OF THE EQUILIBRIUM MOISTURE CONTENT OF A DESICCANT WHEEL

The experimental apparatus and measuring methods adopted in this study follow the JIS's method for measuring the equilibrium moisture content of building materials [2] and the Environmental Standards of the Architectural Institute of Japan [3].

Fig. 4 shows the experimental apparatus. This apparatus is composed of a temperature-controlled bath, an air control unit, which automatically adjusts the mixing ratio of dry air and saturated air to a specified relative humidity and sends air to the measurement chamber, and an air handling unit (AHU), which supplies the air with steady temperature to the temperature-controlled bath. The weight change of the desiccant wheel during moisture adsorption or desorption was gauged by an electronic balance, and the dew-point temperature of the influx/outflux air was measured by a chilled-mirror dew-point thermometer. The temperature inside the temperature-controlled bath was measured by a PT100.

Firstly, a desiccant wheel with a diameter of 150 mm and a thickness of 100 mm was put into an oven, and dried sufficiently at 110 degrees Celsius, and then the reference dry mass was measured. Next, the desiccant wheel was put into the measurement chamber, air with a specified relative humidity of 10% was supplied, and measurement was carried out until the desiccant wheel reached the constant weight (equilibrium state). The relative humidity was increased in a stepwise manner, and the above weight measurement was conducted (adsorption process). After the specified relative humidity was increased to 95%, the relative humidity was decreased in a stepwise manner, and the above weight measurement was conducted until the specified relative humidity reached 10% (desorption process).

Fig. 5 shows measurement results of the equilibrium moisture content. Here, the moisture content higher than 45%RH for 70 degrees Celsius condition was not measured because the measuring limit of dew-point temperature was 50 degrees Celsius, and so the humidity limit was 45% in the 70 degrees Celsius condition. This has no problems because the relative humidity of the air inside the actual desiccant wheel never becomes higher than 45 %RH in the high-temperature condition.

Under the 25 degrees Celsius condition, the moisture content during the desorption process was higher than that during the adsorption process, under the medium humidity condition (20%RH-80%RH). This hysteresis phenomenon, which is commonly seen in porous materials, was observed. In addition, when the ambient temperature was set to be 70 degrees Celsius, the moisture content became lower than that under the 25 degrees Celsius condition, for the same relative humidity. This indicates that the relation between relative humidity and equilibrium moisture content varies according to temperature.

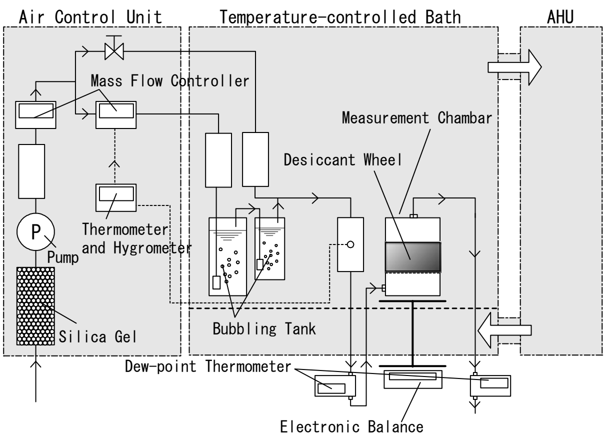

Figure 4. Experimental apparatus for measuring the equilibrium moisture content

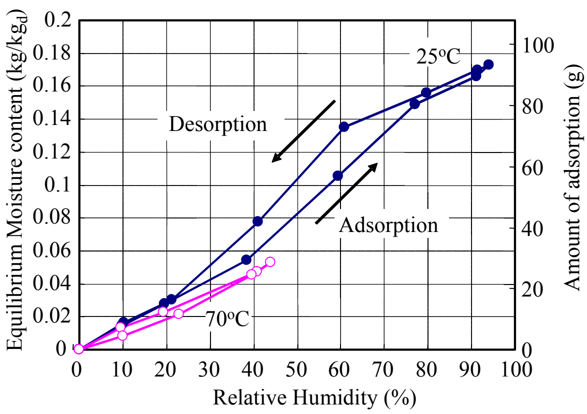

Figure 5. Adsorption isotherm

ALGORITHM OF NUMERICAL CALCULATION

The five equations shown in Table 1 are solved, to obtain the five unknown quantities: X_a, X_b, θ_a, θ_d, w. This algorithm is such that each variable is set at an initial value, Equations (1), (4), (5), and (2) are solved through time evolution, to obtain the values of X_a, θ_a, θ_d, w at the next time step, and then a convergent calculation between Equations (2) and (3) is carried out to determine w and X_b. If the explicit method was used to solve Equations (1), (4), and (5), it is necessary to discretize time extremely finely in order to obtain stable solutions, and so the full-implicit method is adopted in this study. Namely, each of Equations (1), (4), and (5) is solved with TDMA (tri-diagonal matrix algorithm) [4] and then a convergent calculation is conducted between Equations (2) and (3), to determine provisional values of X_a, X_b, θ_a, θ_d, w at the next time step. Using these updated provisional values, the above procedures are repeated with the Gauss-Seidel method [4] until convergence, to determine the values at the next time step. When the values at the next time step are determined, the time step proceeds to the next one. The control volume method is used for the discretization of space, and the upwind scheme is adopted for the advection term [4].

Fig. 6 shows the graph drawn by converting the horizontal axis in Fig. 5 from relative humidity to absolute humidity. The curves in Fig. 6 correspond to the function $f(X_a, \theta_a)$ in Equation (3). Here, the moisture content was defined as the average values of the adsorption and desorption processes, and the unmeasured temperature condition was obtained through the interpolation based on the measurement results at 25 degrees Celsius and 70 degrees Celsius. In the numerical analysis for the Equation (3), the temperature and the absolute humidity on the desiccant surface θ_d and X_b are substituted into the function that was produced by approximating adsorption isotherms in Fig. 6 with a quartic function, to obtain moisture content w in the Equation (3).

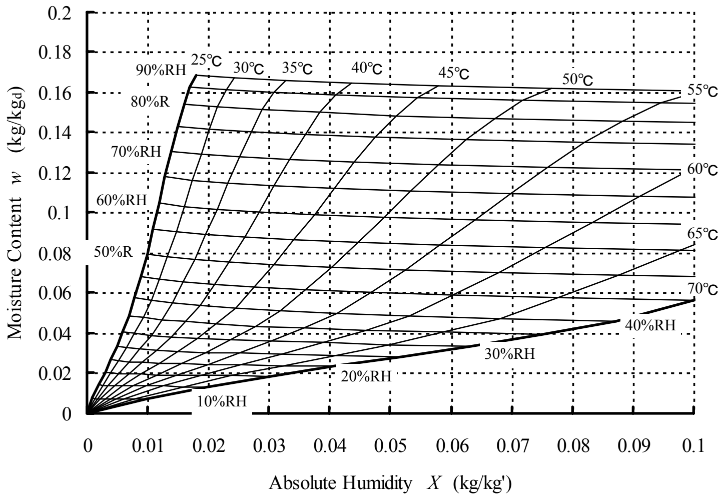

Figure 6. Adsorption isotherm used for the calculation

VALIDATION OF NUMERICAL CALCULATION

In order to check the validity of the formulated equations and the numerical calculation method for solving them, the repeated adsorption-desorption experiment was conducted for comparison. Fig. 7 shows the experimental apparatus. In the experiment, the air of 25 degrees Celsius and 70 degrees Celsius were supplied to a desiccant wheel by turns every two minutes, by operating manual dampers. The desiccant wheel with a diameter of 150 mm and a thickness of 100 mm was fixed so as not to rotate. The air volume was adjusted to 100 m^3/h with an orifice flow meter. In order to insulate heat, glass wool was wrapped around the Y pipe, the straight pipe, and the orifice. The desiccant wheel was placed on an electronic balance, separated from the straight pipe and the chamber box. The both sides were covered with lightweight, thin vinyl film, so as to prevent air leakage and not to receive loads from the both sides as much as possible. The mass of the desiccant wheel was measured with the electronic balance, and the dew-point temperatures at the upstream and the downstream of the wheel were measured with the chilled-mirror dew-point hygrometer.

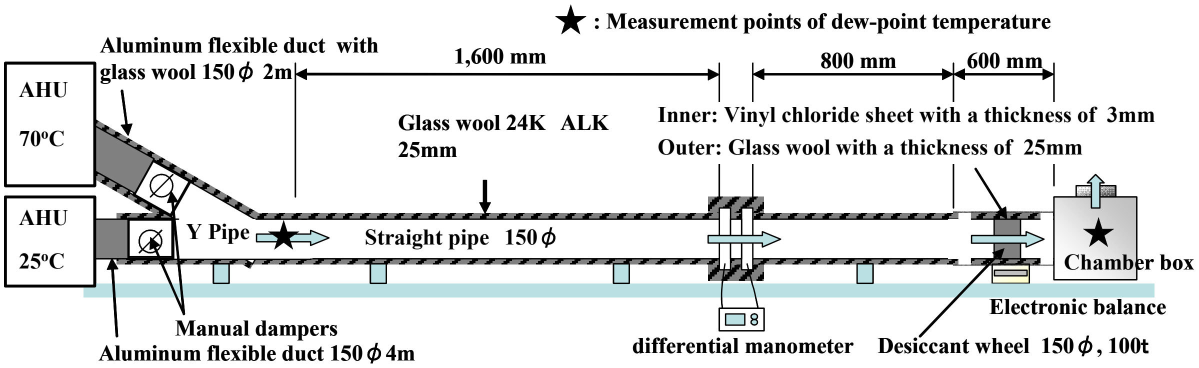

Figure 7. Experimental apparatus for repeated adsorption-desorption experiment

Table 2 tabulates properties used for the numerical calculation. For the computational grid, the 100 mm-thick element was divided equally into 20 parts, and time interval was set at 0.1 seconds.

Fig. 8 shows the calculation result of the moisture content after it reached periodic steady state, with the experimental result. The calculation reproduced the periodic change of the moisture content with the repeat of adsorption and desorption, and the calculated moisture content was within the range of the variation of the experimental data, as a whole.

Table 2. Calculation conditions

$\varepsilon = 0.71$ $\rho_d = 988$ (kg/m^3) $\alpha = 60$ (J/sm^2K) $\rho_a = 101.3/((\theta_a + 273)*0.287*(1+1.61*X_a))$ (kg/m^3)

$u = 1.163$ (m/s) $\lambda' = 0.000032$ {kg/ms (kg/kg')} $\alpha' S = 17.0$ {kg/m^3 s(kg/kg')} $S = 1180$ (m^2/m^3)

$\gamma = 289$ (kg_d/m^3) $L = 2390000$ (J/kg) $C_a = 1006$ (J/kgK) $C_d = 580$ (J/kgK)

$\lambda_a = 0.022$ (J/smK) $\lambda_d = 1.0$ (J/smK)

Inflow Boundary Condition: θ_a = 25K or 70K $X_a = 0.0069$ kg/kg'

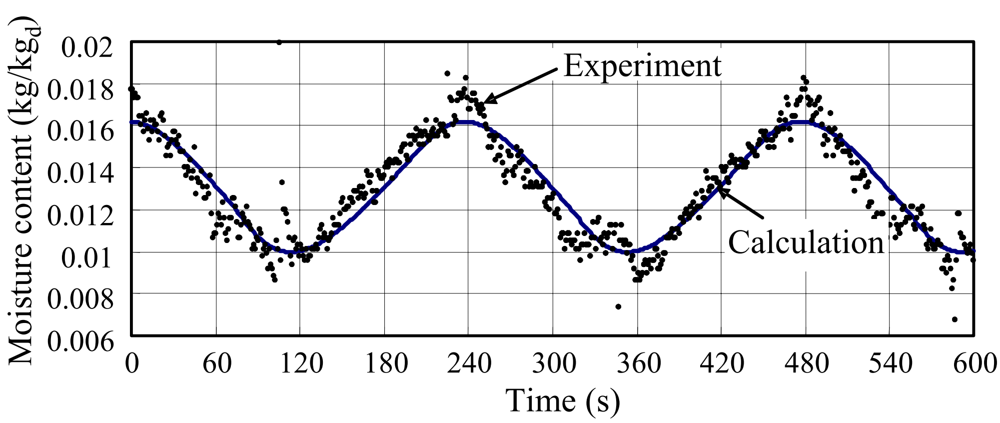

Figure 8. Periodic change of moisture content

NUMERICAL CALCULATION METHOD CONSIDERING THE ROTATION OF A WHEEL

Calculation method

As shown in Fig. 9, the wheel is segmented into each element in the rotational direction, and the above-mentioned numerical calculation is carried out separately. Each segmented element is rotated, and the flow direction and the influx temperature and humidity are switched, according to whether it is located at the adsorption side or the desorption side. By averaging the outflow temperature and humidity of all elements located at the adsorption side, it is possible to obtain the temperature and humidity of the air that has been dehumidified through the wheel at each time step. Until this averaged temperature and humidity become steady, the calculation is continued while rotating the wheel.

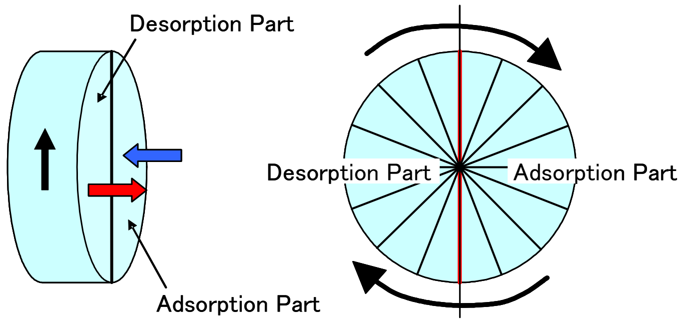

Figure 9. Calculation method for rotating desiccant wheel

Validation of Calculation method

In order to validate the numerical calculation method considering the wheel's rotation, we compared the calculation results with the experimental results for an actual desiccant machine. The actual desiccant wheel had a diameter of 300 mm and a thickness of 100 mm. Table 3 shows the experimental condition. For the inflow boundary conditions of the temperature and the humidity at the process side and the regeneration side, the measurement values were used. The air volume was set at 100 m^3/h, and the rotation speed was set at 30 RPH.

Fig. 10 plots the calculated result and measured result on a psychrometric chart. The air temperature that has passed through the wheel increased from 30 degrees Celsius to 54 degrees Celsius in the experiment, while the calculated temperature became 49 degrees Celsius, which was about 5 degrees lower than the experimental value. The absolute humidity after passing through the wheel was 0.01 kg/kg' in the experiment, while the calculated one was 0.0093 kg/kg'; these values are almost the same. Namely, the amount of dehumidification was evaluated quite accurately by the calculation. As for the regeneration side, there was also some difference in temperature between the experimental and the calculated results, but the results of the absolute humidity were almost the same. With regard to the discrepancy in temperature, this calculation method did not take the heat transfer among the segmented elements into account, but in actual wheels, it is considered that there is heat transfer at the boundary surface between the regeneration and process sides. This is expected to be improved by considering the heat transfer at the boundary surface between the regeneration and process sides in the model of the numerical calculation.

Table 3.
Calculation conditions

Geometry of desiccant wheel	300φ×100 t
Inflow Condition of process air	Temperature 30 K Absolute humidity 14.7 g/kg'
Inflow Condition of regeneration air	Temperature 70 K Absolute humidity 10.7 g/kg'
Air volume	100 m^3/h
Rotation speed	30 RPH

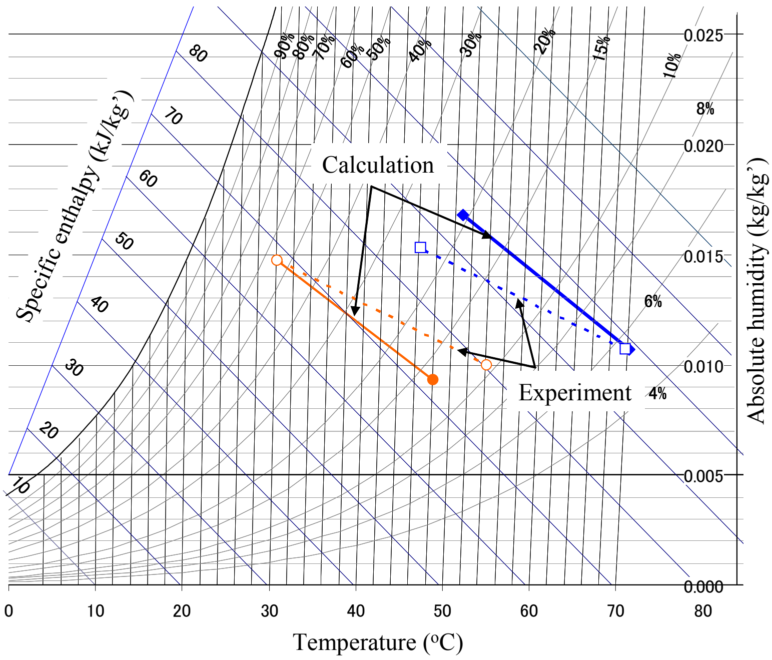

Figure 10. Comparison between calculated results and experimental results for an actual desiccant machine

CONCLUSIONS

The heat and water vapor transfer phenomena inside a desiccant wheel were formulated, and the algorithm of numerical calculation was investigated. Its validity was checked by comparing the calculation results with the results of the repeated adsorption-desorption experiment and the experiment using an actual desiccant machine. With regard to the numerical calculation targeted at the repeated adsorption-desorption experiment, the change in moisture content was accurately reproduced. However, the comparison with the experiment using an actual desiccant machine indicates that this numerical analysis could not accurately predict the air temperature after passing through the desiccant wheel, although it could evaluate the amount of dehumidification quite accurately. The prediction of air temperature is expected to be improved by considering the heat transfer at the boundary surface between the regeneration and process sides in the model of the numerical calculation.

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Controlled active mass for increased thermal comfort

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SUMMARY

The objective of this study was to evaluate the underlying physics and controlling parameters for designing a Controlled Active Mass (CAM) system. The purpose of such a system is to obtain a good thermal indoor climate and facilitate the use of energy sources with low environmental impact. The CAM system will be designed to cover the cooling demands for controlling the internal heat gains in a typical-sized office room for one person. The CAM unit will be constructed as a tank filled with water in which the level and temperature can be controlled depending on demand. Compared to other cooling systems such as passive cooling beams, the system should only be used when there is a cooling demand in the office. We also expect that the CAM system will be possible to achieve more energy savings and better indoor climate compared to other active mass systems.

INTRODUCTION

Office buildings are often subject to substantial temperature variations during the day, with the building's structure acting as an active energy storing mass. Appropriate selection of massive construction materials permits the building's energy storage behaviour to reduce the energy demand of the building.

For many years, there has been an interest in using the heat capacity of different building materials to reduce temperature fluctuations and thus improve the indoor climate and reduce energy use. IEA Annex 37 [1] “Low-exergy Systems for Heating and Cooling of Buildings” has investigated increased use of low-temperature heating systems and high temperature cooling systems in what are known as low-exergy systems. Today, there are a number of different thermally active construction elements available. Examples include the “Thermodeck”, where ventilation air passes through holes in concrete slabs that act as heat stores [2], and embedded water pipes in a flat-slab floor that works as a heat accumulating mass [3]. Similarly, there are systems for cooling buildings where water pipes have been cast into the concrete building elements in floors, ceilings or walls [4, 5, 6, 7].

IEA Annex 44 [8] “Integrating Environmentally Responsive Elements in Buildings” is concerned with various building components known as Responsive Building Elements (RBE). RBE is defined as “a building construction element that assists to maintain an appropriate balance between optimum interior conditions and environmental performance by reacting in a controlled and holistic manner to changes in the external or internal conditions and to occupant intervention”. It concentrates on five specific RBEs that seem to have the most promising potential for improvement and good take-up in the building sector: Advanced Integrated Façades, Thermal Mass, Earth Coupling, Dynamic Insulation Walls and Phase-

Change Materials. This report concentrates on thermal mass, and so does not discuss the other types further here.

Thermal Mass can be utilized in both passive and active ways. In passive thermal mass systems, the thermal energy is stored directly in the building structure hence the mass cannot be altered. When the thermal mass, such as roofs and external walls, is directly exposed to the ambient air, it is classified as external thermal mass. If it is exposed to the indoor air, it is referred to as internal mass. For good performance of passive systems, the mass should have a high heat capacity, such as that of a typical massive concrete structure, or with inner walls inside the building's thermal insulation. Compared to a building with a low thermal capacity, such a design can show an annual energy saving of about 5-10 %.

Active thermal systems, on the other hand, and known as Thermo-Active Building Systems (TABS), consist of thermally massive structures incorporating ducts or pipes for circulation of thermal-energy-transferring media. This permits the flow and the temperature of the thermal mass to be changed to suit demand, and allows the supply of energy to or from the building to be controlled. In this way the thermal mass becomes active instead of passive, and the effect of the thermal mass can be controlled.

The thermal mass of buildings is affected by ambient temperature variations as well as by temperature variations inside the building. Temperature variations over the day can be used to supply heat to (for cooling the building), or abstract it from (for heating the building), the thermal mass. These are the most important variations that the thermal mass has to cope with. Short and more fluctuating temperature variations during the day are also reduced by the thermal mass.

The objective of this study was to evaluate the underlying physics and controlling parameters for designing a Controlled Active Mass (CAM) system. The purpose of such a system is to obtain a good thermal indoor climate and facilitate the use of energy sources with low environmental impact and low exergy value. This paper will describe the definition of a CAM system and some of the underlying theory of the design process.

THEORY

The energy storage capacity of the thermal mass will work together with other energy transfer processes in the office building. Heat is gained or lost in the office through heat conduction through the building envelope, thermal radiation through windows, ventilation through openings and infiltration/exfiltration through leakage. The following subsystems are included in the office; outdoor air, building envelope, furniture, the CAM system itself, ventilation air, room air, person, computer, lights and other equipment.

Thermal mass

The building envelope, furniture and equipment in the building all contribute to thermal mass. The energy that is supplied or stored in the thermal mass will be transferred to the room air and affect the thermal climate in the building. Depending on the shape, volume, location, thermal load etc., the thermal mass will affect the indoor air differently. There are three parameters that mainly govern energy transfer from the thermal mass to the indoor air: the convective heat transfer coefficient on the surface of the thermal mass, the thermal diffusivity within the thermal mass and the specific energy storage capacity of the thermal mass.

By expression the time scales for these parameters, we can quantify how the different parameters will influence the energy transfer from the thermal mass to the room air.

The building envelope is affected by the outdoor air and contributes to the thermal mass. Temperature variations in the outdoor air will penetrate the building envelope, to a depth and at a rate that depends on the physical properties of the wall. The penetrating depth δ, of the temperature variations with frequency omega into the building envelope, is described as:

$$\delta = \sqrt{\frac{2\kappa}{\varpi}} \ ;$$

where κ is the thermal diffusivity (thermal conductivity /(density*specific heat)) of the building envelope. The effective thermal mass is either equal to the thickness of the wall d or the penetration depth δ. If the penetration depth is less than the wall thickness, then the effective thermal thickness is equal to the penetration depth.

Furniture inside the building contributes to passive thermal storage. Furniture with volume $V_{Furniture}$ and surface area $A_{Furniture}$ can be lumped into a sphere with radius d as:

$$d = 3V_{Furniture} \,/\, A_{Furniture}$$

Time scale for thermal diffusion within the thermal mass:

$$\frac{d^2}{\kappa}$$

Here d is a characteristic length of the thermal mass and κ is the thermal diffusivity of the building envelope. This is the characteristic time scale for thermal diffusion to affect mass temperature.

Time scale for convective heat transfer at the surface of the thermal mass:

$$\frac{\rho \ C \ V_{Active}}{S_{Active} \ h}$$

Here ρ is the density and C is the specific heat capacity of the air surrounding the thermal mass, V_{Active} is the volume of the thermal mass, and where S_{Active} is the surface area of the thermal mass. The heat transfer coefficient h can be found from different empirical expressions, such as the type of convection (forced, mixed or natural) at the surface of the thermal mass, and whether the surface is horizontal or vertical [9, 10]. This is the time scale for convection to affect mass temperature.

In this study the thermal mass is an active system, in which the thermal storage mass and its temperature may be varied by changing the water in the CAM tank.

Time constant for charging

$$\tau = \frac{V_{Active}}{q_W}$$

Here V_{Active} is the volume of the thermal active mass and q_w is the volume flow rate of the water filling of the CAM. This is the time it takes to charge the tank with new fresh tap water.

Temperature variations
The temperature of the outdoor and indoor air will affect the heat transfer to the thermal mass and the temperature of the incoming ventilation air will also influence the system and the thermal comfort in the office.

The outdoor air temperature T_a is described by a mean temperature T_0 with amplitude of variation of outdoor temperature ΔT_a, and frequency ϖ [Hz].

$$T_a(t) = T_0 + \Delta T_a \cos(2\pi\varpi t)$$

The characteristic time scale for variation of the outdoor temperature is the *forcing time scale* $1/\varpi$.

There are different parameters and physical phenomena that affect the indoor air temperature in the office.

Time scale for ventilation
The building must be ventilated with fresh supply air to achieve a good indoor air quality. For a room with volume V and ventilation flow rate q, the time scale for the ventilation can be expressed by the nominal time constant for the ventilation:

$$\tau_n = \frac{V}{q}$$

The time it takes to replace the air within the room is at least twice the nominal time constant.

Time scale for mixing within the room

$$\frac{V^{1/3}}{U}$$

Where U is a characteristic velocity which can be set equal to the velocity that is not experienced as a draught, say 15 cm/s, and V is the ventilation flow rate.

Time scale for convection to affect interior temperature

$$\frac{\rho\ C\ V}{S\ h}$$

where S is the room surface and ρ, C and V are properties of the thermal mass in the office building envelop. This corresponds to the energy transfer between the exterior and the indoor air.

Ratio between scales
By using the above defined time scales, we can find relationships between them and also how the different energy transfer processes are affected by each other. We can, for example, quantify how changes in outdoor air will affect the thermal energy transfer of the thermal mass.

The ratio $\frac{\varpi\rho Cd}{h}$ of the time scale for convection at the surface to the forcing time scale indicates whether there is time for significant energy to be transferred to the thermal mass before the outdoor temperature changes appreciably.

The ratio $\frac{d^2\varpi}{\kappa}$ of the time scale for diffusion to the forcing time scale indicates whether there is time for temperature variations to penetrate the thermal mass before the outdoor temperature changes appreciably.

Evaluation of the occupants' thermal comfort situation

To ensure that the CAM system creates good thermal comfort in real conditions, it is essential to focus on the occupants in the system, using thermal comfort evaluation methods originating from people. It is their comfort requirements that finally decide what type of design that will succeed in the long run. It is consequently very important to use comfort evaluation methods that are based on human reactions, and not just temperatures of surfaces and air.

It can be quite difficult to communicate the combined effects from different heat losses to and from individuals. It is therefore very useful to convert these values into something easier to understand, such as the PMV index [11, 12] for whole-body evaluation and the equivalent "experienced" temperature (t_{eq}) [13, 14]. The intentions in this study are to look at how both whole body and local influences of the positioning of the CAM system affect the subjective experience of thermal comfort.

DEFINITION AND PLANNED EVALUATION OF CAM

Here follows the definition and application of the Controlled Active Mass (CAM) system. The methods that will be used to evaluate the CAM system in an office room are presented together with the CAM variables that will be evaluated.

Definition of the CAM system

The purpose of the Controlled Active Mass system is to reduce the need for cooling in offices and at the same time to provide a good indoor climate. CAM can be seen as a combination of active and passive thermal mass systems, with which both the thermal mass and the temperature can be varied. While the CAM installation transfers cooling energy to the office area it can be seen as a passive system, since there is no media flowing in the system. On the other hand, when there is an increased need for cooling in the office, the water in the system can be replaced by new cold water to increase the cooling capacity. CAM thus becomes an active system that can be controlled depending on demand from the occupants of the building. The CAM system allows us to change both the mass and the temperature during the day. The thermal mass is changed by varying the level of filling.

The CAM system will contribute to a better thermal comfort in two ways: by reducing temperature fluctuations, due to an even distribution of cooling in the office, and also through permitting the cooling load to be changed to suit weather conditions and seasons, which will give a more adaptable thermal comfort. Many office buildings have a high cooling demand due to solar radiation through glazed facades and through transmission through walls during the summer. It may therefore be necessary to have a central cooling system connected to the ventilation unit for achieving a comfortable supply temperature. The purpose of the CAM system is not to replace the central system but rather to meet the cooling need for internal heat loads such as computers, lighting, occupants etc.

The thermal mass will be constructed as a tank filled with water, the level of which can be varied depending on demand. The temperature of the water in the tank can also be controlled and varied by changing the water in the tank with new fresh tap water. The temperature of the water in the tank can thus be altered between room air temperature and the temperature of the incoming tap water which, for Swedish conditions, varies between 4 °C and 12 °C, depending on the season and type of water source. The temperature of the incoming water to CAM must however be higher than the dewpoint of the office air in order to avoid condensation.

Application of CAM

During the evaluation the CAM system will be placed in a test room with dimensions 2.6*3.7*2.8 m (*W***L***H*), as shown in Figure 1 below. The test room corresponds to a typical-sized office room for one person. The room is well insulated, and is located inside a large test hall with a normal indoor climate, so that the climate inside the test room is not influenced by its surroundings. There are therefore no transmission losses through the test room envelope.

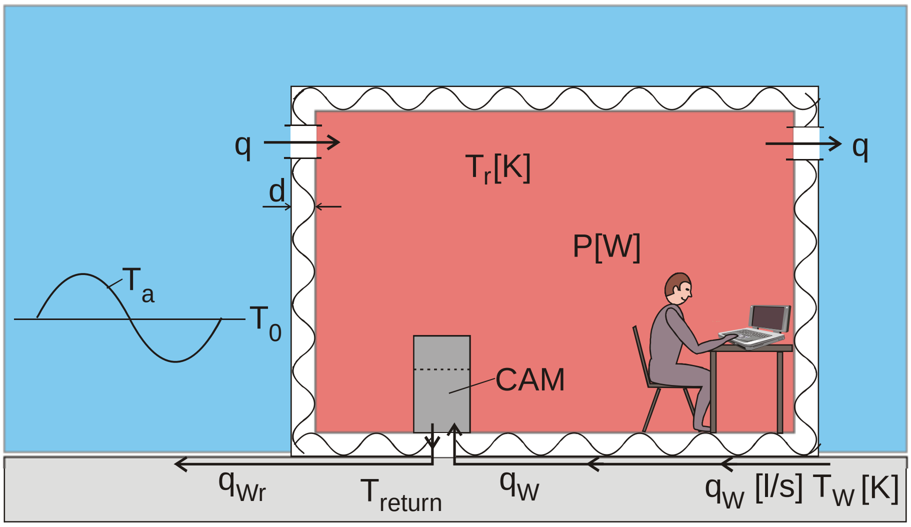

Figure 1. Test room for evaluation of CAM

In this study the CAM system will be designed to cover the cooling demands for controlling the internal heat gains from occupants, computers and lighting. For this specific case, this gives a total cooling requirement of 284 W in order to control the internal heat loads in the room. The occupant for the measurements will be simulated using a mannequin. The internal heat loads affecting the room during daytime in normal working hours, which means that cooling is needed only during these hours. The room is ventilated with a low-velocity inlet device located 2.4 m above the floor, with the exhaust opening located in the opposite wall.

Methods to be used for evaluation of CAM

Experiments and simulations will investigate how CAM will affect the thermal comfort in the room and energy-saving potential. Temperature and air velocity distributions in the office will be measured. Temperature measurements will be made using thermocouples and infrared thermograph. The velocity and turbulence levels in the room will be measured using sensors located around the mannequin and CAM. The air flows in the room will also be visualized using smoke. Measurements will also be used to determine boundary conditions that should be used as input for the simulations. The measured results will also be used for validation of the calculated values.

Simulations will be made to achieve a detailed description of the temperature and velocity distribution in the whole office room. Computational Fluid Dynamics (CFD) will be used for the numerical predictions. The same cases that have been evaluated will be simulated and a detailed comparison between the calculations and the measurements will be made. Some extra additional cases will be performed using CFD to evaluate the potential with CAM and its influence on the indoor climate.

CAM variables to be evaluated
How the CAM system influences the air movements in the office and thermal climate has to be examined. This will be made from various approaches, including the position of the CAM unit, its temperature, the amount of water in it, and when the water should be changed. When performing the CFD calculations the size, shape and the total cooling load will also be evaluated.

For the experimental evaluation, one geometrical configuration of CAM system will be examined, i.e. the volume, size and shape will be constant. In order to change the cooling capacity of CAM, both the temperature and the amount of water will be changed. Performance might also be affected by the position of the CAM unit, and so the experiments will also investigate the effect of different locations on the thermal climate in the office. It is important that the cooling need of the office should be fulfilled, but the unit should also improve occupants' thermal comfort.

DISCUSSION

CAM can be viewed as a cooling sink that should balance the thermal loads of occupants and equipment in the office. Compared to other cooling systems such as passive cooling beams, the system should be used only when there is a cooling demand. However, passive beams provide cooling all day, even though the demand may have decreased. Another advantage is that with CAM one can predict where the chilled air is released to the room and draught problems might be avoided. With beams, the airflow is more unstable and harder to predict, and draught problems might arise.

An evaluation of the CAM system is interesting, as we expect that it should be possible to achieve energy savings and better indoor climate than from other active mass systems. Compared with traditional active systems, such as systems with embedded pipes or ducts, the thermal mass is placed directly in the room, thus giving a more direct energy transfer. There is also no continuous drive energy requirement needed to transfer the energy into the building, since the cooling in the tank is exchanged only when there is a demand.

In real use of CAM systems the best solution must also take into account possible different prices for cooling and heat energy during the day or year. It is also important to know the working time schedule and the heat load from lighting, equipment, occupants etc.

ACKNOWLEDGEMENT

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Indirect evaporative cooling: performance evaluation based on the system's effectiveness

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SUMMARY

Indirect evaporative cooling (IEC) is an interesting passive cooling technique in which the extracted air is cooled by means of adiabatic humidification. By passing over an air/air heat exchanger this air cools down the supply air. A clear interaction can be observed between the relative humidity of the extracted air and the thermal comfort realized in the building.

Using data from measurements carried out in a non-residential building which makes use of this cooling technique, the influence of the room moisture balance on the thermal performance is investigated. Furthermore, dynamic simulations based on the system's effectiveness were performed using the multizone building simulation program Trnsys, in order to evaluate different parameters affecting the performance of IEC.

INTRODUCTION

The relative humidity in a room is influenced by moisture sources (people, cooking …), ventilation with outdoor air, infiltration and the exchange of moisture with walls and furniture. Hygroscopic materials such as wood and textile are able to dampen out relative humidity variations [1] [2]. Nevertheless the uncertainty of many parameters (material data, available surface …) makes it difficult to assess the relative humidity variations well and as a result most building simulation programs, e.g. Trnsys, predict the relative humidity in a simplified way [3] [4].

In indirect evaporative cooling the thermal performances depend on the indoor humidity. In the installation the return air passes over the wet side of an air/air heat exchanger (Figure 1). By adiabatic humidification the air stream is cooled. At the same time fresh air flows over the other side of the heat exchanger and cools down. This method differs from direct evaporative cooling in which the supply air stream is directly humidified. By humidifying the return air the dry bulb temperature of the supply air can be lowered without increasing its humidity ratio. In this way a more comfortable indoor climate can be obtained.

In figure 2 the states of both airstreams during the process are schematically presented making use of a Mollier chart. During humidification, the return air follows lines of equal wet bulb temperature until saturation, while the horizontal lines indicate the cooling of the supply air.

Typically, an IEC installation can operate at different stages. When the outdoor air is fresh enough in summer, the air flow rate is increased and the air is used for free cooling. As soon as the outdoor air temperature is too high, the fresh air is adiabatically cooled by moistening the heat exchanger. In some installations, active cooling is present to contribute to lower the temperature of the supply air, if the desired indoor temperature is not yet reached [5].

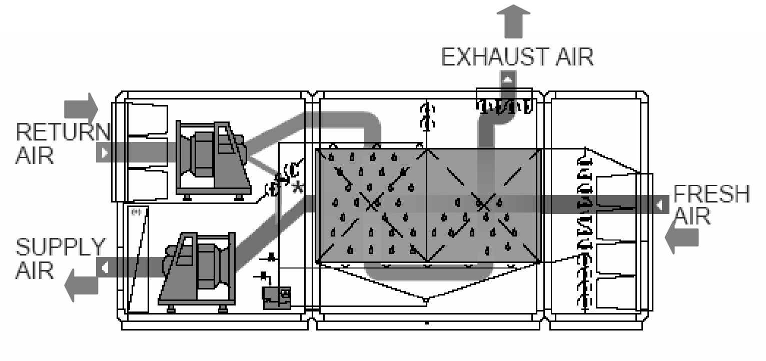

Figure 1. Indirect evaporative cooling [5]

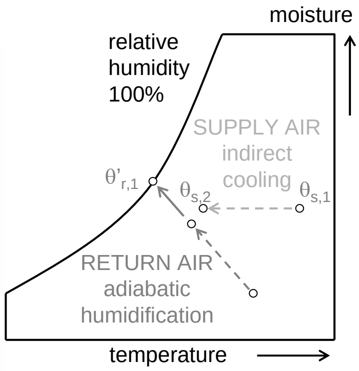

Figure 2. State of the airstreams in IEC

This paper focuses on evaluation of the system's performance. First, making use of measurement data the relation between the indoor humidity and the cooling performance is established. After, dynamic simulations will be used to evaluate various parameters influencing the thermal performance of the technique.

METHOD

A. Experimental study of hygrothermal interaction

The relative humidity of the return air plays an important role in the performance of IEC because it is moistened in the installation. Measurements carried out in non-residential building in Oostende (Belgium) where IEC was applied showed that a linear relation exist between the amount of cooling of the supply air and the difference between the dry bulb temperature of the supply air entering and the wet bulb temperature of the return air leaving the heat exchanger (Figure 3). The smaller this latter temperature difference, the less the fresh air can be cooled because saturation is reached faster when humidifying the return air. This can as also be seen in the Mollier chart (Figure 2).

The effectiveness ε of an indirect evaporative cooling system can be defined. It describes its thermal performance and is given by:

$$\varepsilon = \frac{\theta_{s,1} - \theta_{s,2}}{\theta_{s,1} - \theta'_{r,1}} \quad (1)$$

In Eq. (1) $\theta_{s,1}$ and $\theta_{s,2}$ are respectively the temperature of the supply air entering and leaving the heat exchanger. $\theta'_{r,1}$ is the wet bulb temperature of the return air entering the heat exchanger. The effectiveness thus compares the actual temperature difference realized by IEC with the maximum possible difference.

Calculated from the measurements the effectiveness of the IEC installation in Oostende is on average 82.5 %. The studied system contains a cross flow heat exchanger with total width of 900mm, a length of 1950mm and is made out of polypropylene. The maximum air flow rate of both the supply and return fan is 7100m³/h.

It should be noted that the measured points on figure 3 do not intersect the origin. It shows that a small temperature change occurring in the return air flow will not be perceived by the supply air flow. We assume that this should be explained by the heat resistance of the heat exchanger's walls and the water film that has to be overcome.

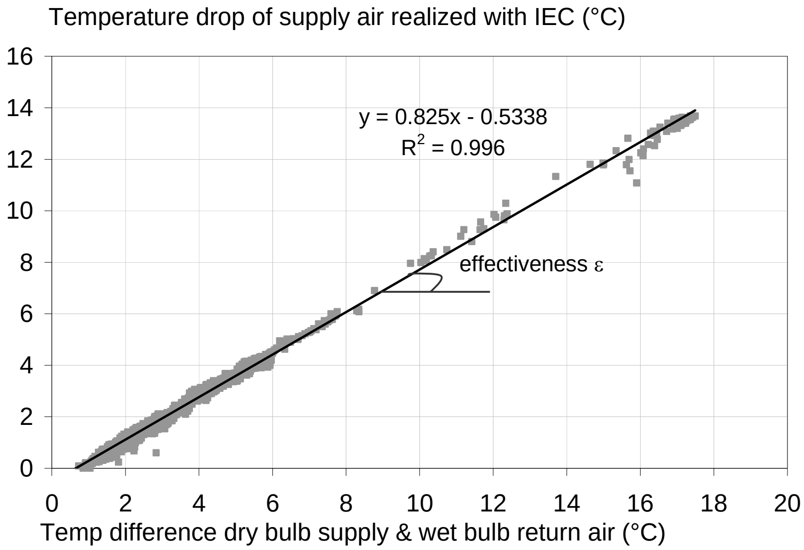

Figure 3. Interaction between indoor humidity and thermal performance (Oostende, 18/07/06 - 22/07/06)

B. Simulation model

The definition of the effectiveness can be used as a first step in studying the performance of an IEC system. As the performance of the installation depends on wet bulb temperature of the return air, the indoor temperature realized using IEC is defined both by the room heat and moisture balance. Therefore in Trnsys [4] a simple model assuming a constant effectiveness was built, which calculates the temperature of the supply air using Eq. (1) in every time step. The model has the advantage to be able to predict the indoor thermal comfort, without the need of numerical models that require a large calculation time.

As a case study, a small office model was designed (Figure 4). The office has a floor surface of 15m² and a height of 2.8m with one window of 3m² facing west, being the most disadvantageous orientation. No shading is present. The office is occupied during the office hours (08-16h) by one person (seated and light writing), who has a heat production of 120W/person (65W dry and 55W latent heat) and a moisture production of 0.07kg/h [4][6]. Apart from this, internal gains due to office equipment (140W/pc) and lights (10W/m²) are introduced.

The office is built up from a traditional heavy construction. Only the west oriented wall is an external wall, all the other boundaries are assumed to be adiabatic. For the floors and ceilings only the layers that are the most important for the storage and conductance of heat, are modelled. There is no lowered ceiling or raised floor taken into account. Due to symmetry, only half of the construction is modelled for the interior walls. For the exterior wall, the whole construction is taken into account. Tables 1 and 2 show the construction details as well as the window and material properties used.

To model the moisture buffering the simplest model from Trnsys, assuming the indoor air is lumped, is used. Buffering in the plastered walls and ceilings and in some basic furniture was taken into account [2] and the capacity ratio was calculated to be 5.

In the office model also a control system was introduced:

Day control:	Ti < 21°C	Heating setpoint 21°C (07-17h)
		Hygienic ventilation air change rate (07-18h)
	Ti > 25°C	IEC (07-18h) until Ti < 22°C: Cooling air change rate
Night control:	Ti < 16°C	Heating setpoint 16°C (17-07h)

The airflow rate for cooling is derived from cooling load calculations [7]. The hygienic ventilation rate is calculated for one person to be 36m³/h to meet the minimal ventilation requirements to obtain an IDA 2 – class in non-residential buildings [8].
Simulations were run for the entire year using a 15 min - timestep. The outdoor climate from Uccle was used in which the air temperatures have a return period of ten years [9]. The summer period extends from 01May to 30September. Weekends are not taken into account as we wanted to include all hot summer days.

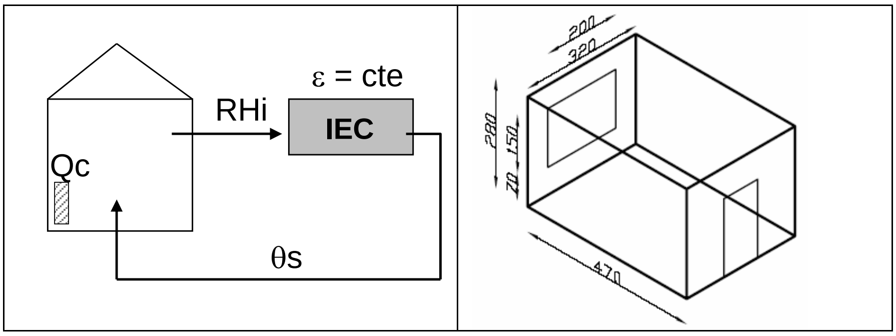

Figure 4. Scheme of the simulation model and office model in Trnsys

Table 1. Construction details and material properties

Construction	Materials	d (m)	λ (W/mK)	ρc (kJ/m³K)
Exterior wall (heavy)	Exterior bricks	0.09	0.5	1050
	Air layer	0.02	R = 0.137	/
	EPS	0.08	0.034	51
	Interior brick	0.14	0.5	1050
	Plaster	0.01	0.35	1008
Interior wall (heavy)	Plaster	0.01	0.35	1008
	Brick	0.09	0.5	1050
	Plaster	0.01	0.35	1008
Floor	Finishing carpet	0.005	0.061	260
	Light concrete	0.07	0.23	630
	(heavy) Concrete	0.10	1.7	1848
Ceiling	(heavy) Concrete	0.10	1.7	1848
	Plaster	0.10	0.35	1008
Exterior wall (light)	Aluminium	0.005	203	2376
	EPS	0.10	0.034	51
	Plaster board	0.01	0.35	840
Interior wall (light)	Plaster board	0.01	0.35	840
	EPS	0.05	0.034	51
	Plaster board	0.01	0.35	840

Table 2. Properties glass and window

4/16/4 2-ins-argon	k_{glass} (W/m²K)	g (-)	U_{window} (W/m²K)
	1.4	0.589	2.27

C. Saved cooling demand and indoor comfort

It is possible to calculate the cooling demand Q_c, that should be delivered by an active top cooling e.g. a ventilo-convector placed in the office, in order to accomplish a maximum indoor air temperature of 26°C during all office hours.

$$Q_c = 0.34\, nV(\theta_i - 26°C) + \frac{h_{ev}\rho_a\, \xi\, nV}{3600}(p_i - p_{sat}(\theta_i)) \quad (2)$$

In Eq. (2) h_{ev} is the latent heat of evaporation (2.5 10^6 J/kg), n is the cooling air change rate (h^{-1}), V is the office volume and ξ is the specific vapour capacity of air (6.1 10^{-6}kg/kg/Pa). As the heat exchanger can be bypassed in the described installation (Figure 1), thus allowing free cooling with outdoor air, the cooling demand that is saved using IEC in comparison to free cooling can be calculated:

$$Q_{c,saved} = Q_{c,FreeCooling} - Q_{c,IEC} \quad (3)$$

The indoor summer comfort is evaluated using the Adaptive Temperature Limits Indicator (ATG) which takes into account the thermal adaptation of occupants to the indoor climate [10] [11]. The method differs between alpha and beta - buildings. The evaluation of the beta building is more severe and is used for our office model. It assumes occupants cannot affect the indoor climate e.g. by use of opening windows). The ATG method divides thermal comfort into three levels. It is required that a building meets the standard level B which corresponds to 80% thermal acceptability to have a 'good' indoor thermal comfort in the period studied. Levels A and C correspond respectively with 65% and 90% acceptability.

In figure 5 the evaluation of the indoor climate using this method is shown for a specific simulation case as an example. On the horizontal axis the running mean outdoor temperature is presented $T_{e,ref}$, which is derived from the external temperature from the current day and that of the three preceding days. The vertical axis shows the indoor operative temperature $T_{i,o}$.

Figure 5 shows that in this specific case a good indoor comfort is expected at average outdoor temperatures. During some hot days ($T_{e,\,ref}$> 21°C), some peaks can be found in the indoor climate, during witch the comfort criteria are not fulfilled.

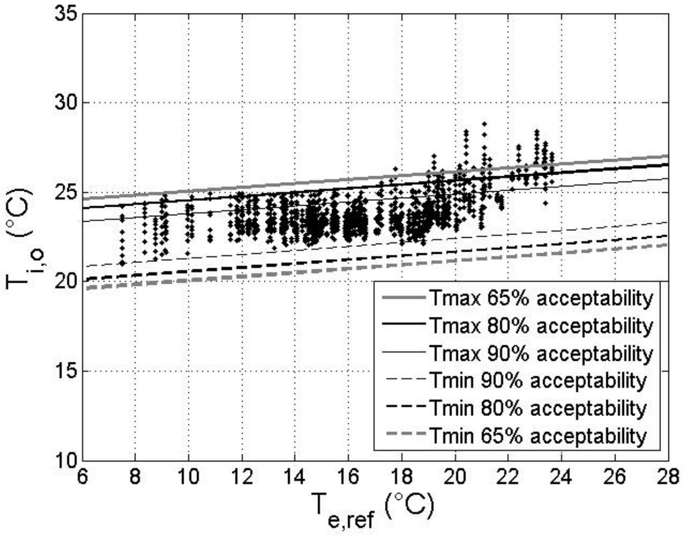

Figure 5. Indoor comfort evaluated using the ATG – method for a beta building with an IEC system: effectiveness 85% and cooling air change rate 7h^{-1} (1May-30Sept: 08-16h).

RESULTS

The cooling air change rate was calculated to be 9h^{-1} (equals 380m³/h). Simulations, assuming an IEC installation with a constant effectiveness of 85%, are compared with simulations in which the heat exchanger in the IEC installation is bypassed (free cooling). Furthermore a case, in which no cooling at all is present, is shown. In this case the hygienic ventilation was working constantly during the office hours.

In figure 6 the percentage of the office hours in which respectively an indoor class A, B and C is obtained, and, in which the IEC/free cooling is working, are given. The cooling demand, which is saved making use of IEC instead of free cooling, is calculated with Eq.(2) and (3).

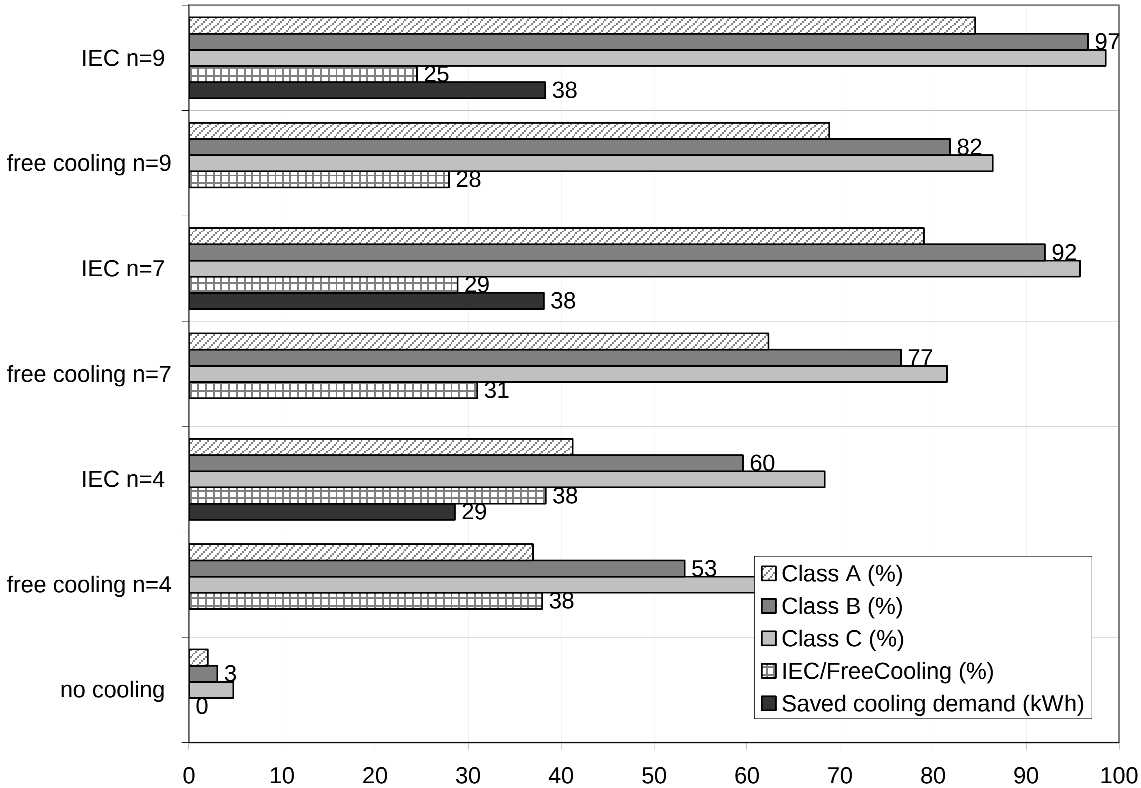

Figure 6. Influence of air change rate on indoor comfort and saved cooling demand

We can conclude that a sufficiently good indoor climate is obtained when applying IEC with a cooling air change rate of 9h^{-1}: during 97% of the occupancy hours the climate is situated in class B. By lowering the cooling air change rate respectively to 7h^{-1} and 4h^{-1}, this value drops with respectively 5% and 37%. The comfort obtained by using IEC is always better than when applying only free cooling. No cooling at all only 3% of the time good comfort is predicted. The saved cooling demand decreases when lowering the air change rate. In case of a cooling air change rate of 7h^{-1} 38 kWh can be saved, while 6.5 kWh would still be necessary to obtain an indoor temperature of maximum 26°C during all the office hours. Free cooling and IEC work for almost the same time in the respective cases.

Apart from this, other parameters influencing the thermal performance of IEC are evaluated. Starting from an IEC system which has an effectiveness of 85%, figure 8 shows which parameters have been changed in the different cases. In all the cases the cooling air change rate is $7h^{-1}$, other parameters are kept constant.

- Changing the systems effectiveness only has a small influence on the performance of IEC. The obtained comfort and the cooling demand saved are similar for an effectiveness equal to 70%, 85% and 95%. Logically, an installation with a higher effectiveness will perform slightly better, as lower supply temperatures can be obtained.
- Increasing the indoor moisture production has a large influence on the room's moisture balance. One person produces about 0.07 kg/h moisture at rest, but other gains such as bathing, washing, … can be introduced (e.g. in health care). Enlarging the intern gains to 2.5kg/h, the time in which comfort is situated in class B reduces with about half. Furthermore, IEC is working 10% of the time more. Apart from a sensible part, also a latent part is present in the cooling demand, being respectively 82% and 18% of the total cooling demand (102kWh).
  In this case the cooling demand for IEC is larger than when applying free cooling. This can be explained by the large internal moisture production, for which the wet bulb temperature of the return air will be larger than the outdoor air temperature. This means the outdoor air will be heated by the heat exchanger instead of cooled which results in higher supply temperatures. In practise the heat exchanger may be bypassed in these situations to allow free cooling with outdoor air.
- Doubling the moisture buffering capacity has almost no influence on the systems performance. This can be explained by the fact that moisture buffering does not have a large effect on the average relative humidity, but rather dampens out the variations. This is also concluded in [2] by experiments.
- In case the office is built up from a light construction the comfort decreases with 6% of the time (class B). The saved cooling demand is in this case 51kWh, compared to 38kWh in case of a heavy construction type. At the other hand, IEC is active during 4% of the time more. Construction and material details can be found in Table 1.

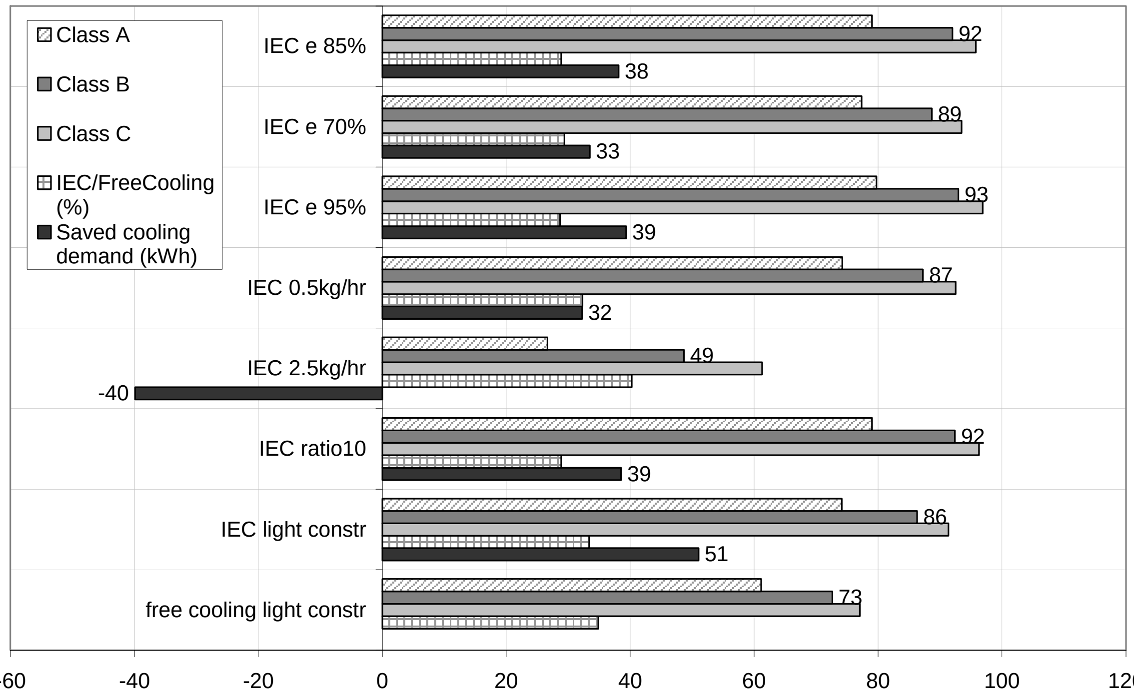

Figure 7. Influence of effectiveness, indoor moisture production and buffering, and construction type on indoor comfort and saved cooling demand ($n=7h^{-1}$)

CONCLUSION

Indirect evaporative cooling is an interesting passive cooling technique in which the thermal performance depends mainly on the indoor humidity. The evaluation of measurement data shows that a linear relation exists between the temperature drop of the outdoor air and the maximum cooling amount that is possible with IEC. Dynamic simulations based on the system’s constant effectiveness showed that the knowledge of the indoor moisture balance is essential to predict the performance of indirect evaporative cooling well.

The indoor moisture production has the most important influence on the system’s thermal performance. On the other hand, the amount of moisture buffering material in the room rather influences the stability of the climate and does not have a large effect on the performances of IEC. Furthermore the moisture balance is also determined by the ventilation air change rate. Lowering the cooling air change rate decreases the established indoor comfort. In the studied installation, varying the system’s effectiveness had only a limited influence on the obtained thermal comfort. Applying the technique in a building with a light construction will reduce the comfort but increases the energy saved in comparison to free cooling.

ACKNOWLEDGEMENT

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The Experimental Works And Some Parametric Investigations of Thermally Activated Desiccant Cooling System

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SUMMARY

This paper presents the study of the desiccant based cooling system in the laboratory. The main purpose of the study is to determine the performance of desiccant system components. Also, using some results from the experimental works, preliminary parametric investigation in TRNSYS had been done. The investigation is centered on the application of the system in different climatic conditions. The results showed the dependence of the desiccant system upon the performance of its components and the climatic conditions.

INTRODUCTION

Desiccant based dehumidification and cooling system is envisioned as an alternative to the refrigerant based cooling system. The main advantage of the desiccant based system is that it can be directly operated using renewable thermal energy sources compared to the refrigerant based which is electrically operated. Several studies are on-going for the application of desiccant system in actual situations. This paper presents the studies conducted in our laboratory regarding this technology. The study is focused on the performance evaluation of the system components including some preliminary modeling and parametric investigations.

EXPERIMENTAL FACILITY AND EVALUATION

Experimental Facility

The thermal conversion facility was constructed to simulate the outdoor and indoor air conditions. The main purpose of the said facility is to test and investigate the performance of the desiccant based system. The experimental facility has a physical set-up shown in Figure 1. It has two controlled chambers in which Chamber A is used to simulate the outdoor air condition and Chamber B for the indoor air condition. The first chamber (A) has temperature range from -10^{0}C to 40^{0}C with accuracy of 2%. The humidity could be varied depending on the needed condition. The accuracy of maintaining the set-value of the air humidity content is within 10%. For Chamber B, the operating temperature range is from 10^{0}C to 40^{0}C with accuracy of within 1%. The air humidity content could be varied as in Chamber A. Its accuracy is within 1%.

The technical description of the facility as shown in Figure 2.The source of raw air for both controlled chambers is from ambient air. It is forced to flow into the chambers by fans. If the temperature set-value is lower than that of the ambient air, air cooling operates. On the other hand, if the set-value of the needed air temperature is higher than that of the ambient, air heater operates. Seemingly similar is the situation for maintaining the air humidity content. If the ambient air humidity is high, dehumidification process operates. And when the ambient air humidity content is lower compared to the set-value the humidifier operates. The conditioned air is mixed in the controlled chambers before supplying it to the testing chamber. The testing chamber is located between the two controlled chambers. The air is moved from the chambers into the testing chambers using fans attached to the ducting system. In the testing chamber any kind of air handling unit and heat exchanger can be tested and evaluated. For this study, the desiccant system was evaluated.

Experimental Evaluation of Desiccant System

The main components of the desiccant based system consist of desiccant dehumidifier, heat recovery wheel, desiccant heater and evaporative cooler. The performance of the whole desiccant cooling system is dependent on the performance of its components. To determine the performance of its components, the performance evaluations of the desiccant wheel and heat wheel were under taken.

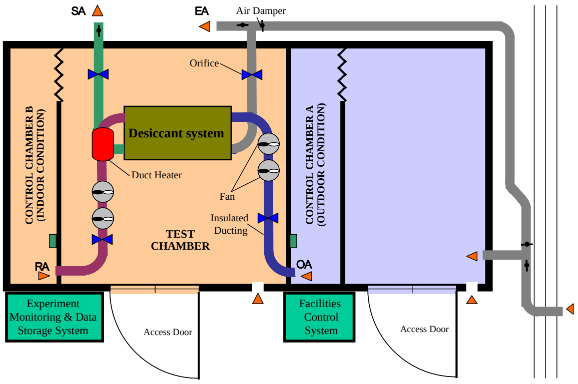


Figure 1. Experimental Facility.

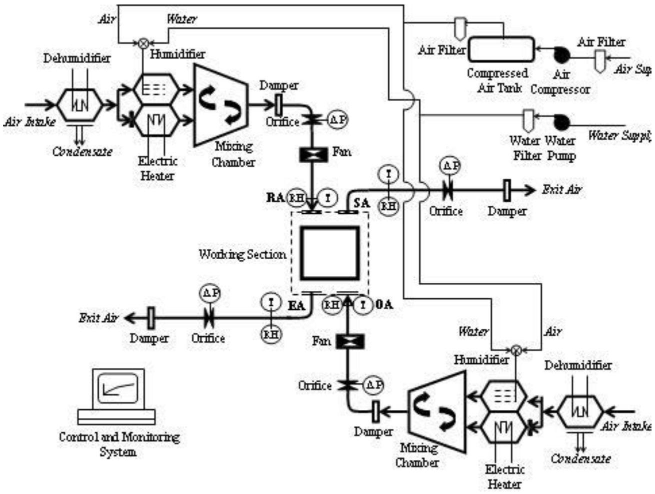


Figure 2. Technical description of the experimental facility.

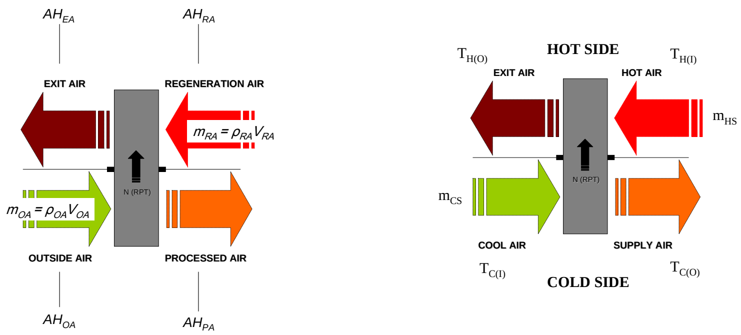


Figure 3. The schematic diagram of the desiccant wheel (left) and heat wheel (right).

Desiccant Wheel Evaluation

The desiccant wheel subjected to evaluation is a Silica-Gel coated wheel. It has 300mm external diameter with 100mm depth. The parameters considered for the evaluation were the rate of volumetric flow, the regeneration temperature and the wheel rotational speed. The considered flow rates were $100m^3/hr$ and $200m^3/hr$. The regeneration temperatures applied were 60^0C, 70^0C and 80^0C. For $100m^3/hr$ the wheel rotational speeds were 5, 10, 15, 20, 25, 30, 35, 40, 50 & 60 RPH and for $200m^3/hr$ the rotational speeds were 5, 10, 20, 30, 40, 50, 60, 70, 80 & 90 RPH. The selection of different wheel speed for the different flow rates was based on the initial test run that as the flow rate increases the optimum speed of the wheel increases. Figure 3 shows the schematic diagram of the wheel and the governing performances were based on NREL testing manual [1]. Based on the manual, the dehumidification performance of the desiccant wheel is based on moisture removal capacity or MRC. It is expressed as,

$$MRC = m_{OA}(AH_{OA} - AH_{PA}) \tag{1}$$

The amount of moisture removal capacity or sorption rate is the same at the regeneration side moisture regeneration rate which is the moisture removal regeneration (MRR) expressed as,

$$MRR = m_{RA}(AH_{EA} - AH_{RA}) \tag{2}$$

To evaluate the characteristic and performance of the experiment, the moisture mass balance (MBB) determined the quality of gathered data and thus the MBB is a checking factor and expressed as,

$$MMB = \frac{MRC}{MRR} \tag{3}$$

For acceptable accuracy of gathered data, the ratio of MMB should be within 0.5 to 1.5. To be within the limits of acceptable accuracy improvements of the experimental conditions and instrumentations are highly necessary. And the nomenclature, m_{OA} and m_{RA} are the mass flow rates (outside air and return air), kg/s. *AH* is absolute humidity content of air, g/kg. *OA*, *PA*, *EA* and *RA* are the outside air, processed air, exit air and return air.

Heat Wheel Evaluation

The evaluated heat wheel is coated with Silicone-Acrylic Compound. The physical appearance and dimension of the wheel is the same as the desiccant wheel. However, the main purpose of the heat wheel is for sensible heat recovery only. In the evaluation, the moisture content of the air was not considered. To create temperature difference between the two counter flowing air streams in the wheel faces, air stream at cold side was maintained at 25^0C and the air stream at hot side 50^0C. For properly designed heat exchanger, variation of air streams temperature was found to have no effect on its performance. Therefore only these two temperatures were considered and the schematic diagram of the heat wheel is shown in Figure 3.

The other parameters used in the evaluation were the volumetric flow rates and the wheel rotational speed. The considered flow rates were $100m^3$/hr and $200m^3$/hr. The wheel rotational speeds were 2.5, 5, 10, 15 and 20 RPM. And, the performance evaluation of the heat wheel was based on its effectiveness in transferring sensible heat shown as [2],

$$Eff_{Average} = \frac{m_{CS}(T_{C(O)} - T_{C(I)}) + m_{HS}(T_{H(I)} - T_{H(O)})}{2m_{Minimum}(T_{H(I)} - T_{C(I)})} \quad (4)$$

For the nomenclature, the $Eff_{Average}$ is the average effectiveness of the heat wheel. The m_{CS} and m_{HS} are the mass flow rates (hot and cold sides), kg/s. $T_{C(I)}$ and $T_{C(O)}$ are temperature of air in the cold side (inlet and outlet), ^{0}C. $T_{H(I)}$ and $T_{H(O)}$ are temperature of air in the hot side inlet and outlet, ^{0}C. And, $m_{Minimum}$ is the minimum flow rate of either cold or hot side, kg/s.

EXPERIMENTAL RESULTS AND DISCUSSION

Desiccant Wheel

Using equations 1, 2 and 3 in analyzing the gathered data and plotted below (Figure 4), the results show that the wheel performance is basically dependent upon its rotational speed. In the case of $100m^3$/hr the optimum speed is 30RPH and for the $200m^3$/hr flow rate the optimum speed is 50RPH. Meaning for the same type and dimension of wheel, increasing the flow rate will also increase the optimum wheel speed. The optimum speed is the same for different regeneration temperature. However, as the regeneration temperature increases, the moisture adsorption increases. At optimum speed, for the case of $100m^3$/hr, the increase of moisture adsorption rate from 60^0C to 70^0C is 13% and from 70^0C to 80^0C the increase is 6 percent. In the case of $200m^3$/hr flow rate the increase of adsorption rate from 60^0C to 70^0C is 14% and from 70^0C to 80^0C the increase is 5%. So, the amount of the regeneration temperature is another factor in the effectiveness of desiccant dehumidifier in moisture adsorption. However, the increase of moisture adsorption rate becomes lower as the regeneration temperature becomes higher. Based on the moisture mass balance presented in Figure 4 the gathered and analyzed data are within the range of 0.95 to 1.05. Thus experimental evaluation is within the accuracy imposed by NREL.

Heat Wheel

For the heat wheel the results of the experimental study are presented in Figure 5. For $100m^3$/hr flow rate, the effectiveness approached 81% for wheel speed of 20RPM and for the $200m^3$/hr the effectiveness approached 69% at 20RPM. Based on the results, the performance of the heat wheel is dependent on the wheel rotational speed and on the amount of flow rate. Results show that the wheel can be operated at 20 RPM as maximum rotational wheel speed.

MODELLING AND PARAMETRIC INVESTIGATION

Modeling

Preliminary modeling of the desiccant based system was implemented in TRNSYS [3]. Standard components were used such as the desiccant dehumidifier, the heat wheel and evaporative coolers. In the TRNSYS, the model of the desiccant dehumidifier is based on the analogy between the heat transfer of the sensible heat exchanger and of the desiccant heat and mass transfer. The input parameters of the model are the two effectiveness values proposed by P.J. Banks which is discussed in TRNSYS Manual. In this study, experimental results of the performance of the desiccant wheel were available. However, it could not be used directly to the model of the desiccant wheel. Thus the comparison between the experimental performance of the desiccant wheel and the model in TRNSYS was done through series of trials in finding the two effectivenesses that can have almost similar result to the experimental works. In the future study, this limitation will be addressed later. The modeling of heat recovery wheel is based on the sensible heat transfer from two streams of air in different temperatures. The actual effectiveness from the experimental work was used in the model. For the evaporative cooler, the modeling was based on effectiveness in reducing the dry bulb temperature to the maximum of its web bulb temperature. Based on the study cited by Daou, et al. 2006 [4], the maximum effectiveness of direct evaporative cooler is 90% and of the indirect evaporative cooler 80%. These parameters were used in the modeling since the current experimental set-up did not include experimental investigations in these devices. In general, the above considerations were used in the modeling and parametric investigation. Thus some deviations occurred in the results when compared with the experimental works.

Figure 6 shows the schematic diagram of the experimental unit which was under experimental investigation. This set-up was also used for comparison with the modeling done in TRNSYS. As stated above, since the model of the desiccant wheel in TRNSYS has no provision for direct comparison with experimental results, series of trials had been done. The finest trials that can be reached are presented in Table 1. Based on the results, the comparisons are within 10 percent. However, some points such as of points 2, 3 and 7 of the relative humidity content, the results are more than 10 percent. These points are in the desiccant wheel. As stated above, since the TRNSYS desiccant model cannot have direct comparison with the experimental results, higher deviation in these states was expected. This is the limitation of TRNSYS and should be subject for more improvements to arrive at a good agreement between the experimental and the simulated results. Considering this limitation, some parametric investigations were done for testing the capability of the model when applied in different conditions.

Simulation

The parametric investigations for the application of the desiccant cooling system was implemented in different climatic conditions; for temperate climate - little bit warm and humid, for the Mediterranean - warm and humid and, of the tropical climate - relatively hot and humid for all-year-round. Table 2 shows the temperature and relative humidity for the countries representing the three different climatic conditions used in the study. Two models of the desiccant cooling system were used for the investigation: the standard desiccant cooling system shown in Figure 7 and, the desiccant cooling cycle with pre-cooling of outdoor air prior to the desiccant wheel shown in Figure 8.

PARAMETRIC INVESTIGATION RESULTS AND DISCUSSION

Using the performance data, stated above, regarding the comparison between the experimental results and modeling and, applied in different climatic conditions the results are presented in Table 3. Based on the presented results the standard desiccant cycle can operate effectively in temperate and Mediterranean climates. However, for the Mediterranean climate, the needed regeneration air temperature (RegA) is high, more than 100^0C. For tropical climate, the standard cycle cannot operate and supply comfortable air condition since the sensible load of air is higher even though the air latent load is lower. On the other hand, the cycle with pre-cooling may attain the same capability in temperate and Mediterranean climates and comparable to standard cycle with lower supply air and regeneration temperature (only small difference). However, in the tropical areas, the system is still short of supplying comfortable air condition as shown in the condition of the supply air temperature. But, by using much more efficient pre-cooling devices such as borehole heat exchanger, absorption chiller and others, the system performance could be further improved as shown in graphical presentation, Figure 9.

Based on this presentation, the desiccant adsorption process increased when the air temperature entering the desiccant wheel decreased. Thus cooling the outdoor air to near saturation may increase the dehumidification performance of the desiccant wheel. However, further study should be done to determine the over-all performance of the system.   Some research works conceptualized the pre-cooling effect intended for hybrid desiccant cooling system [5].

SUMMARY AND CONCLUSIONS

Experimental works and some parametric investigations were conducted. Based on the results, the performance of the desiccant cooling system is dependent on the intrinsic and

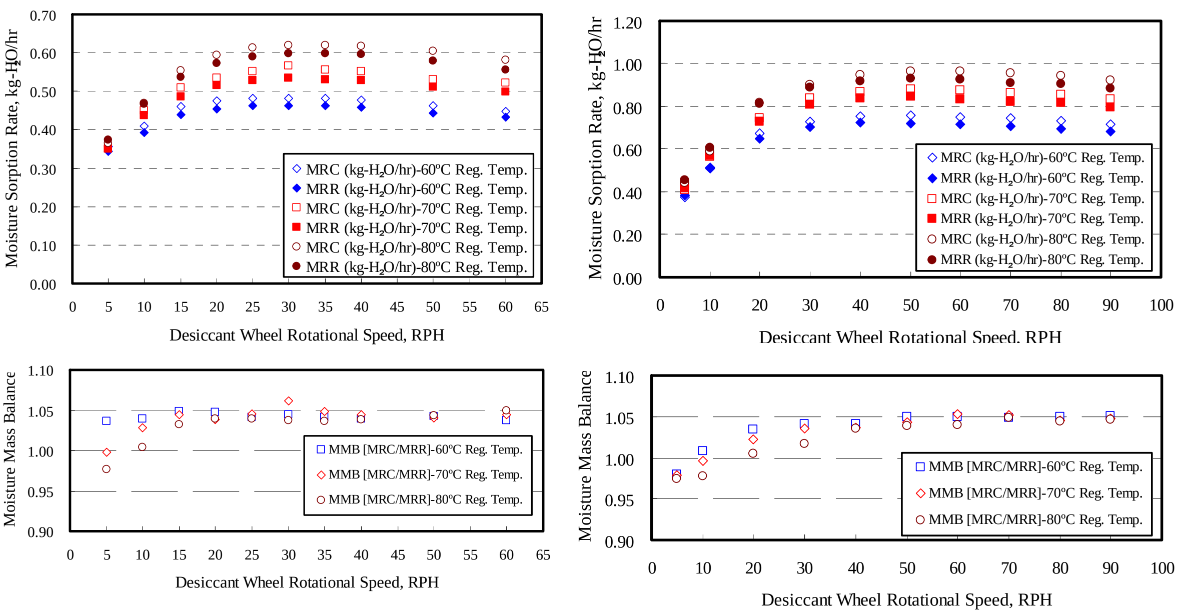


Figure 4. The sorption capacity of desiccant wheel.

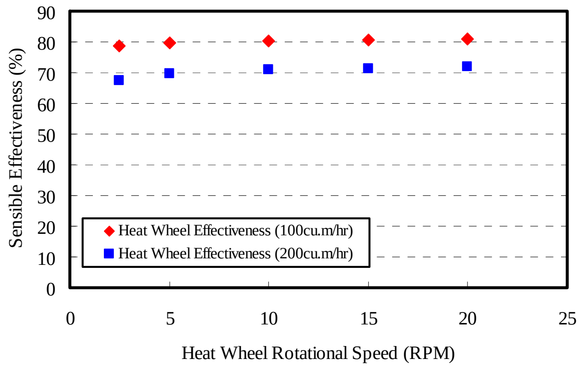


Figure 5. The Heat Wheel Effectiveness.

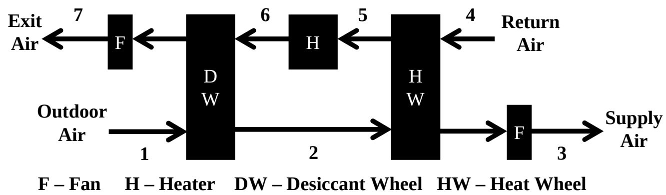


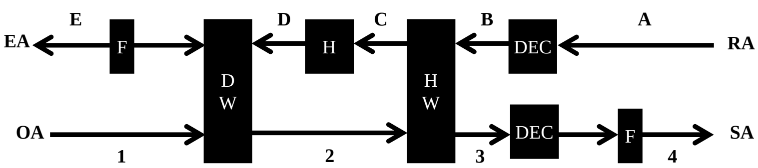
Figure 6. Schematic diagram of the first unit desiccant cooling system.

Table 1. Results of the experimental works and of the simulation (100m^3/hr).

STATE POINT	1	2	3	4	5	6	7
Simulation (T) [ºC]	30.0	55.3	31.7	26.0	49.4	82.3	56.8
Experimental (T) [ºC]	30.8	59.2	33.5	26.1	52.6	80.0	51.9
% Difference	2.5	6.6	5.4	0.4	6.1	2.8	8.7
Simulation (RH) [%]	60.0	7.1	24.1	55.0	15.4	3.6	18.9
Experimental (RH) [%]	58.5	9.1	35.3	55.5	13.3	3.9	20.1
% Difference	2.5	22.5	31.7	0.9	13.6	8.7	6.0

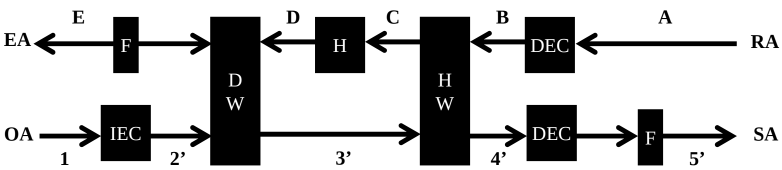
Table 2. The different climatic conditions for parametric investigation.

CLIMATE	COUNTRY	DBT (T) [ºC	RH [%]	AH [g/kg]
TEMPERATE (Warm and Less Humid)	JAPAN (Tokyo)	30	60	16
MEDITERRANEAN (Warm and Humid)	ITALY (Palermo)	30	70	19
TROPICAL (Hot and Very Humid)	PHILIPPINES (Manila)	35	75	27



DEC – Direct Evaporative Cooler

Figure 7. Standard and common desiccant cooling system



IEC – Indirect Evaporative Cooler

Figure 8. The standard desiccant cooling system with pre-cooling.

Table 3. The simulation results between the standard desiccant cycle (Figure 7) and with pre-cooling of outdoor air prior desiccant wheel (Figure 8).

DESICCANT COOLING SYSTEM	CLIMATE	OA(T) [ºC]	OA(RH) [%]	SA (T) [ºC]	SA(RH) [%]	SA(AH) [g/kg]	RegA(T) [ºC]
Model 1 (Standard Cycle)	Temperate	30	60	21.60	57.30	9.20	85.50
Model 2 (With Pre-Cooling Cycle)				21.20	58.20	9.12	82.60
Model 1 (Standard Cycle)	Mideterranean	30	70	22.70	54.90	9.44	98.60
Model 2 (With Pre-Cooling Cycle)				22.40	55.50	9.37	96.40
Model 1 (Standard Cycle)	Tropical	35	75	27.10	46.10	10.32	149.00
Model 2 (With Pre-Cooling Cycle)				26.70	46.80	10.23	146.00

* The Return Air (RA) condition was maintained at 26^0C and 55%.

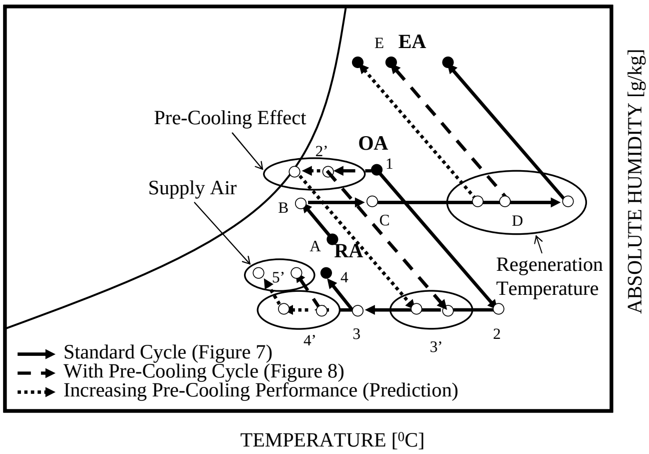


Figure 9. Effect of increasing the performance of pre-cooling prior desiccant wheel.

extrinsic parameters such as effectiveness of its components and outdoor air condition. For practical application of the system in real conditions, further investigation and analyses are imperative and thus additional and thorough research is important. Further investigation of the desiccant cooling system will be the next direction of this research works.

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Performance model for small scale indirect evaporative Cooler.

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SUMMARY

Indirect evaporative cooling an efficient and cheap way to reduce the heat load in a building. For an innovative small scale evaporative cooler (400 m^3/h) a model is made to predict the outlet temperature and cooling capacity for different environmental conditions in order to estimate the seasonal heat load reduction. The cooler is a compact counter flow heat exchanger with louver fin geometry on either side; the secondary side is coated with a water absorbing material. Due to the direct evaporation from the surface and the counter flow operation outlet temperatures below the wet bulb temperature can be achieved. The model uses Chilton & Colburn approximation for the evaporation, an approximation for the j-factor as found by Chang and Wang. The resulting model is based on estimated geometry properties and predicts the outlet temperature for given inlet conditions an airflow rates. The model predicts an over performance of the output temperature. For the inlet temperature below 24°C there a 20 % overshoot of the model, for higher inlet temperatures the overshoot is less than 10%.

INTRODUCTION

The demand for cooling increases due to higher comfort demand, compact way of building, higher internal heat load ect. For small scale building as primary schools, small utility buildings a demand for a simple way of cooling is desirable. Evaporative (adiabatic) cooling is an energy efficient way of cooling, though it has a drawback that with increasing outdoor humidity the cooling capacity decreases. In this paper a performance model for an indirect evaporative cooling is described. The purpose of the model is to use it in order to predict the effective cooling performance for use in building performance studies.

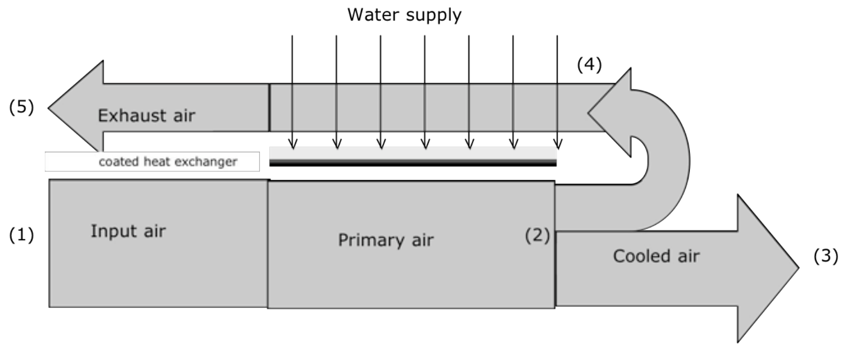


Figure 1. Schematic operation of the indirect adiabatic cooler

Operation

The main advantage of indirect adiabatic cooling is that the water content of the cooled air is not increased. The studied cooler is an innovative design where efficient use is made of counter flow geometry and of direct evaporation of water from the separating wall. The principle operation is given in fig.1.

On the primary side of the heat exchanger outside air (1) is cooled to outlet conditions. About 1/3rd of the air (2) is fed back to the secondary side (4). The secondary side of the heat exchanger has a hygroscopic coating, saturated with water. With regular intervals (typically 10min.) fresh water is sprayed onto the coating. Due to the direct evaporation from the wall a counter flow operation the outlet temperature (3) approximates the dew temperature of the inlet air. The secondary outlet air (5) is exhausted to the environment.

Construction

The heat exchanger consists of fin geometry according to Figure 3. The copper fins (primary and secondary) are glued back to back on a thin plastic foil. The separation plates between the fins achieve construction stability. This results in a very lightweight heat exchanger that after service life easily can be replaced.

Objective.

Measurements of the performance of the cooler have show that the outlet temperatures of the cooler are very near the dew point of the inlet air. This performance is due to the effective counter flow operation and the direct vaporization of water from the coated fins. In order to get insight in the performance of the cooler a model is made to relate the convective heat transfer on the primary side to the combined heat and mass transfer on the secondary side. The mathematical model can be used to enhance the performance of the cooler for different configurations and also can be used to predict the performance of the cooler

METHOD

The main principle of the cooler is the evaporation of water on the secondary side. The heat transfer consists of a convective part and a mass transfer part due to the evaporation. An enthalpy balance is set up for a surface element in the heat exchanger (Figure 2).

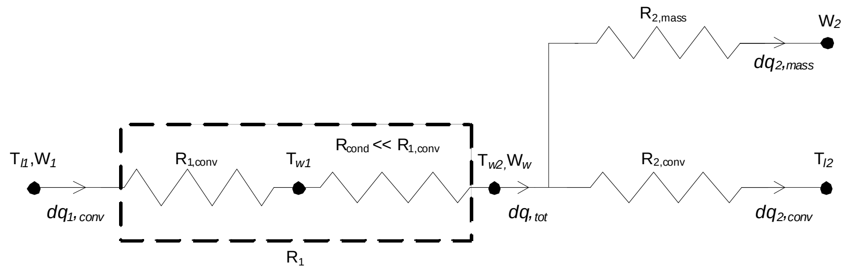


Figure 2. Model of the enthalpy exchange for a surface element of the heat exchanger.

In order to find a solution for (1) approximations have to be found for the convective heat and convective mass transfer coefficients h_c, h_m.

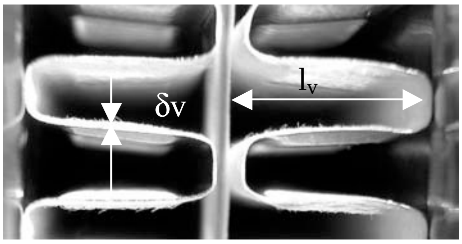
$$d(T_1,T_2,w_1,w_2)\ of\ dq_{tot} = f(\underbrace{\dot{V}_1,\dot{V}_2}_{\text{Operational Flows}}, \underbrace{A_c,A_m}_{\text{constrution}}, \underbrace{T_1,T_2,w_1,w_2}_{\text{Proces variables}}, \underbrace{\rho_l,c_{pl},c_{pw,g},c_{pw,f},h_{fg}}_{\text{material properties}}, \underbrace{h_c,h_m}_{\text{process parameters}}\ \underbrace{T_{w2},w_w}_{\text{other variables}}) \quad (1)$$

Chilton-Colburn correlation.
The heat exchanger is build up with louvered fins according to figure 3. For an estimate of the mass convection coefficient h_m the Chilton-Colburn analogy in a boundary layer is used and expressed in the dimensionless Colburn factor j_c. [1]

$$\frac{Nu}{\mathrm{Re}\,\mathrm{Pr}^{1/3}} = j_c = \frac{1}{2} f \qquad for \quad 0.5 < \mathrm{Pr} < 2 \quad (2)$$

The friction factor *f* and the Colburn factor j_c are supposed to be only a function of the Reynolds number.
The Reynolds number based on the hydraulic diameter of the canals gives *Re* = 570 and suggest a laminar flow. Though to the louvers on the fins flow through the fins leads to a turbulent flow, a so-called "low Reynolds-number turbulent flow" [2].
In the paper "A generalized friction correlation for louver fin geometry" by Chang e.o. [3] experimental correlations are given for a different arrangement of fins and louvers. For this study a general approximation from [3] is used to estimate j_c.

$$j_c = C_{jc}\,\mathrm{Re}^{-n} = 0.425\,\mathrm{Re}_{Lp}^{-0.496} \qquad (3)$$



$l_v = 8$mm
$\delta_v = 0.05$ mm
k_v=401 W/mK

Figure 3 Fin geometry of the heat-exchanger

From (3) an expression fort the heat-transfer coefficient h_c can be derived:

$$h_c = \frac{C_{j_c}\left(\frac{\mu_l c_p}{k_l}\right)^{1/3}}{\left(L_p/k_l\right)}\left(\frac{L_p}{\mu A_k}\right)^{1-n}\left(\dot{m}\right)^{1-n} \qquad (4)$$

For the given configuration with A_k=0.0048m^2, L_p= 1mm, and the general properties of air this gives:

$$h_c = 344\cdot\dot{m}^{0.505} \qquad (5)$$

According to Chilton-Colburn j_m=j_c and from (6) h_m can be expressed in h_c (7)

$$\frac{Sh}{ReSc^{1/3}} = \frac{Nu}{RePr^{1/3}} \Rightarrow h_m = h_c\frac{D_{AB}}{k_l}\underbrace{\left(\frac{k_l}{c_{pl}\rho_l D_{AB}}\right)^{1/3}}_{Lewis\ number} \qquad (6,7)$$

With ((5) follows an estimate for h_m:

$$h_m = 9.46 \cdot 10^{-4} \;\Rightarrow\; h_m = 0.325 \cdot \dot{m}^{0.505} \qquad (8)$$

Assumptions and simplifications.
With the found heat and mass a heat coefficients a heat and mass balance can be derived. The material properties of water and air (Table 2) are assumed to be constant over the length of the heat exchanger. The thermal resistance R_{cond} (figure 2) is assumed to be negligible to the convective heat resistances.
The effective conducting surface is based is based on the number of fins, the length and an estimated fin efficiency according to [4].

$$\eta_v = \frac{\left(tanh\left(ml_v\right)\right)}{ml_v} \quad \text{with} \quad m = \sqrt{\frac{2h_c}{k_v \delta_v}} \qquad (9)$$

With a length of the exchanger of 0.48 m, 108 (two sided) fins, height l_v=0.008 m A_v=1.66 m^2 and with a fin efficiency according to (9) the effective surface can be expressed as a function of h_c.

$$A_{lc} = \eta_{lv} A_v \approx -1.5 \cdot 10^{-3} h_c A_v = -2.49 h_c \qquad (10)$$

The fins are coated only on one side so $A_{lm} = \frac{1}{2} A_{lc}$.

The temperature drop due to the thermal resistance of the wetted coating is neglected. Assumed is that the humidity at the boundary of secondary side is saturated. The capillary resistance of the coating is also neglected.

Model.
The resulting model is a set of 3 coupled (11,12,13) witch can be numerically solved with boundary conditions (14,15,16).

$$\frac{dT_1}{dx} = -\underbrace{\frac{1}{\rho_l \dot{V}_1 (c_{pl} + w_1 c_{pw,g})}}_{Constant} \left(\frac{(T_1 - T_2)}{\frac{1}{A_{1c}h_{1c}} + \frac{1}{A_{2c}h_{2c}}} + \frac{\frac{1}{A_{2c}h_{2c}}}{\frac{1}{A_{1c}h_{1c}} + \frac{1}{A_{2c}h_{2c}}} \cdot \frac{(w_{w2} - w_2)}{\frac{1}{A_{2m}h_{2m}h_{fg}}} \right) \qquad (11)$$

$$\frac{dT_2}{dx} = -\frac{1}{\rho_l \dot{V}_2 (c_{pl} + w_2 c_{pw,g})} \left(\frac{(T_1 - T_2)}{\frac{1}{A_{1c}h_{1c}} + \frac{1}{A_{2c}h_{2c}}} - \frac{\frac{1}{A_{1c}h_{1c}}}{\frac{1}{A_{1c}h_{1c}} + \frac{1}{A_{2c}h_{2c}}} \cdot \frac{(w_{w2} - w_2)}{\frac{1}{A_{2m}h_{2m}h_{fg}}} \right) \qquad (12)$$

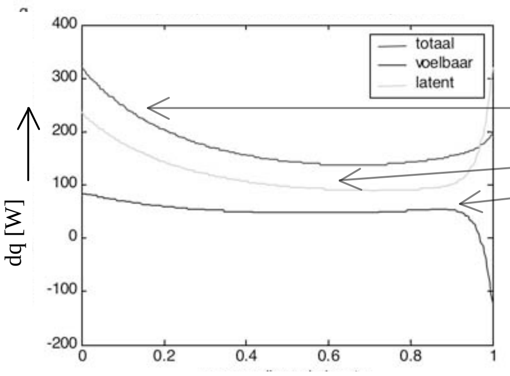
$$\frac{dw_2}{dx} = -\frac{1}{\rho_l \dot{V}_2 (h_{fg} + c_{pw,g} T_2)} \frac{(w_{w2} - w_2)}{\frac{1}{A_{2m}h_{2m}h_{fg}}} \qquad (13)$$

$$T_1(0) = T_{1,in}\ ;\ T_2(1) = T_{1,uit} \qquad \text{and} \quad w_2(1) = w_{1,uit} = w_{1,in} = w_1(0) \qquad (14,15,16)$$

The equations are solved with Matlab® the used routines are collected in [9].

Results

Figure 4 shows the heat transfer along the length of the exchanger. The secondary air enters from the right. |At first the latent heat transfer (2) is also used to cool the secondary air, the sensible heat transfer (2) is negative. Due to the vapour transfer along the total length of the heat exchanger a good performance is achieved. The latent heat transfer is about 2 times larger than the sensible heat transfer.



Heat transfer q in [W] along the length of the heat-exchanger

(1) Total heat transfer
(2) Latent heat transfer
(3) Sensible heat transfer

$\dot{V}_1 = 600 m^3/h \quad T_1(0) = 30^\circ C \quad \varphi = 30\%$

$\dot{V}_2 = 200 m^3/h$

→ norm. length

Figure 4 Latent en Sensible heat transfer along the length x/L of the heat exchanger.

Table 1. shows the calculated results of the model, the sensible cooling strongly depends on the humidity of the inlet air. Between rel. humidity 30 and 70% the temperature difference ΔT drops from 10-15K to 4-5 K. In Table 2 the simulated data is compared to measured data of the cooler [5]

Table 1 Predicted temperature difference between inlet en outlet air.

	Rel. humidity of inlet air.					
	30%		*50%*		*70%*	
T_{in} [°C]	T_{out} [°C]	ΔT [K]	T_{out} [°C]	ΔT [K]	T_{out} [°C]	ΔT [K]
22	10.1	10.0	13.8	8.2	17.2	4.8
23	10.6	10.5	14.5	8.5	18.1	4.9
24	11.1	11.2	15.2	8.8	18.9	5.1
25	11.7	12.0	15.9	9.1	19.8	5.2
26	12.2	12.5	16.6	9.4	20.7	5.3
27	12.7	13.2	17.3	9.7	21.6	5.4
28	13.3	14.0	18.1	9.9	22.4	5.6
29	13.8	14.5	18.9	10.1	23.3	5.7
30	14.4	15.0	19.6	10.4	24.2	5.8
31	14.9	15.7	20.4	10.6	25.1	5.9
32	15.5	16.5	21.2	10.8	26.1	5.9

Table 2. Comparison of model results and measured data [5]

T_{in} [℃]	Rel. humidity of inlet air.								
	30%			*50%*			*70%*		
	Model	Meas.	Δ	Model	Meas.	Δ	Model	Meas.	Δ
22	10.1	12.0	-19%	13.8	15.1	-16%	17.2	18.2	-21%
23	10.6	12.5	-18%	14.5	15.6	-13%	18.1	18.8	-14%
24	11.1	12.8	-15%	15.2	16.2	-11%	18.9	19.3	-8%
25	11.7	13.0	-11%	15.9	16.7	-9%	19.8	20.0	-4%
26	12.2	13.5	-10%	16.6	17.2	-6%	20.7	20.8	-2%
27	12.7	13.8	-8%	17.3	18.0	-7%	21.6	21.8	-4%
28	13.3	14.0	-5%	18.1	18.2	-1%	22.4	22.5	-2%
29	13.8	14.5	-5%	18.9	19.0	-1%	23.3	23.6	-5%
30	14.4	15.0	-4%	19.6	19.8	-2%	24.2	24.5	-5%
31	14.9	15.3	-3%	20.4	20.5	-1%	25.1	25.5	-7%
32	15.5	15.5	0%	21.2	21.2	0%	26.1	26.8	-12%

DISCUSSION

The model gives a good insight in the principal operation of the cooler, although based on experimental approximation for fin geometry; the model gives a reasonable approximation of the performance. The found model gives (Table 2) an over estimation of the temperature drop of max. 20%. The neglect of: heat conductance through the wall, the moisture transport through the coating all causes an overestimation of the cooling effect. For the lower inlet temperatures (<24 ° C) the secondary air over the total length is staying close to the saturation line, when the air at the surface is not fully saturated, as was assumed, this will have a significant effect on the cooling performance. The neglected R_{cond} of the wall will also lead to an increase of error at lower temperature.

To improve the model for the lower temperature range the thermal conductance of the wall and the vapour transport through the coating should be incorporated in the model.

Another thing was not accounted for in the model, is or the long term the saline deposition could lead to an extra thermal barrier. The manufacturer claimed on basis of endurance test that the deposition does not affect the water transport in the coating.

Table 3 Used material properties for air, water and copper

Air	Water	Copper
$\rho_l = 1.185 \quad kg \cdot m^{-3}$ $c_{pl} = 1006.9 \quad J \cdot kg^{-1} \cdot K^{-1}$ $\mu_l = 18.21 \cdot 10^{-6} \quad N \cdot s \cdot m^{-2}$ $k_l = 25.9 \cdot 10^{-3} \quad W \cdot m^{-1} \cdot K^{-1}$ $D_{AB} = 0.26 \cdot 10^{-4} \quad m^2 \cdot s^{-1} \left(T{=}298\ K\right)$	$\rho_{w,f} = 998.0 \quad kg \cdot m^{-3}$ $h_{fg0} = 2502 \quad J \cdot kg^{-1} \left(273.15\ K\right)$ $c_{pw,f} = 4181 \quad J \cdot kg^{-1} \cdot K^{-1}$ $c_{pw,g} = 1868 \quad J \cdot kg^{-1} \cdot K^{-1}$	$k_v = 401 \quad W \cdot m^{-1} \cdot K^{-1}$ Source: [3]

ACKNOWLEDGEMENT

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Used Symbol

A_c	convective surface (m^2)	q_{conv}	Sensible heat transfer (W)
A_m	masstransfer surface (m^2)	q_{mass}	latent heat transfer (W)
ρ_l	density of air (kg/m^3)	q_{tot}	total heat transfer (W)
c_{pl}	spe(m^{2}K/W)	R_{conv}	conv. thermal resistance (m^{2}K/W)
$c_{pw,f}$	cific heat of air (J/kg.K)	R_{cond}	cond. thermal resistance (m^{2}K/W)
$c_{pw,g}$	specific heat water (J/kg.K)	R_{mass}	mass. conv. resistance (m^{2}K/W)
μ_l	specific heat of vapour (J/kg.K)	h_c	thermal convection coeff. (W/m^{2}K)
D_{AB}	dyn. viscosity of air (Ns/m^2)	h_m	mass convection coeff. (kg/m^{2}s)
p	diff. coeff. water in air (m^2/s)	k_l	heat conductance of air (W/mK)
p_{sat}	air-pressure (Pa)	k_w	heat conductance of wall (W/mK)
w_{sat}	sat. vapour pressure (Pa)	k_v	heat conductance of fin (W/mK)
w	sat. moisture content (kg/kg)	L_w	thickness of wall (m)
T	moisture content (kg/kg)	u_m	average airspeed (m/s)
$\dot{V}$	temperature (K)	D_h	hydraulic diameter (m)
$\dot{m}$	volumeflow (m^3/s)	l_v	length of fin (m)
h	massflow (kg/s)	δ_v	thickness of fin (m)
h_{fg}	specific enthalpy (J/kg)	η_v	fin efficiency (-)
	evap. enthalpy water (J/kg)		

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Evaporative cooling and heat pipe recovery systems to improve energy efficiency in air conditioning

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ABSTRACT

The objectives of this work are: to evaluate and optimize a method of operation of a combined system of energy recovery for air conditioning systems by means of one configuration in series of three devices, semi-indirect evaporative cooler of ceramic tubes, which recover sensible and latent energy, using the refrigeration effect by evaporation of water transmitted through its porous ceramic structure to a non saturated airflow, or by the sensible cooling of air by water flow. Heat-Pipes heat recovery battery and an air-air Heat Pump, which allow low temperature residual energy recovery. Experimental design methodology was used in test beds. This combined system will allow reducing investment cost in air conditioning equipment, to lower energy consumption of the buildings and reduce CO2 environmental impact in emissions using less HCFCs.

INTRODUCTION

In 2002 The European Union approved a directive on Energy Performance Building EPBD, one of its specific sections being the introduction of new technologies of air conditioning systems, more energy efficient such as the one proposed in the research project developed, that allows, by means of combination of a system of semi-indirect evaporative cooling to gain heat, a battery of Heat Pipes HP and a Heat Pump HPu to reduce the energy consumption in the thermal conditioning of buildings.

The application of the combined system studied, in addition to preparing the air with high energy efficiency and obtaining a suitable level of comfort in areas allows a configuration of tri recovery, recovering energy of the airflow of the return or expulsion, and therefore to reduce final energy consumption. One of the problems associated to systems of evaporative cooling is proliferation of legionella; in our semi-indirect evaporative system primary airflow does not take place, since the matter exchanged is made from pores of the ceramic material towards the primary airflow avoiding aerosols, as a result of the gradient of water steam concentrations between surface and air. In addition to being a system of highly energy efficient system, allows a reduction in the environmental impact by reducing use of coolants like, HCFCs, and producing less CO2. The study of the behavior of the combined system was conducted in winter and summer conditions, analyzing the different characteristics that might have air in different climate conditions.

Figure 1 shows the experimental installation, its main components numbered, corresponding to climate chamber (1), Air Treatment Unit (ATU) or, simulator of conditions of outer air (2), SEC (3), Heat Pipes HP (4) and Heat Pump HPu (5).

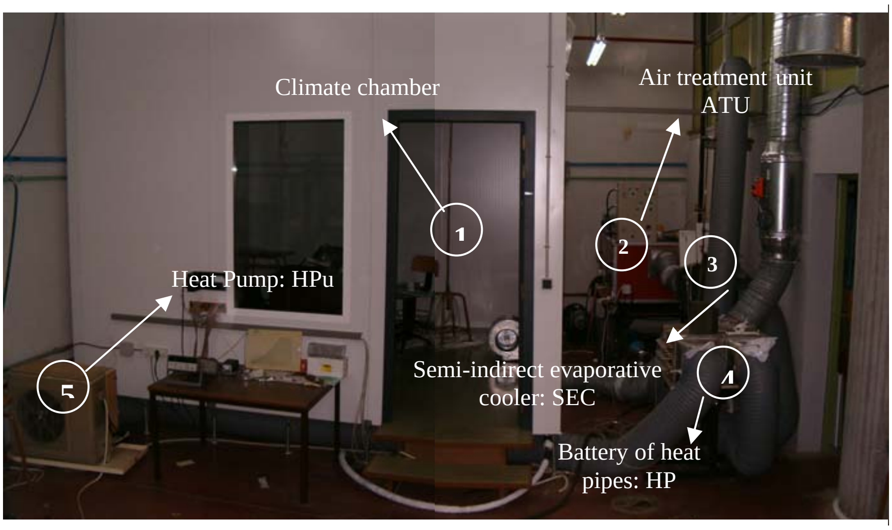


Figure 1.Experimental installation.

EXPERIMENTAL TEST BED, SENSORS AND DATA ACQUISITION

The experimental test bed of the combined system, shown in figure 1, consists of ceramic semi-indirect evaporative cooler equipment SEC, a battery of heat pipes HP and an air-air heat pump HPu linked by means of one recovery configuration to obtain air conditioning comfort on the premises and obtain energy recovery.

The Air Treatment Unit ATU prepares the air to simulate different climate conditions, that can be included in some of the climate conditions analyzed summer in tropical climate (hot air and high specific humidity) and continental summer (hot air and low relative humidity).

The outside air supply (primary current) flows through the SEC where it comes into contact with the ceramic surface, becomes humid, and as the ceramic surface is a porous material, part of the water is transferred to the primary air producing cooling by heat and mass transfer. In crossed current and by the interior of the SEC the return air (secondary current) from the premises is put in direct contact with water in counter flow, producing cooling by heat and mass transfer. Part of the water returns to the tank from where it returns to be pumped to the evaporative cooling system. However another part of the water can cross the porous structure coming into contact with the outdoor air.

The process in the SEC ceramic pipes not only produces cooling in the water, also in the secondary air flow, and in addition, humidifies the latest. This cooled airflow is used in condensation stage of the battery of heat pipes HP, where the primary air supply from the SEC is subjected to further to cooling, this time sensible cooling, with no variation of specific humidity. Figure 2 shows the components of the recovery system.

After the Semi indirect Evaporative Cooler in serial configuration, the primary air is cooled in crosscurrent in a heat pipe battery. The chamber return air, after the adiabatic cooling process, is used in the condensation stage of the heat pipes HP and on its exit, goes to the outer unit of the heat pump HPu, used in the final thermal conditioning of climate chamber used to develop the experimental tests.

Evaporative
cooler of
ceramic pipes
SEC

Heat Pipes recovery system HP.

Outer Unit of the heat pump HPu (left) and inner unit (right)

Figure 2.The components (SEC, HP and HPu) placed in the recovery process.

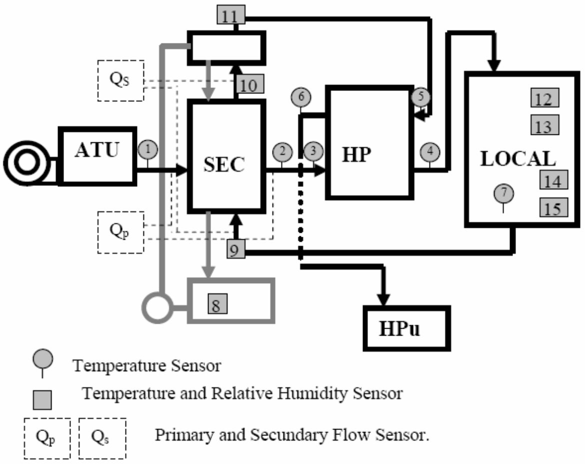
The last stage of energy recovery of the return air flow is to pass through the outer unit of the HPu that operates as, a condenser and is connected to the exhaust duct leaving from the HP battery. The inner unit of the air-air HPu, allows us, to obtain comfort conditions in the chamber.

Finally, the recovery system consists of taking advantage of the higrothermal characteristics of the return air using three processes in series, the evaporative cooler SEC that takes advantage of the cold water of on adiabatic cooling process with the return air, the HP system, that takes advantage of the cooling capacity of the air from the adiabatic cooling process itself and HPu system, since the power characteristics of the return air before being expelled, are usually more favorable for the outer unit that the outside air itself.

In the implementation of the recovery system in the air conditioning system, the installation control system must decide on the direction of the airflows, to optimize system power recovery, determining the most favorable operation conditions based on the specific temperature and humidity characteristics of the air flows involved in the recovery process.

For the measurement of the variables in all the points of the installation, we decided to use platinum resistance sensors, the humidity sensors, differential pressure sensor water flow, and air volume nozzle. The configuration of the sensors in the test bed is shown in figure 3.

A data acquisition system transforms the tension signals from pressure sensors, RTDs, humidity sensors, etc. into numerical values for subsequent later graphical representation and numerical analysis. The data acquisition, shown in figure 3, will receive the information from the sensors described in the previous section, and the software of the data logger “MOCIL” designed by us will be used in the experimental test bed recuperator.



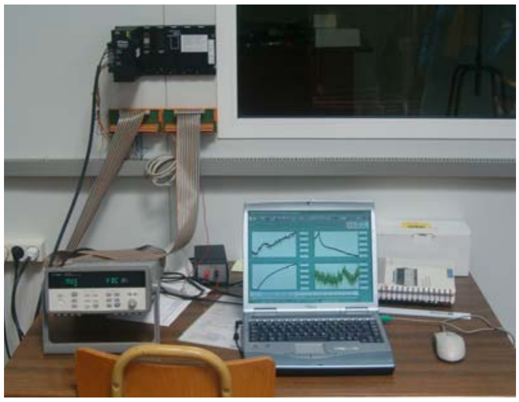


Figure 3.General scheme of the distribution of the sensors and the acquisition system used.

EXPERIMENTAL MEASUREMENTS AND ANALYSIS OF RESULTS

In the present study “design of experiments” methodology has been chosen to develop the experimental stage, using as control factors those variables that are anticipated, will affect the behavior of the recovery systems energy analyzed SEC, HP and HPu. The factors that have been analyzed in the complete design of factorial experiments are those that can be controlled in the experimental device, and which affect the functional characteristics of the equipment. These factors are:

Outer air temperature. Levels:

T1 5 ºC T2 10 ºC T3 15 ºC T4 20 ºC T5 C 25 ºC
T6 30 ºC T7 35 ºC T8 40 ºC T9 45 ºC

Outer Air flow. Levels:

V1 300 V2 400 V3 500

Outer air humidity. Levels:

The hygrometric conditions of the installation, established as its base are those of the laboratory, establishing three levels from the base conditions to the upper level. NH1 conditions base, NH2 base + 3 g w /kg d.a. and NH3 base + 6 g w /kg d.a.

Water flow:

Experimental tests have demonstrated the convenience of evaporating a litre of water each 300 m^3 of air. Provided we remain in that ratio the water volume does not have any appreciable impact. For this reason for all our tests we decided to maintain this constant value at 150 l/h.

Once the corresponding factors and their levels have been determined, we considerate making an experimental design table, ensuring that all the levels of all the factors have been assayed with all the levels of the remaining factors the same number of times. The number of tests agrees with the product of the number of levels of the factors, and in addition it is necessary to guarantee the repeatability between experiments.

The experimental results must be treated so that the required characteristics can be calculated to be able to explain the behavior of the recovery systems involved in the process of the case study.

Different characteristics have been analyzed. As an example we show only total heat flow for all the combined system and latent heat for the SEC. The analysis of the different characteristics is made in two stages. Having presented the first average values of the levels of each factor and their interactions, which establish the behavior of the characteristic with the factor, the contribution of the factors and interactions using ANOVA variance analysis will subsequently be verified.

Sensible heat for SEC+HP set.

Figure 4, shows two graphs and a table. The graph on the left shows the results corresponding to the average values of the characteristic for the different levels from the individual factors. The graph on the right corresponds to the interaction of both factors, temperature x volume finally table 1 shows the evaluated values and contributing percentage with the variance analysis.

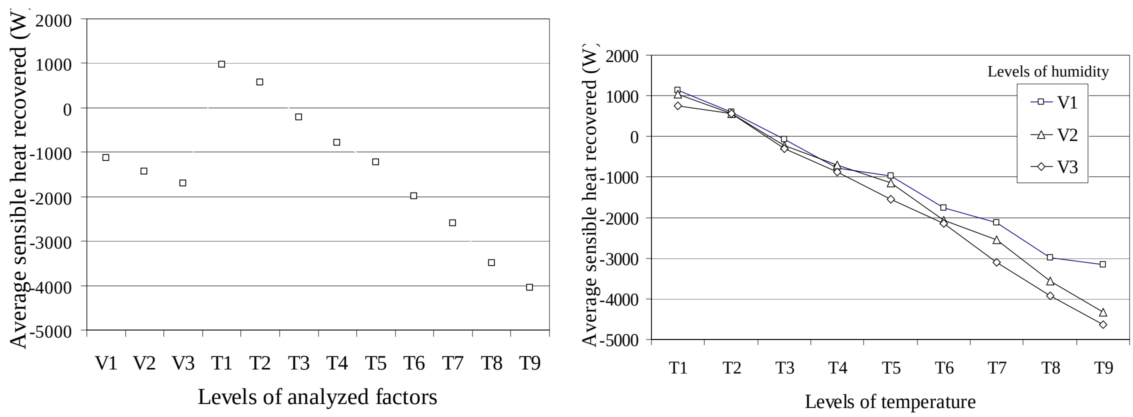


Figure 4. Graph for the levels of the analyzed factors (left) and the interaction of volume x temperature (right).

Table 1 ANOVA for the sensible heat characteristic.

Factors	gdl	SS	V	%
V	2	1453205.1	726602.5	1.91
T	8	73561119.0	9195139.9	96.69
VxT	16	1065784.6	66611.5	1.40
Total	26	76080109.0	2926158.0	100.00

The results from the table give an idea of the behavior of the device with respect to the characteristic:

Air volume. The recovered energy is slightly proportional to the air volume, but varies very little with this factor. It is the least contributing factor with a percentage in variance analysis to close to 2%. The slight change between the different levels in volume causes the global exchange coefficient, to remain constant.

Temperature. As happens in the other heat exchangers systems, when increasing the temperature difference between the outer air and the return air the amount of recovered sensible heat is greater, since the flow interchanged between the currents is directly proportional to the difference of its temperature, and is the factor that contributes most to the characteristics with a percentage close to 97%. In addition, the first graph allows us to see how in the proximities of level 3 temperature (T3), corresponding to 15 ºC, exchange between heating and cooling mode takes place in our device, we can also see in our characterization how positive values change to negative values, lineally becoming increasingly negative at higher temperature level. This fact can be very interesting when internal loads are considered.

Interaction double Temperature x Flow. Its study corroborates the analysis of the individual factors. It can be seen how as temperature and volume increase, sensible energy recovered increases. Interaction between the factors does not appear, the evolution of the energy recovered with respect to the three volumes having an almost parallel behavior.

Contributions of the recuperators SEC and HP separately.

The contributing percentage of each piece of SEC and HP equipment can be analyzed separately on the analyzed characteristic (sensible heat). However, in this case it is evident that SEC contributes more as it has greater thermal gradient between the currents that exchange energy. Figure 5 shows the total HP and SEC equipment sensible heat values compared to volume and temperature. Table 2 shows ANOVA for the recuperators separately.

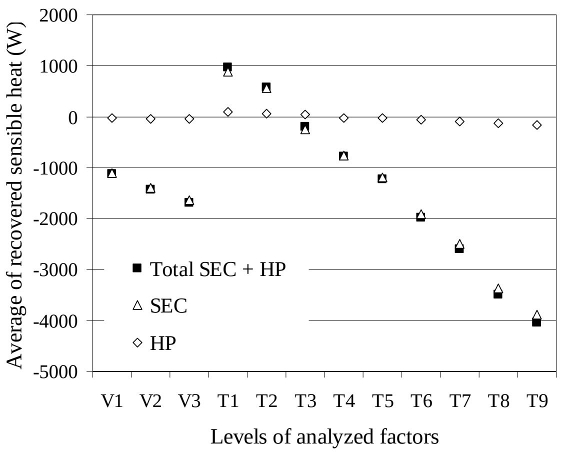


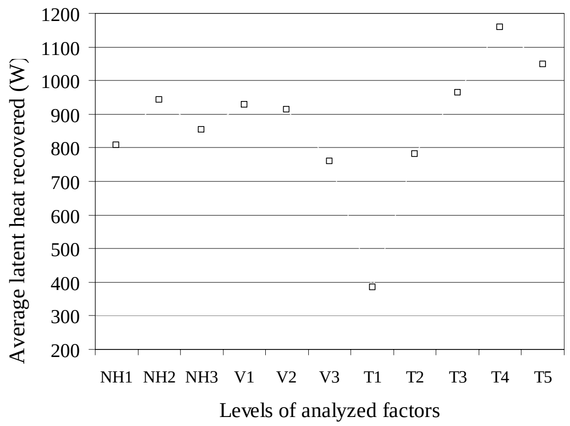
Figure 5.Graphs and sensible heat values for SEC and HP separately.

Table 2 ANOVA for the recuperators separately.

Sensible Heat		Total		SEC		H.P.	
Factors	**gdl**	**SS**	**%**	**SS**	**%**	**SS**	**%**
V	2	1453205.1	1.91	1296814.6	1.87	1727.23	0.92
T	8	73561119.0	96.69	67024437	96.54	181267.12	96.96
VxT	16	1065784.6	1.40	1102928.9	1.59	3956.14	2.12
Total	26	76080109.0	100.00	69424180	100.00	186950.49	100.00

SEC behavior in cooling mode.

In the second design of experiments, humidity factor level has been tested, and addition the temperature factor is analyzed only in summer conditions. As in the previous example figure 6 presents the graphs and the tables of the average values obtained for each level of the analyzed factors and the double interaction. Table 3 contains the ANOVA where error is included as the contributions associated to the higher order interactions.



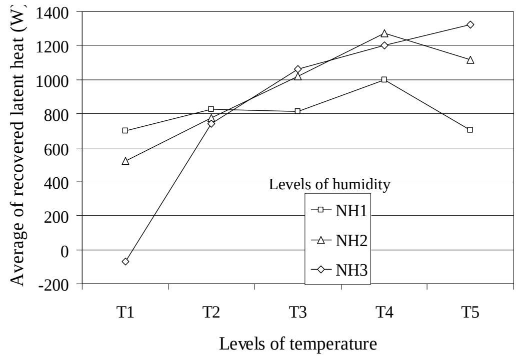


Figure 6.Graphics for the latent heat characteristic in the SEC.

Table 3.Results of the variance analysis (ANOVA).

Factors	**gdl**	**SS**	**V**	**%**
NH	2	138428.81	69214.405	2.25
V	2	263730.73	131865.36	4.29
T	4	3304199.60	826049.90	53.70
NHxV	4	74888.43	18722.11	1.22
NHxT	4	1671083.90	208885.49	27.16
VxT	8	268608.11	33576.014	4.37
Error	16	431728.76	26983.05	7.02
Total	44	6152668.30	139833.37	100.00

The behavior of the device with respect to the characteristic can be seen by observing the results presented:

Humidity level. An increase in humidity can even cause condensation in input air (dehumidification). Therefore it is felt that the system may be applied in tropical climates. When the relative humidity in the input air is low, water evaporation from the outer surface of the ceramics occurs and the treated current is humidified.

Air volume. It has no significant impact. Therefore under the analyzed conditions, it does not contribute to the result of the characteristic.

Temperature. Evaporation is determined by the gradient from partial pressure of the water steam between the surface of the pipes and the primary air. Therefore, when outer air temperature increases, if the constant specific humidity remains stable, relative humidity is reduced. Therefore the exchanged latent heat is greater when the evaporation capacity from the humid surface increases. It can be seen how for the higher humidity value and lower temperature (greater analyzed relative humidity), condensation takes place on the outer surface of the ceramic pipes.

CONCLUSIONS

The main conclusions to be drawn from this work are:

1. The recovery system combining SEC/HP /HPu allows energy exchange between the return current and the outer airflow, reducing consumption of thermal adjustment in all air, air conditioning systems.
2. The SEC recovery system eliminates the possibility of ventilated air contamination, as there is no mixture between the two currents that take part in the exchange process, the structure of the ceramic filter itself acting as a filter. This is why possible contamination from legionella that might appear in the primary air flow by water diffusion in the air, is avoided when no aerosols are formed.
3. The characterization of the SEC – HP equipment has been developed using experimental design techniques, making a complete factorial design, analyzing how they affect the factors considered input air volume, input air temperature and humidity level and the double interactions raised for each characteristic of the case study. Its contribution has been authenticated by means of variance analysis of results obtained. The factors analyzed in the design experiments, have allowed us to characterize the behavior of the SEC/ HP recovery equipment.
4. The efficiency of ceramic evaporative cooling like direct evaporation greatly depends on the ceramic pipes, since in the mass of water exchanged with the main airflow capillary diffusion of the water in the porous wall of the pipes also contributes. The controlling mechanism of the cooling process is due to convective mass diffusion of water through the ceramic pipe fillings.
5. The HP battery behaves like a “thermal shock absorber”, stabilizing temperature oscillations the airflows which cross the system of energy exchange system.

ACKNOWLEDGEMENT

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Renewable energy utilization in indirect evaporative air coolers under combined airflow conditions

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SUMMARY

The present report concerns mathematical simulation of coupled heat and mass transfer in the heat exchangers used in air conditioning units for the purpose of evaporative air cooling. Heat and mass transfer model based on the one-dimensional transfer model has been developed for thermal calculations of indirect evaporative cooler performance. On the basis of numerical investigations of "classical" heat exchangers designs the new design of air cooler is proposed. The original arrangement of matrix makes possible to realise two-stage indirect evaporative air cooling under combined parallel-counter air flow conditions and raise the degree of air cooling and thermodynamic perfection of this unit at the expense of efficient phase change heat utilisation. The received results show high efficiency of indirect evaporative air cooler under combined parallel-counter airflow conditions and offer scope for estimation of optimum operating conditions range variations and suitable climatic zones for the proposed unit.

INTRODUCTION

Keeping a safe and clean environment is currently becoming a more and more serious problem. One of the possibilities to decrease the pollution is the reduction of conventional energy consumption on behalf of more environmental friendly solution such as the increasing of renewable energy share in the total primary energy structure. Evaporative air cooling is recognized as a key demand-side "opportunity technology", which offers compelling energy, economic development and environmental benefits at the expense of utilization of thermodynamic unbalance of the atmospheric air.

In the case of direct evaporative cooler, the air stream to be cooled is in direct contact with a liquid water film and cooling is accomplished by the adiabatic heat exchange between the air stream and the liquid water film. The evaporation of water in the air stream leads to reduction of dry-bulb temperature and increasing absolute humidity of the supplied air. In general, the maximum possible reduction in air dry-bulb temperature depends on the difference between the dry-bulb and the wet-bulb temperature of the air stream. The direct type is simple but introduces moisture into the air-stream while cooling it adiabatically. In many applications, however, the increase of humidity in a supply air stream is not desirable. A promising way to increase the thermal performance of evaporative cooling systems is to decrease the dry-bulb temperature of the air at constant absolute humidity. Such kinds of processes can be realized with the help of indirect evaporative air coolers (IEC) [1–7].

Conventional integrated IEC has two distinct air passages, one named as the primary and the other as the secondary air passage. The primary air is usually outdoor air that is supplied to the room after it has been cooled by the air in the secondary air passages through the heat transfer surface . The secondary air can be either outdoor air or room exhaust air. The surface of the secondary air passage is moisturized, so that heat and mass transfer takes place between the wet surface and the secondary air. The dry primary air stream passes through the heat exchanger and uses the cooled wet secondary stream as its heat sink. Thus, IEC involves sensible cooling of the 'primary' air by an evaporatively cooled secondary air stream. As a result, the temperature of both streams decreases. The two air streams never mix or come in direct contact.

Traditionally, the heat exchanger type employed for IEC is a plate heat exchanger. The evaporative cooling of the wet stream takes place within the IEC itself along the wet passages. Then, the humidified air is dumped into the atmosphere.

There are many construction designs of combined IEC, improving of which goes in the following directions [6–9]:

- application of multistage cooling process within the confines of one unit;
- using capillary porous plates as a structural material for the heat transfer matrix, allowing moisture to fully cover the surface of the secondary channels at the minimum value of water consumption;
- implementation of rational combined management of airflows conditions, allowing to lower the temperature level of the received cold (air coolant);
- searching optimum operating conditions and suitable climatic zones for IEC on the basis of using the modern methods of polyoptimization.

Taking into account that there are merits and demerits for each type of heat exchanger, researches should be carried out not only for the developed air coolers, but also for heat exchangers, realizing "traditional" management of airflows conditions (Fig. 1). Such kind of research allows one to estimate an overall performance of new IEC and their conformity to the predicted purposes and to determine suitable climatic zones of their rational application.

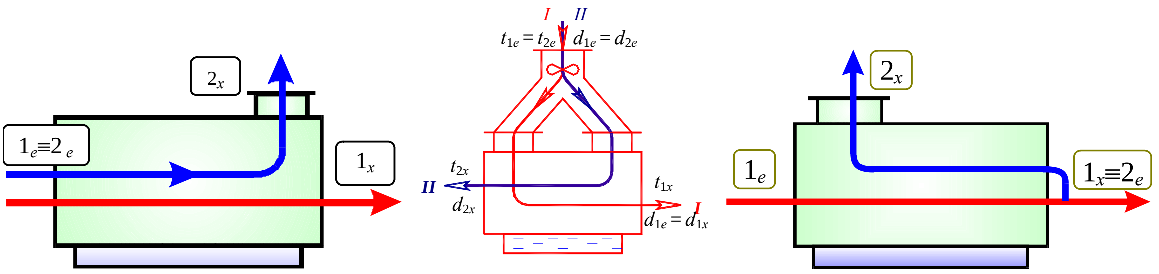

Figure 1. Schematic diagrams of typical IEC designs. a) Direct-flow, b) Counter-flow, c) Regenerative heat exchanger [6].

However, the substantiation of potential efficiency of heat and mass transfer process and rational performance of the considered units calls for a detailed theoretical study. With this purpose mathematical model of heat and mass transfer was developed for the combined IEC on the base of differential equations of the α-model [4, 5, 10]. It is necessary to note, that mathematical modelling of heat and mass transfer in modern IEC appears rather complicated [1–5, 11]. First of all, it is connected with an increase of the number of airflow passages and, consequently, the problem order.

The present paper is concerned with mathematical simulation of coupled heat and mass transfer in the units, used in air conditioning for the purpose of indirect evaporative air cooling.

METHODS

In many problems, including the said one, an engineer is not interested in receiving thermodynamic variables distributions normal to the heat exchanger plate surface, but bulk average values. Therefore, we gave preference to a one-dimensional transfer model (α–model) [4, 5, 10]. In this case, air stream in matrix passages is considered as a gaseous fluid flow with constant temperature and mass transfer potential in direction normal to plate surfaces, which are equalled to bulk average values.

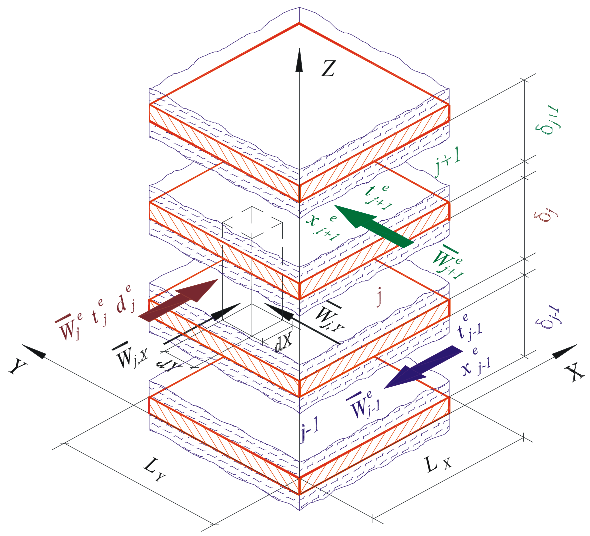

Figure 2. Schematic diagram of multichannel plate IEC.

Now, let us formulate a number of assumptions, concerning the peculiarities of examined heat and mass transfer processes and taken as the base of developed model in the most universal case (Fig. 2):

- driving force of mass transfer is a gradient of absolute humidity of air;
- kinetic properties of air flow and water film are constant, equalled to bulk average values;
- longitudinal molecular diffusion of water vapor in air and longitudinal heat transfer at the expense of thermal conductivity are negligible;
- no wall, air and fluid thermal and moisture diffusivity in the flow directions;
- unrestricted number of passages – n;
- consumed water rate corresponds to sufficient evaporation and keeping up the material of plates in hygroscopic saturated condition, that causes the following water film to air flow heat capacity rate ratio $\overline{W}_w / \overline{W} << 1$;
- the passage walls are impervious to mass transfer;

According to the assumptions of α–model combined heat and water vapour transfer in in the j^{th} passage of the plate IEC is described by partial differential equations of energy and mass balances developed for the air stream

$$\left(\overline{W}_X / \overline{W}\right)_j \frac{\partial t_j}{\partial \overline{X}} + \left(\overline{W}_Y / \overline{W}^e\right)_j \frac{\partial t_j}{\partial \overline{Y}} =$$

$$= \mathrm{NTU}_{j,j-1}\left(t_{wj,j-1} - t_j\right)\left[1 + \left(\frac{c_g}{c_p}\right)_j (\sigma/\mathrm{Le})_{j,j-1}\left(d_{wj,j-1} - d_j\right)\right] +$$

$$+ \mathrm{NTU}_{j,j+1}\left(t_{wj,j+1} - t_j\right)\left[1 + \left(\frac{c_g}{c_p}\right)_j (\sigma/\mathrm{Le})_{j,j+1}\left(d_{wj,j+1} - d_j\right)\right], \qquad (1)$$

$$\left(\overline{W}_X / \overline{W}\right)_j \frac{\partial d_j}{\partial \overline{X}} + \left(\overline{W}_Y / \overline{W}^e\right)_j \frac{\partial d_j}{\partial \overline{Y}} =$$

$$=\left(\mathrm{NTU}\cdot\sigma/\mathrm{Le}\right)_{j,j-1}\left(d_{wj,j-1}-d_{j}\right)+\left(\mathrm{NTU}\cdot\sigma/\mathrm{Le}\right)_{j,j+1}\left(d_{wj,j+1}-d_{j}\right)\ , \qquad (2)$$

and equations, describing conditions of energy flow rate continuity on the phase boundary

$$\left(\overline{W}^{e}_{j+1}/\overline{W}_{\mathrm{o}}\right)\mathrm{NTU}_{j+1,j}\left[\left(t_{wj+1,j}-t_{j+1}\right)+\left(1/c_{pj+1}\right)\left(H_{fg}\sigma/\mathrm{Le}\right)_{j+1,j}\left(d_{wj+1,j}-d_{j+1}\right)\right]+$$

$$+\left(\overline{W}^{e}_{j}/\overline{W}_{\mathrm{o}}\right)\mathrm{NTU}_{j,j+1}\left[\left(t_{wj,j+1}-t_{j}\right)+\left(1/c_{pj}\right)\left(H_{fg}\sigma/\mathrm{Le}\right)_{j,j+1}\left(d_{wj,j+1}-d_{j}\right)\right]=0\ , \qquad (3)$$

$$\left(\overline{W}^{e}_{j-1}/\overline{W}_{\mathrm{o}}\right)\mathrm{NTU}_{j-1,j}\left[\left(t_{wj-1,j}-t_{j-1}\right)+\left(1/c_{pj-1}\right)\left(H_{fg}\sigma/\mathrm{Le}\right)_{j-1,j}\left(d_{wj-1,j}-d_{j-1}\right)\right]+$$

$$+\left(\overline{W}^{e}_{j}/\overline{W}_{\mathrm{o}}\right)\mathrm{NTU}_{j,j-1}\left[\left(t_{wj,j-1}-t_{j}\right)+\left(1/c_{pj}\right)\left(H_{fg}\sigma/\mathrm{Le}\right)_{j,j-1}\left(d_{wj,j-1}-d_{j}\right)\right]=0\ , \qquad (4)$$

where j = 1..., n – the number of the channel (air stream), (j, j–1), (j, j+1) - the indexes describing interrelation between the j^{th} air stream and water film on the plate surfaces of the j^{th} passage, accordingly adjacent to (with) the $(j-1)^{\text{th}}$ and $(j+1)^{\text{th}}$ channels, $\overline{W}_{\mathrm{o}}$ - heat capacity rate of the base air stream, W/ K.
Derivation of the energy balance equations for water films (3), (4) is carried out on the base of the following additional assumptions: temperature difference across the plate and water film on the plate has been assumed to be negligible because of the small thickness of the plate (Bi<<1) and liquid film ($\overline{W}_{wj,j+1} << \overline{W}_{j}$ $\overline{W}_{wj,j-1} << \overline{W}_{j}$), and, hence, the local temperature of water film can be taken to be the same as the local temperature of the plate

$$t_{wj,j+1}=t_{wj+1,j}\ ,\ t_{wj,j-1}=t_{wj-1,j}\ . \qquad (5)$$

Thus, the set of simultaneous equations, describing heat and mass transfer in IEC, includes n differential equations of energy and mass balances for the air streams (1)¸ (2), n the algebraic equations of energy balance for water films (3)¸ (4) and n conditions for local temperature of water films (5). To derive the solution the set of equations (1)–(5) is supplemented by the non-linear empirical relation between absolute humidity of saturated air over liquid film and its temperature $d_{w,sat}=f\left(t_{w}\right)$ [12], the boundary conditions, establishing initial thermodynamic parameters values of exchanged air streams on the entries to appropriate channels of IEC. Taking into account the big variety of IEC construction designs and characters of plate mouisturizability , the boundary conditions should be taken up for each type of heat exchanger separately. The above formulated nonlinear boundary problem is nonlinear and can not be solve on analytic way. Therefore simultaneous integration of the equations set (1)–(5) with proper boundary conditions was made by numerical methods. Numerical simulation was executed with the help of the one-step fourth-order Runge-Kutta method, distinguished by sufficient accuracy and stability at the decision of similar type problems. The values of local plate temperature t_W satisfied by equation (3) and (4) were determined on each step of integration by means of Wegstein's iteration method used for solving general nonlinear equations of the form $t_W = f(t_W)$. The described algorithm was realised in digital computer program including 14 modules. The computer program evaluation tests were performed with the help of calculations of different variants, describing problems with familiar analytical solutions [4, 5, 9]. The accuracy of numerical method was checked by computing the identical variants on various space-time domain grids. Convergence test showed that the optimum number of grid nodes and, consequently, the minimum time of computation within the given tolerance of accuracy (0,5% thermal and mass balance error) was reached at the step size of 0,02 in the direction of axis $\bar{X}$.

RESULTS and DISCUSSION

On the basis of numerous calculations it was established that combined heat and mass transfer in the investigated IEC is characterised by complex and diverse temperatures and humidity profiles, which depend upon the combinations of thermodynamic, hydrodynamic and other factors.
The established distribution of air streams and water film thermodynamic parameters in the direction of primary airflow (Figs. 3, 4) necessitates the need to take into account the changes of plate temperature. Largely it concerns to regenerative IEC, in which changes of water film temperature reach (6–11)°C (Fig. 3.*b*) and to a lesser degree – to direct-flow IEC, where these changes are in an interval (1÷5) °C (Fig. 3.*a*).

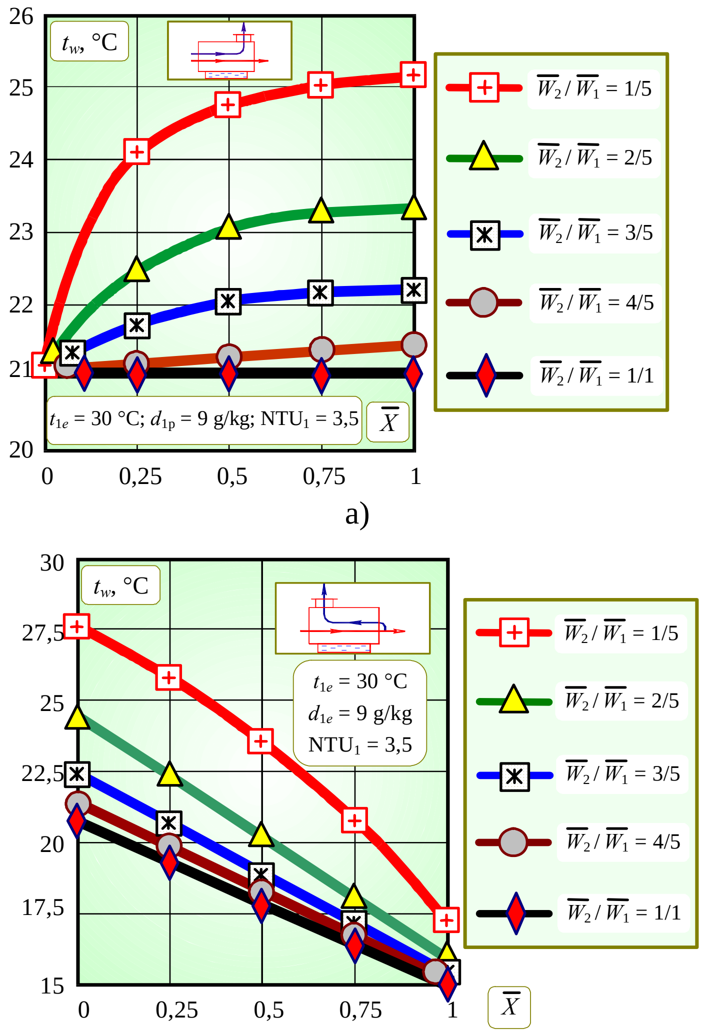

Figure 3. Effect of heat capacity rate ratio $\bar{W}_2/\bar{W}_1$ on the liquid film temperature distribution in the direction of primary airflow for two IEC construction designs.

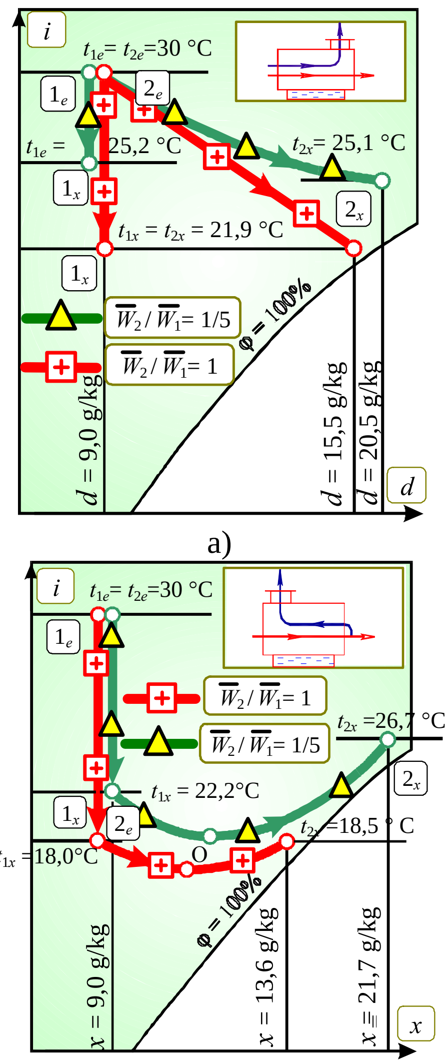

Figure 4. Paths for the ‘primary’ and secondary air streams on the *i-d* chart for two IEC construction designs.

In the case of the regenerative type of IEC the degrease of water film temperature in the direction of primary airflow allows the lowering of the temperature of the received cold (see Figs. 3.*b*, 5).

As shown in the Fig. 4, combined heat and mass transfer in the investigated IEC differs substantially in the case of direct-flow and regenerative type. The character of heat and mass transfer in counter-flow IEC needs special discussion.

The analysis of temperature distribution in secondary counter-flow passages brings out the existence of two particular heat and mass transfer areas (2_e-O and O-2_x in Fig 4.*b*). The air temperature decreases when the secondary air stream enters the channel (2_e-O), reaching a minimum value at a certain distance from the entrance, and then going up in the remainder of the channel (O-2_x). This can be explained as follows: the high values of temperature and absolute humidity differences at the entrance of secondary counter-flow channels (2_e-O) cause intensive water evaporation and secondary airflow cooling as a result. At the same time liquid film temperature t_w gradually increases because virtually constant sensible heat flow from adjacent primary air stream begins to exceed the evaporative cooling effect. Therefore, the temperatures of the secondary air stream t_2 and water film t_w at a certain distance from the flow inlet become equal (point O in the Fig. 4.*b*). As a result of water film heating on the terminal parts of the plate the temperature of liquid film t_w becomes higher then airflow temperature t_2. Thus combined heat and mass transfer process at the exit part of secondary counter-flow channels is characterised by the change of sensible heat exchange direction.

Figure 5 compares the thermal performances of three investigated IEC models for the ranges of heat capacity rate ratio $0 \le \overline{W}_2 / \overline{W}_1 \le 1$.

The received dew-point temperature effectiveness functions of dimensionless operating parameters (Fig. 5.*a*) are the most advantageous for the regenerative model of IEC. However, from the point of view of cooling capacity Q such construction design of IEC are worse then direct-flow and counter-flow heat exchangers (Fig. 5.*b*).

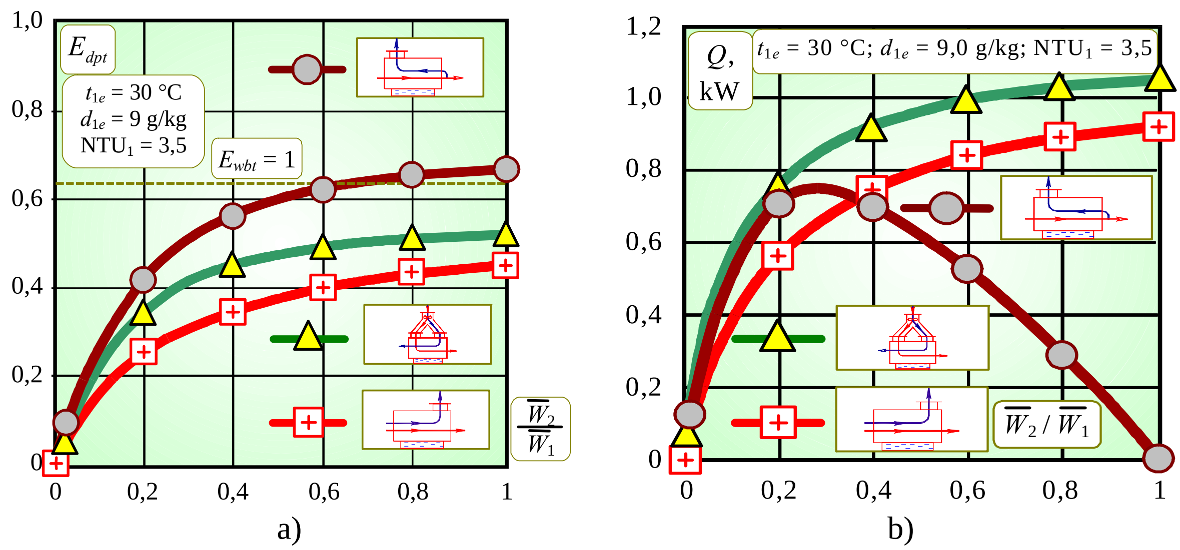

Figure 5. Influence of heat capacity rate ratio $\overline{W}_2 / \overline{W}_1$ dew-point temperature effectiveness E_{dpt} and cooling capacity Q for three IEC construction designs

According to the above specified characteristics of the heat and mass exchange processes the new construction of heat exchanger (Fig. 6) was proposed [7]. This construction – owing to the rational implementation of the combined management of airflow conditions (direct-flow

and counter-flow) – results in the following advantages of two-stage cooling process: effective heat and mass exchange at the entrance areas of plates at the parallel airflow conditions bundle and regenerative counter-flow on the terminal parts of the matrix.
The heat exchanger contains shell 1, pallet 2, suction branch for total airflow 3, and downstream fairings for primary 4 and secondary 5 airflows. Heat transfer bundle of a regular structure located in package 1 is composed of capillary porous plates 6, one side of which is covered by moisture-resistant coating. Plates are arranged so that they form alternate "*dry*" 7 and "*wet*" 8, 9 channels for passing to total ***I*** and secondary ***II***, ***III*** airflows consequently. There are dividing partitions 10 for organising necessary directions of airflows.

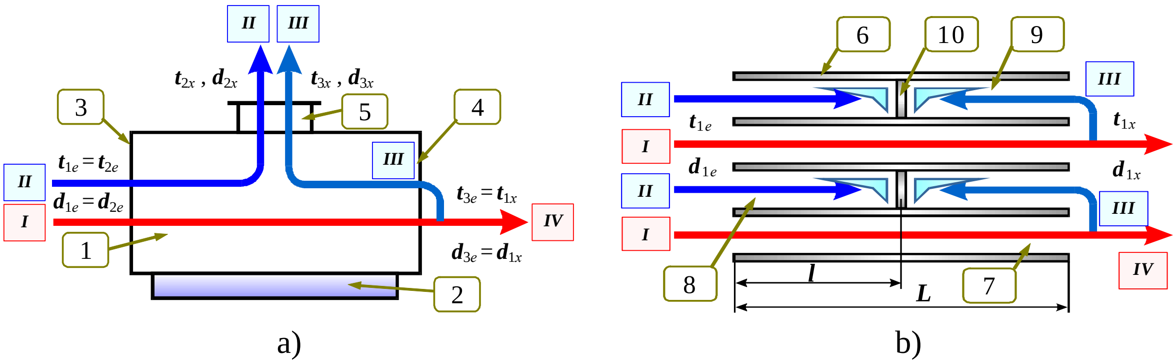

Figure 6. IEC under combined airflow conditions [7]. *a*) Front view, *b*) Top view.

The air cooler works as follows. Outside air enters the heat exchanger through suction branch 3 and then is divided on parallel secondary ***II*** and total ***I*** airflows, the first of which, passing through the "*wet*" channels 8, causes liquid film to be cooled by evaporation and then is discharged through downstream fairing 5 into the atmosphere. The total airflow ***I*** is cooled in "*dry*" air passages 7, formed by moisture-resistant surfaces of plates 6, due to its heat exchange with secondary airflows ***II*** and ***III*** passing through "*wet*" passages, and on the exit from downstream fairing 4 is divided on the primary ***IV*** and secondary regenerative ***III*** airflows. The primary airflow ***IV*** is delivered to air conditioned space, and secondary countercurrent airflow ***III*** (after the cooling of water film via evaporation in channels 9) is discharged through downstream fairing 5 in atmosphere.
The increasing of the air cooler capacity in comparison with the regenerative indirect evaporative air cooler [6] becomes possible due to the reduction of the secondary regenerative airflow consumption without decreasing total airflow temperature difference.
Optimising these distributions with the help of mathematical response surface models offered scope to determine rational performance characteristics of this IEC and the variation range of optimal operating conditions (1,5 <NTU_1< 2,5; 0,8<$\overline{W}_2/\overline{W}_1$ < 1; 0,14 <$\overline{W}_3/\overline{W}_1$ < 0,27; 0,4<$\bar{l}$< 0,55).

NOMENCLATURE

c_p = specific heat of the air, J/(kg dry air·K); c_g = specific heat of the water vapor, J/(kg K); d = absolute humidity of air (air humidity ratio), kg/kg dry air (g/kg dry air); E_{dpt} = $\left(t_{1e}-t_{1x}\right)/\left(t_{1e}-t_{1dpt}\right)$ = dew-point temperature effectiveness, dimensionless; G = air mass flow rate, kg/s; F = heat and transfer area of matrix, exposed to an air stream, m^2; H = latent heat of phase change, J/kg; i = specific enthalpy of the air-water vapor mixture, J/(kg dry air); l = length of parallel airflow zone, m; $\bar{l}=l/L_X$ = relative length of parallel airflow zone, dimensionless; L_X = plate length in X direction, m; L_Y = plate length in Y direction, m; Q =

heat flow rate, W/m^2; t = temperature, °C; w = speed of air flow m/s; $\overline{W}$ = heat capacity rate = ($G\, c_p$), W/ K; X = axial coordinate along the plate in the direction of primary airflow, m; $\overline{X} = X/L_x$, $\overline{Y} = Y/L_y$ = relative coordinates, dimensionless; α = sensible heat transfer coefficient, $W/(m^2{\cdot}K)$; β = mass transfer coefficient, $kg/(m^2{\cdot}s)$; φ = relative humidity, percent; σ = mouisturizability of plate (σ = 0,0 : dry; σ = 1,0 : completely wet), dimensionless.

NONDIMENSIONAL COMPLEXES

NTU = number of heat transfer units = $(\alpha \cdot F)/(G \cdot c_p)$; Le = Lewis number = $\alpha/(\beta \cdot c_p)$.

SUBSCRIPTS

dpt = dew-point temperature; *e* = state of air stream at the entrance of passage; *fg* = vapor-liquid; *g* = water vapor; *p* = at constant pressure; *sat* = saturation state; *x* = state of air stream on the exit of passage; *w* = surface of plate; 1 = primary airflow; 2 = secondary parallel airflow; 3 = secondary regenerative counter-airflow.

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Selection of adsorptive materials for desiccant cooling systems

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SUMMARY

The continuous rise of electric power rates worldwide and environmental issues related to the ozone layer depletion have increased the interest in natural gas cooling technologies over the last years. One of the most promising techniques consists of evaporative cooling aided by solid desiccants. Although silica-gel has been primarily used in such systems, the requirement of encompassing a greater variety of climatic conditions has led to the development of new materials with selective adsorption properties, some of them being able to adsorb more than one chemical species at the same time (co-sorption). Accordingly, the present work analyses the influence of the isotherm shape of different adsorptive materials over the dehumidifying capacity of a desiccant wheel. A mathematical model for the desiccant wheels is developed, and the governing equations for heat and mass transfer are solved employing a totally implicit finite volume technique. The adsorption isotherms are represented by a general equation characterized by a separation factor R, the variation of which allows the behavior of three different materials (silica-gel, molecular sieve and 1M) to be simulated. The results show that the separation factor R has a great influence over the dehumidifying effectiveness, for a given regeneration temperature.

INTRODUCTION

The adsorption phenomena occur in almost all solid-vapor interfaces, but its effects are negligible unless the solid possess certain properties that allow a great affinity with the vapor. For instance, the internal area of 1 cm^3 of silica-gel is estimated to be 400m^2. The average size of the porous is also of great importance, as it is the factor which determines the size of the molecules to be absorbed. The structural integrity in face of the continuous thermal inversion is also a factor to be considered. One of the earliest works devoted to the dehumidification modeling was performed by Bullock and Trelkheld [1], and used a predictor-corrector method to solve a quite simplified model. Maclaine and Cross [2] presented a characteristics potential method, which allowed the coupled problem to be solved in an analogous heat transfer problem. Jurinak, Mitchell and Beckmann [3] proposed alternative design options combining evaporative coolers and desiccant wheels, expanding the range of applicability of such equipment. Zheng and Worek [4] used an explicit finite difference scheme to solve the governing equations, using experimental data for the silica-gel heat of adsorption and isotherm shape. Zhang, Dai and Wang. [5] presents a very accurate model for the cyclic adsorption, which however involves too many parameters. They also provide the experimental data for the heat of adsorption used by the present simulation. The present work uses a totally implicit finite volume technique to solve the non-dimensional governing equations, which contains less parameters when compared to previous efforts. It also proposes the use of a generalized equation for isotherm shape [6], which can be set to represent a particular material (silica-gel, molecular sieve 13 or 1M) by choosing an appropriate value for the separation parameter R. Desiccant wheels consist of a porous disc impregnated with hygroscopic material, which by definition attract and retain water vapor. Figure 1 shows a schematic of a desiccant wheel, operating between two air streams. At the "cold side", a fresh air stream (process air) is forced through the wheel, giving up its moisture to the hygroscopic material and leaving the wheel at a much lower humidity ratio. As the wheel rotates, the hygroscopic material eventually switches to the regeneration stream, where hot air from a hot source is forced through the wheel, drying the hygroscopic material and dumping the water vapor back to the atmosphere. As depicted in Figure 1, the porososity pattern is not random, but formed by small channels, evenly distributed throughout the disc. Each channel has a structural layer (usually aluminum) and an adsorptive layer, above which the air flows.

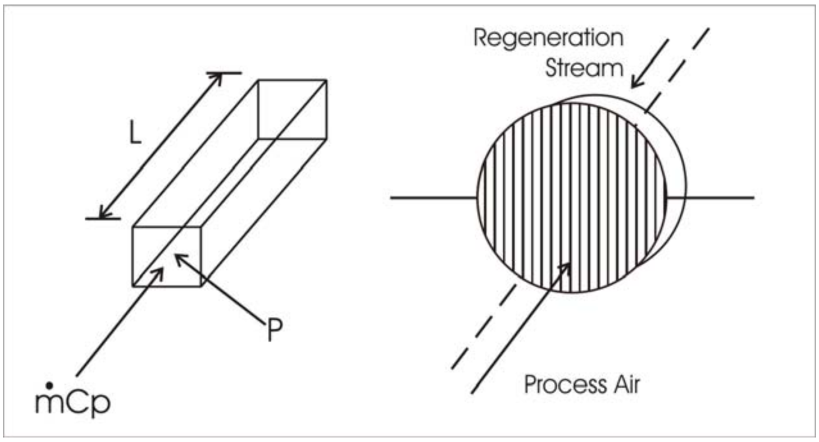

Figure 1: Schematic representation of a desiccant wheel

METHODS

An element of the channel of length Δx, in the direction of the flow, as shown in figure 2, is considered.

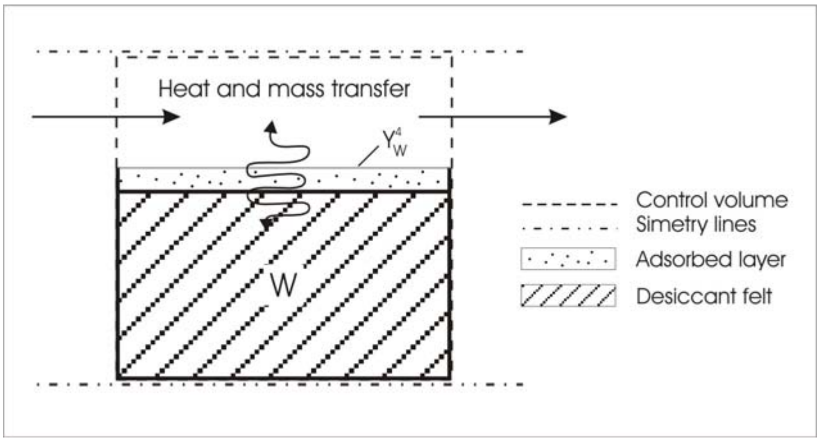

Figure 2: Elementary Control Volume

Some simplifying assumptions are necessary to the establishment of a mathematical model:

1) The flow is hydrodynamic developed.
2) The heat and mass transfer coefficients are constant along the channel.
3) Energy and mass storage within the air are negligible when compared to the solid.
4) Uniform temperature and concentration values in the direction perpendicular to the flow.
5) Symmetry lines represent perfectly adiabatic and impermeable surfaces.
6) Thermal and mass resistances are negligible in the direction perpendicular to the flow.
7) Thermal and mass resistances are infinite in the direction of the flow.

Assumption (1) is reasonable because of the low viscosity of air and the (small) ratio between the air channel height and length. Assumption (2) holds because the flow is usually laminar in the range of interest, whereas assumption (3) reflects the smaller thermal and mass capacitances of air when compared to those of the hygroscopic material. Assumption (4) reflects the typical dimensions of desiccant wheels, in which the channel length is usually a hundred times greater than the channel height and the desiccant layer. Assumption (5) reflects the homogeneity of the porous media, and both assumptions (6) and (7) are consequences of assumption (4).

Accordingly, applying a mass balance to the elementary control volume shown on Figure (2), enclosing both the flow channel and the hygroscopic material,

$$\underbrace{\rho_1\,\Delta x\;b\,a_1\,\frac{\partial Y^*}{\partial t_A}}_{\substack{transient\\ within\,the\,channel}}+\underbrace{f\;\rho_w\,\Delta x\;b\,a_w\,\frac{\partial W}{\partial x}}_{\substack{transient\\ within\,the\,solid}}+\underbrace{\frac{\dot{m}_1}{n}\left[\left(Y^*+\frac{\partial Y^*}{\partial x}\Delta x\right)-Y^*\right]}_{net\;\;flux}=0 \qquad (1)$$

in which

ρ_1 = *density of air*

Δx = *elementary length*

b = *channel and solid width*

a_1 = *channel height*

Y^* = *air absolute humidity*

W = *solid humidity content*

ρ_w = *density of solid*

f = *solid void fraction*

t_a = *time*

using the following definitions

$$\dot{m}_1=\rho_1\,a_1\,y_{AF}\,u_1\,n$$

$$m_w=\rho_w\,a_w\,x_{AF}\,y_{AF}\,n$$

one obtain

$$\dot{m}_1 \left[\frac{1}{u_1} \frac{\partial Y^*}{\partial t_A} + \frac{\partial Y^*}{\partial x_A} \right] + \frac{f\ m_w}{x_{AF}} \frac{\partial W}{\partial t_A} = 0$$

The mass transfer between the solid ant the air stream is given by

$$f\ \rho_w\, b\, \Delta x_A\, a_w \frac{\partial W}{\partial t_A} = 2h_y\, b\, \Delta x_A \left(Y^* - Y^*{}_w \right)$$

or

$$\frac{f\ m_w}{n\ x_{AF}\ y_{AF}} \frac{\partial W}{\partial t_A} = 2\, h_y \left(Y^* - Y_w^* \right) \tag{2}$$

in which

m_w total mass of desiccant in the wheel
n number of channels
x_{af} length of the wheel
h_y mass transfer coefficient

Applying a mass balance to the control volume shown on Figure (2), enclosing both the flow channel and the hygroscopic material,

$$\underbrace{\rho_1\, \Delta x_A\ y_{AF}\ a_1 \frac{\partial H_1}{\partial t_A}}_{\substack{transient \\ within\ the\ channel}} + \underbrace{\rho_w\, \Delta x_A\ y_{AF}\ a_w \frac{\partial H_w}{\partial t_A}}_{\substack{transient \\ within\ the\ solid}} + \underbrace{\frac{\dot{m}_1}{n} \left[\left(H_1 + \frac{\partial H_1}{\partial x_A} \Delta x_A \right) - H_1 \right]}_{net\ \ flux} = 0$$

or

$$\dot{m}_1 \left[\frac{1}{u_1} \frac{\partial H_1}{\partial t_A} + \frac{\partial H_1}{\partial x_A} \right] + \frac{m_w}{x_{AF}} \frac{\partial H_w}{\partial t_A} = 0 \tag{3}$$

The heat transfer between the solid ant the air stream is given by

$$\underbrace{\rho_1\, b\, \Delta x_A\, a_1 \frac{\partial H_1}{\partial t_A}}_{\substack{transient \\ within\ the\ channel}} + \underbrace{\frac{\dot{m}_1}{n} \left[\left(H_1 + \frac{\partial H_1}{\partial x_A} \Delta x_A \right) - H_1 \right]}_{\substack{net\ enthalpy \\ flux}} = \underbrace{2h_y\, b\, \Delta x_A \left(Y^* - Y_w^* \right) \frac{\partial H_1}{\partial Y}}_{\substack{heat\ released\ due\ to \\ adsorption}} + \underbrace{2\, h \left(T_w - T_1 \right)}_{\substack{heat\ transfer \\ between\ air\ and \\ solid}}$$

$$\frac{\dot{m}_1}{n b} \left[\frac{1}{u_1} \frac{\partial H_1}{\partial t_A} + \frac{\partial H_1}{\partial x_A} \right] = 2\, h_y \left(Y_w^* - Y^* \right) \frac{\partial H_1}{\partial Y} + 2\, h \left(T_w - T_1 \right) \tag{4}$$

in which

$$H_1 = aT_1 + Y^* \left(d + cT_1 \right) \tag{5}$$

$$a = 1.0046465\, KJ / Kg\, ^\circ C \quad d = 2467.4304\, KJ / Kg$$

$$c = 1.8837122\, KJ / Kg\, ^\circ C$$

Equations (1) through (4) are transformed into equivalent dimensionless forms after extensive algebra,

$$\frac{\partial Y^*}{\partial x^*} = Y_w^* - Y^* \quad (6)$$

$$\frac{\partial W}{\partial t^*_{h,c}} = \lambda_2 \left(Y^* - Y_w^* \right) \quad (7)$$

$$\frac{\partial T_1}{\partial x^*} = T_w - T_1 \quad (8)$$

$$\frac{\partial T_w}{\partial t^*_{h,c}} = \left(T_1 - T_w \right) + \lambda_1 \left(Y^* - Y_w^* \right) \quad (9)$$

where h and c refer to the hot and cold periods, respectively, and

$$\lambda_2 = \frac{C_{wr}}{f\left(\frac{\partial H_1}{\partial T_1}\right)} \qquad \lambda_1 = \frac{Q}{\left(\frac{\partial H_1}{\partial T_1}\right)}$$

The heat of adsorption Q is expressed in terms of the latent heat h_v and is experimentally obtained [5] as

$$Q = h_v \left(1.0 + 0.2843\, e^{-10.28W}\right) \quad (10)$$

We have now four equations (6) to (9) and five unknowns, Y^*, Y, W, T_1 and T_w . The missing equation is the adsorption isotherm, which relates the humidity content of the hygroscopic material, its temperature and the humidity ratio of the air layer in equilibrium with the solid,

$$W = W\left(T_w, Y_w^*\right) \quad (11)$$

with boundary conditions are given by

$$T_1(0,t^*) = T_{hin} \qquad ,0 < t^* < P_h^* \quad (12)$$

$$Y_1(0,t^*) = Y_{hin} \qquad ,0 < t^* < P_h^* \quad (13)$$

$$T_1(x_f^*, t^*) = T_{cin} \qquad ,P_h^* < t^* < P^* \quad (14)$$

$$Y_1(x_f^*, t^*) = Y_{cin} \qquad ,P_h^* < t^* < P^* \quad (15)$$

and the periodicity conditions are given by

$$T_{wc}(x^*, P^*) = T_{wh}(x^*, 0) \quad (16)$$

$$W_c(x^*, P^*) = W_h(x^*, 0) \quad (17)$$

The adsorption isotherm (Eq.11) is specific for each hygroscopic material. Simonson and Besant [6] suggest that although the humidity content W is a function of both temperature and relative humidity of the air layer, its dependence on the later is much stronger than on the former. Accordingly, a variety of adsorptive materials can be represented by

$$\frac{W}{W_{\max}} = \frac{1}{(1 - R + \frac{R}{RH})} \tag{18}$$

in which

W_{max}	maximum humidity content of the solid
R	separation factor
RH	relative humidity of the air layer

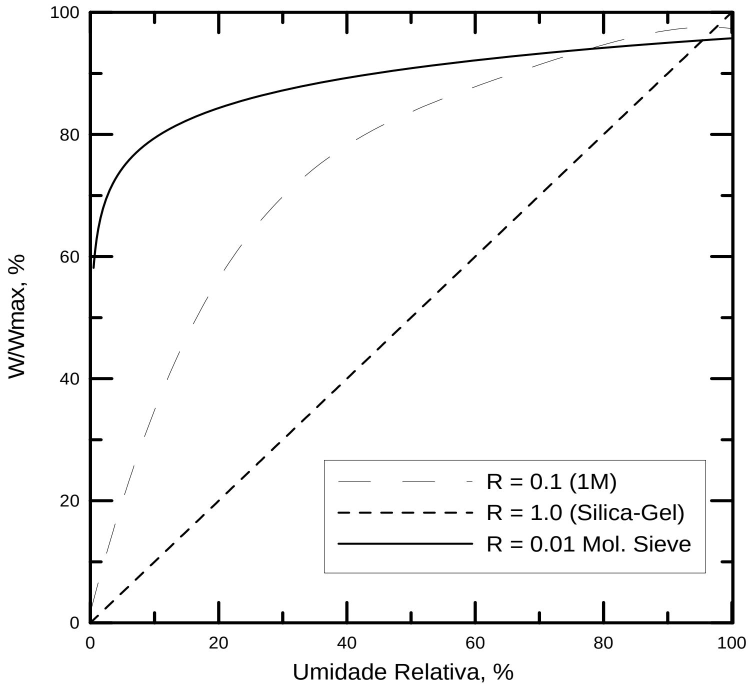

Figure 3: Generalized Adsorption Isotherm

The separation factor R can be adjusted so as to make the curve a fair representation of selected hygroscopic materials, as illustrated in Figure (3). The definition of effectiveness for a desiccant wheel is not as straightforward as it is in heat exchangers. In a desiccant wheel both heat and mass are being exchanged, and mass and heat transfers have opposite directions. Taking the regenerative stream (hot period) as an example, one would note that its temperature decreases as it flows through the channel, which tends to decrease its enthalpy. However, the hot stream is also humidified as it flows, which causes its enthalpy to increase. Moreover, both fresh air and the regeneration streams are supplied by ambient air, so they should have the same inlet humidity ratio (i.e., $Y_{hi} = Y_{ci}$). Accordingly, results are expressed by the dehumidification effectiveness given by

$$\varepsilon_d = \frac{\left(Y_{ci} - Y_{co}\right)}{\left(Y_{ci}\right)} \tag{19}$$

The domain was discretized using a finite-volume technique, using a fully-implicit formulation for the transient terms and an upwind formulation for the convective terms. Since both initial mass and temperature distributions are not known a priori, an iterative procedure, which compares the initial fields with the fields calculated at 360° is required. The enthalpy flux of the inlet streams must equal the average enthalpy flux of the outlet streams, as otherwise the wheel wouldn't be operating in a cyclic condition. The heat balance error HBE given below was found to be smaller than 0,1% for all simulations carried out.

$$HBE = \frac{\dot{m}_h\, h_{hi} + \dot{m}_c\, h_{ci} - (\dot{m}_h \frac{1}{p_h}\int_0^{p_h} h_{ho}\, dt^* + \dot{m}_c \frac{1}{p_c}\int_0^{p_c} h_{co}\, dt^*)}{\dot{m}_h\, h_{hi} + \dot{m}_c\, h_{ci}} \qquad (20)$$

RESULTS

Figures (4) and (5) show the influence of the adsorptive material selection over the effectiveness of dehumidification for two regeneration temperatures. For both cases the best performance is obtained for R=0,1 (1M). For a moderate value of the regeneration temperature (90°C), silica-gel exhibits a better performance than molecular-sieve, whereas for a higher regeneration temperature (160°C) the results are reverted. This result is consistent with the fact that molecular sieves are much stronger adsorbents than silica-gel, and thus require higher regenerative temperatures for desorption to take place. Interesting to note that a relatively weak adsorbent like silica-gel might not be indicated because of the relatively low capacity of removing humidity. Conversely, a strong adsorbent such as a molecular sieve might not be recommended as well, because an excessively high affinity to the water vapor might result in an uncompleted desorption, compromising the cyclic operation of the desiccant wheel. Accordingly, a moderate value for the separation factor seems to be the best choice. The existence of an optimum value of NTU can be explained by considering that an excessively long channel will result in continuous cooling of the regenerative stream with subsequent resorption of the water vapor. The order of magnitude of the optimum value for NTU is consistent with independently obtained results [7].

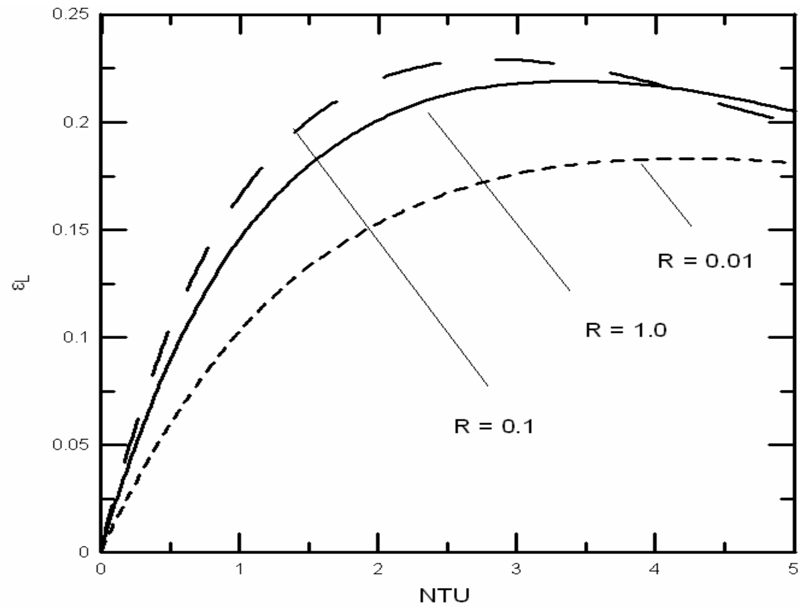

Figure 4: Dehumidifying effectiveness, T_{reg} = 90ºC

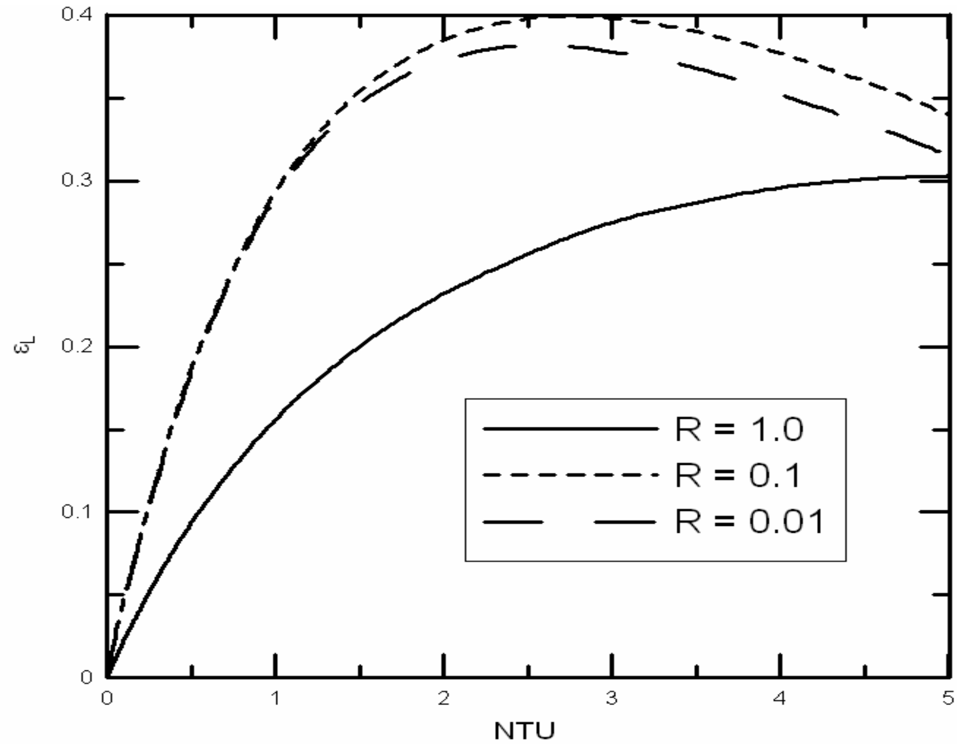

Figure 5: Dehumidifying effectiveness, T_{reg} =160ºC

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Optimisation and Cost Analysis of a Lithium Bromide Absorption Solar Cooling System

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SUMMARY

This paper presents the optimisation of the various components of a lithium bromide (LiBr) absorption solar cooling system such as the type, slope and area of solar collector and storage tank size. The collector types considered are the flat plate, compound parabolic and evacuated tube collectors. The optimisation is based on an energy benefit analysis, i.e., the amount of useful energy collected against the life cycle cost of the solar system. The above analysis considers the current prices for fuel and equipment costs, which have increased greatly during the last 5 years. A solar absorption system consisting of a lithium bromide-water unit, a solar collector and a storage tank is modelled with the TRNSYS computer program using a typical meteorological year (TMY) for a hot climate (Nicosia, Cyprus). For this analysis a typical house is modelled for a full year. The solar system can be used during summer to provide part of the heat required by the absorption unit and during winter to provide part of the heating load. The results indicate that due to the present high cost of fuel a large part of the building load can be covered with solar energy.

INTRODUCTION

Solar energy is in abundance in Cyprus. In summer, mean monthly temperatures for Nicosia at 14.00 hours in July, are 35.4°C with the temperature sometimes reaching 43°C. Therefore, there is a need to lower the indoor temperature considerably in order to be able to provide comfort. Solar cooling of buildings seems to be one of the most attractive solutions. This is an application in which the demand for cooling energy closely matches the availability of solar energy, not only to the seasonal but also to the daily variation.

Many researchers have developed solar assisted absorption refrigeration systems. Most of them have been produced as experimental units and computer codes were written to simulate the systems. Relevant studies are shown in [1-4].

The objective of this paper is to model a complete system, consisting of a number of solar collectors, storage tank, a boiler and a LiBr-water absorption refrigerator, which will cover a typical house load during the whole year. Conclusions are drawn by comparing the economics of the system between the years 2001 and 2007 when there was a dramatic change in the fuel price.

For this analysis the typical house indicated in Fig. 1 is considered. The house thermal load is minimised by considering an insulated roof, insulated walls, double glazed windows, internal shading and night ventilation (3 ach) in summer. The above factors were studied and found to be economically viable [5]. The double-walls are made of 0.10 m hollow brick and 0.02m plaster on each side and a layer of 0.05 m insulation in between. The roof is constructed from fair-faced 0.15 m heavy concrete, 0.05 m polystyrene insulation, 0.07 m screed and 0.004 m asphalt, covered with aluminum paint of 0.55 solar absorptivity. The above construction requires an annual cooling load at 25°C of 17,600 kWh with a peak load of 10.3 kW and an annual heating load at 21°C of 3,530 kWh with a peak load of 5.5 kW.

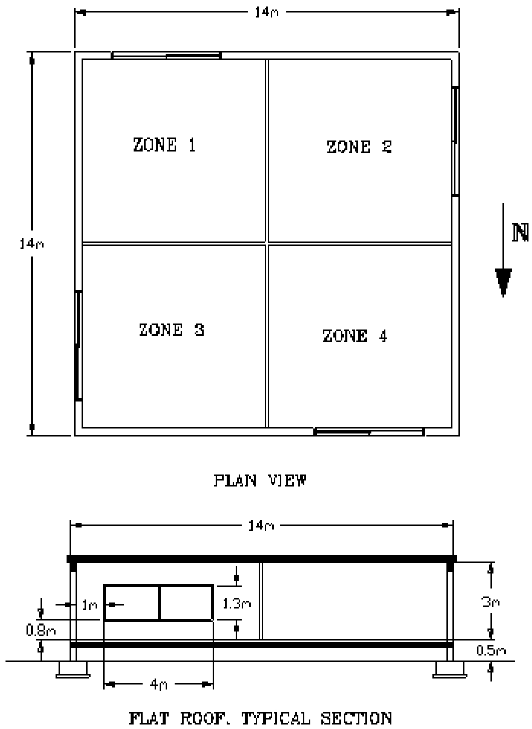

Fig. 1 Model house

The TRNSYS program is used to model the complete system (house load estimation with solar powered heating and absorption cooling), together with the weather values of a typical meteorological year (TMY) file for Nicosia, Cyprus.

The solar powered system consists of an array of solar collectors, boiler, storage tank, 11 kW absorption cooling unit, pumps and thermostats.

Using this approach a system optimisation is performed in order to select the right equipment, i.e., the collector type, the storage tank volume and the collector slope angle and area. The collector area is decided by performing an economic analysis of the system. Also the long-term integrated system performance and the dynamic system behaviour is evaluated and an economic comparison is made between the years 2001 and 2007.

CHARACTERISTICS OF AN 11 KW WATER-LIBR ABSORPTION CHILLER

The characteristics needed in TRNSYS deck file are the generator load, the mass flow of the heating water to the generator heat exchanger and its input and output temperatures. Referring to previous work [6] the calculated 11 kW unit characteristics are as shown in Table 1. Also other parameters needed for the TRNSYS input file were calculated based on this previous work.

Table 1 Water-LiBr absorption refrigeration system calculations based on a generator temperature of 75°C and a solution heat exchanger exit temperature of 55°C

Point	**H (kJ/kg)**	$\dot{m}$ **(Kg/s)**	**P (kPa)**	**T (°C)**	**%LiBr (X)**	**Remarks**
1	83	0.05691	0.93	34.9	**55**	
2	83	0.05691	4.82	34.9	55	
3	124.7	0.05691	4.82	**55**	55	Sub-cooled liquid
4	183.2	0.05217	4.82	**75**	**60**	
5	137.8	0.05217	4.82	51.5	60	
6	137.8	0.05217	0.93	44.5	60	
7	2612	0.00474	4.82	**70**	0	Superheated Steam
8	131.0	0.00474	4.82	31.5	0	Saturated liquid
9	131.0	0.00474	0.93	6	0	
10	2511.8	0.00463	0.93	6	0	Saturated vapour
11	23.45	0.00011	0.93	6	0	Saturated liquid

Description	**Symbol**	**kW**
Capacity (evaporator output power)	$\dot{Q}_e$	**11.0**
Absorber heat, rejected to the environment	$\dot{Q}_a$	14.1
Heat input to the generator	$\dot{Q}_g$	14.9
Condenser heat, rejected to the environment	$\dot{Q}_c$	11.8
Coefficient of performance	COP	0.74

THE COMPLETE SYSTEM CHARACTERISTICS

The complete system besides the absorption refrigerator consists of a number of solar collectors, a thermally insulated vertical storage tank, a conventional boiler and interconnecting piping. A schematic of the system showing also the simulation program information flow is shown in Fig. 2.

The system was modelled with the TRNSYS simulation program. The program consists of many subroutines that model subsystem components. The type number of every TRNSYS subroutine used to model each component is also shown in Fig. 2.

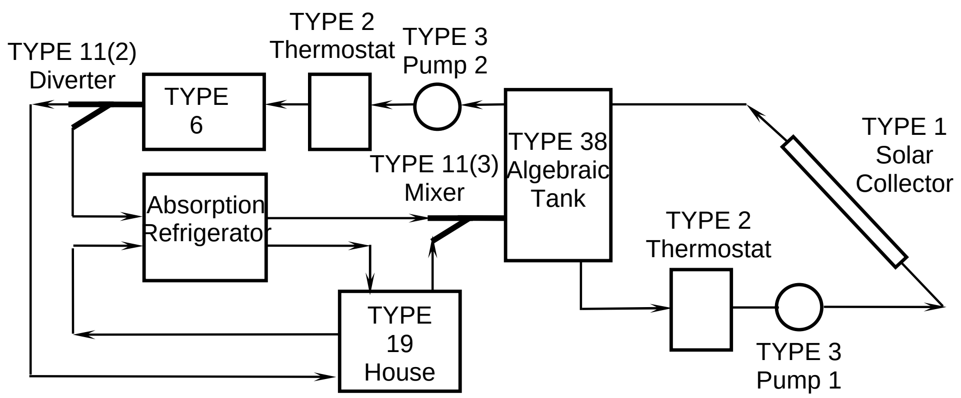

Fig. 2 Circuit diagram and TRNSYS types used for modelling the system.

The construction and type of the solar collectors is important and relevant to the operation and efficiency of the whole system. Three types of solar collectors, modelled with TRNSYS Type 1, are evaluated in this study as follows:

Flat Plate Collectors: These are predominantly used for domestic hot water production. The external casing of the collectors considered in this study, which are manufactured locally, is made of high corrosion resistant galvanised steel sheet, sprayed with aluzinc paint. The casing is covered with a 4 mm thick single glass and sealed with a rubber gasket. The absorber plate is made of copper. High radiation absorption is achieved by the use of black fine matt finish on the copper surface, which has a high absorption coefficient. The underside of the absorber plate and the side casing are well insulated to reduce conduction losses with 50 mm and 30 mm fibreglass insulation respectively. Each flat plate collector consist of 12 evenly spaced parallel copper pipes, 15 mm in diameter, embossed by semi-circular grooves formed in the flat plate absorber.

Conventional flat plate collectors are developed for use in sunny and warm climates. During cold, cloudy and windy days their performance is greatly reduced. Furthermore, condensation and moisture can cause early deterioration of internal materials resulting in reduced performance and system failure.

Compound Parabolic Concentrating Collectors (CPC): These collectors use curved reflecting surfaces to concentrate sunlight onto a small absorber area. CPC’s are used for higher water temperature applications than the flat plate collectors. Such a focusing collector performs very well in direct sunlight but, depending on the concentration ratio, does not perform well under cloudy or hazy skies because only a few rays are captured and reflected onto the absorber. A compound parabolic collector system can work either as a stationary system, or it can track the sun. In stationary systems, much of the sunlight that hits the concentrator's reflector often misses the absorber. Tracking devices allow the collector to follow the sun's movement across the sky. This ensures that the concentrator always faces toward the sun. Such systems are usually employed for higher temperature applications. Since residential applications only require medium temperatures, stationary systems are usually employed. Stationary systems are also less expensive and are easier to install and maintain. Concentrating collectors work best in climates with a high amount of direct solar radiation as in Cyprus.

Evacuated Tube Collectors: These collectors are highly efficient, made of an absorber pipe enclosed within a larger glass tube. The absorber pipe may also be attached to a black copper fin that fills the tube (absorber plate). The space between the glass and the absorber is evacuated. Only radiant heat is transmitted from the absorber pipe to the outside of the collector and in this way the efficiency of the collector is substantially increased.

The tubes are usually mounted into a manifold and the heated liquid is either used directly or circulates through a heat exchanger and gives off its heat to water. The coated surface of the absorber pipe has high absorption (> 92 %) of solar radiation and low emittance (< 6 %) for the infrared heat radiation. The absorber coating also has high resistance to long-term vapour condensation and high operating temperature.

The performance of the system employing these three types of collectors is investigated in order to select the most suitable for the present application. The final selection is made by considering the financial viability of the system.

Hot water is stored in a TRNSYS Type 38 tank. The vertical cylinder construction is made of copper and is thermally insulated with polyurethane. Also the tank is protected by a galvanised outer shell 0.6 mm thick.

The backup boiler (TRNSYS Type 6) is assumed to have a maximum heating rate of 18 kW and a set upper temperature of 93°C.

A number of thermostats (TRNSYS Type 2) are also used in order to control:

a. The flow to the solar panels, allowing the fluid to circulate only when the temperature of the fluid returning from the collectors to the storage tank is higher than that of the fluid delivered to the load; and

b. The operation of the boiler, allowing the boiler to operate only when the temperature of the fluid delivered to the load is below an optimum value. In this case the boiler will keep the water temperature delivered to the absorption cooler always above 85°C.

SYSTEM OPTIMISATION AND ENERGY FLOWS

A number of simulations were carried out in order to optimise the various factors affecting the performance of the system. The parameters considered are as follows:

The collector slope angle: The solar heat gain from the system for various collector slope angles was examined. The optimum angle in the Cyprus environment is:

i. 25°-30° for the flat plate collector
ii. 30° for the compound parabolic collector, and
iii. 30° for the evacuated tube solar collector

This is due to the solar altitude angle, which for the latitude of Cyprus (35°) can reach to 78° during noon in June. Also, because of the load characteristics, with the total cooling loads being about 6 times bigger than the heating loads, the optimum angle should be such that the collectors are absorbing greater heat during summer.

Storage tank size: This factor also plays a role in the optimisation of the system. Simulations show that for all three types of collectors, a smaller tank size results in slightly less energy consumption by the boiler and slightly less energy collected by the solar collectors.

The storage tank size is therefore decided only from the length of time intervals between firing the boiler. Between these intervals the tank should be able to supply the system with the needed water mass flow at the correct temperature in the summer. As it results, when the storage tank size is small the temperature in the cylinder cannot be kept above 85°C for long periods of time.

Assuming that the boiler is firing every 12 minutes, which may be a reasonable interval, the optimum storage tank size needed for the system would be:

i. 0.8 m^3 for the flat plate collector
ii. 0.6 m^3 for the compound parabolic collector and
iii. 0.6 m^3 for the evacuated tube solar collector

Finally the effect of the collector area is evaluated against the boiler heat required. As it is expected the greater the collector area the less the boiler heat needed as indicated in Fig. 3 and the more the collected heat as indicated in Fig. 4.

To determine the optimum collector area the cost of the solar system must be compared against the fuel saved due to its use. The cost of the various types of collectors is shown in Table 2.

Table 2 Cost of solar collectors

Collector type	2001 Cost (€/ m^2)	2007 Cost (€/ m^2)
Flat plate collector	186	220
Compound parabolic collector	305	370
Evacuated tube collector	425	460

The economic method used, is the life cycle analysis [7]. For solar systems the following equation can be used:

System annual cost = Extra mortgage payment + Maintenance cost – Extra fuel savings

– Extra electricity savings – Extra tax savings (1)

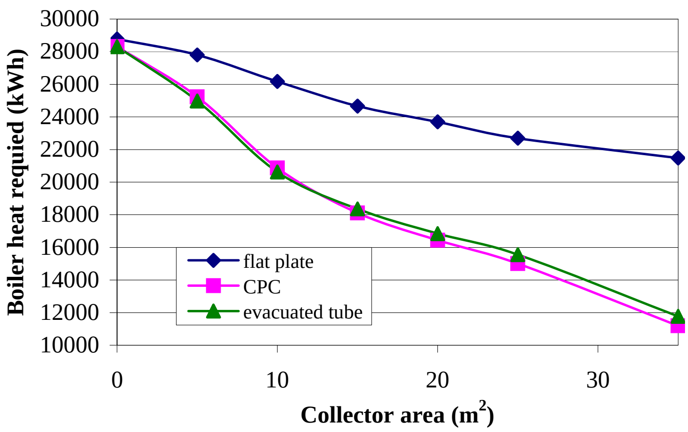

Fig. 3 Effect of the collector area on the boiler heat required by the system

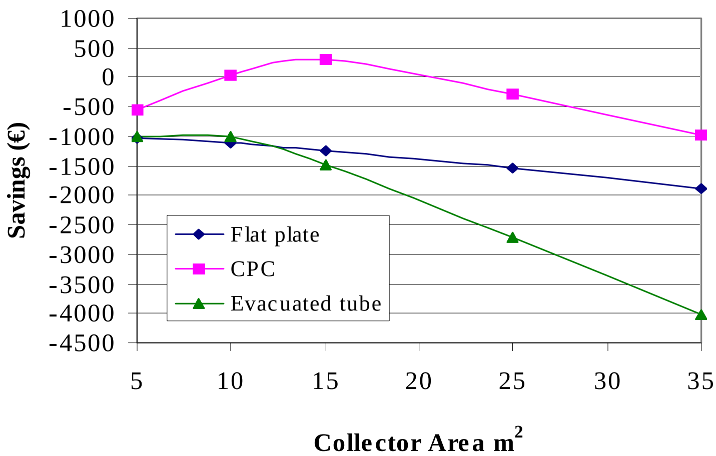

Fig. 4 Effect of the collector area on the collector heat gain

The scenario considered is that 30% of the initial cost of the system is paid at the beginning and the rest is paid in equal instalments in the next 10 years. The evaluation is based on the price of the diesel in 2001, which was € 0.29 per lt and the current price which is € 0.73 per lt.

The results of the economic analysis are presented in Fig. 5. As it is observed the only economically viable solution in 2001 was to use the CPC with a collector area of 15 m^2. The flat plate collector and the evacuated tube collector were not suitable for this application then due to the low price of fuel.

Now, in 2007, the situation has changed and CPC and evacuated tube collectors can be used economically as indicated in Fig. 6.

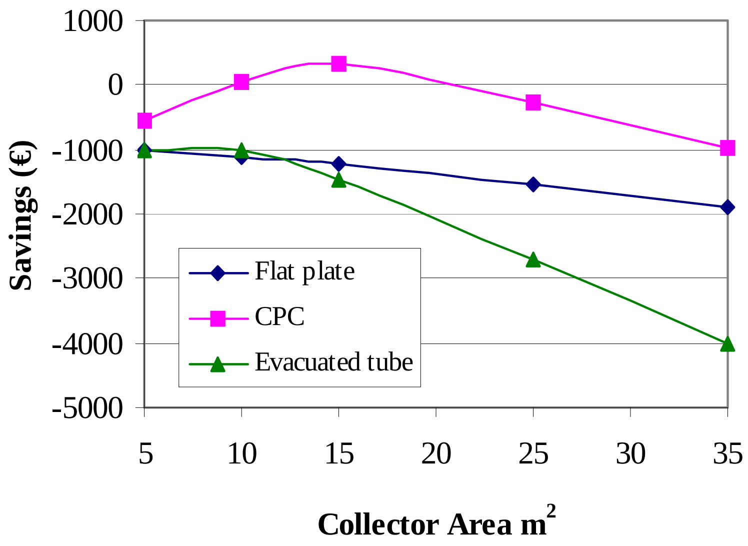

Fig. 5 Collector area against life cycle savings in € for a fuel price of €0.29 per lt and a 20 year period.

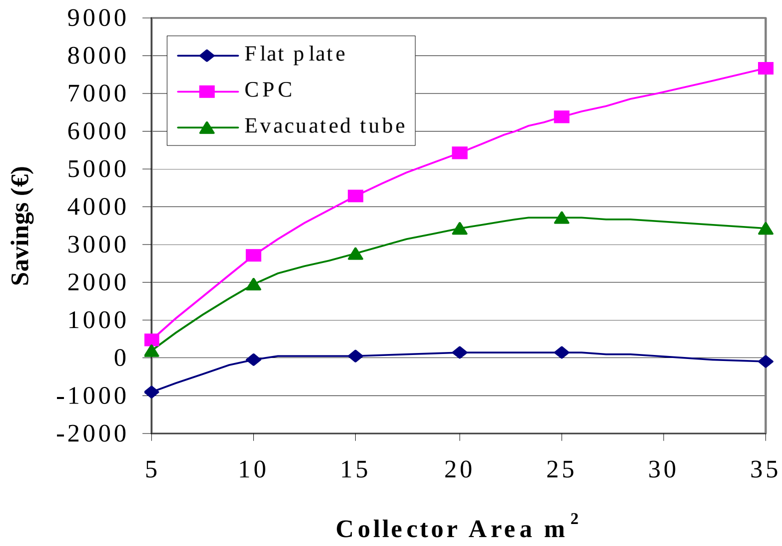

Fig. 6 Collector area against life cycle savings in € for a fuel price of €0.73 per lt and a 20-year period.

SYSTEM LONG-TERM PERFORMANCE AND ECONOMIC ANALYSIS

The specifications of the final system obtained from the optimisation study are shown in Table 3.

The energy flows of the system are shown in Figs. 7 and 8 for the years 2001 and 2007 respectively. The cooling load of the building reaches a maximum monthly value of 4200 kWh (in July), whereas the maximum monthly heating load occurs during January and is equal to 1250 kWh. The heat required from the conventional boiler is also shown in the above figures. The maximum monthly load supplied by the solar system is 1500 kWh and as can be seen from the difference of the curves for the cooling load and boiler heat, nearly all collector heat can be utilised for cooling or heating purposes.

Table 3 The final optimum system in 2001 and 2007

Item	Year 2001	Year 2007
Collector type	CPC	CPC
Collector area	15 m^2	35 m^2
Collector slope	30°	30°
Storage tank size	600 lt	600 lt

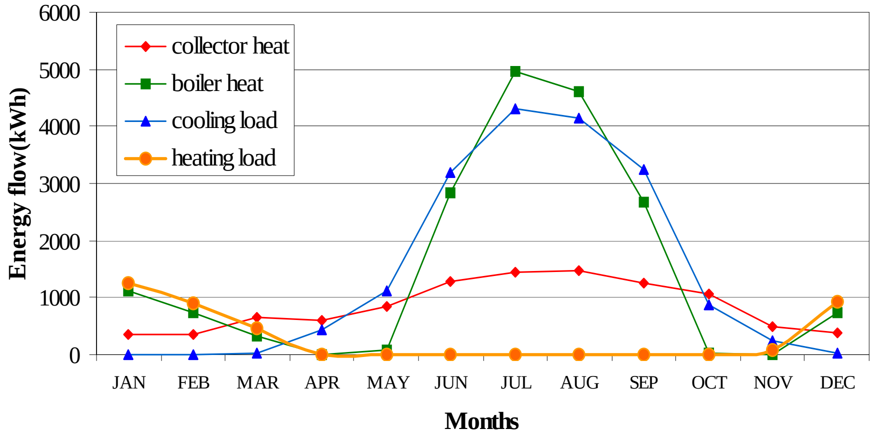

Fig. 7 System energy flows for 2001

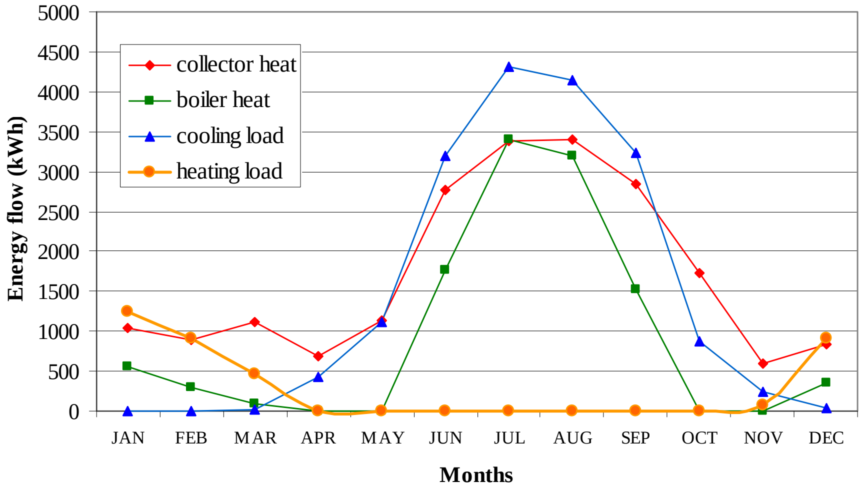

Fig. 8 System energy flows for 2007

The annual cooling load of 17,600 kWh is covered with a total supply of 15,220 kWh of boiler heat, supplemented by 8500 kWh of solar heat, offered by the solar system. The annual heating load of 3530 kWh is covered with a total supply of 2880 kWh of boiler heat and 1500 kWh of solar heat.

In 2001, the total life cycle cost of a complete system comprising the collector and the absorption unit, for a 20-year period was €22,700 with an estimated price for the absorption unit together with its accessories equal to € 8,140. In order to have the same total life cycle cost the price of the absorption unit together with its accessories should not be higher than € 3,400. This could be possible only when absorption units were mass-produced.

Today (in 2007) the final optimum system, as obtained from the complete system simulations, consists of 35 m^2 compound parabolic collectors. The only limitation is the available area on the house roof, which cannot be completely blocked. The price of diesel in 2007, which is € 0.73 per lt indicates that it is economical to replace about 20,000 kWh with solar energy collected with 35 m^2 of CPC and the life cycle savings of such a system are € 7,700. The total life cycle cost of a complete system comprising the collector and the absorption unit, for a 20-year period is € 38,200.

CONCLUSIONS

The final optimum system for 2007 as obtained from the complete system simulations, consisted of 35 m^2 compound parabolic collector tilted at 30° from horizontal and 600 liters hot water storage tank. The typical insulated house considered, requires an annual cooling load at 25°C of 17,600 kWh with a peak load of 10.3 kW and an annual heating load at 21°C of 3,530 kWh with a peak load of 5.5 kW. This annual load could be met by using about 28,100 kWh of boiler heat. The price of diesel in 2001, which was € 0.29 per lt indicated that it was economical to replace about 10,000 kWh with solar energy collected with 15 m^2 of CPC and the life cycle savings of such a system were € 320. The life cycle savings for the present system and 2007 fuel price (€ 0.73 per lt) give life cycle savings of € 7,700. The increase of the cost of fuel compels consumers to move to other energy forms in order to meet their needs. So gradually renewable energy forms can economically replace part of the fuel energy as the fuel price increases.

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The Effects of External Fluids Parameters on the Performance of District Heat Water Driven Absorption Chillers

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SUMMARY

District heating and cooling plays a great role to improve CHP plants efficiency and reduce global warming gas emission to meet Kyoto Protocol. DH energy driven absorption cooling can balance the production of heat and power in CHP plants, and reduce the summer power peak load caused by huge cooling load of building air conditioning systems. All fluids loops related to chillers have a great influence on the chillers' performance. With a simulation program developed by the author, the paper studies and simulates their effect on chillers operation so as to find the solutions to improve its performance.

1. INTRODUCTION

In some European and Asian countries, district heating (DH) systems are commonly used due to the cold winter [1], but summer cooling by separate summer electric power driven compression chillers. Two sets of system means big initial investment for building air conditioning. The integration of building cooling with existing heating systems sounds a nice solution to provide building air conditioning round the year.

District heating and cooling plays a great role to improve CHP plants efficiency and reduce global warming gas emission to meet Kyoto Protocol [2]. DH energy driven absorption cooling can balance the production of heat and power in CHP plants [3], so as to reduce the summer power peak load due to the huge cooling load of building air conditioning systems. In addition to the impacts of chillers' design on their performance, all fluids loops related to chillers have also big influence on the performance. This paper studies and simulates the effect of the factors below on the performance of DH water driven absorption chillers [4].

a. Diluted water/lithium bromide solution flow
b. Cooling water temperature
c. Chilled water outlet temperature
d. Chilled water inlet temperature
e. Driving water inlet temperature to chillers
f. The temperature drop of driving water through chillers
g. Driving water flow
h. Chillers' operating output

2. SIMULATION

One simulation program is developed by the author with EES (Engineering Equation Solver) as the platform which was developed by F-Chart Software [4]. EES is mainly designed for thermodynamics and heat transfer simulations, which has been extensively used as the platform for other further simulation programs development.

LG low temperature water driven absorption chiller is taken as the prototype for the simulation [5]. Figure 1 presents the single effect absorption chiller's (SE ACC) model. The nominal chilled water capacity is 1000kW$_c$.

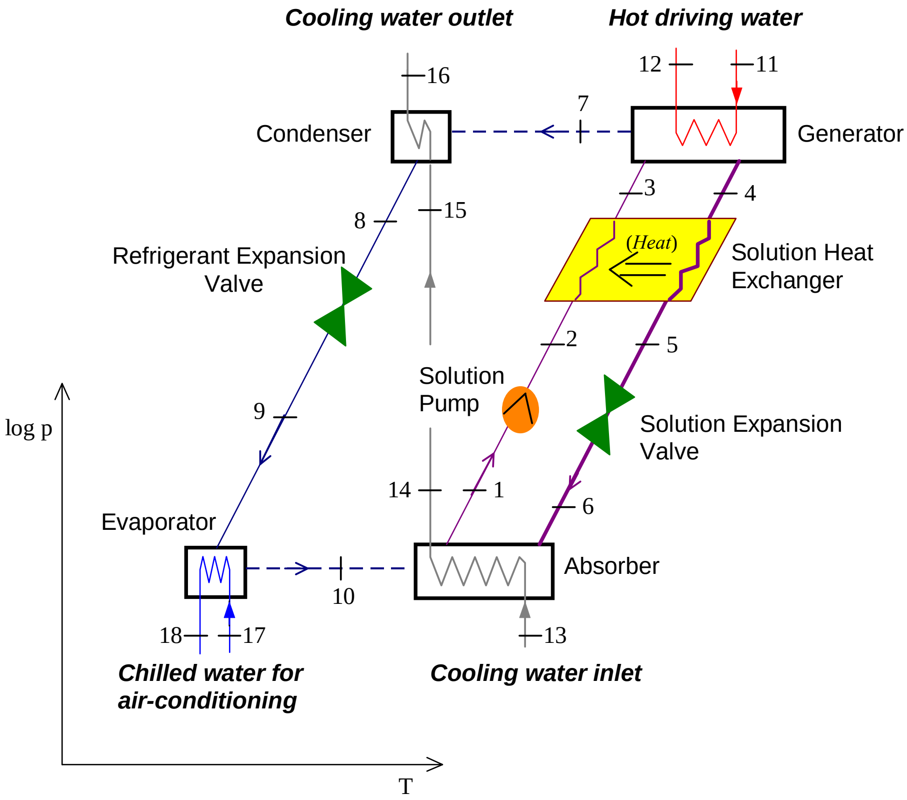

Figure 1 SE ACC model

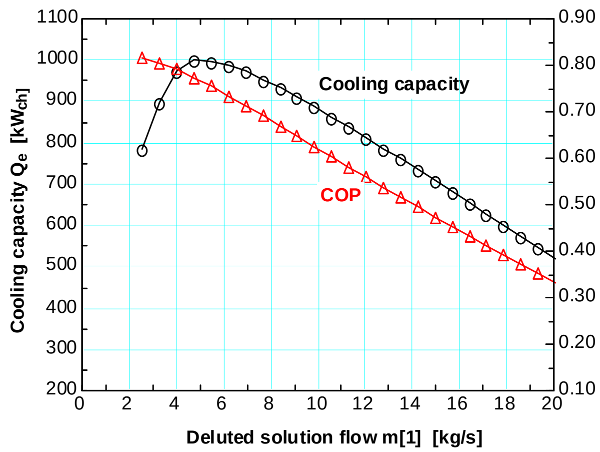

Figure 2 The impact of diluted solution flow on chillers' performance.

2.1 Diluted water/lithium bromide solution flow

Normally, SE ACCs in the market have constant diluted solution flow rate, i.e. constant flow of solution pump is installed. However, diluted solution flow rate has a strong effect on the performance when other parameters keep constant shown in Figure 2. There is a maximum value when diluted solution flow at 6kg/s, depart from that, the cooling capacity is lower.

Simulations show that the diluted solution flow at which cooling capacity reaches it maximum value, called as optimised diluted solution flow (ODSF) changes when other 6 external temperature changes. As driving water temperature increases and cooling water temperature decreases, ODSF goes up so that maximum cooling capacity is higher; whereas, when driving water temperature decreases and cooling water temperature increases, ODSF goes down resulting in a lower cooling capacity. Therefore the diluted solution pumps should be speed-controlled to meet the ODSF at different external temperature.

2.2 Cooling water temperature

Cooling water temperature has a great influence on chillers' performance. Maintaining driving water temperature, chiller's cooling output and COP increase as the decrease of cooling water temperature, shown in Figure 3. When cooling water temperature decreases from 31^{o}C at design mode to 25^{o}C, and driving hot water temperature keeps at 88/78^{o}C, cooling capacity of the chiller increases from 1000kW to 1526kW, and COP goes up from 0.691 to 0.755; on the contrary, if cooling water temperature increases from 31 to 35^{o}C, cooling capacity decreases from 1000kW to 440kW linearly, and COP goes down from 0.691 to 0.57.

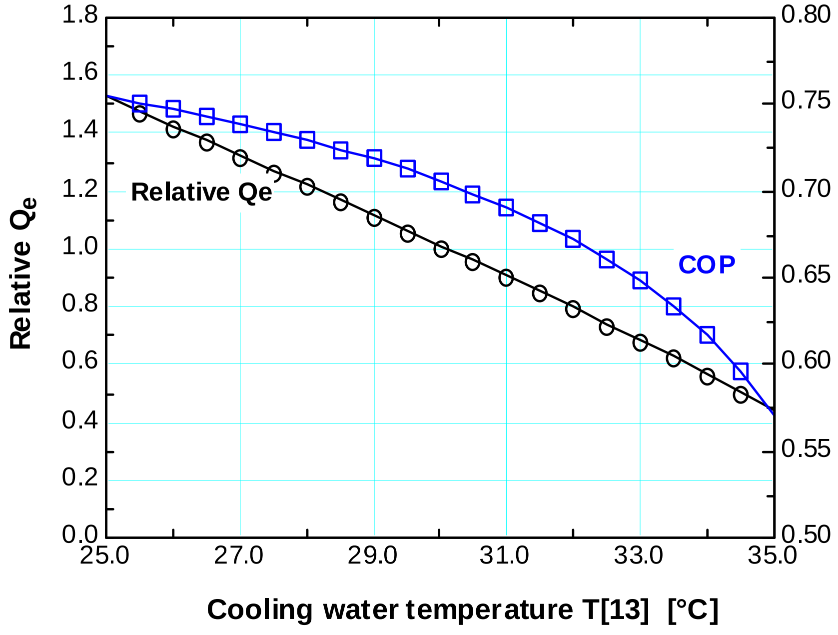

Figure 3 Cooling water temperature VS Chillers' performance

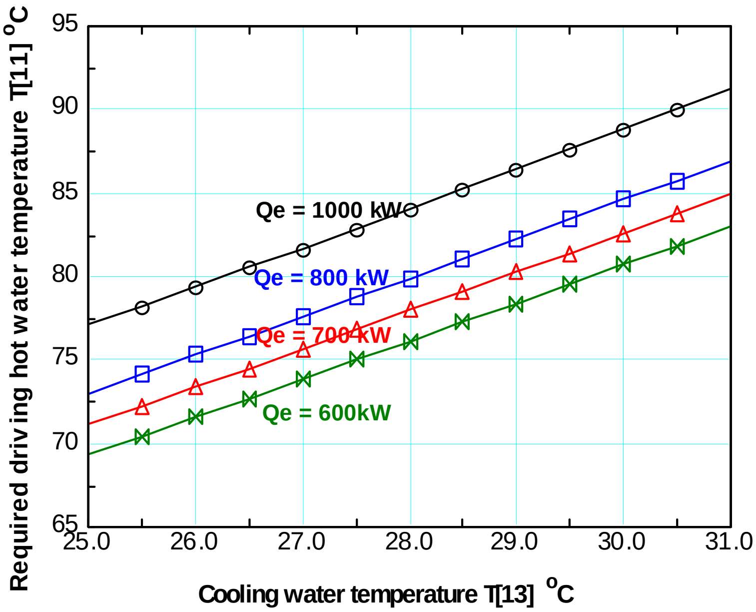

Figure 4 Cooling water temperature VS driving hot water temperature

It is obvious that cooling production of the chiller decreases 11.8%, i.e. 118kW when cooling water temperature has 1°C increase.

At design mode, the cooling water temperature at 31/36°C, but if cooling water temperature drops to 25°C by some ways, for instance heating swimming pool water with cooling water, the required driving hot water temperature can be decreased considerably shown in Figure 4. When cooling water is 25°C, it is possible to reduce the driving hot water temperature from 88 to 77°C that is very suitable for DH system in summer. As the decrease of chiller's capacity, the required driving hot water temperature decreases correspondingly. When chiller capacity is kept as 800kW, the required driving water temperature is 85 and 73°C when cooling water temperature are 30 and 25°C respectively.

Since the benefits of lower cooling water temperature, and its effective influence on chillers' performance, the solutions to reduce cooling water temperature have to be found.

2.3 Chilled water outlet temperature

The simulation results when change chilled water temperature are shown in Figure 5 and 6, under certain amount of chiller output, increase chilled water temperature level can improve chiller's performance and reduce required driving water temperature.

Increasing chilled water temperature results in the higher heat transfer potential of evaporator, i.e. greater temperature difference between refrigerant and chilled water. When keeps other parameters, increased chilled water temperature causes higher chiller output. While keeps the certain amount of chiller output, driving energy input decreases, thus COP goes up.

Refer to Figure 6, DH supply water temperature entering to chiller generator keeps at 88°C and chilled water output at 800kW_{ch}, when chilled water outlet temperature increases from 8°C to 14°C, driving water exit temperature will decrease from 76.2°C to 66°C, as the result it can return to DH network directly, and driving water flow decreases from 22 to 11.8kg/s.

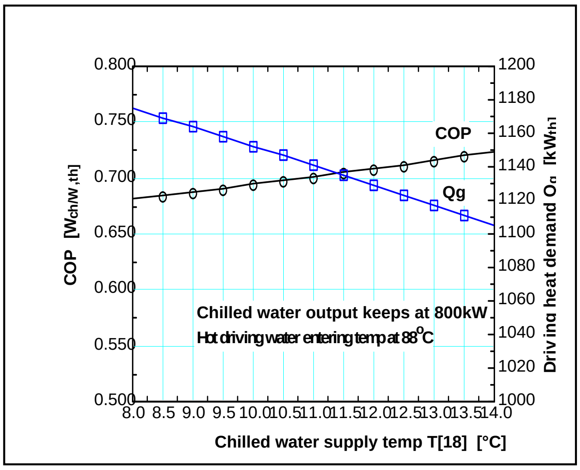

Figure 5 Chilled water temp vs chiller COP and driving heat demand

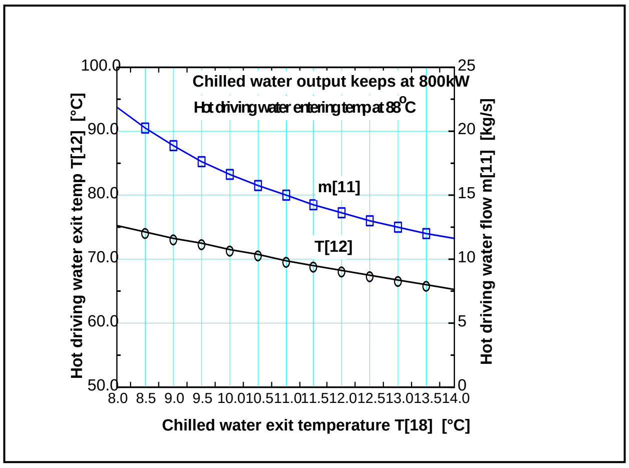

Figure 6 Chilled water temperature vs driving water exit temperature and flow

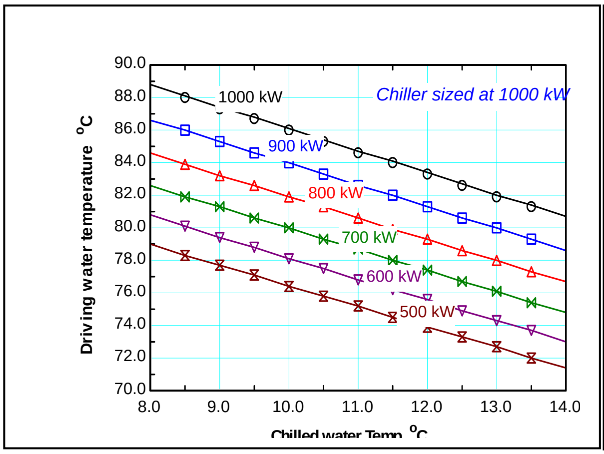

Figure 7 Driving water temperature VS chilled water temperature at different chillers output

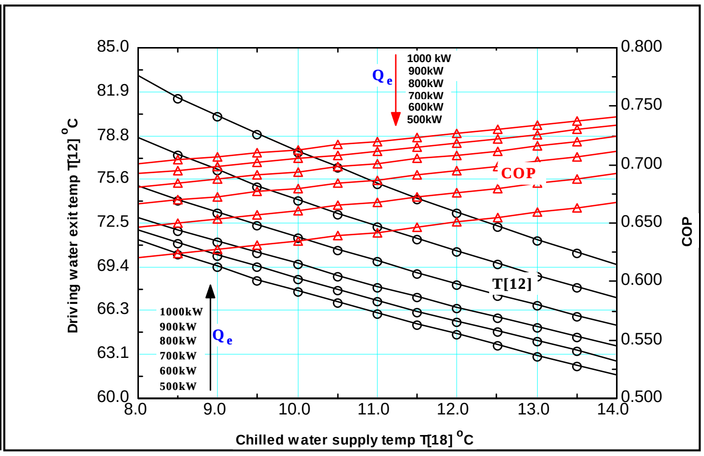

Figure 8 Driving water exit temperature and COP VS chilled water temperature at different loads

Figure 7 shows the results of how chilled water temperature influences the required driving water temperature when chiller's output varies with the chiller sized at 1000 kW. If the chiller runs at its full capacity 1000kW, the required driving water temperature decreases from 88 to 80.7°C when chilled water temperature increases from 8 to 14°C. When the chiller's output is 500kW, the required driving water temperature decreases from 79 to 69.5°C corresponding the same range of increase of chilled water temperature.

When driving water temperature keeps at 88°C, the driving water exit temperature and COP values at different chiller's output are illustrated in Figure 8 if chilled water temperature increases from 8 to 14°C. With the increase of chilled water outlet temperature, the driving water exit temperature decreases. the driving water exit temperature goes down as the reduction of chiller's output.

Figure 9 presents the chiller's output at increased chilled water outlet temperature. When the outlet temperature rises from 8 to 13°C, the chiller output increases from 884 to 1195kW, and COP increases from 0.691 to 0.743.

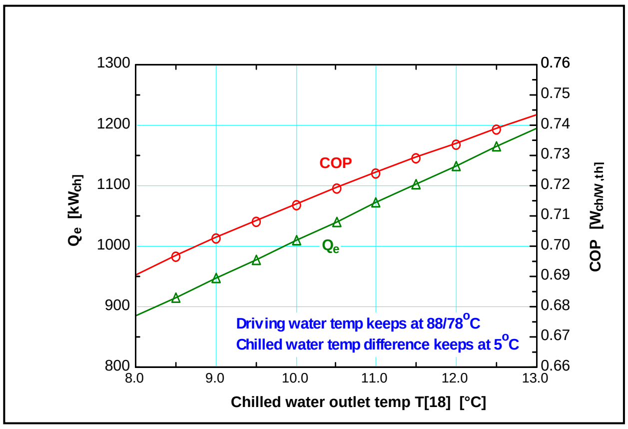

Figure 9 Chilled water outlet temperature VS COP and chillers' output

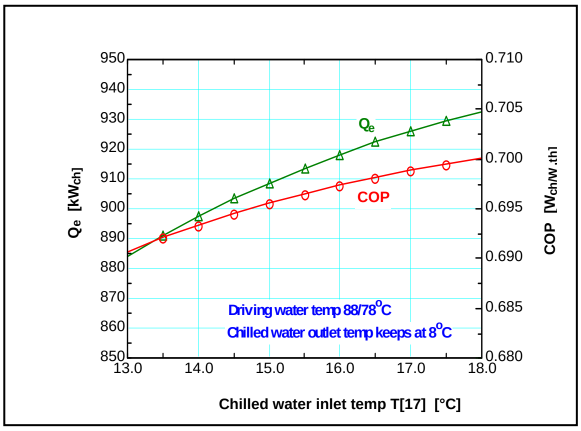

Figure 10 Chilled water inlet temperature VS COP and chillers' output

2.4 Chilled water inlet temperature

Chilled water inlet temperature impacts chillers' performance as well. Shown in Figure 10, keep chilled water outlet temperature at 8^{o}C, but increase chilled water inlet temperature, the chilled water output and COP goes up. When the inlet temperature rises from 13 to 18^{o}C, the chiller output increases from 884 to 933kW, and COP increases from 0.691 to 0.7.

To increase of chilled water outlet temperature has much stronger influence on chiller's output and performance than that of chilled water inlet temperature does.

2.5 Driving water temperature to chiller

Keeping all other parameters at design mode, the influence of driving water temperature on chillers' performance is simulated, and the results are shown in Figure 11 and 12 for driving water temperature drop at 5^{o}C and 10^{o}C, respectively. If driving water temperature drop keeps at 5^{o}C, when driving water temperature decreases to 75^{o}C, chilled water output decreases to 365kW. If driving water temperature drop keeps at 10^{o}C, the minimum driving water temperature is 78^{o}C when the chilled water output is only 311kW. The chiller output is more sensitive to driving water entering temperature than exit temperature.

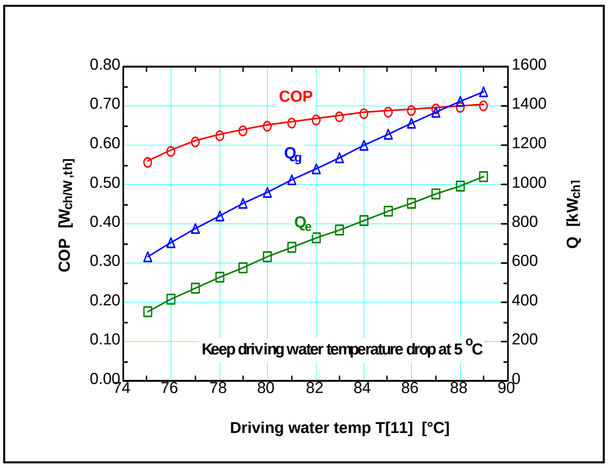

Figure 11 Driving water temperature VS COP & chiller output and driving heat demand in case of driving water T drop at 5^{o}C

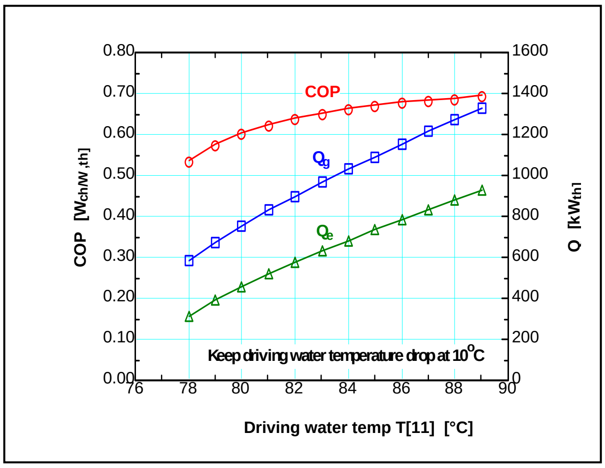

Figure 12 Driving water temperature VS COP & chiller output and driving heat demand in case of driving water T drops at 10^{o}C

2.6 The Temperature drop of driving water through chillers

It is found that driving water temperature drop through chillers' generator impacts its performance. Maintaining driving water temperature at 88°C and other parameters, to simulate the influence of variable temperature drop, the results are shown in Figure 13.

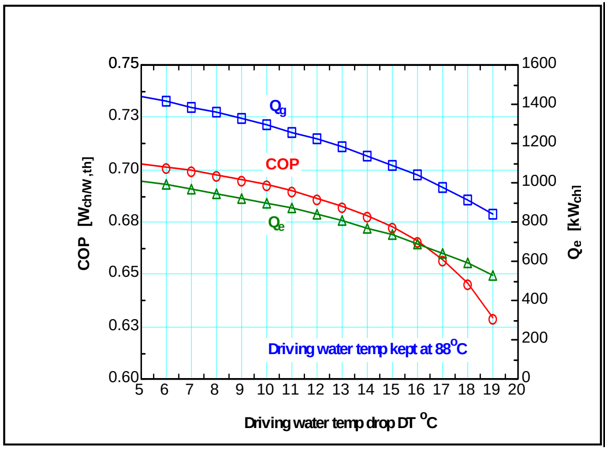

Figure 13 Driving water temp drop VS performance **[Chiller sized at 1000kW when the drop is 5°C]**

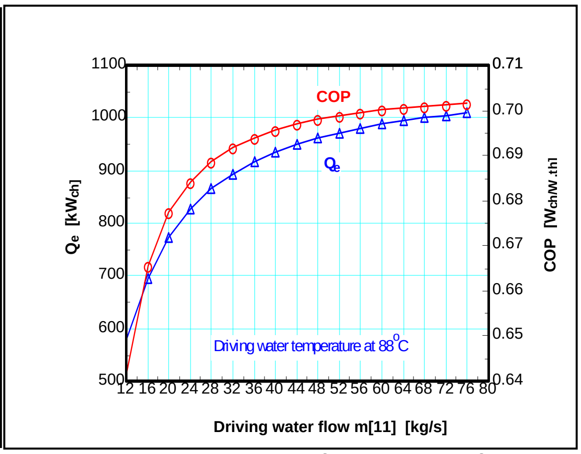

Figure 14 Driving water flow VS performance

If driving water temperature drop increases, the performance of chiller depreciates. When it rises from 5 to 19°C, chiller's output decreases from 1000kW to 527kW, and COP from 0.703 to 0.629.

2.7 Driving water flow

Figure 14 gives the correlation between driving water flow and chillers' output when other parameters are kept as constant. The higher of driving water flow, the better of performance. Figure 15 presents the change of driving water exit temperature as the flow varies.

When driving water flow is small, to increase the flow can bring a big rise of chillers' output and COP. But as the continuously increasing, the effect becomes weak.

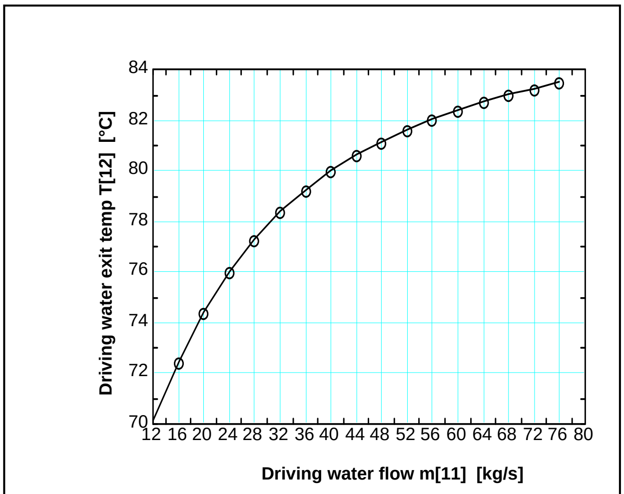

Figure15 Driving water exit temperature VS driving water flow

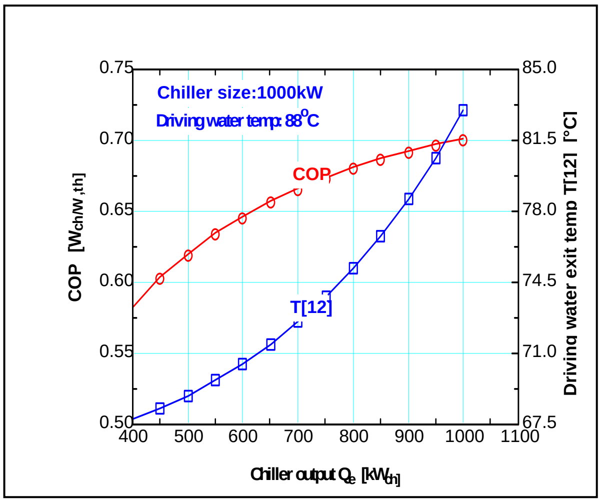

Figure 16 Chiller performance VS its operating output

2.8 Chillers' Operating Output

For the chiller sized at 1000kW at 88/83°C, as its operating output changes, the performance varies as well, Figure 16 gives the simulation results.

When its operating output goes down from 1000kW to 400kW, COP value drops from 0.7 to 0.583, but the driving water temperature drop increases. As the result, the performance depreciates as the decreasing operating output.

3. CONCLUSIONS

Simulation is the basis for manufactures to optimise their products, for designers to design a perfect district cooling station, for operators to run the station in an economical manner. The simulations implemented in this paper give the hints how the external fluids and their parameters influence the cooling capacity and performance of certainly sized absorption chillers.

Diluted solution flow has strong effect on the performance of chillers. When external fluids parameters vary, the optimised diluted solution flow at which maximum cooling capacity is reached will change as well. As the facts of the varied external fluids parameters, the solution pumps should be speed-controlled to reach the optimised performance of chillers.

Absorption chiller's performance can be improved through the following solutions:

- Control diluted water/lithium bromide solution flow
- Lower cooling water temperature;
- Higher chilled water temperature;
- Higher driving water temperature;
- Lower driving water temperature drop through chillers' generators;
- Higher driving water flow;
- Operate chillers at high output.

ACKNOWLEDGEMENT

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High Performance Cooling in Buildings
- The Centre for Sustainable Building (ZUB)[*]

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SUMMARY

The process of developing environmentally friendly and sustainable building structures is an interdisciplinary task. Only with a close teamwork of people from different building related disciplines, is it possible to face the challenge of solving the real life problems involved in an integral planning process. The construction of the ZUB is an example of what today's state of the art of technology for low energy demand and sustainable building can look like.
Planned as an example of a low energy building, the new office of about 1300 m^2 is attached to an existing preserved building and consists of three different main parts: one for exhibitions and events, one for offices, and an experimental part for different kinds of research in innovative building techniques and building services. A special focus was given to the realisation of a sustainable cooling concept, using a renewable energy source: the coolness of the ground underneath the building.

During the first two and a half year long intense monitoring period, about 1300 points of data were recorded in the entire building, the achievements have been proven and optimisations based on the measurements have been made. Results and findings from the intense measurements and from an additional three year long measurement period have also been collected. The project offers an opportunity to obtain actual proof and quantify the achievements of a good sustainable building. In addition, it highlights today's possibilities in the field of energy conservation design.

INTRODUCTION

Building a sustainable building is an interdisciplinary task, in which a close teamwork of people from all different building related disciplines is needed to face the challenge of answering the real life problems of this process. That is one reason for the foundation of the Centre of Sustainable Building at the University of Kassel, Germany. However, not only building research and the development of integrated planning processes in the field of building are conducted within the walls of the centre. Moreover, the centre is a platform for the fast implementation of research results into practical building processes. It offers a variety of different services in research and development, and in education for all kinds of people who are interested in building (professionals and non-professionals). Furthermore, public relation activities are carried out at the ZUB to present ideas for better, more efficient and reliable building while having the smallest possible impact on the environment. The building of the ZUB is an example of what today's state of the art technology of low energy demand and sustainable building can be like [1],[2],[3].

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[*] Zentrum für Umweltbewusstes Bauen in Kassel, Germany. (The Centre for Sustainable Building)

In modern and energy efficient office buildings, there is interaction between the architectural design, the construction of the building envelope, and the building services equipment. These factors combine to integrally influence the resulting indoor climate. During the project presented here, these interactions have been identified. By optimising all factors, it is possible to secure a comfortable indoor environment by consuming only a minimum amount of energy.

THE BUILDING CONCEPT

The office building of the Centre for Sustainable Building is situated at the University of Kassel, in an old urban neighbourhood. The new building of the ZUB closes a gap in an ensemble of old houses. An atrium, used as a gap of light, containing the entrance zone and the staircases, joins the old brick building of the Faculty of Architecture to the modern concrete construction, thus combining old and new.

Figure 1. The office building of the Centre for Sustainable Building in Kassel, Germany [4].

The ZUB office building consists mainly of three different parts: one part for exhibitions and events, one part for offices and an experimental part for different kinds of research in innovative building techniques and building services concepts. Some space, with non-shaded conditions, has been provided for test equipment on the flat roof.

The building envelope construction

The load bearing skeleton in reinforced concrete consists of round pillars with 5.40 m spaced between them and flat concrete slabs for the floor/ceiling construction. The U-value of the exterior concrete walls, with 30 cm of polystyrene insulation, is 0.11 W/m^2K, and triple glazing with a U-value of 0.6 W/m^2K and a g-value of 0.48 have been chosen for the mainly south facing large window area. The minimal frame-fraction of the wooden façade construction helps to reduce heat losses. Furthermore, the problem of thermal bridges has been diminished by carefully planning all joints and details. Good thermal insulation is mandatory in achieving minimal heating energy demand for the entire building.

Table 1. Main building facts

		Building part	U-value
Volume	6882 m^3		
Net floor area	1348 m^2	Exterior walls	0.11 W/m^2K
Main floor space	830 m^2	Roof	0.16 W/m^2K
		Windows	0.80 W/m^2K
Area/volume ratio	0.34 m^{-1}	Wall/floor against ground	0.26 W/m^2K
		Mean U-value	0.32 W/m^2K

The high insulation standard has dramatically reduced the energy demand. For the ZUB building, the calculated heating demand according to the WSVo'95 (old German energy code) turned out to be 25 kWh/m^2a, which is 27% of the limiting value. The heating demand according to the new and actual German energy code EnEV is 21.3 kWh/m^2a [2].

These higher standards caused, on one hand, some more initial costs but, on the other hand, some special savings became possible. One example of this is that it was possible to reduce the size of the building equipment. Finally, the costs of the ZUB building were in the range between a commonly built and a low cost office building, according to the German building costs index [5]. Also, the other buildings examined during the named research program, the EnBau program, show the same cost structure as commonly built offices in Germany.

Table 2. Building costs per net floor area of the ZUB building compared to different German standards

	ZUB	Low cost [5]	Common [5]	Advanced [5]
Construction	768.50 €/m²	772 € /m²	979 € /m²	1284 € /m²
Services equipment	300.60 €/m²	151 € /m²	268 € /m²	408 € /m²
Building costs	1069.10 €/m²	923 € /m²	1247 € /m²	1692 € /m²

THE SUCCESSFUL BUILDING SERVICE SYSTEM CONCEPT

During the planning phase, there was already an emphasis on a good interaction between the used building construction and the chosen building service systems. A lot of passive and natural means were utilised for the heating, cooling and ventilation of the building, and it was possible to reduce the use of building service equipment to a minimum. The results of the evaluation from the concomitant scientific research program, as shown in the figure, point out the success of the concepts which have been executed.

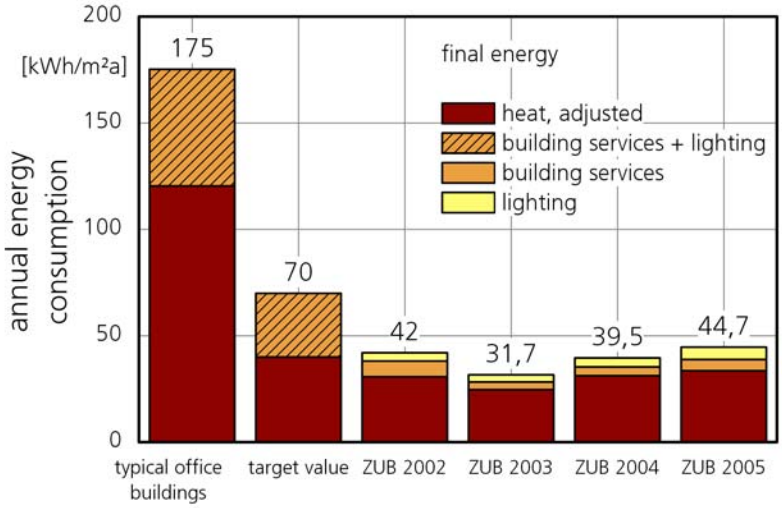

Figure 2. Specific annual energy consumption for heating, building services and lighting of the ZUB building compared to the target values of the German research program EnBau and to existing typical office buildings [6].

The central points of the outlined achievements are the good insulation standard and the utilisation of aligned building service equipment, which is described in its three most important parts: heating and cooling, ventilation and lighting.

Soft heating and cooling with thermally activated building constructions

All office rooms and the lecture hall of the ZUB building are equipped with a surface heating and cooling system, with thermally activated building constructions. On all floor slabs, conventional floor heating has been installed in addition to the activated ceilings. This has been done for research purposes in order to test and verify different regulation strategies. The pipes of the ceiling systems are laid about 15cm apart, directly on the lower reinforcement layer. Each office room is equipped with a separately regulated heating/cooling circuit in the ceiling and in the floor slab. Via a regulation of the mass flow of the heat carrier, water, an individual energy supply and demand controlled heating and cooling is possible. In the case of heating, the systems are supplied with district heating, which is available from the neighbouring university building (see Figure 1).

Figure 3 . The pipes of the thermally activated slabs directly on the lower reinforcement layer

The main general difference between the other radiant systems and thermally activated building construction, as used in the ZUB, is the possible asynchrony of the operation of the conditioning plant and thermal loads, i.e. the slab has the opportunity to store heat or cool at different times, with respect to thermal loads.

The main advantages for the thermally activated constructions are the following [7]:

- the thermal load is distributed in a longer period, which leads to lower peak loads, thus allowing the use of conditioning plants of reduced sizes
- the possibility to use two radiant surfaces leads to more uniform conditions in the conditioned space and better thermal comfort conditions
- the buildings have reduced dimensions in relation to buildings with suspended ceilings
- it is possible to use conditioning plants suitable for temperatures close to room temperature, i.e. heat pumps, condensation boilers, solar collectors, or ground heat exchangers as used in the ZUB
- for cooling purposes, the night over-ventilation can be used as well
- low installation costs and low operation costs are possible.

The use of thermally activated systems on the other hand requires the following [7]:

- active thermal slabs are used in multi-storey buildings with a central plant
- attention has to be paid in the case of raised floors, while the ceiling surfaces have to be free from obstacles (no suspending ceilings can be used, which could cause difficulties for the room acoustics)
- the design of this type of radiant systems is very critical; adequate solar radiation screens are needed as is good thermal insulation of the envelopes, which has to be lower than 1 W/m^2 K and better than 0.6 W/m^2 K, the transmitting insulation (shading elements). The ZUB building is a good example which shows how these challenges can be met.

A great glazing fraction of a façade, such as that of the south facing façade of the ZUB , leads to an increased demand in cooling if no highly efficient sun shading is installed. This has been done in the ZUB by installing external blinds. To reach particular comfortable indoor conditions, the thermally activated constructions are also used as a soft cooling system. The operational temperatures for these systems are relatively low for heating (27°C) and high for cooling (20°C). This fact allows for the possibility of using renewable cooling sources, e.g. the coolness of the ground under the building. In the ZUB, a so-called ground heat exchanger has been installed.

The ground heat exchanger

The utilisation of low water supply temperature (of 25 – 40°C) for in the case of heating and high temperatures (16 – 20 °C) for cooling, makes it possible for renewable energy sources to be used in the system. In cooling conditions, it is possible to use renewable energy sources as well, for example ground heat exchangers (as used in the ZUB) or rain water ground accumulators. Due to the high thermal inertia, the thermally activated building constructions can even be loaded during the night.

Similar to the construction of the activated floor slabs, there are pipes in the bottom slab of the building. Via thermal contact to the ground, the flowing water inside the pipes is cooled and via the circulation, it is supplied to the ceiling and floor systems in the office rooms. Measurements have shown that the ground heat exchanger works with a calculated COP of 23, in comparison to a normal mechanical cold production with COP’s of about 3.5. The system used in the ZUB is about 6.5 times more energy efficient than a conventional cooling system. A common German office building uses about 30 kWh/m²a primary energy for the climatisation (cooling) of the office spaces [5]. In this figure, the energy use for the pumps is not included. In the ZUB building approximately 5300 kWh of cooling is delivered per year to the offices, using approximately 270 kWh of electricity for the pumps. The primary energy use turns out to be 0.6 kWh/m²a for the cooling.

The cooling power, which can be utilised at low operational costs, is not arbitrarily adaptable. It depends strongly on the temperature and water flows (moisture) of the ground. For very dry grounds, the cooling potential could be worn out after a few weeks as the ground temperature rises and only a very limited cooling power can be used. In the case of flowing ground water, it is possible to use greater cooling power and rooms can be cooled more intensively.

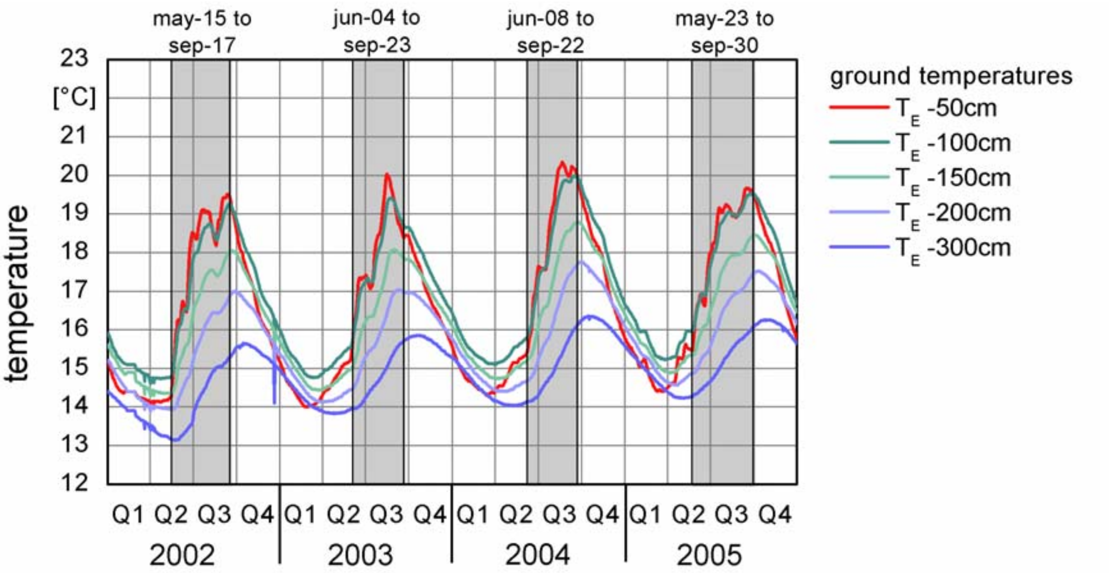

Figure 4. Measured temperatures in the ground below the ZUB building, which is used as the heat sink for the cooling system.

As shown in the figure, the ground temperature rose slightly from the cooling season in 2002 to 2004 by about 1K/a in 3m below the ZUB. This trend was stopped in the later cooling seasons, the ground temperature returned to the same values after cooling during periods. The management of the cooling source works.

Thermal conditions in the ZUB offices

A main concern during the course of the project was the question whether or not the limited power output of the thermally activated constructions would be sufficient to cover the demands.

Simulations showed that a constant cooling power of about 8 W/m^2 is reasonable for the ZUB. The maximum measured cooling power was 40 W/m^2 and the maximum heating power was 80 W/m^2, which is also shown in the following figure. These values are in the same range as indicated in the literature [7].

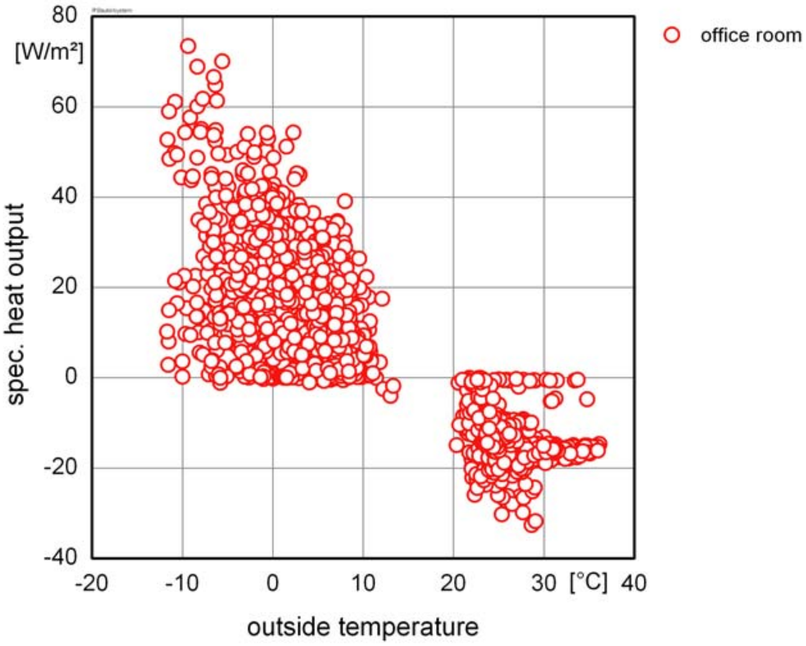

Figure 5. Measured specific heating and cooling power of an office room in the ZUB

With these values, thermal comfort could be secured during the measurement period. The overheating hours, hours with room temperature above 26°C, could be diminished by using this system up to only 125 h or 4% of the occupancy time in a representative office room during the “summer of the century” in 2003. An intensive study on the comfort conditions in the ZUB based on questionnaires showed that the occupants were very pleased with the conditions in the building and accepted the slight changes in room due to the heating, cooling and ventilation system. This is proof of the excellent thermal comfort of the ZUB building.

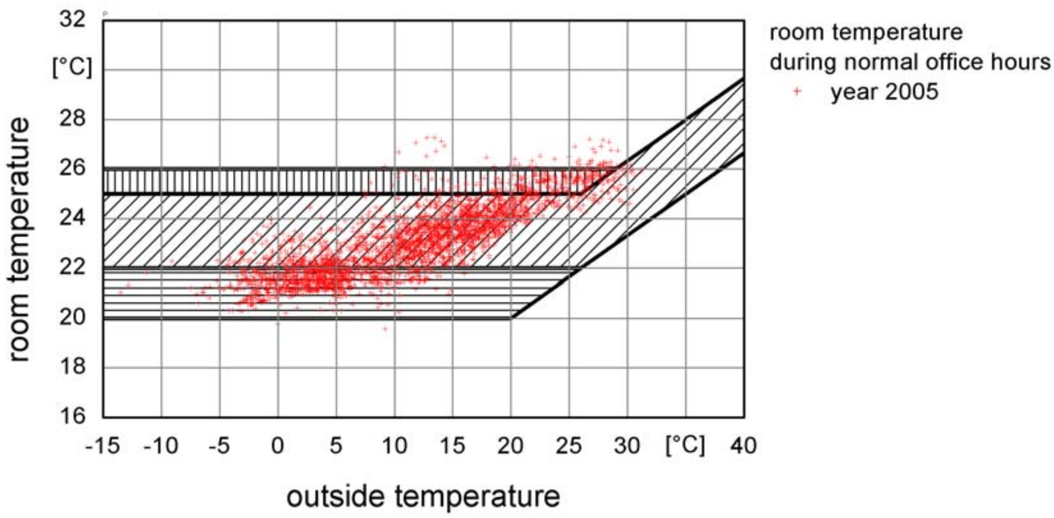

Figure 6. Measured room temperatures in an office room in the ZUB depended on the comfort ranges given in the German Standard DIN 1946 part 2 for the year 2005.

CONCLUSIONS

The project of the ZUB building is actual proof that high demands on commercial buildings with regard to energy efficiency in heating and cooling, as well as a rational use of electricity, can be met. The first two year long intensive monitoring period provides evidence for the fact that if all crafts are integrated into the planning and building process in a holistic manner, it is possible to build houses which are trendsetting in the sense of energy efficiency and indoor comfort. The combination of a high insulation standard and the solar optimised orientation of the building, with thermally activated building constructions for heating and passive means for cooling, are the framework for energy optimised buildings in Northern (European) climates. The use of the ground heat exchanger, which is especially expedient when external and internal load from machines, computers and persons can be reduced, shows a very high acceptance and a high energy efficiency. By using external sun shadings and energy saving office equipment, good indoor thermal comfort can be reached at all times.

ACKNOWLEDGEMENT

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Comparative study of the performances of a buried-pipe ventilation system and an indirect evaporative cooler operating in a care home for old people

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SUMMARY

Since the heat wave of 2003, summer thermal comfort in buildings and more particularly in residences for the elderly has become in France one of the major concerns of building owners. Hence they have been requesting for finer thermal simulation studies. Within the framework of a care home project for dependent elderly people, two ventilation systems have been studied: a double-flow system with possible adiabatic indirect evaporative cooling and a buried-pipe system coupled to a double-flow-type blowing network.

The building model studied has been defined using TRNSYS software. Simulations were carried out using the annual weather data file of Toulouse (south of France). The performance analysis in terms of energy consumption and thermal comfort (evaluated by a number of hours of overheating) of the double-flow and buried-pipe systems is carried out over the whole year and seems favourable compared to the results of a basic simple-flow ventilation system.

INTRODUCTION

In France, since the heat wave of summer 2003, the thermal study for new buildings has become finer especially in the case of care homes for dependent elderly people. Now, this type of building has to satisfy the new French thermal regulations (RT 2005) in terms of energy consumptions for space heating and thermal comfort during summer and in-between seasons (between winter and summer and between summer and winter). These regulations also encourage not to use cooling systems in order to reduce summer temperatures. This study compares the performances of different ventilation systems. A double-flow system using an air heat exchanger to recover the heat of the extracted air in winter and equipped with an adiabatic indirect evaporative cooling system in summer, and a buried-pipe system are studied and compared to a reference solution, which is a simple-flow ventilation system. The 45-bedroom care home for elderly people is situated in Toulouse (south of France).The thermal simulations are run using TRNSYS software.

DOUBLE-FLOW VENTILATION SYSTEM

Description of the system

The ventilation box (figure 1) is composed of a blowing fan, an extraction fan, a static heat recovery exchanger and a lost water humidifier placed on the extraction side. This feature is only working during summer. An electronic gate commands the cold water (CW) that flows on a web weaved in polyester threads. The excess of water is evacuated through a siphon. The box is also equipped with dampers controlled by an on-off system that allow the by-pass of the exchanger in the hotter seasons.

Thus outside air can follow three ways: a preheating in winter through the heat recovery exchanger, an indirect adiabatic cooling in summer and a direct blowing with no previous treatment when the exterior and interior conditions are favourable during the in-between seasons (free cooling).

The conditions of operation of the double-flow system are detailed in Table 1. To avoid any conflict in the regulation strategy and to limit short sequences, the laws of control integrate a hysteresis of 1°C on set temperatures.

Table 1. Operation modes of the double-flow ventilation system

Period	**Temperature conditions**	**Air treatment type**
Winter	Tamb ≤ 17 °C	outside air preheating
Winter	Tamb > 17 °C	Direct ventilation
Summer	Tin > 22 °C et Tamb > Tint	Indirect evaporative cooling
Summer	Tin ≤ 22°C	Free cooling

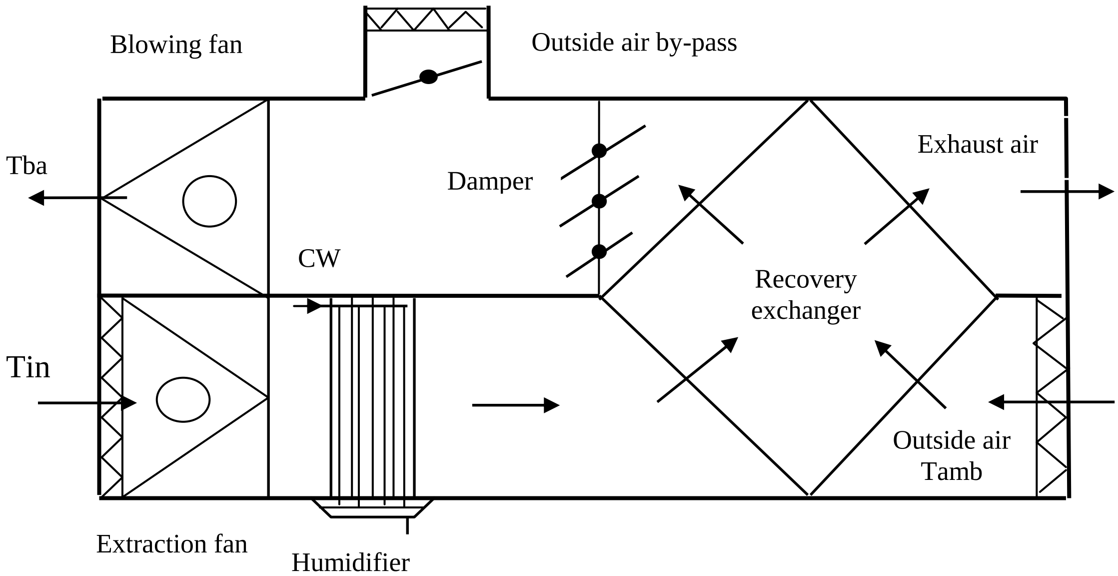

Figure 1. Double-flow ventilation box

Outside air preheating

In winter, the outside air is preheated through the heat recovery system by exchange with the exhaust air. Considering the efficiency ε of the exchanger and an increase of 0.4°C by passing through the fan, the supply air temperature is given by equation 1:

$$T_{sa} = T_{amb} + \varepsilon \times (T_{in} - T_{amb}) + 0{,}4 \qquad (1)$$

Where T_{sa} is the supply air temperature, T_{in} is the inside temperature and T_{amb} is the ambient temperature. The mean value of the efficiency is estimated at 0.75, which corresponds to measurements already carried out on mechanical double-flow ventilation systems [1]. For ambient conditions of 21°C and 50% of relative humidity and exterior conditions of 5°C and 80%, the supply air temperature is 17.4°C. The evolution of the air characteristics is represented on figure 2.

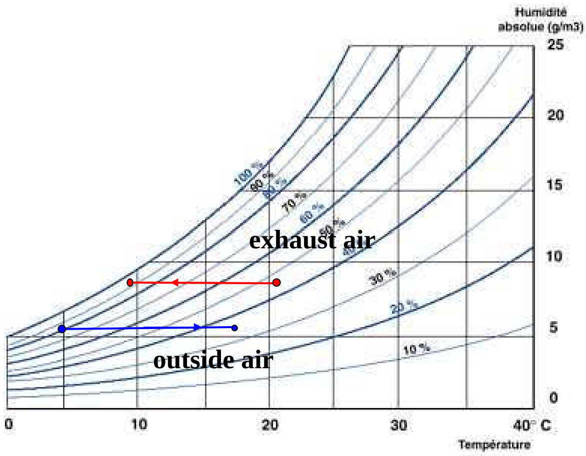

Figure 2. Evolutions in outside air preheating on the psychometrics diagram

Direct ventilation, free cooling

During hotter seasons, the building can be ventilated directly. The dampers of the ventilation box enable the outside air to by-pass the recovery exchanger. The fan slightly increases the temperature of the air (0.4°C in equation 2). Depending on the temperature levels inside and outside, this operation mode can lead to a direct refreshment of the building (free cooling).

$$T_{sa} = T_{amb} + 0{,}4 \qquad (2)$$

Indirect adiabatic evaporative cooling

In summer, when the inside temperature goes over 22-23 °C and the outside temperature is even higher, the double-flow system can work as a cooler. Exhaust air is adiabatically cooled through the humidifier following a wet-bulb isotherm. The temperature of the exhaust air decreases by around 6°C. Then in the static exchanger the outside air is refreshed by the humidified and cooled exhaust air. The evolution of the characteristics of the supply air and the exhaust air are represented on figure 3.

The value of the humidifier's efficiency ε_h is estimated at 0.9 and the lost water flow is around 15% of the evaporated water in the air flow. These values correspond to mean values measured on this type of systems [1]. The characteristics of the outside and exhaust air are given by equations 3, 4 and 5.

$$\varepsilon_h = \frac{(\omega_{hea} - \omega_{in})}{(\omega_{sat} - \omega_{in})} = 0{,}9 \qquad (3)$$

$$T_{wb,in} = T_{wb,sat} = T_{wb,hea} \qquad (4)$$

$$T_{sa} = T_{amb} + \varepsilon \times (T_{hea} - T_{amb}) + 0{,}4 \qquad (5)$$

hea, wb and sat in subscript mean respectively humid exhaust air, wet bulb and saturation. For example, if the inside temperature is 25°C and the outside temperature 30°C, the humidified air temperature is 19°C and the supply air temperature is 22.2°C.

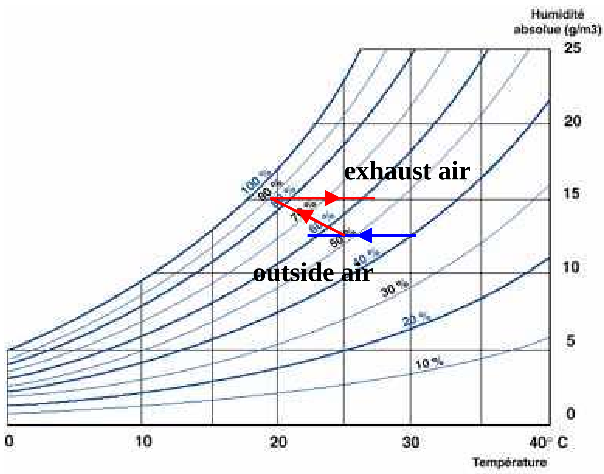

Figure 3. Evolutions in indirect evaporative cooling on the psychometrics diagram

BURIED-PIPE VENTILATION SYSTEM

Description of the system

The buried-pipe system enables air preheating in winter and also air cooling in summer. It is connected to a blowing network similar to the one used in a double-flow ventilation system. The exhaust air is extracted from the technical premises by several air outlets linked to a fan of a simple-flow mechanical ventilation system.

The buried pipes are in polyethylene. Their diameter is 184 mm and their thickness 5 mm. They are placed in parallel and between 1.8 and 2 m deep. The distance between the tubes is around 2.5 m. The air flow per pipe is 240 m^3/h. The length of the tubes is 50 m with a slope of around 2% in order to evacuate the possible condensates. The air inlet is raised at 1 meter over the ground and is equipped with an anti-rain system and a grate fine enough to prevent dead leaves, rodents and insects from entering.

A direct inlet in the blowing network by-passes the buried-pipe system depending on the inside and outside temperatures. The ventilation box is composed of two dampers (Figure 4). The regulation system enables a continuous functioning of the buried-pipe system all year round (Table 2). In winter when the ambient temperature is lower than the ground temperature, the buried-pipe system preheats the outside air. During the in-between seasons when the ambient temperature is higher than the ground temperature, the buried-pipe system is by-passed. The supply air is then directly blown in the premises. This operation mode can also be used in summer to provide direct refreshment (free cooling). When the inside temperature is near 22°C and the ambient temperature is higher than the ground temperature, the buried pipe system is used to cool the outside air efficiently before being blown in the building.

Table 2. Operation modes of the buried-pipe ventilation system

Period	Temperature conditions	Air treatment type
Winter	Tamb ≤ Tground	Outside air preheating using the buried-pipe system
Winter	Tamb > Tground	Direct ventilation
Summer	Tin ≤ 22°C	Direct ventilation
Summer	Tin > 22 °C Tamb ≤ Tground	Direct ventilation
Summer	Tin > 22 °C Tamb > Tground	Refreshment using the buried-pipe system

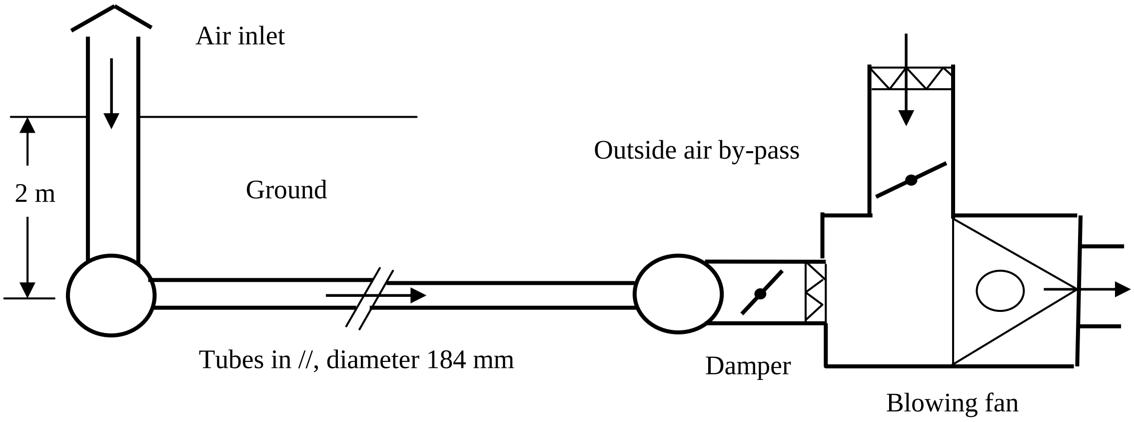

Figure 4. Sketch of the buried-pipe system

Temperatures of supply air

In direct ventilation or free cooling, the supply air temperature is given by equation 2 that takes in account a slight increase in temperature passing through the fan. When the outside air flows in the pipes the air temperature is given by equation 6 that is valid for heating and cooling. The outlet temperature is function of the length of the pipes l and of the overall heat transfer coefficient U between the air and the ground.

$$T_{sa} = T_{ground} + \left| T_{amb} - T_{ground} \right| \times e^{-\frac{Ul}{q_m C_{air}}} + 0{,}4 \quad (6)$$

Where q_m is the air flow rate and C_{air} is the air specific energy. The ground temperature varies in a sinusoidal way [3]. The minimum temperature is in February (10.1°C) and the maximum in September (19.2°C).

BUILDING CHARACTERISTICS AND SIMULATION CONDITIONS

Description of the care home for elderly dependent people

The care home for elderly dependent people has 45 bedrooms and is composed of a building of three levels with the main frontages towards north and south and a second building on one level where are located a reception, two offices, a restaurant, a leisure room and a care room. The building with heavy floors and an interior isolation is considered having average inertia.

The building has 7 bedrooms on the ground floor and 19 on each upper level. Each bedroom has a surface area of 20 m² and is led into by lifts and staircases. The bedrooms are nearly equally distributed between north and south.

The windows are equipped with double glazing argon-filled and with low-emissivity coating (low-e). Overhang shadings and interior Venetian blinds limit direct solar gains. In every zone it is simulated that the Venetian blinds are shut down when the solar radiation is over 500 W/m² and are put up under 400 W/m². The rooms are assumed to be occupied continuously from 21:00 until 7:00 in the morning. The light gain is taken equal to 5 W/m² from 21:00 to 23:00 every evening. During all the heating period, the temperature is maintained at 21°C. Each room includes a bath room, where is placed the ventilation outlet. Ventilation is continuous in the whole building and corresponds to 1350 m^3/h that is 30 m^3 per hour and per room.

The common part of the building (reception, administration and restaurant) communicating with the room part has a surface area of 250 m² and counts three frontages towards north, west and south. This part of the residence is occupied every day of the year from 7:00 to 21:00. The restaurant welcomes 60 persons every day from 8:00 to 9:00, from 12:00 to 14:00 and from 18:30 to 20:30. During the rest of the time the occupants and the staff are supposed to be in the other common rooms. The artificial light gains are 10 W/m² when natural lighting is not sufficient and between 7:00 and 21:00. The two offices are equipped with computers. The common part of the building has a discontinuously occupancy with intermittency of heating and ventilation. Heating and ventilation are cut from 21:00 to 7:00 during the night. During the day, the total ventilation flow is 1080 m^3/h and is produced by a second ventilation set. During the heating season, the daytime set temperature is 21°C.

Electric power of the fans

Table 3 shows the powers of the blowing and extraction fans depending on the type of ventilation. Because of the use of the exchanger and its filters the double-flow ventilation fans have higher powers. Concerning the buried-pipe system, the blowing fan power is the same as for the double-flow system and the extraction fan power is more or less the same as for a simple-flow ventilation system.

Table 3. Electric power of the fans

	Simple flow	**Double flow**	**Buried-pipe**
Bedroom area Qv = 1350 m^3/h			
Blowing fan	-------	600 W	600 W
Extraction fan	350 W	600 W	350 W
Common area Qv = 1080 m^3/h			
Blowing fan	-------	500 W	500 W
Extraction fan	270 W	500 W	270 W

Simulation tool

Thermal zones, wall compositions, glazed surfaces, solar protections, ventilation systems, occupancy and lighting schedules have been defined using the multi-zone model of the transient simulation program TRNSYS [5]. The simulation has been run with an hourly weather data file from Toulouse (south of France). Depending on the ventilation type (simple-flow, double-flow or buried-pipe) different features are compared: the energy consumptions during the heating season, the temperatures in each zone in order to evaluate a number of hours of overheating and eventually the number of hours of functioning per mode (outside air preheating, direct ventilation or free cooling and refreshment).

COMPARISON OF SIMULATED PERFORMANCES

Heating energy consumptions

The simulations run with the climatic data of Toulouse enable to calculate the heating energy consumptions for each system and to evaluate the economy done using outside air preheating compared to a basic simple-flow solution. The energy consumptions are counted separately for the common area and for the bedroom area of the building where the operating is different as far as heating and ventilation are concerned.

Table 4 shows that the two double-flow ventilation systems with heat recovery exchanger are the most performing and enable to save 32626 kWh compared to a simple-flow system. The registered energy economy using a buried-pipe system is 6207 kWh. For outside air preheating, it is to say that the buried-pipe system is of very little interest.

Table 4. Thermal energy consumptions for heating

	Simple flow	**Double flow**	**Buried-pipe**
Bedroom area Energy consumptions in kWh	81949	57919	77311
Common area Energy consumptions in kWh	19965	11369	18396
Total consumptions in kWh	101914	69288	95707

Annual electric energy consumptions of the fans

The values of table 5 show that the double-flow ventilation system with heat-recovery exchanger has the higher electric energy consumption of the fans. Compared to a simple-flow solution, the annual over-consumption of the whole building raises 11180 kWh. In the same conditions, the over-consumption of the buried-pipe system is 7810 kWh.

Table 5. Annual electric energy consumptions

	Simple flow	**Double flow**	**Buried-pipe**
Bedroom area Energy consumptions in kWh	3066	10512	8322
Common area Energy consumptions in kWh	1380	5110	3935
Total consumptions in kWh	4446	15622	12257

Comparison of the operation times per mode

For the bedroom area of the care home and for the double-flow and the buried-pipe ventilation system, table 6 gives the number of hours of functioning per mode: outside air preheating, direct ventilation or free cooling and outside air cooling. This table confirms that in winter, with the chosen operating strategy and the relative temperature levels of the ground and the ambient air, the buried-pipe system is less performing than the double-flow system.

Table 6. Number of hours of operation per mode

	Outside air preheating	**Direct ventilation**	**Cooling**
Double-flow	4585	2702	1473
Buried-pipe	3441	3846	1473

Comparison of the temperature levels during summer

Table 7 shows the number of hours for which the inside temperature is above 27°C for the three studied types of ventilation and for three zones of the building.

Table 7. Number of hours of overheating

	Simple flow	**Double flow**	**Buried-pipe**
Bedroom area			
Zone at the north	314	110	3
Zone at the south	646	316	70
Common area			
Leisure room	607	249	93

A double-flow ventilation system with indirect evaporative cooling enables to divide by 2 or 3 the number of hours of overheating depending on the thermal zone concerned. A buried-pipe is very efficient in summer: the overheating risk in the care home is really limited.

CONCLUSIONS

This simulation study enables to validate that a double-flow ventilation system using a heat recovery exchanger between the exhaust air and the outside air is very performing as far as energy economy is concerned. Nevertheless the fans have to be carefully chosen. These equipments can lead to a subsequent energy over-consumption. A buried-pipe ventilation system has poor performances when preheating outside air. However it seems to be very efficient to cool premises in summer. When lowering the maximum temperatures, the internal summer comfort is improved, which is a critical subject concerning care homes for elderly people.

This work will be implemented by the study of the performances of a buried-pipe system coupled to a double-flow ventilation system. This solution could preheat efficiently the outside air in winter and maintain coolness in summer with no air-conditioning system. It is also seen as a lead to evaluate the performances of this system under other weather conditions and to study in a more precise way the electric energy consumptions of the fans.

The research of French organization CSTB (Scientific and Technical Centre of Buildings) in the area of buried pipes should soon lead to new regulations of dimensioning and implementation to limit sanitary risks around legionella in condensates and radon radiations.

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Passive Downdraught Evaporative Cooling, Humidity Control and Water Resources: defining strategies using Traditional Chimney as Case Study in Portugal

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SUMMARY

Passive Downdraught Evaporative Cooling (PDEC) towers applied indoors produce high rates of humidity (RH) that should be controlled not exceed 70% inside the tower and the room in order to reduce risk for public health and fabric detioration. An experiment using PDEC on an existing fabric in a hot-dry area of Portugal, combined with Computer Fluid Dynamics (CFD) and a Thermal Model (TM) indicated excessive production of RH inside the tower and besides windows, where risk of condensation is higher due to solar radiation from glass. Some recommendations are proposed to reduce the excessive RH production according with fabric constrains in order to prevent from Legionella, condensation, microbiological growth and fabric deterioration, such as treatment of water system and maintenance of the PDEC tower.

INTRODUCTION

PDEC originates from the vernacular architecture of the Middle East [1] and it has been the object of great interest worldwide since 70´s energy crises. Several experiments have been carried out to extend its effectiveness to both outdoors and indoors for uses such as domestic, offices and refurbishments.

The device consists of single or multiple towers equipped with a water/vapor supply placed on the top. During the constant injection of water, droplets descend through the tower and conditions close to saturation along its length. Cool air descends the tower and exits at its base where it is delivered to the adjacent spaces.

The concept is based on the relatively large amount of energy required to convert water from its liquid to gaseous form within a local thermal imbalance with subsequence differences in air density. This leads to the movement of air from a zone of high pressure, where air is hot and less dense (top of the tower) to a zone of lower pressure, where air is colder and denser (bottom of the tower). The physical process occurs when water evaporates within a stream of ambient air without a supply of external heat, resulting in the drop of the Dry Bulb Temperature (DBT), increase of the RH while the Wet Bulb Temperature (WBT) remains approximately constant. The capacity of air to hold vapour is indicated by its DBT and RH. Traditionally, the temperature drops to 2-3 ^{o}C above WBT [2]. This phenomena is the result of the principle of latent heat of vaporization that is the amount of “hidden energy” that is required to convert a unit of mass water from its liquid to its gaseous/vapour phase. The transfer of heat energy from ambient air to water to support the phase change is the process that reduces ambient air temperature. The latent energy required to evaporate more water is supplied by the surrounding air, so the later is cooled.

PDEC is specially useful in hot-dry climates and it has several advantages: it is an adiabatic process because there is no addition or extraction of energy (heat) from the system, therefore, it can be completely passive; it supplies ambient air with moisture raising the RH levels, thus improving the thermal comfort and reducing problems resulting from low RH; it is a quite flexible system that can be easily adapted to new and existing constructions.

Regarding refurbishments, a PDEC shower system was introduced and tested on an existing brick chimney traditionally used for summer ventilation throw “chimney” effect in the southeast of Portugal. This experiment included measurements that indicated RH, T and Air Speed (AS) values inside both PDEC and room. In order to validate and complement data collected from experiment it was created a TM according to building fabric. A CFD model was also created and combined with the TM to evaluate predicted Air Movement (AM), AS, T and RH beside all the room geometry and fabric, including chimney.

METHODS

The chimney object of the case study is located on the first floor of, a small cultural centre from the 19th century located in Moura (38º13N:07º13W). The use of these chimneys was particular successful in obtaining cooling by stack effect.


a)

b)

Figure 1 a) Location of Moura b) photo of the exterior of the chimney

Construction details include internal and external walls 60 cm thick, made from adobe and finished with lime.

The chimney has a cylinder shape measuring 4,8 m in height (measured from the floor of the room) and 0,9 m in diameter. It is located on the first floor adjacent to a room with approximately 19 m² floor area and 63 m³ of volume. The room has tree old wooden doors (two interior and one exterior), and one north-facing window with single glazing and aluminum frame. The chimney walls are in adobe, 0,30 m thick. On the upper limit there are a few openings. There is no lime rendering on it.

An evaporative system project within the chimney was designed according to the experiments made by Givoni and Pearlmutter et al. It included a single shower placed 4,5 m high above floor plan, with coarse water. The water was provided from an existing tap on the exterior that could be easily controllable. The water flow rate was 10,5 l/min.

The measurements took place in the 6th of August, every hour, between 11:30 and 15:30. The room’s exterior and interior atmospheric conditions are registered on row 1 and 2 of Table 1. The drop of temperature and relative humidity registered at 12:30 is due to a breeze.

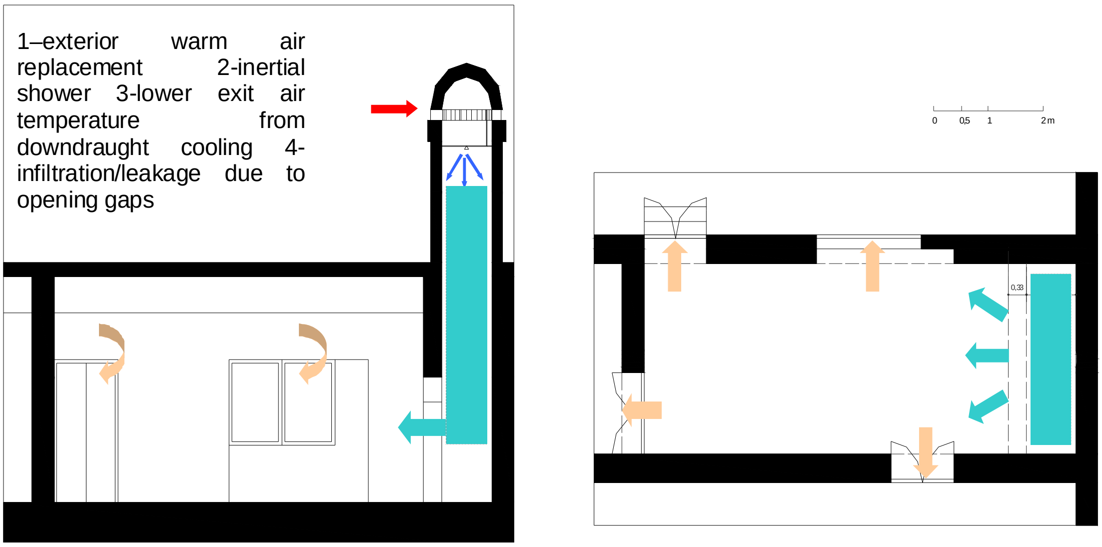


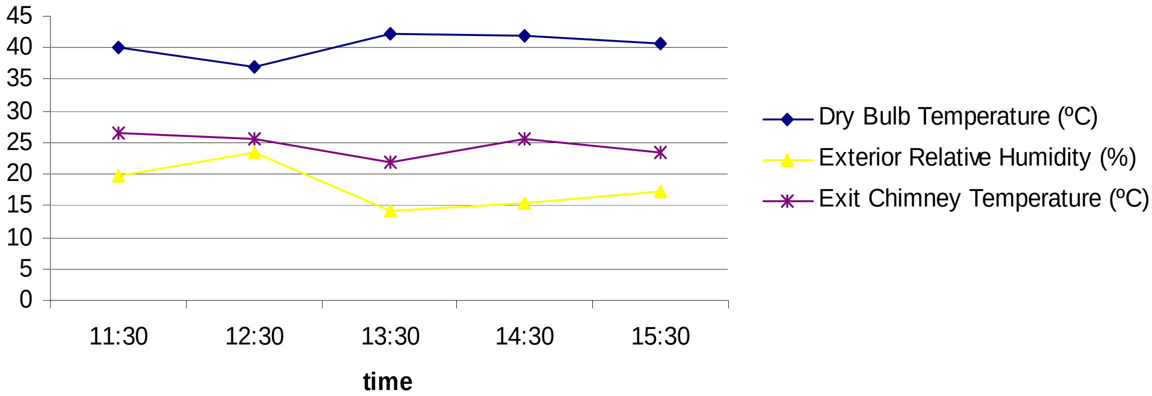
Figure 2: Section and Plan of the chimney and adjacent room (not on scale)

Water did not circulate continuously in order to avoid flood problems and the shower operated for only a few minutes. Also, PDEC configuration ended up being less detailed than initially thought. Testing conditions were not ideal, especially due to the use of course instead of fine spray.

RESULTS

Results show that after the shower system was activated, both exit air temperature and RH were not stable on the measured periods. Temperature ranged from 22 ^{O}C to 26.6.^{O}C and RH from 62 % to 95.5 %. Graphic 1 shows that there is a relation between temperature and RH: a higher humidity corresponds to lower exit air temperature. According to Graphic 2, room conditions after PDEC are very close to saturation, i.e., air cannot hold more moisture. Also, line of enthalpy is not absolutely constant, therefore WBT changes slightly.

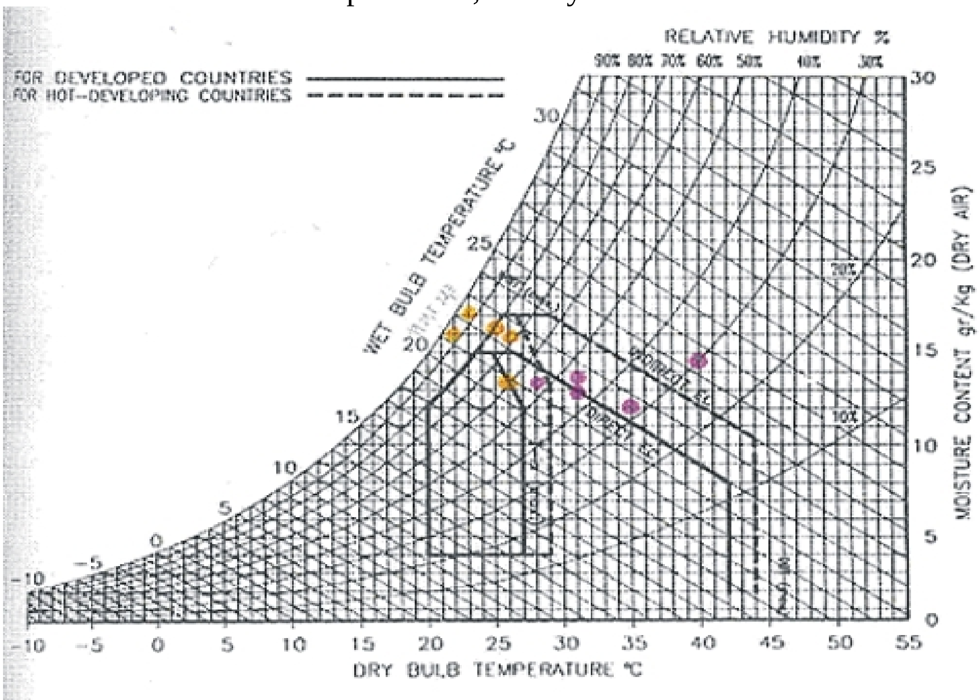
Exterior Temperature, Relative Humidity and Interior Temperature



Graphic1: Measurements

When Moura´s chimney PDEC was activated, it produced very high values of relative humidity (more than 90 %), especially in the first minutes of operation. Such high humidity can lead to surface condensation, creating the risk of mold growth or other moisture problems.

Therefore, temperature depression of the chimney should be controlled so that relative humidity should not exceed 70 % in all zones [3]. It is important to note that although PDEC was activated every hour thus not providing continuous cooling, results are quite similar to those observed on other experiments, namely Givoni´s.



Graphic2: Givoni´s psychometric chart.

Also, water particles did not fully evaporate before reaching the ground. People placed near the chimney could easily get wet from not evaporated water droplets. Evaporation depends upon droplet diameter and length of time water stays within the air stream. Therefore water droplet should be small enough or the distance from the sprays to the openings great enough to guarantee total evaporation. Since height of the shower could not be maximized because cooling source was placed on the top of it, a fine spray mist of water (like filtered micronisers) should have been used instead.

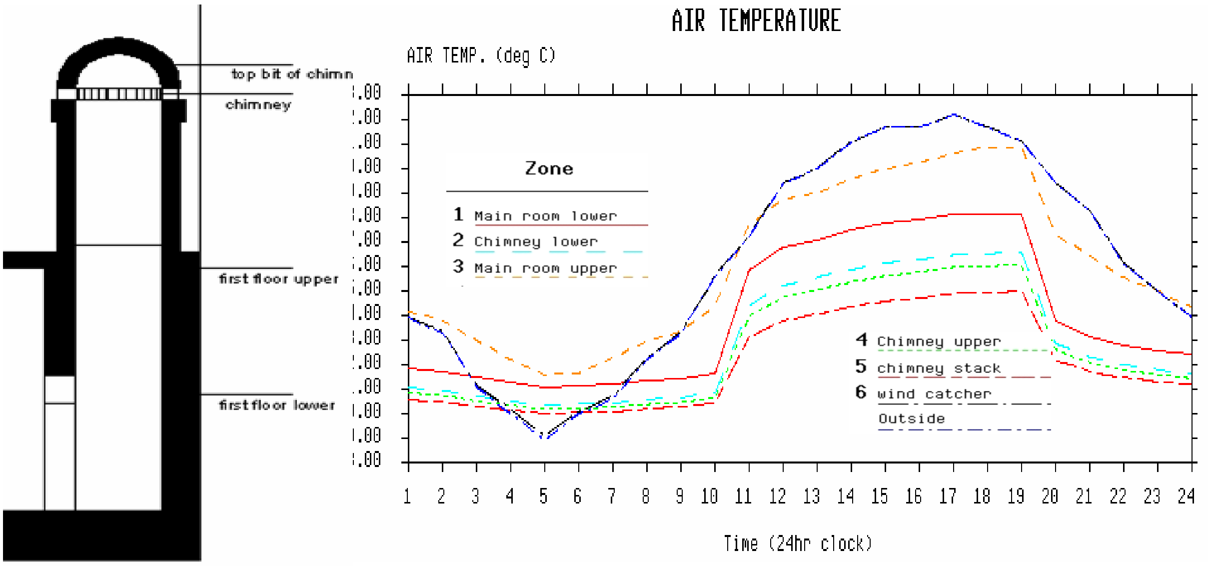
Finally, exit airflow of 0.7 m/s rate was far above than the 0.25 m/s recommended avoiding unpleasant draughts [4].

Since field-testing had several limitations to produce valid data, it seemed important to create a thermal and a dynamical model of the PDEC tower and adjacent rooms to investigate cooling performance of the system. There are few programs that allow a PDEC simulation because there is the need to recognize top-down airflows and driven forces. PDEC are often described as “reverse chimneys” because the column of cool air falls.

However, internal conditions can be reproduced according to an occupancy pattern (number of users of the space/building, heat gains from people, machinery, etc) and physical properties of the construction (envelope, heat losses from envelope, etc) in a thermal model of the PDEC chimney and adjacent room. This model can be combined with a CFD program that analyses how airflow rates from PDEC supply coolness into the adjacent room. CFD does not simulate cooling systems where driven forces and cooled air occur in the same pressure zone, therefore heat transport does not correspond to air movement inside the cooling chimney. The downdraught evaporative cooling should be on the bottom of the tower. A CFD can be very efficient to ensure that exit AS, T and RH are precisely distributed within a space. This is especially important to identify eventual stagnant zones where air does not flow.

THERMAL AND COMPUTER FLUID DYNAMICS (CFD) MODEL

TAS 8.4 was used to create a TM model.



Graphic 3: TAS output data.

AMBIENS is a CFD two dimensional Cartesian grid system, in which a transversal “slice” 1m tick including exit air from chimney, was modeled. The grid represents the computational cells that will be the basis of the simulation. Building elements were not placed reproducing reality thus maximizing output information to better understand indoors thermal comfort. Therefore window and outlets (corresponding to doors gaps) are placed in the same plan, although this does not happen in any 1m slice.

Features represented are:

a) one window b) one cooling source (air inlet) c)two door gaps (air inlet and air outlet)

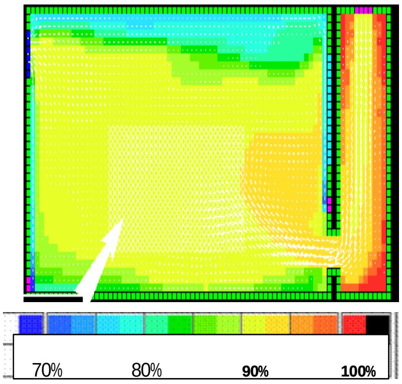
Note that inlet air volume should equal outlet air volume.

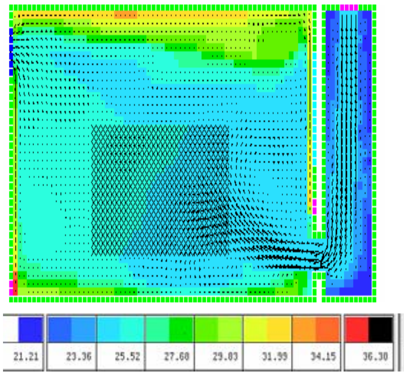
Definitions of modeling included:

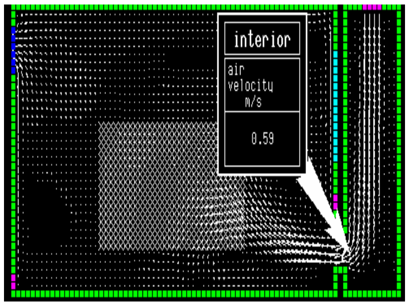
a) each pixel is equivalent to 0.1m height or width and 0.1 m² area b) 4 pixels for each opening, equivalent to 0.41 m² assuming 2.5 m² in reality

Input data for modeling included:

a) 63 m³ of room volume b) 2 ACH of infiltration, equivalent to 0.035 m³/s of Airflow Rate in the whole room, therefore 5.83 x 10 -3 m³/s in the “slice” c) air speed of 1 pixel equivalent to 0.0075m/s at 0.6m/s or 8 pixels at 0.3m/s i.e., “slice” volume divided by area of each pixel d) 0.6m/s definition of exit air speed, either 4 pixels







Interior: air velocity 0.59 m/s

Figure 3 : Relative Humidity

Figure 4 : Temperature

Figure 5 : Air Speed

Temperature surface (obtained from TAS thermal model) included: roof - 34^{o}; floor - 31^{o}C; walls - 32^{o}C ; window - 40^{o}C.

DISCUSSION

According to AMBIENS model, there is an excessive production of humidity inside the chimney (between 90% and 100%). Junctions of chimney walls and ground are especially affected and water particles that did not fully evaporate before reaching the ground can warmer situation on the bottom PDEC tower. Air beside a window has a higher RH than the rest of the room, excluding the chimney since its surface has a higher temperature. (Figure 3). Here risk of condensation is higher. However, air is distributed in a very satisfactory way within the room, and door gaps are important to help extract "old" air.

Exit air openings can play an important role defining water flow rate. Results suggest that a system with a flexible size opening could be used to deliver the exact amount of cooling required, although this is not practical. Results also show that even though thermal comfort benefits from reducing on exit air speed, increase of temperature penalizes it. By reducing the water flow rate by 2.5l/min, exit air temperature will increase 1^{o}C.

Regarding ventilation supply, the system is efficient since 25 people require approximately 13 ACH (assuming 8l per person per second), and the system can supply 38 ACH.

Simulation data suggests that humidity produced inside traditional chimney PDEC can represent a problem to public health and building maintenance:

- Risk of diseases (such as pneumonia) associated with production of Legionella;
- Condensation produced inside the tower will create corrosion and fabric deterioration;
- Risk of sedimentation/deposition of solids (organic and inorganic) in tower surfaces.

Therefore some recommendations can be made according to limitations of the building fabric, namely architecture form:

- Placement of an exhaust (like a fan) on the opposite side of the of the PDEC tower to extract "old" air thus renewing the air and creating a replacement cycle;
- Placement of an microbial bio film on all tower surfaces in contact with water, since these films can consume certain inhibitors and prevent access of inhibitors [5];
- Water particles that did not fully evaporate before reaching the bottom of the tower should not be induced again to feed the PDEC system before being previously filtered and heated to a temperature above 55ºC to inhibited microbiological growth . Then, water temperature should be lowered and then induced in the system again;

- All pipe system and water reservoirs (such as to collect particles that did not fully evaporate before reaching the ground) should be in copper since it is inhibitory to support the growth of Legionella and other bacteria [6];
- Water should be previously treated before feed PDEC system. It is suggested disinfectants such as chlorine and bromide or ultra-violet light. Sea water could also be used since surviving of Legionella is limited by both salinity and warmer temperature [6];
- Pipe installation should avoid any contact between main and PDEC water supply system to avoid contact between different types of water and filtration;
- Maintenance and cleaning of the PDEC system should be at least twice a year;

However PDEC is a specific technique can help reducing the impact of global warming thus providing thermal comfort in building and occupants.

In *2004 the* annual temperature of Portugal increased 0.6 ºC[7] and *recent studies predict* that the *dry condition will affect 1/3 of the Portuguese territory* [8]. The spectrum of buildings and urban areas that can use these techniques is wide enough: it can be applied both in indoors and outdoors, new buildings and refurbishments, all scales and type of buildings with architectural expression according to the occupants and buildings aesthetical form. The benefits affect several areas:

- Indoors temperature can reach 24 – 30 Cº, and HR 40-70 %, when outdoors conditions can reach 40-45 Cº and 15-20 % RH whenever EC systems are applied. It can improve occupant's thermal comfort, allowing the use of natural elements such as ventilation;
- PDEC techniques can help reaching Kyoto Protocol objectives that in Portugal should be reduction in 27% the emission of pollutants in the period 2008-2012 [9]. In Portugal, in the period 1990-2003 the residential and commercial sectors registered an increase of 100% of emission of polluents into atmosphere mainly due AVAC systems[10]. Measures promoted by European Community and Portuguese governments obliged the use of renewable energies in all sectors, including buildings [11].
- In PDEC, all white waters can be recycled and re-used. Typically a shower consumes 20-40 litters and a bath 50-100 liters [12]. To provide a cooling power of 3 Kw, a PDEC system can consume 40 litters of water a day that represents a small percentage of residual domestic water [13]. The Department of Water Resources recommends the recycling of residual waters in semi-arid or arid in the world, especially from domestic sources. In Israel, it represents 70% [14] of total residual water. In Portugal, residential is the second sector that consumes more water (8%) and the majority of it goes to flush toilets, showers and baths, with an efficiency of only 58%[14].
- Compared to conventional air-conditioning systems, evaporative cooling techniques can represent a significant reduction of energy consumption and money savings. The European Programme Energy Manage promoted the application of an evaporative cooling system in an office building in Malta. It was proved that it reduced energy consumption in 48% and CO_2 emissions in 28 152 kg compared with a building that used electrical cooling. It represented money savings of £ 3·080/year [15].

PDEC should always be considered as a part of a cooling strategy that may include high thermal mass or proper shading. These techniques may be high or low tech, according to the cooling power required in the building/outdoor area or the capital costs involved in the construction/maintenance of the system. Quality of the water piping system and treatment of deposited water can be considered the higher financial investment in PDEC solutions.

ACKNOWLEDGEMENT

The authors are grateful to Dr. Alan Young from the Bartlett School of Graduate Studies from University College London.

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Study on the heat transfer model and the application of the underground pipe system

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SUMMARY

The underground pipe system (UPS) cools the intake outdoor air through the underground pipe before supplying it to the indoor space, so that the cooling load of the air-conditioning system of the building can be obviously reduced. In this paper, a quasi-three-dimensional heat transfer model was developed, which was made up with a two-dimensional unsteady heat transfer model of earth and a one-dimensional fluid model of air. The total heat transfer process was considered when the condensation happened on the pipe wall during the calculation. Moreover the model was validated by the comparison between simulated result and measured data of a built project applied UPS. By means of this model, the earth-to-air heat exchanging process was simulated and the effect of the pipe parameters, such as the length， depth of burial and ventilation velocity of underground pipe system, on the cooling/heating capability was analyzed. The simulated result was also used to direct the design, operation and management of the UPS.

INTRODUCTION

Underground Pipe System (UPS) utilizes the energy accumulating in the soil to cool or heat outdoor air, and then supplies it into rooms. The system can ameliorate indoor environment and save energy. Moreover its air-conditioning efficiency is limited by outdoor climate, structure of underground pipe, operating schema, etc. Therefore the mathematic model which can be used to simulate the earth-air heat exchange process accurately is a very important tool for engineering design.

In most studies of the earth-air heat transfer models, the models can be separated into two kinds: one is steady models; the other is unsteady model. A steady model based on the assumption of constant heat flux and half-infinite object heat transfer was simple and fit for engineering design [1]. Another steady model utilized the empirical equation which was regressed by the measurement data [2]. The unsteady model includes regression model, finite difference model and finite element model. Some researchers introduced the finite difference model to solve the 3-dimension unsteady sensible heat transfer process, and analyzed the influencing factors of operating effect [3][4]. Furthermore, a 3-dimension finite-element model, which can judge whether moisture will condense on the wall of UPS, was developed to depose the simulation of total heat transfer process while the air went through the underground pipe [5].

In this field great progress has been made, however the present models still don’t satisfy the requirements. The steady models mentioned above can’t predict the temperature variation during the operation of UPS. The 3-demintional models required a lot of computations, so it was hard to accomplish annual simulation of UPS due to the complex computations. There is still dearth of accuracy, general and swift computation model to direct the design of UPS projects.

In this paper, a quasi-three-dimensional heat transfer model was developed, which was made up with a two-dimensional unsteady heat transfer model of earth and a one-dimensional fluid model of air. By means of this model, the effect of the pipe parameters, such as the length, depth of burial and ventilation velocity of underground pipe system, on the cooling/heating capability was analyzed. The simulated result can also be used to help the designers to determine the structure and the operation scheme of the UPS.

METHODS

2.1 Numerical simulation model

The heat & mass exchanging process can be predicted by a quasi-three-dimensional model, which assumes that heat transfer of soil along the direction of pipe can be ignored. For the z basic calculate surface in Figure 1, energy balance equation of soil and air are respectively presented as:

$$\frac{\partial t_{soil}}{\partial \tau} = a\left(\frac{\partial^2 t_{soil}}{\partial x^2} + \frac{\partial^2 t_{soil}}{\partial y^2}\right) \qquad (1)$$

and

$$V \rho_{air} \frac{\partial i_{air,z}}{\partial \tau} = G \cdot (i_{air,z-1} - i_{air,z}) + dQ \qquad (2)$$

where t_{soil} is temperature of soil(˚C), τ is time(s), a is thermal diffusivity(m^2/s), V is volume of air for a basic grid(m^3), $i_{air,z}$ is air enthalpy of computational unit in Z coordinate axis(kJ/kg), ρ_{air} is density of air(kg/m^3), G is fresh air volume(m^3/s), and dQ is heat transfer quantity between air and earth(W).

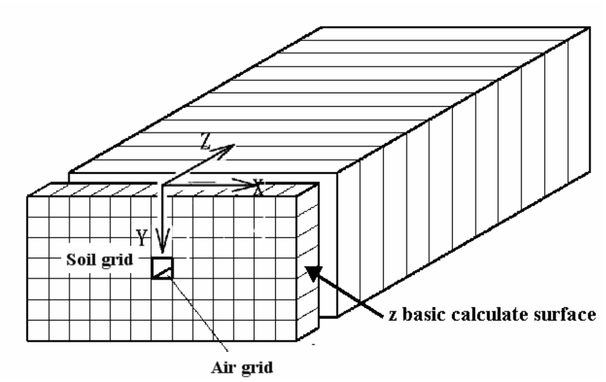


Figure 1. Grid division for numerical simulation

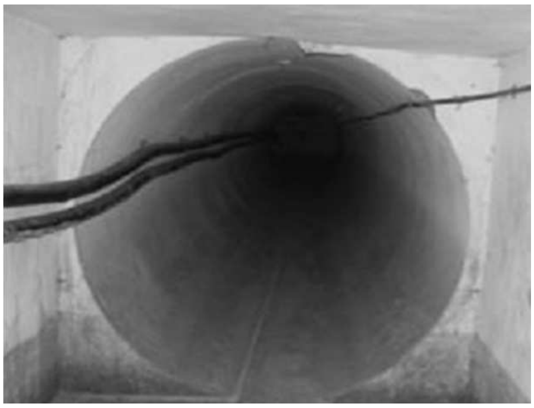


Figure 2. Cross-section of the UPS

Because latent heat exchange may exist between air and earth, dQ can be calculated by Eq.(3) for only sensible heat exchange or by Eq.(4) for total heat exchange.

$$\begin{aligned} dQ &= \sum_{j=1}^{J} hF_{Z,j}(t_{b,Z,j} - t_{k,Z}) \\ &= \sum_{j=1}^{J} hF_{Z,j}(t_{b,Z,j} - i_{k,Z} + 2500 \cdot d_{k,Z}) \end{aligned} \qquad (3)$$

$$dQ = \sum_{j=1}^{J} h' F_{Z,j} (i_{b,Z,j} - i_{k,Z}) \qquad (4)$$

where j is No.j earth-to-air heat transfer surface, h is convective heat transfer coefficient (w/(m^2.˚C)), h' is convective mass transfer coefficient (kg/(m^2.s)), $F_{Z,j}$ is area of earth-to-air heat transfer surface of computational unit Z (m^2), $t_{b,z,j}$ is surface temperature of the pipe of No,j surface, $t_{k,Z}$ is air temperature in the pipe of computational unit Z, $i_{k,Z}$ is saturated enthalpy corresponding to temperature of pipe wall(kJ/kg), $d_{k,Z}$ is absolute humidity of air (kg/kg). Finite difference method is applied to solve the problem, and temperature and humidity of air extracted from the UPS can be achieved.

2.2 Experimental Measurement of a UPS

To validate the availability of numerical calculation model, the airflow velocity, inlet and outlet air temperature and humidity of a UPS was measured. The experimental result would be compared with the simulation result.

The Building applied UPS system is in Beijing. The shape of cross-section of the UPS is shown in Figure.2. Table.1 reveals the main parameters of the system. The experiment time was from Jul.15th, 2002 to Aug.18th, 2002. The system operated from 7:00am to 19:00pm everyday.

Table 1. Parameters of an UPS in a college in Beijing

Length	Diameter	Material of pipe wall	Burial depth	Mean annual temperature of soil surface
250m	1.4m	Steel concrete	3m	13.1 ˚C

2.3 Effect of factors of UPS

Based the heat transfer model, the effect of different parameters, such as length, burial depth, airflow velocity and operating mode of UPS, on air-conditioning performance was investigated. The outdoor air temperature and humidify data from Jul.7th, 2002 to Jul.9th, 2002 were selected as computational conditions (Figure.3). The cross-section size of pipe is 1m×1m. In the system, only one pipe was buried under the ground.

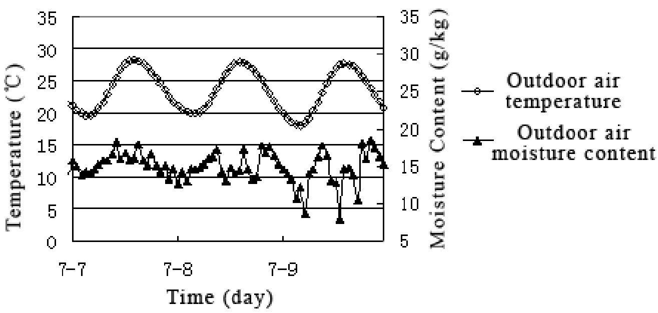


Figure 3. Weather parameters from July 7th to 9th in Beijing

RESULTS

3.1 Measurement result of UPS

The airflow velocity and temperature in the outlet of the UPS was measured. When the system was running, the wind speed maintained at about 3 m/s (Figure.4). Figure.5 shows the comparison between the measured outlet temperature and the simulated temperature; the outdoor temperature is also presented in the figure. From Jul. 17th to Jul. 29th, the measured data accorded with the simulated data on the whole. The average temperature which was measured in the outlet was 20.3˚C; and the computed temperature was 20.6˚C, which was 0.3˚C higher than the measured data.

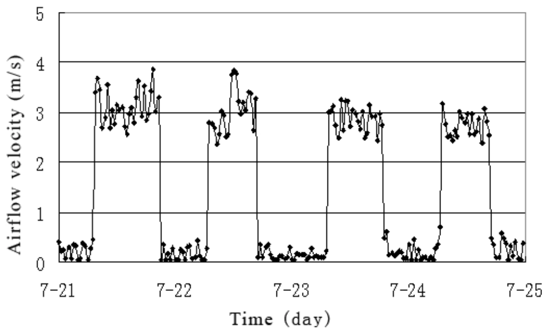


Figure 4. Measured air velocity

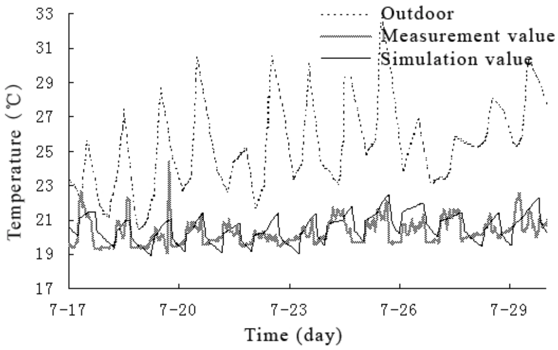
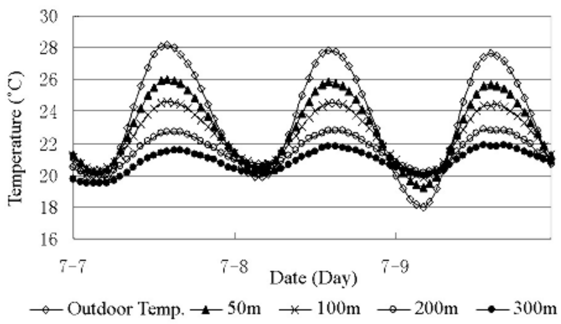


Figure 5. Measured and simulated air temperature

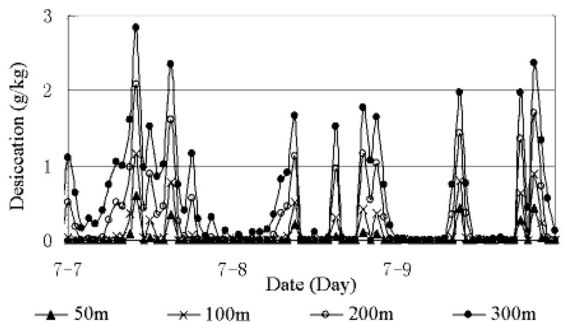
3.2 Influence of UPS parameters on the operating effects

3.2.1 Length of UPS

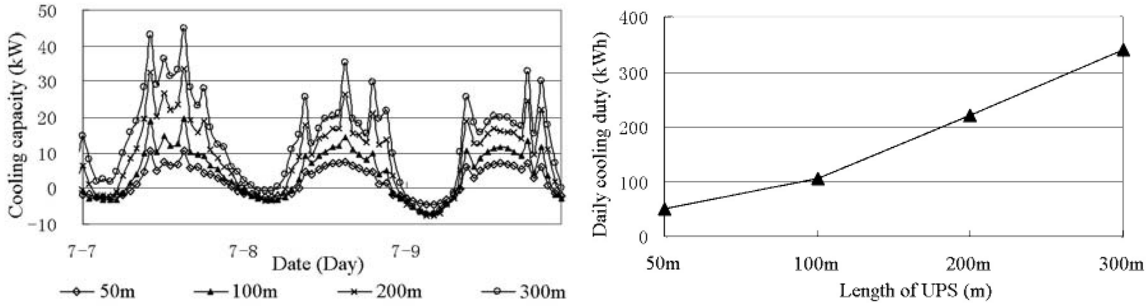
As the calculated examples, the depth of burial of pipe was 3.5m; the air velocity was 3m/s, the length was 50m, 100m, 200m and 300m respectively. The outlet air temperature, desiccation and cooling capacity are shown in Figure.6. When the length increased, the outlet temperature decreased; and the desiccant and cooling capacity rose. The amplitude of temperature fell after the outdoor air went through the system. The outdoor air can be cooled down below 26˚C by a 50m long pipe, and below 22˚C by a 300m long pipe (Figure.6a). For example, the maximal desiccant capacity of a 300m long pipe was 2~3g/kg (Figure.6b); its maximal cooling capacity reached 30~40kw (Figure.6c); and its average daily cooling duty was 340kWh (Figure.6d).



a. Outlet air temperature of UPS



b. Desiccant capacity of UPS

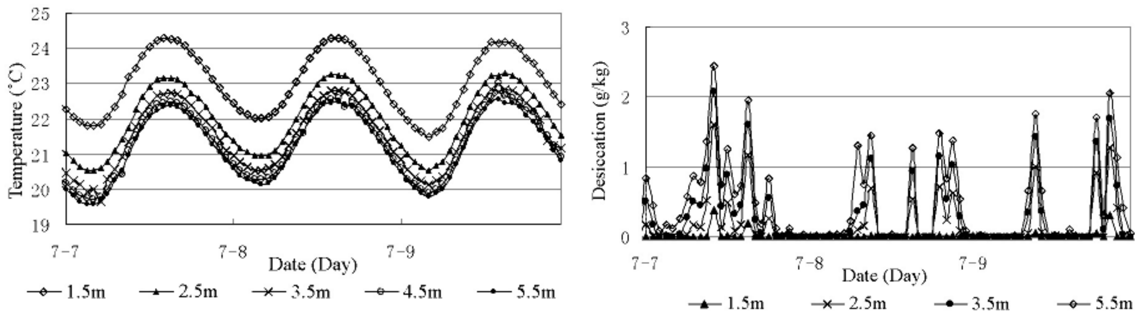


c. Cooling capacity of UPS d. Average daily cooling duty of UPS

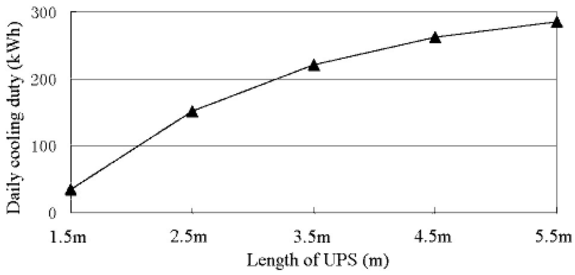
Figure 6. Influence of different length of UPS

3.2.2 Burial depth of UPS

In the simulation, the length of pipe was 200m; airflow velocity in the pipe was 3.5m/s; burial depth was 2.5m, 3.5m, 4.5m, 5.5m respectively. Figure.7 shows the variation of outlet air temperature, desiccant capacity and average daily cooling duty of the UPS.



a. Outlet air temperature of UPS b. Desiccant capacity of UPS



c. Average daily cooling duty of UPS

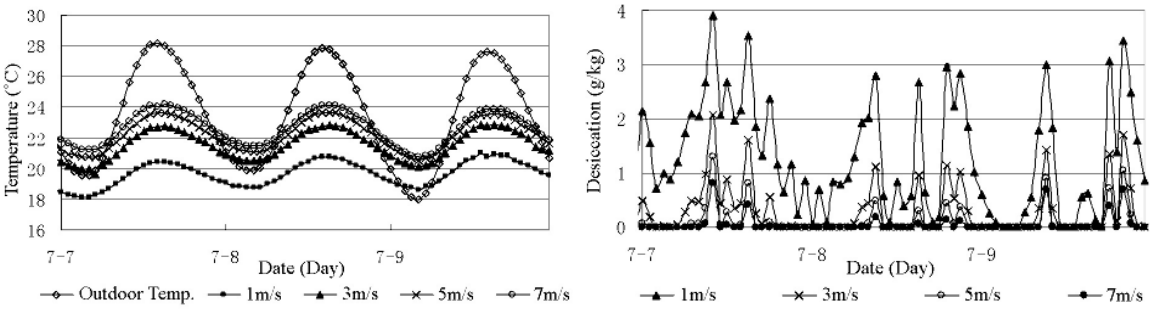
Figure 7. Influence of different depth of UPS

3.2.3 Airflow velocity of UPS

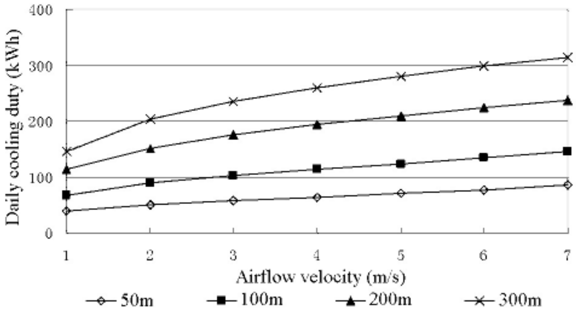
The simulated UPS was 200m long, 3.5m deep from the ground. The airflow velocity in the pipe was 1m/s, 3m/s, 5m/s and 7m/s respectively. The lower the velocity was, the lower the outlet air temperature was (Figure.8-a), the higher the desiccant capacity was (Figure.8-b), and the smaller the average daily cooling duty was (Figure.8-c). When the wind speed was 1m/s, the outlet air temperature was in the range of 18~21˚C. And the maximal desiccant capacity can reach 4g/kg.

3.2.4 Operating mode of UPS

In this case, the length of pipe was 200m; the burial depth was 3.5m; the airflow velocity was 3m/s. The system was operated in the two different modes: One was running continuously, the other was only operated at 8:00~18:00 daily. The outlet air temperature and cooling capacity are shown in Figure.9.

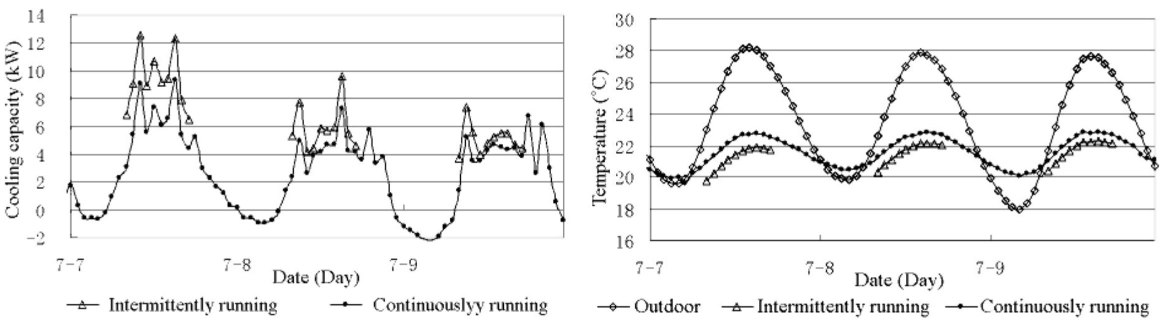


a. Outlet air temperature of UPS b. Desiccant capacity of UPS



c. Average daily cooling duty of UPS

Figure 8. Influence of different air velocity



a.Outlet air temperature of UPS b. Cooling capacity of UPS

Figure 9. Influence of different working mode

DISCUSSION

In this study, a heat transfer model was developed to predict the operating effects of UPS. The influence of UPS parameters on outlet air temperature, desiccant and cooling capacities and average daily cooling duty was investigated.

4.1 Length of UPS

The simulated data indicated that the underground pipe system (UPS) can dehumidify outdoor air in summer, if the system is long enough. When moisture is removed from air, the heat transfer capability between air and pipe wall is enhanced. In this paper, while the length of UPS was greater than 100m, the air intake can be cooled below its dew-point temperature; and then heat transfer process was stronger due to desiccant capacity of the pipe. The average daily cooling duty increased by 12kWh as the length of pipe was lengthened by 10m. Beside these, while the length was less than 100m, the system had less dehumidification capability and less cooling effect than longer system; an increment of 10kWh in the average daily cooling duty associated with a 10m increase in length.

4.2 Burial depth of UPS

As a result of our simulation, when the burial depth was 1.5m, the UPS had little dehumidification capability; the moisture content of air change little while it went through the pipe. When the burial depth increased to 2.5m, the desiccant and cooling capacity rose significantly. However the cooling capability increased slowly as the burial depth was deeper than 3.5m. As the deeper the burial depth was, the more expensive the project cost was; it is likely that the optimum burial depth of UPS was 3~4m.

4.3 Airflow velocity of UPS

The average daily cooling duty increased as the airflow velocity increased, because the coefficient of heat transfer of pipe was positively proportional to the velocity 0.8-power. The increment of velocity led to the growth of the cooling capacity. Moreover while the velocity increased, the outlet air temperature rose; and the desiccant capacity decreased. It is because heat exchange increased slower than the flow rate of air, as the velocity increased the unit mass heat transfer of the air decreased; the air temperature in the pipe increased. So the latent heat exchange part is shortened, and the unit mass desiccant capacity decreased. Accordingly, if the airflow velocity is low in UPS, the system has lower cooling capacity and application effect. In contrast if the velocity is too high, the outlet air temperature and humidity is excessively high; thus the supplied air has little air-conditioning effect. For the UPS with specific length, there should be an optimal range of the airflow velocity in which the system operate effectively and economically. This optimal velocity would rise when the length of the pipe increase.

4.4 Operating mode of UPS

When the UPS ran in the intermittent mode, the outlet air temperature was 1˚C lower than that in the continuous mode. Because the quantity of heat absorbed by the pipe wall in the daytime can be transfer to the deep soil when the system stopped operating at night, the temperature of the wall was lower. Thus the cooling capacity of system rose by 40 percent in the intermittent mode relative to that in the continuous mode. Therefore the rationalized design can improve the cooling capacity of the system effectively.

This paper only used the pipes with some specific size as the calculated examples to obtain the results. If the pipe parameters are changed, the quantitative results should be varied. But the qualitative conclusions presented in the paper are still available.

CONCLUSIONS

Based on the experimental investigations and numerical simulations presented in this paper, the following conclusions may be made:

a) The quasi-three-dimensional heat transfer model was developed to predict the heat and mass transfer process between air and soil. It was verified by experimental measurement and can be used to direct the design, operation and management of the UPS.

b) The cooling capacity and the length of UPS were positively related. The increment of cooling capacity of longer pipe was greater than that of short pipe, when the length increase by the same distance.

c) The deeper the burial depth was, the higher the cooling capacity was. However the cooling capability increased slowly as the burial depth was deeper than 3.5m. It appeared that the optimum burial depth of UPS was 3~4m.

d) While the airflow velocity rose, the quality of heat exchange increased; the outlet air temperature and humidity went up also. A UPS with specific length should have an optimal velocity at which the cooling effect and economical efficiency can be performed. The optimal velocity was positively related with the length of system.

e) In the intermittent operating mode, the cooling capacity was increased by 40% than that in the continuous operating mode.

In the development of the paper we considered that the comprehensive analysis of the factors investigated hereinbefore should be presented to discover the interrelationship among these factors. To study the economical efficiency of UPS is also among our future research goals.

ACKNOWLEDGEMENT

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