CALCULATION OF THERMAL COMFORT FROM CFD-SIMULATION RESULTS

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ABSTRACT
Thermal comfort in ventilated spaces depends mainly on air temperature, air speed and turbulence intensity. The indices for assessing thermal comfort, such as PMV and Draught rating, are based on measurements of the time-averaged mean air speed with omnidirectional hot sphere sensors. At present, thermal comfort is often estimated computationally by CFD-simulations. The mean air speed is not directly obtained from CFD-simulation and therefore the velocity vector magnitude is usually used instead. Since it is smaller than the mean air speed, this method may lead to underestimation of thermal discomfort. A correction method has been earlier proposed for calculating thermal comfort from CFD-simulation results. In this study, the correction method was implemented in CFD-software. Air speed and draught rating values were calculated for two simulated case examples and the effect of the turbulence correction was studied. The correction was found to be significant in room areas with high turbulence intensities.

INDEX TERMS
Thermal conditions, Draught risk, CFD-simulation

INTRODUCTION
Air speed and thermal comfort
Thermal comfort in ventilated spaces depends mainly on air temperature, air speed and turbulence intensity. Current standards and guidelines for thermal comfort are based on measurements of air speed with hot sphere sensors. These sensors are omnidirectional, which means that they measure air speed, the magnitude of air velocity vector, regardless of the direction of the airflow. The time-averaged value of air speed is a relevant parameter for thermal comfort assessment because it is related to the cooling effect of the airflow on the skin.

Mean air speed is used for the determination of the *PMV* index, which concerns the whole body thermal comfort in ISO 7730 standard (ISO, 1994). In a turbulent flow, the fluctuation of air speed has also an effect on thermal comfort. The effect of fluctuations has so far been incorporated into the prediction of draught, which is defined as unwanted local body cooling because of air motion. Fanger has suggested Equation 1 for calculating the draught rating *DR*, i.e. percentage of people dissatisfied due to draught (Fanger et al., 1988).

$$DR = (34 - T_a)(V_o - 0.05)^{0.62}(37 \cdot I_o V_o + 3.14) \quad (1)$$

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where T_a is the air temperature, V_o is the mean air speed (≥ 0.05 m/s) and I_o is the turbulence intensity. I_o is defined as S_o/V_o, where S_o is the standard deviation of air speed ($\leq V_o$).

CFD-simulation and thermal comfort

Thermal comfort is often estimated based on the results of CFD-simulation (Computational Fluid Dynamics) of room airflows. These results, however, do not include mean air speed V_o or turbulence intensity I_o. Instead, they typically include the magnitude of the mean velocity vector V_V and the turbulence kinetic energy *k*. The turbulence intensity I_V, which is defined as S_V/V_V, can be obtained from Equation 2.

$$I_V = \frac{\sqrt{\frac{2}{3}k}}{V_V} \tag{2}$$

In CFD-simulations, *PMV* and *DR* have usually been calculated from the vector values V_V and I_V instead of the omnidirectional values V_o and I_o, because these cannot be directly obtained from V_V and I_V. In turbulent room airflows, the difference between the vector values and the omnidirectional values can, however, be significant. Recently, Koskela et al. presented a correction formula for calculating estimates for V_o and I_o from V_V and I_V (Koskela, 2001). They analysed the effect of the correction to laboratory measurement results and found notable differences in air speed (5 cm/s) and DR-values (7 %-units).

The purpose of this paper is to present practical correction procedures for calculating mean air speed V_o and draught rating *DR* from CFD-results and to study the effect of the correction in two CFD-simulation cases: an office room and a scale model of an industrial hall.

METHODS

Correction formulas

By assuming isotropic turbulence and normal distribution of turbulent velocity components, the mean air speed V_o and turbulence intensity I_o can be estimated by Equations 3 and 4 (Koskela, 2001).

$$\frac{V_o}{V_V} = 1 + I_V{}^2,\ I_V \leq 0.45; \qquad \frac{V_o}{V_V} = \frac{1.596 \cdot I_V{}^2 + 0.266 \cdot I_V + 0.308}{0.173 + I_V},\ I_V > 0.45 \tag{3}$$

$$I_o = \sqrt{(1 + 3I_V{}^2)\frac{V_V{}^2}{V_o{}^2} - 1} \tag{4}$$

The implementation of these corrections in CFD-software is rather straightforward. The post-processor macro language procedures for the correction formulas are given in Figure 1 for the CFX software package that was used in the first simulation case of this study. The second software package in this study was Fluent, where the corrections were implemented using C programming language in so-called user defined functions.

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Vv  =   max(Velocity,0.0001[m s^-1])
Iv  =   sqrt(2/3*ke)/Vv
Vo  =   step(0.45-Iv)*Vv*(1+Iv^2)+step(Iv-0.45)*Vv*(1.596*Iv^2+0.266*Iv+0.308)/(0.173+Iv)
Io  =   sqrt(max((1+3*Iv^2)*Vv^2/Vo^2-1,0))

DR =    max((34+273.15-T/1[K])*max(Vo/1[m s^-1]-0.05,0)^0.62*(37*min(Io*Vo,Vo)/1[m s^-1]+3.14),100)
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Figure 1. CFX 5.5 CEL macro language implementation of the correction formulas.

CFD-simulation case 1: Office room

The first case was adopted from Chen et al. (Chen, 2001). They studied a laboratory model of an office room with two tables, two computers, two persons, two cupboards and four lamps (Figure 2). The study included several air distribution methods from which the slot diffuser was selected for this study. The size of the office room was 5.16 m x 3.65 m x 2.43 m. The dimensions of the obstacles and the boundary conditions of the heat sources and the supply device are given in the reference (Chen, 2001).

Chen et al. carried out measurements of air speed, turbulence intensity and temperature profiles at five locations 0.85 m from the back wall. The calculated flow pattern in this measurement plane is shown in Figure 2.

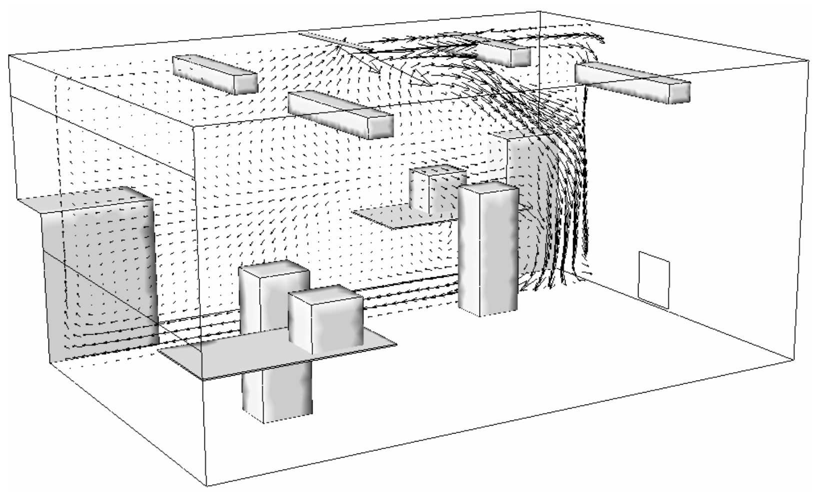

Figure 2. Simulated office room with the calculated flow pattern at the measurement plane.

The CFD-simulation was carried out with CFX 5.5 software by using k-ε turbulence model and second order discretisation. The computational grid was unstructured and included ca. 110 000 nodes. The slot diffuser was modelled as a 0.12 m x 1.2 m surface on the ceiling with a constant supply velocity of 1.44 m/s in the angle of 45° downwards. A momentum source with a height of 0.1 m and strength of 0.346 N was added below the inlet surface in order to meet the measured momentum flow of the supply device. The air change rate was 9.2 times per hour and the temperature difference between inlet and outlet air was 5.1 °C.

CFD-simulation case 2: Scale model of industrial hall

The second case was taken from Ala-Juusela et al. (Ala-Juusela, 2001). They studied ventilation in a 3:10 scale model of an industrial hall. The length, width and height of the small-scale room were 7.2 m, 3.6 m and 2.4 m, respectively. Two air distribution methods were modelled. The first one used nozzles in the upper part of the room as shown in Figure 3 (from now on referred as nozzle case). The other method supplied air from grilles at a lower level on the front wall (grille case). The air change rates for the two cases were 4.7 and 5.9 air changes per hour. The supply air temperature was 4.5 °C and 11 °C below the room temperature in the two cases.

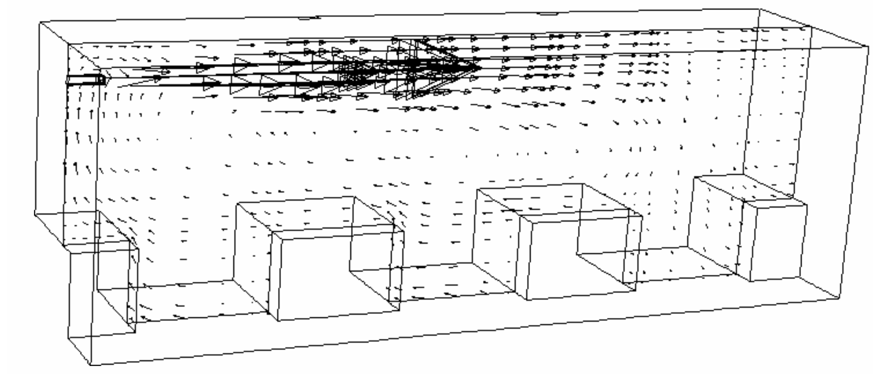

Figure 3. One half of the simulated industrial hall with the nozzle air distribution. The interaction between the supply air jet and the warm obstacles created a complicated and turbulent airflow pattern.

The CFD-simulations presented here were carried out using the Fluent 4.5 software, k-ε turbulence model and second order discretisation. The computational grid consisted of more than 500 000 control volumes. The room temperature level was set to about 22 °C in the simulations.

RESULTS

The omnidirectional values of air speed and turbulence intensity were calculated from the directional values of the CFD-simulation results with the correction Equations 3-4. The draught rating *DR* was then calculated by using Equation 1. The directional value of draught rating was calculated directly from the CFD-results without the corrections. In the office room case, the corrections were carried out with the macro language formulas of CFX shown in Figure 1. In the industrial hall case, the corrections were done with the user-defined functions of Fluent.

CFD-simulation case 1: Office room

The profiles of both the directional and the omnidirectional values at one measurement location, 1.78 m from the left wall, are shown in Figure 4 together with the measurement results. The average and maximum values at all measurement locations in the occupied zone are shown in Table 1.

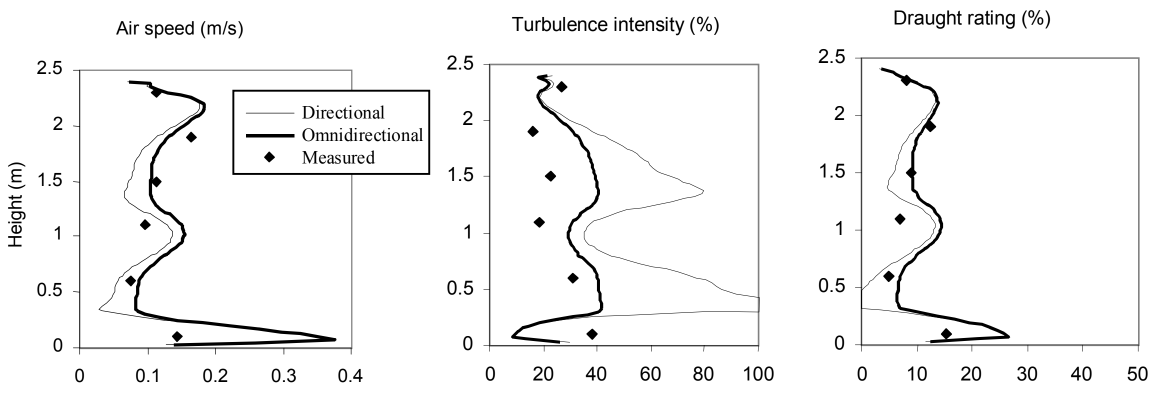

Figure 4. Effect of the turbulence correction on CFD- results at one measurement location. The omnidirectional values are appropriate for assessing thermal comfort.

Table 1. Effect of the turbulence correction on CFD-results at the measurement locations in the occupied zone of the office room.

	Directional	Omnidi-rectional	Difference
Average velocity (m/s)	0.143	0.159	0.016
Max. velocity (m/s)	0.846	0.853	0.007
Average turb. intensity (%)	44	28	16
Max. turb. intensity (%)	289	42	-247
Average draught rating (%)	10.9	12.6	1.7
Max. draught rating (%)	66.5	66.8	0.3

CFD-simulation case 2: Scale model of industrial hall

The average and maximum values of the relevant variables in the occupied zone are shown in Table 2. They were calculated for the same 90 points where the measurements took place. The comparison with the measured results is not shown here, it has been done elsewhere (Ala-Juusela, 2001).

Table 2. Effect of the turbulence correction on CFD-results at the measurement locations in the occupied zone of the scale model industrial hall.

	Grille air supply			Nozzle air supply		
	Direc-tional	Omnidi-rectional	Differ-ence	Direc-tional	Omnidi-rectional	Differ-ence
Average velocity (m/s)	0.137	0.180	0.043	0.274	0.339	0.065
Max. velocity (m/s)	0.291	0.312	0.021	0.580	0.596	0.016
Average turb. intensity (%)	73	35	-38	68	32	-36
Max. turb. intensity (%)	222	42	-180	335	42	-293
Average draught rating (%)	14.2	18.1	3.9	32.7	38.1	5.4
Max. draught rating (%)	29.7	30.8	1.1	59.7	62.1	2.4

DISCUSSION

The effect of the turbulence correction on the average value of the draught rating was 3.9 and 5.4 %-units in the industrial hall case and 1.7 %-units in the office room case. In the earlier study, the effect on the measured value of the draught rating was 7 %-units (Koskela, 2001), which is higher than the values found in this study. The effect of the correction on the maximum value of the draught rating was smaller, 1.1 and 2.4 %-units in the industrial hall case and only 0.3 %-units in the office room case.

The average air speed was clearly higher than the directional air velocity. The corrections obtained for the air velocity were 0.043 and 0.065 m/s for the industrial hall case and 0.016 m/s for the office room case. The relative corrections were 31 %, 24 % and 11 %. The measured correction in the earlier study was 0.05 m/s, which is close to the values in the industrial hall case. The effect on maximum velocities was smaller, being 0.016 and 0.021 m/s (7 % and 3 %) for the industrial hall case and 0.007 m/s (1 %) for the office room case.

The effect of the correction on turbulence intensity was large in all cases. The average omnidirectional turbulence intensity was 36 and 38 %-units lower than the directional value in the industrial hall case. In the office room case the difference was 16 %-units. The maximum values of the directional turbulence intensity were typically 5 times higher than the maximum omnidirectional values.

The fact that the omnidirectional turbulence intensity is much lower than the directional one decreases the differences between the directional and omnidirectional draught ratings. Still the corrected average draught ratings were noticeably higher than the incorrect, directional draught ratings. However, at the point of the maximum draught rating, the effect of the correction was small.

The importance of the turbulence correction is dependent on the flow conditions of the studied case and the turbulence level in the flow. In CFD-simulations, the obtained turbulence level depends on the performance of the turbulence model applied. In this study, the effect of the correction was notably higher in the industrial hall case, where the turbulence level was higher, than in the office room case.

In the earlier experimental study, the effect of the turbulence correction was larger than in the CFD-results of this study. In real room airflow conditions, the velocity fluctuations are caused, in addition to turbulence, also by the fluctuations of the room airflow pattern. This time-dependent nature of the flow cannot be properly predicted by the steady state CFD-simulations using turbulence models. Also, the assumption of isotropic and normally distributed turbulence is not generally valid in room airflows. Therefore, the steady state CFD-simulations can be expected to underestimate the velocity fluctuations. These problems could possibly be overcome by using time-dependent large eddy simulation (LES).

CONCLUSIONS

CFD-simulations tend to underestimate thermal discomfort if the CFD-results are applied without turbulence correction. The effect of the correction on air speed and draught rating values was found to be significant in room airflows with high turbulence intensity. It is especially large when considering average values in the occupied zone of room. The correction for the maximum speed or maximum draught rating in the room was, instead, smaller. It is recommended that the turbulence correction should be applied to CFD-results when calculating estimates for air speed, thermal comfort and draught risk. The method can easily be implemented in modern post-processors.

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