The Potential for Solar Powered Desiccant Cooling

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Abstract
The air conditioning industry has, in recent years, come under sustained attack from a range of economic, environmental and regulatory pressures. Global environmental concerns, improving standards of ventilation and increasing concerns about indoor air quality have all contributed to a sea-change in design thinking. Professional guidance is increasingly steering clients and consultants away from full air conditioning to natural ventilation and mixed mode solutions in new and refurbishment projects. Nevertheless inefficient lighting, increases in computing equipment, and architectural fashion mean that overheating is now the predominant design consideration for new offices in the UK. Older buildings often constrain natural solutions. The situation is likely to worsen if global warming scenarios are accurate. It is vitally important that the building industry seeks innovative methods of maintaining and improving the quality of the indoor environment, under potentially more demanding performance criteria, without increasing the environmental impact. Solar air conditioning potentially offers an elegant exemplar of a clean, sustainable technology which is consistent with UK and International commitment to Sustainable Development. By using solar/gas hybrid desiccant cooling and dehumidification technology it may be possible to offer a benign solution to cooling of buildings, in new as well as refurbishment situations, and significantly reduce greenhouse gas emissions. Desiccant cooling is a potentially environmentally friendly technology which can be used to condition the internal environment of buildings. Unlike a conventional air conditioning system which relies on electrical energy to drive the cooling cycle, desiccant cooling is a heat driven cycle. The feasibility of solar desiccant cooling in the UK is examined in this paper; an investigation on the possibility of using solar energy to drive the desiccant cooling cycle is conducted. Through the use of a parametric study, the energy consumption and costs associated with desiccant cooling are evaluated.
Keywords: desiccant cooling, heat driven cooling, heat powered cycles, solar

1 Introduction

Desiccant materials such as silica gel have long been used to dehumidify air. In such processes, moist air is passed over a surface which is coated with the desiccant substance. As the moist air passes across the surface the desiccant material absorbs moisture from the air, thus dehumidifying the air stream. In order to drive off the moisture absorbed by the desiccant surface, the desiccant then has to be physically moved into a dry hot air stream. In the case of desiccant wheels (one of the most commonly used desiccant systems), the moisture laden section of the wheel slowly rotates (i.e. at 16 revs/hour) from the moist air stream to the hot dry air stream where it is dried out in a process called 'regeneration'.

In a typical desiccant cooling system a desiccant wheel is coupled to a thermal wheel in a single air handling unit (AHU), to produce a system which is capable of heating, cooling, and dehumidifying air, with little or no need for refrigeration [1]. Because it is a heat driven cycle, such a system has the potential to reduce both energy costs and environmental pollution, when compared with conventional vapour compression based systems [2]. Electrical energy consumption is replaced by heat consumption, which in most countries produces much less carbon dioxide (CO,). In the UK the burning of natural gas in a boiler produces 0.21 kg CO_2/kWh whereas delivered electricity produces 0.68 kg CO_2/kWh [3]. In addition, the significant reduction in size, or elimination, of refrigeration plant, results in reduction in the refrigerant charge required. Desiccant cooling affords an opportunity to utilise waste heat and the regeneration temperatures required mean that theoretically they could be coupled to solar collectors to produce an extremely benign cooling system.

2 The system

A typical desiccant cooling system is illustrated in Figure 1, and comprises a thermal wheel and a desiccant wheel located in series. On the supply air side of the AHU, a cooling coil and/or an evaporative cooler is located after the thermal wheel. A heating coil may also be located after the thermal wheel, for use in winter time if required. An evaporative cooler is located in the return air before the thermal wheel to enhance the heat transfer across the wheel. For the purpose of driving the cycle and regenerating the desiccant wheel a heating coil is located between the thermal wheel and the desiccant wheel.

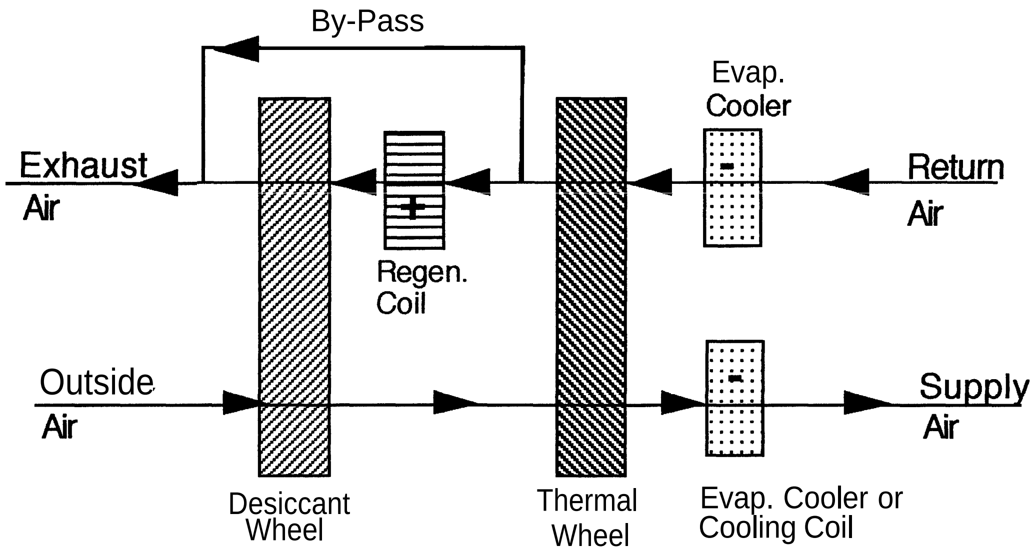

Figure 1: A typical desiccant cooling air handling unit

The cooling/dehumidification process is illustrated by the psychrometric chart shown in Figure 2. During the summer time, warm moist air at for example 26°C and 10.7g/kg moisture content is drawn through the desiccant wheel so that it comes off at say, 39°C and 7.3g/kg moisture content. The psychrometric process line for the air passing through the desiccant wheel on the supply side, has a gradient approximately equal to that of a wintertime room ratio line of 0.6 on the psychrometric chart. The supply air stream then passes through the thermal wheel where it is sensibly cooled to say, 23°C. The air then passes through a small direct expansion (DX) or chilled water cooling coil and is sensibly cooled to the supply condition of say, 17°C and 7.3g/kg moisture content. It should be noted that if humidity control is not required in the space, then the cooling coil can be replaced by an evaporative cooler with an adiabatic efficiency of approximately 85%. In which case, air may be supplied to the room space at say, 16.2°C and 10.2g/kg moisture content.

On the return air side, air from the room space at for example, 22°C and 8.6g/kg moisture content is first passed through an evaporative cooler so that it enters the thermal wheel at approximately 16.2°C and 10.8g/kg moisture content. As the return air stream passes through the thermal wheel, it is sensibly heated to approximately 35°C. The air stream is then heated up to approximately 55°C in order to regenerate the desiccant coil. It should be noted that in order to save energy approximately 20% of the return air flow by-passes the regenerating coil and the desiccant wheel [4].

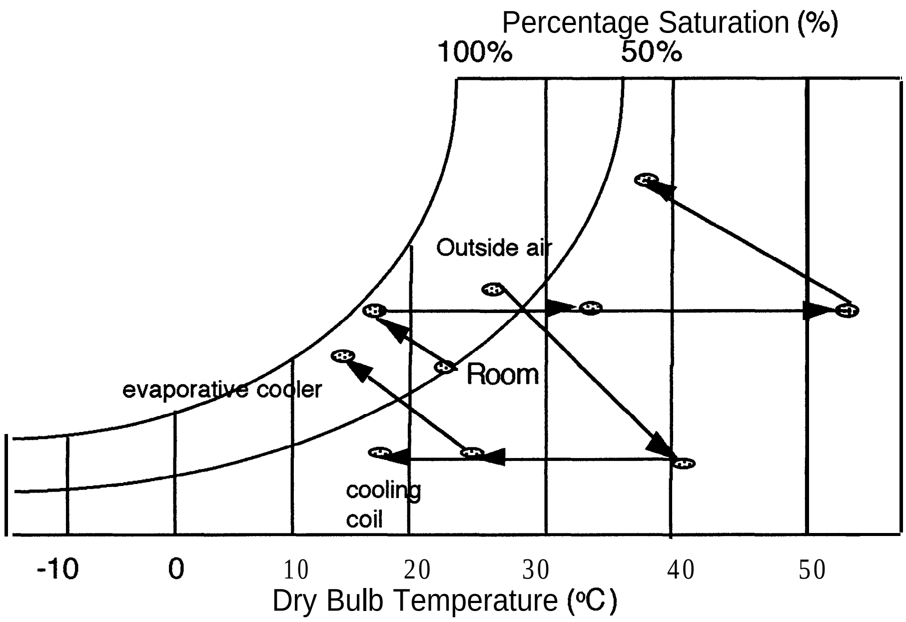

Figure 2: Desiccant system in cooling/dehumidification mode

3 Evaporative cooling

Through the use of a desiccant cooling air handling system it is possible to both dehumidify and sensibly cool the fresh-air supply. However, this often requires the installation of a chilled water cooling coil in the supply air stream. Whilst the installation of a cooling coil might result in an overall energy saving, the downside from the environmental perspective, is the introduction of refrigerants into the system. It is possible to avoid the use of a cooling coil, by installing an evaporative cooler in the supply air stream, to provide the required sensible cooling. However, this strategy has two major drawbacks:

- The air supplied to the room space is humid. Consequently, although the system might achieve the required degree of sensible cooling, it may not be able to provide adequate latent cooling, with the result that the occupants may find the environment uncomfortable. In addition, the high humidity in the room space is likely to lead to condensation problems.
- In order to achieve the required degree of sensible cooling, it is often necessary to perform a large amount of dehumidification on the desiccant wheel. This high level of dehumidification necessitates a high regeneration temperature, and consequently the energy consumption is increased dramatically.

The above drawbacks are significant and as a result most desiccant cooling systems incorporate some form of cooling coil. However, it is important to note that this cooling coil will be considerably less than that which would be incorporated into a conventional fresh air dehumidification system.

4 Solar application

Since desiccant cooling is a heat driven cycle, it would appear logical and synergistic that it might be powered by solar energy. However, the use of solar energy introduces constraints on the application of desiccant cooling. Assuming a ratio of solar collectors to building floor area of 1: 10, then the available heat (in northern Europe) to power the cooling cycle will be (conservatively) in the region of 25 to 50 W/m^2, depending on the climate, type and orientation of the solar collector. Consequently, the heat must be harnessed effectively. The desiccant cooling cycle is an open cycle, which rejects moist air at a high temperature, which is unsuitable for recirculation. Also the parasitic losses may be significant. The greater the air volume flow rate supplied to the room space, the greater the fan power required and the heat energy consumed. Therefore, if desiccant cooling is used in an all air application, the regeneration heat load is many times greater than the available solar energy. However, if the bulk of the sensible cooling within a space is carried out using a water based system such as a chilled ceiling, with the desiccant AHU dehumidifying and 'tempering' the incoming fresh air, then the air volumes handled will be much less and the 'solar energy' may make a significant contribution.

Theoretical work by Beggs and Warwicker has shown that desiccant cooling is best applied to installations in which the bulk of the sensible cooling is performed by a system such as a chilled ceiling [2]. In such an application the desiccant system treats only the incoming ventilation air. Because chilled ceilings are designed to have a dew point of approximately 17°C, it is possible to use low grade chilled water which can be produced using evaporative cooling towers. This avoids the need to install vapour compression refrigeration machinery,

5 The solar desiccant model

In order to investigate the potential for coupling a desiccant system to solar collectors in a European application, a solar desiccant computer model was developed at the University of Leeds. The model simulated the psychrometric and thermodynamic processes associated with desiccant cooling, using the following assumptions:

- The desiccant cooling system was employed solely to dehumidify the incoming fresh air supply, and to provide when required supplementary sensible cooling. It was assumed that the bulk of the sensible cooling would be performed by a separate water based system.

- The desiccant cooling system did not contain a cooling coil. The required degree of sensible cooling being achieved through the use of an evaporative cooler.
- The desiccant cooling system contained a solar heating coil located directly before the regeneration coil.
- The desiccant cooling system incorporated a 20% bypass on both solar heating and regeneration coils.
- Regeneration and supply air temperatures were specified, and the room condition was allowed to vary.

The solar desiccant cooling model considered only the primary and delivered energy consumption associated with the thermal aspects of the desiccant cooling cycle. The associated fan energy consumption was ignored.

6 Methodology

In the theoretical study, the system shown in Figure 3 was analysed in order to determine the running cost, primary energy consumption and CO, emissions attributable to a desiccant cooling system on a m^2 of floor area basis.

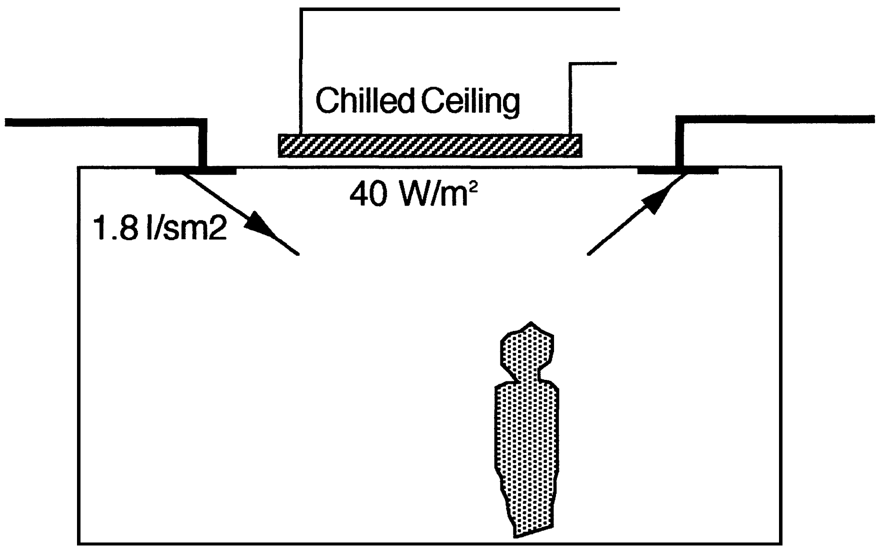

Figure 3: A chilled ceiling application in an office building where fresh air is introduced at high level

The system was analysed in cooling mode only, under the part-load conditions shown in Table 1. The study focused solely on the energy consumption of the desiccant fresh air system. In the study the energy consumption of the chilled ceiling was ignored, since this cannot be connected to a solar energy supply. The maximum sensible cooling output of the chilled ceiling was assumed to be 40 W/m^2. It was assumed in the study that under peak load conditions the desiccant cooling system would make-up the 10 W/m^2 short fall in sensible cooling. However, under part load the desiccant system would supply air to the space at room temperature. The psychrometric processes involved in the study are shown in Figure 4.

	Sensible Heat Gain (W/m^2)	Latent Heat Gain (W/m^2)	Outside Air Temp (^{0}C)	Outside Air Percent. Sat (%)	Outside Air Moist.Cont. (g/kg)
Loading					
Peak	50	7	30	50	13.7
Mid	40	7	26	50	10.7
LOW	30	7	22	50	8.4

Table 1: Part-load data used in Parametric Study

Throughout the study it was assumed that the solar heating coil was connected to solar panels via insulated pipework, containing a glycol/water mixture having a specific heat capacity of 3.7 kJ/kgK. The glycollwater flow rate was assumed to be 0.0016kg/s per m^2 of floor area, which equated to a solar collector to floor area ratio of 1: 10. The glycollwater flow and return temperatures were assumed to be a constant 60.1°C and 53.9°C respectively, providing 36.7 W/m^2 of heat energy to the desiccant cooling system. In reality this value would vary considerably depending on time, solar insolation, orientation and collector type. However, the purpose of the study was to investigate the feasibility of using solar energy to drive a desiccant cooling installation, and so a constant relatively pessimistic value was chosen.

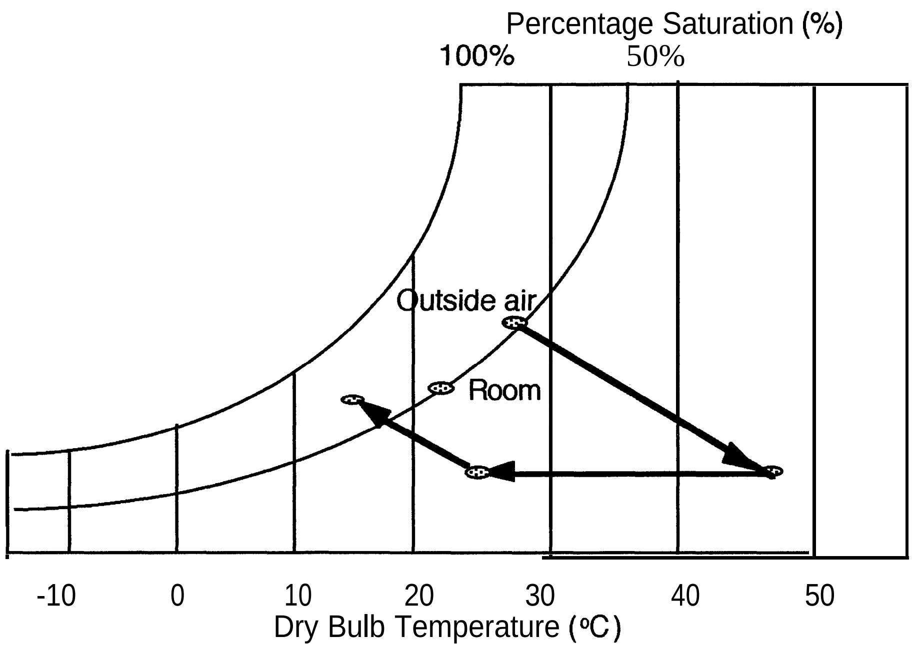

Figure 4: Psychrometric process (For clarity the return air cycle is omitted)

The operating data for the study is shown in Table 2. It should be noted that the degree of sensible cooling achieved by the desiccant cooling system is governed by the degree of dehumidification achieved by the desiccant wheel,-and that this in turn is governed by the regeneration air temperature. In short, the greater the regeneration air temperature, the lower the supply air temperature achievable to the room space. In order to determine the impact of varying the regeneration air temperature on the energy consumption of the cycle, the peak load condition simulation was carried out twice; once with a regeneration temperature of 55°C and a supply air temperature of 20.5°C, and once with a regeneration temperature of 75°C which enabled the supply air temperature to be reduced to 20°C. There was no benefit to be gained from raising the regeneration air temperature

- Section 7, and the remaining part-load simulations were undertaken using a regeneration air temperature of 55°C. The energy data applied in the study is shown in Table 3.

Loading	Supply Air Volume Flow Rate (l/sm 2)	Regen Air Temp. (°C)	Moisture Cont. Leaving Desic. Wheel (g/kg)	Ceiling Cooling output (W/m^2)	Sensible Cooling From Desiccant Cooling System (W/m^2)	Supply Air Condition Peak
Peak	1.8	55	10.2	40.0	10.0	20.5°C & 13.8 glkg
Peak	1.8	75	7.7	40.0	10.0	20.0°C & 12.2 glkg
Mid	1.8	55	7.3	40.0	0.0	22.0°C & 8.4 glkg
Low	1.8	55	5.1	30.0	0.0	22.0°C & 5.1 gtkg

Table 2: Operating data for the various loadings

Unit cost of gas	**1 SO p/kWh**
Unit cost of electricity	**5.00 p/kWh**
Efficiency of heating system	**70 %**
COP of DX/chilled water coil system	**2.5**
Electrical generation efficiency	**35 %**
CO, coefficient for gas consumed	**0.21 g/kWh**
CO, coefficient for electricity consumed	**0.68 kg/kWh**

Table 3: Cost and energy data used in analysis

7 Results

The results of the study are shown in Table 4.

System	Regen. Air Temp. (°C)	Supply Air Condition	Room Air Condition	Regen. coil Duty (W/m^2)	Delivered gas ($Wh/h/m^2$)	Cost per hour ($p/h/m^2$)	CO_2 produced ($kg/h/m^2$)
Non-solar (Peak)	55	20.5°C & 13.8 glkg	25.0°C & 15.2 glkg	30	43	0.07	0.009
Solar (Peak)	55	20.5°C & 13.8 g/kg	25.0°C & 15.2 g/kg	1	2	0.00	0.000
Non-solar (Peak)	75	20.0°C & 12.2 g/kg	24.5°C & 13.5 g/kg	54	77	0.12	0.016
Solar (Peak)	75	20.0°C & 12.2 glkg	24.5°C & 13.5 g/kg	25	35	0.05	0.007
Non-solar (Mid.)	55	22.0°C & 8.4 glkg	22.0°C & 9.7 glkg	38	54	0.08	0.011
Solar (Mid.)	55	22.0°C & 8.4 g/kg	22.0°C & 9.7 g/kg	9	12	0.02	0.003
Non-solar (Low)	55	22.0°C & 5.1 g/kg	22.0°C & 6.4 glkg	45	64	0.10	0.013
Solar (Low)	55	22.0°C & 5.1 g/kg	22.0°C & 6.4 glkg	16	22	0.03	0.005

Table 4: Analysis results (system incorporating evaporative cooler)

It can be seen from the results presented in Table 4 that coupling the desiccant cooling system to solar collectors produces significant savings in both running cost and CO, emissions. The greatest saving was achieved at peak load. This is because the return air leaves the thermal wheel at a higher temperature, and consequently, the solar heater is almost able to achieve an air temperature of 55°C without much help from the regeneration coil. However, despite the impressive energy savings achieved, based on these assumptions, the results indicate that operation of the system without a cooling coil is quite impracticable for most applications, since it results in high room air humidities. Indeed, for the installation shown in Figure 3 the results indicate that for much of the year condensation would occur on the surface of the chilled ceiling. However, this problem can be overcome by replacing the evaporative cooler by a small DX or chilled water cooling coil. If the study is repeated using a cooling coil instead of an evaporative cooler, then the results achieved are those shown in Table 5.

System	Regen. Air Temp. (°C)	Supply Air Condition	Room Air Condition	Regen. Duty (W/m²)	Delivered gas (Wh/h/m²)	Delivered electricity (Wh/h/m²)	Cost per hour (p/h/m²)	CO_2 produced (kg/h/m²)
Non-solar (Peak)	55	17.5°C & 10.2 g/kg	22.0°C & 11.5 g/kg	32.1	7.0	45.8	0.10	0.014
Solar (Peak)	55	17.5°C & 10.2 glkg	22.0°C & 11.5 glkg	2.5	7.0	3.6	0.04	0.006
Non-solar (Mid.)	55	22.0°C & 7.3 glkg	22.0°C & 8.6 glkg	38.9	0.4	55.6	0.09	0.012
Solar (Mid.)	55	22.0°C & 7.3 glkg	22.0°C & 8.6 g/kg	1.1	0.4	1.6	0.00	0.001
Non-solar (Low)	55	22.0°C & 5.1 g/kg	22.0°C & 6.4 glkg	45.0	0.0	64.2	0.10	0.013
Solar (Low)	55	22.0°C & 5.1 glkg	22.0°C & 6.4 glkg	16.0	0.0	22.9	0.03	0.005

Table 5: Analysis results (system incorporating cooling coil)

Table 5 shows that the inclusion of a refrigerated cooling coil increases CO, production under peak-load conditions, however for most of the year it will be inoperative and thus the overall increase in CO, would be minimal. It should also be noted that despite the removal of the supply air evaporative cooler, the solar collectors are still able to significantly reduce the overall energy consumption, and CO, emissions of the system.

The results in Table 5 indicate that for most of the operating year the conditions achieved in the room space would be acceptable. However, the peak load results suggest that condensation might still be a problem if a regeneration air temperature of 55°C is used. Consequently, under peak load conditions it would be advisable to increase the regeneration air temperature to say 75°C which would achieve an acceptable room air condition of 22°C and 9.1g/kg moisture content. If the operating conditions of the solar panel are optimistic then it is likely that these regeneration temperatures could be achieved during peak demand.

8 Conclusions

The energy study reported in this paper represents an initial study to determine feasibility of coupling solar collectors to a desiccant cooling system in an northern European application. To this end the study has been successful, since it has demonstrated that the inclusion of a solar heater into the desiccant cooling cycle can result in large energy savings. Indeed, under the mid and low load scenarios the thermal energy costs are negligible.

The main conclusions to be derived from the energy study are as follows:

(i) The inclusion of a solar heater into the desiccant cooling cycle can lead to significant savings in primary energy consumption and associated CO, emissions.

(ii) Due to desiccant cooling being an open cycle, solar energy can only be effectively used in applications where the supply air volume flow rate is small. This effectively limits its application to installations where the bulk of the sensible cooling system is undertaken using a water based system.

(iii) The regeneration air temperature should be kept as low as is practically possible, in order to minimise fossil fuel energy input.

(iv) There is little or no benefit to be derived from using an evaporative cooler on the supply air side. The inclusion of such a device can result in increased energy consumption and unacceptably high air humidities in the room space. It is preferable to include a small refrigerated cooling coil, which is only used under conditions of peak load.

The study is at an exploratory stage and further work will investigate the consequences of varying supply volumes and regeneration temperatures. Solar data from a sample site will be used to provide realistic profiles of water supply temperature.

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References

1. Busweiler U.(1993) Air conditioning with a combination of radiant cooling, displacement ventilation, and desiccant cooling, *ASHRAE Transactions*, pp. 503-10
2. Beggs C. B., and War-wicker B. (1998) Desiccant cooling: Parametric Energy Study *Building Services Research and Technology*, Vol. 19, Number 2.
3. Smith J.T., and Webb B.C. (1995) A comparison of the CO, emission rates of gas-fired and electrically driven chiller/heater machines under dynamic loading. *Proceedings of the CIBSE National Conference, Eastbourne*, pp. 151-9
4. Munters Ltd. (undated) *MCC - Series Cooling Cassette*