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NOVEL VENTILATION COOLING SYSTEM FOR REDUCING AIR CONDITIONING USE IN BUILDINGS

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ABSTRACT

Night-time cooling of buildings is a recognised way of reducing the use of air conditioning, and hence energy consumption. The paper describes the construction and testing of a *latent* heat storage system, which uses a novel combination of night cooling, heat pipes and phase change materials (PCMs) and has the distinct advantage that it is suitable for fitting to existing buildings. The design of the heat pipe/PCM heat storage unit is briefly discussed. The test rig was constructed in a room of approximately 14 m^2 floor area, and consisted of a number of heat pipe/PCM units mounted radially close to the ceiling with forced convection from a large propellor fan. At night, cool outdoor air is passed over one end of the heat pipe to freeze the PCM. During the day, warm air from the room is circulated over the heat pipe to melt the PCM and thereby cool the air. Test results from the prototype over a range of summer weather conditions are presented, showing a significant heat storage during the summer day with little external energy input. The advantages of the system compared to existing concrete beam systems are briefly described.

KEYWORDS

night cooling, heat pipe, PCM, latent heat storage, air conditioning, energy saving, environmental benefit

ENERGY-EFFICIENT COOLING

It is estimated that best practice building design coupled with natural ventilation could reduce energy use and costs by up to 80% of the air conditioned solution (AIVC, 1996; Turrent & Barlex, 1997). In addition to the high financial cost of air conditioning, the environmental cost of non-renewable energy consumption is high. Studies such as reported in CIBSE (1997) suggest that up to 90% of UK office workers preferred the more variable environment provided by natural ventilation to air conditioning, provided the building was kept within reasonable comfort limits. Passive cooling techniques can therefore provide environmental, financial, health and satisfaction benefits.

One established method of passive cooling is the use of ventilation during the night to pre-cool exposed mass inside the building (eg. Birtles et al. (1996). However, large exposed mass is difficult to fit in existing buildings. Latent heat storage (LHS) systems have more potential. A phase change material (PCM) will store a given amount of latent heat in a much smaller volume than the same amount of sensible heat. For example, paraffin wax has a latent heat storage (LHS) capacity of 196 MJ m^{-3}, compared with 3.8 MJ m^{-3} of sensible heat storage for a 2°C temperature rise in concrete. LHS also takes place at a constant temperature, reducing the likelihood of overcooling. PCMs have been used in conventional building materials (Kaasinen, 1992). The main problem with using PCMs in building materials is achieving a sufficiently high heat transfer rate between the room air and PCM. The solution adopted here is to use *heat pipes* (Dunn & Reay, 1994), with forced convection on the air side of the pipe provided by a low-energy propellor fan. Heat pipes have been used for a wide variety of applications (eg. review by Seshan & Vijayalakshmi, 1986), such as solar energy collectors and waste heat recovery.

Figure 1 shows a cross-section of the system which consists of a number of identical heat pipe units arranged radially above a propellor fan. One half of each heat pipe is embedded in the PCM and the other side is exposed to the air. Air is drawn over the exposed ends of the heat pipes with the low powered propellor fan, which serves the dual purpose of providing convective cooling of the occupants. During the day, heat is transferred to the PCM, melting it and reducing the temperature in the room. At night, cool outside air is drawn over the heat pipes, and heat is extracted from the freezing PCM. The heat pipes are therefore reversible, which is a new development over previous heat pipe-PCM exchangers (eg. Abhat, 1978). The heat pipes avoid the need for complex heat exchange geometries on the surface of the PCM, and the system is easily retrofitted. This paper presents an outline of the construction and testing of a prototype PCM heat transfer module, and results from experiments cooling a real office in summer 1999.

HEAT PIPE/PCM DESIGN

A simple model was designed to predict the heat exchange between air and the PCM, via the heat pipe. The purpose of the model was principally to size the units for design and testing, and details are given in Turnpenny et al. (2000). Each unit consisted of a heat pipe (Isoterix Ltd) 1000 mm length by 19 mm diameter with a copper outer container, since copper's high thermal conductivity minimised the temperature difference between the two ends of the pipe. Methanol was used as the working fluid. Glaubers' Salt ($Na_2SO_4 \bullet 10H_2O$) was used as the PCM, and was modified by the addition of freeze point depressants to give melt/freeze temperatures of about 21 °C. The end of the pipe embedded in the PCM was plated with tin since the freeze point depressants were corrosive of copper. Further details of the unit design, including the wire finning arrangements can be found in Turnpenny et al. (2000).

The heat storage rate in the PCM over a melting cycle was calculated from:

$$Q = \frac{L_s}{t} \tag{1}$$

where L_s is the total latent heat capacity of the PCM (J) and t is the time for the whole volume to change phase (s). The mass of PCM required was calculated on the basis of 30 W m^{-2} heat gains over an 8 hour working day, or 3600 Wh in a 14 m^2 room. Since the latent heat of fusion of the PCM is 293 MJ m^{-3}, the total volume of PCM required was about 45 *l*. With each unit approximately 8 *l*, or 11.6 kg of PCM, six units were required, assuming the heat transfer rate is sufficient to melt all the PCM in 8 hours. Ten units were fitted to allow for lower heat transfer rates.

CONSTRUCTION OF RIG

The test rig consisted of a number of units arranged radially and mounted close to the ceiling on a stand, with the propellor fan axis in the centre of the circle (Figure 1). One of the units contained thermistors arranged to allow analysis of the melting and freezing behaviour of the PCM. The test room was 6 m × 4.8 m, facing south-south-east and receiving considerable solar gain in the summer. Two small windows were replaced with vents as part of the night ventilation strategy. In addition, new vents were cut in the plasterboard of the opposite wall to allow entry and exit of air. The room was partitioned down the middle with a wall of 6 cm-thick strawboard with a thermal conductivity of 0.1 W m^{-1} K^{-1}, and the cooling unit placed in one half. The floor area to be cooled was therefore approximately 14 m^2.

Each vent had a free area of about 0.1 m^2. During the day, the vents were shuttered and the extract fan switched off. At night, cool air was drawn inwards from upper-level vents, guided above the units by a movable board and passed over the heat pipes using the ceiling fan blowing downwards. The warmed air then passed through the exit vent, drawn by an extractor fan delivering about 8 air changes per hour sealed over the inside of the exit vent.

PRT sensors were positioned 70 cm above the floor - a height representative of a seated person - to determine a representative room temperature. Another PRT probe was sited outside, 140 cm above the ground and 10 cm away from the outside wall. Two PRT sensors were arranged above and below the units to assess the direct effects of their presence. These temperatures are indicated by T_u and T_d respectively. Three thermistors were embedded in the PCM, half way along the PCM container, at 10, 25 and 40 mm from the outer wall of the container to analyse the phase change behaviour of the PCM. The mean of these three temperatures is T_{pcm}. A movable hot-wire anemometer measured air speeds in the room. Smoke flow visualisation confirmed that the required flow patterns were achieved both during the day and at night. The air speed over the heat pipe was about 2 m s^{-1} at the fan speed used, which produced a satisfactory volume flow rate. Care was taken to ensure that the heat pipes were level. To generate the required temperatures in the room and hence in the PCM, it was necessary to provide heating with an electric fan heater (1 kW or 2 kW), sited at floor level.

RESULTS

A typical 24-hour test period consisted of (i) a period of heating during the day e.g. 4 - 6 hours at 1 or 2 kW with the extract fan OFF followed by (ii) overnight cooling with the extract fan ON. The ceiling fan was kept ON for the whole period with a constant flow rate and direction. The results of a run of one day and night are shown in Figure 2. The PCM temperatures at all three depths are within 1°C of each other, and the rapid response of the PCM to changes in air temperature (eg. in under an hour at about 450 minutes) indicates a good heat transfer into the PCM. Figure 2 shows a good melting response by the PCM to conductive heating from the heat pipe. When the PCM is below 20.5°C, the temperature falls by a small amount from nearest the heat pipe to the outer edge of the container (40 mm probe to 10 mm probe). As the PCM melts, between 20.5 and 22.5°C, the rate of temperature rise nearest the heat pipe falls, indicating latent heating, while the outer parts of the PCM continue to warm sensibly.

Figure 3 shows some typical results over a 32-hour period for (a) temperature difference between T_u and T_{pcm}, ie. the total temperature gradient for heat transfer, and (b) $T_u - T_d$. The first heating cycle was simulated using 1 kW heat input, and the second cycle used 2 kW. When the mean PCM temperature rises above about 23°C, it is reasonable to assume all the PCM is melted, and hence the heat added is the latent heat of storage for the full mass. Complete melting occurs over a period of about 8 hours when $T_u - T_{pcm} = 2°C$ (ie. when the room is 2°C warmer than the mealt temperature of the PCM) and over a period of about 3 hours when $T_u - T_{pcm} = 3.5°C$. Since the latent heat capacity of each pipe unit is 0.6 kWh, this implies heat transfer rates of 80 W and 200 W per unit respectively, or total heat transfer rates of 800 W and 2000 W for the room with 10 units. Under real operating conditions this is a good performance i.e. it should absorb internal gains at a fast enough rate to prevent excessive overheating of the room.

CONCLUSIONS

The present system has been compared with the well-known passive cooling system whereby a concrete slab of high thermal mass is cooled by night ventilation, and/or chilled water is passed through pipes embedded within the slab. A concrete slab of 5 tonnes (which would be very difficult to retrofit) would be needed to provide the same cooling rate as the present system with 10 heat pipe units. Moreover, with the present system there is no risk of condensation whereas reduction of the concrete temperature below the dew point of the ambient air can occur with chilled-ceilings. The current units therefore provide a much more weight- and installation-efficient system than the chilled ceiling option.

The financial and environmental benefits of the heat pipe/PCM system have also been compared with a standard air-conditioning unit. For example, fitting the proposed system in 2000 offices instead of air conditioning would reduce CO_2 emissions by approximately 430 tonnes per annum. The financial savings (in terms of capital and running costs) are also significant.

In summary, it is believed that the proposed system offers a significant improvement to the technology for the night-time cooling of buildings. Heat transfer rates of 200 W have been observed, which is sufficient to ameliorate the effects of high summer temperatures in a small office. There are substantial cost and energy-saving benefits to the system, both compared to conventional air-conditioning and other technologies such as cooled beams, and the system can easily be retro-fitted.

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FIGURE 1. CROSS-SECTION OF SYSTEM
(shown in daytime operation)

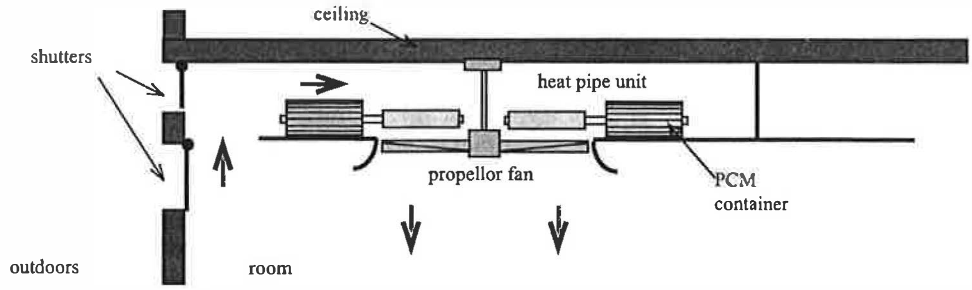

FIG. 2: PLOT OF THREE PCM TEMPERATURES FOR ONE COMPLETE CYCLE

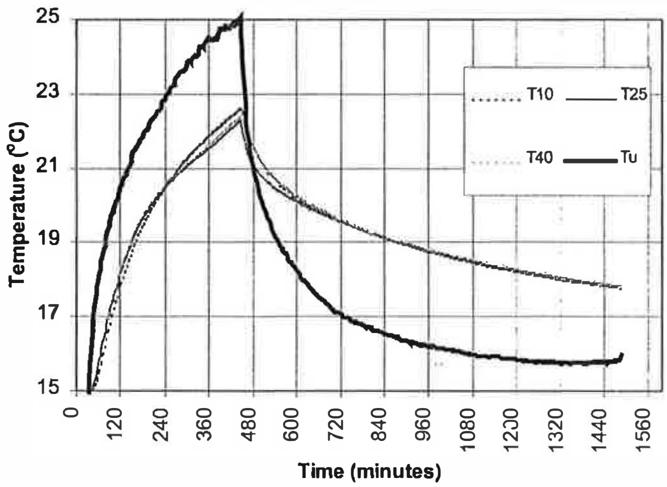

FIG. 3: TEMPERATURE DIFFERENCES T_u - T_d AND T_u - T_{pcm}, 1 AND 2 kW HEAT INPUT, 16 - 17 SEP 1999