SIMULATION OF A NOVEL HEAT PIPE/PCM SYSTEM FOR COOLING OF NATURALLY VENTILATED BUILDINGS

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ABSTRACT

The paper describes the numerical simulation of a novel ventilation cooling system. The ultimate aim of the mathematical model is to facilitate the design of the system for specific buildings and climates. The model directly takes account of time dependency in that the diffusion equation for the heat flow in the fabric of the room is solved. This approach means that important parameters (e.g. ventilation rates and internal heat gains) can be arbitrarily specified as functions of time. For convenience the model equations are solved with a spreadsheet, which means that the explicit method of solution has to be used. The model was tested against an analytic solution and close agreement was obtained. The model has then been used to investigate the performance of the system under a wide range of conditions. Examples of results relating to three important parameters are presented. It is also shown that the behaviour observed in the field with simple thermostatic control is reproduced by the model.

KEYWORDS

Natural ventilation, mathematical model, cooling, phase change material, carbon emission

INTRODUCTION

Brief description of system

In (Etheridge and Murphy 2004) a novel system for reducing or eliminating the need for air conditioning in buildings and thereby reducing energy consumption and carbon emissions is described. The novelty arises from the use of a phase change material (PCM) in combination with heat pipes and a fan. The principle of operation of the system is quite straightforward. During the night, cool air is used to “freeze” the PCM and during the day heat is extracted from the room air which “melts” the PCM. This cycle is repeated on a daily basis. The crucial process is the transfer of heat between the air and the PCM. Heat transfer coefficients need to be high, because the temperature differences between the air and PCM are low, typically less than 6 K. To determine the effectiveness of the system under real conditions, field tests were carried out and it was found that with simple thermostatic control the system could maintain a constant room air temperature in a similar way to a conventional air conditioner.

Objectives of the present work

The ability of the system to maintain a controlled temperature depends to a large extent on the mass of the PCM (the latent heat capacity) and on the heat transfer rate. However there are many other factors involved e.g. the nature of the internal heat gains and the thermal properties of the building. All of the factors need to be taken into account when designing a system for a particular application. The objective of the present work was to develop a simple mathematical model that could be used for a parametric study and subsequently as an aid to design.

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MATHEMATICAL MODEL

The current model simulates a single space as illustrated in Figure 1. The other main simplifications are:-

- the fabric surrounding the envelope is represented by a single wall, consisting of a homogenous material that separates the room from the exterior. In effect the heat transfer to other parts of the building is neglected. This corresponds to an internal envelope with low thermal mass and low heat transfer rates to surrounding spaces.
- radiation heat flows are not specifically included.
- the air within the space is assumed to be well-mixed, such that the temperature T_{air} is uniform at any given time.

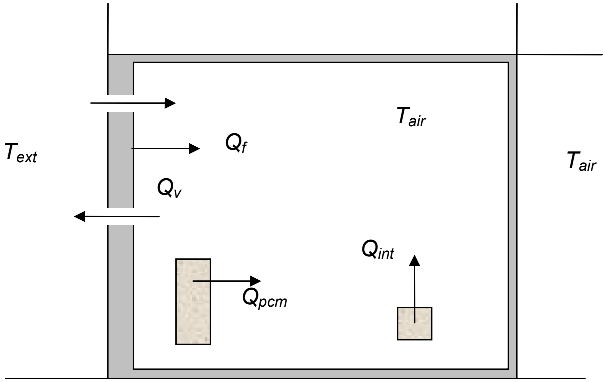

Figure 1. Basic configuration used for simulation

Governing equations

Since the air within the room is uniform at all times, the rate of change of T_{air} with time is given to a close approximation by the following equation (conservation of thermal energy for air in room)

$$\rho c V \frac{dT_{air}}{dt} = Q_f + Q_{\text{int}} + Q_{pcm} + Q_v \qquad (1)$$

where V, ρ and c denote room volume and the density and specific heat capacity of air. The terms on the right hand side are the heat flows to or from the air (taken as positive when the flow is in to the air) i.e. fabric Q_f, internal heat gain Q_{int}, the PCM system Q_{pcm} and ventilation Q_v respectively.
To solve equation 1 the heat flow rates were calculated as follows.

The fabric of the envelope acts as a thermal store i.e. the temperature distribution in the material changes with time, with corresponding changes to the heat flows to the external and internal air. The heat flow within the fabric is governed by the diffusion equation. Here we make use of the one-dimensional form i.e. the temperature varies with distance only in the x direction. The fabric has density ρ_f, specific heat capacity c_f and thermal conductivity k_f, thus the equation is

$$\rho_f c_f \frac{\partial T}{\partial t} = k_f \frac{\partial^2 T}{\partial x^2} \qquad (2)$$

which can be written as

$$\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2} \tag{3}$$

where α denotes the thermal diffusivity

$$\alpha = \frac{k_f}{\rho_f c_f} \tag{4}$$

The heat flow from the fabric to the room air is given by

$$Q_f = -A_f k_f \left.\frac{\partial T}{\partial x}\right|_{x=L} \tag{5}$$

where A_f denotes the fabric surface area and L the thickness of the wall.

The internal heat gain Q_{int} is specified as a function of time in terms of the heat gain per unit floor area

$$Q_{\mathrm{int}} = q_{\mathrm{int}} A_{fl} \tag{6}$$

where q_{int} denotes the heat gain per unit area [W/m^2] and A_{fl} denotes the floor area.

The heat flow between the PCM and the air is expressed in terms of an overall heat transfer coefficient for the heat exchanger, h_{pcm}, based on the difference between the temperatures of the air and the PCM

$$Q_{pcm} = h_{pcm}(T_{pcm} - T_{air}) \tag{7}$$

The ventilation rate is specified as a function of time in terms of the air change rate R, so that

$$Q_v = \rho\, VRc\ (T_{ext} - T_{air})/3600 \tag{8}$$

Boundary conditions

To solve equations 1 and 3 it is necessary to specify the following boundary conditions:-

(i) the temperature distribution in the fabric and the room air temperature at time t = 0

(ii) the temperatures of the wall surfaces as a function of time i.e. T_{wext} and T_{wint}.

The first condition can be specified approximately and the calculations carried out over a sufficient time period such that the results are independent of the initial values. The second boundary condition is more important. For our case we do not know T_{wext} and T_{wint} directly, but we can specify the heat transfer coefficients at the two wall surfaces (external and internal).

At the external wall the heat flux from the air to the material can be written both in terms of conditions in the material and conditions in the air i.e. taking the heat flux q as positive in the x direction

$$q_{fext} = h_{fext}(T_{ext} - T_{wext}) \quad \text{and} \quad q_{fext} = -k_f \left.\frac{\partial T}{\partial x}\right|_{x=0} \tag{9}$$

so that

$$\left.\frac{\partial T}{\partial x}\right|_{x=0} = -\frac{h_{fext}}{k_f}(T_{ext} - T_{wext}) \tag{10}$$

Similarly for the internal wall surface

$$q_{f\,\mathrm{int}} = h_{f\,\mathrm{int}}(T_{w\mathrm{int}} - T_{air}) \quad \text{and} \quad q_{f\,\mathrm{int}} = -k_f \left.\frac{\partial T}{\partial x}\right|_{x=L} \tag{11}$$

so that

$$\left.\frac{\partial T}{\partial x}\right|_{x=L} = -\frac{h_{f\,\mathrm{int}}}{k_f}(T_{w\,\mathrm{int}} - T_{air}) \tag{12}$$

In equations 10 and 12 the surface heat transfer coefficients, h_{fext} and h_{fint}, are empirical values, known to lie within a certain range, and which can include an approximate allowance for radiation.

Numerical solution

Equations (1) and (3) need to be expressed in finite difference form. For a time step Δt equation 1 can be written as

$$\Delta T_{air} = \frac{1}{\rho_a c_a V}(Q_f + Q_{\mathrm{int}} + Q_{pcm} + Q_v).\Delta t \tag{13}$$

where

$$T_{air}\{t + \Delta t\} = T_{air}\{t\} + \Delta T_{air} \tag{14}$$

where the heat flow rates are evaluated at time *t*.

The diffusion equation (3) is expressed in its finite difference form by dividing the wall thickness *L* into *N* equal parts, each of width Δx. The *explicit* method of solution (Price and Slack 1952), sometimes referred to as the Euler forward difference method, is used here. This has the advantage that the solution at a given time is obtained directly from the solution at the previous time, which means that one can use a spreadsheet to solve the equation. The disadvantage is that, for a given time step, it is not as accurate as other methods that rely on the solution of simultaneous equations e.g. the Crank-Nicholson method. For the explicit method the error is proportional to Δt, whereas for the latter method the error is proportional to Δt^2. It is also possible for the explicit solution to become unstable when Δt is too large.

Stability and numerical errors

Stability of the explicit method and accuracy of results places limits on the values of Δt and Δx. For the present calculations, the wall was divided into 11 equal slabs (N = 11). This led to Biot numbers based on Δx, $B = 0.5 h_f \Delta x / k_f$, that were less than 0.2.

A stability criterion is the Fourier number, $F = \alpha \Delta x / \Delta t^2$, and the value of Δt has to be chosen such that *F* is much less than 0.5. For the present calculations typical values were less than 0.1. In fact, it is easy to check for stability and numerical errors, by carrying out calculations with different values of Δt. For the calculations presented here Δt was usually taken as 30 s. Even with such a small time step the calculation times were only of order 1 s. Thus the use of the explicit method, although relatively inefficient compared to other methods, was perfectly acceptable.

Verification

To verify the written code, the results obtained were compared with those from an analytical verification test (Xiao et al 2005) i.e. the “homogenous/nonadiabatic/step” test. The comparison in terms of predicted load is given in Figure 2 for two time steps (10 s and 40 s). There is negligible effect of time step and the errors in the heat load were less than 2 %. It should be noted that this percentage is based on the actual value at a given time. The errors quoted in (Xiao et al 2005) for other mathematical models use the maximum load to evaluate the percentage, so the present errors are much smaller in real terms.

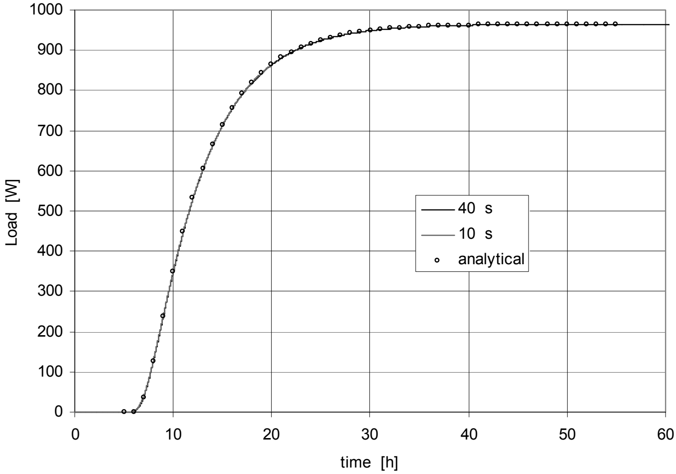

Figure 2. Calculated results compared with analytic solution

RESULTS OF SIMULATIONS

Parameters

There is a large number of parameters involved which can be classified under the following headings

- room and fabric properties
- PCM system
- ventilation rate
- internal heat gain (which includes the waste heat from the fan)
- external temperature.

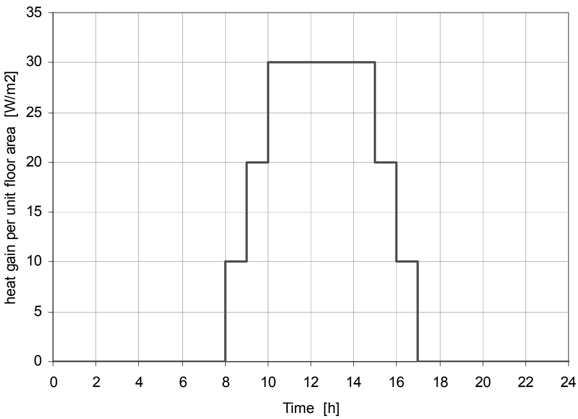

Figure 3. Example of variation of internal heat gain with time

The main room parameters are volume and floor surface area. The main fabric properties are density, specific heat, thermal conductivity, thickness, wall surface area and the heat transfer coefficients for the internal and external surfaces.

The properties of the system of importance are the latent heat storage capacity, E_{pcm}, the heat transfer rate to and from the PCM, the PCM transition temperature and the control system (primarily the thermostat setting). For the present calculations it was assumed that transition occurred at a discrete temperature (19 C) and that this was the same for "melting" and "freezing".

The ventilation rate (the flow rate of external air into the space) was specified by two values i.e. a day rate and a night rate.

The internal heat gain per unit floor area was specified as a step function of time as shown in Figure 3.

The external temperature was taken to be a sinusoidal function of time, with a period of 24 h. The mean value, the amplitude, the frequency and the phase were obtained from curve fits to actual records.

Method of solution

Each calculation was carried out for a period of three days. Where necessary, the initial values were adjusted until the results for the second and third days were nominally identical.

Results - effect of latent heat storage capacity

Figure 4 illustrates the importance of the storage capacity of the system. The sine curve shows the variation of external temperature over the 24 hour period. This corresponds to a fairly severe summer day in the UK. The other curves show the internal temperature as a function of time for a range of values of storage capacity i.e. E_{pcm} from 0 to 2.5 kWh in steps of 0.5 kWh. The thermostat was set to 23 [C] and all other parameters were kept constant. (The kinks in the curves between 08.00 and 17.00 correspond to the steps in the internal gains, as shown in Figure 3.)

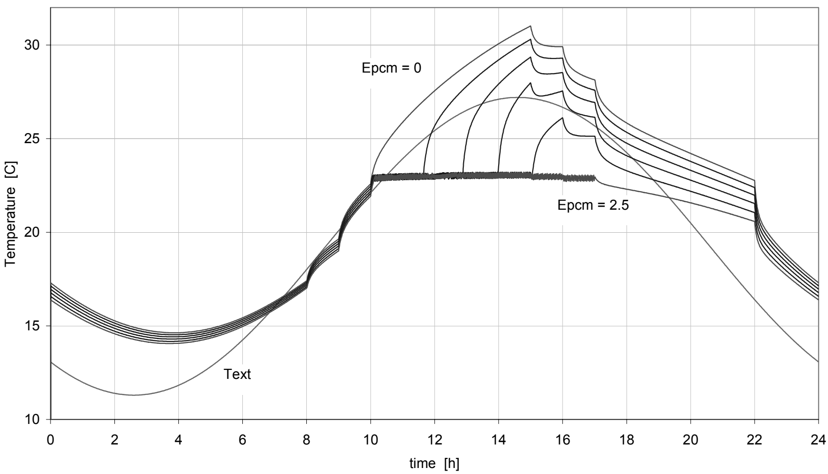

Figure 4. Effect of increasing storage capacity from 0 to 2.5 kWh in 0.5 kWh steps (h_{pcm} = 120 W/K)

The curve with E_{pcm} = 0 is the natural ventilation case with night cooling (R = 1 h^{-1} during the day and 5 h^{-1} during the night) and it can be seen that high temperatures occur, reaching a peak of 31 C at 15.00 h. The remaining five curves show the effect of the system with E_{pcm} values of 0.5, 1.0, 1.5, 2.0 and 2.5 kWh. It can be seen that the system is capable of controlling the room temperature. In fact it cycles ON and OFF, as observed in the field trial of the real system. However, once the storage capacity has been used the temperature increases rapidly to a level that is close to the natural ventilation. For this particular case a storage capacity of 2.5 kWh is required (the real system has a capacity of 4 kWh).

Results - effect of heat transfer rate

The rate at which heat is transferred from the room air to the PCM is an important parameter. For a given heat exchanger (heat pipes) the relevant coefficient is h_{pcm} (in W/K) in equation 7. The value of h_{pcm} can be increased by increasing the fan flow rate, but only at the expense of increased electrical power consumption. Figure 5 shows the effect on T_{air} of increasing h_{pcm} from zero to 120 W/K in steps of 20 W/K. As before, the thermostat was set to 23 C and all other parameters were kept constant.

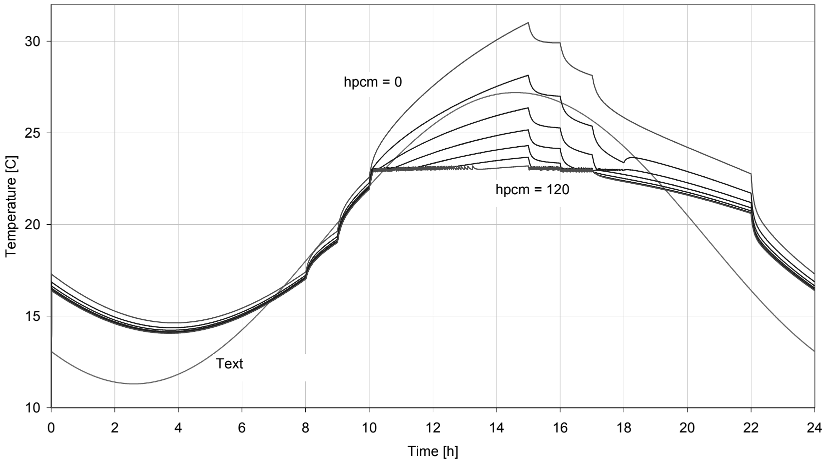

Figure 5. Effect of increasing heat transfer coefficient from 0 to 120 W/K (E_{pcm} = 2.5 kWh)

It can be seen that a value of 120 W/K is needed to maintain control. Close examination of the curve reveals that the system (fan) cycles ON and OFF between 10.00 and 13.15, but the fan is ON continuously from 13.15 to 15.00. At 15.00 it returns to cycling. This behaviour was also observed in the field trial.

Results – effect of fabric storage

Figure 6 shows the effect of increasing the specific heat capacity of the fabric from 600 to 1400 J/kgK in steps of 200. The latent heat capacity of the PCM is constant at 2.0 kWh and again all other parameters were also constant. It can be seen that the latent capacity of the PCM is effective for a longer time when the fabric heat capacity is increased. This is due to the fact that thermal storage by the fabric is still beneficial, although its effect is reduced by the lower room air temperatures.

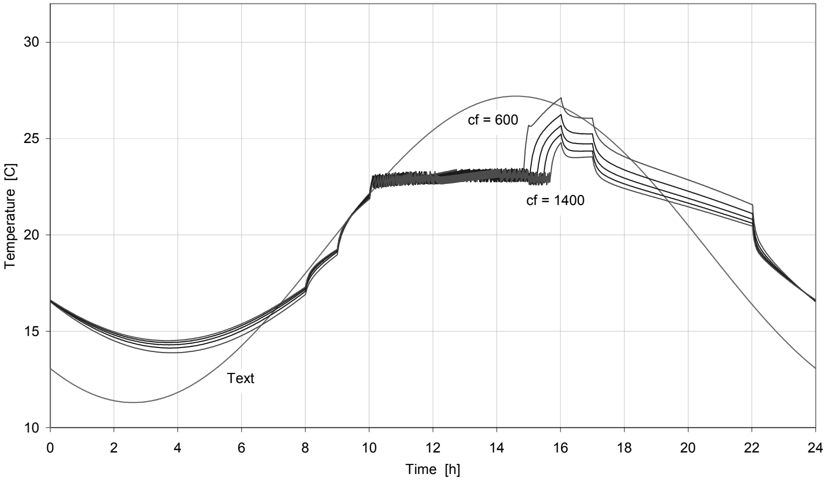

Figure 6. Effect of increasing specific heat capacity of fabric from 600 to 1400 J/kgK

CONCLUSIONS

A simple mathematical model has been developed to simulate the performance of a novel PCM cooling system. The ultimate objective is that the model can be used to assist the design of such systems. Results from a parametric study show the importance of the latent heat capacity and the overall heat transfer coefficient of the system. The results also display qualitatively similar behaviour to that observed in a field trial.

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