A SIMPLE MODEL FOR FREE COOLING CALCULATIONS

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ABSTRACT

We present a simple model to calculate the energy loss by free cooling at night. The time dependence of the exhaust air and wall surface temperatures is predicted by a simplified dynamic model that couples air flow, heat transfer, and wall temperature. For given ventilation rate the model predicts that the total heat extracted from the building during the night can be maximized by increasing the heat exchanging surface area and the thermal effusivity, $\sqrt{(\lambda\rho c)}$, of the wall materials. The influence of ventilation rate on the heat removed by freecooling at night is discussed. The model is tested in field measurements where the air flow is known from the stack effect.

RESUME

Nous proposons un modèle simplifié pour le calcul du refroidissement des bâtiments par ventilation nocturne. La température de l'air intérieur d'une zone est obtenue en établissant le bilan thermique entre pertes par ventilation d'une part et le tranfert de chaleur entre l'air et les parois d'autre part. La variation de la température de surface des parois dépend de la densité de flux de chaleur et de l'effusivité thermique, $\sqrt{(\lambda\rho c)}$, moyenne des matériau de construction. L'influence du taux de ventilation sur la quantité de chaleur éliminée est discutée. Le modèle est évalué par comparaison avec des mesures effectuées en vraie grandeur, et oû le débit d'air résulte de l'effet de cheminée.

1. INTRODUCTION

Convective air cooling is useful in heavy buildings located in climates where large diurnal temperature variations occur. Large thermal capacity structures cause the indoor temperature to be close to the average outdoor temperature. Freecooling at night removes heat from the building mass, resulting in a reduced average indoor air temperature.

Key issues of energy calculations of ventilative cooling are [1,4] the evaluation of the exhaust air temperature (dependence on heat transfer and flow rate) and the estimate of the mass flow rate (dependence on internal flow resistances), but to exploit freecooling at night, architects still have to dimension the openings, relying more on their feeling than on calculation. In this paper we present a model which can help in developing detailed guidelines for ventilative cooling.

In the following we first describe the algorithm for calculating the temperature of the exhausted inside air during cooling, then we present a comparison of the model with field measurements and finally, we discuss the parameters influencing the cooling potential of freecooling at night [3,4].

2. MODEL FOR VENTILATIVE COOLING

From the mass flow rate $\dot{m}$, and the temperature of the exhausted inside air T_{in}, the heat loss rate (taken positive from inside to outside) is given by

$$Q = C_p \dot{m} (T_{in}-T_{out}) \tag{1}$$

where T_{out} is the temperature of the outside air entering the building.The model couples the ventilative heat loss rate with the heat transfer between the air and the walls and a decreasing wall surface temperature. The mass transfer and thermal models form a four-node network (Figure 1).

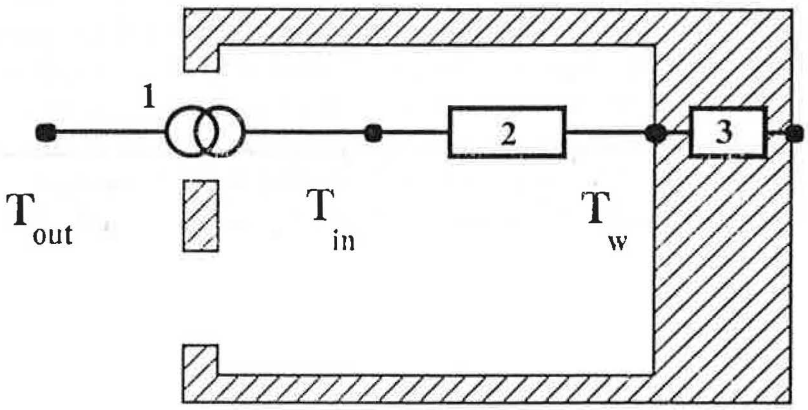

Figure 1. The model for ventilative cooling can be represented by a four-node network. (1) current source for ventilation heat loss (2) boundary layer resistance, (3) dynamic wall resistance. T_b is a reference wall temperature.

2.1 Mass flow rate

The mass flow due to the stack effect, can be calculated for various opening configurations. Our test case is a volume with two openings (area A_t at the top and A_b at the bottom). The influence of wind (in general an enhancement of the air flow) was not considered. For openings A_t and A_b much smaller in height than the distance H between the middle of the two openings, the mass flow rate is given by :

$$\dot{m} = \rho_{in} A_t C_1 \sqrt{\frac{2g}{T} \frac{(T_{in} - T_{out}) H}{1 + (T_{in} A_t / T_{out} A_b)^2}} \qquad (2)$$

where a single discharge coefficient C_1 (≈ 0.6) is assumed [2]. The ventilation heat flow is represented in Figure 1 by a heat source (1) , its value being given by Equations 1 and 2.

2.2 Heat transfer

The temperature difference between the wall surface and the air is described by a convective heat transfer coefficient h_c, and equals $T_w - T_{in} = q/h_c$, where q is the heat flux density. A fixed value of $h_c = (6\pm1) W/m^2K$ appeared to be consistent with the measurements [4]. The value of q is obtained by dividing the total heat loss Q by the heat exchanging wall surface area

$$T_w - T_{in} = q/h_c = Q/C_2 S_i\, h_c \qquad (3)$$

The coefficient C_2 is the fraction of the total internal wall surface area S_i, which is active in the heat transfer process. The heat transfer is represented in Figure 1 by a thermal resistance (2). The heat capacity of the air adds a small time constant [4] which is here neglected.

2.3 Wall surface temperature

We are interested in the time dependence of the surface temperature of the wall, T_w. The walls are considered to be in thermal equilibrium with the building until ventilation starts at t=0, $T_w(t=0)=T_b$. For a constant heat loss density q, the wall surface temperature of a semi-infinite solid is given by

$$T_w(t) = T_w(0) - \frac{2q}{b}\sqrt{\frac{t}{\pi}} \qquad (4)$$

where $b=\sqrt{\rho c \lambda}$ (the square root of the product of thermal conductivity, density, and specific heat) is the thermal effusivity of the wall material, see [4]. In the present model, this expression is used even when q does vary with time. This appears to be an acceptable approximation, as long as the actual value of q(t) is close to its time averaged value.

2.4 Solving

After substituting Q and $T_w(t)$ in Equation 3, one solves the resulting non-linear expression for T_{in} in a few iterations, and the cooling rate Q is evaluated with (1).

3. EXPERIMENTAL TEST

The model has been tested for the case of the cooling of office rooms (volume 50-100 m^3) by a single open window. The agreement was quite good [3].

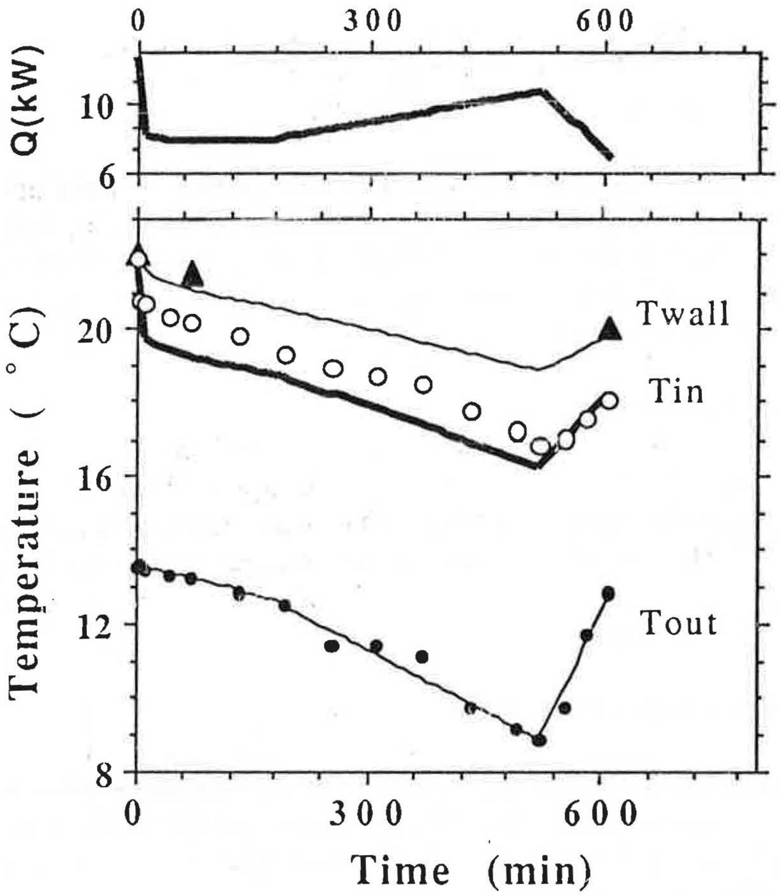

Figure 2. Free cooling at night of a $540m^3$ staircase from 10 p.m. to 8 a.m. The measured and calculated exhausted inside air temperature, T_{in}, the outside temperature, T_{out}, the wall temperature, T_{wall}, and the calculated heat loss rate, Q, as a function of ventilation time. $A_b=1m^2$, $A_t = 1.8\ m^2$, H=9.6m. The calculated energy loss over 10 h is 90kWh.

In this paper we consider another configuration, i.e. the high mass staircase of a three level office building (volume 4.5 by 8 by 15 = 540m^3) with ventilation openings at the roof and ground level (H≈10m, C2=1). The walls are characterized by an average thermal effusivity of b=1000 [J/ (m^2 K.s$^{0.5}$)] with a 20% uncertainty, as discussed in [4]. The total wall surface area of the staircase is S_i=700 m^2 with a 10% uncertainty. Measured are the temperatures of the in- and outflowing air, and of the wall as a function of time.

Figure 2 shows an example of freecooling at night lasting from 10 p.m. to 8 a.m. The openings are A_b=1 and A_t=1.8m^2. Are plotted both the calculated and measured temperature of the outflowing air (T_{in}), the measured outdoor temperature and a few values of the wall temperature as a function of ventilation time. The heat loss rate Q is given in the upper part of the plot. The total cooling energy calculated over the 10 hour period is 90kWh.

The calculated exhaust air temperature, T_{in}, is about one degree too low which is nearly 10% of T_{in}-T_{out} and Q is therefore underestimated by up to 15%, showing the kind of accuracy we can expect from such a simplified model.

We note that both the inside and outside air temperature lowered at about the same rate, and the ventilation rate, being almost constant over the night, was 10 air changes per hour.

4. DISCUSSION

The simplified model has two main parameters which determine the cooling potential (i) the active heat exchanging wall surface area C_2S_i and (ii) the thermal effusivity b of the wall material, characterizing the rate at which the inside surface temperature decreases.

The cooling rate increases with ventilation rate but is not proportional to it as Equation (1) would suggest. This is because the inside air temperature decreases further below the wall surface temperature.

The influence of S_i on the relation between cooling and mass flow rate is simple when we take $\dot{m}$ and T_w constant. After elimination of T_{in}, Equations (1) and (3) give then the following expression for the heat loss

$$Q = \frac{C_p\, m\,(T_w - T_{out})}{1 + \dfrac{C_p\, m}{S_i\, h_c}} = \frac{S_i h_c\, CF\,(T_w - T_{out})}{1 + CF} \tag{5}$$

The maximum value for the heat loss $Q=S_ih_c(T_w - T_{out})$ is approached when the cooling factor $CF=C_p\dot{m}/S_ih_c$ is large. For CF=1, Q is at 50% of its maximum value, and to increase Q further one has to increase the mass flow $\dot{m}$ in a disproportionate way. In conclusion, for a given value of the wall surface area S_i, we estimate an air flow rate of about (S_i/200) m^3/s to correspond to CF=1. For the case of Figure 2, the air flow rate was nearly constant at 1.6 m^3/s and CF≈0.5.

To show the relative importance of wall area and material choice, we calculated the total energy loss over a 10 hours period, for a few values of S_i and b and varying initial inside-outside temperature difference, $T_w(t=0)$ -T_{out}. The outside temperature was set constant. The results are plotted in Figure 3. Thick curves correspond to a heavy building structure (e.g. concrete and masonry) and the thinner lines to a light-weight structure with a wall material with four times lower thermal effusivity (e.g. lightweight concrete or thick wood).

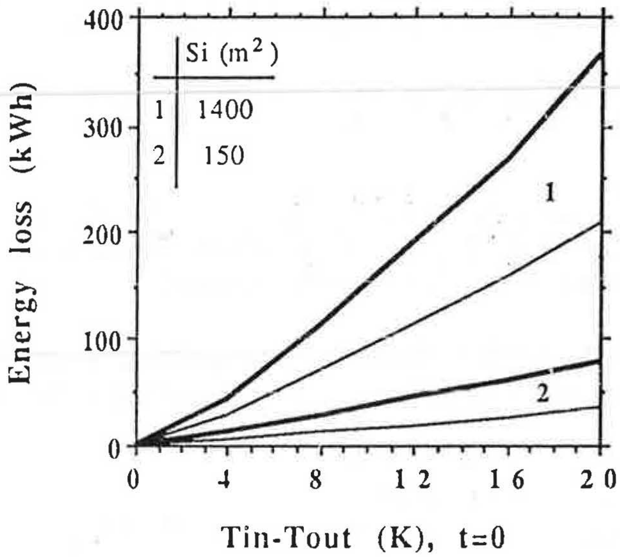

Figure 3. Simulation of energy loss by free cooling at night. Calculated is the total energy loss integrated over 10 hrs, for two values of S_i and b and as a function of initial inside-outside temperature difference, $T_w(t=0)$-T_{out}. T_{out} is assumed constant. Thick curves, b=1000 (e.g. concrete and masonry), thin curves b=250J/[$m^2Ks^{0.5}$] (e.g. lightweight concrete or thick wood); A_{in}=A_{out}=2m^2; H=10m.

5. CONCLUSION

This paper concerns a model to estimate the ventilative cooling potential of buildings. Tested on a high mass staircase cooled by stack ventilation, the time dependence of the air and wall temperatures is correctly explained without the need for adjusting coefficients. The free parameters of the model are the heat exchanging wall surface area and the thermal effusivity b of the wall material.

6. ACKNOWLEDGEMENT

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NOMENCLATURE

A_b = bottom opening area [m^2]
A_t = top opening area [m^2]
b = thermal effusivity, $(\lambda\rho c)^{0.5}$ [J/ (m^2 K.$s^{0.5}$)]
c = wall specific heat [J/kg.K]
C_p = specific heat of air at constant pressure [J/kg.K]
C_1 = flow discharge coefficient
C_2 = wall surface fraction for heat transfer
g = gravitational acceleration [m/s^2]
h_c = convective heat transfer coefficient [W/m^2·K]
H = vertical distance between centres of openings [m]
$\dot{m}$ = mass flow rate [kg/s]
q = density of heat flow rate [W/m^2]
Q = ventilation heat flow rate [W]
S_i = internal wall surface area [m^2]
t = time [s]
T = mean absolute air temperature[K]
T_{in} = inside air temperature [°C]
T_w = wall surface temperature [°C]
T_{out} = outside air temperature [°C]
ρ = density [kg/m^3]

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