EFFECT OF THERMAL MASS ON NATURAL VENTILATION IN A SINGLE-ZONE BUILDING WITH TWO OPENINGS

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ABSTRACT

This paper considers an ideal naturally ventilated building model that allows a theoretical study of the effect of thermal mass associating with the non-linear coupling between the airflow rate and the indoor air temperature. The thermal mass number and the convective heat transfer air change parameter are suggested to account for the effect of thermal mass heat storage and convective heat transfer at the thermal mass surfaces. The new thermal mass number measures the capacity of heat storage, rather than the amount of thermal mass. From the numerical analysis, the maximum indoor air temperature phase shift induced by the direct outdoor air supply without control was shown to be 6 hours. The use of thermal mass should be optimized, as excessive thermal mass is not beneficial when the phase shift approaches to its maximum value.

KEYWORDS

Thermal mass, natural ventilation, periodic heat flow, thermal coupling, buildings, passive design

NOMENCLATURE

A_b	area of the bottom opening 'b', m^2
A_M	effective heat transfer area of the thermal mass, m^2
A_t	area of the top opening 't', m^2
A^*	effective opening area of a building, m^2
B	buoyancy flux, m^4/s^3
C_d	discharge coefficient for the two openings
C_M	heat capacity of the thermal mass, J/kg ^oC
C_p	heat capacity of air, J/kg ^oC
E	effective total heat power, W
g	acceleration of gravity, m/s^2
h	height between two vertical openings 't' and 'b', m
h_M	convective heat transfer coefficient at surface of thermal mass materials, W/m^2 ^oC
M	mass of thermal mass, kg
q	ventilation flow rate, m^3/s
t	time, hr
T_i	indoor air temperature, K
T_M	thermal mass temperature, K
T_o	outdoor air temperature, K
$\tilde{T}_o$	mean outdoor air temperature, K
$\Delta\tilde{T}_o$	amplitude of fluctuation of outdoor air temperature, K

Greek symbols

α	buoyancy air change parameter, m^3/s
φ	thermal mass number
θ	outdoor temperature fluctuation air change parameter, m^3/s
θ_E	heat source induced air temperature parameter, K
q_H	convective heat transfer air change parameter, m^3/s
ρ	air density, kg/m^3
w	frequency of outdoor temperature variation, hr^{-1}

Subscripts

b	bottom opening
i	indoor
M	thermal mass
o	outdoor
t	top opening

1. INTRODUCTION

Thermal mass can be used to reduce energy consumption of mechanical cooling and heating systems in buildings. Thermal mass stores heat in both the building envelope materials and the interior mass such as partitions, ceiling and floor to reduce cooling load in summer and satisfy partly the heating requirement in winter by shifting the load from daytime to nighttime. The main focus of this paper is to study theoretically and numerically the non-linear coupling between ventilation and *internal* thermal mass in naturally ventilated buildings.

Balaras (1996) reviewed a large number of the previous theoretical, computational and experimental studies on thermal mass effects in buildings. Sixteen different parameters were found for describing the thermal mass effects, such as the admittance factor (Burberry, 1983), the effective heat storage (Mathews et al., 1991) and the total thermal time constant (Givoni, 1981). Classical texts in analytical heat conduction such as Carslaw and Jaeger (1959) have found their use in buildings through the well-known work such as Danter (1960).

Thermal mass is effective for dampening the wide range temperature fluctuation from the outdoor and maintaining the indoor air temperature within a comfortable range (Asan and Sancaktar, 1998; Maloney et al., 1982; Threlkeld, 1970). A good example of combined use of ventilation and thermal mass is the so-called night ventilation technique (Blondeau et al., 1997; van der Maas and Roulet, 1991). Night ventilation can be either natural or mechanical. When it is mechanical, the analyses are much simpler than when it is naturally ventilated. Kammerud et al. (1984) presented a study of the effect of ventilation cooling for a group of residential buildings. Their analyses assumed a fan-forced ventilation rather than natural ventilation. For stack-driven natural ventilation, the ventilation flow rate and the temperature difference between indoor and outdoor air are interdependent. Ventilation flow rate and indoor air temperature are coupled in a non-linear manner. Almost all existing studies have treated the problems as linear systems. Van der Maas and Roulet (1991) developed a simple dynamic model that couples airflow, heat transfer and a thermal model for the wall.

The present work identifies an ideal building model that allows a theoretical study of the effect of thermal mass. Two key thermal mass parameters are identified. Effect of thermal mass, ventilation flow rate, convective heat transfer at the thermal mass surface and the total heat gain in the building are investigated. With some modification, the present analyses are also applicable to situations when phase change materials are used for thermal storage.

2 A SIMPLE BUILDING MODEL WITH STACK-DRIVEN VENTILATION

We consider an ideal building model with two openings (Figure 1). The building is ventilated by stack force. The fully mixed assumption is used here because it leads to relatively simple equations, which nonetheless displays interesting behaviour and because this assumption is used in the simpler treatments of natural ventilation of multi-zone buildings. The building envelope is perfectly insulated and due to the uniformity of indoor air temperature, the thermal radiation between the room surfaces do not exist. All heat gain and heat generation in the building can be lumped into one heat source term, E. The temperature distribution in the thermal mass materials is also assumed to be uniform. We assume that the outdoor temperature can be expressed as a sinusoidal wave with a period of 24 hours.

$$T_o = \tilde{T}_o + \Delta\tilde{T}_o \sin(\omega t) \tag{1}$$

Let us adopt the convention that upward ventilation flows are positive and downward flows are negative. The flow rate is given, according to Li and Delsante (2001), by

$$q = C_d A^* \operatorname{sgn}(T_i - T_o)\sqrt{\left|2gh(T_i - T_o)/\tilde{T}_o\right|} \quad \text{or} \quad q|q| = (C_d A^*)^2 \cdot 2gh(T_i - T_o)\Big/\tilde{T}_o \tag{2}$$

where $A^* = A_t A_b \Big/ \sqrt{A_t^2 + A_b^2}$ is the effective area. Noting that the indoor temperature can be either higher or lower than the outdoor temperature. The positive sign is used in Eqn. 2 if indoor air temperature is higher than outdoor temperature, while the negative sign is used if the indoor air is cooler.

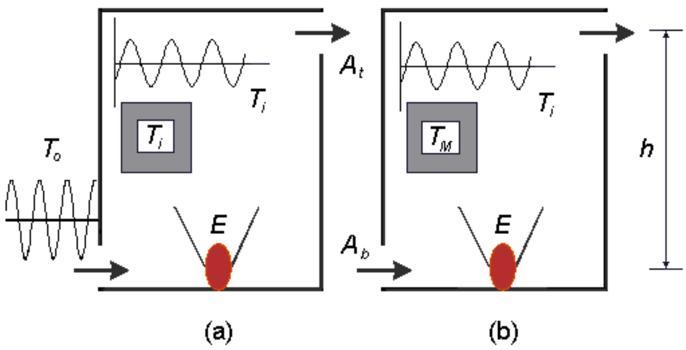


Figure 1: A simple two-opening one-zone building model with periodic outdoor air temperature variation. The shaded area represents the thermal mass. (a) The thermal mass is in equilibrium with the room air. (b) The thermal mass is not in equilibrium with the room air.

2.1 The thermal mass is in equilibrium with the room air

The heat balance equation for the building is

$$\omega M C_M \frac{\partial T_i}{\partial(\omega t)} + \rho C_p |q| (T_i - T_o) = E \tag{3}$$

Combining Eqn. 1-3 we obtain the following governing equation for the ventilation flow rate,

$$2\varphi\theta|q|\frac{\partial q}{\partial(\omega t)} = -q^3 + 2\alpha^3[1 - \varphi\cos(\omega t)] \tag{4}$$

where $\alpha = (C_d A^*)^{2/3}(Bh)^{1/3}$, $B = (Eg/\rho C_p \tilde{T}_o)$, $\theta = (E/\rho C_p \Delta\tilde{T}_o)$ and $\varphi = (MC_M \omega \Delta\tilde{T}_o / E)$. Similarly, we can obtain the governing equation for indoor air temperature T_i from Eqn. 1-3

$$\frac{\varphi}{\sqrt[3]{2}}\left(\frac{\theta}{\alpha}\right)\frac{\partial Y}{\partial(\omega t)} + Y|Y|^{1/2} = 1 - \varphi\cos(\omega t) \tag{5}$$

where $Y = (T_i - T_o)/\theta_E$ and $\theta_E = [E^2 \tilde{T}_o / 2gh(C_d A^*)^2 \rho^2 C_p{}^2]^{1/3}$. The fourth-order Runge-Kutta method was used to solve Eqn. 4 and Eqn. 5. Figure 2a shows the variations of the phase shifts for both the flow rate and the indoor temperature compared to the outdoor air temperature. The phase shift of the airflow rate is bounded by 6 and 12 hours and the phase shift for the indoor air temperature ranges from 0 to 6 hours. For both phase shifts, the rates of increase are initially very high and then gradually decrease until the phase shifts become constants. This indicates that the use of thermal mass should be optimized, as excessive thermal mass is not beneficial when the phase shift approaches to its maximum value. The effect of the ratio θ/α on the phase shift variation is also shown in Figure 2a. The higher the θ/α value, the greater is the phase shift for the same thermal mass number. Figure 2b shows the changes of the ratio of indoor temperature fluctuation to the outdoor temperature fluctuation with respect to the thermal mass number φ. The ratio decreases from 1 to 0 as the thermal mass parameter increases from zero to infinity for all θ/α values. When the effect of thermal mass is absent, the indoor air temperature fluctuates at the same amplitude as the outdoor temperature. As the effect of thermal mass is very large (approaching infinity), the outdoor air temperature fluctuation is dampened completely. These are the obvious physical features of the system.

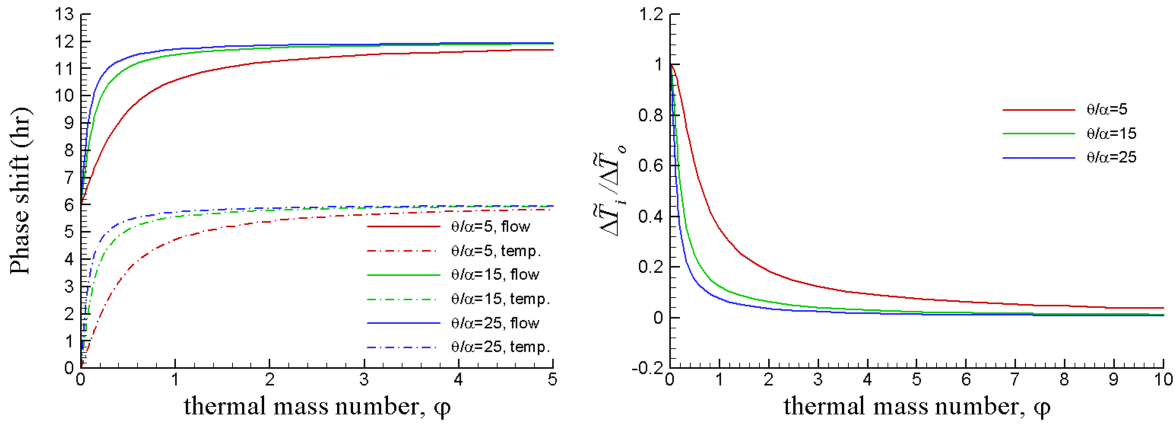


Figure 2: Numerical results for the case when $T_M = T_i$. (a) Phase shifts for both the ventilation flow rate (solid lines) and the indoor air temperature (dash-dotted lines). (b) The ratio of indoor air temperature fluctuation to outdoor temperature fluctuation.

2.2 The thermal mass is not in equilibrium with the room air

There are two heat balance equations, one for the room air (Eqn. 6) and one for the thermal mass (Eqn. 7).

$$\rho C_p |q| (T_o - T_i) + h_M A_M (T_M - T_i) + E = 0 \tag{6}$$

$$MC_M \frac{\partial T_M}{\partial t} + h_M A_M (T_M - T_i) = 0 \tag{7}$$

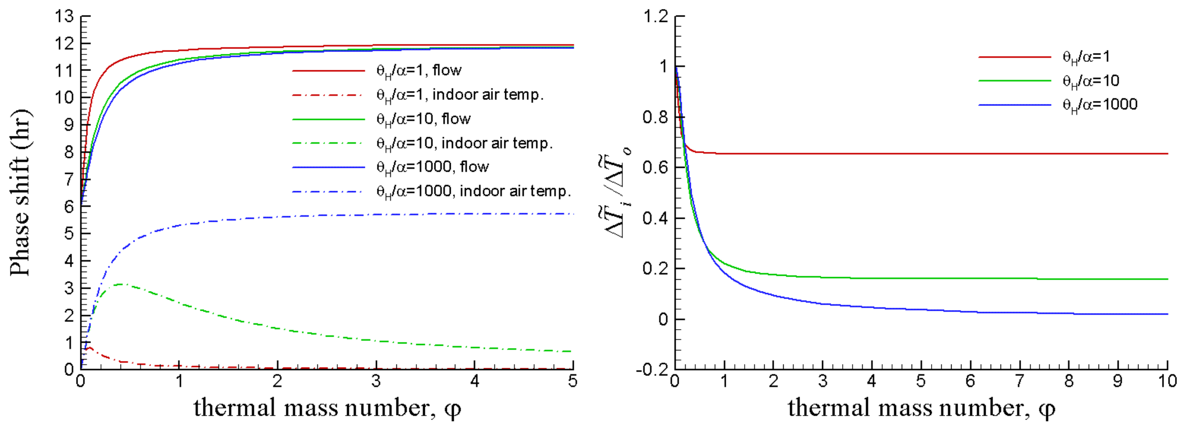
Using Eqn. 6-7 and Eqn. 1-2, and after simplification, we obtain

$$\varphi\left(\frac{\theta}{\alpha}\right)\left[\frac{1}{\sqrt[3]{2}}\left(\frac{\theta_H}{\alpha}\right)+\frac{3}{2}|Y|^{1/2}\right]\frac{\partial Y}{\partial(\omega t)}+\left(\frac{\theta_H}{\alpha}\right)Y|Y|^{1/2}=\left(\frac{\theta_H}{\alpha}\right)\left[1-\varphi\cos(\omega t)\right] \tag{8}$$

where $Y = (T_i - T_o)/\theta_E$, and $\theta_H = (h_M A_M / \rho C_p)$. As the convective heat transfer air change parameter θ_H goes to zero, there is no thermal link between the thermal mass and the air. The indoor air temperature is then in phase with the outdoor temperature. Alternatively, we can obtain the governing equation for the flow rate q

$$\varphi\left(\frac{\theta}{\alpha}\right)\left[\frac{3}{2}\left(\frac{q}{\alpha}\right)^2+\left(\frac{\theta_H}{\alpha}\right)\left|\frac{q}{\alpha}\right|\right]\frac{\partial\left(q/\alpha\right)}{\partial(\omega t)}+\frac{1}{2}\left(\frac{\theta_H}{\alpha}\right)\left(\frac{q}{\alpha}\right)^3=\left(\frac{\theta_H}{\alpha}\right)\left[1-\varphi\cos(\omega t)\right] \qquad (9)$$

Figure 3a shows the changes of phase shifts for both the airflow rate and the indoor air temperature as the thermal mass number changes. For simplicity, a fixed value of 10 for θ/α was chosen, and it was observed that the results for other values of θ/α were similar. Results for different normalized convective heat transfer air change parameter θ_H/α are shown. As θ_H/α is greater than 1000, the curves approach to that for the situation as described in Section 2.1. However, a smaller value of θ_H/α reduces the phase shift significantly. This means that the convective heat transfer process predominates the thermal mass effect when the thermal mass is not in thermal equilibrium with the indoor air. In Figure 3b, the normalized indoor air temperature fluctuation is shown as a function of the thermal mass number for different values of θ_H/α. When θ_H/α is small (less than 10), the normalized indoor air temperature fluctuation is controlled by the convective heat transfer process at the thermal mass surfaces. Further increase of thermal mass would not reduce the fluctuation to zero. Again, this suggests the importance of the thermal interaction (convective heat transfer) between the mass and the indoor air.



(a) **(b)**

Figure 3: Numerical results for the case when $T_M \neq T_i$. (a) Phase shifts for both the ventilation flow rate (solid lines) and the indoor air temperature (dash-dotted lines). (b) The ratio of indoor air temperature fluctuation to outdoor temperature fluctuation.

3 CONCLUSIONS

It is possible to design a system with a predicted time lag in a naturally ventilated building using the simple methods derived in this paper. Unlike the periodic heat transfer through the building envelope which can introduce a large time lag for the indoor air temperature, we have shown here that the maximum indoor air temperature phase shift induced by internal thermal mass with direct outdoor air supply is 6 hours. The thermal mass number and the convective heat transfer air change parameter are suggested to account for the thermal mass heat storage and the convective heat transfer at the thermal mass surfaces. The new thermal mass number measures the capacity of heat storage, rather than the amount of thermal mass. By calculating the values of these parameters, engineers can estimate whether the thermal mass design would be suitable for the application. Appropriate amount of thermal mass should be used in passive building design, as addition of thermal mass beyond the optimum limit would not increase the phase shift of the system, nor enhance the dampening effect on temperature fluctuation. Convective heat transfer at the thermal mass surfaces is very important and its quantitative impact shows that the phase shift of the indoor air temperature can be reduced even the thermal mass number is increased if the convective heat transfer parameter is small. It is hoped that the present study will pave the way for further understanding of the thermal mass effect and develop design guidelines for natural ventilation and passive design of buildings.

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