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A semi-empirical model for studying the impact of thermal mass and cost-return analysis on mixed-mode ventilation in office buildings

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ABSTRACT

Mixed-mode ventilation that combines natural ventilation and mechanical ventilation has great potential to save cooling energy when compared to mechanical systems and is more reliable than natural ventilation systems. This paper presents a semi-empirical model for studying the impact of window opening area, insulation, and thermal mass on the cooling energy saving of mixed-mode ventilation for three office buildings in different types of US climates using EnergyPlus simulations. The results show that electricity use can be reduced by 6–91% depending on the climate. In addition to climate, thermal mass has a large impact on the performance of mixed-mode ventilation. This investigation developed a semi-empirical model to predict the impact of thermal mass on energy use, and optimized the thermal mass for maximum monetary return based on the model.

1. Introduction

In the United States, buildings consume about 40% of total primary energy [1], and the energy consumption of office buildings comprises about 10% of the total building energy usage [2]. Natural ventilation has great potential for reducing the energy consumption in buildings [3,4]. However, several studies have found that natural ventilation may not provide good thermal comfort during a certain time of year in many locations [5,6], especially for commercial buildings such as office buildings [7]. Moreover, natural ventilation may not be used when it is raining or too windy. A more reliable ventilation system is needed that can provide the same thermal comfort as a mechanical system and consume less energy. Mixed-mode ventilation that combines the natural and mechanical cooling modes is a potential solution.

The mixed-mode system uses the natural cooling mode when the outdoor climate is suitable. The mechanical mode is used as a backup when outdoor conditions are not favorable. This system can therefore save energy and provide better indoor air quality than a pure mechanical system [8,9]. The system can also provide better thermal comfort than a pure natural ventilation system [10,11]. Furthermore, this system is highly integrable and can be coupled with, for example, a night-cooling strategy to further reduce the energy consumption in buildings [12].

Although mixed-mode ventilation has great potential to reduce energy consumption and to improve indoor air quality, design optimization is still needed to ensure the optimal performance of this system. Most of the current research focuses on active optimization, namely, the use of an advanced control algorithm to achieve better performance. Some researchers have developed advanced automatic control strategies for mixed-mode ventilation [13,14]. They have deployed advanced control algorithms, such as a predictive algorithm with automatic windows and multiple sensors to control the natural ventilation mode and mechanical mode. Such a system, although it has the potential for significant energy saving, is expensive and may easily lead to fouling of the system. On the other hand, some researchers have focused on occupants' control of mixed-mode ventilation [15,16]. Control based on occupant behavior is more feasible because the majority of buildings require occupants' active interaction with window control when mixed-mode ventilation is used. However, because there are a number of uncertainties in occupant-based control, a deterministic solution is difficult to obtain and therefore to optimize [17,18].

The other type of design optimization is a passive approach: changing the building construction materials to achieve better performance. To date, few researchers have addressed this approach specifically for mixed-mode ventilation. However, there have been studies of passive building optimization for pure natural ventilation [19–21], in which the researchers identified the thermal mass as a very important factor. They found that an increase in thermal mass can reduce the peak temperature by 1–3 K for buildings using free-running natural ventilation with a night-cooling strategy. Because we could also use a night-cooling strategy for the natural

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Nomenclature

A	area
C	model constant
C_d	discharge coefficient
C_p	pressure coefficient
c_p	specific heat of thermal mass
d	thickness
E_{ME}	energy consumption by mechanical ventilation
E_{saving}	energy saving by mixed-mode ventilation
h	convective heat transfer coefficient
h_1	elevation of the lower edge of the window
h_2	elevation of the upper edge of the window
l	opening width
P	pressure
$\bar{Q}$	mean flow rate
$\bar{Q}_{in}$	mean inflow rate
$\bar{Q}_{out}$	mean outflow rate
T	temperature
t	building lifetime in years
$\bar{U}$	mean wind velocity
V_{mass}	total thermal mass
z	vertical location
z_0	vertical location of the neutral plane
ρ	density
τ	time constant of thermal mass

Subscript

i	indoor air
m	thermal mass
o	outdoor air
ref	reference value at 10 m

ventilation mode in mixed-mode ventilation, thermal mass could have a large impact on the energy performance in our investigation.

This study, therefore, focused on the passive approach to improving energy efficiency for mixed-mode ventilation. This investigation aimed to demonstrate the impact of several important building envelope factors, such as thermal mass, insulation, and window opening area, on mixed-mode ventilation performance. A cost-return analysis was conducted to find the optimal design for thermal mass in order to yield the maximum return for office buildings, taking into account the capital cost and the return from energy saving during the summer.

2. Research method

This study investigated buildings with mixed-mode ventilation in five different cities: Miami (Climate Zone 1, very hot and humid), Phoenix (Climate Zone 2, very hot and dry), Las Vegas (Climate Zone 3, hot and dry), San Francisco (Climate Zone 3, marine climate), and Philadelphia (Climate Zone 4, warm and humid) [22] using EnergyPlus simulations. Because mixed-mode ventilation has much larger cooling saving than heating saving in the US [23], this study focused on cooling performance. Therefore, cold climates were not studied, and the time period of the simulation was from May 1 to September 30.

This investigation studied typical office buildings of three different sizes. The smallest one had a floor area of 225 m^2, representing typical small office buildings in the US [1]. The medium one had a floor area of 600 m^2, which, according to Deru et al. [24], covers 70% of typical commercial buildings in the US. The largest building had a floor area of 1500 m^2 to provide a wider range of data. Buildings larger than 1500 m^2 are often uniquely designed and cannot be represented by one specific model [25]; thus, they are not included in this study.

Fig. 1 shows the various building zones as represented by different colors. For the 225 m^2 building, as shown in Fig. 1(a), this study used three zones. Each zone could be naturally ventilated because the building depth was small [6]. For the other two buildings, as depicted in Fig. 1(b), five zones were used, and only the four perimeter zones could be naturally ventilated because the core zone did not have direct exposure to outdoor air. Each zone was conditioned by a separate constant air volume (CAV) system to enable individual control [24]. The mechanical system was a packaged rooftop heat pump, and it was automatically sized according to the design day for each climate. Because humidity is a problem in some climates, both humidity and temperature were controlled.

Table 1 lists the detailed information for the building enclosure used in this study. The building envelope constructions were from the Online Building Component Library [26] and based on ASHRAE Standard 90.1 [22]. The baseline buildings had no thermal mass in the building envelope, and differing amounts of thermal mass were added to the building in order to study the impact of thermal mass on cooling energy use. For the baseline building which has no concrete in building envelope, the floor slab contained only carpet. Although this floor structure is not possible in reality, this configuration was used to make the thermal mass comparison more consistent. The insulation for the baseline buildings was based on ASHRAE Standard 90.1 [22] for small or medium office buildings. Additional insulation was added to non-baseline buildings to study its impact. ASHRAE Standard 90.1 [22] requires the glazing-to-wall ratio to be within 0–40%. This study chose a ratio of approximately 20% for each building. The operable window area for natural ventilation was assumed to be half of the total glazing area. The schedules and corresponding values for occupants, lighting, and electrical equipment were based on ASHRAE Standard 90.1 [22] for working and non-working hours [26]. A humidistat was used to avoid condensation when relative humidity was high by overcooling 2 K lower than the cooling setpoint. Natural ventilation would be used when the outdoor temperature was between 15 °C and 22 °C and the indoor temperature was higher than 19 °C during working hours. During non-working hours, natural ventilation would be used when the outdoor temperature was between 10 °C and 22 °C in order to utilize night cooling.

This study considered only single-sided ventilation because in typical office buildings, the interior doors between rooms are usually closed for privacy. Also, even though buildings may have operable windows on each side of the envelope, cross-ventilation is still difficult to realize because of the large depth of buildings or interior partitions. Moreover, single-sided ventilation would provide us with the baseline ventilation rate for the worst-case scenario, which is suitable for design analysis. A modified model that includes the effects of both wind and buoyancy, based on Wang and Chen [27], was used to predict the mean single-sided ventilation rate. The pressure difference between the indoor space and outdoor environment at height z along the opening was calculated based on the stack and wind pressure difference across the opening, using the following equation:

$$\Delta\bar{P}(z) = \frac{1}{2}\rho_o C_p \frac{\bar{U}^2}{{z_{ref}}^{2/7}}(z^{2/7} - {z_0}^{2/7}) - \rho_i g(z - z_0)\frac{T_i - T_o}{T_o} \quad (1)$$

The neutral level, z_0, is an additional unknown which can be calculated from the mass balance equation between the incoming and outgoing ventilation rates through the opening as:

$$\bar{Q} = \bar{Q}_{in} = \bar{Q}_{out} \quad (2)$$