Topic 2. Ventilation and Sustainable Development

Technical Feasibility Study of Hybrid Ventilation for a High-rise Office Building in Shanghai

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ABSTRACT In this paper, a study to evaluate the feasibility of implementing natural ventilation using operable windows in an airtight high-rise office building, which is located in Shanghai (China), is presented. In this building, the natural ventilation would work collaboratively with the mechanical cooling/ventilation system. In order to help the building operator to decide whether the natural ventilation can be used, a simple rule based on the outdoor air parameters such as wind speed, air temperature and humidity is developed according to the following studies: firstly the outdoor wind environment simulation is performed using CFD (Computation Fluid Dynamics) method to obtain the flow rate of the air into the building under certain wind speed, based on which the cooling load can be calculated; then the calculated cooling load is treated as input variable for simulation of the mechanical cooling/ventilation system to gain the supply air temperature; lastly the supply air temperature is used in the indoor environment simulation to yield the indoor air temperature distribution pattern, based on which the indoor thermal comfort can be assessed. The results show that it is possible to employ natural ventilation without sacrifice of thermal comfort in this building when the outdoor air parameter meets certain requirement.

Keywords: Hybrid ventilation, High-rise office building, CFD, Analytical method

INTRODUCTION

Natural ventilation (NV) is now considered necessary primarily for sanitary purposes in office building which is suffering from increasing indoor pollutant level due to higher density of occupant, equipment and furniture in modern office buildings. However, traditionally NV has been only applied in low-rise buildings (Heiselberg, 2002) because increasing wind speeds associated with building height may be too high for comfort and would cause problems such as paper flying around. A possible solution to this problem is to employ hybrid ventilation system which consists of the natural ventilation and mechanical cooling/ventilation system. However, since the natural ventilation is mainly driven by the wind which changes frequently both in speed and direction while the indoor thermal system is typical non-linear system which would be affected not only by the cooling load but also the cooling system, it is hard to evaluate indoor thermal comfort level under the hybrid system.

In the previous scientific research, many authors have focused on the relationship of NV performance and indoor environment (Emmerich et al., 2005; Luo et al., 2007; Zhai et al., 2011). To help assess and optimize hybrid ventilated buildings, several methods are available to analyze air flows and thermal environment in hybrid ventilation systems, such as simple analytical method and computational fluid dynamics (CFD) method (Heiselberg, 2002), among which CFD method is used most extensively. Besides, CFD is usually found in ventilation studies (Tan et al., 2005; Ji et al., 2009; Cheung et al., 2011).

CFD simulation can provide information of thermal environment and contaminant in detail. Recently CFD has been used as a relatively reliable tool for thermal comfort and indoor air quality evaluation. Nevertheless, the application of CFD is limited in real building design because it requires very detailed boundary condition data as well as parameters such as pressure and temperature of inlet/outlet air in the control volume, which are often unavailable for real design. Besides, the excessive computer resources and the long running time are usually unaccepted for actual cases. In comparison, simple analytical method (Tan et al., 2005) takes a building as one homogeneous node. Due to its “well-mixed” assumption, less input information is required for calculation. Additionally, it is easily understood in terms of problem definition. Compared with the CFD method, simple analytical method needs less running time and relatively lower requirement for computer hardware. However, no detailed air temperature and airflow distributions within the thermal zone could be provided.

Therefore, both CFD simulation and simple analytical method have limitations, which should be considered in the method selection for certain problems. In this paper, a new analytical method (combined method), which integrates the CFD simulation with simple analytical method to make a balance between simpleness and reliability, is proposed. A case study of a high-rise office building in Shanghai is also carried out to elaborate the application of such method and to analyze the feasibility of hybrid ventilation. The study consists of two parts: 1) Calculation of outdoor air flow rate through windows on different facades; 2) Simulation of indoor air temperature, humidity and velocity distribution with opening windows.

METHODOLOGIES

According to Figure 1, the analysis process of combined method could be divided into three main stages, including HVAC (Heating, Ventilation and Air Conditioning) load increment calculation, HVAC system simulation and indoor thermal environment analysis.

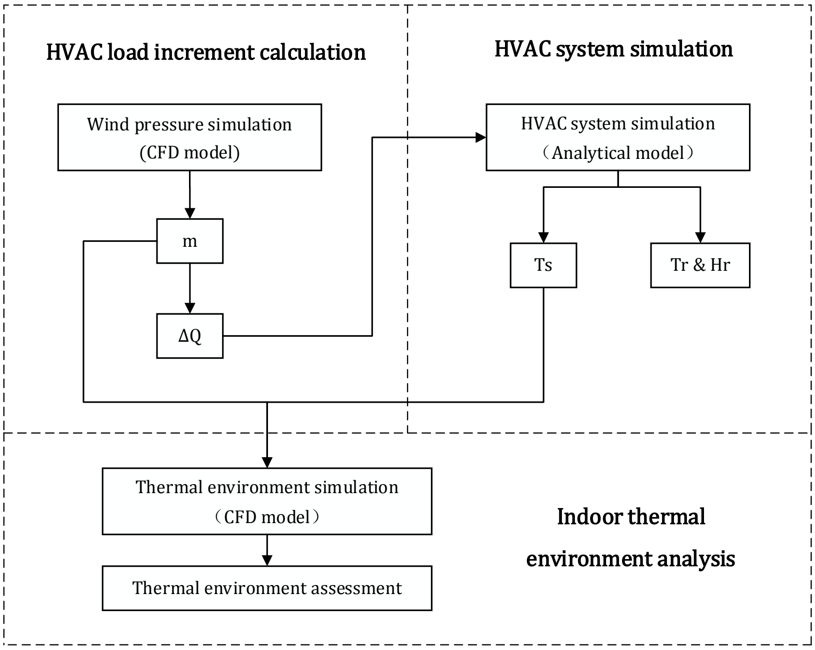


Figure 1. Combined method solution process.

HVAC Load Increment Calculation

When natural ventilation operates, the hot and wet outdoor air will be inducted into thermal zone, which may increase the air-conditioning loads, as described (Cai et al., 1999) by:

$$\Delta Q = m(h_o - h_r) \tag{1}$$

$$m = A\sqrt{\frac{2\rho|P_e - P_r|}{\varepsilon}} \tag{2}$$

We assume that P_r will keep constant as set point for the HVAC system (normally 10Pa for office buildings), then the only unknown variant for Equation (2) is P_e. To get the value of P_e, a CFD simulation was made, as shown in Figure 2. The RNG k-ε model is used in the CFD simulation because of its steady and easily convergent advantages (Tan et al., 2005). In Figure 2, a control volume is used to model the wind field surrounding the target building. Under real conditions, the outside wind is very difficult to simulate due to its fluctuation. According to the local predominant wind direction, the inlet and outlet building boundary could be set and the wind velocity for wind inlet could be described (Heiselberg, 2002) by:

$$U = U_0\left(\frac{H}{H_0}\right)^n \tag{3}$$

where U_0 is the wind velocity at reference height H_0, which is 10m in this study. The constant n, depending on the roughness of the ground, was 0.33 in this example. With the above model, the wind pressure distribution across the outside surface could be obtained, thus HVAC load increment could be calculated by Equations (1) and (2).

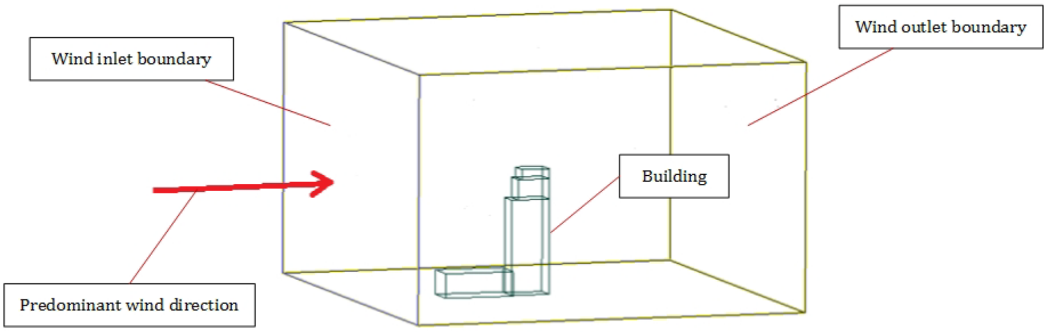


Figure 2. Wind pressure simulation.

HVAC System Simulation

Component-based method is employed to simulate the dynamic characteristics of thermal zone combined with HVAC system. The output of one module would be treated as the input for another module, which makes all modules connected with each other as an information loop. In this HVAC system simulation, the system was divided into two separate modules: thermal zone module and AC system module, as shown in Figure 3.

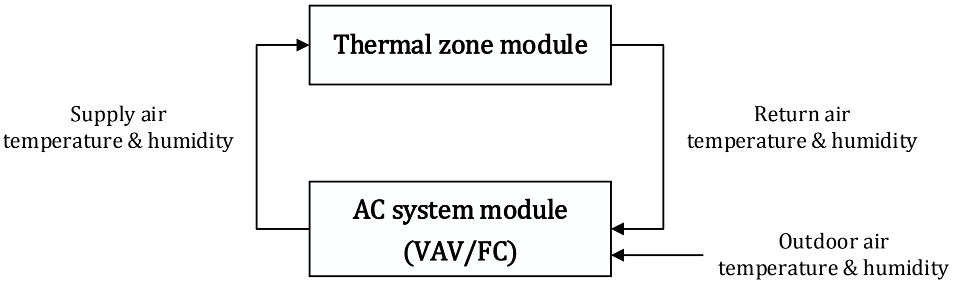


Figure 3. HVAC system simulation.

According to the First Law of Thermodynamics and the Law of Conservation of Mass, the heat and humidity balance of the thermal zone could be described as follows:

$$\rho V C_p \frac{dT_r}{dt} = Q_{con} + K\left(T_o - T_r\right) + C_p m_s\left(T_s - T_r\right) + C_p m\left(T_o - T_r\right) \tag{4}$$

$$\rho V \frac{dD_r}{dt} = D_{con} + m_s D_s + C_p m D_o - \left(m_s + m\right) D_r \tag{5}$$

Thus, T_r and D_r can be solved in terms of T_o, T_s, D_o and D_s as:

$$T_r = f\left(T_o, T_s, m\right),\ \ D_r = f\left(D_o, D_s, m\right) \tag{6}$$

Likewise, the heat and humidity balance of AC system can be illustrated by:

$$m_s h_s - m_r\left[C_p T_r + \left(C_{pw} T_r + I\right) D_r\right] - m_f D_o\left(C_{pw} T_o + I\right) - Q_{capacity} = 0 \tag{7}$$

($m_f = 0$ if the fan coil system is used) where h_s and D_s is given by:

$$h_s = C_p T_s + \left(C_{pw} T_s + I\right) D_s \tag{8}$$

$$D_s = a h_s^3 + b h_s^2 + c h_s + d \tag{9}$$

Thus, T_s and D_s can be solved in terms of T_o, T_r, D_o and D_r as:

$$T_s = f\left(T_o, T_r\right),\ \ D_s = f\left(D_o, D_r\right) \tag{10}$$

Based on the analysis above, it could be found that Equations (6) and (10) are coupled and could be solved simultaneously using the iterative algorithm.

Indoor Thermal Environment Analysis

In this stage, one or a few thermal zones were selected for further and detailed analyses for thermal comfort assessment with the CFD simulation, as shown in Figure 4. The boundary conditions for the CFD simulation are listed in Table 1. According to output parameter (T_r), the effect of natural ventilation could be evaluated for decision-maker's reference.

Table 1. Relevant parameters for CFD simulation.

Parameter	Specification
m	Provided by the HVAC load increment calculation stage
T_o	Determined by local climate data
T_s	Provided by the HVAC system simulation stage
T_r	The output data for this CFD simulation

In summary, through the three stages above, a feasible study could be finished to make more specific illustrations, and then a case study was performed.

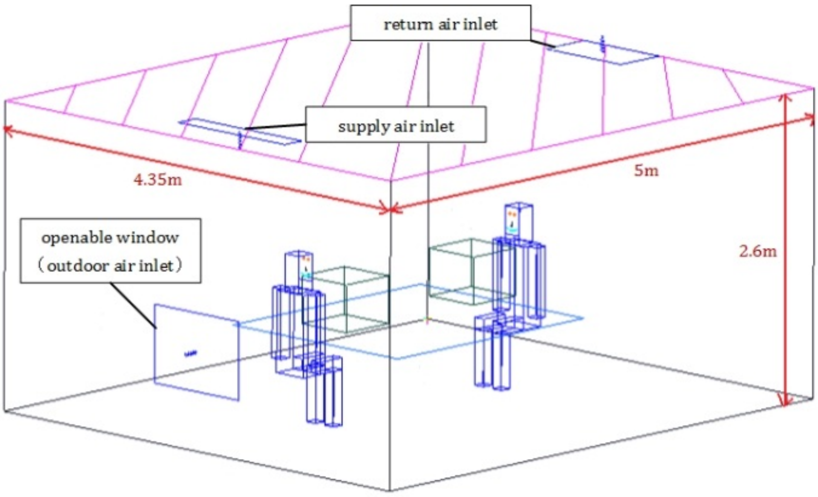


Figure 4. Indoor thermal environment analysis.

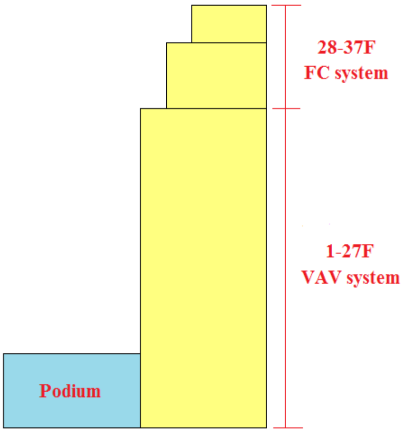


Figure 5. Shanghai Future Tower.

CASE STUDY

Shanghai Future Tower is located in Lujiazui Financial and Trade Zone, covering a gross area of 75000m^2 with 140m high, which has 37 floors on the ground and 3 floors underground. This high-rise building has two separate HAVC systems, as shown in Figure 5. Shanghai Future Tower has a podium on 1F-8F, whose sizes are 78.3m×39.75m×27.3m. The sizes of standard offices are 43.5m×26.1m×106.05m on 1F-27F, 36.2m×26.1m×22.5m on 28F-34F and 31.8m×26.1m×11.25m on 35F-37F. The building model is established based on these information and the wind field surrounding the building is also simulated, as shown in Figure 2. The dimensions of the computational domain for wind field are 450m×500m×280m (Empirically the dimensions of the computational domain are about 10 times larger than the dimensions of the building model).

The standard offices are located on the 9th to the 27th floor, with an area of 1000m^2 each floor. An AHU (Air Handling Unit) is installed every floor (ventilation rate is 6.94m^3/s and cooling capacity is 145.8kW). The FCU (Fan Coil Unit) system is applied on the 28th to the 37th floor. The conditioned area is 760m^2 every floor from the 28th to the 34th floor with 23 fan coils (cooling capacity is 8.24kW and heating capacity is 13.55kW), as well as 629m^2 every floor from the 35th to the 37th floor with 19 fan coils (cooling capacity is 8.24kW and heating capacity is 13.55kW).

Though the total fresh air supply in Shanghai Future Tower reaches the correlative standard (GB50019-2003), certain floor areas lack fresh air for the reason that outdoor air cannot be distributed equally to every space without control system. Thus these areas need to open windows to induce natural ventilation. The negative effects from opening windows are obvious: firstly, the ventilation increases the indoor loads and cannot ensure temperature and humidity comfort level when outdoor air enthalpy is high; secondly, people in the office would feel uncomfortable when the ventilation rate is high. Therefore, the indoor air temperature, humidity as well as air velocity should be predicted when the window is open if the capacity of HVAC system keeps constant.

RESULTS AND DISCUSSION

For the simulated VAV system, the highest office in the system (27F) was selected as subject for investigation considering the worst situation after opening the windows. Similarly, the highest office in the FCU system (37F) was studied for the simulated FCU system. Concrete analysis of the problems above is illustrated in Section 2. The simulation results of indoor air temperature and humidity are shown in Figures 6-9.

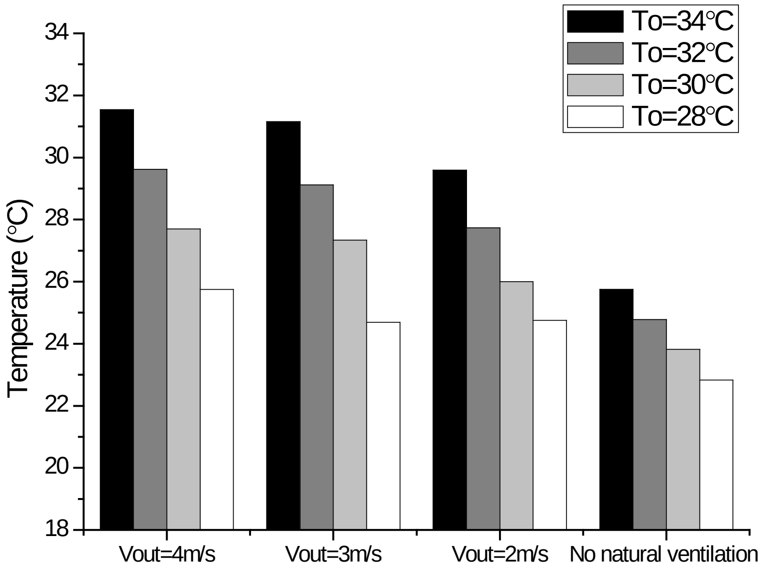


Figure 6. The maximum average indoor air temperature for VAV area.

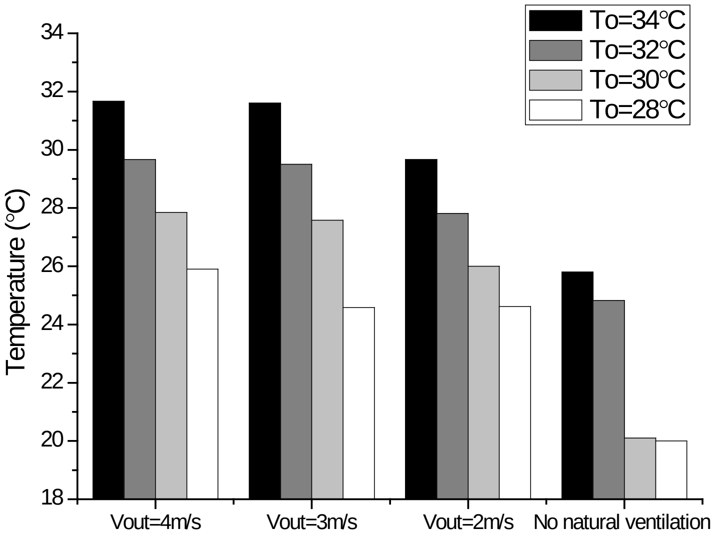


Figure 7. The maximum average indoor air temperature for FC area.

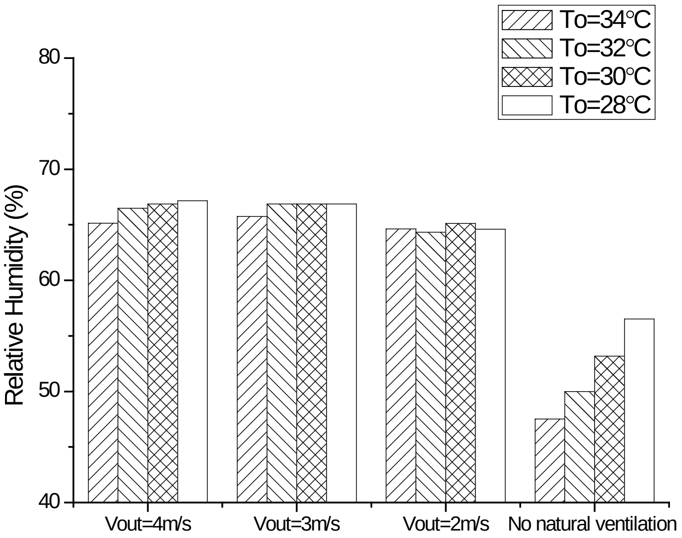


Figure 8. The average indoor air humidity for VAV area.

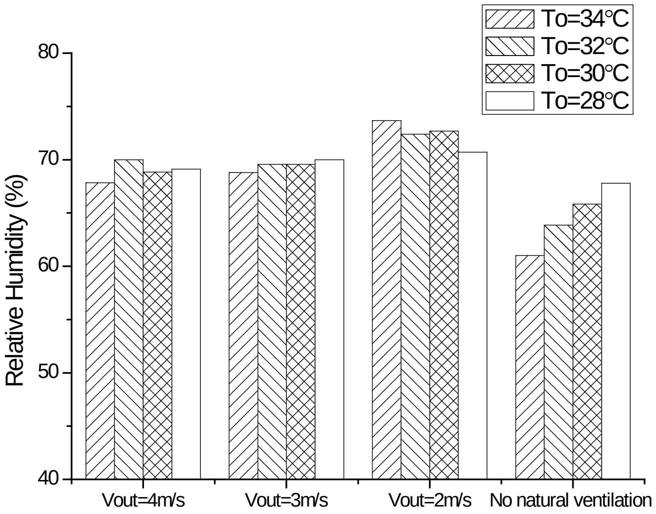
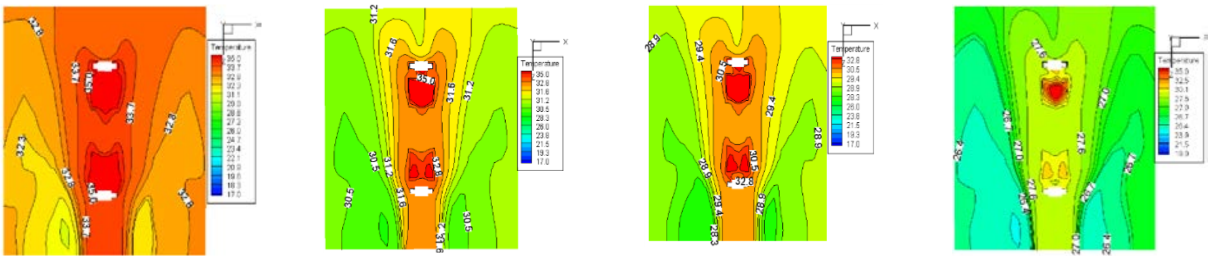


Figure 9. The average indoor air humidity for FC area.

From Figure 6 to Figure 9, it comes to the conclusion that: When the windows are open, the minimum difference between the indoor and outdoor air temperature is around 2°C, which would increase as the outdoor air velocity decreases. For example, if the outdoor air temperature is 30°C, the difference between the indoor and outdoor air temperature will be 2°C when the outdoor air velocity is 4m/s, while such difference will increase to 3.3°C when the outdoor air velocity is 2m/s. Therefore, if we select 26°C as the upper limit of indoor comfort temperature and the outdoor relative humidity is maintained under 70% with the common wind velocity in Shanghai (2-4m/s), the window should not open when the outdoor air temperature is higher than 28°C.

To show the effect of outdoor air temperature and velocity on the temperature and indoor air velocity distribution, the VAV area in the worst situation is investigated as an example. The indoor temperature plant distributions in VAV area under each condition are shown in Figure 10 (See Figure 4 for the CFD model diagram). The results show that, when the outdoor air temperature is 34°C, 32°C or 30°C, the indoor air temperature will be usually too high, resulting in a bad thermal environment. While the indoor air temperature may approximately range within the thermal comfort zone if the outdoor air temperature is 28°C. Thus, the CFD simulation results are correspondent with the conclusion derived from Figures 6-9. Also, we could find that the temperature drops across the direction of the outdoor wind into the conditioned zone.

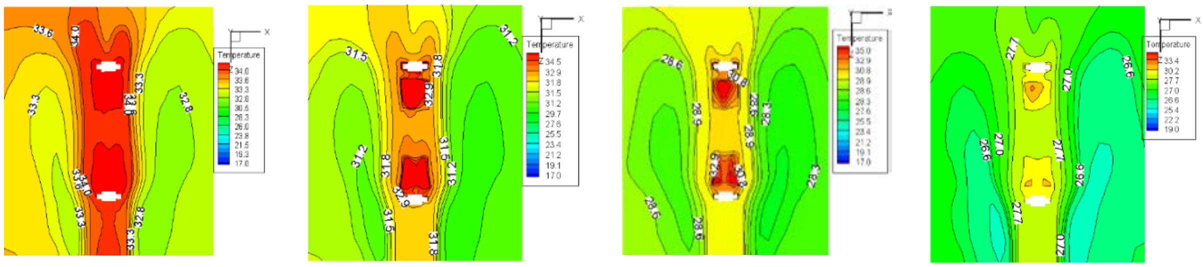


Y=0.8m, T_o=34°C, V_o=4m/s

Y=0.8m, T_o=32°C, V_o=4m/s

Y=0.8m, T_o=30°C, V_o=4m/s

Y=0.8m, T_o=28°C, V_o=4m/s

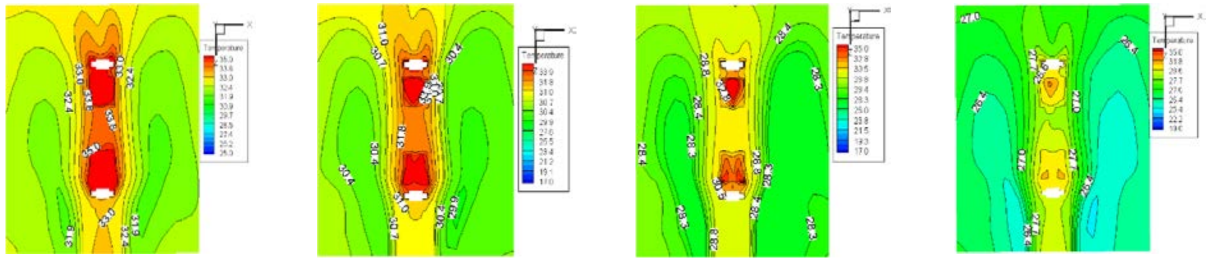


Y=0.8m, T_o=34°C, V_o=3m/s

Y=0.8m, T_o=32°C, V_o=3m/s

Y=0.8m, T_o=30°C, V_o=3m/s

Y=0.8m, T_o=28°C, V_o=3m/s



Y=0.8m, T_o=34°C, V_o=2m/s

Y=0.8m, T_o=32°C, V_o=2m/s

Y=0.8m, T_o=30°C, V_o=2m/s

Y=0.8m, T_o=28°C, V_o=2m/s

Figure 10. Indoor temperature plant distribution in VAV area (Y=0.8m).

CONCLUSIONS

The main purpose of this paper is to propose a combined method for predicting the hybrid ventilation in which natural ventilation assists the central HVAC system. Results from the case study show that: when the windows are open, the minimum difference between the indoor and outdoor air temperature is around 2°C, which would increase as the outdoor velocity decreases. If we select 26°C as the upper limit of indoor comfort temperature and the outdoor relative humidity is maintained under 70% with the common wind velocity in Shanghai (2-4m/s), the window should not open when the outdoor air temperature is higher than 28°C. Therefore, it is possible to employ natural ventilation without sacrifice of thermal comfort in this building when the outdoor air parameter meets certain requirement.

The combined method has several benefits: it takes less running time compared to pure CFD simulation while it obtains more detailed output results than simple analytical method. It is anticipated that this method could be used for further assessments of natural ventilation which would ultimately impel the application of hybrid ventilation in high-rise buildings. During the study, however, some assumptions are made to simplify the calculation. Furthermore, the impacts of neighboring tall structures on the wind and natural ventilation performance are not considered, which could influence the accuracy of the method to some extent. In the future, it is hoped that more experimental research or on-site measures could be performed to further validate this combined method.

NOMENCLATURE

A	across flow area for natural ventilation, m^2
a, b, c, d	coefficients of the empirical equation
C_p	air specific heat, kJ/(kg·K)
C_{pw}	water specific heat, kJ/(kg·K)
D_{con}	moisture load, hardly change with the temperatures
D_o, D_r, D_s	outdoor, indoor and supply air absolute humidity, kg/kg
H	building height, m
H_0	reference height, H_0=10m
h_o, h_r, h_s	enthalpy of outdoor, indoor and supply air, kJ/kg
I	latent heat of vaporization, kJ/kg
K	heat transfer coefficient, W/(m^2·K)
m	mass flow rate of natural ventilation, kg/s
m_f	mass flow rate of fresh air, kg/s
m_r, m_s	return and supply mass flow rate, kg/s
n	exponent factor for atmospheric boundary layer
P_e, P_r	ambient and indoor pressure, Pa
ΔQ	HVAC load increment due to natural ventilation, W
$Q_{capacity}$	rated refrigeration capacity of air handling unit, W
Q_{con}	cooling load, hardly change with the temperatures
T_o, T_r, T_s	outdoor, indoor and supply air temperature, °C
U, U_0	wind velocity at height H and H_0, m/s
V	thermal zone volume, m^3
ε	resistance factor
ρ	air density, kg/m^3

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