Title page

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Effects of key factors on the thermal insulation performance of the hollow block ventilated wall

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Abstract: Taking use of the air flowing through the cavities, the hollow block ventilated wall (HBVW) can be cooled down in summer and warmed in winter. The heat transferred between the outdoor and indoor environments as well can be removed. However, both simulation and experimental studies on the HBVW are not sufficient and factors that affect the thermal insulation performance of the HBVW needs to be identified. In this paper, frequency domain finite difference method (FDFD) and number of transfer units method (NTU) are adopted to build the coupled 3D heat transfer model to investigate the temperature distribution of wall surfaces and the exhaust airflow. Then experiments are carried out to validate the 3D model and comparisons prove that the model has considerable accuracy. In addition, both the numerical and experimental studies show that HBVW can significantly reduce the heat transferred from outdoor environment to indoor room through the ventilation in summer. The effects on the thermal insulation performance of the HBVW caused by influential factors, including the airflow velocity, inlet temperature of the airflow and the cavity size, are investigated based on the heat transfer model. Results show that the temperature of inner surface of the HBVW at four orientations is reduced and ranges from 24.8~27.0 °C during a typical summer day. The heat flux through the inner surface is reduced by 55.08%, 55.07%, 56.03% and 55.19% respectively for the east, west, south, and north HBVWs, indicating the energy saving potential of HBVW. The most critical factor is the cavity size. When the cavity dimension is constant, the influence of the airflow velocity on thermal insulation is much larger compared with the airflow temperature, and the suggested airflow rate is 0.8 m/s. Results show that a higher airflow rate, higher inlet airflow temperature and larger cavity size enable the HBVW achieves better thermal insulation performance.

Keyword: hollow block ventilated wall; thermal insulation; heat transfer; frequency domain finite difference; thermal comfort; energy saving

1. Introduction

Energy for residential and commercial use make up respectively about 20% and 18% of total energy consumption in 2017 [1], indicating that buildings account for great responsibility for energy shortage. Furthermore, it has been stated that 20-50% of the cooling and heating load is caused by building envelopes [2]. Improving the thermal insulation is important to reduce the energy consumption of buildings. Many researchers have done comprehensive work related to wall [3–5] and window [6–8], aiming to propose effective methods to reduce cooling and heating load resulting from the building envelope. One alternative is to decrease building energy consumption by adopting air layers within building envelops [9], which may include Trombe walls, internal hollow composite walls, double-skin façades, PV façades, multi-layer windows, and air flow window, etc. Ana Briga-Sá et al. [10] researched the Trombe wall in both analytical and experimental aspects and found out that more than 30 °C of external surface would be decreased when the wall was equipped with occlusion device, and switching off the ventilation openings leads to much more heat delay. Ahmed Faheem et al. [11] integrated a ventilated hollow core slabs with mPCM (micro-encapsulated phase change materials), presented numerical model which was validated by experiments and pointed out that applying mPCM enhanced the cooling potential in high thermal mass building. Souza et al. [12] established a test cell to collect temperature data of a double skin façade and results showed that double skin façade could reduce the indoor temperature. Studies related to these kinds of building envelops can be divided into two categories and their difference is that whether there is ventilation in the air layer. When there is no ventilation in the air layer, which can be regarded as steady, the thermal resistance of the wall is increased for the building envelope mainly due to low thermal conductivity of air. While there is ventilation in the air layer, which can be regarded as unsteady, natural or forced convective heat transfer would happen between the air layer and the building envelop, and then remove the heat transferred into the indoor room. The heat flux of façade without ventilation is much higher than the façade with ventilation. Moreover, the heat flux decreased by improving the airflow velocity in the ventilation façade [13].

The hollow block ventilated wall (HBVW) is a novel ventilation wall and different from the conventional ventilation wall with air layer. Due to the heat exchange between the airflow inside cavities and the wall before all the heat transferred through the wall, the thermal insulation performance of the wall is improved and then the cooling and heating loads are reduced. The cavities are composed with the construction of concrete hollow blocks, so no additional façade configuration is needed. The airflow sources can be the exhaust air of air conditioning system, tunnel wind, night breeze, and outdoor cool air [14]. The exhaust air can take away the heat stored in the structure in summer and warm the structure in winter. Practically, when the outdoor air temperature is lower than the indoor air in summer, the cool air (i.e., underground tunnel air, night breeze or at the place in temperate climate area) can be utilized to eliminate stored heat.

The conventional (non-ventilation) hollow block wall has been studied by numerical or experimental methods and combination of these two [15–17]. A mathematic model was built by Boukendil et al. [15] to describe the heat transfer of double hollow brick wall, using the finite difference method and SIMPLE algorithm to obtain heat transfer coefficient and heat flux. Their work indicated that there was a large energy saving potential of buildings by utilizing materials with low emissivity. They also showed that the thicker mortar would result in more heat transferred through the wall. Another type of fired hollow block with 29-row holes had been tested by Wu et al. [16] through a hot box experiment, which showed that the measured average U-value met well with the requirement of National Standard GB 50189-2005 [2]. Nevertheless, this type of hollow block need complicated manufacturing process due to much more holes. Zhang et al. [17] carried out both theoretical and experimental studies to investigate the thermal properties of a 3-row holes hollow block. Combining methods of TCHCM, FDM and PIM, they discussed comprehensive possible impact factors and concluded that it was better to reduce the thermal conductivity of block material and enhance its thermal capacity. Increasing the block thickness and hole rows can achieve better thermal performance.

Although there are many studies on the air layer ventilation wall and conventional (non-ventilation) hollow block wall, studies on HBVW are rarely reported. Our previous work has put forward a heat transfer model of HBVW built through frequency domain finite difference(FDFD) method, and analyzed the results in frequency domain under steady-state condition [14]. There is still lack of an effective dynamic heat transfer model of HBVW in time domain which can more intuitively describe the heat insulation performance and calculate the heat transfer between the airflow and wall body. The thermal performance of HBVW mainly depends on the parameters such as airflow rate, airflow temperature and the cavity size, it is necessary to fill the vacancy that the influence of these parameters on the heat transfer performance of the wall and its influence on the optimal design of the structure. This paper presents a coupled heat transfer model that can calculate the dynamic heat transfer happening on HBVW. The influence of three key factors on thermal performance will as well be analyzed so as to provide reasonable suggestion for improvement of heat insulation performance and optimal design for HBVW.

2. Description of the heat transfer model

A schematic diagram of HBVW is presented in Fig. 1, showing the specific path of airflow. As shown in Fig. 2, the heat transfer in the hollow block ventilated wall contains the following processes: the solar radiation on the outer wall, the long wave radiation heat exchange between the wall outer surface and surroundings, the convective heat transfer between the wall outer surface and outdoor air, the heat conduction of the wall, the convective heat transfer between the interior surfaces of the cavities and the airflow inside, the radiation heat exchange among the interior surfaces of the cavities, the convective heat transfer between the inner wall surface and indoor air, and the radiation heat exchange between the inner wall surface and other surfaces. In order to simplify the heat transfer calculation, instead of outdoor air temperature, the outdoor sol-air temperature is used as one of the boundary conditions to consider solar radiation, the long wave radiation heat exchange and convective heat transfer on outer surface. The radiation heat exchange between the wall inner surface and other surfaces is ignored. The heat transfer model includes two sub-models. One is the FDFD model used for calculating the temperature distribution along the thickness and length directions of the ventilated wall. The other is a steady state NTU model used for calculating the outlet temperature of airflow and the heat exchange between the airflow and wall mass. The heat transfer model is explained as follows:

(1) A small section of HBVW along height is concerned in this study. The temperatures of the airflow and wall mass along the height of the wall are assumed to be constant firstly. FDFD method is used to build the basic two-dimensional heat transfer model to analyze the heat transfer in the horizontal directions.

(2) Considering the airflow temperature variation inside the cavities along with the height of HBVW, the NTU model is adopted to calculate the outlet temperature of airflow. Then the coupled heat transfer model is established by integrating the FDFD model and the NTU model through the average temperature of the wall, and the heat exchange between the airflow and wall mass can be analyzed.

(3) Effects of influential factors on the thermal performance of HBVW are investigated using the coupled heat transfer model, including airflow velocity, airflow inlet temperature and cavity size.

2.1 FDFD model for heat transfer in the horizontal directions.

The frequency domain finite difference method is firstly utilized to develop the basic two-dimensional heat transfer model. The FDFD model is used to analyze the heat transfer in the horizontal directions of the wall from the outdoor environment to the indoor space. Fig. 3 shows the cross section diagram of HBVW.

Due to the symmetrical distribution of the cavities, the section circled by dotted line is chosen as the study object of the basic FDFD model in which the left and right borders are considered as adiabatic. The selected section is divided into 192 heat transfer cells as shown in Fig. 4. Each cell consists of one heat capacitor and 4 thermal resistors.

The heat balance equation of each single cell can be written as Eq. (1):

$$s_1(\theta_1-\theta_0)+s_2(\theta_2-\theta_0)+s_3(\theta_3-\theta_0)+s_4(\theta_4-\theta_0)=dxdy\rho c\frac{\partial\theta_0}{\partial t} \tag{1}$$

Where, θ is the plural temperature (^{o}C), s is the heat transfer factor between two nodes (W/(m^2·K)), ρ is the density of the hollow block (kg/m³), c is the specific heat of the hollow block (J/(kg·K)), t is the time of heat transfer (h), dx and dy are the length of two sides of each heat transfer cell (m). The value of s differs when the position of the heat transfer cell changes.

When the heat transfer cell is on the inner surface of the wall:

$$s_1=h_{in}\cdot dy \tag{2}$$

When the heat transfer cell is on the outside surface of the wall:

$$s_2=h_{out}\cdot dy \tag{3}$$

When the heat transfer cell is on the surfaces between each block:

$$s_3=s_4=0 \tag{4}$$

When the heat transfer cell is on the interior surfaces of cavity:

$$s_j=\begin{cases} h_{zj}\cdot dy & (j=1,2)\\ h_{zj}\cdot dx & (j=3,4)\end{cases} \tag{5}$$

When the heat transfer cell is not on aforementioned surfaces:

$$s_j=\begin{cases} \dfrac{dy_0}{dx_j/2\lambda_j+dx_0/2\lambda_0} & (j=1,2)\\ \dfrac{dx_0}{dy_j/2\lambda_j+dy_0/2\lambda_0} & (j=3,4)\end{cases} \tag{6}$$

Where, h_{in}, and h_{out} are the convective heat transfer coefficients of the inner and outer surfaces of HBVW. h_{zj} is the integrated heat transfer coefficient considering radiation and convection on each surface of the cavity.

The plural temperature with frequency ω can be described as Eq. (7), in which real and imaginary components are included:

$$\theta=T\,exp(i(\omega t+\varphi))=Texp(i\varphi)\,exp(i\omega t)=(u+iv)\exp(i\omega t) \tag{7}$$

To obtain the values of u and v, the Euler equation is introduced as follows:

$$exp(i\varphi) = (cos\varphi + isin\varphi) \tag{8}$$

Combining equation (1) and (7), the heat transfer function of each single cell can be written as Eq. (9):

$$\begin{bmatrix} s_1+s_2+s_3+s_4 & -pc\omega dxdy \\ pc\omega dxdy & s_1+s_2+s_3+s_4 \end{bmatrix} \begin{bmatrix} u_0 \\ v_0 \end{bmatrix} = \begin{bmatrix} s_1u_1+s_2u_2+s_3u_3+s_4u_4 \\ s_1v_1+s_2v_2+s_3v_3+s_4v_4 \end{bmatrix} \tag{9}$$

Uniting all the heat transfer functions of the cells, the large sparse matrix equation with 384 rows can be obtained as Eq. (10).

$$\text{diag}[A_1, A_2, \cdots, A_{192}][X_1, X_2, \cdots, X_{192}] = [B_1, B_2, \cdots, B_{192}] \tag{10}$$

Where, $A_k = \begin{bmatrix} \sum_{j\neq k}^{l} s_j & -pc\omega dxdy \\ pc\omega dxdy & \sum_{j\neq k}^{l} s_j \end{bmatrix}$, $X_k = \begin{bmatrix} u_k \\ v_k \end{bmatrix}$, $B_k = \begin{bmatrix} \sum_{j\neq k}^{l} s_j\, u_j \\ \sum_{j\neq k}^{l} s_j\, v_j \end{bmatrix}$, $k = 1,2,\cdots,192$. And k is the quantity of cells and l represents how many cells involved in the heat transfer with number j cell.

The matrix equation (10) can be expressed as the linear simultaneous equations shown in Eq. (11):

$$\begin{bmatrix} \sum_{j\neq 1}^{n} s_j & -dxdy\rho c\omega & & & & & & & \\ dxdy\rho c\omega & \sum_{j\neq 1}^{n} s_j & & & & & & & \\ & & \sum_{j\neq 2}^{n} s_j & -dxdy\rho c\omega & & & & & \\ & & dxdy\rho c\omega & \sum_{j\neq 2}^{n} s_j & & & & & \\ & & & & \ddots & & & & \\ & & & & & \sum_{j\neq N-1}^{n} s_j & -dxdy\rho c\omega & & \\ & & & & & dxdy\rho c\omega & \sum_{j\neq N-1}^{n} s_j & & \\ & & & & & & & \sum_{j\neq N}^{n} s_j & -dxdy\rho c\omega \\ & & & & & & & dxdy\rho c\omega & \sum_{j\neq N}^{n} s_j \end{bmatrix} \begin{bmatrix} u_1 \\ v_1 \\ u_2 \\ v_2 \\ \vdots \\ u_{N-1} \\ v_{N-1} \\ u_N \\ v_N \end{bmatrix} = \begin{bmatrix} \sum_{j\neq 1}^{n} s_j u_j \\ \sum_{j\neq 1}^{n} s_j v_j \\ \sum_{j\neq 2}^{n} s_j u_j \\ \sum_{j\neq 2}^{n} s_j v_j \\ \vdots \\ \sum_{j\neq N-1}^{n} s_j u_j \\ \sum_{j\neq N-1}^{n} s_j v_j \\ \sum_{j\neq N}^{n} s_j u_j \\ \sum_{j\neq N}^{n} s_j v_j \end{bmatrix} \tag{11}$$

The 2D basic FDFD model has been built above, in which the matrix equation is in the form of AX=B. To be more specific, the constant matrix B consists of three external temperature disturbances: the outdoor sol-air temperature, the indoor air temperature and the airflow temperature inside the cavity (T_a). Nevertheless, these temperatures in time domain could not be directly put into the matrix. That explains why the Fourier series expansion should be introduced below.

$$T(t) = T_0 + \sum_{n=1}^{h} T_n sin(\omega_n \cdot t + \varphi_n) = T_0 + \sum_{n=1}^{h}[u_n sin(\omega_n \cdot t) + v_n cos(\omega_n \cdot t)],$$

$$t=0,\ 1,\ 2,\ \cdots,\ 23,\quad h \leq 12,\quad \omega_n = 2\pi \cdot n/24 \tag{12}$$

Where t stands for the 24 hours in one day, h is the order of the Fourier series expansion and the highest order equals to 12.

As one of the external disturbances, the sol-air temperature in a typical day outside the southern wall in Wuhan can be expanded as the superposition of simple harmonic waves. As shown in Fig.5, the temperature values obtained from the superposition of harmonic waves have come close to the original values when the highest order gets 4. The maximum temperature of the outdoor air temperature is 45.0 ℃, at 12:00 noon; the lowest value is 29.1 ℃, at 04:00; its temperature amplitude is 15.9 ℃, and the average temperature is 35.3 ℃. Fig. 6 gives four diagrams of harmonic waves except the constant, i.e. T_0. The temperatures of indoor air and airflow in frequency domain can also be got through this method so as to calculate the FDFD model. In addition, the Fourier inversion is utilized to transform the results from frequency domain to time domain, which is the way to obtain final results in time domain.

The heat transfer processes inside the cavity include the convective heat transfer between the interior cavity surfaces and the airflow, and the radiation heat exchange among the interior cavity surfaces. With regard to calculating the radiation heat transfer inside the cavity, the convective heat transfer and the radiation heat exchange between the surface1 and surface 2 (shown in Fig. 4) are considered, the simplified dynamic heat network model is shown in Fig. 7 (a) [18]. The triangle network composing of two convection resistances and one radiation resistance can be converted to the star network, and the equivalence of triangles and stars is given in Fig.7 (b). Three resistances and four temperatures describe the radiation heat transfer inside the cavity and the convective heat transfer between airflow and interior surfaces.

In Fig.7 (a) and (b), T_{b1} and T_{b2} are the temperatures of the cavity surface 1 and 2, respectively; R_{c1} and R_{c2} represent the convective thermal resistances happening between airflow and cavity surface 1 and 2, respectively; and R_r is the radiation heat transfer thermal resistance. T_a is the airflow temperature, and $T_a{}'$ is the hypothetical temperature of air flow. R_4, R_5 and R_6 are the equivalent resistances calculated by the convective resistance and radiation resistance, which are obtained by Kirchoff's law and shown in Eq. (13).

$$\begin{cases} R_4 = \frac{R_{c1}R_{c2}}{R_{c1}+R_{c2}+R_r} \\ R_5 = \frac{R_{c1}R_r}{R_{c1}+R_{c2}+R_r} \\ R_6 = \frac{R_{c2}R_r}{R_{c1}+R_{c2}+R_r} \end{cases} \tag{13}$$

The relationship between temperatures and thermal resistances can be written as Eq. (14):

$$T_a'\left(\frac{1}{R_4}+\frac{1}{R_5}+\frac{1}{R_6}\right)-\frac{T_{b1}}{R_4}-\frac{T_{b1}}{R_5}-\frac{T_a}{R_6}=0 \tag{14}$$

The value of each thermal resistance can be got by the Eq. (13) and further combining the Eq. (14) leads to $T_a{}'$, which is used to replace the disturbance T_a in the basic model so as to establish the developed FDFD model considering the radiation inside the cavity.

2.2 NTU model for heat exchange between airflow and wall mass

The heat transfer occurs in both the horizontal and the vertical (along the height of the wall) directions. The airflow temperature would not stay the same because the heat exchange happens all the time as the airflow rises along the height of the wall. This cannot be taken into consideration in the 2D FDFD model. Therefore, the method of NTU (number of transfer units) is introduced to evaluate the heat transfer in the vertical direction of the wall.

The NTU method is usually used to design or calibrate the heat exchanger [19], which can be also utilized to calculate the unknown outlet temperature of the fluid during the analysis of heat exchange. Ren and Wright et al. [20] had taken use of NTU method to calculate the heat exchange happening in the ventilated hollow concrete board, and Zhu et al. [21] also used it to investigate the heat transfer between the water and the pipe-embedded building envelop. Similar to these two structures, the outlet temperature of the airflow in the cavity can as well be studied by NTU method.

In the presented NTU model, the airflow and the block represent the two "fluid" performing heat exchange in the "heat exchanger" HBVW. The specific heats of block and airflow are 1198 J/kg·K and 1010 J/kg·K, respectively, but the weight of the block is much larger than that of the airflow. The temperature change for the block of a transfer unit is very small when the heat exchange between the airflow and the block happens. That explains why the block represents the fluid with a bigger specific heat capacity and the airflow is the fluid with a smaller specific heat capacity, which is named minimum specific-heat-capacity fluid, i.e. $(Mc)_{min}$ in NTU method. NTU represents dimensionless number kA/C_{min} called the number of transfer units.

$$\mathrm{NTU} = \frac{kA}{C_{min}} = \frac{kA}{(Mc)_{min}} \tag{15}$$

Where k is heat transfer coefficient of convection between the airflow and the wall mass (W/(m^2·K)), A is the heat transfer area (m^2), M is the mass flow rate of the fluid (kg/s), and c is the specific heat of the fluid (J/(kg·K)).

2.3 The coupled model of HBVW

The FDFD model and NTU model are coupled together by the intermediate variable of wall average temperature, the heat transfer model of HBVW is obtained, and thus the outlet temperature of the airflow can be calculated. Take the structure (height: 2.4m, sectional area: 200mm×190mm) with one single cavity for example, the wall is divided into several heat transfer units along the height which all have corresponding inlet temperature and outlet temperature of airflow, and the outlet one can be calculated by the equation written below:

$$T_{b,n} = T_{m,n} + \left(T_{b,n-1} - T_{m,n}\right)e^{-NTU}, n = 1,2,3, \ldots \quad (16)$$

Where $T_{b,n}$ is the outlet airflow-temperature of number n heat transfer unit, meanwhile $T_{b,n-1}$ is the outlet airflow temperature of number (n-1) heat transfer unit, and it is also the inlet temperature of number n heat transfer unit. $T_{b,0}$ represents the initial value of inlet airflow temperature, and $T_{m,n}$ is the wall average temperature of number n heat transfer unit, i.e. the average temperature of all the heat transfer units on certain height, which can be obtained by FDFD model. The 3D heat transfer can be used to further investigate the outlet temperature of the airflow, so as to prove the accuracy of the coupled heat transfer model in return.

3. Experiment validation of the coupled model of HBVW

3.1 Experiment set up

To validate the coupled heat transfer model, an experiment rig was built. Detailed information about the test rig and experimental study is given in reference [22]. Two cases are presented here to show the comparison between the numerical and experimental results: one is the case without ventilation, the other is the case with ventilation with an air velocity of 0.28 m/s. The airflow for ventilation comes from the exhaust air of the air-conditioning system, so the inlet airflow temperature is equal to the indoor temperature (24 ℃). In these two cases, the indoor air temperature is 24 ℃, and the outdoor solar-air temperature is shown in Fig.6 (a) (the original value). The physical parameters of wall materials are shown in table 1.

3.2 Validation of the coupled model of HBVW

For the first case without ventilation, the temperature of inner and outer surface are mainly concerned. As shown in Fig. 8, both the inner or outer surface temperature from the simulation and the experiment match well with each other. To be more specific, the maximum relative error of inner surface temperature is 2.8% between heat transfer model and experiment. When it comes to outside surface temperature, this value is up to 4.9%, which is still acceptable.

For the second case, the air velocity in HBVW cavity is 0.3 m/s. Similar to the first condition, inner and outer surface temperature from the simulation and experiment all match well. The maximum relative error of inner and outer surface temperature is 1.6% and 2.2%, respectively, which proves the model has a high accuracy.

Besides the temperatures of inner and outer surfaces of the wall, the outlet temperature of airflow should also be validated in order to validate the accuracy of the NTU model concerning the temperature rise of airflow inside the cavity. As shown in Fig. 9 (c), the calculated outlet temperature is close to the experimental one. The maximum relative error is 3.4%, proving that the NTU model has sufficient accuracy. All these results show that calculated results match well with experiments so that the coupled heat transfer model can be reliably utilized to investigate the thermal insulation performance of HBVW.

4. Results and discussions

4.1 Evaluation indexes

To access the thermal performance of HBVW, dimensionless decrement factor (DF) and the time lag (TL) are introduced. TL is the time for a temperature wave propagating through a structure from the outside to the inside, and the DF indicates the decreasing ratio of the temperature amplitude during the above process. The decrement factor DF and time lag TL can be expressed as Eq. (17) and Eq. (18) [23,24]:

$$DF = \frac{t_{is}^{max} - t_{is}^{min}}{t_{\mathrm{os}}^{max} - t_{\mathrm{os}}^{min}} \tag{17}$$

$$TL = \begin{cases} \tau_{is}^{max} - \tau_{\mathrm{os}}^{max} + 24, & \tau_{is}^{max} < \tau_{\mathrm{os}}^{max} \\ \tau_{is}^{max} - \tau_{\mathrm{os}}^{max}, & \tau_{is}^{max} > \tau_{\mathrm{os}}^{max} \end{cases} \tag{18}$$

t_{os}^{max} and t_{os}^{min} are the maximum and minimum outer surface temperatures of HBVW. t_{is}^{max} and t_{is}^{min} are the maximum and minimum inner surface temperatures, respectively. τ_{is}^{max} and τ_{os}^{max} are the moments when the maximum inner surface temperature and maximum outer surface temperature appear, respectively. The lower the decrement factor is, the smaller the amplitude of inner surface temperature is, and the more comfortable the indoor thermal environment becomes.

In addition, average inner surface heat flux q_{is} (W/m^2) during a day is an important aspect that can reflect the heat transferred from the outside to the inside and represent the heat insulation performance of HBVW. It can be calculated by the Eq. (19):

$$q_{is} = \frac{1}{24}\sum_{l=0}^{24} h_{in}\left(T_{is,l} - T_{in,l}\right) \tag{19}$$

Where, h_{in} is convective heat-transfer coefficient on the inner surface of wall, here taking the value of 4.9W/m^{2}K which is obtained from the experiment; $T_{in,l}$ and $T_{is,l}$ are the hourly indoor air temperature and inner wall surface temperature at the moment l, respectively.

4.2 Thermal performance of four orientations of HBVW in Wuhan

The thermal performance of HBVW in Wuhan is calculated and analyzed by using the validated heat transfer model. Four orientations including east, west, south and north are considered. The airflow resource is the exhaust of the air-conditioning system. The airflow rate inside the cavity is 0.3 m/s. The indoor air temperature is 24 °C. The inlet air temperature is assumed to be equal to the indoor air temperature. The solar-air temperatures of four orientations at a typical summer day, as well as the calculated inner and outer HBVW surfaces temperatures, are shown in Fig. 10. Since the solar radiation intensities of four orientations are different, the maximum solar-air temperature and corresponding time it appears differ from each other. The maximum solar-air temperature is 58 °C for the west wall, 55 °C for the east wall and 45 °C for the south wall which appear at 16:00, 9:00 and 12:00, respectively. The north solar-air temperature is mainly affected by the scattered radiation of the sun, so its maximum solar-air temperature appears at 17:00 throughout the day. The temperatures of different orientations are the same at night during the 19:00-5:00.

It can be seen from Fig. 10, the outer surface temperature response of HBVW toward four orientations is similar to the corresponding outdoor solar air temperature trend, but in comparison, the inner surface temperature responses of the four walls are gentler. The outer surface temperatures of the east wall and west wall reach the maximum at 10:00 am and 17:00 pm, respectively. The inner surface temperatures reach maximum at 14:00 pm and 20:00 pm. For the south wall, the peak time of the outer surface temperature is 14:00 pm, which is between those of the east and west wall. The inner temperatures range from 24.8~27.0 °C during the day for the four ventilation walls.

Inner surface temperature responses of four walls are compared. It can be seen that the most obvious inner surface temperature fluctuation occurs in the west wall and the fluctuation amplitude is 2.5 °C, followed by the east wall and the south wall. The inner surface temperature of north HBVW is the gentlest fluctuation and the fluctuation amplitude is 1.2 °C.

The inner surface heat flux, DF and TL of HBVW are shown in Table 2. The DF of HBVW with the airflow rate of 0.3 m/s is 0.209, 0.221, 0.209 and 0.233 when it towards east, west, south and north, respectively. The delay time of the east HBVW is the longest, which is 4 hours. The delay time of the north wall is the shortest, which is only 2 hours. Compared with the non-ventilation wall, the inner surface heat flux is decreased by 55.08%, 55.07%, 56.03% and 55.19% for east, west, south, and north HBVWs, respectively. The thermal performance of the south wall shows the best in hot summer and cold winter area.

4.3 Effects of factors on heat insulation performance of HBVW

Ventilation in the cavity makes use of the airflow to improve the thermal performance of hollow block wall. The velocity and temperature of the airflow, as well as the cavity size would affect the thermal performance of the wall. These three factors are investigated through the coupled heat transfer model to optimize the thermal insulation performance of HBVW.

Taking the south wall as an example, the other boundary conditions are kept unchanged during numerical calculation. The outdoor air temperature is taken from the typical weather day of Wuhan, and the outdoor solar-air temperature used for coupled heat transfer model is the Fourier series expansion of the original values (Fig. 5). The indoor temperature set point is 24 °C; and the exhaust air of the air-conditioning system is used for ventilation. Therefore, the airflow temperature at the inlet is assumed to be the same as the indoor air temperature.

4.3.1 Air velocity

The validated heat transfer model is used to analyze the thermal performance comparison of HBVW. First, the comparisons of six surfaces' temperature distributions for non-ventilation wall and ventilation wall with air velocity of 0.3 m/s are carried out. Second, the effects of different air velocities on thermal performance of HBVW are analyzed.

The surface temperatures obtained from coupled heat transfer model on different surfaces are shown in Fig. 11 (a) and (b), which represent the non-ventilation and ventilation wall with air velocity of 0.3 m/s, respectively. The boundary conditions of indoor temperature and outdoor solar air temperature are set as identical for the two cases. Results show the same tendency that as the surface gets closer to indoor room, the temperature gets lower, the delay time gets longer and the decay rates gets bigger. Moreover, this trend becomes more obvious in the ventilation wall. The inner surface temperature fluctuation of ventilation wall is gentler than that of non-ventilation wall, and the inner surface temperatures are lower for HBVW (shown in Fig. 11 (b)). It indicates that inner surface temperature gets more stable when the wall is ventilated using exhaust air of the air-conditioning system. The inner surface temperature decreases by 1.8 ℃ averagely compared with non-ventilation wall. The results would lead to more stable indoor temperature and better indoor thermal comfort. For the two cases, Surface 3 of the cavity has the same temperature with Surface 4 because of symmetry, and these two surfaces have higher temperature than Surface 1.

When the air velocities are 0.3, 0.8, 1.3 and 1.8 m/s, respectively, the inner surface temperature distribution is shown in Fig. 12. Compared with the situation without ventilation, the inner surface temperature decreases sharply when the cavity is ventilated with airflow at 0.3 m/s. It can be seen that the temperature drops slightly while the velocity increase from 0.3 m/s to 1.8 m/s.

Table 3 shows the decrement factor, delay time, and average inner surface heat flux under different air velocities. When the cool air flows through the cavity, the heat transferred from outside to the wall mass can be partly taken away, the inner surface heat flux reduces, and the inner temperature amplitude as well declines. With the increase of airflow rate, the decrement factor and average inner surface heat flux decrease. For the non-ventilation wall, the DF is 0.409, the average inner surface heat flux q is 21.08W/m^2, and the time lag is 3 hours. Compared with non-ventilation hollow block wall, when the airflow rate increases to 0.3, 0.8, 1.3 and 1.8 m/s, the DF decreases to 0.209,0.182, 0.165 and 0.152, the average inner surface heat flux decreases to 9.27, 7.70, 6.37 and 5.36W/m^2, respectively. As the air velocity increases, the decrement factor decreases. The airflow rate has little effect on delay time: the time lag is 3 hours for airflow rate ranging from 0 to 1.3, and 2 hours for the airflow rate of 1.8 m/s.

From table 3, we can see that the increase of airflow velocity can enhance the convective heat transfer between the wall and the airflow, so just little amount of heat would get through the wall into indoor area. When the velocity is increased from 0 to 0.3 m/s, the average inner surface heat flux decreases from 21.08W/m^2 to 9.27W/m^2 (reduced by 56.02%). Then it slowly decreases to 5.36W/m^2 at the velocity of 1.8 m/s, nearly 74.57% of heat is taken away. It is very sensitive and there is a sharply decrease for both DF and q when the airflow rate is lower than 0.3 m/s. When the airflow rate is higher than 0.8 m/s, both the reductions for DF and q become very gradual.

4.3.2 Airflow inlet temperature

Utilizing the exhaust of indoor air conditioning system means that the airflow inlet temperature should be kept the same with indoor air temperature. Fig.13 presents the inner surface temperature at different airflow inlet temperatures. Fig.14 shows the difference between inner surface temperature and indoor air temperature, i.e. $(T_{is} - T_{in})$.

Higher inlet temperature of the airflow will result in higher temperature of the inner surface. Therefore, $(T_{is} - T_{in})$ is presented here, representing that the temperature difference decreases as the airflow inlet temperature increases. Lower $(T_{is} - T_{in})$ corresponds to better indoor thermal comfort. The temperature difference is less than 2 ℃ when the indoor air temperature is 26 ℃, which shows the best performance. The thermal insulation performance for HBVW at different airflow inlet temperatures is shown in table 4. The inner surface heat flux q has the biggest decline when the airflow inlet temperature is 26 ℃, i.e. 67.75% of heat is taken away, which shows again that 26 ℃ of airflow temperature is the best. The inner surface heat flux is reduced by 56.03% at airflow inlet temperature of 24 ℃, and 47.69% at 22 ℃ compared with non-ventilation hollow block wall. The DF and TL show little change with the variation of the inlet temperature.

4.3.3 Cavity size

The cavity usually occupies 25%-50% of the total block area. The cavity size affects the heat exchange areas between the airflow and the wall mass, so three sizes are studied to optimize the design of hollow block. The air volume ventilated in the cavity is kept the same when studying the effect caused by different cavity sizes. The airflow velocity varies under this circumstance. The inner surface temperature is given in Fig. 15. It can be seen that with the decrease of the cavity area, the inner surface temperature increases, and the moments of peak and trough show time delays.

Table 5 shows the heat insulation performance of HBVW with different cavity sizes. It shows that the wall with bigger cavity size can take away more heat from the ambient environment, even though the smaller cavity size will cause a higher airflow velocity. By changing the cavity area from130×130mm^2 to 110×110 mm^2 and 100×100 mm^2, the airflow rate would increase from 0.3 m/s to 0.4 and 0.5 m/s when the air volume ventilated in the cavity keeps constant. The time lag is increased from 3 hours to 4 and 5 hours, respectively. The average inner surface heat flux increases by 20.23% and 35.86%, respectively. The DF increases by 17.70% and 27.93%, respectively. Results show that the improvement on convective heat transfer resulting from the higher airflow rate, cannot compensate the influence leaded by cavity size. Hence, the effect of the cavity size on the thermal insulation performance is more sensitive than that of air velocity.

4.4 Discussions

HBVW is constructed with the hollow blocks and have similar characteristics compared with the conventional ventilation wall. They all rely on the airflow to take the heat/cold energy away from the wall in summer/winter in the vertical direction. However, the construction and the heat transfer are different. Interlayer ventilation wall usually contains an air gap layer inside the normal wall, so heat transfer of airflow in horizontal plane occurs only between two surfaces of the air gap layer. For HBVW, the airflow passage is rectangular, and heat transfer occurs between the airflow and the four surrounding surfaces.

The impacts of the cavity size, airflow rate and temperature are investigated to evaluate thermal insulation performance of HBVW. Results show that decrement factor decreases as the airflow rate and the cavity size increase, while it is basically the same when the airflow temperature varies. The time lag increases as the cavity size decreases, while the variations of airflow rate and temperature do not have a significant effect on time lag.

When the cavity dimension is constant, the influence of the airflow rate is much greater than the airflow temperature. When the airflow volume is constant, the cavity dimension and airflow velocity are changed simultaneously, and results tell that HBVW with big cavity and low airflow velocity shows better insulation performance than small cavity and high airflow velocity. In order to obtain good performance of the heat insulation, cavity size should be as large as possible within an acceptable porosity, which is the most important factor.

The heat insulation performance of HBVW is better compared with the non-ventilated wall. Koray Ulgen [25] investigated the effects of thermal physical properties on wall's TL and DF. Several normal wall (non-ventilation) are studied. For instances, the construction of one wall is 190mm VCbrick covered with 3mm plaster on the outside and 2mm on the inside. The decrement factor is 0.676, and the time lag is 9.39 hour. Another one is a wall with an air layer (3mm) and an EPfoam insulation layer (3mm) between the outside layer of Cbrick (9mm) and main layer of HCbrick (13.5mm), and the inner surface is inner plaster (20mm). The decrement factor is 0.24, and the time lag is 16.37 hour. It indicates that the air layer and insulation layer can substantially decrease the DF and increase the TL. Because of the insulation layer, the DF is lower than hollow block wall (non-ventilation wall), but still higher than that of the HBVW.

For the non-ventilation hollow block wall, the DF is 0.409, the average inner surface heat flux q is 21.08W/m^2, and the time lag is 3 hours. For HBVW, when the airflow velocity is 0.3 m/s, the decrement factor of HBVW is 0.209

which decreases 48.9%. The inner surface heat flux is 9.27 W/m^2 which decreases by 56.0%, and the time lag is 3 hours. When the airflow rate is 1.8 m/s, the decrement factor of HBVW is 0.152 which decreases by 62.8%. The inner surface heat flux is 5.36 W/m^2 which decreases about 74.6%. The time lag changes to 2 hours. Apparently, HBVW performs much better in the aspect of thermal insulation.

The decrement factor of HBVW is close to that of interlayer ventilation wall. Feng [26] carried out an experiment study to investigate the thermal performance of the interlayer ventilation wall. The interlayer ventilation wall is composed of the external wall, air interlayer and internal wall. The external wall consist of normal Portland cement blocks (240 mm) with cement mortar finish (20 mm). The internal wall is gypsum and the air interlayer is 4 cm between them. The exhaust air of the HVAC system is used to flow through the air interlayer, and the ventilation rates of the test room are controlled between 4 and 9 air changes per hour. The calculated air flow rate is between 0.17 and 0.38 m/s. The decrement factor (namely heat transfer attenuations) is 0.2228 and 0.1989 for the air interlayer wall and the ventilation interlayer wall, respectively. The time lag is 9.75 and 10.10h respectively. Compared with the HBVW at the same airflow rate, the DF is about 0.192 and the TL is about 4 hour. The decrement factor of HBVW proves to be a little better than the Interlayer ventilation wall, but the TL is much shorter.

5. Conclusions

The heat transfer model for the HBVW is established by coupling FDFD model and NTU model. It is a 3D model, which can reflect the heat transfer in both the horizontal and vertical directions of the wall. The model is validated through experiment, by comparing the surface temperatures calculated by coupled heat transfer model and the experimental results. Results show that the model has considerable accuracy.

Effects of influential factors on the thermal performance of HBVW are investigated using the coupled heat transfer model, including airflow velocity, airflow inlet temperature and cavity size. The surface temperature and the average inner surface heat flux q_{is} are calculated. The decrement factor and time lag are analyzed to evaluate this ventilation wall. The conclusions are as follows:

(1) The thermal insulation of HBVW is better than the non-ventilation hollow block wall. The insulation performance of the HBVW towards four orientations are studied, and results show that the inner surface temperature of HBVW ranges from 24.8~27.0 °C during the day of ventilation. The DF of HBVW with the airflow rate of 0.3m/s is 0.209, 0.221, 0.209 and 0.233 when it towards east, west, south and north, respectively. Compared with the non-

ventilation wall, the inner surface heat flux is decreased by 55.08%, 55.07%, 56.03% and 55.19% for east, west, south, and north HBVWs, respectively. Hence, HBVW can cut down heat gain and consequently decrease building energy consumption.

(2) The most critical factor that influences the thermal performance of the HBVW is the cavity size. The decrement factor decreases as the airflow rate and the cavity size increases. When the airflow volume is constant, HBVW with bigger cavity and lower airflow velocity gets lower value of decrement factor. The influence of the cavity dimension on the heat insulation is larger than the airflow rate. When the cavity dimension is constant, the influence of the airflow rate is much greater than the airflow temperature. The suggested airflow rate is 0.8 m/s because DF and heat flux become very gradual.

(3) The time lag of the HBVW ranges from 2-5 hours. The time lag decreases as the cavity size decreases, and there is almost no change when airflow rate and temperature changes.

(4) When the exhaust air of the air-conditioning system is used for ventilation in the HBVW, the inlet airflow temperature has a significant influence on the inner surface temperature and average inner surface heat flux q_{is}, but has little effect on decrement factor and time lag. The inner surface heat flux q decreases when the airflow inlet temperature is higher. When the airflow temperature is 26 ℃, 67.75% of heat is taken away, which is the largest. As the temperature continues to increase, the percent of heat removed keep almost constant when the airflow temperature varies.

(5) The heat insulation performance can be optimized by increasing the airflow velocity, the indoor air temperature, as well as the cavity size. In addition, improving the indoor air temperature can reduce the temperature difference between the wall surface and indoor air, which achieve a better thermal comfort.

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References

[1] EIA, Annual Energy Outlook 2017, Energy Inf. Adm. (2017). doi:DOE/EIA-0383(2012) U.S.

[2] Design Standard for Energy Efficiency for Public Buildings, Ministry of Construction of the People's Republic of China, 2005.

[3] H. Omrany, A. Ghaffarianhoseini, A. Ghaffarianhoseini, K. Raahemifar, J. Tookey, Application of passive wall systems for improving the energy efficiency in buildings: A comprehensive review, Renew. Sustain. Energy Rev. 62 (2016) 1252–1269. doi:10.1016/j.rser.2016.04.010.

[4] Y. Cui, J. Xie, J. Liu, S. Pan, Review of Phase Change Materials Integrated in Building Walls for Energy Saving, in: Procedia Eng., 2015: pp. 763–770. doi:10.1016/j.proeng.2015.09.027.

[5] H. Huo, J. Shao, H. Huo, Contributions of energy-saving technologies to building energy saving in different climatic regions of China, Appl. Therm. Eng. 124 (2017) 1159–1168. doi:10.1016/j.applthermaleng.2017.06.065.

[6] E. Cuce, P.M. Cuce, Vacuum glazing for highly insulating windows: Recent developments and future prospects, Renew. Sustain. Energy Rev. 54 (2016) 1345–1347. doi:10.1016/j.rser.2015.10.134.

[7] J. (Jialiang) Wang, D. Shi, Spectral selective and photothermal nano structured thin films for energy efficient windows, Appl. Energy. 208 (2017) 83–96. doi:10.1016/j.apenergy.2017.10.066.

[8] M. Casini, Active dynamic windows for buildings: A review, Renew. Energy. 119 (2018) 923–934. doi:10.1016/j.renene.2017.12.049.

[9] T. Zhang, Y. Tan, H. Yang, X. Zhang, The application of air layers in building envelopes: A review, Appl. Energy. 165 (2016) 707–734. doi:10.1016/j.apenergy.2015.12.108.

[10] A. Briga-Sá, J. Boaventura-Cunha, J.C. Lanzinha, A. Paiva, Experimental and analytical approach on the Trombe wall thermal performance parameters characterization, Energy Build. 150 (2017) 262–280. doi:10.1016/j.enbuild.2017.06.018.

[11] A. Faheem, G. Ranzi, F. Fiorito, C. Lei, A numerical study on the thermal performance of night ventilated hollow core slabs cast with micro-encapsulated PCM concrete, Energy Build. 127 (2016) 892–906. doi:10.1016/j.enbuild.2016.06.014.

[12] L. C. O. Souza, H. A. Souza, E. F. Rodrigues, Experimental and numerical analysis of a naturally ventilated double-skin façade, Energy Build. 165 (2018) 328–339. doi:10.1016/j.enbuild.2018.01.048.

[13] L. Liu, Z. Yu, H. Zhang, Simulation study of an innovative ventilated facade utilizing indoor exhaust air, in: Energy Procedia, 2017: pp. 126–133. doi:10.1016/j.egypro.2017.08.009.

[14] J. Yu, J. Yang, C. Xiong, Study of dynamic thermal performance of hollow block ventilated wall, Renew. Energy. 84 (2015). doi:10.1016/j.renene.2015.07.020.

[15] M. Boukendil, A. Abdelbaki, Z. Zrikem, Numerical simulation of coupled heat transfer through double hollow brick walls: Effects of mortar joint thickness and emissivity, Appl. Therm. Eng. 125 (2017) 1228–1238. doi:10.1016/j.applthermaleng.2017.07.094.

[16] J. Wu, G.L. Bai, H.Y. Zhao, X. Li, Mechanical and thermal tests of an innovative environment-friendly hollow block as self-insulation wall materials, Constr. Build. Mater. 93 (2015) 342–349. doi:10.1016/j.conbuildmat.2015.06.003.

[17] Y. Zhang, K. Du, J. He, L. Yang, Y. Li, S. Li, Impact factors analysis on the thermal performance of hollow block wall, Energy Build. 75 (2014) 330–341. doi:10.1016/j.enbuild.2014.02.037.

[18] C. Xiong, J. Yu, J. Yang, Study of the Simplified Dynamic Thermal Network Model for the Hollow Block Ventilated Wall, in: Procedia Eng., 2015: pp. 1304–1311. doi:10.1016/j.proeng.2015.09.008.

[19] X. Zhang, Z. Ren, F. Mei, Heat transfer, 5th ed., China Architecture & Building Press, 2007.

[20] M.J. Ren, J.A. Wright, A Ventilated Slab Thermal Storage System Model, Build. Environ. 33 (1998) 43–52. doi:10.1016/S0360-1323(97)00030-9.

[21] Q. Zhu, X. Xu, J. Gao, F. Xiao, A semi-dynamic model of active pipe-embedded building envelope for thermal performance evaluation, Int. J. Therm. Sci. 88 (2015) 170–179. doi:10.1016/j.ijthermalsci.2014.09.014.

[22] J. Yu, J. Huang, X. Xu, H. Ye, C. Xiong, J. Wang, L. Tian, A semi-dynamic heat transfer model of hollow block ventilated wall for thermal performance prediction, Energy Build. 134 (2017) 285–294. doi:10.1016/j.enbuild.2016.11.001.

[23] H. Asan, Y.S. San, Effects of Wall ' s thermophysical properties on time lag and decrement factor, Energy Build. 28 (1998) 159–166. doi:10.1016/S0378-7788(98)00007-3.

[24] L.E. Mavromatidis, M. EL Mankibi, P. Michel, M. Santamouris, Numerical estimation of time lags and decrement factors for wall complexes including Multilayer Thermal Insulation, in two different climatic zones, Appl. Energy. 92 (2012) 480–491. doi:10.1016/j.apenergy.2011.10.007.

[25] K. Ulgen, Experimental and theoretical investigation of effects of wall's thermophysical properties on time lag and decrement factor, Energy Build. 34 (2002) 273–278. doi:10.1016/S0378-7788(01)00087-1.

[26] J. Feng, Z. Lian, Z. Hou, An innovation wall model based on interlayer ventilation, Energy Convers. Manag. 49 (2008) 1271–1282. doi:10.1016/j.enconman.2007.08.011.