A simplified model for calculating heat transfer through the double skin facade

G He¹ ², Y Meng¹, J Zhu¹ and S Zhang¹
¹ College of civil engineering and architecture, Zhejiang University, Hangzhou, China
² Center for Balance Architecture, Zhejiang University

Guoqinghe@zju.edu.cn

Abstract. Double skin façade (DSF) has been recognized as a flexible type of envelope that can adapt to various building needs, such as insulation, solar heat gain, ventilation, and shading. This adaption ability makes the DSF a potentially high performance envelope. However, the reliable calculation of the heat flow in the DSF has been a challenging task due to the complex heat transfer process involved in the DSF. In this study, we propose a simple model that aims to simplify the heat transfer calculation involved in the DSF. In this model, a characteristic function of heat transfer coefficient (CFHTC) was proposed for the heat transfer between the inner layer and the outside air, which would otherwise call the complex convective heat transfer in the cavity. We use experimental data to demonstrate that this function can be expressed as a function of the incident solar intensity. This CFHTC is supposed to be dependent on the geometry of the DSF. With the CFHTC, the calculation of the heat transfer between the inner layer of the DSF and the outside air is simplified and can be incorporated in energy simulation tools.

1. Introduction

Double skin façade (DSF) has been recognized as a flexible type of envelope that can adapt to various building needs, such as aesthetic, daylighting, insulation, solar utilization, ventilation, and shading. This adaption ability makes the DSF a potentially high performance envelope. One of the intriguing questions associated with the double skin façade (DSF) is whether this type of envelope is energy efficient for a particular climate. It is probably not much of a question that the DSF is more efficient than the single skin façade that has only one layer of double glazing (DG) [1]. This appears to be intuitive as the DSF excels both in solar shading coefficient [2] and insulation [3,4]. A more interesting question is whether the DSF can outperform the traditional envelope with wall and windows. This remain a question because of the difficulties associated with the understanding of its energy performance given the complex heat transfer involved in the DSF.

Many studies have been conducted to understand the performance of the DSF [5,6]. CFD simulation is a popular and effective method [7], for steady state performance analysis. However, for dynamic annual energy analysis, one would require a fast and reliable analytical that can be incorporated into energy simulation tools. The energy flow in the DSF involves the solar transmission, convection heat transfer, heat conduction, and radiation heat transfer. Among these components, modelling of the natural convection in the cavity is probably the most challenging. Some researchers used correlations developed for natural convection along vertical plates, which was assumed reasonable for wide cavity [8-11]. Ghadamian et al. [12] used empirical correlations for the heat...
transfer in the air spacing. A single zone model for solar chimney [13] was often used to obtained the mass flow in the ventilated cavity, which was then used to estimate the convection heat transfer coefficients along the cavity [14].

Attempts have been made to obtain a more direct and simpler approach for calculating the total energy through the DSF. For steady state, He et al. [15] showed that the total heat gain can be expressed as a linear function of the solar intensity and the temperature difference between the outside and the inside. Xue and Li [16] used a lumped model for the heat flow between the exterior surface of the inner double glazing and the room. Lee et al. [17] adopted similar approach but used a response factor method that also accounts for the past temperature history of the glazing.

This study took a slightly different approach by developing a characteristic function of heat transfer coefficient (CFHTC) for the heat transfer between the inner layer and the outside air. We use experimental data to demonstrate that this function can be expressed as a function of the incident solar intensity. This objective is to develop a fast and reliable method to access the annual energy performance of a DSF design.

2. Methods

2.1. Experiments

Figure 1. shows the schematic diagram and photos of the test system. The system was consisted of a test cell that was divided by a DG into two spaces: the room chamber and the DSF cavity. In the room chamber, a water-cooled absorbing plate was placed in the middle to function as a heat sink. The DSF structure used the DG as its inner layer in contact with the room chamber and the single glazing as its outer layer. The single glazing was shorter than the total inner height of the cell so that there was an opening at the top and the bottom each. The optical parameters of the glazing used are shown in Table 1, which are derived from the general glass simulation analysis software windows 7.2. Temperatures at various points were monitored as shown in Figure 1. Two duct type platinum resistance thermometers (PT100) were used to measure the water temperature at the inlet and outlet of the collector plate. Eighteen thermocouples (T-type, ±0.5℃) were used to measure the temperatures of the air, the glazing panel, the absorbing plate, and the cell walls. The cell walls were all polystyrene materials and the outer surfaces of the cell were covered with aluminum sheet to reduce absorption and radiative heat transfer. All thermocouples used for measuring air temperature was protected from direct radiative heat transfer by placing the sensor in 3 cm long 0.5 cm diameter aluminum tubes. All temperature readings by the thermocouples were recorded through a NI-9213 data acquisition module every 10s. The circulating water flow rate was measured using micro-motion two-wire K-series Coriolis mass flowmeter (Emerson, ± 0.5%). The flow rate signals were collected via RS-485 communication interface. The radiation intensity of the solar radiation was measured by a pyranometer (2%). Four hot sphere anemometers (Swema 03, 0.03 m/s) were used to measure the airflow velocity at the outlet of the DSF cavity. One anemometer of the same type was positioned close to the outlet to measure the outdoor wind speed. The two PT100 sensors were calibrated in a water bath together so that their reading difference had an accuracy to 0.2 ℃.

Table 1. Optical parameters for the glazing used in the DSF.

| Glazing                  | Optical parameters | Heat transfer coefficient | Emissivity |
|-------------------------|--------------------|--------------------------|------------|
|                         | Transmittance      | Reflectance              | Solar heat gain coefficient |               |
| 6mm float tempered      | 82%                | 8%                       | 0.75       | 2.692      | 0.837       |
| SG 4mm+0.38PVB+4mm      | 75%                | 7%                       | 0.81       | 5.60       | 0.837       |
2.2. Modelling

Figure 2 shows the proposed simplification of the model.

The balance equation for the temperature of the inner glazing ($T_{ig}$) is
\[
\frac{d(mcT)_{ig}}{A_{ig}dt} = h_{ig-in}(T_{in} - T_{ig}) + \varepsilon_{ig-in}\sigma(T_{in}^4 - T_{ig}^4) + \varepsilon_{ig-Al}\sigma(T_{Al}^4 - T_{ig}^4) + \alpha I_{sol} - q_{ig-out} \quad (1)
\]

where \( m \) is the mass, \( c \) is the specific capacity, \( A \) is the area, \( h \) is the convection coefficient, \( T \) is the temperature, \( \varepsilon \) is the emissivity, \( \sigma \) is the Boltzmann constant, \( \alpha \) is the absorptivity of the DG (\( \alpha = 0.15 \)). The subscription \( ig \) is the inner surface of the inner glazing pane; \( in \), the room; \( sol \), the solar; \( out \), the outside air. The first term on the right is the heat gain by convection heat transfer between the DG and indoor air. The second term is the heat gain by radiative heat transfer between the DG and the room. It is assumed that the wall and the air share the same temperature. The third term is the heat gain by radiative heat transfer between the DG and the absorbing plate. The fourth term is the absorbed solar heat gain. The last term is heat transfer between the outside and the inner glazing.

The room chamber temperature \( (T_{in}) \) is balanced by the heat transfer from the inner glazing and the aluminum board.

\[
(mc)_{in} \frac{dT_{in}}{dt} = A_{ig}h_{ig-in}(T_{ig} - T_{in}) + 2 \times A_{Al}h_{Al-in}(T_{Al} - T_{in}) \quad (2)
\]

The balance equation for the temperature of the aluminum absorbing plate \( (T_{Al}) \) is

\[
(m_{Al}c_{Al} + m_w c_w) \frac{dT_{Al}}{dt} = 2A_{Al}h_{Al-in}(T_{in} - T_{Al}) + A_{Al}\varepsilon_{Al-ig}\sigma(T_{ig}^4 - T_{Al}^4) + A_{Al}\varepsilon_{Al-in}\sigma(T_{in}^4 - T_{Al}^4) + A_{Al}I_{sol} - C_w m_w \Delta T_w \quad (3)
\]

where the third term on the right is the radiative heat transfer between the room (walls) and the plate. The fourth term is the absorbed solar radiation passing through the DSF. The last term is the heat taken away by the cooling water.

The heat balance for the outer pane of the DG is

\[
(mc)_{og} \frac{dT_{og}}{dt} = -A_{og}h_{og-out}(T_{og} - T_{out}) + 2 \times A_{Al}h_{Al-in}(T_{Al} - T_{in}) \quad (4)
\]

In Equation (1), typically the heat transfer between the outer pane of the DG and the ambient air, \( q_{ig-amb} \), is solved in a complicated process involving the convection, radiative heat transfer, ventilation of the cavity. In this study, it is modelled as

\[
Q_{ig-amb} = A_{ig}h_{ig-out}(T_{out} - T_{ig}) \quad (5)
\]

where \( h_{ig-out} \) is effective heat transfer coefficient between the inner glazing and the outside air. Presumably, this coefficient is dependent on the heat source (solar absorption) and cavity configuration. We hypothesize that there exits a characteristic function for this coefficient for a given configuration of the cavity. Thus

\[
h_{ig-out} = f(I_{sol}, h_{base}) = (a I_{sol}^{b} + 1)h_{base} \quad (6)
\]

\( f \) is called the characteristic function of heat transfer coefficient, or CFHTC. \( h_{base} \) is the heat transfer coefficient without the solar radiation, which can either be measured [3,4] or determined using CFD method.

The convective heat transfer between the glazing and the room air is [18]

\[
h_{ig-in} = 2.92 \times |T_{ig} - T_{in}| \quad (7)
\]

For convective heat transfer between the isothermal absorbing plate and the room air, the following empirical correlation is used [19].

\[
Nu = C(GrPr)^\eta \quad (8)
\]
with \( c = 0.59 \) and \( n = 0.25 \) for \( 1.43 \times 10^4 \leq Gr \leq 3 \times 10^9 \), \( c = 0.0292 \) and \( n = 0.39 \) for \( 3 \times 10^9 \leq Gr \leq 2 \times 10^{10} \), and \( c = 0.11 \) and \( n = 1/3 \) for \( 2 \times 10^{10} \leq Gr \).

For radiative heat transfer between two glazing panes of the DG, the view factor is assumed to be 1. All surfaces are assumed to be grey surfaces.

For the studied set-up, the base heat transfer coefficient of the cavity is predetermined as \( h_{\text{base}} = 1.5 \) W/(m\(^2\)·K). So far, all parameters in Equations (1)-(6) are or can be pre-determined except for \( a \) and \( b \). The remaining two undetermined parameters will be obtained by fitting the model to the experimental data.

Finally, the total heat flow through DSF into the room chamber, \( Q \), is composed of three parts:

\[
Q = \mathcal{A} G [h_c(T_{ig} - T_{in}) + \varepsilon_{ig-in} \sigma(T_{ig}^4 - T_{in}^4) + \tau I_{sol}]
\]

3. Results

3.1. Measurements results

Figure 3. shows the measured temperatures, solar radiation, and wind speed in four days of tests conducted in November and December of 2019. The duration of each test lasted from 1 to 7 hours during the daytime. The two tests in November lasted only for about 1 hour in order to maintain a relatively constant tilt angle. The DSF test cell was tilted to reduce the solar shadow. Absolute perpendicular was not possible as the sun rose fast in the morning. To acquire a relative long period of test, the DSF system was tilted at a fixed angle (10° for all tests) and the orientation was adjusted intermittently to follow the track of the sun as it was moving from east to the south and to the west. In December, the solar altitude was reduced. It was possible to obtain a longer period with relatively constant solar angle. The two days’ tests in October lasted about 1 hour and the solar radiation on the glazing surface was relatively constant. The two days’ tests in December lasted about 4 to 6 hours and the solar radiation experienced a fast drop short before the sunset. Figure 4. show the measured wind speed during the two tests in December. Much larger air velocity was registered at the cavity outlet in December 11 test than in December 27 test.

3.2. Regression result

The effective transmissivity were very similar for the four measurements: \( \tau = 0.261, 0.286, 0.269, \) and 0.273 for 11/5, 11/6, 12/11, and 12/27, respectively

The regression produces the following correlation between \( h_{ig-out} \) and solar intensity:

\[
h_{ig-out} = \left( 1.69 \times 10^{2.67 \times 10^{-5}} + 1 \right) h_{\text{base}}
\]

where \( h_{\text{base}} = 1.5 \) W/(m\(^2\)·K). Figure 5 shows comparisons of temperatures between the simulation and measurement for the four cases. It can be seen that the temperatures are predicted with reasonable accuracy. Especially, the decreasing trends at the drop of solar radiation in the late day were predicted. Figure 6 shows the comparison of heat flow through the DSF between the simulation and the measurement for the two cases in December. It can be seen that the energy flow, or the room heat gains were predicted with good accuracy.
4. Conclusion

In this study, we propose a simple model that aims to simplify the heat transfer calculation involved in the DSF. In this model, a characteristic function of heat transfer coefficient (CFHTC) was proposed for the heat transfer between the inner layer and the outside air, which would otherwise call the complex convective heat transfer in the cavity. We used experimental data to demonstrate that this function can be expressed as a function of the incident solar intensity. This CFHTC is supposed to be dependent on the geometry of the DSF. With the CFHTC, the calculation of the heat transfer between the inner layer of the DSF and the outside air is simplified and can be incorporated in energy simulation tools.
The study is part of a continuous project. Next, the CFD is to be used to obtain the CFHTC of a DSF design. The ultimate goal is to develop a method that can calculate the energy performance of a DSF design using the CFD method.

Figure 5. The comparison of temperatures of room air and plate between simulation and measurement.

Figure 6. Comparison of net energy flow through the DSF between simulation and measurement.
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