Thermal behavior of a novel type see-through glazing system with integrated PV cells

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Abstract

Steady natural convective airflow in a novel type glazing system with integrated semi-transparent photovoltaic (PV) cells has been analyzed numerically using a stream function vorticity formulation. Based on the resulting numerical predictions, the effects of Rayleigh numbers on airflow patterns and local heat transfer coefficients on vertical glazing surfaces were investigated for Rayleigh numbers in the range of $10^3 < Ra < 2 \times 10^5$. Significant agreement for the Nusselt numbers was observed between numerical simulation results in this study and those of earlier experimental and theoretical results available from the literature. In addition, the effect of air gap thickness in the cavity on the heat transfer through the cavity is evaluated. The optimum thickness of the air layer in this research is found to be in the range of 60–80 mm. This novel glazing system type could not only generate electricity but also achieve potential energy savings by reducing the air conditioning cooling load when applied in subtropical climatic conditions and simultaneously provide visual comfort in the indoor environment.

Keywords:
Convective heat transfer
Fluid flow
Glazing system
Semi-transparent photovoltaic (PV)
Building integration

1. Introduction

Advanced glazing systems have been substantially applied in both high-rise office buildings and commercial buildings in Hong Kong, over the past few years. The application of a double-skin façade can achieve energy savings through the reduction of air conditioning (A/C) energy consumptions. Natural lighting utilization and sound insulation can also be enhanced. A quiet and healthy working environment could thus be provided in a metropolitan city like Hong Kong. A new application of the glazing system is to integrate building facades with semi-transparent photovoltaic (PV) cells. These solar cells can, not only generate electricity, but also provide shade. The void between the two glass panes could serve as a thermal insulation layer and reduce the amount of energy consumed by the air-conditioning system in the building due to low conductivity of the air in the cavity. When the glazing is integrated with photovoltaic cells, an additional function of the building envelope is also created. This kind of solar energy applications is called building-integrated photovoltaics (BIPV). In this application, the windows are integrated with semi-transparent PV cells, which generates electricity as an energy source, thus contributing a dual function to the building envelope. Additional structural installation and maintenance cost could, therefore, be substantially reduced.

Traditionally, windows play an important role in controlling heat transmission in indoor space during the summer period and external ambient heat loss during the winter period. In order to decrease solar transmission through windows, shading and blinds in the air cavities of the double-glazed window are, therefore, often employed. Avedissian and Naylor [1] investigated numerically, the free convective heat transfer in an enclosure with an internal louvered blind. The effect of Rayleigh numbers, enclosure aspect ratios, and blind geometry on the convective heat transfer, in this study is evaluated and compared with experimental results. Experimental results and other theoretical work related to heat transfer by natural convection or multi-mode heat transfer, for traditional double-skin facades, have been given by Todorovic and Cvetkovic. [2] and Krishnan et al. [3]. Gosselin and Chen [4] investigated heat transfer and airflow through a dual-airflow window. Yang and Zhu [5–7] extensively investigated natural convective heat transfer in both inclined channels and tall cavities. Laminar or turbulent natural convection heat transfer and transition of both flows were considered. This natural convective heat transfer in cavities has been of substantial interest owing to the practicality of its application, one example being the use of a double-skin façade to reduce heating and cooling consumption, others include heat loss from flat-plate solar collectors and so on. In addition to the results from theoretical studies, many well-recorded data from experimental tests are also commonly available in the literature for validation of the theoretical mode for evaluation of the buoyant cavity flow [8].
As well as the use of blinds in the cavities of double-glazed windows for shade and thermal control, PV solar cells could be integrated with the glass to absorb solar radiation. Yang et al. [9] theoretically investigated the energy performance of a PV wall and massive wall structure, and revealed that photovoltaic integration in building walls could reduce the cooling load by 33–50%. Fung and Yang [10] developed a one-dimensional transient heat transfer model to evaluate the heat gain of semi-transparent PV modules for BIPV applications. Related research work was conducted by Yang et al. [11,12]. Fossa [13] investigated the physical mechanisms which influenced the thermal behavior of a double-skin photovoltaic (PV) façade, and the experimental results showed that variation in the air gap width could decrease the glass surface temperature and thus enhance PV conversion efficiency. Fluid flow and heat transfer in the PV façade cavities are considerably important in the evaluation of the energy performance and thermal behavior of the BIPV system. Various theoretical models and simulations of BIPV systems and double-skin façades are found in the literature. Moshfegh and Sandberg [14] investigated fluid flow and heat transfer in a vertical channel heated from one side by PV elements. In their study, a two-dimensional finite element model was employed in a computational fluid dynamics study. Charron and Athienitis [15,16] presented a theoretical study of double-façades with integrated PV and motorized blinds, analyzed various design parameters which affected the conversion of solar radiation, and investigated a two-dimensional model of a double-façade with integrated photovoltaic panels.

However, so far, no research has been conducted on the natural convective heat transfer in windows integrated with semi-transparent PV cells, in order to assess the local and average heat transfer coefficients. Determining the local and average heat transfer coefficients is important for estimating the amount of cooling load reduction and conversion efficiency improvement of the BIPV façade. To contribute further information on this issue, heat transfer through windows with semi-transparent PV, is investigated in this study. The various parameters affecting the local heat transfer coefficients on vertical glazing surfaces are also evaluated and based on simulation results, the optimum thermal thickness of the air layer is provided.

### 2. Problem description

Incorporating energy efficiency or sustainable features in building construction design in urban areas could eventually help to reduce the reliance on fossil fuel resources which, in turn, could contribute to the reduction of the effect of global warming. A PV module integrated in a building façade could generate electricity and substantially reduce the amount of carbon dioxide emissions from power stations. Fig. 1 shows the building façade in which the glazing is integrated with an amorphous solar cell. In this application, solar cells provide dual functions, i.e. they provide the building with both shading and electricity.

#### Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>aspect ratio ($-$)</td>
</tr>
<tr>
<td>$g$</td>
<td>gravitational acceleration (m/s$^2$)</td>
</tr>
<tr>
<td>$H$</td>
<td>height of the cavity (m)</td>
</tr>
<tr>
<td>$L$</td>
<td>length of the cavity (m)</td>
</tr>
<tr>
<td>$Nu$</td>
<td>Nusselt number ($-$)</td>
</tr>
<tr>
<td>$n$</td>
<td>number of iterations</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number ($-$)</td>
</tr>
<tr>
<td>$Ra$</td>
<td>Rayleigh number ($-$)</td>
</tr>
<tr>
<td>$T$</td>
<td>temperature (K)</td>
</tr>
<tr>
<td>$x, y$</td>
<td>coordinates</td>
</tr>
<tr>
<td>$X, Y$</td>
<td>non-dimensional coordinates</td>
</tr>
<tr>
<td>$u$</td>
<td>velocity component in $x$-direction (m/s)</td>
</tr>
<tr>
<td>$v$</td>
<td>velocity component in $y$-direction (m/s)</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>thermal diffusivity (m$^2$/s)</td>
</tr>
<tr>
<td>$\beta$</td>
<td>thermal expansion coefficient (1/K)</td>
</tr>
<tr>
<td>$\psi$</td>
<td>stream function (m$^2$/s)</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>constant ($10^{-4}$)</td>
</tr>
<tr>
<td>$\phi$</td>
<td>general non-dimensional variable ($-$)</td>
</tr>
<tr>
<td>$\Psi$</td>
<td>non-dimensional stream function ($-$)</td>
</tr>
<tr>
<td>$\Omega$</td>
<td>non-dimensional vorticity ($-$)</td>
</tr>
<tr>
<td>$\Theta$</td>
<td>non-dimensional temperature ($-$)</td>
</tr>
<tr>
<td>$\nu$</td>
<td>kinematic viscosity (m$^2$/s)</td>
</tr>
<tr>
<td>$\omega$</td>
<td>vorticity (1/s)</td>
</tr>
</tbody>
</table>

#### Subscripts

- $C$: cold surface
- $g$: glass pane
- $o, out$: outdoor
- $PV$: photovoltaic
- $H$: hot surface
- $in$: indoor
- $X, Y$: non-dimensional coordinates

### Fig. 1

Glazing with integrated semi-transparent PV module for potential application in an office room.
3. Mathematical model

3.1. Governing equations for airflow in the cavity

In this study, a mathematical model and computer code are developed to study the heat transfer by natural convection of air in the selected cavity. The airflow was assumed to be two-dimensional, incompressible laminar natural convection flow with constant fluid properties. The Boussinesq approximation was used to estimate the effect of the density variation. Viscous heat dissipation in the fluid was neglected. By elimination of pressure between two momentum equations, the stream function equation, vorticity transport equation and energy equation are obtained as follows.

\[
\frac{\partial^2 \psi}{\partial x^2} + \frac{\partial^2 \psi}{\partial y^2} = -\omega
\]  

\[
\frac{\partial \psi}{\partial y} \frac{\partial \omega}{\partial x} - \frac{\partial \psi}{\partial x} \frac{\partial \omega}{\partial y} = \rho \left( \frac{\partial^2 \omega}{\partial x^2} + \frac{\partial^2 \omega}{\partial y^2} \right) - \frac{\partial \psi}{\partial x} \frac{\partial T}{\partial y}
\]  

\[
\frac{\partial \psi}{\partial y} \frac{\partial \omega}{\partial x} - \frac{\partial \psi}{\partial x} \frac{\partial \omega}{\partial y} = \alpha \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right)
\]

where the stream function, \( \psi \), and the vorticity, \( \omega \), are defined by:

\[
u = \frac{\partial \psi}{\partial y} = -\frac{\partial \omega}{\partial x}
\]

\[
\omega = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}
\]

For the boundary conditions, the temperatures of the two vertical walls are assumed to be uniform, and the bottom wall and top wall are adiabatic, as shown in Fig. 2, so that

\[
\frac{\partial \psi}{\partial x} = \frac{\partial \psi}{\partial y} = 0 \text{ on the walls}
\]

\[
\frac{\partial T}{\partial y} = 0 \text{ at } y = 0, y = H
\]

\[
T = T_H, T = T_C \text{ at } x = 0, x = L
\]

The nondimensional governing equations are obtained by defining the following dimensionless parameters:

\[
\psi = \frac{\psi \rho L^2}{\nu} \quad \Omega = \frac{\omega L^3}{\nu}
\]

\[
X = \frac{x}{L} \quad Y = \frac{y}{L}
\]

\[
\Theta = \frac{T - T_C}{T_H - T_C}
\]

The nondimensional equations are

\[
\frac{\partial^2 \psi}{\partial X^2} + \frac{\partial^2 \psi}{\partial Y^2} = -\Omega
\]

\[
\frac{\partial^2 \Omega}{\partial X^2} + \frac{\partial^2 \Omega}{\partial Y^2} = \frac{1}{Pr} \left( \frac{\partial \psi}{\partial Y} \frac{\partial \omega}{\partial X} - \frac{\partial \psi}{\partial X} \frac{\partial \omega}{\partial Y} \right) + Ra \frac{\partial \Theta}{\partial X}
\]

\[
\frac{\partial^2 \Theta}{\partial X^2} + \frac{\partial^2 \Theta}{\partial Y^2} = \frac{\partial \psi}{\partial Y} \frac{\partial \Theta}{\partial X} - \frac{\partial \psi}{\partial X} \frac{\partial \Theta}{\partial Y}
\]

The boundary conditions in terms of dimensionless variables are:

\[
\psi = 0 \text{ on the walls}
\]

\[
\frac{\partial \Theta}{\partial Y} = 0 \text{ at } Y = 0, Y = A
\]

\[
\Theta = 1, \Theta = 0 \text{ at } X = 0, X = 1
\]

where \( A = H/L \) is the aspect ratio of the enclosure.
3.2. Evaluation of Nusselt numbers

The physical quantities of interest in this problem are the local and average Nusselt numbers. The local and average Nusselt number for heated wall are given by

$$Nu_Y = \frac{Q}{\pi \mu X} \left( \frac{C_1}{C_2} \right)^2$$

$$Nu = \frac{1}{\pi \mu A} \int_0^A Nu_Y \, dY$$

The heat flux through the double-glazed PV window is determined by

$$q = \frac{T_{out} - T_{in}}{L_{g,o} / k_g + (L_{PV} / k_{PV}) + (1/h_o) + (1/h_{in})}$$

where $h$ is the convective heat transfer coefficient due to natural convection of the air in the gap, determined by

$$h = \frac{Nuk_{air}}{L_{air}}$$

$T_{out}$ is outdoor ambient temperature, $T_{in}$ is indoor design temperature, these temperatures are described in section 6.2; $L_{PV}$ is the thickness of the photovoltaic module, $k_{PV} = 237 \, \text{W/m K}$; $L_g$ is the thickness of the glass sheet, $L_{g,o} = 4 \, \text{mm}$ and $L_{g,in} = 10 \, \text{mm}$; $h_o$ and $h_{in}$ are outside and inside convective heat transfer coefficients respectively, which can be obtained from ASHRAE [17].

4. Numerical methods

The present investigation deals with numerical predictions of air flow and temperature distribution in a two-dimensional air cavity, which is isothermally heated from one side and cooled on the other side. The Navier–Stokes equations and energy equation in terms of stream function and vorticity formulation are solved numerically by iterative finite difference technique. The governing equations are substituted by a set of algebraic equations, used in all grid points in the calculation domain being considered. To obtain a grid-independent solution, various grid dimensions were employed. Fig. 3 shows the variation of the vertical velocity profile at mid-height for various grid sizes. It was found that a $36 \times 36$ grid size ensured the independence of solution on the grid with

Fig. 4. Comparisons of Nusselt numbers with literatures.

Fig. 5. Non-dimensional temperatures distribution isotherms for $Ra = 1 \times 10^3, 3 \times 10^4, 5 \times 10^4, 9 \times 10^4, 10^5, Pr = 0.7$ (end walls are adiabatic).
reasonable computation time. The computations are performed on Pentium(R) D CPU 3.40 GHz 3.39 GHz, 1.99 GB of RAM PC using a FORTRAN compiler.

The set of resulting algebraic equations were solved iteratively and were performed by Successive Under-Relaxation (SUR) [18]. SUR is often employed to avoid divergence in the iterative solution of algebraic equations and to accelerate the convergence. Convergence of iteration for the solution is obtained until the values of the variables cease to change from the previous iteration to the next by less than the prescribed value. To examine whether the solution has converged, the following criterion is employed:

\[
\left| \frac{\phi^{n+1} - \phi^n}{\phi^{n+1}} \right| \leq \varepsilon
\]  

(22)

where \( \phi \) stands for \( \Psi, \Omega \) and \( \Theta \) respectively, and \( n \) stands for the number of iterations. The value of \( \varepsilon \) is chosen as \( 10^{-4} \).

5. Validation of the model

The present numerical results are evaluated for accuracy against numerical results published in previous works and experimental results reported by various authors. The comparisons of the average Nusselt numbers for laminar natural convective flow in an enclosure between the results from the present computer code and the published results are shown in Fig. 4. Significant agreement, which demonstrates the validity of the formulation and the computer code, is observed. The correlations between the average Nusselt numbers with other parameters, proposed by Yin et al. [19], are listed below:

\[
\text{Nu} = 0.23A^{0.131}Ra^{0.269} \text{ for } 10^3 \leq Ra \leq 5 \times 10^6 \text{ and } 4.9 \leq A \leq 79.7
\]  

(23)

The correlations from Elsherbiny et al. [20] are listed below:

\[
\begin{align*}
\text{Nu}_1 &= 0.0605 \times Ra^{1/3} \\
\text{Nu}_2 &= \left[ 1 + \left( \frac{0.104Ra^{0.293}}{1 + (6310/Ra)^{1/36}} \right) \right]^{1/3} \\
\text{Nu}_3 &= 0.242 \left( \frac{Ra}{A} \right)^{0.272} \\
\text{Nu} &= \max(\text{Nu}_1, \text{Nu}_2, \text{Nu}_3)
\end{align*}
\]  

(24)

(25)

(26)

(27)

for \( 5 \leq A \leq 110, \quad A = 20:Ra < 2 \times 10^6, \quad A = 40:Ra < 2 \times 10^5, \quad A = 80:Ra < 3 \times 10^4. \)

6. Results and discussion

6.1. Airflow in the cavity

The temperature distribution and flow field of the buoyancy force induced natural convective airflow in the rectangular cavity, have

Fig. 6. Streamlines for \( Ra = 1 \times 10^3, 3 \times 10^4, 5 \times 10^4, 9 \times 10^4, 10^5, \Pr = 0.7 \) (end walls are adiabatic).
been analyzed numerically for different Rayleigh numbers. Numerical results in terms of streamlines and isotherms are presented at various Rayleigh numbers. The numerical results, including the streamlines, temperature, local Nusselt number profiles and average Nusselt number for heated wall, are discussed respectively.

Fig. 5 shows the corresponding isotherm contours for different Rayleigh numbers in the range of $10^3 \leq Ra \leq 10^5$. The results are illustrated for the aspect ratio of 8 and the end walls are assumed adiabatic. In the case of lower Rayleigh numbers, i.e. $Ra = 10^3$, the dominant mode of heat transfer is observed by conduction, as shown in Fig. 5. The isotherms present nearly pure conduction characteristics, almost parallel to both the heated and cooled vertical surfaces. The isotherms are compressed to two vertical walls with the increase of the Rayleigh numbers. It can also be seen that warmer fluid becomes dominant in the rectangular cavity with an increase of the Rayleigh numbers when two thermal boundaries are gradually established. The temperature gradients in the $x$-direction at the bottom left and top right areas of the rectangular

![Figure 7](image1.png)

**Fig. 7.** Vorticity for $Ra = 1 \times 10^3, 3 \times 10^4, 5 \times 10^4, 9 \times 10^4, 1 \times 10^5, Pr = 0.7$ (end walls are adiabatic).

![Figure 8](image2.png)

**Fig. 8.** Variation of the local Nusselt number with coordinate $Y$ for $Ra = 10^3$.

![Figure 9](image3.png)

**Fig. 9.** Variation of the local Nusselt number with coordinate $X$ for various Rayleigh numbers.
cavity are relatively large in comparison with the gradients near the top left and bottom right ends of the glass pane surfaces.

Streamlines and vorticity contours for $Pr = 0.7$ and aspect ratio $A = 8$, presented in Figs. 6 and 7, show the effect of various Rayleigh numbers on the buoyancy induced natural convective flow in the rectangular cavity for the Rayleigh number in the range of $10^3 \leq Ra \leq 10^5$. Normally, aspect ratio in a general case will be quite larger than 8. In this study, an aspect ratio that equals to 8 has been chosen in order to show a complete flow pattern in the air cavity in a small scale. The streamlines and isotherms for Rayleigh number in the range of $1 \leq Ra \leq 10^3$ are neglected in this study because of the small variation in the flow patterns. This paper, therefore, presents the effect of $Ra$ on flow patterns and heat transfer for $Ra > 10^3$. The flow patterns have a significant effect on the rate of heat transfer. As shown in Fig. 6, streamline contours exhibit circulation patterns as the fluid motion is affected by both vertical heated and cooled glass panes. The direction of the flow, due to the thermal buoyancy force, is clockwise in this geometry and a single cell is observed in the cavity for all Rayleigh numbers.

Fig. 8 shows the variation of the local Nusselt number with the coordinate $Y$ at the Raleigh number of $10^3$. In low Rayleigh ranges, the dominant mode of heat transfer is conduction, and small variations in both the airflow pattern and heat transfer are observed especially when the Rayleigh number is less than $10^3$. The figure shows that the local Nusselt number decreases with the increase of the height of the cavity. Fig. 9 shows the variation of the local Nusselt number with coordinate $X$ at the mid-height of the cavity for various Rayleigh numbers and $A = 8$. Fig. 10 shows the vertical velocity profiles at mid-height for various Rayleigh numbers. It was found that the velocity increases with the increase of the Rayleigh numbers.

### 6.2. **Optimization of the air layer**

The effect of the thickness of the air layer between two glass panes on the heat flux through the PV window was investigated for different temperature differences ($T_{out} - T_{in}$). In Hong Kong, the outdoor design temperatures for A/C systems are $32 \degree C$ for summer and $10 \degree C$ for winter. The indoor air temperature in a typical office in Hong Kong is supposed to be maintained, for energy savings, at $25 \degree C$ in summer and $22 \degree C$ in winter. Fig. 11 shows the effect of the thickness of the air layer in relation to two temperature differences, i.e. $12 \degree C$ and $7 \degree C$. It is indicated in the figure that heat transfer through the window can be considerably reduced by optimizing the thickness of the air layer. Heat flux through the window decreases considerably with the increase of the thickness of the air layer when $L$ is less than $10 \text{mm}$ where heat transfer by conduction of the air gap is the dominant mode. The air layer in the window also served as additional thermal insulation. Heat transfer by convection of the air layer becomes pronounced with the increase of the thickness. In this circumstance, the change of heat transfer through the window will be balanced by the increase of conduction heat transfer and the decrease of convection heat transfer due to natural convection of the air in the cavity. The overall heat transfer decreases slightly when the thickness of the air layer is up to $60 \text{mm}$. Therefore, the optimum thickness of the air layer could be chosen as $60–80 \text{mm}$ when energy saving due to less energy transport through the window is considered.

### 6.3. **Power generation from the amorphous silicon cell**

A see-through a-Si solar cell was incorporated in the outer pane of the window for possible application in the office room façade in Hong Kong. The novel type glazing system allows day lighting transmission and without sacrificing vision contact between the occupants and the outdoor environment simultaneously. This design has also been reported by various authors [21,22]. Over the past few years, according to reference [23,24], a see-through a-Si solar cell has been used in windows and skylights by many Japanese architects. In Hong Kong, a ventilated solar window system was developed and analyzed by Chow et al. [25] for possible application in warm climates. They claimed that in the warm climatic regions, the reversible mechanism is not required for the ventilated window. In their recent work [26], a PV module is integrated in the window for cogeneration of electrical power and day lighting utilization.

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**Fig. 10.** The vertical velocity profiles at mid-height for various Rayleigh numbers.

**Fig. 11.** Variation of the heat flux (W/m²) through the PV double-glazed window with different thickness of the air cavity.

**Fig. 12.** Characteristic curves of the semi-transparent PV module at specified conditions.
The I–V characteristics of the see-through a-Si solar cell module have been measured in the Solar Simulation Laboratory of The Hong Kong Polytechnic University. Fig. 12 shows the characteristic curves of the semi-transparent PV module under specified conditions (solar radiation intensity = 815 W/m², measured temperature = 21 °C). Table 1 shows the technical data of the a-Si cell used in the glazing system. About half of the PV panel area is filled with thin film solar cells. The efficiency of the a-Si cell is approximately 6%, which is lower, compared with other types of solar cells, i.e., mono-silicon cell. However, it has a lower manufacturing cost and has little reliance on the Si as well as little effect on the environment.

7. Conclusions

A numerical analysis of the steady buoyancy induced natural convective airflow in a cavity of a novel type see-through glazing system with integration of semi-transparent PV cells has been conducted in the study reported in this paper. Various factors affecting the local and average heat transfer coefficients along the vertical surfaces have been evaluated. It was found that the convective flow strength and heat transfer increased with the increase of the Rayleigh number. The numerical results have been compared with previous experimental results and other theoretical results available in the literature. Significant agreement was observed, thus validating the formulation and computer code, adopted by this study and presented in this paper.

This study provides accurate heat transfer variables, necessary for predicting the PV conversion efficiency, when considering solar cell temperature in an investigation of the thermal behavior of building elements integrated with PV, such as a semi-transparent PV window. The effect of cavity air thickness on the overall heat transfer through the window has been evaluated, and the optimum thickness for the air layer was found to be in the range of 60–80 mm for the case study. Energy losses through the window could be reduced and additional power generation from the PV could also be obtained by applying the optimized semi-transparent PV window.

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