DETERMINATION OF CONVECTIVE HEAT TRANSFER FOR FENESTRATION WITH BETWEEN-THE-GLASS LOUVERED SHADES

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ABSTRACT
A two-dimensional steady laminar natural convection model of a window cavity with between-panes louvered shades (i.e., slats) was developed by approximating the system as a vertical cavity with isothermal walls at different temperatures, and with rotateable baffles located midway between the walls. The baffles were set to a third temperature so that night-time and day-time conditions, and the effects of low emissivity coatings (low-e), could be considered. It was found that the system is suited to a traditional one-dimensional analysis. A novel approach that allows the use of standard vertical cavity convection correlations and a modified cavity half-width is described, and a cavity modification factor, \( n^* \), is presented. Finally, the \( n^* \) factor and vertical cavity convection correlation are joined with a longwave radiant model, and the results are compared to experimental results. The models show good agreement with the experiments.

INTRODUCTION
The objective of this paper is to develop a correlation that predicts the natural convection heat transfer in window cavities containing rotateable louvered shades. Such systems have become increasingly popular, and accurate heat transfer correlations are required for rating purposes and building energy analysis. Such systems have been examined extensively in recent times by Garnet [1], Yahoda [2], Yahoda and Wright [3,4], and Naylor and Collins [5]. To date, however, none have looked at the situation where the blind is sunlit. Recently, Huang [6] conducted an experimental investigation similar to that of Garnet [2]. He examined the effects of louvered on the convective and radiative heat transfer inside a vertical window cavity using two sets of glazings; clear/clear and low-e/clear. His experiment used isothermal vertical surfaces at various pane spacings and louver angles to examine the centre-glass \( U \)-factor. The results showed better window performance when the louveres were tilted from their fully open position and also when the low-e coating was used. A simplified convective heat transfer model was developed which was subsequently combined with Yahoda’s [2] longwave radiation model to predict the centre-glass \( U \)-factor. The new model reproduced experimental data accurately.

In this study, natural convection heat transfer was studied numerically and a correlation was developed that predicts convective heat transfer in the cavity. Convective heat transfer was considered for situations when the blind was at a third prescribed temperature relative to the glass temperatures. As a 3-temperature analysis, simulation of heat transfer can be performed for cases where the shade is hotter than the glass; simulating absorbed solar radiation. That is, the system was analyzed for situations that represent sunlit conditions. Full details of the numerical model are provided in Tasnim et al. [7]. The correlation has been coupled with Yahoda’s [2] longwave model, and comparisons were made to the results of Huang [6].

NUMERICAL MODEL
In the numerical model, a tall vertical enclosure was chosen to represent the glazing cavity, and baffles located on the vertical centre line of the enclosure represented the blind louver (Figure 1). The two window panes (AB and CD) were set apart at a distance, \( W \), and a height, \( H \), and were assumed to be isothermal. The end walls (BC and DA) were assumed to be adiabatic. The blind consisted of a set of evenly spaced isothermal baffles of width, \( w \), and pitch, \( P \). (pitch is the vertical distance between two consecutive louveres), which could be rotated about their centre to an angle, \( \phi \), from the horizontal. The baffles were assumed to be made of a material with high thermal conductivity and no thermal storage capacity, and flat with zero thickness.

Three temperatures were required to model the system. In this study, \( T_1 \) and \( T_2 \) are the left wall (AB) and right wall (CD) temperatures, and \( T_3 \) is the baffle temperature. For convenience, the temperature difference across the cavity and dimensionless baffle temperature are defined as \( \Delta T = T_2 - T_1 \) and \( \Theta_3 = (T_3 - T_1)/(T_2 - T_1) \), respectively. Air properties were evaluated at a reference temperature, \( T_{ref} \), that represents all three temperatures in the system with the baffle temperature predominating.
The air properties at $T_{\text{ref}}$ were taken from Hilsenrath \[8\].

\[
T_{\text{ref}} = \frac{1}{2} \left( \frac{1}{2} T_1 + \frac{1}{2} T_2 + T_3 \right)
\]  \hspace{1cm} (1)

To understand the flow field and heat transfer characteristics of the system, a matrix of three different wall spacings ($W=17.8$ mm, 25.4 mm, and 40.0 mm), three different wall-to-wall temperatures ($\Delta T=35^\circ C$, 10 $^\circ C$, and -15 $^\circ C$), three different baffle temperatures ($\Theta_3= 0$, 0.5, and 1), and three different baffle angles ($\phi=0^\circ$, 45$^\circ$, and -45$^\circ$) were considered. Some additional baffle temperatures were also included for $W=17.8$ mm and 25.4 mm to account for significant solar input to the shade layer. Table 2 presents the matrix of conditions considered in this study.

Steady laminar natural convective heat transfer in the system is described by the fundamental conservation laws of mass, momentum, and energy. The Boussinesq approximation was applied to the $y$-momentum equation, and the assumptions of an incompressible fluid flow with negligible viscous dissipation, and constant thermo-physical properties was made. No slip conditions were applied to all surfaces, the temperature was specified for both side walls and the baffles, and the end surfaces were adiabatic.

The steady state governing equations were discretized by the finite-volume-method using a third order Quick scheme \[9\]. The solution procedures included the conjugate gradient method and the PISO algorithm (Pressure-Implicit with Splitting of Operations) \[10\] to ensure correct linkage between pressure and velocity. The typical number of iterations needed to obtain convergence was between 5,000 and 10,000. The tolerance of the normalized residuals upon convergence was set to $10^{-5}$ for every calculation case.

The numerical model was an approximation of a real fenestration. For an actual window, there would be frame effects and only the center-glass region would be nearly isothermal. The idealized system was, however, consistent with the experimental setup used in the examination conducted by Huang \[6\]. Geometric parameters that remained constant for all numerical simulations are given in Table 1.

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To provide confidence in the numerical model, grid dependency was examined and steady laminar natural convection in a vertical cavity was also studied and compared to published solutions. The results of those tests provided confidence in the numerical model.

Complete details of the numerical model development, validation, and production and analysis of results can be found in Tasnim et al. [7].

**NUMERICAL RESULTS**

From the numerical results, it was shown that the local Nusselt number, \( N_{u_{local}} \), reached a steady-periodic state over a very short distance (Figure 2). This result is a fortunate occurrence in that most windows are analyzed from a one-dimensional centre-glass perspective, and ultimately a center-glass \( U \)-factor would be required for use in window rating software and building energy studies. As such, a center-glass Nusselt number, \( N_{u_{cg}} \), was calculated at the center of the cavity between two consecutive louvers:

\[
N_{u_{cg}} = \frac{1}{2P} \int_{y_{i}}^{y_{j}} N_{u_{local}} dy
\]  

(2)

\( N_{u_{local}} \) is given by

\[
N_{u_{local}} = \frac{q''W}{\Delta T k}
\]  

(3)

Here, \( k \) is the conductivity of the air, \( q'' \) is output by the software as \( q'' = -k \beta \partial T / \partial x \) at the wall, and \( y_{i} \) and \( y_{j} \) are the vertical locations of consecutive slats.

\( N_{u_{cg}} \) is compared to the Rayleigh number in Figure 3 where \( Ra \) is given by:

\[
Ra = \frac{\rho^2 g \beta AT W \delta}{\mu^2 Pr}
\]  

(4)

where \( \rho \), and \( \mu \) are the density and dynamic viscosity, respectively. \( \beta \) is the volume expansion coefficient where \( \beta = 1/T_{ref} \), and \( g \) is gravitational acceleration. \( Pr \) is the Prandtl number

\[
Pr = \frac{\mu C_{p}}{k}
\]  

(5)

Here, \( C_{p} \) is the specific heat of the air. Fluid properties were evaluated at the reference temperature.

Figure 3 shows \( N_{u_{cg}} \) for all situations of \( \theta_{3} = 0, 0.5, \) and 1, and on the right wall. As the system is symmetrical, \( N_{u_{cg}} \) at the left wall can be examined via the same plot where \( \theta_{3, left} = |\theta_{3} - 1| \) and the \( N_{u_{cg, left}} = -N_{u_{cg}} \). For example, to examine \( N_{u_{cg}} \) on the left wall for \( \theta_{3} = 1 \), then look at the results for \( \theta_{3} = 0 \), and take the negative value of the resulting \( N_{u_{cg}} \).

Examination of the numerical results suggested that a number of assumptions can be applied to the formulation of a simplified heat transfer model. These assumptions relate to the treatment of direct convection between the glass, the intra-louver heat transfer, and the glass-to-louver heat transfer characteristics.

- The energy transfer that would occur at the end regions, when the flow reverses cavity sides, and by air entrainment directly through the louvers, was found to be negligible. From the numerical model, \( N_{u_{local}} \) was influenced at the ends of the cavity over a small distance, and therefore, the turn-around region is also small. Flow across the cavity was also negligible due to the formation of cells between the louvers. For these reasons, the convective heat transfer could be represented as the convective heat transfer from the glass-to-blind and blind-to-glass, without including a glass-to-glass term.

- \( N_{u_{local}} \) reached a steady-periodic state over a very short distance. Practically, this supports the one-dimensional centre-glass analysis preferred by building modelers.

- It was shown that the temperature drop across the cavity exists mostly between the blind tips and the glass. The convective cells that form between the slats create mixing which makes the blind-section of the cavity essentially isothermal (i.e., with negligible resistance to heat flow). Therefore, no resistance needs to be assigned to the blind section.
The isotherms spread slightly into the spaces between the louvers. On the basis of this observation it seemed reasonable to treat convective heat transfer between the glass and the blind using established vertical cavity correlations, where the width of the cavity is based on the glass-to-blind spacing with some sort of geometric correction factor applied. That is $Ra$ would be calculated on the basis of a cavity width which is a strong function of slat angle.

Combining these conclusions, the convective heat transfer in a window cavity with a blind can be treated as a combination of two vertical cavities from the glass-to-blind and blind-to-glass without accounting for the blind section. The cavity width will be some modified width based on the slat geometry and the slat tip-to-glass spacing.

$\theta_3 = |\theta_3 - 1|$

$Nu_{cg} = N_{cg}$

$\theta_3 = 0, 0.5, 1.0$ plotted.

Figure 3: $Nu_{cg}$ for $\phi=45^\circ$ (top), $\phi=0^\circ$ (mid), and $\phi=-45^\circ$ (bottom) on the right wall. For left wall: $\theta_{3,\text{left}} = |\theta_3 - 1|$ and $Nu_{cg,\text{left}} = -Nu_{cg}$. Only $\theta_3=0, 0.5, 1$ plotted.
CORRELATION DEVELOPMENT

Based on the previous analysis, it was decided that an attempt would be made to apply the vertical cavity correlation by Shewen et al. [11] to predict heat transfer in either side of the blind layer. The correlation is

\[
Nu = 1 + \left[ \frac{0.0665Ra^{\frac{1}{5}}}{1 + \left( \frac{9000}{Ra} \right)^{\frac{1}{4}}} \right]^{\frac{1}{2}}
\]

(6)

To do so, however, a fictitious cavity width, \( L \), would need to be established which is comprised of the louver tip-to-glass spacing, \( b \), plus a modifying distance, \( c \), that accounts for the fact that the flow on either side of the louver layer does broaden in the region between the louvers. Figure 4 shows the system and parameters under consideration. To further quantify the modifying distance, a fluid layer width modification factor, \( n^* \), was also established where

\[
n^* = \left( 1 - \frac{2c}{W \cos \phi} \right)
\]

(7)

To find the parameter \( n^* \), the following approach was established.

1. A half-cavity Rayleigh number, \( Ra' \), was calculated based on the estimated fictitious cavity width, \( L \).

\[
Ra' = \frac{\rho^2 g \beta \Delta T' L^3}{\mu^2 Pr}
\]

(8)

where \( \Delta T' \) is the temperature difference between the glass and the louvers, and the fluid properties were evaluated at the average of the glass and the louver temperature, \( T_{ref} \). Therefore, on the right side cavity:

\[
\Delta T' = (T_2 - T_3)
\]

\[
T_{ref}' = \frac{T_2 + T_3}{2}
\]

on the left side cavity:

\[
\Delta T' = (T_1 - T_3)
\]

\[
T_{ref}' = \frac{T_1 + T_3}{2}
\]

2. The half-cavity Nusselt number, \( Nu' \), was calculated using Eqn. (6).

3. Recognizing that \( Nu' \) could also be represented by

\[
Nu' = \frac{q'L}{\Delta T' k}
\]

(9)

then

\[
L = \frac{Nu' \Delta T' k_{ref} W}{Nu_{avg} \Delta T k_{ref}}
\]

(10)

Using the new value of \( L \), repeat steps 1 through 3 until convergence.

4. When converged, \( c \) can be calculated using

\[
c = L - \frac{W - W \cos \phi}{2}
\]

(11)

and \( n^* \) is found using Eqn. (7)

Only the results for the 17.8 mm and 25.4 mm pane spacings were used in the aforementioned process. Results from the 40 mm pane spacing were excluded because it was thought that the \( Ra \) numbers were large enough to invalidate the laminar flow assumption, and because that case represents a window which is rarely built due to structural limitations. Values of \( n^* \) are shown in Tables 3 and 4 for \( W = 25.4 \text{mm} \) only. Results for \( W = 17.78 \text{mm} \) have been omitted for brevity. \( \theta_i = 0 \) for the left side, and \( \theta_i = 1 \) for the right side have been omitted because there is no temperature difference between the glass and shade for those cases. Results for \( \phi = -45^\circ \) have also been omitted for brevity.

Table 3

---

Figure 4: Parameters used in correlation development.
Values of $n^*$ for $\phi=45^\circ$, $W=25.4$ mm, $Nu' = 1$.

<table>
<thead>
<tr>
<th>$T_1$ (°C)</th>
<th>$T_2$ (°C)</th>
<th>$W$ (mm)</th>
<th>Left Side</th>
<th>Right Side</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>$\Theta_3$</td>
<td>$Nu_{cg}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>-10.0</td>
<td>25.0</td>
<td>0.0254</td>
<td>0.5</td>
<td>-1.39</td>
</tr>
<tr>
<td>15.0</td>
<td>25.0</td>
<td>0.0254</td>
<td>0.5</td>
<td>-1.34</td>
</tr>
<tr>
<td>40.0</td>
<td>25.0</td>
<td>0.0254</td>
<td>0.5</td>
<td>-1.33</td>
</tr>
<tr>
<td>-10.0</td>
<td>25.0</td>
<td>0.0254</td>
<td>1.0</td>
<td>-2.77</td>
</tr>
<tr>
<td>15.0</td>
<td>25.0</td>
<td>0.0254</td>
<td>1.0</td>
<td>-2.66</td>
</tr>
<tr>
<td>40.0</td>
<td>25.0</td>
<td>0.0254</td>
<td>1.0</td>
<td>-2.66</td>
</tr>
<tr>
<td>-10.0</td>
<td>25.0</td>
<td>0.0254</td>
<td>2.0</td>
<td>-5.64</td>
</tr>
<tr>
<td>15.0</td>
<td>25.0</td>
<td>0.0254</td>
<td>4.0</td>
<td>-10.98</td>
</tr>
<tr>
<td>40.0</td>
<td>25.0</td>
<td>0.0254</td>
<td>-1.0</td>
<td>2.63</td>
</tr>
</tbody>
</table>

Using these results, an attempt was made to produce a correlation for $n^*$. Correlation tests, however, showed $n^*$ was only weakly correlated to $\cos \phi$, $T_{ref}$, and $Ra$. This was not surprising in that one of Huang's [6] conclusions was that, while $n^*$ was important, it did not strongly influence the predicted convective heat transfer. As such, he found that setting $n^*$ to a constant value of 0.70 produced excellent results unless the cavity spacing was wide. Following this conclusion, the average $n^*$ value was found to be 0.61 with a standard deviation of ±0.04. Producing a weighted average, that increased the importance of $n^*$ when the blinds were open and the cavity was narrow (i.e., small $b$), made no difference in the $n^*$ constant quoted above.

**COMPARISON TO HUANG [6]**

As was previously mentioned, Huang [6] modeled the centre-glass $U$–factor using the concept of a thermal resistance network. The convection model described in the previous section (using $n^* = 0.70$) was integrated with the radiation model developed by Yahoda and Wright [9], and used to simulate the glazing system samples tested in his experiments. Using this approach, he accurately reproduced experimentally determined $U$-factors.

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Table 5
Comparison of predicted versus experimental [8] U-factors. \( T_1 = 283 \text{ K} \) and \( T_2 = 303 \text{ K} \) in all cases.

<table>
<thead>
<tr>
<th>Slat Angle</th>
<th>Glass(^2)</th>
<th>W (mm)</th>
<th>(U_{\text{meas}}) (W/m(^2)K)</th>
<th>(n^*) = 0.61</th>
<th>Error</th>
<th>(U_{\text{meas}}) (W/m(^2)K)</th>
<th>(n^*) = 0.70</th>
<th>Error</th>
<th>(U_{\text{meas}}) (W/m(^2)K)</th>
<th>(n^*) = 1.00</th>
<th>Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>75 Cl-Cl</td>
<td>17.8</td>
<td>2.28</td>
<td>2.29</td>
<td>0.4%</td>
<td>2.32</td>
<td>1.8%</td>
<td>2.37</td>
<td>3.9%</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>60 Cl-Cl</td>
<td>17.8</td>
<td>2.50</td>
<td>2.48</td>
<td>0.8%</td>
<td>2.53</td>
<td>1.2%</td>
<td>2.66</td>
<td>6.4%</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>30 Cl-Cl</td>
<td>17.8</td>
<td>2.87</td>
<td>2.79</td>
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</table>

1 Huang also presented results for \( T_1 = 293 \). These results were included in Figure 6, but have been excluded from the Table for brevity. In all cases, the different temperature resulted in less that a 2% difference in the measured \( U \)-factor.

2 Cl-Cl is Clear glass on both sides. Low-\(\epsilon\)-Cl has a low-\(\epsilon\)-coating (\( \epsilon = 0.164 \)) on the outdoor glass.

The comparison between experimentally measured and predicted \( U \)-values are presented in Figure 5 and Table 5 for \( n^* \) values of 0.61, 0.70, and 1.00. Full details of the experimental parameters can be found in Huang [6]. In comparison to experimental results, an \( n^* \) of 0.61 predicts the \( U \)-factor with an average RMS error of 3.2% if the 40 mm results are excluded, and the maximum error is within 6% with one exception. Generally, the new constant is low in its prediction. Comparatively, results produced using \( n^* = 0.70 \), as proposed by Huang [6], give the \( U \)-factor with an average RMS error of 1.5% with a maximum error within 3%. If no \( n^* \) constant is used (i.e., \( n^* = 1.00 \)), the RMS error is almost 10% irregardless of which spacings are considered, and the maximum error reached above 25% for a number of cases. Including the 40 mm cases, the average RMS errors are 6.6%, 5.0%, and 10.1% for \( n^* = 0.61, 0.70, \) and 1.00 respectively. Colour images are welcome as the proceedings will be in digital format only.

Figure 5: Experimental vs predicted \( U \)-factors for different values of \( n^* \). Measured values have been obtained from Huang [6].
It is not surprising that the $n^*$ value of 0.70 works better than the 0.61 value. Huang's constant was determined using his own experimental results and is therefore 'tuned' to the particular conditions of his tests. The value of $n^*=0.61$, however, was determined by an independent numerical study that, in difference to Huang's [6] experiments, had isothermal, flat, and curveless louvers, different glass and louver temperatures, and some different slat angles. It is surmised that, because of these differences, the value of $n^* = 0.70$ is still the better choice. The numerical model, however, as an approximation of the experimental setup, provide excellent confidence in the present approach in addition to insight into the flow structures occurring in the system.

CONCLUSIONS

Using numerical results, a cavity width modification factor of 0.61 was predicted for between-the-glass louvered shades, and a methodology is described for using this value to predict convective heat transfer within the glazing cavity. The convective predictions were coupled with a longwave radiative model and compared to published experimental results.

The new value, while performing well, does not predict a system's $U$-factor better than the 0.70 value put forward by Huang [6]. Approximations made in the numerical model are likely the case. It is suggested, therefore, that Huang's 0.70 value be used in practice. The numerical results do, however, provide confidence in the approach in addition to insight into the flow structures occurring in the system.

ACKNOWLEDGEMENTS

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REFERENCES


