SENSITIVITY ANALYSIS OF HEAT TRANSFER FROM AN IRRADIATED WINDOW AND HORIZONTAL LOUVERED BLIND ASSEMBLY

ABSTRACT

The present study investigates the heat transfer from the indoor glazing surface of a window with adjacent irradiated Venetian blind, using a steady-state, laminar, two-dimensional, conjugate conduction / convection / radiation finite element model of a vertical isothermal plate with heated, horizontal, and rotateable louvers. The sensitivity of radiative and convective heat transfer from the plate has been examined with respect to six variables: blind to glass spacing, blind slat angle, irradiation level, glass temperature, and blind and glass emissivity. The effect of the variables on the heat transfer rate from the plate surface has been demonstrated.

INTRODUCTION

It is common for a louvered shading device, such as a Venetian blind, to be mounted on the indoor surface of a window to provide privacy and to control day-lighting. In addition, the presence of these shading devices will affect natural convection and radiant heat transfer from the window. As a result, there will be a change in the heat transmission and solar heat gain, through the window.

At present, advances are being made that demonstrate the complex thermal interaction between a shade and a window, in the absence of solar irradiation. Several previous studies have examined the effect of a Venetian blind on the free convective heat transfer at an indoor glazing surface when there is no solar irradiance (i.e., for “nighttime” conditions). Machin et al. (1998) performed interferometry and flow visualization. Their experiment used a Mach-Zehnder Interferometer to examine the local and overall convection coefficients from the surface of an isothermal plate at various blind to plate spacings and louver angles. They found that when an aluminum blind was placed close to the plate surface, the slats caused a strong periodic variation in the local Nusselt number distribution. Ye et al. (1999) conducted a two-dimensional finite element study of this problem. In that study, the effects of thermal radiation were neglected and the blind slats were modeled as zero thickness no-slip, impermeable surfaces. A similar numerical study was done by Phillips et al. (1999), which included the effect of heat conduction along the blind slats, but neglected radiation heat exchange. However, the agreement with the experimental data of Machin et al. (1998) was poor, except when the blind slat temperatures were fixed at the measured experimental values. It was concluded that the effect of radiation was significant and needed to be included in the model. Phillips et al. (2000) then numerically determined the effects of horizontal louvers on the coupled convective and radiative heat transfer at an indoor window glazing. Their improved model showed excellent correlation with interferometric data. A more recent study by Collins et al. (2000, 2001) made a move from the previously mentioned “nighttime” models, to a “daytime” model by examining the influence of heated horizontal louvers (representing irradiated blind slats) on the local and average radiative heat transfer from a vertical isothermal surface (representing a window glazing surface). They performed an interferometric validation on a modified version of the numerical model produced by Phillips et al. (2000). Experimental and numerical work showed excellent agreement.
The present numerical study represents a sensitivity analysis of six variables which influence radiative and convective heat transfer from the interior surface of a window, using the model by Collins et al. (2000, 2001). The results will be used to aid in the design of a full parametric analysis of the system.

**PROCEDURE**

In the numerical study, the indoor glazing surface is idealized as an isothermal vertical flat plate of height \( l \) and emissivity \( \varepsilon_p \), that is heated to temperature \( T_p \) above the ambient room temperature \( T_\infty \). A Venetian blind consisting of seventeen horizontal louvers, is positioned a nominal distance \( b \) from the plate surface and the individual slats are inclined at an angle with respect to the horizontal \( \phi \). A heat flux \( q_b \) is put on one side of each slat to simulate the solar radiation absorbed by the blind. The slat geometry and properties were determined using a commercially available aluminum Venetian blind. The slats had a width \( w \), thickness \( t \), an arc length and a radius of curvature \( r_c \), and a pitch \( p_s \) which provided a slat pitch ratio \( p_s/w = 7/8 \) that is typical of commercially available Venetian blinds. The slat thermal conductivity \( k_b \) and emissivity \( \varepsilon_b \) were also included. Figure 1 shows the system geometry. Model parameters that remained constant for all simulations are given in Table 1.

<table>
<thead>
<tr>
<th>( l ) (mm)</th>
<th>( p_s ) (mm)</th>
<th>( w ) (mm)</th>
<th>( t ) (mm)</th>
<th>( r_c ) (mm,deg)</th>
<th>( \varepsilon_c )</th>
<th>( T_\infty ) (K)</th>
<th>( k_b ) (W/mK)</th>
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<tbody>
<tr>
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<td>22.2</td>
<td>25.4</td>
<td>0.17</td>
<td>52.3, 27.3</td>
<td>1.00</td>
<td>297</td>
<td>120</td>
</tr>
</tbody>
</table>

The numerical setup is only an approximate model of actual fenestration. For an actual window, there will be frame effects and only the center-of-glass region will be nearly isothermal. Also, the actual indoor glazing temperature will increase with the solar irradiance, rather than being constant. However, these simplifications eliminate several secondary parameters, such as the frame geometry and the glazing external thermal boundary conditions.

Fluid properties were evaluated at an estimated film temperature of 300 K and were taken from Touloukian et al. (1970a, 1970b, 1975). This approach was taken because the inclusion of a hot blind between the glass and the ambient would cause an under-prediction of the film temperature using the traditional method. Average fluid temperatures predicted by the model showed that the fluid temperature did not deviate significantly from 300 K in any case.

For each of the three slat angles, a parametric of glass temperature was performed for \( b = 30 \) mm and \( n = 18 \) mm, with \( \varepsilon_b = \varepsilon_p = 0.6 \), and \( q_b = 60 \) W/m². Previous investigations (Matchin et al. 1998, Phillips et al. 2000) suggested that convective heat transfer from the interior glazing was more strongly influenced by tip spacing \( n \) as opposed to nominal spacing \( b \). It is not certain if this trend also applied to radiative heat transfer. Additional parametrics were subsequently performed using two base cases consisting of 0° slat angle with glass temperatures of 287 and 307 K (10 K above and below ambient). Depending on the temperature of the glass with respect to the ambient, it was known that the interaction between the heated blind and heated or cooled glass could produce aiding or counter flow. The differences between these two cases were considered to be important enough to justify the performing of a sensitivity analysis for each case. Blind flux, glass and blind...
emissivity, and nominal spacing were each examined independently at four additional levels around the conditions of each base case. The full range of investigation for each variable was largely chosen from experience (Collins et al. 2000, Collins et al. 2001), and are thought to be representative of conditions which may occur in an actual window and shade installation. Tables 2 and 3 show the numerical model conditions.

Table 2: Numerical conditions of initial parametric series.

<table>
<thead>
<tr>
<th>φ (deg)</th>
<th>b (mm)</th>
<th>n (mm)</th>
<th>ε_b</th>
<th>ε_p</th>
<th>q_b (W/m²)</th>
<th>T_p (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-45</td>
<td>30</td>
<td>20</td>
<td>0.6</td>
<td>0.6</td>
<td>60</td>
<td>277, 287, 297, 307, 317</td>
</tr>
<tr>
<td>0</td>
<td>30</td>
<td>18</td>
<td>0.6</td>
<td>0.6</td>
<td>60</td>
<td>277, 287, 297, 307, 317</td>
</tr>
<tr>
<td>45</td>
<td>30</td>
<td>22</td>
<td>0.6</td>
<td>0.6</td>
<td>60</td>
<td>277, 287, 297, 307, 317</td>
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<tr>
<td>-45</td>
<td>27</td>
<td>18</td>
<td>0.6</td>
<td>0.6</td>
<td>60</td>
<td>277, 287, 297, 307, 317</td>
</tr>
<tr>
<td>45</td>
<td>25</td>
<td>18</td>
<td>0.6</td>
<td>0.6</td>
<td>60</td>
<td>277, 287, 297, 307, 317</td>
</tr>
</tbody>
</table>

Table 3: Numerical conditions of secondary parametric series. All parametrics performed at φ = 0. Base conditions indicated in bold.

<table>
<thead>
<tr>
<th>b (mm)</th>
<th>ε_b</th>
<th>ε_p</th>
<th>q_b (W/m²)</th>
<th>T_p (K)</th>
</tr>
</thead>
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<td>0.6</td>
<td>60</td>
<td>287, 307</td>
</tr>
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<td>20, 25, 30, 35, 40</td>
<td>0.6</td>
<td>0.6</td>
<td>60</td>
<td>287, 307</td>
</tr>
<tr>
<td>30</td>
<td>0.2, 0.4, 0.6, 0.8, 1.0</td>
<td>0.6</td>
<td>60</td>
<td>287, 307</td>
</tr>
<tr>
<td>30</td>
<td>0.2, 0.4, 0.6, 0.8, 1.0</td>
<td>0.6</td>
<td>60</td>
<td>287, 307</td>
</tr>
<tr>
<td>30</td>
<td>0.6</td>
<td>0.2, 0.4, 0.6, 0.8, 1.0</td>
<td>60</td>
<td>287, 307</td>
</tr>
<tr>
<td>30</td>
<td>0.6</td>
<td>0.2, 0.4, 0.6, 0.8, 1.0</td>
<td>60</td>
<td>287, 307</td>
</tr>
<tr>
<td>30</td>
<td>0.6</td>
<td>0.6</td>
<td>0, 30, 60, 90, 120</td>
<td>287, 307</td>
</tr>
<tr>
<td>30</td>
<td>0.6</td>
<td>0.6</td>
<td>0, 30, 60, 90, 120</td>
<td>287, 307</td>
</tr>
</tbody>
</table>

GOVERNING EQUATIONS

In developing the numerical model, a number of assumptions have been made. These include

- The flow is steady, laminar, incompressible and two-dimensional. These assumptions were later confirmed experimentally (Collins et al. 2001).
- The thermo-physical properties are constant, except for fluid density, which is treated by means of the Boussinesq approximation.
- Grey diffuse radiation exchange between the glazing surface, blind and room has been considered, and the fluid is a non-participating medium. All surfaces were assumed to be uniform in temperature.

For the purposes of model execution, the problem has been non-dimensionalized. The dimensionless governing equations for the fluid are:

\[
\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0
\]

(1)

\[
\frac{1}{\text{Pr}} \left( U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} \right) = -\frac{\partial P}{\partial X} + \left( \frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} \right)
\]

(2)
\[
\frac{1}{\text{Pr}} \left( U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} \right) = -\frac{\partial P}{\partial Y} + \left( \frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} \right) + Ra_R \left( T^* - 1 \right) \]
\[
U \frac{\partial T^*}{\partial X} + V \frac{\partial T^*}{\partial Y} = \left( \frac{\partial^2 T^*}{\partial X^2} + \frac{\partial^2 T^*}{\partial Y^2} \right) \]

The above equations have been cast in dimensionless form using the following dimensionless variables:

\[
X = \frac{x}{l}, \quad Y = \frac{y}{l}, \quad U = \frac{u}{\alpha_f |l|} \]
\[
V = \frac{v}{\alpha_f |l|}, \quad P = \frac{P L^2}{\mu_\alpha}, \quad T^* = \frac{T}{T_\infty} \]

Note that the temperature has been scaled using the absolute ambient temperature because of the coupled radiative heat transfer. As a result, in Eq. (3) the modified or "Radiation" Rayleigh number (RaR) has been defined as:

\[
Ra_R = \frac{g \beta_f T_\infty l^3}{\alpha_f \nu_f} \]

However, the results will be presented in terms of the conventional Rayleigh number (Ra):

\[
Ra_i = \frac{g \beta_f (T_p - T_\infty) l^3}{\alpha_f \nu_f} \]

The relationship between RaR and Ra is:

\[
Ra_i = Ra_R (T_p^* - 1) \]

Steady-state conduction heat transfer in the solid blind is governed by Laplace's equation:

\[
\frac{\partial^2 T^*}{\partial X^2} + \frac{\partial^2 T^*}{\partial Y^2} = 0 \]

The dimensionless flux to the top of each slat surface is

\[
q_b^* = \frac{q_b l}{k_f T_\infty} \]

The radiative heat transfer was calculated using the net radiation method (Siegel and Howell 1970), assuming each surface to be grey and diffuse with constant emissivity and uniform temperature. With N surfaces in an enclosure, the dimensionless radiative heat flux from each sub-surface (q*) was calculated as follows:

\[
\sum_{j=1}^{N} \left( \frac{\delta_{ij}}{\varepsilon_j} - F_{ij} \frac{1 - \varepsilon_j}{\varepsilon_j} \right) q_j^* = N_{rc} \sum_{j=1}^{N} F_{ij} (T_{k}^{*4} - T_{j}^{*4}) \]

where
\[ q_j^* = \frac{q_j l}{k_f T_{\infty}} \quad \text{and} \quad N_{RC} = \frac{\sigma T_{\infty}^3 l}{k_f} \]

In Eq. (11), \( N_{RC} \) the radiation-to-conduction interaction parameter.

A sketch of the dimensionless computational domain is shown in Fig 1. Referring to Fig 1, the dimensionless boundary conditions are:

\[
\begin{align*}
U &= V = 0, T^* = T_p^* & \text{GH} \\
U &= V = 0, T^* = T_p^* & \text{FG, HA} \\
\partial U/\partial X &= V = 0, T^* = 1 & \text{BCDE}
\end{align*}
\]

At the surface of the slats, no-slip and impermeability conditions apply \((U=V=0)\). Continuity conditions for temperature and heat flux also apply at this solid-fluid interface, which can best be expressed in dimensionless form as:

\[
K_{bf} \left. \frac{\partial T^*}{\partial N} \right|_{solid} = \left. \frac{\partial T^*}{\partial N} \right|_{fluid} + \frac{q_j l}{k_f T_{\infty}} - \frac{q_j l}{k_f T_{\infty}}
\]

where \( N \) is the normal vector and \( K_{bf} \) is the blind to fluid conductivity ratio.

Equations (1) through (11) have been solved subject to the specified boundary conditions using a finite element method. Nine-node quadratic elements with biquadratic interpolation functions were used for temperature and velocity. Pressure was eliminated from the momentum equations using the penalty formulation (Fluent 1999). The discretized equations were solved using successive substitution, with incremental loading and under-relaxation to speed convergence.

Extensive grid density and far field boundary testing has been done (Phillips et al. 2000). Based on this testing, a graded mesh with approximately 27000 nodes was used. Referring to Fig 1, the upper (FE) and lower (AB) entrance regions were set at a dimensionless height of 0.105, and the dimensionless domain width was set at 0.171. At these values, the average convective and radiative Nusselt number data were found to be grid and boundary independent.

**RESULTS AND DISCUSSION**

The ultimate objective of this research is to determine the effects of a sunlit shade on heat transfer in the center-of-glass region of a window. The results of Collins et al. (2000, 2001) showed that the blind layer generally suppressed significant changes in heat transfer, and allowed a center-of-glass analysis to be performed on the middle section of the window model. As such, average and local heat flux rates will only be presented for the glass region located between the midpoint of the 5th and 6th slats, to the midpoint of the 12th and 13th slats (slats 1 and 17 being the bottom and top slats respectively). This vertical section includes 7 blind slats, and 0.16 m of glass. A complete analysis and discussion of the heat transfer in the end regions can be found in Collins et al. (2000, 2001). For the present convention, positive flux is from the plate, while negative is into the plate. Average heat flux has been summarized in Tables 4 and 5 and Fig. 2.
Table 4: Numerically predicted average convective and radiative heat flux from the center-of-glass region of the plate. All models executed at $\varepsilon_b = \varepsilon_p = 0.6$, $q_b = 60 \text{ W/m}^2$.

<table>
<thead>
<tr>
<th>$\phi$</th>
<th>blind spacing</th>
<th>Average flux (W/m²)</th>
<th>$T_p$ (K)</th>
<th>277</th>
<th>287</th>
<th>297</th>
<th>307</th>
<th>317</th>
</tr>
</thead>
<tbody>
<tr>
<td>-45</td>
<td>$b = 30 \text{ mm}$</td>
<td>convective</td>
<td>-65.5</td>
<td>-31.3</td>
<td>-7.1</td>
<td>22.8</td>
<td>60.7</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>radiative</td>
<td>-62.7</td>
<td>-38.6</td>
<td>-11.0</td>
<td>20.7</td>
<td>55.3</td>
<td></td>
</tr>
<tr>
<td>0</td>
<td>$b = 30 \text{ mm}$</td>
<td>convective</td>
<td>-66.7</td>
<td>-31.8</td>
<td>-8.0</td>
<td>22.0</td>
<td>59.8</td>
<td></td>
</tr>
<tr>
<td></td>
<td>$n = 18 \text{ mm}$</td>
<td>radiative</td>
<td>-65.9</td>
<td>-40.1</td>
<td>-10.5</td>
<td>23.2</td>
<td>60.0</td>
<td></td>
</tr>
<tr>
<td>45</td>
<td>$b = 30 \text{ mm}$</td>
<td>convective</td>
<td>-65.1</td>
<td>-26.4</td>
<td>-4.4</td>
<td>24.5</td>
<td>62.4</td>
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<tr>
<td></td>
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<td>radiative</td>
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<td>$n = 18 \text{ mm}$</td>
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<td>21.5</td>
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<tr>
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<td>$n = 18 \text{ mm}$</td>
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<td>-30.9</td>
<td>-7.0</td>
<td>22.0</td>
<td>59.6</td>
<td></td>
</tr>
</tbody>
</table>

Table 5: Numerically predicted average convective and radiative heat flux from the center-of-glass region of the plate. Results are for tests indicated in Table 3. All models executed at $\phi = 0$. Unless otherwise stated, all models executed at $b = 30 \text{ mm}$, $\varepsilon_b = \varepsilon_p = 0.6$, $q_b = 60 \text{ W/m}^2$.

<table>
<thead>
<tr>
<th>$T_p$ (K)</th>
<th>Average flux (W/m²)</th>
<th>$b$ (mm)</th>
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<th>25</th>
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<th>35</th>
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<tr>
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<td>convective</td>
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<td>-35.1</td>
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<tr>
<td>307</td>
<td>convective</td>
<td>7.5</td>
<td>17.9</td>
<td>22.0</td>
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<td>21.8</td>
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<th>$T_p$ (K)</th>
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</tr>
<tr>
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<td>radiative</td>
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<td>-40.1</td>
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<td>-49.3</td>
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</tr>
<tr>
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<td>23.9</td>
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<td>27.8</td>
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</tbody>
</table>
The effect of glass temperature and slat angle on the average convective and radiative flux can be seen in Fig. 2 (a) and (b) and Table 4. Fig. 2 (a) and (b) present the results of the glass temperature and slat angle parametric at a constant nominal spacing and tip spacing respectively. Both radiative and convective heat transfer rates demonstrate a linear relationship with temperature whereby the average flux becomes more positive with increasing glass temperature. At all three slat angles, linear data fits to the convective data produced correlation coefficients of 0.992 or greater when \( b = 30 \text{ mm} \) and 0.995 when \( n = 18 \text{ mm} \). Considering the radiative heat transfer results, linear data fits to the data produced correlation coefficients of 0.995 when \( b = 30 \text{ mm} \) and 0.994 when \( n = 18 \text{ mm} \). A more significant result can be found by examining the difference of calculated heat flux with varying blind slat angle. By ignoring slat angle and fitting a straight line to the convective flux data, we find that the correlation coefficient of 0.993 for \( b = 30 \text{ mm} \) and 0.995 for \( n = 18 \text{ mm} \) are produced. This confirms the findings of Machin et al. (1998) and Phillips et al. (2000) who found that tip spacing is more important than slat angle when determining convective heat transfer from the glass. A slightly different result occurs when considering radiative heat transfer. As before, by ignoring slat angle and fitting a line to the radiative flux data, we find that the correlation coefficient of 0.994 for \( b = 30 \text{ mm} \) and 0.993 for \( n = 18 \text{ mm} \) are produced. The radiative heat transfer rate is more dependent on nominal spacing than blind slat angle. These results suggest that slat angle is not important when determining radiative heat transfer, and need only be considered in calculating tip spacing when determining convective heat transfer. It should also be noted that for the cases chosen, average radiative and convective flux are of the same magnitude.

The effect of blind emissivity on convective and radiative heat transfer from the glass can be seen in Fig. 2 (c) and Table 5. Average convective heat fluxes were not significantly affected by changes in blind emissivity. The magnitude of change in convective heat transfer over the range of blind emissivities examined was not significant. When considering the warm glass case (\( T_p = 307 \text{ K} \)), the convective flux changed by less than 1.2 W/m² or 5.6% over the entire range. The cold glass case (\( T_p = 287 \text{ K} \)) was more significant at 5.0 W/m² or 14.5%. Quadratic fits of the data produced correlation coefficients in excess of 0.990 in both cases while linear fits were slightly worse with correlation coefficients of 0.903 at \( T_p = 287 \text{ K} \) and 0.947 and \( T_p = 307 \text{ K} \). It may be possible to accurately predict convective heat transfer without using the blind emissivity as input. When considering the radiative heat flux, changes in slope were more significant than seen in the convective data. Greater blind emissivities reduce the resistance of the blind to receiving radiative heat flux from the glass. Therefore, radiative flux from the glass becomes more negative with increasing blind emissivity. Quadratic fits of the data produced correlation coefficients in excess of 0.992 in both cases while linear fits were slightly worse with correlation coefficients of 0.914 at \( T_p = 287 \text{ K} \) and 0.866 and \( T_p = 307 \text{ K} \).

The effect of glass emissivity can be seen in Fig. 2 (d) and Table 5. Convective heat flux from the glass is not influenced by the glass emissivity. Convective flux is reduced by only 1 W/m² or 4.4% for the hot plate, and increased by 2.9 W/m² or 8.7% for the entire range of emissivities examined. As with the blind emissivity, it may be possible to ignore glass emissivity when predicting convective heat transfer. A linear fit to the data works well in both cases with correlation coefficients of 0.975 at \( T_p = 287 \text{ K} \) and 1.000 at \( T_p = 307 \text{ K} \). In contrast however, the radiative heat transfer changes significantly with glass emissivity. Increasing glass emissivity would reduce the resistance of the glass to radiative heat transfer, and would therefore increase the magnitude of flux emitted or absorbed at the plate surface. Both cases show excellent linearity with correlation coefficients of 0.997 for each glass temperature.
The effect of nominal blind spacing can be seen in Fig. 2 (e) and Table 5. As the blind gets closer to the glass, an increased effect is clearly indicated. When considering convection, closer spacing causes a more negative shift in the direction of heat flux. The heat lost from a warm plate is reduced, while heat gained by a cold plate increases. A 2nd order polynomial fit to the data produces a correlation coefficient of 0.985 for both glass temperatures. Changes in radiative heat flux are less pronounced with proximity, although a slight decrease in heat transfer is experienced as the plate gets closer to the glass. Because the blind consisted of the hottest surfaces in the system, an increase in the glass to ambient viewfactor and an associated decrease in the glass to blind viewfactor would reduce the radiant exchange to the glazing surface. Quadratic data fits produce correlation coefficients of 0.960 at $T_p = 287$ K and 0.990 at $T_p = 307$ K.

The final variable examined is the level of absorbed blind flux. Figure 2 (f) and Table 5 show the effect of blind flux on heat transfer from the glass. Higher blind flux results in higher blind slat temperatures, which cause both the convective and radiative flux to become more negative. Linear trendlines fit to each data set produce correlation coefficients in excess of 0.997. A more important result was found by examining the slope of each trendline. The change in convective heat flux changes from $-0.06$ to $-0.10$ W/m$^2$ per W/m$^2$ of absorbed flux between $T_p = 287$ K and 307 K respectively. Between those same temperatures, radiative heat flux changes from $-0.15$ to $-0.17$ W/m$^2$ per W/m$^2$ of absorbed flux. Small changes of slope with changing temperature suggest that these variables are not strongly coupled.

An example of local convective and radiative heat flux rates for the complete $0^\circ$ slat angle temperature parametric can be seen in Fig. 3 where slat positions are shown in gray. Local radiative heat flux does not change with vertical location on the glass, indicating insignificant changes in blind slat temperature. However, local convective heat transfer rates do change with distance up the glass, indicating boundary layer growth. The magnitude of this change increases as the plate temperature deviates further from the ambient temperature. That is, the convective flux increases from the glass over a height of 0.16 m by only 1.8 W/m$^2$ at $T_p = 297$ K, while at $T_p = 317$ K the convective flux into the glass decreased by 17 W/m$^2$. While this suggests that window height should be included in the analysis, software limitations prevent expansion of the model at this time. Fortunately, significant changes in local convective flux only occur at extreme glass temperatures. Boundary layer growth is unavoidable and must be accepted as a limitation of the analysis. Complete local heat flux results can be found in Collins (2001).

CONCLUSIONS

Radiative and convective heat transfer from a horizontal Venetian blind adjacent to an indoor window glazing has been obtained using a validated conjugate heat transfer model of the same system. The following conclusions were drawn from the results, and can be applied within the range of the parameters investigated.

- Blind slat angle does not effect the average convective or radiative heat flux from the glass. Convective heat flux determined at different slat angles with equivalent tip to glass spacings ($n = 18$ mm), and radiative heat flux determined at different slat angles with equivalent nominal spacings ($b = 30$ mm) were similar.
- The average radiative and convective heat flux increases linearly with increased plate temperature. For the cases examined, radiative and convective fluxes were also of the same magnitude.
• The average convective flux from the plate is not significantly affected by either the plate or the blind emissivity, and may not be required in a predictive equation. The average radiative flux from the plate increases linearly in magnitude with increasing glass emissivity, and becomes more negative with increasing blind emissivity.

• Nominal blind distance has more influence on heat transfer as the blind gets closer to the glass surface. Average convective and radiative heat transfer rates are well represented by quadratic correlations.

• Convective and radiative heat flux become more negative with increasing levels of absorbed blind flux. In addition, both fluxes change linearly with $q_b$. More importantly, insignificant changes in the rate of change in flux from the glass with changing glass temperature suggest that these variables are not coupled.

• Instantaneous flux rates show that while radiative flux is steady under all the conditions examined, convective flux does change slightly with distance along the glass surface. These changes in convective flux have been accepted and noted as a deficiency in the analysis.

While this analysis has shown the effect of model parameters, it does not give sufficient information concerning parameter interactions. Inferences can be made, however, concerning the glass temperature verses slat angle and glass temperature verses blind flux interactions. A three level factorial design can be used to assess those interactions.

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NOMENCLATURE

\begin{align*}
  b &\quad \text{nominal louver spacing, mm} \\
  c_p &\quad \text{specific heat, J/kg-K} \\
  F &\quad \text{radiation shape factor, dimensionless} \\
  g &\quad \text{gravity, m/s}^2 \\
  h &\quad \text{heat transfer coefficient, W/m}^2\text{-K} \\
  k &\quad \text{conductivity, W/m-K} \\
  K &\quad \text{conductivity, dimensionless} \\
  l &\quad \text{plate height, mm} \\
  n &\quad \text{louver tip to plate spacing, m} \\
  Nu &\quad \text{Nusselt number, dimensionless} \\
  N &\quad \text{normal vector} \\
  N_{rc} &\quad \text{radiation-to-conduction interaction parameter, dimensionless} \\
  p &\quad \text{pressure, Pa} \\
  P &\quad \text{pressure, dimensionless} \\
  Pr &\quad \text{Prandtl number, dimensionless}
\end{align*}
**Symbols**

- $\alpha$: thermal diffusivity, $m^2/s$
- $\beta$: volume expansion coefficient, $1/K$
- $\delta$: Kronecker delta, dimensionless
- $\varepsilon$: emissivity, dimensionless
- $\phi$: louver angle, deg.
- $\mu$: dynamic viscosity, $kg/m\cdot s$
- $\nu$: kinematic viscosity, $m^2/s$
- $\rho$: density, $kg/m^3$
- $\sigma$: Stefan-Boltzmann Constant, $W/m^2\cdot K^4$

**Subscripts**

- $C$: convective
- $b$: blind/louver
- $f$: fluid
- $p$: plate/glass
- $R$: radiative
- $\infty$: ambient/room

**Other**

- *: alternative dimensionless notation

**REFERENCES**


Figure 1: System geometry (left), computational domain (right).
Figure 2: Average convective and radiative heat flux from the center-of-glass region of the plate. Model conditions are presented in Tables 2 and 3. (a) temperature parametrics for $b = 30$ mm, (b) temperature parametrics for $n = 18$ mm, (c) blind emissivity parametric, (d) glass emissivity parametric, (e) nominal spacing parametric, and (f) blind flux parametric.
Figure 3: Local convective and radiative heat flux in the center-of-glass region with changing glass temperature ($\phi = 0^\circ$, $b = 30$ mm, $\varepsilon_b = \varepsilon_p = 0.6$, $q_b = 60$ W/m$^2$). Slat positions are shown in gray.