A NUMERICAL STUDY OF THE EFFECT OF A VENETIAN BLIND ON THE CONVECTIVE HEAT TRANSFER RATE FROM A RECESSED WINDOW WITH TRANSITIONAL AND TURBULENT FLOW

by

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Abstract

The presence of a blind adjacent to a window affects the natural convective air flow over the window and natural convective heat transfer from the window to the room. Most numerical studies of convective heat transfer between a window-blind system and a room are based on the assumption that the flow remains laminar. However, in the case of larger windows it is to be expected that transition to turbulent flow will occur in the flow over the window. The aim of the present study was to numerically determine the effect of Venetian blind on laminar-to-turbulent transition in the flow over a simple recessed window and on the convective heat transfer from the window. An approximate model of a recessed window that is covered by a venetian blind has been considered. The fluid properties have been assumed constant except for the density change with temperature that gives rise to the buoyancy forces, this being dealt with using the Boussinesq approach. Radiant and conductive heat transfer effects have been neglected. However in the present study the case where there is a constant heat generation rate in the blind slats, as the result of solar radiation absorbed by the slats of the blind, has been considered. The k-epsilon turbulence model with the full effects of the buoyancy forces being accounted for has been used in obtaining the solution. The turbulent, steady and two dimensional governing equations have been solved using the commercial finite-volume based CFD code FLUENT. Results are generated for different blind slat angles, for different distances of the pivot point of the slats from the window and for different constant heat generation rates in the slats. The results show that over a wide range of Rayleigh number, the distance of the blind to the window has a stronger effect on the convective heat transfer from the window and also on the laminar to turbulent transition in the flow over the window than the blind slat angle. Heat generation in the slats increases the Mean Nusselt number and this effect increases as the Rayleigh number decreases.
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<th>Symbol</th>
<th>Description</th>
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</tr>
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<tr>
<td>$C_p$</td>
<td>Specific heat</td>
<td>$J/kg.K$</td>
</tr>
<tr>
<td>$c_1$</td>
<td>Constant used in k-e turbulent model</td>
<td>-</td>
</tr>
<tr>
<td>$c_3$</td>
<td>Constant used in k-e turbulent model</td>
<td>-</td>
</tr>
<tr>
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<td>$c_e$</td>
<td>Constant used in k-e turbulent model</td>
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</tr>
<tr>
<td>$E$</td>
<td>Turbulence kinetic energy dissipation rate</td>
<td>$m^2/s^3$</td>
</tr>
<tr>
<td>$g$</td>
<td>Gravity acceleration</td>
<td>$m/s^2$</td>
</tr>
<tr>
<td>$Gr$</td>
<td>Grashof number</td>
<td>-</td>
</tr>
<tr>
<td>$k$</td>
<td>Conductivity</td>
<td>$W/m.K$</td>
</tr>
<tr>
<td>$K$</td>
<td>Turbulent kinetic energy</td>
<td>$m^2/s^2$</td>
</tr>
<tr>
<td>$L$</td>
<td>Window height</td>
<td>m</td>
</tr>
<tr>
<td>$Nu$</td>
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</tr>
<tr>
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<td>Pressure</td>
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</tr>
<tr>
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<tr>
<td>$q$</td>
<td>Heat flux</td>
<td>$W/m^2$</td>
</tr>
<tr>
<td>$Q$</td>
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<td>W</td>
</tr>
<tr>
<td>$r$</td>
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<td>$R$</td>
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</tr>
<tr>
<td>$Ra$</td>
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<tr>
<td>$T$</td>
<td>Temperature</td>
<td>C/K</td>
</tr>
<tr>
<td>$u$</td>
<td>Velocity component in the window direction</td>
<td>$m/s$</td>
</tr>
<tr>
<td>$U_r$</td>
<td>Friction velocity</td>
<td>-</td>
</tr>
<tr>
<td>$v$</td>
<td>Velocity component normal to the window direction</td>
<td>$m/s$</td>
</tr>
<tr>
<td>$x$</td>
<td>Horizontal coordinate</td>
<td>-</td>
</tr>
</tbody>
</table>
\( y \)  
Vertical coordinate  
-  
\( y^+ \)  
Dimensionless wall distance  
-  
**Greek Symbols**  
\( \beta \)  
volumetric thermal expansion coefficient  
\( \frac{1}{K} \)  
\( \epsilon \)  
Eddy viscosity  
Pa.s  
\( \rho \)  
Fluid density  
\( \frac{kg}{m^3} \)  
\( \mu \)  
Dynamic fluid viscosity  
Pa.s  
\( \theta \)  
Blind slat angle  
-  
\( \varepsilon \)  
Turbulent dissipation rate  
\( \frac{m^2}{s^3} \)  
\( \nu \)  
Kinematic viscosity  
\( \frac{m^2}{s} \)  
**Subscripts**  
A  
Air in the room  
c  
convective  
f  
film  
H  
Based on height of window  
m  
mean  
s  
Blind slat  
T  
turbulent  
W  
window  
\( \infty \)  
Air far from window  
*  
Alternative dimensionless notation  
_  
Time average quantity  
'  
Fluctuating component
Chapter 1

Introduction

1.1 Introduction

There are a limited number of renewable energy sources on Earth. As a result, the importance of conserving the current energy supply for future generations, in addition to reducing the impact that the consumption of nonrenewable sources, like oil and coal, have on the environment, has made the conservation of energy an important topic.

The total energy that is consumed in a building can be divided into three parts: initial embodied energy, recurring embodied energy, and operational energy. The relationship between operating energy and embodied energy, for a typical office building in Canada (Vancouver and Toronto) for a 50-year life cycle, has been investigated by Cole, R.J. and Kernan, P.C. (1996) shown in Figure 1-1. These results (Figure 1-1) indicate that the amount of embodied energy is very small in comparison with operational energy. Operational energy is the energy that is used in a building for cooling, heating, ventilation, and lighting.

In a study that was done by Ayoub et al (2000), commercial and residential buildings were found to be a consumer of almost 30% of the total end-use energy in Canada. In an article published by Natural Resources Canada (2001), it has been shown that the largest components of end-use energy in a residential and commercial building is for space heating, as is demonstrated in Figures 1-2 and 1-3.
Figure 1-1 Components of energy use during a 50-Year Life Cycle of a typical office building in Vancouver and Toronto (Cole, R.J., Kernan, P.C., 1996)

Figure 1-2 End-use energy breakdown for residential buildings (Natural Resources Canada, 2001)
Windows, doors, and window’s covering have a very significant effect on energy usage in buildings for heating and cooling. By using appropriate fenestration and sun-shading strategies, there is the potential to reduce energy consumption in a building. Venetian blinds have the advantage of efficiently using daylight and reducing the use of artificial lighting, as well as contributing to reducing the cooling and heating loads. To help in developing a good understanding of the effects of Venetian blinds on the heat-transfer rate from a window to room and of its effects on air flow over a window several numerical and experimental studies have been undertaken. In most of these studies, the window is assumed to be as an isothermal flat vertical plate and the Venetian blind is assumed to be a series of equally spaced, thin, flat, horizontal surfaces with the same size. In the study of window and blind combinations, there are three heat-transfer processes:

- Natural convective heat transfer from the window
- Conductive heat transfer in blind slats
- Radiative heat transfer from window to the blind and the room
In the first part of the present study attention has only been given to natural convective flow and heat transfer. In the second part of the study the effect of a constant heat flux on slats has been considered. This heat flux is the result of solar energy which passes through the window and is absorbed by slats i.e. irradiated blind slats are considered.

Previous studies, related to the present study, are reviewed in following sections of this chapter.

1.2 Previous Studies on the effect of Venetian blinds on heat transfer from a window to room

Since the early 1950s, several studies have been undertaken in order to investigate the effect of Venetian blinds on the solar and thermal performances of a window. A study by Parmelee and Aubele (1952) was performed to investigate the effectiveness of louvered blinds in reducing heat gain, when the blinds are mounted inside and outside of a single pane of a sunlit glazing. In this study, a mathematical analysis that determined layer specific absorption and reflection and transmission of the shade layer was introduced. These properties were assumed to be functions of optical properties of the shade material and the geometry of fenestration systems such as slat angle, slat width, and pitch ratio, as well as the function of position of the sun to the window.

Parmelee et al. (1953a) presented another paper on the experimental verification of results obtained in their first paper. In this study the experimental results were in good agreement with the computational result for the case of direct radiation, but didn’t compare favorably with
values for the case of diffuse radiation. In their final work, Parmelee and Vild (1953b) presented a table of design data for use in predicting the shading coefficient of Venetian blinds.

Farber et al. (1963) developed a thermal resistance network to investigate convective and radiative gain. Because of the complexity of the geometry of this model, some assumptions had to be made in order to predict the convective heat-transfer coefficient but later, in another study by Pennington et al. (1964), it was shown that those assumptions were in good agreement with experimental results.

Owens (1974) mathematically analyzed the optical performance of blind. In this study, he implemented a matrix technique to find the optical properties of the blind and used an energy balance to determine the heat flow and heat absorption in each blind slat.

Several studies have also been undertaken on shading layers, which are located between the glazings. Ozisik and Schutrum (1960) did a study to investigate solar heat gain using the Parmelee et al. (1952) method. In their model, the total solar heat gain was the sum of absorbed energy in the blind and the transmitted solar energy. They didn’t consider different types of glasses or geometries, but the results of their model still agree with experimental results that were presented by Smith and Pennington (1964).

Rheault and Bilgen (1989) undertook a study of the overall heat transfer rates for a case in which Venetian blinds were installed between the panes of a double paned window for typical Canadian winter and summer condition. Conduction was neglected within the blind slats and only radiative and convective heat transfers were considered. In this study, the slat tips were at a relatively large distance from the window. Therefore, when the slats were all in a horizontal position, they didn’t have any effect on the cavity flow, but when the slats were positioned vertically they were considered as a dividing wall which made two side-by-side cavity flows. The
results showed that the presence of louvered blinds between window glazings will reduce heat loss from a window. Later, Rheault and Bilgen (1990) validated their numerical results with an experimental study.

An experimental study similar to the studies of Rhealt and Bilgen (1989, 1990) was undertaken by Garnet et al. (1995) using commercially-available Venetian blinds placed between window glazings. In their study, the slat tip to window space was smaller and the blind could be moved. It was observed that when the slat angles increased from a horizontal position to a vertical position, the heat loss from the window decreased. But when the slats were in a horizontal position, the heat loss was greater than when there was no fenestration. This is because the conduction effects in the blind slats were more significant than the convection effects in a horizontal slat case. Also, blind slats with higher angles blocked the radiation which caused a decrease in heat loss.

It is worth mentioning that Klem et al (1992, 1994a, 1994b, 1995, 1996a, 1996b, 1997) presented a comprehensive mathematical method for all types of fenestration devices with any combination and properties. However, this method is very complex and requires a large and accurate input of data, which “takes years to provide” (Collins, 2001). Therefore, no researcher had performed this task until now.

Today, because of the importance of this topic and its contribution to the conservation of energy, several studies have been performed on the effects of Venetian blinds on the free convective heat transfer from an indoor glazing system. These studies were completed as part of an NSERC strategic project, which involved Queen’s University, Ryerson University and the University of Waterloo.
Machin (1997) examined the change of local and overall convection coefficients from the surface of an isothermal plate as an indoor glazing surface with various blind-to-plate spacings and louver angles using interferometric and flow-visualization methods, a Mach-Zehnder Interferometer being used. The results of this experiment illustrated that Venetian blinds have a strong influence on the local heat-transfer coefficient over the length of the plate, but they slightly lowered the average convective heat transfer in comparison with for a vertical plate without blinds. Strong periodic variations in the local Nusselt number were also observed when the blind was located close to the plate surface.

Ye (1997) completed a numerical study in conjunction with Machin’s (1997) work at Queen’s University. In his two-dimensional finite elements model, radiation effects were neglected. The blind slats were assumed to be flat with zero thickness and non-conducting. The effect of Venetian blind slat angles and blind-to-plate spacing, on the average, and the local heat transfer rates were studied. It was found that the numerical results were lower than the experimental results of Machin (1997). Ye (1997) concluded that neglecting conduction in Venetian blinds caused this difference.

Another numerical study was performed by Phillips (1999), which included the effect of conduction and the shape and thickness of blind slats. In his first study, Phillips neglected radiative heat transfer and the results were in poor agreement with Machin’s (1997) experimental data except when the blind slat temperature was fixed. As a result, Phillips (1999) performed another study considering the coupled effects of radiation and convection heat transfer for the case in which the Venetian blind slats were horizontal. This was in good agreement with Machin’s (1997) interferometric results.
After gaining good agreement between experimental and numerical results for a window and indoor Venetian blinds under “nighttime” conditions, the study moved to a “daytime” situation in which the effects of incident solar energy were included. With the assumption that long- and short-wave radiation does not interfere with one another and can be examined independently, the effects of solar irradiation were considered from a thermal perspective in which absorbed solar energy was represented by heating the shade layer.

Duarte (2000) performed an interferometry and flow-visualization examination of the effects of a heated Venetian blind on free convective heat transfer from an indoor glazing surface using a Mach-Zehnder interferometer. He studied the effects of blind-slat angle, blind-to-plate spacing and blind irradiation on the heat-transfer rate from an isothermal plate at a temperature of 15K above the ambient temperature on the local heat-transfer coefficient. The results showed that heating of a Venetian blind has a significant impact on local heat-transfer coefficients.

Naylor and Duarte (2000) again used the interferometric method to study the free convective heat transfer from a vertical isothermal plate adjacent to a heated aluminum Venetian blind. Their results also showed that heat generation in a blind has a strong influence on local and average heat transfer coefficients along with the plate. It was also discovered that blind-to-plate spacing has a substantial impact on convection and when this spacing is small, the heated slats cause a pronounced periodic variation in the local heat-transfer distribution on the plate. When the blind heat generation was increased, it was found that the convective heat-transfer rate from the plate decreased and the average convection coefficient became more dependent on both slat angles and blind-to-plate spacing.

Collins (2001) undertook an investigation in which the effects of incident solar energy were studied. In this study, the system was examined using a solar-thermal separation method in which
short-wave solar radiation and long-wave radiative heat transfer could be examined separately for a given system. In his study, the influence of heated horizontal and rotatable louvers, representing irradiated blind slats, on the local and average convective and radiative heat transfer from a vertical isothermal surface representing a window, was examined. For cases in which blinds were at a higher temperature than the surface of the window, a negative and indeterminate heat-transfer coefficient would occur. As the louver was moved away from the surface, the local convective and radiative heat flux was less affected by the individual angle of each louver. Collins (2001) also undertook an experimental study using the Mach-Zehnder interferometer method to validate the aforementioned numerical model. The local and average convective heat-transfer coefficients were found to agree closely in magnitudes and trends.

1.3 Studies on the development of turbulent flow over a window

Most of the studies that have been undertaken on the effects of Venetian blinds on the thermal performance of a window have assumed that the natural convective flow will remain laminar over the window. However in large windows a transition to turbulent from laminar flow may occur.

In a study that was completed by Cable et al (2007), different turbulence models for the numerical study of mixed and natural forced convective flow in atriums have been evaluated to determine which model is in better agreement with the experimental data. All the turbulence models gave satisfactory results, while the choice of model to be used depended on the required computational power and time.

Oosthuizen (2009) completed a numerical two-dimensional study for conditions under which the transition from turbulent to laminar occurs. In this study, the convective heat-transfer
rate was investigated when the flow is turbulent. The effects of the recessed depth of a window on the development of turbulence flow and mean heat-transfer rate were also considered. This study didn’t include the radiation effects and the window was assumed to be an isothermal vertical flat plate. The recessed depth on the window was found to have no significant effect on the mean heat-transfer rate when the blind is fully opened, but when the blind is fully closed a significant reduction to the mean heat-transfer rate was observed especially at high Rayleigh numbers. In Figure 1-4, variation of the mean Nusselt number with the Rayleigh number for different recessed depths is shown.

![Figure 1-4 Variation of the mean Nusselt number with a Rayleigh number for a recessed window with a blind, Oosthuizen (2009)](image)

Oosthuizen and Naylor (2010) also investigated the effects of a partially open plane blind when the transition from laminar to turbulence occurs at “nighttime” condition. In this study, a model of recessed window covered by a partially open plane blind was considered. The window
was modeled on an isothermal vertical surface. In this two-dimensional study, the k-epsilon turbulence model with the full effects of buoyancy forces was used and the governing equations were solved using the finite volume-based CFD code, FLUENT. The Results of this study were in good agreement with previous studies done by Oosthuizen (2009) that indicate that when a blind is fully closed the mean heat-transfer rate from the window decreases. In addition, a nearly closed blind delays the development of turbulent flow to higher Rayleigh numbers. The procedure used in this study was the first to be applied to the case in which there was no blind and the window was nonrecessed. This methodology ensured a good prediction of where the transition happens and allowed for a comparison to available experimental results. A plot of these results is presented in Figure 1-5. This result is important, because the methodology that is being used in the current study is very similar to it.

![Figure 1-5 Variation of the mean Nusselt number with a Rayleigh number for a non-recessed window with no blind, Oosthuizen & Naylor (2010)](image)

Figure 1-5 Variation of the mean Nusselt number with a Rayleigh number for a non-recessed window with no blind, Oosthuizen & Naylor (2010)
1.4 Studies on three-dimensional effects on the flow over a window

One of the common assumptions in all existing studies of convective heat transfer in window-blind systems is that the flow is two-dimensional. However, in some cases, three-dimensional flow effects can become very significant. Oosthuizen (2007) numerically investigated a situation in which three-dimensional flow effects are important, while examining the way that they change to convective heat transfer in the window-blind arrangement. In his study, radiative heat transfer was neglected and the blind was assumed to be a simple-blind type. This study was done on two different cases: one in which the window was hotter than the room and one in which the window was colder than the room. The results indicate that three-dimensional flow effects were important in both cases, but were more significant when the width of a window was small and when the Ra value was low. Some of the results are presented in Figure 1-6 and 1-7.

![Graph showing variation of Nu with dimensionless plate width W for a "hot" and a "cold" window for Ra= 1.0E5 and a dimensionless window-blind gap of 0.08, Oosthuizen (2007)]
It will also be seen that for the higher Rayleigh number, the Nusselt number values for the 'hot' the 'cold' window cases are essentially equal at the higher values of $W$ considered, i.e., as the effectively two-dimensional flow situation is approached. At the lower Rayleigh number, the Nusselt number values for the 'hot' window case are always higher than those for the 'cold' window case although the percentage difference between the $Nu$ values for the two cases decreases as $W$ increases.

### 4. Conclusions

The results of this study indicate that the basic form of the variation of the mean Nusselt number with the dimensionless parameters $w$, $W$, and $Ra$ is the same for the 'hot' and 'cold' window cases. However, the effect of the dimensionless window-blind gap $w$ on the Nusselt number varies significantly between the 'hot' and 'cold' cases. The lack of numerical studies on the effect of Venetian blinds on the transition from a laminar to turbulent flow over windows and its effect on the thermal performance of windows, were the main motivation for the current study. It was intended that a two-dimensional numerical model of the development of a turbulent flow from a vertical isothermal recessed surface (representing a window) with adjacent heated louvers (representing irradiated blind slats) would be created. The development of a computer model for this situation, using a finite volume-based CFD code, FLUENT, will help to obtain a better understanding of the effect of Venetian blinds on windows.

**1.5 Research objectives**

In Section 1.3, it was stated that with large windows, transition from laminar to a turbulent flow may occur in the flow over the window. The lack of numerical studies on the effect of Venetian blinds on the transition from a laminar to turbulent flow over windows and its effect on the thermal performance of windows, were the main motivation for the current study. It was intended that a two-dimensional numerical model of the development of a turbulent flow from a vertical isothermal recessed surface (representing a window) with adjacent heated louvers (representing irradiated blind slats) would be created. The development of a computer model for this situation, using a finite volume-based CFD code, FLUENT, will help to obtain a better understanding of the effect of Venetian blinds on windows.
In the first part of this study, only convective heat transfer is considered and radiation effects are neglected. This situation is the so-called “nighttime” situation. It would also be assumed that there is no conduction along the slats. In the second part of this study, the irradiated slats are considered the heating at the blind slats being the results of solar radiation.

The goal of this study was to develop a CFD model of a room with a large window that would allow for an examination of slat-to-plate spacing, slat angles and heat generation in slats on the natural convective heat transfer. It would also consider mean heat transfer from an isothermal surface and on the development of a turbulent flow over the height of the plate.

1.6 Methodology

The investigation of published works on the CFD modeling of window-blind systems, the effects of Venetian blinds on heat transfer from windows, and the laminar-to-turbulent transition over the height of windows, with or without blinds, is presented in the current chapter.

In Chapter 2, the assumptions and governing equations that would apply over the solution domain will be studied. The k-epsilon turbulent model and the Boussinesq approach will also be studied in the same chapter.

The geometrical and meshing details of the model in GAMBIT and the assumption that is determined in FLUENT will be presented in Chapter 3. In addition, two validation cases will be presented in this chapter.

The results of the numerical study, including charts, temperature, and velocity contours, and other tables and contours, will be presented and discussed in Chapter 4.
This chapter will summarize the discussions and conclusions presented in previous chapters, in addition to recommendations for future studies.
Chapter 2

Background theory and Numerical model

2.1 Introduction

Over the last two decades Computational Fluid Dynamics (CFD) has been one of the most rapidly developing areas in engineering. The development of commercial CFD codes has increased both computational and processing power and has played an important role in industry by providing robust solutions for flow and heat transfer problems.

In using the commercial code FLUENT, the computational domain is established in GAMBIT in which the grid is generated and the boundary conditions for the solution domain are defined. The model is then exported to FLUENT. FLUENT solves the governing equations of the flow using the finite-volume method (Beausoleil-Morrison, 2000).

In the following sections, the governing equations, the turbulence models, and important information about the numerical model, are reviewed.

2.2 Boussinesq Approximation

Natural convective-flow problems, due to their complicated nature, are challenging. When setting up a CFD-based problem involving natural convection, the density must be modeled as a variable since changes in density are what generate buoyancy forces which cause the flow. These changes are analyzed in the present study using the Boussinesq approximation.

In the Boussinesq approximation it is assumed that only density changes due to temperature variations are important but changes in all other fluid properties are negligible. The
Buoyancy force is what causes the fluid motion in free convection. Density changes, due to temperature variation in a fluid at constant pressure, are represented by $\beta$ the volumetric thermal expansion coefficient, which is a thermodynamic property of a fluid;

$$\beta = -\frac{1}{\rho} \left( \frac{\partial \rho}{\partial T} \right)_p$$

(2.1)

And can be approximated by;

$$\beta \approx -\frac{1}{\rho} \frac{\rho_\infty - \rho}{\rho_\infty - T}$$

(2.2)

It should be stressed that the Boussinesq approximation can only be used when the temperature variation in the solution domain is not large $\beta(T - T_\infty) \ll 1$.

### 2.3 Governing equations

One of the most common examples of natural convection occurs when a vertical isothermal flat plate is subjected to a cooler or warmer surrounding fluid. Figure 2-1 shows the temperature and velocity boundary layers for a case in which the fluid is cooler. When the surrounding fluid is warmer, the physical appearance is inverted but the method of analysis remains the same. The assumptions in the present study are:

- Two-dimensional flow
- Steady flow
- Boussinesq approximation
For this case, the governing equations are:

Continuity: \[ \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \] \hspace{1cm} (2.3)

Energy: \[ u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = k \rho C_p \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \] \hspace{1cm} (2.4)

Momentum in x direction: \[ u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \theta \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) + \beta g (T - T_\infty) \] \hspace{1cm} (2.5)
2.3.1 Dimensionless parameters

The dimensionless parameters involved in free-convective flow and heat transfers are obtained by non-dimensionalizing the governing equations. The following dimensionless variables are introduced as;

\[ x^* \equiv \frac{x}{L}, \quad y^* \equiv \frac{y}{L} \]
\[ u^* \equiv \frac{u}{\sqrt{g \beta (T_w - T_\infty)L}}, \quad v^* \equiv \frac{v}{\sqrt{g \beta (T_w - T_\infty)L}} \]
\[ T^* \equiv \frac{T - T_\infty}{T_w - T_\infty} \]

Substituting these variables in to the governing equations will provide the following dimensionless parameters;

\[ Pr = \frac{c_p \mu}{k}, \text{ the Prandtl number that is the ratio of kinematic viscosity over thermal diffusivity; } \]
\[ Gr = \frac{\beta g (T_w - T_\infty)L^3}{\theta^2}, \text{ the Grashof number, which indicates the ratio of the buoyancy force to the viscous force acting on the fluid} \]

In the example of the vertical isothermal flat plate that is illustrated in Figure 2-1 at some distance from the leading edge, a transition from laminar to turbulent flow may occur depending on fluid properties and the thermal gradient. In this case, another dimensionless number is defined;
In calculating the Rayleigh number all the fluid properties are constant and at their film temperature is defined by: 

\[ T_f = \frac{T_w + T_\infty}{2} \]

In some textbooks (Oosthuizen, Naylor, 2007), it has been assumed that the transition occurs when the Rayleigh number is equal to \(10^8\).

### 2.3.2 Empirical correlations

The Nusselt number for free convection over an isothermal vertical flat plate is a function of the Grashof and Prandtl numbers. The following equation is based on engineering experimental data that is applicable for many geometric cases:

\[
Nu_L = \frac{hL}{k} = C(GrPr)^n
\]  

(2.6)

Where \(n\) is equal to \(\frac{1}{4}\) for laminar flow and \(\frac{1}{3}\) for turbulent flow. And \(C\) is equal to 0.59 for laminar and 0.13 for turbulent flow.

### 2.4 Turbulence modeling

Turbulence is characterized by random and chaotic fluid motion. When turbulence is present, it usually dominates all other flow phenomena and results in increased energy dissipation, mixing, heat transfer, and drag (George). Solving the aforementioned governing equations for turbulent flow to obtain the variation of the flow variables with time requires very high computational resources. Therefore, in most of the analysis of turbulent flows, the Reynolds average equations are used in which the variables are composed at their mean, or time-averaged values and the fluctuation from their value i.e.
\( u = \bar{u} + \hat{u}, \quad v = \bar{v} + \hat{v}, \quad T = \bar{T} + \hat{T}, \quad p = \bar{p} + \hat{p} \)

Where \( \bar{\text{---}} \) represents the mean (time-averaged) quantity and \( \hat{\text{---}} \) is the fluctuating component of this mean value, which is time-dependent. When these values are substituted into the governing equations and the equations are time average, the following are obtained:

**Continuity:**
\[
\frac{\partial \bar{u}}{\partial x} + \frac{\partial \bar{v}}{\partial y} = 0 \tag{2.7}
\]

**Momentum:**
\[
\bar{u} \frac{\partial \bar{u}}{\partial x} + \bar{v} \frac{\partial \bar{u}}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \frac{\partial}{\partial y} \left( \bar{\theta} + \epsilon \right) \frac{\partial \bar{u}}{\partial y} \tag{2.8}
\]

**Energy:**
\[
\bar{u} \frac{\partial T}{\partial x} + \bar{v} \frac{\partial T}{\partial y} = \nu \frac{\partial}{\partial y} \left( \frac{1}{Pr} + \frac{\epsilon}{Pr_T} \right) \frac{\partial T}{\partial y} \tag{2.9}
\]

Where \( \epsilon \) is known as the eddy viscosity, and \( Pr_T \) is the turbulent Prandtl number.

It should be noted that these two terms are not properties of the fluid involved alone, but they depend primarily on the turbulence structure at the point under consideration.

**2.4.1 k-epsilon turbulence model**

The governing equations for turbulent flows, in addition to the mean flow variables, depend on the fluctuating turbulent quantities. Therefore, there are more variables in these governing equations than there are equations. In order to solve turbulence problems, turbulence models are introduced.

The k-epsilon turbulence model has been adopted in the present study, due to its generality and the good agreement given using this model with experimental results as found throughout case studies conducted by Cable et al. (2007). This model consists of two extra transport equations which give the turbulent properties of the flow. The first transport variable is
turbulent kinetic energy, $K$ that determines the energy in the turbulence while the second transport variable is the turbulent dissipation rate, $E$. These two variables are determined by the following equations;

\[
\begin{align*}
\overline{u} \frac{\partial K}{\partial x} + \overline{v} \frac{\partial K}{\partial y} &= \varepsilon \left( \frac{\partial u}{\partial y} \right)^2 + c_1 \frac{\partial}{\partial y} \left( \varepsilon \frac{\partial K}{\partial y} \right) - E \\
\overline{u} \frac{\partial E}{\partial x} + \overline{v} \frac{\partial E}{\partial y} &= c_3 \varepsilon \left( \frac{\partial u}{\partial y} \right)^2 \frac{E}{K} + c_4 \frac{\partial}{\partial y} \left( \varepsilon \frac{\partial E}{\partial y} \right) - c_5 \frac{E^2}{K}
\end{align*}
\]

(2.10)  

(2.11)

Where $\varepsilon = c_e \frac{K^2}{E}$

The $K-E$ turbulent model, which is known as k-epsilon, contains some empirical constants that have been determined through experiments on simple flows. It is assumed here that these constants are;

\[c_1 = 1, \quad c_3 = 1.44, \quad c_4 = 0.77, \quad c_5 = 1.92, \quad c_e = 0.09\]

The standard k-epsilon model, with full effects of buoyancy forces, has been adopted in this study. In this study the default value of $Pr$ is equal to 0.7.

**2.5 Heat generation in the slats**

Heat generation in the slats in the present study is the result of solar radiation passing through the window and falling on, and being absorbed by, the slats of the blind. The situation considered in the present study does not closely model the real situation involving heat transfer from Venetian-blind-covered windows. In the real situation both convective and radiant heat
transfer are involved, but here attention is restricted to convective heat transfer. In a real blind situation, the solar radiation often does not fall uniformly over the blind. However, in the present study, in order to examine in a general way the effect of heat generation on the heat transfer rate, a uniform rate of heat generation has been assumed over the entire slat.

### 2.6 Numerical model

The accuracy of the results obtained using FLUENT is strongly dependent on the mesh configuration, as well as other inputs such as the turbulence model, nearby wall functions, and interpolation schemes.

#### 2.6.1 Mesh configuration

The mesh size that is being used should be able to resolve both velocity and thermal boundary layers. In addition to the mesh size, the mesh shape is also important. For a rectangular or quad mesh, the cells shape should not be distorted and should, ideally, be square. For triangular mesh, the cell shape should be close to an equilateral triangle. It should be kept in mind that the higher the number of cells, the greater the numerical time; therefore, the total cells count should be kept as low as possible.

Domain size is important since the bigger the domain size; the more computational power is required. Furthermore, if the domain size is too small it will not be able to give true results; therefore, the domain size should be large enough so that the effects of geometry on the flow field are not felt at the domain boundaries. The domain geometry will be discussed in the next chapter.
2.6.2 Near wall region

The mesh size near the boundary is critical. Turbulence quantities close to the wall are negligible, because the turbulent velocity fluctuations, normal to the wall, are damped by the presence of the wall (Oosthuizen and Naylor, 2007). This region is termed the “laminar sublayer”. The laminar sublayer is very thin and the variation of shear stress and heat-transfer rate through this layer are negligible; therefore, the velocity and temperature distributions in this region are linear.

The variable $y^+$ the dimensionless wall distance is introduced here and is used to find a suitable grid size near the walls and represents the distance of the cell center to the wall. This variable is given by the following equation;

$$y^+ = \frac{y U_r}{\bar{v}}$$  \hspace{1cm} (2.12)

Where $U_r$ is the “friction velocity”

The laminar sublayer extends from $y^+=0$ to $y^+=5$. When $y^+$ increases from 5, the turbulence effects become important. The size of the turbulent “eddies” near the wall increases linearly with the distance from the wall. Between $y^+=5$ and $y^+=30$, both the molecular and turbulent stresses are important; as a result, this region is termed a “buffer” region. When $y^+$ is higher than 30, the turbulence effects are dominant on the flow. This outer region is called the “fully turbulent layer” or “log-law region”. In Figure 2-2, the three main regions near the wall are shown.
The k-epsilon turbulence model, with standard wall function, is used in the present study, but to resolve the flow in this region standard wall functions require a range of $y^+$ between 30 and 300 for an accurate prediction of turbulent flow over a wall.

Figure 2-2 Subdivisions of the near-wall region, $U_t$ being the friction velocity (FLUENT, 2006)

### 2.6.3 Numerical solution method

The governing equations, together with the equations for turbulent kinetic energy and rate of dissipation of turbulent kinetic energy, forms, in general a set of six differential equations that describe the flow in the control volume. As it would be impossible to analytically solve these
equations, a numerical solution is obtained using the finite-volume method. There are two numerical models in FLUENT:

- Pressure-based solver
- Density-based solver

In both methods, the velocity field is obtained from the momentum equations. In the present study, the pressure-based solver is adopted. In either method, FLUENT will solve the governing integral equations for the conservation of mass and momentum and, when appropriate, for energy and other scalars such as turbulence and chemical species. In both cases a control-volume-based technique is used that consists of (FLUENT, 2006):

- Division of the domain into discrete control volumes using a computational grid.
- Integration of the governing equations on the individual control volumes to construct algebraic equations for discrete dependent variables ("unknowns") such as velocities, pressure, temperature, and conserved scalars.
- Linearization of the discretized equations and solution of the resultant linear equation system to yield updated values of the dependent variables.

Figure 2-3 demonstrates the segregated solution method for solving these algebraic equations sequentially.
The pressure-based solver uses a solution algorithm in which the governing equations are solved sequentially (i.e., segregated from one another). Because the governing equations are nonlinear and coupled, the solution loop must be carried out iteratively in order to obtain a converged numerical solution.
2.6.4 Solver setting

In the present study, the flow is assumed to be in a steady state and two-dimensional. The PRESTO (pressure-staggering option) scheme in FLUENT was used for pressure interpolation. PRESTO is recommended for problems with strong body forces, such as gravitational forces and pressure gradients, as in high Rayleigh number convection (FLUENT, 2006). FLUENT documentation also includes guidelines for solving problems that involve natural convection. Other solver settings are as follows:

- Pressure-based, two-dimensional, and steady solver
- Second order upwind discretization for momentum, turbulent kinetic energy, turbulent dissipation rate, and energy
- Simple pressure-velocity coupling
- Under-relaxation factors values indicated in the following table:

<table>
<thead>
<tr>
<th>Under-relaxation factor</th>
<th>Pressure</th>
<th>Density</th>
<th>Body force</th>
<th>Momentum</th>
<th>Turbulent kinetic energy</th>
<th>Turbulent dissipation rate</th>
<th>Turbulent viscosity</th>
<th>Energy</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.3</td>
<td>0.8</td>
<td>0.8</td>
<td>0.5</td>
<td>0.8</td>
<td>0.8</td>
<td>0.8</td>
<td>0.8</td>
</tr>
</tbody>
</table>

Table 2-1 Under-relaxation factors modified in FLUENT for the current study

- Convergence criteria of 0.001
Chapter 3

CFD modeling

3.1 Physical description of the model

The solution domain used in the present work, is shown in Figure 3-1. The present numerical model was based on the experience gained in previous studies of window-blind systems. In the solution domain, a vertical isothermal flat surface (IH) represents the interior surface of a window with adjacent louvers as blind slats (Collins, 2001). The blind slats were modeled as a series of equally spaced thin and flat surfaces, which were pivoted around their center point (Sun, 2002).

In the present study, the system geometry consisted of an isothermal plate and horizontal flat louvers. The window, IH, is idealized as a vertical isothermal plate with height (l) and temperature (T_w), which is lower than ambient room temperature (T_A). The window is recessed in a frame, which is represented by horizontal adiabatic surfaces (GH and IJ) that are mounted in vertical walls (FG and AJ) that are represented by the vertical adiabatic surfaces and exposed to a large surrounding room. The slats are pivoted around their center point and have an angle (θ) with respect to the horizontal that changes. The effects of changes distance of the center point of the slat to the plate (r) have also been examined. The slats are in close proximity to the isothermal surface, but are not in contact with it. In part of the present study there is a uniform heat generation in the slats (Q_s) to simulate the solar radiation absorbed by the slats.
3.2 Mesh generation

The accuracy of the results obtained from FLUENT is strongly dependent on the mesh configuration and size; a coarse grid will lead to large numerical errors and a fine grid is computationally expensive, taking a long time to converge. Moreover, as explained in Section 2.6.2, the required $y^+$ values for standard wall functions are between 30 and 300. To obtain the required $y^+$ values, the cells adjacent to the window had to be very small. To avoid excessive
computational effort, a mesh boundary layer was adopted to blend the fine mesh near the window to the coarser mesh found in the core region.

A mesh boundary layer was developed for the region close to the window, shown in Figure 3-2. In creating this boundary layer, the first row was located 1.0 mm away from the window and the next four rows increase with a 1.2 growing factor.

![Figure 3-2 Boundary layer for an isothermal vertical wall](image)

By using the mesh boundary layer it was possible to make a more accurate prediction of flow and convective heat transfer near the window. In Figure 3-3 a typical variation of $y^+$ over the window’s height with horizontal blind slats is shown.
3.3 Mesh independency testing

A mesh independency test was undertaken to ensure that the results derived were independent of the number of mesh points used. As described previously, increasing the number of cells will increase the computational time. By using this rule of thumb, increasing the cell numbers by a factor of 2 will increase the computational time by a factor of 4 (Collins, 2001). In addition, if the cell numbers becomes large it will use the entire RAM of the computer. This will lead to using the virtual memory for additional information, which will increase the time required to obtain the solution.
A starting point, attention was given to the grid density that was used by Collins (2001). Considering the fact that in that study the numbers of blind slats was 17 and in the present study it is 31, and because the solution domain in that study was smaller than that used in the present study, the starting number of nodes for the grid independency check was 138,600. Six different grids were adopted for the mesh independency test which was undertaken for the case where there was no blind. In Table 3-1, the number of cells for each grid is shown.

<table>
<thead>
<tr>
<th>Grid</th>
<th>Cell count</th>
</tr>
</thead>
<tbody>
<tr>
<td>Grid 1</td>
<td>15400</td>
</tr>
<tr>
<td>Grid 2</td>
<td>61600</td>
</tr>
<tr>
<td>Grid 3</td>
<td>138600</td>
</tr>
<tr>
<td>Grid 4</td>
<td>248400</td>
</tr>
<tr>
<td>Grid 5</td>
<td>385000</td>
</tr>
<tr>
<td>Grid 6</td>
<td>554400</td>
</tr>
</tbody>
</table>

Table 3-1 Number of cells for a grid independency test

The variation of the mean Nusselt number for the window with increasing Rayleigh numbers for each of these grids is shown in Figure 3-4.
As illustrated in the above figure, the greatest difference between the mean Nusselt numbers in the grids occurred in the region at which the Rayleigh number is relatively high. The percentage difference between different grids is shown in Table 3-2.

![Figure 3-4 Variation of the mean Nusselt number with a Rayleigh number for different grids for a recessed window without a blind](image)

<table>
<thead>
<tr>
<th>Rayleigh Number</th>
<th>Grid 1-2</th>
<th>Grid 2-3</th>
<th>Grid 3-4</th>
<th>Grid 4-5</th>
<th>Grid 5-6</th>
</tr>
</thead>
<tbody>
<tr>
<td>2e7</td>
<td>0.649%</td>
<td>0.135%</td>
<td>0.057%</td>
<td>0.018%</td>
<td>0.003%</td>
</tr>
<tr>
<td>2e8</td>
<td>0.952%</td>
<td>0.171%</td>
<td>0.091%</td>
<td>0.022%</td>
<td>0.006%</td>
</tr>
<tr>
<td>2e9</td>
<td>0.504%</td>
<td>0.166%</td>
<td>0.074%</td>
<td>0.034%</td>
<td>0.014%</td>
</tr>
<tr>
<td>2e10</td>
<td>1.034%</td>
<td>0.984%</td>
<td>0.877%</td>
<td>0.767%</td>
<td>0.203%</td>
</tr>
<tr>
<td>2e11</td>
<td>3.080%</td>
<td>1.917%</td>
<td>1.392%</td>
<td>1.066%</td>
<td>0.349%</td>
</tr>
</tbody>
</table>

Table 3-2 Percentage difference between the mean Nusselt numbers for different Rayleigh numbers in different grids

34
Based on these results, it was determined that it would be best to use the cell count of grid 3. At the higher density grids the computational time was expected to be too large.

### 3.4 Boundary Conditions

The last step after developing the mesh is specifying the boundary conditions that are required to get a solution. Zone-type specifications in GAMBIT define the physical and operational characteristics of the model at its boundaries within specific regions of its domain (FLUENT, 2006). In Table 3-3, the zone-types defined in GAMBIT for the present model are presented.

<table>
<thead>
<tr>
<th>Name</th>
<th>Id</th>
<th>Zone type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Window</td>
<td>IH</td>
<td>Wall</td>
</tr>
<tr>
<td>Adiabatic walls</td>
<td>FG, GH, AJ, JI</td>
<td>Wall</td>
</tr>
<tr>
<td>Roof and floor</td>
<td>AB, EF</td>
<td>Wall</td>
</tr>
<tr>
<td>Blind slat</td>
<td>Slats</td>
<td>Wall</td>
</tr>
<tr>
<td>Outlet</td>
<td>BE</td>
<td>Pressure outlet</td>
</tr>
</tbody>
</table>

**Table 3-3 Zone type selection**

After exporting the mesh to FLUENT, the boundary conditions need to be specified. In Figure 3-5, the boundary IH represents the window. A no-slip boundary condition has been applied to it and, since it is an isothermal window, a constant temperature has also been applied. Boundaries HG, GF, AJ and IJ are adiabatic walls which have similar conditions as the window, except that there is zero heat flux in them. The “pressure outlet” boundary condition was used to define the domain boundaries in cases of unbounded subsonic flows in Section BE. This boundary condition requires the specification of static pressure at the outlet boundary, which was
set to 0. A set of "backflow" conditions is also specified in FLUENT should the flow reverse direction at the pressure outlet boundary during the solution process. The backflow temperature was set for this case, because it entails energy calculation. The backflow temperature is equal to air ambient temperature, while the temperature of the air inside the room is equal to the ambient temperature. Sections EF and AB represents the upper and lower boundaries that are also assumed to be adiabatic walls with no-slip condition. Finally, at the surface of the slats, a no-slip condition applies. In the situation in which there is heat generation in the slats, the amount of heat flux was added to the boundary condition.

Figure 3-5 Boundary conditions on a solution domain
As previously discussed, fluid properties have been assumed to remain constant except for the density change with temperature which gives rise to buoyancy forces. This has been treated using the Boussinesq approach. The flow properties are taken as these at the mean film temperature, and are given in Table 3-4.

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Density $\frac{kg}{m^3}$</th>
<th>$C_p \frac{j}{kg \cdot K}$</th>
<th>Thermal conductivity $\frac{W}{m \cdot K}$</th>
<th>Viscosity $\frac{kg}{m \cdot s}$</th>
<th>Thermal expansion coefficient $\beta \left( \frac{1}{K^0} \right)$</th>
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</thead>
<tbody>
<tr>
<td>Air</td>
<td>1.225</td>
<td>1006.43</td>
<td>0.0242</td>
<td>1.789e-05</td>
<td>0.00333</td>
</tr>
</tbody>
</table>

Table 3-4 Constant flow properties

### 3.5 Validation

#### 3.5.1 Validation using Collins (2001) results

In order to validate the numerical models, results given using the present solution procedure are compared to the results obtained in an experimental Collins (2001) study of a window-blind system. Since the experimental system uses different dimensions from those considered above, another model was produced in GAMBIT, Figure 3-6. This model is similar to Collins (2001) physical model; except that all the boundary conditions discussed above for the present study were used.
A comparison of the mean convective heat transfer from the window to the room was made for the two different cases, presented in Table 3-5.

<table>
<thead>
<tr>
<th>Case</th>
<th>$Q_s \frac{W}{m^2}$</th>
<th>$T_w (K)$</th>
<th>$T_A (K)$</th>
<th>$\theta$</th>
<th>$r (mm)$</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>125</td>
<td>297</td>
<td>283</td>
<td>0</td>
<td>15.4</td>
</tr>
<tr>
<td>2</td>
<td>125</td>
<td>297</td>
<td>283</td>
<td>45</td>
<td>15.4</td>
</tr>
</tbody>
</table>

Table 3-5 Validation case specification
For clarity in presenting the results, the dimensional flux is presented instead of the Nusselt number in Table 3-6. There is negative convective heat flux for the plate, since the isothermal plate (window) is colder than the surroundings.

<table>
<thead>
<tr>
<th>Case</th>
<th>( \overline{q_c} \text{ Experimental } (\frac{W}{m^2}) )</th>
<th>( \overline{q_c} \text{ Numerical } (\frac{W}{m^2}) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>-75.1</td>
<td>-76.81</td>
</tr>
<tr>
<td>2</td>
<td>-53.1</td>
<td>-41.78</td>
</tr>
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</table>

Table 3-6 Comparison of Numerical and experimental results

In the first case, the experimental data from Collins (2001) correlated well with the numerical results, the percentage difference is 2.27%. In the second case, the percentage difference between the experimental and numerical result is higher being approximately 20%. This discrepancy might be due to the experimental model and limitations in the experimental method (Collins, 2001). Furthermore, radiation effects and the effects of conduction in the slats will also cause discrepancies.

3.5.2 Validation using Oosthuizen and Naylor (2010) results

An indication of the adequacy of the present study numerical model is also given by comparing the results obtained here for the case of natural convection from a recessed vertical isothermal flat plate without a blind with an existing numerical and experimental results of Oosthuizen and Naylor (2010) numerical model (Figure 3-7).
Numerical results obtained in the current study are in good agreement with the trend of the Oosthuizen and Naylor (2010) results. There are some differences in magnitude; the present results being lower in the turbulent flow region than the Oosthuizen and Naylor results and the present numerical model gives higher values in the laminar flow region. The transition region covers the same range of Rayleigh numbers in all three cases considered.

Figure 3-7 Comparison of the current model's numerical result with Oosthuizen and Naylor’s (2010) result for variation of the mean Nusselt Number with a Rayleigh number without blind case
Chapter 4

Results and discussion

4.1 Introduction

This chapter presents a comparison of the numerical results that have been obtained. The main focus is the convective heat transfer for the so-called “nighttime” condition. Contour plots based on the numerical results are also presented. In the last section, the effects of heat generation in the slats, resulting from irradiation of the slats, is studied.

For the “nighttime” condition, the effect of changes in the model parameters on the transition from laminar to turbulent over the window’s height, and variation of the local and mean Nusselt number are investigated.

4.2 Model parameters

The solution has the following parameters;

1- The Rayleigh number, $Ra_R$, based on the plate height and the overall temperature difference;

$$Ra_L = \frac{\beta g (T_A - T_w) L^3}{\alpha}$$

2- The Prandtl number. The results have been obtained for Pr=0.7 (i.e., the approximate value for air).
3- The dimensionless blind-to-slat spacing, which is measured from the slat pivot point to the window;

\[ R = \frac{r}{L} \]

4- The blind slat angle, \( \theta \), results being obtained have four values of 0, -22.5, -45, or +45 degrees

5- Where there is no blind or if the blind is fully closed.

Results have been obtained for the case in which the “window” surface is at a lower temperature than the temperature of the air in the room. At the second part of the present study, results have also been obtained for the case where there is a heat generation in the blind slats.

The mean and local heat transfer rates from the window, \( q_m \) and \( q \), have been expressed in terms of the mean and local Nusselt numbers, \( Nu \) and \( Nu_y \), i.e. in terms of:

\[ Nu = \frac{q_m L}{(T_A - T_W)k} \]

And

\[ Nu_y = \frac{q_y}{(T_A - T_W)k} \]
4.3 Effect of blind-to-window spacing on the local and mean Nusselt number

Variations of the mean Nusselt number with the Rayleigh number for different recessed depths for four different blind slat angles are presented in Figures 4-1 through 4-4. The results for the cases in which the blind is fully closed are demonstrated by the results given in Figure 4-5.

Figure 4-1 Variation of the mean Nusselt number with a Rayleigh number for $\theta=45^\circ$ for three values of $R$
Figure 4-2 Variation of the mean Nusselt number with a Rayleigh number for $\theta = 45^\circ$ for three values of $R$

Figure 4-3 Variation of the mean Nusselt number with a Rayleigh number for $\theta = 22.5^\circ$ for three values of $R$
Figure 4-4 Variation of the mean Nusselt number with a Rayleigh number for $\theta=0^\circ$ for three values of $R$

Figure 4-5 Variation of the mean Nusselt number with a Rayleigh number for a fully closed blind for three values of $R$
According to Figures 4-1 through 4-4, when the Rayleigh number is high the mean Nusselt number is almost the same as for a plane recessed vertical surface. When the Rayleigh number is low, however decreasing the blind-to-window spacing, R reduces the mean Nusselt number. When the blind is fully closed, Figure 4-5, the mean Nusselt number is significantly lower than in the other case. In this case the mean Nusselt number varies only slightly with variation of R.

The effect of the variation of blind-to-window spacing, R, on the velocity boundary layers can be seen from the velocity contours for the highest and the lowest Rayleigh numbers for three different R values for $\theta=0$ from Figures 4-6 through 4-8.
Figure 4-6 Velocity contours for a horizontal blind slat and R=0.08 for Ra=2.0E7 (up) and Ra=2.0E11 (down)
Figure 4-7 Velocity contours for a horizontal blind slat and $R=0.05$ for $Ra=2.0\times10^7$ (up)

and $Ra=2.0\times10^{11}$ (down)
Figure 4-8 Velocity contours for a horizontal blind slat and $R=0.03$ for $Ra=2.0E7$ (up) and $Ra=2.0E11$ (down)
An observation of the results indicates that decreasing the Rayleigh number will cause an increase in thickness of the velocity boundary layer. This will decrease the mean Nusselt number, especially when at smaller blind-to-window spacings. This outcome was expected because the intensity of the flow over the window decreases as a result of the presence of the blind, especially when the blind is close to the window. The effect of the presence of the blind on the intensity of the flow over the window can be seen by comparing the previously given velocity contours with those for the case where there is no blind, these being presented in Figure 4-9. The Rayleigh number is equal to 2.0E07.

Figure 4-9 Velocity contours isothermal recessed vertical window without a blind where Ra=2.0E7
The variation of local Nusselt number with dimensionless window height for the highest and the lowest Rayleigh numbers considered for various slat angles are presented in Figures 4-10 through 4-13.

**Figure 4-10** Variation of the local Nusselt number with a dimensionless window height for a Rayleigh number equal to 2.0E7 for \( \theta = 45^\circ \), for different R values

**Figure 4-11** Variation of the local Nusselt number with a dimensionless window height for a Rayleigh number equal to 2.0E11 for \( \theta = 45^\circ \), for different R values
Figure 4-12 Variation of the local Nusselt number with a dimensionless window height for a Rayleigh number equal to 2.0E7 for $\theta=0^\circ$, for different $R$ values.

Figure 4-13 Variation of the local Nusselt number with a dimensionless window height for a Rayleigh number equal to 2.0E11 for $\theta=0^\circ$, for different $R$ values.
Figures 4-10 through 4-13 show periodic variation in the local Nusselt number when the blinds are very close to the window.

4.4 Effect of blind slat angle on local and mean Nusselt number

The effect of blind slat angle, $\theta$, on the variation of mean Nusselt number with Rayleigh number is illustrated by the results presented in figures 4-14 through 4-16.

![Graph](image)

**Figure 4-14** Variation of the mean Nusselt number with a Rayleigh number for $R=0.03$ and different values of $\theta$.
Figure 4-15 Variation of the mean Nusselt number with a Rayleigh number for $R=0.05$ and different values of $\theta$

Figure 4-16 Variation of the mean Nusselt number with a Rayleigh number for $R=0.08$ and different values of $\theta$
As indicated in these figures the blind slat angle changes do not have a big effect on the mean Nusselt number, except when the blind is in close proximity to the window and when the angle is small and the tip of the blind is close to the window. The velocity contours when the Rayleigh number is at its lowest for four different blind slat angles and when the blind is close to the window, are shown in Figures 4-17 through 4-20.

Figure 4-17 Velocity contours for \( \theta=\pm 45 \) blind slat and \( R=0.03 \) in which \( Ra=2.0E7 \)
Figure 4-18 Velocity contours for a $\theta=-45$ blind slat and $R=0.03$ in which $Ra=2.0E7$

Figure 4-19 Velocity contours for a $\theta=-22.5$ blind slat and $R=0.03$ in which $Ra=2.0E7
Figure 4-20 Velocity contours for a $\theta=0$ blind slat and $R=0.03$ in which $Ra=2.0\times10^7$

The variation of the local Nusselt number with a dimensionless window height is presented in Figures 4-21 and Figure 4-22 for two different Rayleigh numbers.
Figure 4-21 Variation of the local Nusselt number with a dimensionless window height for Rayleigh number equal to 2.0E7 and R=0.03 for different θ values

Figure 4-22 Variation of the local Nusselt number with a dimensionless window height for Rayleigh number equal to 2.0E11 and R=0.03 for different θ values
4.5 Effect of a blind on laminar to turbulent transition

4.5.1 Effect of a blind to window spacing on transition

The position at which turbulence develops in the flow is dependent on the blind-to-window spacing. This is illustrated by the results given in Figure 4-23 through 4-25, which show typical contours of the turbulent viscosity ratio, $\frac{\vartheta_T}{\vartheta}$, for a given Rayleigh number and blind slat angle for three values of the dimensionless blind to window spacing, R.

Figure 4-23 Distribution of the turbulence viscosity ratio for R= 0.08, a Rayleigh number of 2.0E11 and a horizontal blind slats
Figure 4-24 Distribution of the turbulence viscosity ratio for R= 0.05, a Rayleigh number of 2.0E11 and a horizontal blind slats

Figure 4-25 Distribution of the turbulence viscosity ratio for R= 0.03, a Rayleigh number of 2.0E11 and a horizontal blind slats
As seen in Figure 4-25, for the smallest blind-to-window spacing (i.e., 0.03) turbulence developed in the flow over the outside of the blind, while with the largest blind opening (i.e., 0.08) the turbulence developed in the flow between the window and blind.

The figures giving the variation of the mean Nusselt number with a Rayleigh number presented in Section 4.3 indicate that the transition will occur at approximately the same Rayleigh number range as for the flow over a plane recessed surface without a blind, except when the blind is very close to the window. However, it must be noted that the Rayleigh number at which transition begins does tend to increase somewhat as R decreases.

The transition range can be deduced from the variation of Nusselt number with Rayleigh number. Variations cease to be similar to the laminar flow flat plate variation by noting the values of Rayleigh at which the Nusselt number variation becomes similar to the turbulent flow flat plate variation.

### 4.5.2 Effect of blind slat angle on transition

As indicated in Figure 4-14, for R= 0.03 and different blind slat angles, at which the angle is equal to 0 and -22.5, the transition occurs at a higher Rayleigh number. This was expected, since the distance between the tip of the blind slats and window is lower in these cases. The contours for the turbulent viscosity ratio for different blind slat angles for a given Rayleigh, R equal to 0.03, and different blind slat angles are shown in Figures 4-26 through 4-29.
Figure 4-26 Distribution of the turbulence viscosity ratio for $R = 0.03$, a Rayleigh number of $2.0E11$ and $\theta = -45$.

Figure 4-27 Distribution of the turbulence viscosity ratio for $R = 0.03$, a Rayleigh number of $2.0E11$ and $\theta = +45$. 
Figure 4-28 Distribution of the turbulence viscosity ratio for R= 0.03, a Rayleigh number of 2.0E11 and θ=0

Figure 4-29 Distribution of the turbulence viscosity ratio for R= 0.03, a Rayleigh number of 2.0E11 and θ=90 (fully closed)
In Figures 4-26 and 4-27, when the slat angles are +45 and -45, turbulence in the flow develops over the window at approximately the same position. However, when the blind slat angle is 0, turbulence in the flow develops over the blinds. In Figure 4-29, the turbulent viscosity ratio for the case when the blind is fully closed is shown. In this case, turbulence in the flow develops completely in the flow over the blind.

4.6 Effect of heat generation in the slats

4.6.1 Effect of heat generation in the slats on the local and mean Nusselt number

Heat generation in the slats is the result of solar radiation passing through the window and falling on, and being absorbed by, the slats of the blind. The variation of the mean Nusselt number with a Rayleigh number for a given blind slat angle and R value, is shown in Figure 4-30 for four different heat fluxes on the slats.
In the present study, the isothermal vertical plate (window) is at a lower temperature than the room air temperature; therefore, the presence of heat generation in the blind slats tends to increase the heat transfer rate to the window. At higher values of the heat flux and lower values of Rayleigh number, the mean Nusselt number is higher. By looking at the velocity contours, Figures 4-31 through 4-33, where there is a constant heat generation in the blind slats, it can be seen that at high values of slat heat fluxes, another flow will occur in the room in the opposite direction.

Figure 4-30 Variation of the mean Nusselt number with a Rayleigh number for R=0.03 and θ=+45 and different values of Qs
Figure 4-31  Velocity contours of an isothermal recessed vertical window with a $\theta=+45$ blind where
\[
Ra=2.0\text{E}11, R=0.03 \text{ and } Q_s=25 \left( \frac{W}{m^2} \right)
\]

Figure 4-32 Velocity contours of an isothermal recessed vertical window with a $\theta=+45$ blind where
\[
Ra=2.0\text{E}11, R=0.03 \text{ and } Q_s=50 \left( \frac{W}{m^2} \right)
\]
Figure 4-33 Velocity contours of an isothermal recessed vertical window with a $\theta=+45$ blind where $Ra=2.0E11$, $R=0.03$ and $Qs=100 \text{ W/m}^2$

4.6.2 Effect of heat generation in the slats on laminar-to-turbulent transition

The contours of the turbulent viscosity ratio for different slat heat fluxes for a given Rayleigh and $R$ equal to 0.03, and a slat angle of 45 degrees are shown in Figures 4-34 through 4-36.
Figure 4-34 Distribution of the turbulence viscosity ratio for $R=0.03$ with a $\theta=+45$ blind and a Rayleigh number of $2.0E11$, $Q_{s}=25 \left( \frac{W}{m^2} \right)$

Figure 4-35 Distribution of the turbulence viscosity ratio for $R=0.03$ with a $\theta=+45$ blind and a Rayleigh number of $2.0E11$, $Q_{s}=50 \left( \frac{W}{m^2} \right)$
Figure 4-36 Distribution of the turbulence viscosity ratio for $R=0.03$ with a $\theta=+45$ blind and a Rayleigh number of $2.0E11$, $Q_s=100 \left(\frac{W}{m^2}\right)$

Figure 4-34 shows a secondary turbulent flow in top of the domain. When the heat flux in the slats increases, this secondary turbulent flow becomes stronger as indicated in Figures 4-35 and 4-36.
Chapter 5

Conclusion and Recommendations

5.1 Introduction

The present work was undertaken because there are few available studies of convective heat transfer from window-blind systems in which transition from laminar to turbulent occurs in a flow. A numerical model of a window-Venetian blind combination has been developed using commercial finite-volume software, FLUENT. In the second part of the study, heat generation in the blind slats resulting from absorbed solar irradiation has been considered.

5.2 Conclusion

The main conclusions that can be drawn from the results obtained in the present study are:

i. At higher Rayleigh numbers considered (over $10^9$) the presence of a blind, with different slat angles and blind to window spacing, does not have a significant effect on the mean Nusselt number.

ii. The presence of blinds at lower Rayleigh numbers considered (below $10^9$) when the blind-to-window spacing is small, does have a significant effect on the mean Nusselt number, decreasing its value.
iii. When the slat angle is small (i.e., the slat is near the normal to the window and the blind to window spacing is small), the presence of the slats tends to delay the development of turbulent flow in a window blind system to higher Rayleigh numbers.

iv. In the smaller slat angles when the blind to window spacing is small, the presence of the slats tends to change the development of a turbulent flow in a window blind system to higher Rayleigh numbers.

v. The presence of a fully closed blind considerably decreases the mean Nusselt number.

vi. Strong periodic variations are observed the local Nusselt number variation when the blind-to-window spacing is small.

vii. Heat generation in the slats, which is representative of absorbed solar radiation, increases the mean Nusselt number and this effect increases as the Rayleigh number decreases.

viii. At high Rayleigh numbers (i.e., over $10^{11}$), increasing the heat generation in the blind slats decreases the mean Nusselt number and this effect increases as the Rayleigh number decreases.

ix. As the heat generation in the blind slats increases, convective heat transfer from the blind to the room occurs.
5.3 Recommendations

i. In the numerical modeling of Venetian blinds, conduction in the slats and radiant heat transfer should be included in the analysis.

ii. In the present study the blind slats were assumed to be flat for simplicity in modeling. An investigation should be made for the case where the blinds have the geometry of commercially-available blinds.

iii. Experimental studies should be taken in order to validate the results obtained using the numerical model.
Bibliography


FLUENT, 2006. In: FLUENT Documentation, ANSYS Inc..


Appendix A

Input values for FLUENT

Numerical model

FLUENT

<table>
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<tr>
<th>Version</th>
<th>2d double precision (2ddp)</th>
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<td>Full Simulation</td>
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Model

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<tr>
<td>Energy</td>
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Materials

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<td>1006.43 ( \frac{j}{kg-K} )</td>
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<td>Thermal conductivity</td>
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<td>0.0242 ( \frac{W}{m-K} )</td>
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<td>Thermal expansion coefficient</td>
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Operation Conditions

Operation pressure: 101325 (Pascal)
Gravity: On
Operating Temperature: 310 (K)
Gravitational Acceleration: X=0, Y=-g \( \frac{m}{s^2} \)

Boundary Conditions

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<tr>
<td>Motion Type: Stationary</td>
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### Solver Controls

<table>
<thead>
<tr>
<th>Equation solved</th>
<th>Flow, Turbulence and Energy</th>
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<tbody>
<tr>
<td>Under Relaxation Factors</td>
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<tr>
<td>Pressure</td>
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<tr>
<td>Density</td>
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<tr>
<td>Body forces</td>
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<tr>
<td>Momentum</td>
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<tr>
<td>Turbulent Kinetic energy</td>
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<tr>
<td>Turbulent Dissipation rate</td>
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<tr>
<td>Turbulent Viscosity</td>
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<tr>
<td>Energy</td>
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<tr>
<td>Discretization</td>
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<tr>
<td>Pressure</td>
<td>PRESTO!</td>
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<tr>
<td>Momentum</td>
<td>Second Order Upwind</td>
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<tr>
<td>Turbulent Kinetic energy</td>
<td>Second Order Upwind</td>
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<tr>
<td>Turbulent dissipation rate</td>
<td>Second Order Upwind</td>
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<tr>
<td>Energy</td>
<td>Second Order Upwind</td>
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<tr>
<td>Pressure-velocity coupling</td>
<td>Simple</td>
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