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A NUMERICAL STUDY OF THE EFFECT OF A BELOW WINDOW HOT AIR VENT ON THE CONVECTIVE HEAT TRANSFER RATE FROM A COLD WINDOW COVERED BY A TOP-DOWN, BOTTOM-UP PLANE BLIND SYSTEM

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ABSTRACT

When heating systems are in use during cold winter months hot air from a floor-mounted vent located below a window will often flow over the cold window. The nature of the air-flow pattern over the window tends to be changed by the presence of the vent flow thus altering the rate of convective heat transfer to the window. The effect of the vent flow on the convective heat transfer rate from the window will be influenced by the presence of a blind system over the window. In recent times top-down, bottom-up plane blind systems in which the blind can both be raised at the bottom and lowered at the top have become popular. In the present study the effect of the hot air flow from a below window vent on the heat transfer rate from a recessed cold window covered by a top-down, bottom-up plane blind system has been numerically studied. The window, represented by a plane isothermal section recessed into the wall, is colder than the room air far from the window. It is assumed that the floor-mounted vent is located against the wall and has a uniform discharge velocity which is normal to the vent surface. The flow has been assumed to be steady. Situations involving both laminar and turbulent flow have been considered. The fluid properties have been assumed constant except for the density change with temperature that gives rise to the buoyancy forces. This was dealt with using the Boussinesq approach. Radiant heat transfer effects have been neglected. The governing equations have been solved using the commercial CFD code ANSYS FLUENT©. The *k*-epsilon turbulence model with buoyancy force effects fully accounted for was used. Results have been obtained for a Prandtl number of 0.74, i.e., effectively the value for air. The effect of the dimensionless top and bottom openings on the window Nusselt number for a wide range of Reynolds and Rayleigh numbers has been studied.

INTRODUCTION

Ensuring the thermal comfort of the occupants of a room is one of the main reasons that air vents are often located below a window. Generally, the presence of an air vent below a window alters the air flow pattern near the window and changes the rate of convective heat transfer from the window. The presence or absence of a blind system on the window will modify the effect of the vent flow on the window heat transfer rate.

When a blind system is present, the effect of the vent flow on the window heat transfer rate will be influenced by the type of blind system and the blind opening. In the present study, this effect has been investigated for the particular case of a top-down, bottom-up plane blind system. These blinds can be raised at the bottom and/or lowered at the top. By permitting the controlled use of sunlight to illuminate the building (daylighting) and/or the use passive solar room heating while still providing shade from direct sunlight and privacy for the occupants such blinds have the potential to reduce energy consumption in buildings. However, the effects of such a blind system on the convective heat transfer rate from the window to the room to which it is exposed have not been studied extensively. Determining the effect of vent discharge velocity on the convective heat transfer rate from a window with a top-down, bottom-up plane blind system was the basic purpose of this study. The main contribution of this work arises from the fact that it deals with top-down, bottom-up blinds, such blinds not having been considered in earlier work on window-hot air vent flows.Winter conditions, when hot air is discharged from the floor-mounted vent and when the window is at a lower temperature than the air in the room away from the wall containing the window, have been considered.

NOMENCLATURE

B	[-]	Dimensionless window recess depth, b/h
b	[m]	Window recess depth
g	[m/s^2]	Gravitational acceleration
h	[m]	Height of window
H_{bot}	[-]	Dimensionless bottom blind opening, h_{bot}/h
h_{bot}	[m]	Bottom blind opening
H_{top}	[-]	Dimensionless top blind opening, h_{top}/h
h_{top}	[m]	Top blind opening
H_w	[-]	Dimensionless height of bottom of window from floor, h_w/h
h_w	[m]	Height of bottom of window from floor
k	[W/mK]	Thermal conductivity
Nu	[-]	Nusselt number
Pr	[-]	Prandtl number
q_m'	[W/m^3]	Mean surface heat flux
Ra	[-]	Rayleigh number
Re	[-]	Reynolds number based on vent velocity
T_F	[K]	Temperature of fluid far from window and vent
T_v	[K]	Temperature of fluid discharged from vent
T_w	[K]	Temperature of window
u_v	[m/s]	Vent discharge velocity
V	[-]	Dimensionless vent size, v/h
v	[m]	Vent size

Greek Symbols

α	[m^2/s]	Thermal diffusivity
β	[1/K]	Bulk coefficient
ν	[m^2/s]	Kinematic viscosity

The situation that has been investigated in the present study is a very approximate model of real situations. The flow has been assumed to be two-dimensional. A plane isothermal section recessed into the wall represents the window. This window section is colder than the room air far from window and colder than the air discharged from the vent. It is assumed that the vent is located against the wall and that it has a uniform discharge velocity. This discharge from the vent has been assumed to be normal to the vent surface, i.e., to be vertical. The flow situation considered is shown in Figure 1. Because of the flow conditions considered both laminar and turbulent flow can exist. Radiative heat transfer effects have been neglected as this study has been restricted to the convective heat transfer from the inside of the window to the room.

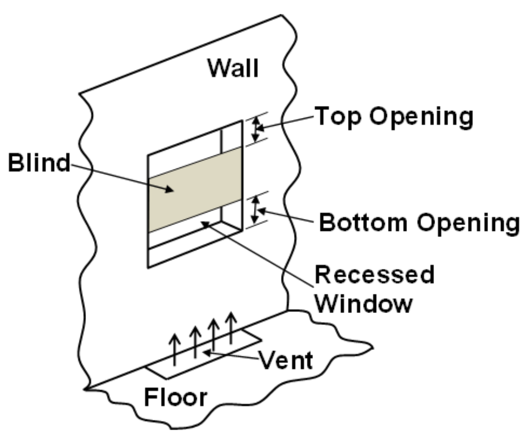

Figure 1 Flow situation considered

Of the many studies of the effect of blinds on the heat transfer rate between the room-side of a window and the room, most have involved traditional bottom-up type blinds. References [1-7] are typical of studies involving plane blind systems. Such studies involving Venetian blind systems have also been undertaken and some typical studies are described in [8-14]. Convective heat transfer from windows covered by top-down, bottom-up blinds has also been studied [15-18]. The effect of the presence of a floor-mounted vent on the convective heat transfer from a window system has been studied [19-21]. The effect of the presence of a floor-mounted vent on the convective heat transfer from a window with a top-down, bottom-up plane blind system has been studied [22] but attention in this study was restricted to summer conditions when the window is at a higher temperature than the cold vent air flow. Although turbulent flow can occur in a number of real situations, many past studies of convective heat transfer from the inner surface of a window have assumed that the flow remains laminar. However, some attention has been given to situations in which transitional and turbulent window system flows occur [23-25]. Using a similar turbulence modeling approach to that adopted here, numerical studies of turbulent building air flows have also been undertaken. Experimental validation of the numerical results is presented in some of these studies [26-45]. As is the case in many of these previous studies of window system heat transfer, the present study focused only on the convective heat transfer. However, the radiant heat transfer can be very important in some window heat transfer situations and can interact with the convective flow [46].

SOLUTION PROCEDURE

The flow has been assumed to be two-dimensional, i.e., the effect of the window and vent width has not been considered. The mean flow has been assumed to be steady. Except for the density change with temperature that gives rise to the buoyancy forces, which was dealt with using the Boussinesq approach, the fluid properties have been assumed constant. Reynolds averaged Navier-Stokes equations in conjunction with the *k*-epsilon turbulence model including full buoyancy force effects were used to obtain the solution. The commercial cfd code ANSYS FLUENT© has been used to solve the governing equations. The temperature of the fluid far from the window has been assumed to be constant and specified. As discussed above, radiant heat transfer effects have been neglected.

The following boundary conditions were assumed:

- the velocity on all solid surfaces was zero
- the temperature on the surface of the window was specified
- the temperature gradient normal to all other solid surfaces was zero
- the pressure was equal to the ambient pressure on the room side of the solution domain. Any fluid entering through this room side surface was at ambient temperature and crossed the outer surface at right angles to the surface
- the vent flow temperature and velocity over the surface of the hot air vent were uniform and specified.

Extensive grid- and convergence criterion independence testing was undertaken for this study. This testing indicated that the heat transfer results here presented are within 1% independent of the number of grid points and of the convergence-criterion used. Also evaluated was the effect of the position of the outer room-side surface from the window. This effect was found to be negligible for the position used in deriving these results.

RESULTS

The window height, h , has been used as the length scale with the other geometrical parameters being defined in Figure 2.

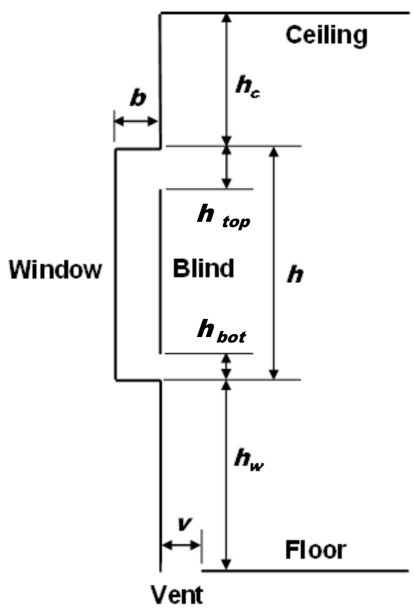

Figure 2 System dimensions considered

The solution has the following parameters:

1. The Rayleigh number, Ra, based on the window height and the overall temperature difference, i.e.:

$$Ra = \frac{\beta g \left(T_v - T_w\right) h^3}{\nu \alpha} \tag{1}$$

2. The Reynolds number, Re , based on the window height and the vent discharge velocity, i.e.:

$$Re = \frac{u_v h}{\nu} \tag{2}$$

3. The Prandtl number, Pr
4. The dimensionless depth of the hot-air vent, i.e., $V = v / h$
5. The dimensionless height of the bottom of the window from the floor, i.e., $H_w = h_w / h$
6. The dimensionless recess depth of the window, $B = b / h$
7. The dimensionless top and bottom blind openings, i.e., $H_{top} = h_{top} / h$ and $H_{bot} = h_{bot} / h$
8. The ratio of the hot-air vent temperature to the room temperature difference to the room temperature to the window temperature difference, $(T_v - T_F)/(T_w - T_F)$.

The distance of the ceiling from the top of the window, *hc*, was found to have essentially no effect on the window heat transfer rate for the values considered here.

Results have been obtained ony for Pr = 0.74, i.e., effectively the value for air, because of the application being considered. Results have also only been obtained for the case where the window surface is at a lower temperature than the room air. Results given here are for the case where V = 0.05, B = 0.05, H_w = 1, and $(T_F - T_v) / (T_F - T_w) = 0$, i.e., for the case where the air in the room away from the window is effectively at the same temperature as the air discharging from the vent. This is not the case existing in most real situations when, in winter, the vent air temperature is usually higher than that of the air far from the window. However, results obtained with other values of these parameters show the same basic characteristics as those presented here. Results were obtained for various values of the remaining solution parameters, H_{top} , H_{bot} , Ra, and Re.

In order illustrate the effect of the vent flow on the heat transfer rate from the window to the room, typical variations of the mean Nusselt number for the inner (room side) surface of the window with the Rayleigh numbers for the case where there is no vent are shown in Figure 3.

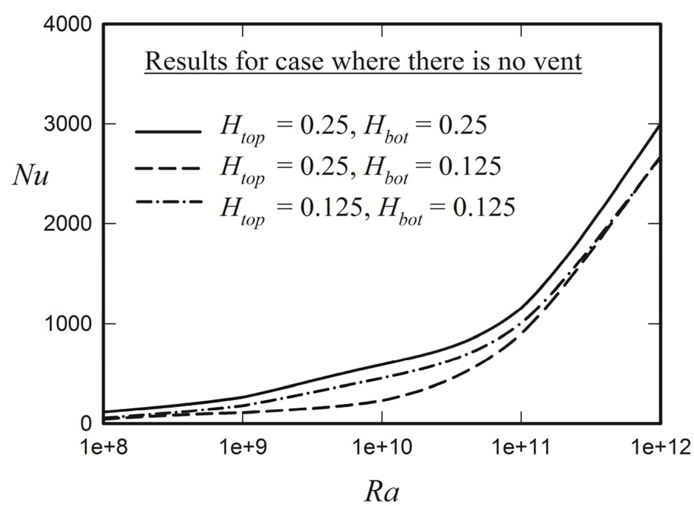

Figure 3 Variation of window Nusselt number with Rayleigh number for the case where there is no vent flow for various dimensionless top and bottom blind openings

Consideration will next be given to situations in which there is a vent flow. Attention will first be given to the two limiting cases where (i) the blind is fully closed, and (ii) the blind is fully open. The variations of the mean Nusselt number for the inner (room side) surface of the window with the Rayleigh numbers for various values of the Reynolds Number for these two cases are shown in Figures 4 and 5.

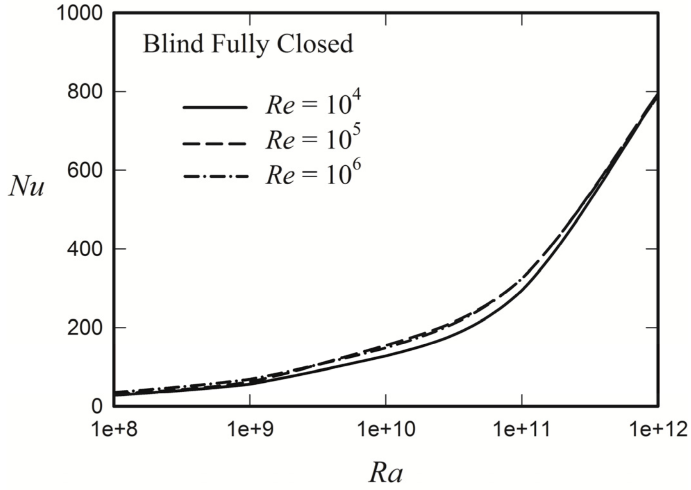

Figure 4 Variation of window Nusselt number with Rayleigh number for various Reynolds numbers for the case where the blind is fully closed

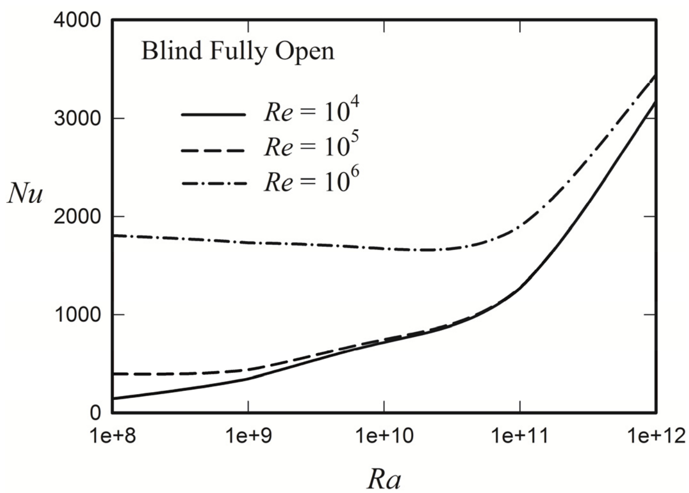

Figure 5 Variation of window Nusselt number with Rayleigh number for various Reynolds numbers for the case where the blind is fully open

It will be seen from the results given in Figure 4 that when the blind is fully closed the vent Reynolds number has very little effect on the Nusselt number. This is because when the blind is fully closed the heat transfer rate is dominated by the natural convective enclosure-like flow between the window and the inner surface of the closed blind, the thermal resistance across this part of the flow being far higher than that between the outer surface of the blind and the room air. By contrast it will be seen from the results given in Figure 5 that when the blind is fully open the vent Reynolds number has very significant effect on the Nusselt number at the higher Reynold numbers considered. At the lower Reynolds numbers the downward natural convective flow over the window diverts the

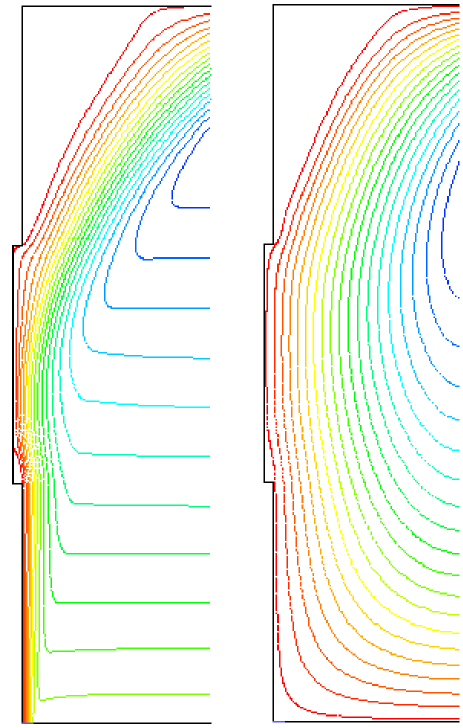

Figure 6 Streamline line patterns for a Rayleigh number of 10^8 and a Reynolds number of 10^6 (left) and for a Rayleigh number of 10^{12} and a Reynolds number of 10^4 (right) for the case where the blind is fully open

upward forced vent flow away from the window and the natural convective flow is then the dominat factor in determining the Nusselt number. However, at the higher Reynolds numbers the upward forced vent flow diverts the downward natural convective flow away from the window and passes directly over the window leading to the higher observed Nusselt numbers. Typical near-window streamline patterns for high and low Reynolds numbers that illustrate the flow pattern changes for this open blind case are shown in Figure 6.

Attention will next be turned to the case where the blind is partially open. Typical variations of the mean Nusselt number for the inner (room side) surface of the window with the Rayleigh numbers for various values of the Reynolds Number for various combinations of dimensionless top and bottom blind opening are shown in Figures 7 to 10. It will be seen from the results given in these figures that in all cases at the lower Rayleigh number values considered the Nusselt numbers tend to be independent of the Rayleigh number, i.e., the window heat transfer rate is mainly the result of the upward vent flow. At the higher Rayleigh number values considered however the Nusselt numbers tend to be independent of the Reynolds number, i.e., the window heat transfer rate is mainly determined by the downward natural convective flow over the window.

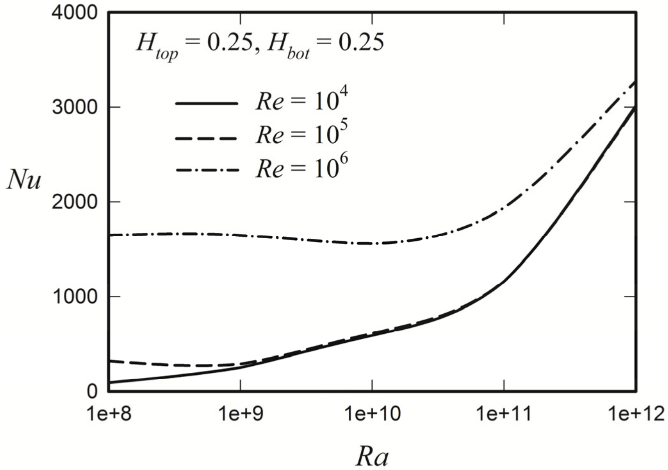

Figure 7 Variation of window Nusselt number with Rayleigh number for various Reynolds numbers for the case where the dimensionless top and bottom blind openings are both 0.25

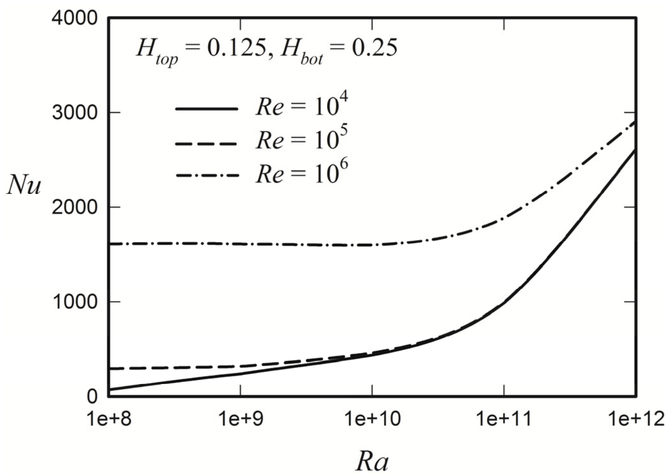

Figure 8 Variation of window Nusselt number with Rayleigh number for various Reynolds numbers for the case where the dimensionless top and bottom blind openings are 0.125 and 0.25 respectively

This is further illustrated by considering the variation of the mean Nusselt number for the inner (room side) window surface with Reynolds number for various Rayleigh number values. Typical variations are shown in Figures 11 and 12. These results again show that at the lower Reynolds numbers considered the Nusselt numbers tend to be independent of the Reynolds number whereas at the higher Reynolds numbers considered the Nusselt numbers tends to increase with the Rayleigh number. From the results given in Figures 7 to 10 it will also be seen that the larger of the top and bottom dimensionless blind opening is the most important in determining the window heat transfer rate. This is shown by comparing the results given in Figures 7 to 9 where the larger dimensionless blind opening is 0.25 with the results given in Figure 10 where the larger dimensionless blind opening is 0.125.

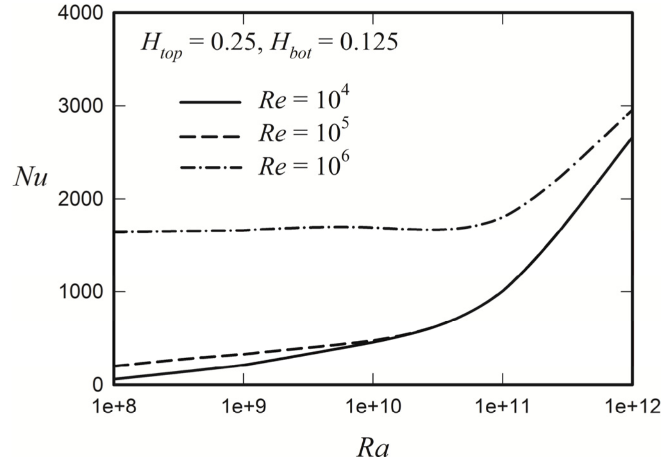

Figure 9 Variation of window Nusselt number with Rayleigh number for various Reynolds numbers for the case where the dimensionless top and bottom blind openings are 0.25 and 0.125 respectively

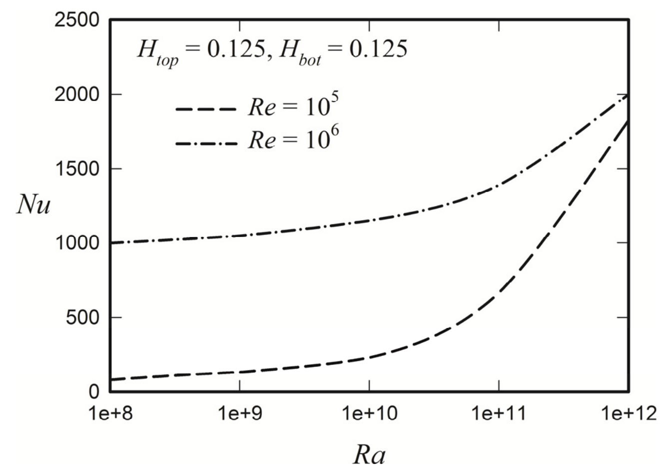

Figure 10 Variation of window Nusselt number with Rayleigh number for two Reynolds numbers for the case where the dimensionless top and bottom blind openings are both 0.125

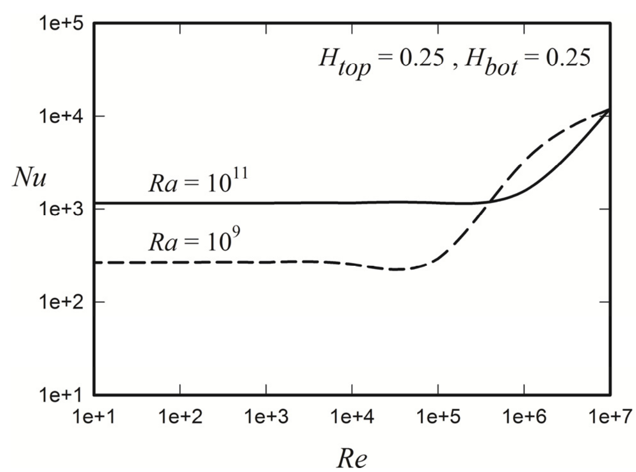

Figure 11 Variation of window Nusselt number with Reynolds number for two Rayleigh number values for the case where the dimensionless top and bottom blind openings are both 0.25

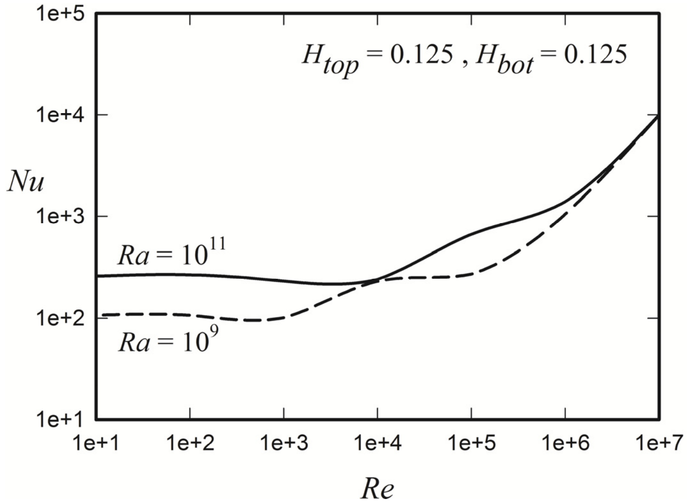

Figure 12 Variation of window Nusselt number with Reynolds number for two Rayleigh number values for the case where the dimensionless top and bottom blind openings are both 0.125

The changes in the form of the Nusselt number variation with Reynolds number at a given Rayleigh number from that which exist at the lower Reynolds numbers considered to that which exist at the higher Reynolds numbers considered is the result of changes in the nature of the flow over the window. At low Reynolds numbers the flow is dominated by the downward natural convective flow over the window which diverts the upward forced vent flow away from the window. At higher Reynolds numbers the upward forced vent flow diverts the natural convective flow away from the window. These changes in the flow pattern are illustrated by the typical near-window streamline patterns for high and low Rayleigh numbers shown in Figures 13 and 14. These changes in the flow pattern near the window and their effect on the temperature distribution near the window have a significant effect on the thermal comfort conditions near the window.

CONCLUSIONS

The results of the present study indicate that:

1. When the blind is fully closed the vent Reynolds number has very little effect on the window Nusselt number.
2. At the lower Reynolds numbers considered the downward natural convective flow over the window is the dominant factor in determining the window Nusselt number whereas at the higher Reynolds numbers considered the upward forced vent flow is the dominant factor in determining the window Nusselt number.
3. The larger of the top and bottom dimensionless blind openings is the most important in determining the window heat transfer rate.
4. The changes in the flow pattern near the window with Reynolds and Rayleigh numbers have a significant effect on the temperature distribution near the wall. These factors together have a strong influence on the thermal comfort conditions existing in the room near the window.

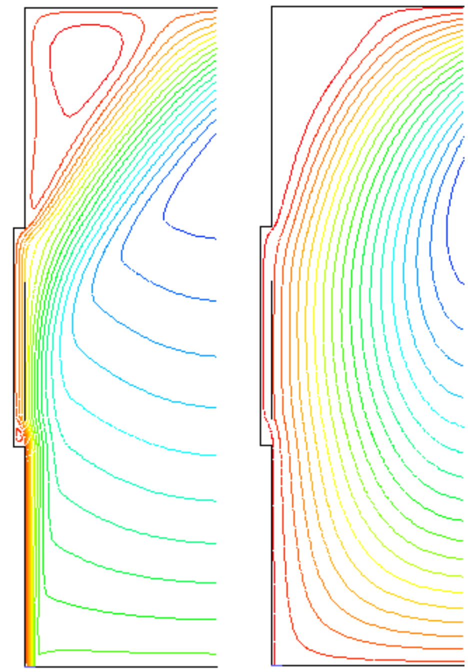

Figure 13 Streamline line patterns for a Rayleigh number of 10^9 and a Reynolds number of 10^6 (left) and for a Rayleigh number of 10^{12} and a Reynolds number of 10^6 (right) for the case where the dimensionless top and bottom blind openings are 0.25 and 0.125 respectively

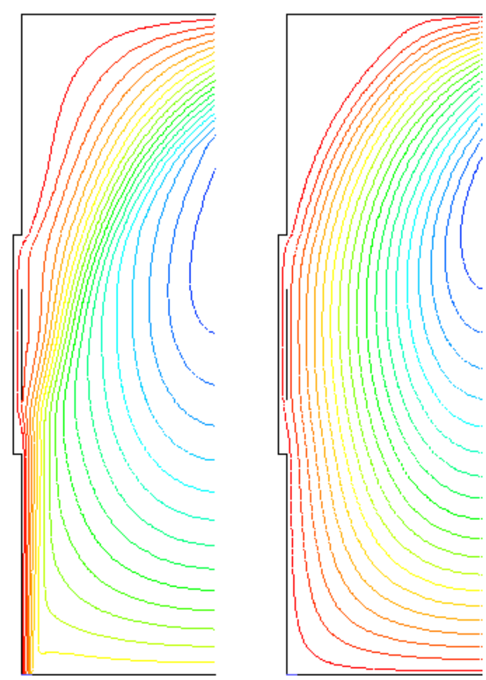

Figure 14 Streamline line patterns for a Rayleigh number of 10^8 and a Reynolds number of 10^5 (left) and for a Rayleigh number of 10^{12} and a Reynolds number of 10^5 (right) for the case where the dimensionless top and bottom blind openings are both 0.25

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