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A Numerical Study of the Effect of a Below-Window Convective Heater on the Heat Transfer Rate from a Cold Recessed Window

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An approximate numerical model of convective heat transfer to a window below which is mounted a natural convective heater has been used to study how the relative temperature of the heater affects the convective heat transfer rate to the window and the flow in the room. The heater and the window are represented by plane isothermal sections on the wall surface, the window section being recessed into the wall and colder than the room air far from the wall and the heater section being flush with the wall and hotter than the room air far from the wall. The width of the heater is greater than that of the window and the flow, which is assumed to be steady and laminar, is therefore three-dimensional. The numerical results which were obtained using a commercial cfd code show the effect of the Rayleigh number and of the dimensionless heater temperature on the window Nusselt number. The conditions under which the downward flow from the window is diverted upward by the flow from the heater have also been investigated.

1. Introduction

For various reasons, such as improving the thermal comfort of the occupants of a room, natural convective heaters are often mounted below a cold window. This alters the rate of convective heat transfer to the window and the flow pattern in the room. The basic purpose of the present study was to determine how, in a very basic way, the relative temperature of the heater and the Rayleigh number affect the convective heat transfer rate to the window and the flow pattern in the room. The basic situation considered in the present study is shown in Fig. 1 and is, as can be seen from this figure, a very approximate model of most real situations. The heater and the window are here both represented by plane isothermal sections, the window section being recessed into the wall and the heater section being flush with the wall. The window section is colder and the heater is warmer than the room air far from the wall containing the window and the heater. The heater and the window, in general, have different widths and the flow is therefore three dimensional. Several geometrical arrangements have been studied but all of the results given here are for the arrangement shown in Fig. 1. These results are typical of the form of the results obtained with all the arrangements considered.
This study attempts to predict for the first time the conditions under which the flow changes from one in which the cold downward flow from the window predominates to

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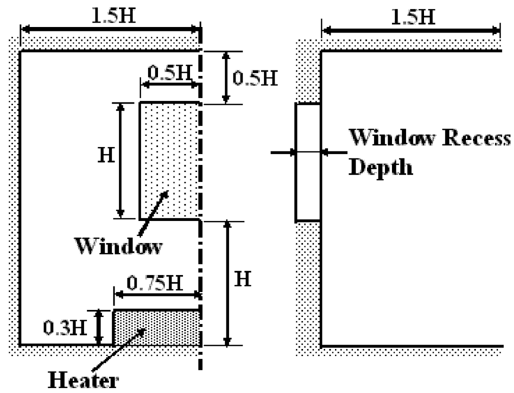

Figure 1 Flow situation considered

one in which the warm upward flow from the heater predominates, this having important implications for thermal comfort near the window.

There have been a number of studies of flow near window-wall heater systems. Typical studies include those undertaken by Turkoglu and Yucel (1995), Savin (1991), Shia-hui and Peterson (1995), Lankhorst and Hoogendoorn (1990) and, Spolek (1986). Studies of systems involving floor heating have been undertaken, for example, by Mohamad et al. (2006) and Peng and Peterson (1995). Studies of thermal comfort near heaters and windows have been undertaken, for example, by Larson and Moshfegh (2002), Jurelionis and Edmundas, (2008) and Kruger (1996). There have also been a number of basic studies of flow in enclosures with heated and cooled wall sections. Typical such studies include those undertaken by Kulkarni and Cooper (1995) and Sigey et al. (2004). None of these papers considers in a basic way how the geometric arrangement affects the flow, temperature distribution and heat transfer rates. It was for this reason that the present study was undertaken.

As is the case in many of the previous studies, the present study considers only the convective heat transfer. Radiant heat transfer can have a significant influence on the wall temperature distribution and thus influence the convective heat transfer. However, because these radiation effects are unlikely to have a strong influence on the way in which changes in the relative heater surface temperature and in the Rayleigh number effect the window heat transfer rate and the room air flow pattern they have not been considered here.

2. Solution Procedure

The flow has been assumed to be steady and laminar. 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. The governing equations have been written in dimensionless form using the window height as the length scale and the difference between the temperature of the window and the temperature of the undisturbed air far from the heater as the temperature scale. The resultant dimensionless equations have been solved using the commercial cfd code

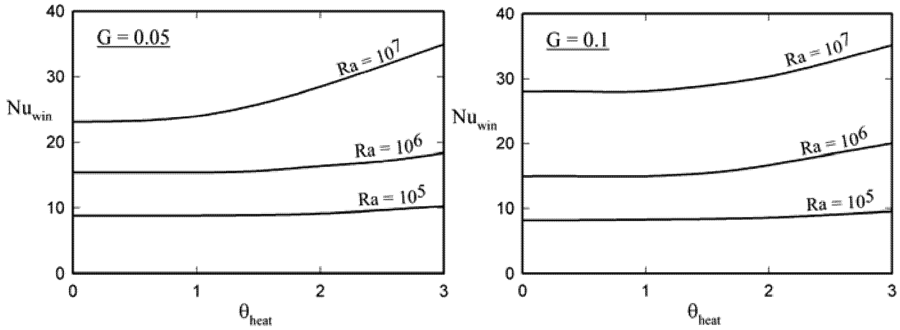

Figure 2 Variation of mean window Nusselt number with dimensionless heater temperature for various Rayleigh numbers for a dimensionless window recess depth of 0.05 (left) and 0.1 (right)

FLUENT. The solution for the particular geometrical arrangement that is here being considered then has the following parameters:

- the Rayleigh number based on the height of the window,
- the dimensionless depth that the window is recessed,
- the dimensionless heater-to-undisturbed-air temperature difference,
- the Prandtl number

Results have only been obtained for a Prandtl number of 0.7. The effects of the other dimensionless variables on the window Nusselt number and on the flow pattern and air temperature distribution near the window have been numerically studied. The dimensionless heater surface temperature is defined by:

$$\theta_H = (T_H - T_a) / \left|(T_W - T_a)\right| \quad (1)$$

where T_H is the uniform heater surface temperature, T_W is the uniform window surface temperature and T_a is the temperature of the undisturbed air far from the heater and window. The dimensionless window recess depth is given by $G = r / H$ where r is the window recess depth and H is the height of the window. The heat transfer rate to the window has been expressed in terms of the following Nusselt number:

$$Nu_w = q_w H / \left|(T_w - T_a)\right| k \quad (2)$$

where q_w is the mean heat transfer rate to the window and k is the thermal conductivity of the air.

Extensive grid- and convergence criterion independence testing were undertaken. This indicated that the heat transfer results presented here are to within 1% independent of

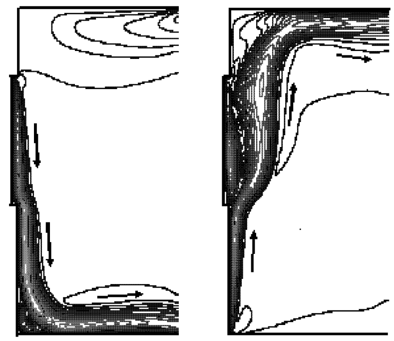

Figure 3 Flow on centre-plane for a Rayleigh number 10^6, a dimensionless recess depth of 0.05 and a dimensionless heater temperature of 1 (left) and 3 (right)

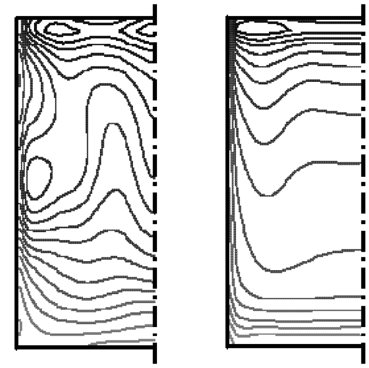

Figure 4 Heat flux distribution over window for a Rayleigh number 10^6 and dimensionless recess depths and dimensionless heater temperatures of 0.05 and 2.5 (left) and of 0.1 and 1 (right)

the number of grid points and of the convergence-criterion used. While the present results are not directly comparable to those obtained in previous related studies because of differences between the conditions considered, the general form of the results obtained here is in good agreement with those found in previous studies and extrapolation of the numerical results obtained in previous studies indicates satisfactory agreement between the present numerical results and those obtained in previous studies.

3. Results

The effect of the dimensionless heater temperature on the window Nusselt number for two dimensionless window recess depths is shown in Fig. 2. It will be seen that in both cases the Nusselt number is essentially constant at the lower values of the dimensionless heater temperature but then increases with increasing θ_H at the larger values of θ_H. This change results from the fact that at the lower values of θ_H the flow is dominated by the downward flow from the window and the flow over the window is essentially unaffected by the presence of the heater and so is essentially unaffected by θ_H. At the larger values of θ_H, however, the upward flow from the heater is dominant and the temperature of the air in the vicinity of the heater then increases with increasing θ_H and under these circumstances the heat transfer rate to the window consequently increases with increasing θ_H. The changes in the flow pattern are illustrated by the typical centre-plane streamline patterns shown in Figs. 3. These flow patterns are for the centre plane. To illustrate the three-dimensional nature of the flow, typical local heat flux distributions over the window are shown in Fig. 4.

It will also be noted from the results given in Fig. 2 that the value of θ_H at which the upward flow from the heater starts to dominate the flow pattern is the approximately the same for the two values of G considered, i.e., there is a relatively distinct value of θ_H at which the flow changes from being dominantly downwards to be dominantly upwards.

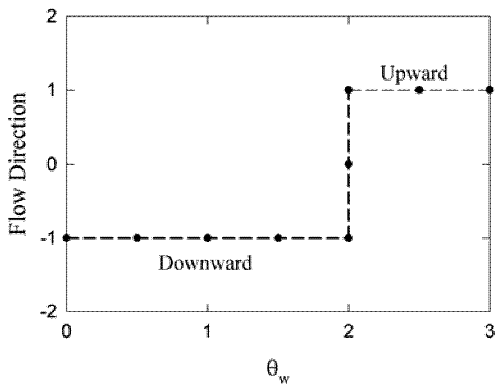

Figure 5 Variation of flow direction with dimensionless heater temperature for all Rayleigh numbers and dimensionless recess depths considered surface of height L

This rise can be expected to occur when the characteristic velocity associated with the upward heater flow becomes comparable to the characteristic velocity associated with the downward window flow. The characteristic velocity induced by a vertical surface kept at a surface temperature to surrounding air temperature difference of (*Ts* - *Ta*) is dependent on (α/L) ($Ra\ Pr$)$^{0.5}$ where α is the thermal diffusivity of the surrounding fluid, L is the height of the surface, Ra is the Rayleigh number based on L and on (*Ts* - *Ta*) and Pr is the Prandtl number. Hence since for the situation here being considered, the change from a dominantly downward flow to a dominantly upward can be expected to occur when the characteristic velocity for the flow over the heater is approximately equal to the characteristic velocity for the flow over the window, i.e., when:

$$\left(\frac{\alpha}{L}\right)_W \left(Ra_W Pr\right)^{0.5} = \left(\frac{\alpha}{L}\right)_H \left(Ra_H Pr\right)^{0.5}$$

where the subscripts W and H refer to the window and the heater respectively. From this equation it follows that the flow change occurs approximately when:

$$\frac{1}{H}\left[\left(\left|T_W - T_a\right|\right)H^3\right]^{0.5} = \frac{1}{l}\left[\left(\left|T_H - T_a\right|\right)l^3\right]^{0.5} \quad \text{i.e., } \theta_H = \frac{H}{l} \tag{3}$$

where H is the height of the window and l is the height of the heater. This equation indicates that the value of θ_H at which the flow change occurs depends only on the geometry and not on the Rayleigh number nor on the window recess depth. If the net flow direction is defined as upwards or downwards or horizontal, the present results give the flow direction variation with θ_H shown in Fig. 5 for all Rayleigh numbers considered and for both recess depths considered. It will be seen from this figure that there is a relatively sharp change from a downward flow to an upward flow, this change occurring when $\theta_H = 2$.

4. Conclusions

The results of the present study indicate that:

1. At the lower dimensionless dimensionless heater temperatures the presence of the heater has little effect on the window Nusselt number. However when the dimensionless heater temperature θ_H exceeds a value of approximately 2 for the geometry here being considered the presence of the heater is significant and produces an increase in the window Nusselt number.
2. The interaction between the flows over the heater and the window produces a net downward flow into the room when the heater has little effect on the window Nusselt number and a net upward flow into the room when the heater has a significant effect on the window Nusselt number.
3. Recessing the window has little effect on the results.

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