Technical Notes

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Influence of Aperture Location on Free Convection in Partially Divided Enclosures

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Introduction

NATURAL convection heat transfer in rectangular enclosures has been the subject of considerable discussion and study. Recently, natural convection in enclosures fitted with internal, two-dimensional partitions has been studied by a number of investigators.[1-13] These studies have considered either a single baffle or two identical baffles, one extending downward from the upper wall and the other upward from the lower one. Natural convection is an enclosure with baffles of different heights along the upper and lower walls has, however, not been studied. In the present study, natural convection of air (Prandtl number $Pr=0.7$) in an externally heated vertical square enclosure with four different baffle configurations is studied. Baffle thickness d is maintained at 10% of the enclosure height H. In each case, the baffles are centrally located with a constant aperture ratio (dimensionless vertical distance between baffles) of 0.5. The four configurations result from increasing the dimensionless height of the lower baffle (denoted by h_ℓ/H) from 0.25 to 0.5 (i.e., $h_\ell/H=0.25$, 0.3, 0.4, and 0.5), while correspondingly reducing the height of the upper baffle. The vertical walls are isothermal (T_{hot} and T_{cold}) with the horizontal connecting walls being either perfectly conducting or adiabatic. The flow is assumed to be steady, laminar, and two-dimensional with the axis of the flow pattern parallel to the third dimension. Experimental work by Elder,[14] Landis and Yanowitz,[15] Linthorst et al.[16] and Nansteel and Greif[3] confirm the validity of these assumptions at the Rayleigh number range under consideration in this study ($Ra<10^6$). The governing equations are solved using the Boussinesq approximation and the assumption of negligible radiation effects.

All results in this paper have been obtained by a finite-difference solution of the governing differential equations. This finite-difference method is based on the control volume and is described in Refs. 12, 13, and 18. For both end-wall conditions, a nonuniform 40×40 grid configuration with grid points "packed" toward the enclosure walls and baffles is chosen. Results are compared with the solutions on a "packed" 80×80 grid. The peak differences between the two solutions for the different baffle configurations studied are less than 3.7%.

Results

In addition to the dimensionless height of the lower baffle (h_ℓ/H), the two parameters of interest are the Grashof number, $Gr=g\,\beta\,\Delta T\,H^3/v^2$ and dimensionless baffle conductivity $k_r=k_b/k$, where g is the acceleration due to gravity, β the coefficient of thermal expansion, ΔT the temperature difference between the hot and cold walls, v the kinematic viscosity, and k_b and k the baffle and air thermal conductivity, respectively. For the adiabatic end-wall condition, results are obtained only for a dimensionless baffle conductivity of $k_r=2$ and for Gr values up to 1×10^6. For the perfectly conducting end wall, solutions are computed for both a low ($k_r=2$) and high ($k_r=500$) value of the baffle conductivity and for Grashof numbers up to 500,000.

Streamlines and Temperature Plots

For the enclosure with adiabatic end walls, Fig. 1 shows the contour plots for the four aperture locations studied and at a Grashof number of 1×10^6. As h_ℓ/H is increased from 0.25 to 0.5, there is an increasing tendency of the fluid flowing down the cold vertical wall to detach from the wall at a y/H approximately equal to the dimensionless height of the lower baffle. When the aperture is at the top of the enclosure ($h_\ell/H=0.5$), the long lower baffle shields the bottom left of the enclosure from the hot vertical wall. The resulting strong thermal stratification decelerates the flow down the cold wall, as is evident by the downward midheight boundary layer velocities that are nearly 45% lower compared to the corresponding velocities when $h_\ell/H=0.25$. The aforesaid observation on flow detachment from the cold wall is consistent with the predictions of Winters.[8] It is observed that the aperture location (i.e., h_ℓ/H) has a signficantly stronger effect on the thermal boundary

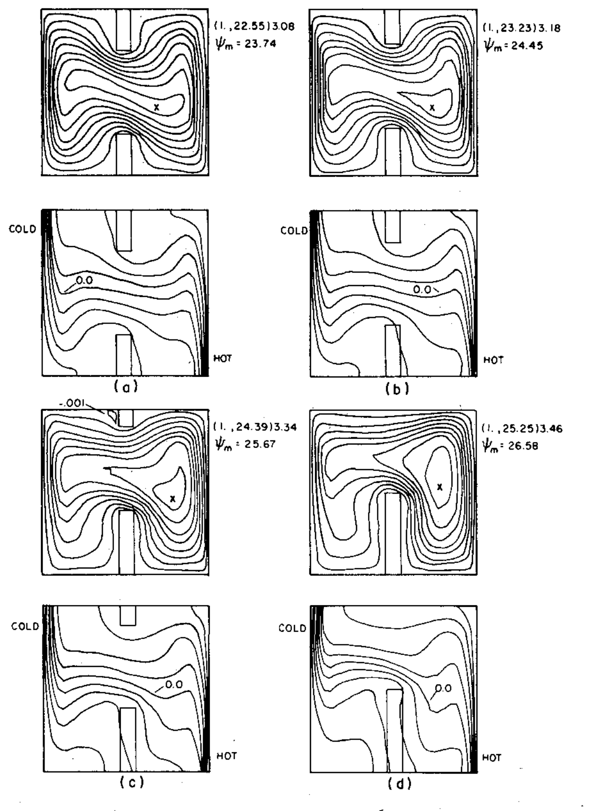

Fig. 1 Streamlines and isotherms, $Gr=10^6$, $k_r=2$, adiabatic end walls: a) $h_\ell/H=0.25$; b) $h_\ell/H=0.30$; c) $h_\ell/H=0.40$; d) $h_\ell/H=0.50$. (The numbers to the right of the streamline plots are the limiting values; the value outside theparenthesis is the uniform increment. Ψ_m is the maximum streamfunction value. Isotherms are plotted in uniform increments of 0.01 and range is from 0.5 to -0.5.)

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layer along the cold wall compared to its effect on the hot-wall boundary layer.

The influence of the end-wall condition can be seen in Fig. 2. The perfectly conducting end walls either cool (along the upper wall) or heat (along the lower wall) the near-wall flow and therefore result in lower temperature gradients along the hot and cold walls. Further, with perfectly conducting end walls, the lower baffle is at a higher temperature and the upper baffle at a lower temperature compared to the corresponding baffle temperatures with adiabatic end walls. Thus, with perfectly conducting end walls, the flow along the lower wall is preheated by both the end wall and the baffles; this preheating causes the flow negotiating the lower baffle tip to be more bouyant, in turn facilitating the separation behind the baffle, as seen in Fig. 2. Also, preheating or precooling of the flow by the end walls decreases the thermal stratification in the lower-baffle/cold-wall or the upper-baffle/hot-wall regions, thus permitting the fluid to penetrate further into the stratified corner regions.

The effect of increasing h_ℓ/H when the end walls are conducting can also be seen in Fig. 2. As in the adiabatic end-wall case, thermal stratification is increased and flow penetration into the lower-baffle/cold-wall region is decreased with increasing h_ℓ/H. The thermal conductivity of the baffle in the perfectly conducting end-wall enclosure has a strong effect on the flow penetration into the stratified lower-baffle/cold-wall region (or the upper-baffle/hot-wall region) of the enclosure. The lower baffle is at a higher temperature when it is more conducting. (See Figs. 2a and 2d.) The higher temperatures of the lower baffle has two important effects: 1) it reduces the thermal stratification in the lower-baffle/cold-wall region and thus permits greater penetration of the downflow along the cold wall; and 2) the greater preheating of the fluid negotiating the lower-baffle tip, which makes it more buoyant and susceptible to flow separation behind the lower baffle (Fig. 2d).

At this point, it is worthwhile to make a qualitative comparison of the flow patterns predicted in this study with those reported in other experimental studies. Duxbury's experiments[7] were conducted at moderate Rayleigh numbers and the flow behind the baffle was observed to be separated. Predictions of Winters[6] and those of the present study with adiabatic end walls show no separation. However, It has been shown earlier[12,17] that, with air as the working fluid, adiabatic end-wall conditions are not obtained experimentally (even if made from a nonconducting material such as Plexiglass) and that a linear temperature variation along the end wall is more realistic. This is borne out by the present predictions with perfectly conducting end walls for which the flow separates behind the baffle as in Duxbury's experiment.[7] The single-baffle experiments in Refs. 3–5 were conducted at high Rayleigh numbers (10^9–10^{11}) and a weak recirculation was noted in front of the baffle. Winters' calculations[8] have shown that, as the Rayleigh number is increased, the lower-baffle/cold-wall

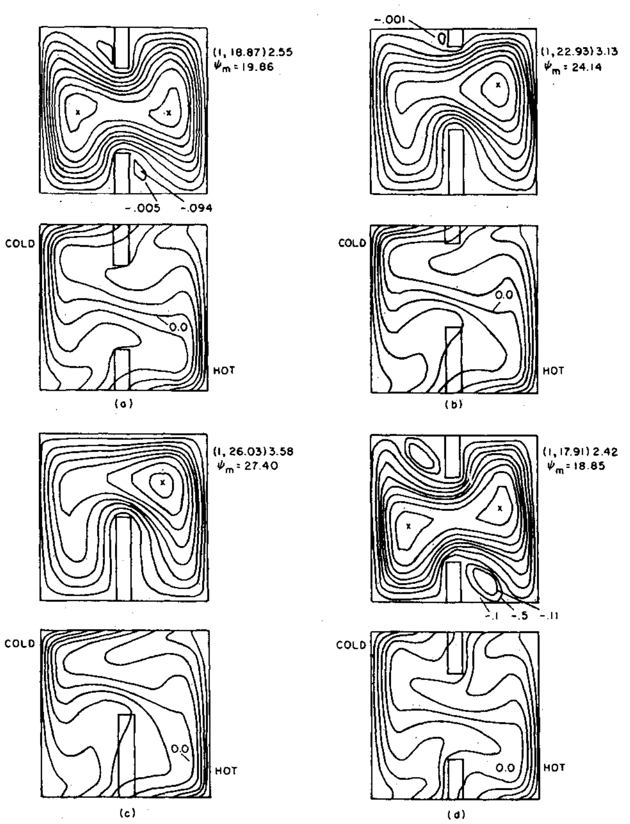

Fig. 2 Streamlines and isotherms, $Gr = 500{,}000$, perfectly conducting end walls: a) $h_\ell/H = 0.25$, $k_r = 2$; b) $h_\ell/H = 0.40$, $k_r = 2$; c) $h_\ell/H = 0.50$, $k_r = 2$; d) $h_\ell/H = 0.25$, $k_r = 500$.

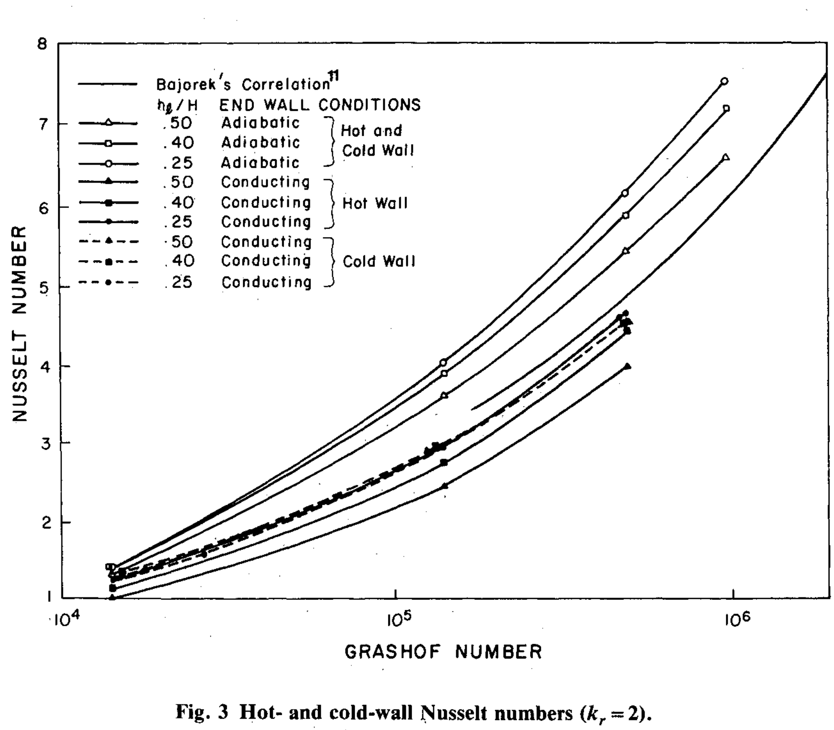

Fig. 3 Hot- and cold-wall Nusselt numbers ($k_r = 2$).

space becomes increasingly stratified and, for sufficiently high Rayleigh numbers (order of 10^9), the flow down the cold wall does not penetrate this region at all, but instead separates from the wall toward the baffle tip, thus forming a weak recirculation in the lower-baffle/cold-wall region.

Nusselt Number Results

Figure 3 compares the hot- and cold-wall Nusselt numbers for different aperture locations. Also included in the figure is Bajorek and Lloyd's[11] experimentally determined correlation for an air-filled enclosure with h_ℓ/H equal to 0.25 and aperture ratio of 0.5. As expected, the perfectly conducting predictions with $h_\ell/H = 0.25$ agree reasonably well with the correlation, while the predictions with the adiabatic end walls are substantially higher. The lowest heat transfer occurs for $h_\ell/H = 0.5$ and Nu increases with decreasing h_ℓ/H. Similar behavior is noted along the cold wall, except that when the end walls are conducting the average Nusselt number appears to be relatively insensitive to the value of h_ℓ/H. To explain some of these trends, it should be noted that, as h_ℓ/H is increased, thermal stratification in the lower-baffle/cold-wall region becomes stronger, resulting in lower heat transfer between the fluid stream and the lower portion of the cold wall. The local Nusselt number along the hot wall also decreases with increasing h_ℓ/H. At a higher value of h_ℓ/H the flow detaches earlier from the cold wall; therefore, the flow up the hot wall is correspondingly less cold, leading to a decrease in Nusselt number.

Conclusion

A numerical study has been made to determine the influence of the aperture location in an enclosure with centrally located upper and lower baffles and dimensionless aperture opening of 0.5. The aperture location is found to have a significant influence on the heat-transfer, velocity, and temperature profiles along the cold wall, but a weaker influence on the corresponding quantities along the hot wall.

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Energy Transfer in an Elliptic Annulus with Uniform Heat Generation

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Nomenclature

C,c	= dimensional and dimensionless constant temperature gradients, Eq. (9)
E,e	= dimensional and dimensionless excess temperatures, Eq. (9)
E_{in},e_{in}	= dimensional and dimensionless excess temperatures of the inner pipe
k	= thermal conductivity, $W/(m^2-K)/m$
L	= radius of outer periphery, m
m	= ellipticity of the annulus
P,P	= dimensional and dimensionless pressure, Eq. (2)
Pe	= Peclet number, $Re_L Pr$
Pr	= Prandtl number, ν/α
q	= characteristic heat flux, Eq. (9)
Q	= mass flow rate through the annulus, kg/s
Re_L	= Reynolds number, WL/ν
T	= temperature, $[CZ+E(X,Y)]$, K
T_0,T_i	= outer and inner wall temperatures, K
$X,x=X/L$	= dimensional and dimensionless transverse coordinate

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