Convective Heat Transfer Inside An Enclosure With Double Discrete Heaters And Exit Configurations

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Abstract

Mixed convective cooling of a two-dimensional vented cavity with multiple heaters at the bottom wall and cold upper wall is investigated numerically. The nonheated parts of the bottom wall and also the vertical side walls are considered to be adiabatic. The top inlet opening allows an external flow through the cavity and exists from another two openings placed at the lower half of the vertical walls. This configuration of mixed convective heat transfer has application in building energy systems, cooling of electronic circuit boards, solar collectors etc. Navier-Stokes equations are solved by using Penalty finite element method. Parametric simulations are carried out for three typical values of the Reynolds numbers, Re = 100, 200 and 300, and Richardson number, Ri as $0.1 \leq Ri \leq 10$ and Prandtl number is taken as 0.71. Results are presented in the form of streamline and isotherm plots as well as the variation of the average Nusselt number to explain the heat transfer characteristics inside the cavity. Analysis shows that the heat transfer coefficient is strongly affected by Reynolds number and Richardson number.

Key Words : Mixed convection, Penalty finite element, Nusselt number.

Mathematics Subject Classification : **76E06.**

1. Introduction

Mixed convection cooling is one of the preferred methods for cooling computer systems and other electronic equipments due to its simplicity and low cost. Again, the demand for faster and denser circuit technologies and packages has been accompanied by increasing heat fluxes at the chip and package levels, the application of air cooling techniques, involving either free or forced convection, plays a significant role over the years. In many modern buildings, mechanical ventilation is provided as a means of room load removal and provision of good indoor air quality. In displacement ventilation, cool air is supplied at a low level in the room to displace the warm room air, which is then extracted at a high level in the room. The indoor air flow and heat transfer characteristics are therefore, determined by the interaction between the natural convection and the forced one. The non-linear dynamical behaviors of mixed convection in a chemical vapor deposition reactor and a ventilated room were respectively investigated by Santen et al. [1] and Chow et al. [2]. However, there are still various applications of mixed convection in enclosures due to multiple discrete heat sources (Hsu and Wang [3]).

Raji and Hasnaoui [4] investigated the mixed convection in ventilated cavities where the horizontal top wall and the vertical left wall were prescribed with equal heat fluxes. Morrison et al. [5] studied the flow field numerically and included a limited amount of experimental data. The effect of different parameters such as the inlet position on the flow was established. The flow in a 10 mm wide rectangular cavity has been investigated by Rosengarten et al. [6], who showed the behavior of the flow and the local heat transfer for a limited range of conditions. In the present analysis, a square cavity ($L \times L$) with two discrete isoflux (q) heaters of length, $L_s = 0.2L$ on the bottom wall, the inflow opening located on the middle position of the top wall which is considered to be at low temperature, and the outflow openings on the lower half of both vertical walls have been considered. The physical model and the coordinate systems are shown in Fig. 1. The length of the all three openings is kept same and equal to 20% of

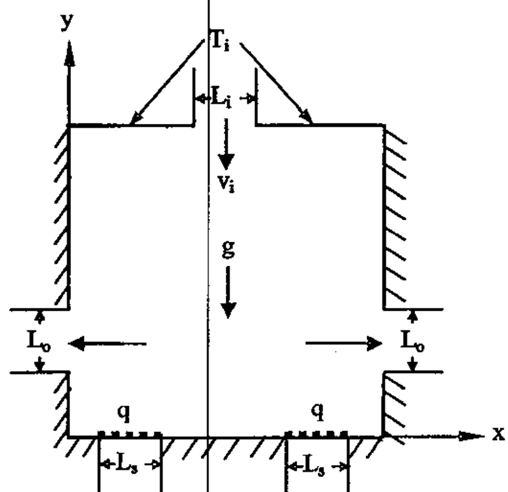

Fig. 1: Physical model of the problem

the cavity length. The other parts of the cavity are considered adiabatic. In this work, heat transfer in a square cavity with an inlet for inflow of external cold air while outflow of hot air occurs through two outlet openings is hereby examined.

2. Mathematical Model

Considering a steady, two-dimensional laminar flow of incompressible fluid, with negligible viscous dissipation effect and applying the Boussinesq approximation, the nondimensional forms of the governing conservation equations can be expressed as:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \tag{1}$$

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\frac{1}{Re}\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right) \tag{2}$$

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\frac{1}{Re}\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)+Ri\,\theta \tag{3}$$

$$U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=\frac{1}{Re\,Pr}\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right) \tag{4}$$

where, U and V are the velocity components in the X and Y directions, respectively, θ is the temperature, P is the pressure. The Grashof number, Reynolds number, Richardson number and Prandtl number are defined as follows.

$$Gr=\frac{g\beta q L^4}{k\upsilon^2},\ Re=\frac{v_i\,L}{\upsilon},\ Ri=\frac{Gr}{Re^2}\ \text{and}\ Pr=\frac{\upsilon}{\alpha} \tag{5}$$

The average Nusselt number can be written as,

$$Nu=\frac{1}{0.2}\left[\int_{0.1}^{0.3}\frac{1}{\theta_S(X)}dX+\int_{0.7}^{0.9}\frac{1}{\theta_S(X)}dX\right] \tag{6}$$

where, $\theta_S(X)$ is the local dimensionless temperature of the heated surface. The Simpson's 1/3 rule is used for numerical integration to obtain the average Nusselt number.

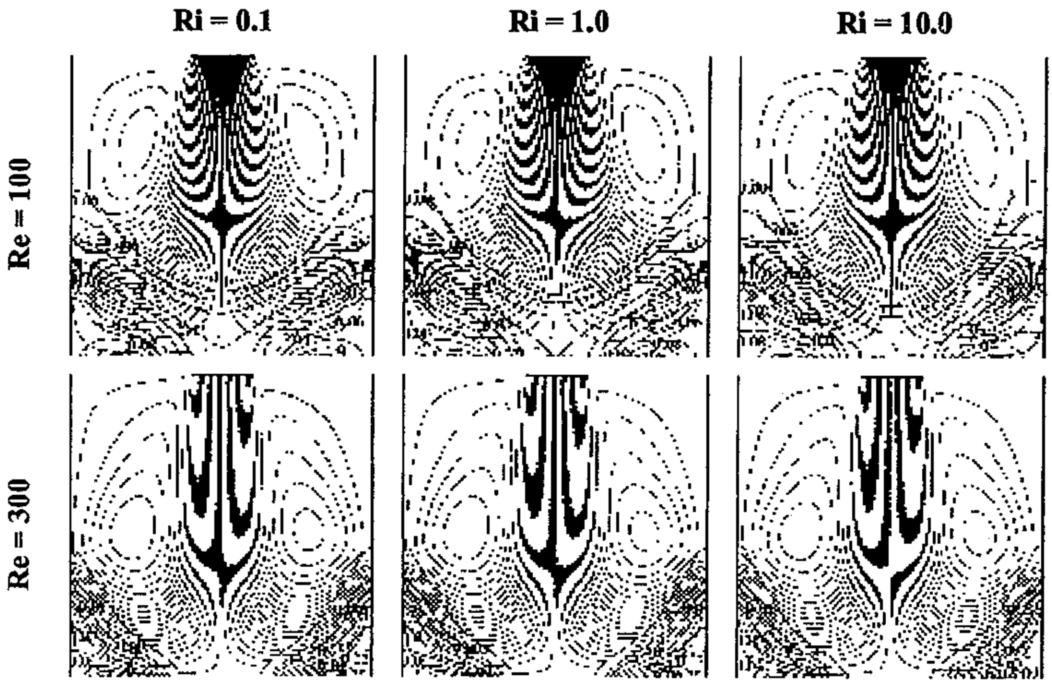

Fig. 2 : Streamlines and isotherm patterns for different Re and Ri

3. Results and Discussion

In the present research work, the Penalty finite element method has been adopted to obtain the solution of nonlinear partial differential equations (1)-(4) numerically. To ensure convergence of the numerical solution, the grid sizes have been optimized and the results presented here are independent of grid sizes. The code has been validated with the solutions are available in the literatures. The test was with mixed convection from a multiple ventilated rectangular enclosure when $Re = 100$ and $0.1 \leq Ri \leq 10$ as performed by Saha et al. [7].

In order to understand the effect of Richardson number on the viscous flow and heat transfer phenomena, a parametric representation of Ri varying from 0.1 to 10 is carried out. Figure 2 represents the streamline and isotherm plots of the vented cavity for $Ri = 0.1$, 1 and 10 respectively. As expected due to a different position of discrete heater placed at the bottom wall and external cold air flow from the mid position of the top wall, fluids move along the heat source surface and rise up forming two cells with counter and anti-counter clockwise rotation near the adiabatic side wall top corners.

The outflow openings exist in bottom side of the vertical walls and take away the heat swiftly from the heater surfaces. At very low values of Richardson number, such as 0.1 and 1, the forced convection due to the driven force which dominates the flow structure. At this order of Ri, the inertia and buoyancy forces balance each other, which then results in a mixed convection. But for high Richardson number, the buoyancy force becomes the dominant mechanism to drive the convection of the fluid.

More pronounced effect of heat transport by fluid convection closer to the heated surface is observed. Also the temperature gradient increases near the heated surface as the Richardson number increases. The characteristics of the mixed convection phenomena can be well understood by plotting the streamlines and isotherms for various Reynolds number as shown in Fig. 2. For all values of Re, the streamline pattern inside the vented cavity is found to vary with Ri in a regular fashion and the increase of Re enhances the heat removal through the exit. Nusselt number as a function of Richardson number for different Reynolds numbers is shown in Fig. 3. Here, three different trends of variation of average Nusselt number with the change of Richardson number are observed. When Ri increases from 0.1 to 1, the average Nusselt number is nearly invariant with Ri, but for increment of Ri from 1 to 5, the average Nusselt number increases slowly and further increase of Ri cause a rapid change of Nu. Also the heat transfer rate is minimum when $Re = 100$ and becomes maximum at $Re = 300$. These results simply conclude that with the increase of Re, heat transfer enhancement above the forced convection limit occurs sharply.

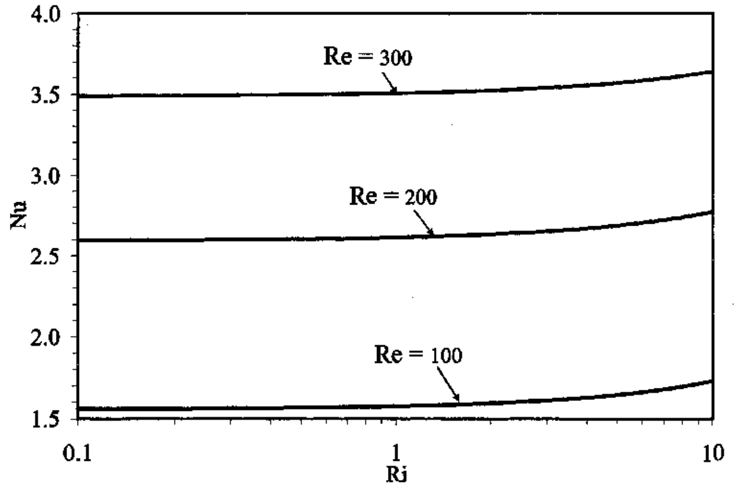

Fig. 3: Variation of Nu with Ri for different Re

4. Conclusion

Mixed convection arises as the buoyancy-induced hot flow from the source interacts with an externally induced cold air flow. The numerical solutions indicate that increasing the value of Re leads to higher heat transfer coefficient as well as higher intensity of recirculation. The governing parameter affecting heat transfer is the Richardson number. The mixed convection regime is observed when Ri = 1. The average Nusselt number plotted along the variation of Richardson number indicates that the heat transfer from the discrete heat source surface increases rapidly when Ri > 5. The average bulk fluid temperature seems to be remain constant in the highly buoyancy dominated convection regime.

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