COMBINED FREE AND FORCED CONVECTION INSIDE A TWO-DIMENSIONAL MULTIPLE VENTILATED RECTANGULAR ENCLOSURE

Sumon Saha[1], Goutam Saha[2], Mohammad Ali[1] and Md. Quamrul Islam[1]
[1]Department of Mechanical Engineering, Bangladesh University of Engineering and Technology, Dhaka, Bangladesh
[2]Department of Mathematics, Dhaka University, Dhaka, Bangladesh
E-mail: sumonsaha@me.buet.ac.bd

ABSTRACT

Combined free convection and forced convection from a flush-mounted uniform heat source on the bottom of a horizontal rectangular enclosure with side openings is studied numerically. The inlet opening allows an externally induced air stream at the ambient temperature to flow through the cavity and exits from another two openings placed top of the both side walls. Two-dimensional forms of Navier-Stokes equations are solved by using control volume based finite element technique. Three typical values of the Reynolds numbers, based on the enclosure height, are chosen as Re = 50, 100 and 200, and steady, laminar results are obtained in the range of Richardson number as $0 \leq Ri \leq 10$ and a fixed Prandtl number of 0.71. The parametric studies for a wide range of governing parameters show consistent performance of the present numerical approach to obtain as stream functions and temperature profiles. Heat transfer rates at the heated walls are presented in terms of average Nusselt numbers. The computational results indicate that the heat transfer coefficient is strongly affected by Reynolds number and Richardson number. An empirical correlation is developed by using Nusselt number, Reynolds number and Richardson number.

Keywords: free, forced, convection, finite, element, Richardson, heat, flux.

INTRODUCTION

In a mixed convection, both natural convection and forced convection participate in the heat transfer process. The bulk fluid flow direction can be any of the three possible directions in a horizontal channel, forward, backward or upward. The forced flow can be in the same direction as the flow created by natural convection, and this flow condition is called assisting mixed convection. Whereas, for the other case, forced flow direction is in an opposing direction to the flow that is created by buoyancy, and this flow condition is referred to as opposing mixed convection. But in some cases, convection from a horizontal heated enclosure, the forced flow is perpendicular to the buoyancy induced flow and this situation is called transverse mixed convection (Incropera and Dewitt, 1998). Mixed convective cooling is one of the preferred methods for cooling computer systems and other electronic equipments (Paterson and Ortega, 1990, Kennedy and Zebib, 1983), 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 an enclosure, the interaction between the external forced stream and the buoyancy driven flow induced by the increasing high heat flux from electronic modules leads to the possibility of complex flows. Therefore it is important to understand the heat transfer characteristics of mixed convection in an enclosure.

In many modern buildings, mechanical ventilation is provided as a means of room load removal and provision of good indoor air quality. The two methods of ventilating a room are mixed flow air distribution, where air is delivered into a ventilated space by turbulent air jets, and low velocity air supply into an occupied zone. These are known as mixing (dilution) ventilation and displacement ventilation, respectively (Awbi, 1998). In mixing ventilation, air is generally supplied to the room through a high-level air terminal device with the aim of mixing the supply air with the room air. 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. Two kinds of convection are involved in a ventilated room, i.e., internal buoyancy-induced natural convection by the discrete heat sources and external mechanical-driven forced convection by the ventilation, which results in double-diffusive mixed convection. In a mechanically ventilated room, the motion of air is significantly greater and one would expect to calculate convective output for many building thermal models. The indoor air flow and heat transfer characteristics are therefore determined by the interaction between the natural convection and the forced one (Deng, 2003).

The direct and indirect solar chimney principle has been used for heating of dwellings, drying of crops, dehumidification of air and as cooling tower (Sayigh, 1979). In heating applications, for example, the dwelling is simulated as an enclosure having a solar chimney located towards the sun, which serves as a receiver of the solar radiation and also serves as an energy storage device. The dwelling air enters the chimney from orifices on its lower part, is heated in the chimney as a result of which it rises and is returned to the dwelling through openings on the top. The air circulates in the dwelling, even after the sunset due to heat accumulated, partly, in the storage device, which is usually a massive wall. In this case, the heat is released from the surface of massive wall as if it had a flush mount heat source on it (Yedder and Bilgen, 1991). The same principle may also be used for space ventilation. In this case, dwellings have openings in opposite sides; the solar chimney is located towards the sun and serves as a heat source to drive the air. It enters from outside through openings opposite to the solar chimney and exits from openings at the top of it.

A search of the literature has shown that there has been little work into mixed convective heat transfer from the heated surfaces of ventilated rooms or enclosures. However, the main concerns of the previous studies were directed to assisting or opposing mixed convections, while very few works have dealt with transverse mixed convections. This is partly due to the difficulties inherent in transverse convections. Since the directions of the forced and natural convective flows are perpendicular to each other, the flows are substantially unstable. Thus, the flows will easily undergo a turbulent transition and will show a very complex nature even at small Reynolds and Richardson numbers. This makes analytical and experimental treatments very difficult.

REVIEW OF LITERATRE

A literature review is carried out in two areas of applications, which are directly related to the present study. In heat management applications, electronic components often flush mounted on boards are cooled by conduction and natural convection in enclosures. There is usually an upper limit for temperature allowed of these components and the heat management becomes an important aspect. A brief literature review shows that the case with enclosures having openings using forced, natural, mixed convection and conduction has been studied in a few cases. The most noted study was by Chen *et al.* (1989). The purpose of their work was to obtain convective heat transfer coefficients for a ventilated room with mixed convection (laminar and turbulent flows) for a relatively low Reynolds number when the boundary layers are not fully developed. The experiments that were carried out in a 5.6 × 3.0 × 3.2 m high chamber concerned the measurements of enclosure surface temperatures, air temperature differences between the enclosure surfaces and the air points 100 mm from the surfaces, the total heat fluxes through the surfaces and the air velocity 100 mm from the surface. Spitler (1990) also conducted measurements in a rectangular office-sized enclosure 5.48×3.65×3.35 m high for low ventilation flows that could fall into the mixed convection flow regime. Fisher (1995) extended Spitler's work by investigating buoyant, wall and free jets over a range of room inlet conditions using the same enclosure. Pavlovic and Penot (1991) reported the results of mixed convection experiments in an open cubic cavity at various flow incidence angles. A 0.6-m cavity was mounted in a low speed wind tunnel and the ambient temperature, approach velocity and cavity surface temperatures were measured.

Fundamental solutions to laminar forced convective heat transfer in ducts are well established (Shah and London, 1978). The thermal boundary condition that consists of having one wall at a constant temperature and the others adiabatic is commonly called a boundary condition of the third kind. The fully developed Nusselt number based on the hydraulic diameter and the log-mean temperature difference is known to be constant, and is equal to 4.86 for a parallel plate duct with boundary conditions of the third kind. The same value applies to an annular duct with a radius ratio of unity, and similar boundary conditions.

Converting the annular geometry into its rectangular equivalent leaves the heat transfer surface vertical, so that the natural convection flow is perpendicular to the main bulk flow direction. This combination is called transverse (or sometimes cross-flow) mixed convection and has been studied for uniformly heated vertical walls in a rectangular cross-section channel with various restricted inlets and outlets (Neiswanger et al, 1987, Rahman and Carey, 1990, Gao and Rahman, 1999). In these studies, the vertical heat transfer walls were sufficiently separated to allow the development of a distinct natural convection boundary layer up the vertical wall that was almost de-coupled from the main flow. Accordingly, a heat transfer correlation was possible in terms of a combination of the Rayleigh and Reynolds numbers.

Du *et al.* (1998) studied mixed convection heat transfer in vertical channels opened at the bottom and top, with protruding discrete heaters installed on one side. They found that the cooling of the electronic components was affected by the imposed flow, the strength of the natural convection, the aspect ratio and their position in the channel. Thermal management of a tape ball array package, which was attached to a plate and placed in a channel, was studied by Sathe and Sammakia (2000). They used a three dimensional conjugate heat transfer model to evaluate various parameters' effects on the chip junction temperatures. They showed good agreement with experimental results and importance of radiation heat transfer in thermal management. Harman and Cole (2001) studied conjugate heat transfer problem with two-layer substrate. They considered a horizontal system of infinite lateral extent with a flush mounted heater generating heat uniformly. The top layer was subjected to a shear flow, and the radiation and natural convection were neglected. They used an analytical-numerical approach to solve governing equations and derived appropriate design correlations. More complex geometrical problems have also been studied. For example, Thrasher et al (2000) studied experimentally a thermal system which consisted of pin-fin heat sink and chimney. They studied the effect of various parameters, namely, porosity, pin spacing and pin numbers on the heat transfer and showed good agreement with the theoretical results of a previous study.

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Jilani *et al*. (2001) performed a numerical analysis to study heat management of a flush mounted electronic component installed in an enclosure with a single opening. They investigated, at small Rayleigh numbers, the effects of heat flux, opening size and in and out going air flow from the opening on the heat transfer rate.

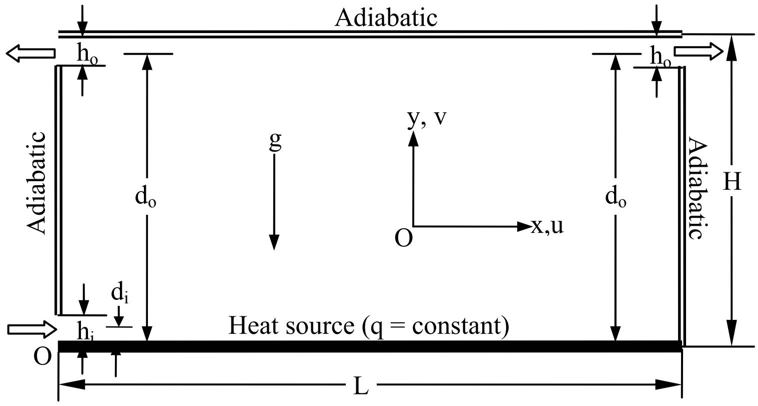

Figure-1. The geometry of the flow configuration under consideration, along with the coordinate system.

Ligrani and Choi (1996) performed an experimental study of mixed convection in straight and curved channels with the buoyancy orthogonal to the main forced flow. Fully developed laminar mixed convection in an annulus has been studied where one wall had a constant heat flux and the other was insulated (Hattori, 1979). Laminar developing mixed convection in a horizontal annulus has been studied numerically for a radius ratio of 0.5 (Karki and Patankar, 1989). More recently, annuli with radius ratios of 0.25, 0.5 and 0.75 were modeled for water and air using boundary conditions of the second and third kinds (Nonio and Giudice, 1996). These results showed that the effect of the Rayleigh number on the mean Nusselt number decreased with increasing radius ratio, although in the study by Nonio and Giudice (1996) the radius ratio was limited to 0.75.

Morrison *et al.* (1998) 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.* (1999), who showed the behavior of the flow and the local heat transfer for a limited range of conditions. They concluded that localized turbulence near the inlet has little effect on the local heat transfer below a critical Reynolds number.

Mixed convection in enclosures with one isolated heat source, in which the interaction between an external forced flow and an internal buoyancy flow determines the fluid flow and heat transfer structures, has received considerable attentions in the last decade. Papanicolaou and Jaluria (1990, 1993) numerically studied two-dimensional laminar mixed convection in a rectangular enclosure with a discrete heat source mounted on the wall. In addition, the effect of the thermal conductivity of the cavity walls on the heat transfer phenomena was investigated by Papanicolaou and Jaluria (1993). A later investigation (Papanicolaou and Jaluria, 1995) further presented turbulent flow in a cavity by k-ε model.

Two related studies of mixed convection in a partially divided rectangular enclosure were respectively carried out by Hsu et al (1997) and by How and Hsu (1998). The effect of an internal volumetric heat-generating and conducting solid body on the mixed convection in a square cavity was investigated by Yilbas *et al.* (2002). Laminar mixed convection in a two-dimensional enclosure heated from one sidewall and submitted to an either aiding or opposing jet was numerically studied in the work of Angirasa (2000) and Raji and Hasnaoui (1998).

The non-linear dynamical behaviors of mixed convection in a chemical vapor deposition (CVD) reactor and a ventilated room were respectively investigated by Santen *et al.* (2001) and Chow *et al.* (2001). However, there are still various applications of mixed convection in enclosures due to multiple discrete heat sources (Hsu and Wang, 2000). Raji and Hasnaoui (2000) investigated the mixed convection in ventilated cavities where the horizontal top wall and the vertical left wall were prescribed with equal heat fluxes. Mahaney *et al.* (1989, 1990) both numerically and experimentally studied the effect of buoyancy-induced flow on the forced flow in a three-dimensional horizontal channel with an array of 4 × 3 discrete heat sources mounted on the bottom wall. Choi and Ortega (1993), and Yücel *et al*. (1993) conducted a numerical study for mixed convection heat transfer in an inclined channel with discrete heat sources. In order to simulate the mixed convection occurring in multi-layered boards, Kim *et al.* (1992) analyzed a horizontal channel with rectangular heating blocks attached on one side of the wall by imposing periodic boundary conditions on the horizontal walls.

Bilen *et al.* (2001) experimentally investigated the effect of the geometric position of wall mounted rectangular blocks on the mixed convection heat transfer in a rectangular channel, taking into account the angular

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displacement of the block in addition to its spanwise and streamwise disposition. Wang and Jaluria (2002) numerically studied the three-dimensional mixed convection flow in a horizontal rectangular duct with multiple discrete heat sources flush-mounted on the bottom surface, and main attention is focused on the characteristics of the instability and the resulting effect on the heat transfer. Hung and Fu (1999) discussed the passive enhancement of mixed convection heat transfer in a horizontal channel with inner rectangular blocks by geometric modification.

Based on the survey, it was found that no work has been reported for mixed convection in a vented rectangular horizontal enclosure with two exit ports located at the top of the vertical walls. In this article we describe heat transfer in a rectangular enclosure with an inlet for inflow of external cold air while outflow of hot air occurs through two outlet openings. The heat source is a constant flux flush-mounted heater placed at the bottom of the cavity. The case of pure forced convection (Ri = 0) will be briefly mentioned, because it is of help in the interpretation of cases where convection is present. The purpose of this paper is to investigate the effect of the characteristic parameters, Reynolds and Richardson numbers on mixed convection in a cavity. The phenomena of both flow and thermal fields are displayed for effective comparison as well.

Geometric model

We consider a rectangular cavity of aspect ratio, H/L = 2.0 with a uniform constant-flux heat source q embedded on the bottom wall; the inflow opening located on the lower left vertical wall; and the outflow openings on the top of both vertical walls. Details of the geometry and coordinate system are shown in Fig. 1. The depth of the cavity is presumed to be long enough so that the whole model is two-dimensional. For simplicity, the height of the all three openings is kept same and equal to 0.1H. The other walls of the cavity are taken as adiabatic. The flow velocities of the fluid through the inflow opening are assumed to be uniform (u_{in}) at constant temperature T_{in}.

Mathematical model

The governing equations are those expressing the conservation of mass, momentum, and energy. As noted earlier, the flow is considered to be steady, laminar and two-dimensional. Constant thermal properties are assumed except for the density in the body force term of the y-momentum equation which is modeled by the Boussinesq approximation. The non-dimensional forms of the governing conservation equations can be written as

$$\nabla \cdot \vec{V} = 0 \tag{1}$$

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

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

$$\left(\vec{V} \cdot \nabla\right) \theta = \frac{1}{Pe} \nabla^2 \theta \tag{4}$$

The boundary conditions used are:

$U_{in} = 1$, $V_{in} = \theta = 0$ at inlet, and

$\frac{\partial U}{\partial X} = \frac{\partial V}{\partial X} = \frac{\partial \theta}{\partial X} = 0$ at the exit

$U = V = 0$, $\frac{\partial \theta}{\partial Y} = -1$ along the heated wall, and

$U = V = \frac{\partial \theta}{\partial X} = 0$ along the vertical insulated walls,

$U = V = \frac{\partial \theta}{\partial Y} = 0$ along the top insulated wall.

Non-dimensional parameter

The dimensionless parameters are used to describe the performance of fluid flow and heat transfer characteristics. Reynolds number based on enclosure height (H) is defined as

$$Re = \frac{u_{in} H}{\upsilon} \tag{5}$$

where u_{in} is inlet velocity of air,
Grashof number for constant heat flux (q) source is

$$Gr = \frac{g\beta q H^4}{k\upsilon^2} \tag{6}$$

and Prandtl number is

$$Pr = \frac{\upsilon}{\alpha} \tag{7}$$

The other two governing parameters, Richardson number and Peclet number are defined as

$$Ri = \frac{Gr}{Re^2} \tag{8}$$

$$Pe = Re \cdot Pr \tag{9}$$

The Nusselt number (Nu) is one of the important dimensionless parameters to be computed for heat transfer analysis in mixed convection flow. We define the local heat transfer coefficient $h = q / (T_H - T_{in})$ at a given point on the heat source surface where T_H is the local temperature on the surface of the heat source. Accordingly the local Nusselt number and the average or overall Nusselt number can be obtained respectively as

$$Nu_l = \frac{h \cdot x}{k} = \frac{X}{\theta_H} \text{ and}$$

$$Nu_{av} = \int_0^1 Nu_l dX = \frac{1}{L_H} \int_0^{L_H} \frac{X}{\theta_H} dX \tag{10}$$

where θ_H is the local dimensionless temperature and L_H is the dimensionless length of the heated wall.

Computational procedure

The computational procedure is similar to the one described by Baliga and Patankar (1983), and Gresho *et al.* (1984). The resulting system of the coupled equations (1-4) with the associated boundary conditions

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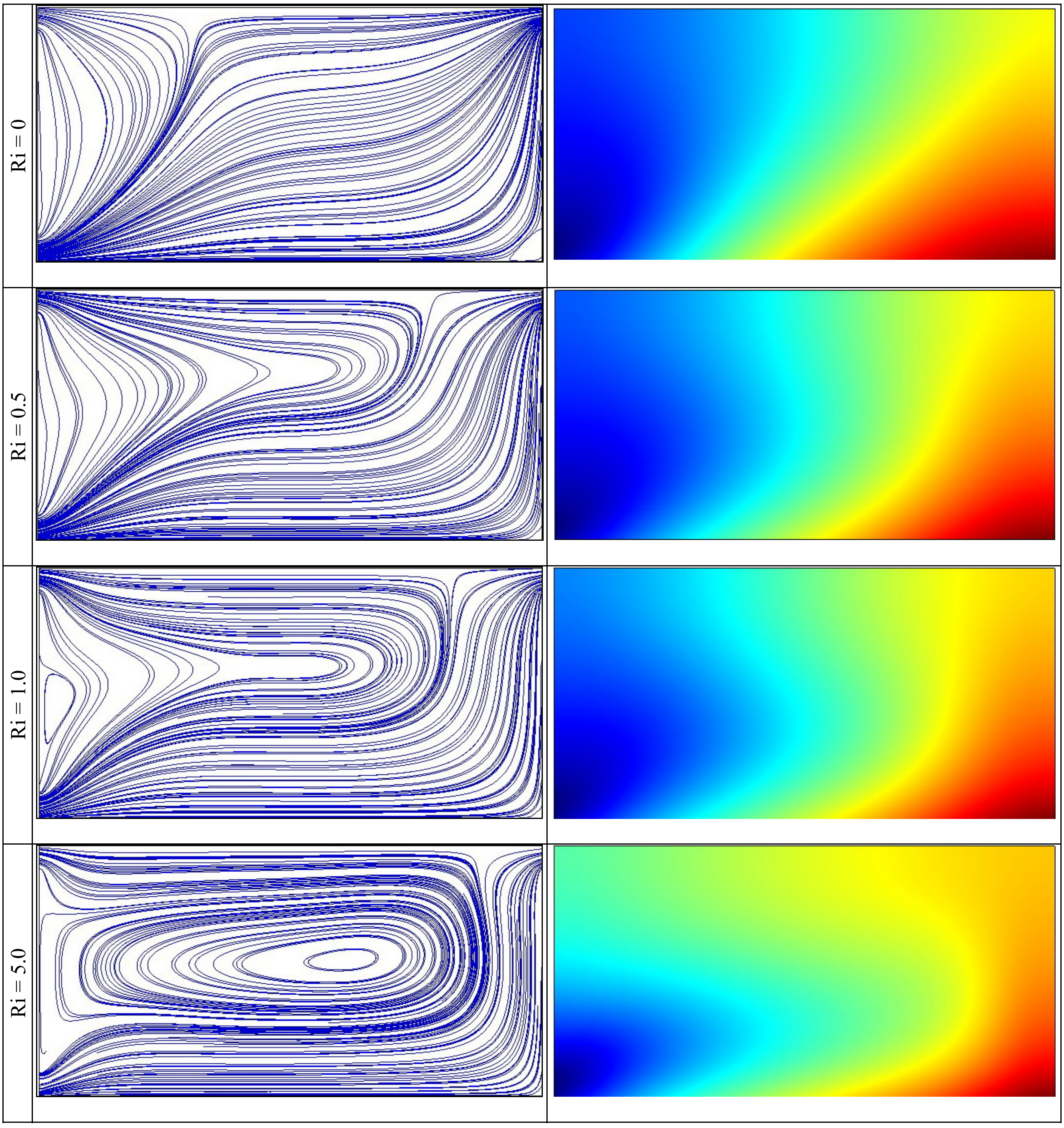
have been solved numerically using control volume based finite element method. The computational domain consists of 80 × 60 bi-quadratic elements which correspond to 63348 grid points. The control volume based finite element method provides the smooth solutions at the interior domain including the corner regions. To ensure the convergence of the numerical solution to the exact solution, the grid sizes have been optimized and the results presented here are independent of grid sizes.

Grid refinement tests have been performed for the case Re = 100 and Ri = 10 using three non-uniform grids 50 × 40, 65 × 55 and 80 × 60. Results show that when we change the mesh size from a grid of 65 × 55 to a grid of 80 × 60, the average Nusselt number (Nu_{av}) and the maximum temperature (θ_{max}) undergoes an increase of only 0.21% and 0.17%, respectively; then, because of calculation cost, the 80 × 60 grid is retained.



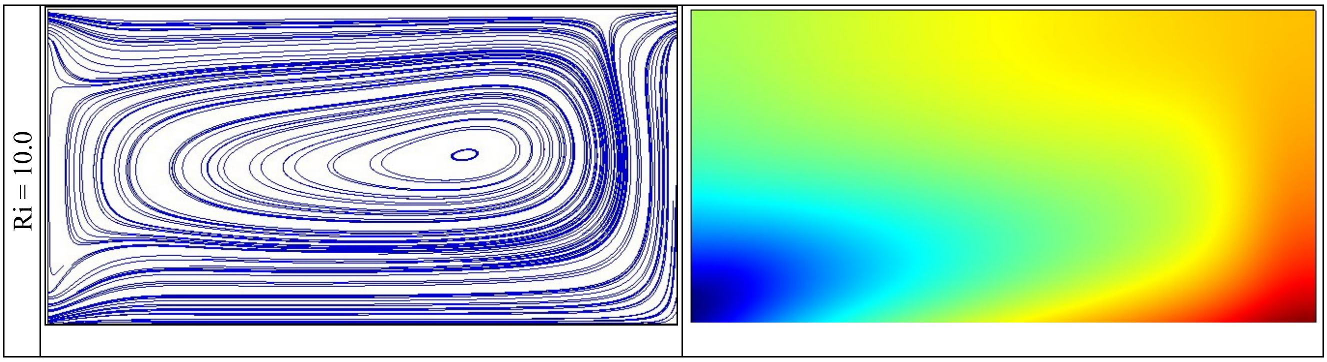
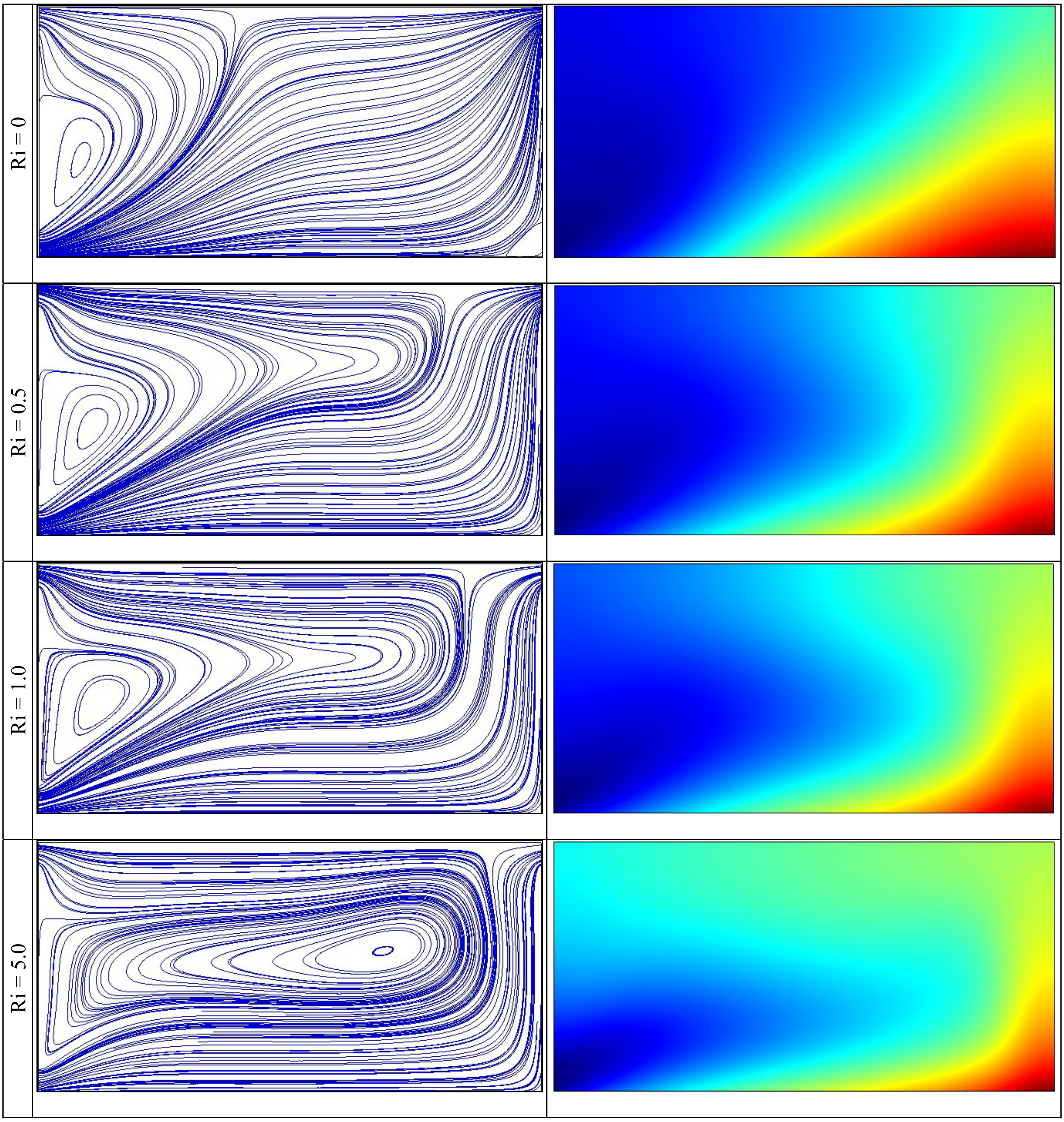


Figure-2. Streamline plots and contour map of temperature for different Richardson numbers at Re = 50.



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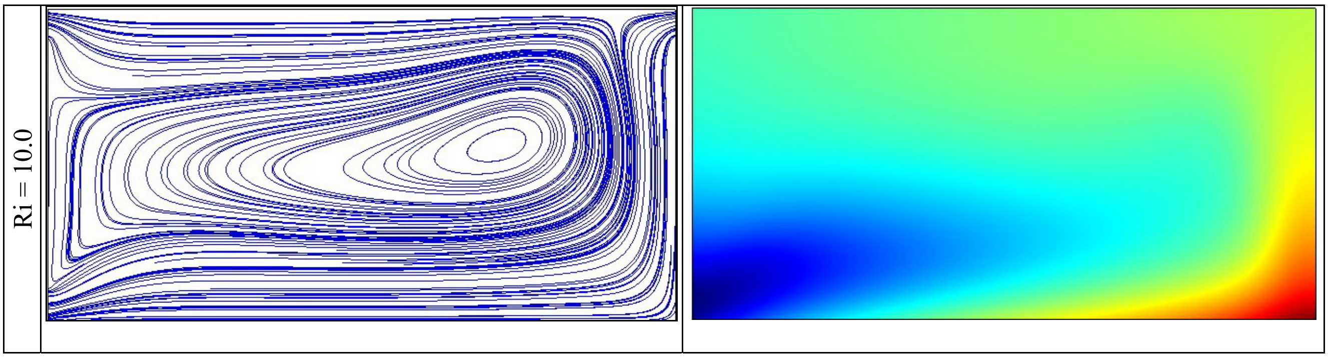
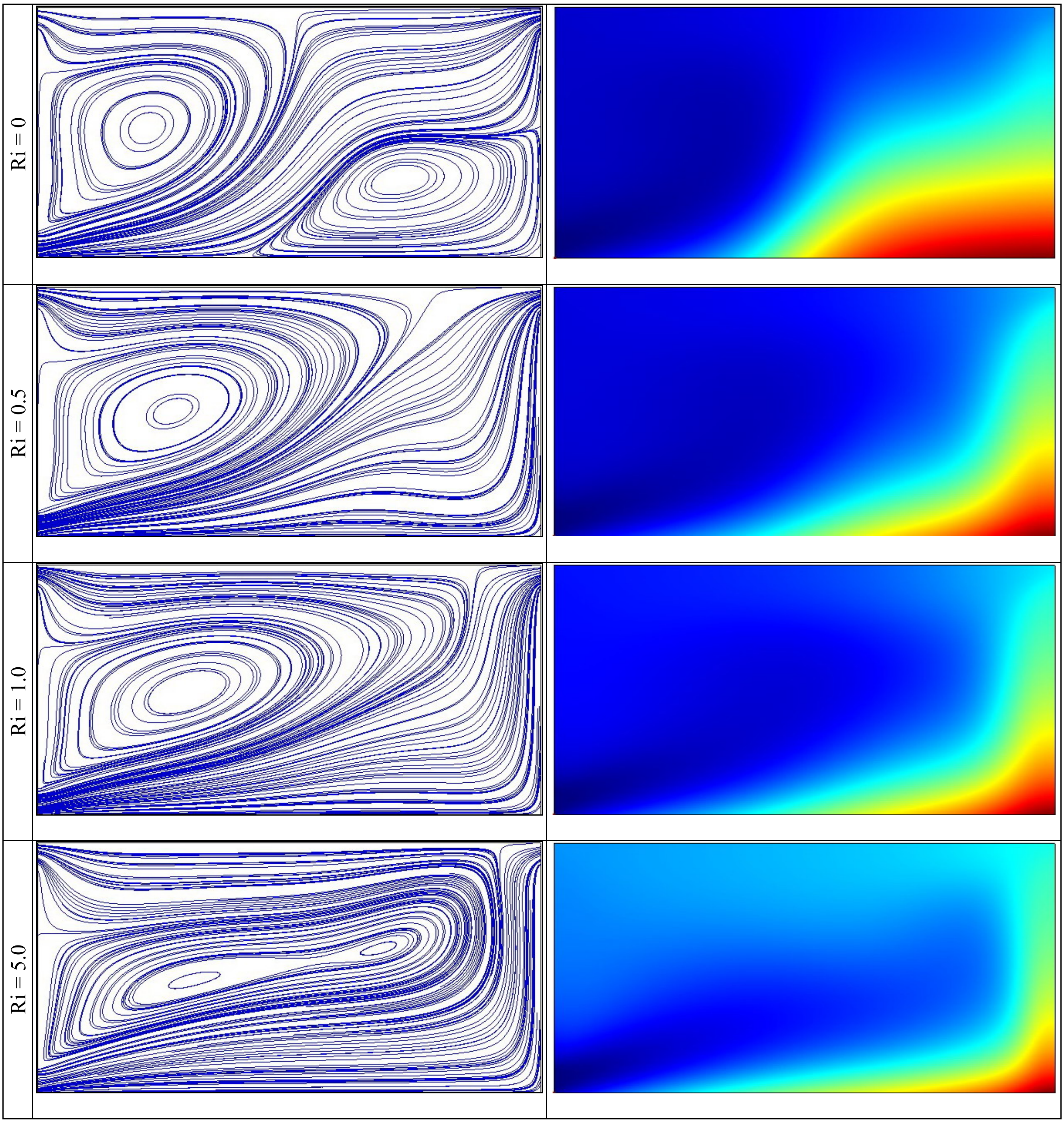


Figure-3. Streamline plots and contour map of temperature for different Richardson numbers at Re = 100.



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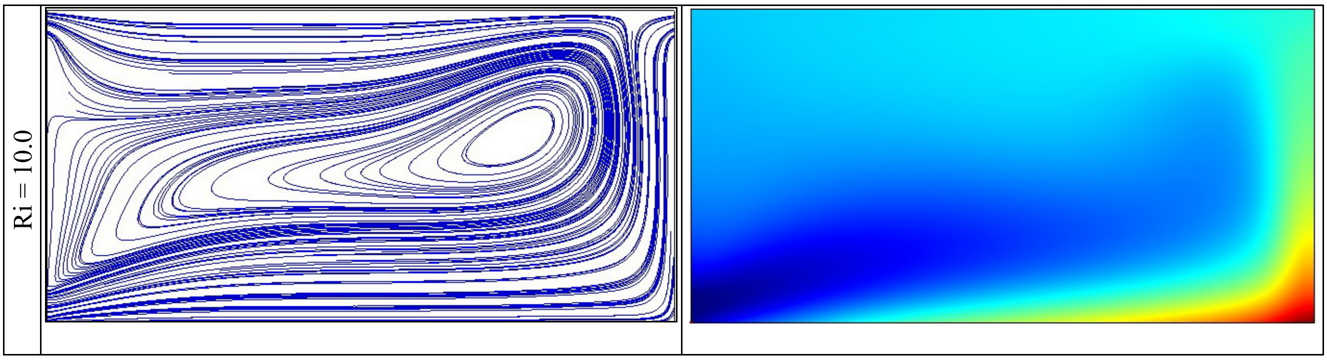

Figure-4. Streamline plots and contour map of temperature for different Richardson numbers at Re = 200.

The computer code has been validated with the solutions are available in the literatures. The test was with mixed convection from an isolated heat source in a rectangular enclosure when Re = 100 and Ri = 0.1 as considered by Manca et al (2003). Good agreement was obtained by Singh and Sharif (2003). Further, we have tested our algorithm based on the grid size (80 × 60) for a rectangular enclosure with a side wall heated and the results were in well agreement with Raji and Hasnaoui (1998) for Ra = 10^6 and Re = 100.

RESULTS AND DISCUSSION

The numerical results are performed for air with Pr of 0.71. The governing physical parameters in the problem considered are Richardson number and Reynolds number. All these values are varied over wide ranges to study the effects on the thermal transport and fluid flow phenomena. Some geometrical relationships shown in Fig. 1 are specified as follows: L/H = 2, h_i/H = 0.1 and h_o/H = 0.1. The inflow opening is located at d_i/H = 0.05 and outflow openings are at d_o/H = 0.95. The values of Re and parameter Ri are in the ranges of Re = 50 – 200 and Ri = 0 – 10. The values chosen for Re, are in the laminar regime. It is observed that, owing to the accompanying periodic oscillation behavior, the steady state solution for values of Ri higher than 10 could not be obtained for the discretization chosen. The numerical results are displayed as stream functions and isotherm contours in Figs 2-4. Variation of the average dimensionless bulk fluid temperature and average Nusselt number at the heated wall with Richardson numbers are analyzed for different Reynolds number in Figs. 5 and 6, respectively.

Characteristics of flow and thermal fields

Stream function and isotherm contours for various Ri = 0-10, and Re = 50, 100, and 200 with uniform heating of the bottom wall are displayed in Figs 2-4, respectively. As expected due to a hot horizontal wall and external cold air flow from the bottom over the heated wall, fluids move along the heated wall and rise up forming a roll with counter-clockwise rotation inside the cavity. Although the outflow openings exist in both sides of the vertical walls, the main flow exits always towards the right side opening. A diagonal nature of main flow at first is predicted for low Re and Ri. As Ri increases from 0 to 1, the values of the stream function increases, i.e., the fluid flow intensity increases. Temperature isograms confirm the above statement. No thermal stratification appears at Ri = 1, and the isotherms are only slightly distorted by the weak flow emerging from the hot source. Figure 2 shows that the isotherm surface changes its value smoothly from the hot wall to the top of the left side insulated wall. Flow advances slowly in the x direction and is slightly deviated toward the active and passive lateral walls. Some crowding of the isotherms is seen at the active corner region between the right insulated wall and the wall containing the heat source. When we increase the Richardson number up to 10, a big convective cell appears in the cavity and moving in the direction of the main flow. When the Grashof number is increased sufficiently, thermal buoyancy becomes important and fluid particles will rise from the bottom to the top along the right insulated vertical wall. The mass conservation dictates an increased fluid velocity near the bottom wall and convective cells approach the bottom wall. Convective cells located near the hot wall are crushed by the dominated buoyancy flow at high Reynolds number, whereas those located near the side wall have sufficient space to increase considerably.

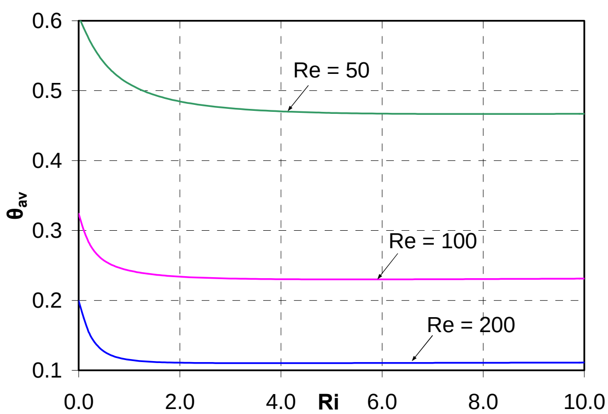

Figure-5. Variation of average dimensionless bulk fluid temperature with Richardson numbers.

Effect of Richardson Number

In order to understand the effect of Richardson number on the viscous flow and heat transfer phenomena,

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a parametric study of Ri varying from 0 to 10 is carried out. Three kinds of heat transfer regimes are observed according to the magnitude of Ri: a forced-convection-dominated regime, a mixed-convection regime, and a buoyancy-dominated (or natural-convection) regime. Figures 2-4 represent the streamline plots of the cavity for Ri = 0, 0.5, 1.0, 5.0 and 10.0, respectively. At very low values of Richardson number, such as 0 and 0.1, the forced convection due to the driven force dominates the flow structure. At this order of Ri, the inertia force of the fluid is dominant compared to the buoyancy force. As the Richardson number increases to Ri = 1, the inertia and buoyancy forces balance each other, which then results in a mixed convection. When the Richardson number further increases to Ri = 10, the buoyancy force becomes the dominant mechanism to drive the convection of the fluid, and the flow is in the regime of natural convection. More pronounced effect of heat transport by fluid convection closer to the hot walls is observed. Moreover, since the other walls are assumed to be adiabatic for heat transfer, the temperature gradient increases steeply near the boundary as the Richardson number decreases. Figures 2-4 also represent the temperature contours plotted for Ri = 0, 0.5, 1.0, 5.0 and 10, respectively. One can observe the rapid changes occurring conspicuously in the temperature contours, which indicate the effect of the increase in the Richardson number on the heat transfer process.

As shown by Singh and Sharif (2003), an arithmetic average of temperatures (θ_{av}) in the enclosure nodes is a meaningful index of heat transfer regimes when active surfaces have different boundary conditions. Figure 5 helps to find the onset of the convection regime. With the increase of the heat transfer rate, or fluid mixing, causes decreasing of θ_{av}. In the present analysis, for Ri > 1, θ_{av} depends strongly on Re, but is nearly invariant with Ri, keeping essentially the levels of pure natural convection. This suggests that in this range of Ri the degree of mixing is either very low or the mixing pattern is such that it does not contribute any significantly to convective heat transfer. A dominance of the mixed convective heat transfer mechanism is therefore inferred for Ri < 1. Conversely, it can be found that for $1 \leq Ri \leq 10$, the average temperature tends to become independent of Ri.

Nusselt number as a function of Richardson number based on enclosure height for different Reynolds numbers is shown in Fig. 6. Here three different trends of variation of average Nusselt number with the change of Richardson number are observed. When Ri increases from 0 to 1, the average Nusselt number increases very sharply. This behavior results from the onset of thermal instabilities and the probable development of secondary flow due to uniform heating and forced flow, causes rapid mixing from the bottom to the top inside the enclosure. For increment of Ri from 1 to 4, the average Nusselt number increases moderately which indicates the onset of the buoyancy dominated regime. Further increase in the value of Ri causes a slow increase in the average Nusselt number.

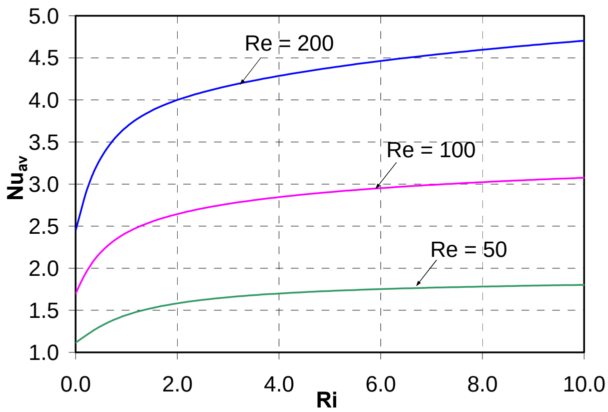

Figure-6. Variation of average Nusselt number with Richardson numbers.

Effect of Reynolds Number

The characteristics of the mixed convection phenomenon can be well understood by plotting the streamlines for various Reynolds number as shown in Figs. 2, 3 and 4. For all values of Re, the stream line patterns inside the enclosure is found to vary with Ri in a regular fashion as expected within the laminar regime. Figure 2 shows, for Re = 50 and Ri = 10, a big cell rotating in the counter-clockwise direction and located in the upper region of the cavity. It is noteworthy that the cold inlet flow descends at the heated horizontal wall and imposes rotation to the warm fluid in contact with the vertical surface. Figure 3 shows similar temperature contours at Re = 100 for various Richardson numbers. Figure 4 depicts the isotherm surface at Re = 200 for various Richardson numbers. Variation of average temperature (θ_{av}) is shown in Fig. 5 from which we can find that θ_{av} decreases markedly with the increase of Re. This was also noted by Papanicolaou anf Jaluria (1994) and Hsu *et al.* (1997). In fact, the increase of Re (increase of the forced convection effect) enhances the heat removal through the exit, i.e., to increase the air flow rate inside the cavity.

Table-1: Absolute maximum values of velocities and temperature and average Nusselt number at Ri = 0 as a function of Reynolds number.

Re	U_{max}	V_{max}	θ_{max}	Nu_{av}
50	1.0779	0.489	1.3574	1.1158
100	1.1040	0.3931	0.9883	1.6968
200	1.1355	0.4403	0.7700	2.4451

Since the Nusselt number is another characteristic concerning parameter in convective heat transfer problems, its variation along the change of Reynolds numbers is also computed and plotted in Figure-6. In this problem, Nu_{av} is a function of Re and Ri. Table-1 shows overall results for the pure forced convection case (Ri = 0). U velocities are always higher than V velocities for all Reynolds numbers, showing that the flow is dominated by an intense lateral mixing. In pure forced convection, a series of computations are carried out by increasing the Reynolds number from 50 to 200 as in Figures 2 and 4. In the present problem Nu_{av} grows very fast at low values of Ri. This shows a predominantly conductive heat transfer regime in the lower range of Ri. As the Reynolds number is increased, heat transfer enhancement above the forced convention limit initially occurs sharply up to Ri = 2. In the upper range, for individual Re the increment of Nu_{av} with Ri is higher for increasing Re. On the other hand, a reduction of Reynolds number increases the strength of the buoyancy-induced secondary flow but the corresponding effect to the increment of heat transfer rate is very slow at higher Richardson number.

Although the results appear to be strongly dependent on Reynolds number, the relative importance of forced, mixed and free convection is determined by the Richardson number Ri. It has been proposed that mixed convection is to be considered when Ri is of the order of unity. For Ri >>1, one can assume pure natural convection and for Ri << 1 it shows forced convection behavior clearly. Therefore it can be decided that the value of Ri = 1 indicates the dominance of the convective nature of flow as free or forced convection. There is a clear indication from Figures 5 and 6 that around unity of Ri, there is a change in heat convection mode.

Table-2: Nusselt numbers and average fluid temperature at Re = 200 for different outlet orientations.

Ri	Left outlet	Right outlet	Present
	Average Nusselt number (Nu_{av})		
0	1.5941899	3.1919985	2.4451845
1	3.3920577	3.9024966	3.680015
10	4.668328	4.749648	4.703847
	Average fluid temperature (θ_{av})		
0	0.6615833	0.16074586	0.19946215
1	0.2262307	0.12134964	0.11498149
10	0.16622116	0.12912723	0.111080505

Effect of outlet port orientation

The average Nusselt number and average bulk fluid temperature of the present configuration are compared with another two different cases of exit port orientation such as enclosure having only right exit port and only left exit port at the top of the side walls. The results are shown in Table-2. As the effect of Reynolds number on heat transfer rate always occurs in a consistent fashion, comparison of governing parameters is shown at Re = 200 for different Richardson numbers. From these results, highest heat transfer rate is found in case of right outlet only, whereas the present configuration ensures only moderate Nusselt number. But at low Ri, these effect significantly marked up to Ri = 1. For free convection dominant regime, increment of Nusselt number for all configurations decreased considerably. Average fluid temperature always remains highest for the left outlet exit configuration, resulting in lowest heat transfer rate in compare with other two configurations. But the present orientation provides minimum bulk fluid temperature inside the cavity for Ri ≥ 1, whereas for Ri < 1 only, the right exit port orientation maintain lowest average fluid temperature though having maximum heat transfer rate.

Heat transfer correlation

The simulated data obtained for all governing parameters are then used for the development of an empirical correlation for general convection. The complex dependence of Nu_{av} on Ri and Re apparently rules out obtaining a single equation that could correlate all results. A least-squares method is used to correlate the data in Fig. 6 for each variation of Re. It is proposed that the empirical correlation for general convection to be the following form:

$$Nu_{av} = (C \times Ri^{m} + D) \times Re^{n} \quad (11)$$

is sought, based on the numerical results at 0 ≤ Ri ≤ 10 and Re = 50, 100 and 200. The best fit is found at values: C = 0.04392, m = 0.251, D = 0.073, n = 0.6516 and finally shown as:

$$Nu_{av} = (0.04392 \times Ri^{0.251} + 0.073) \times Re^{0.6516} \quad (12)$$

This has an overall correlation coefficient of 0.994, which shows that the fit is very good, in the range of parameter ranges considered.

CONCLUSION

Laminar convection in a two-dimensional, horizontally driven rectangular enclosure with a prescribed constant heat flux source mounted on the bottom wall is simulated numerically in this work. Mixed convection arises as the buoyancy-induced cold flow from the source interacts with an externally induced cold air flow. The effects of different ventilation orientations are also described to figure out the best cooling performance. The heat transfer results explain the importance of the non-dimensional parameters like Reynolds number and Richardson number in the natural and mixed convection regime. The effects of these parameters on the flow fields are also investigated. A sufficiently large eddy has been observed and investigated at low Ri and high Re. The important observations made on the mixed convection in a cavity are summarized as follows.

i. The numerical solutions indicate that increasing the value of Re or Gr leads to higher heat transfer coefficient, higher heat source temperature, and higher intensity of recirculation.
ii. The governing parameter affecting heat transfer is the Richardson number. For Ri > 1, the heat transfer is dominated by natural convection. When Ri < 1, the flow and heat transfer are dominated by forced convection. The mixed regime is obtained when Ri = 1.
iii. The average Nusselt number plotted along the variation of Richardson number indicates that the heat transfer from the heated wall increases rapidly

up to mixed convection regime, then it increases at a slow rate.

iv. The average bulk fluid temperature seems to be remain constant in the highly buoyancy dominated convection regime.

A correlation among Nu_{av}, Re and Ri is derived to evaluate the effect of Re and Ri on Nu_{av}. It can be found that the average Nusselt number from the source is only slightly affected by the Reynolds number and is found to vary as $Ri^{0.251}$.

Nomenclature

C_p	specific heat of the fluid at constant pressure ($Jkg^{-1}K^{-1}$)
C	pre-factor in correlation
D	additional factor in correlation
d_i	distance of the middle of the inlet port from the bottom (m)
d_o	distance of the middle of the exit port from the bottom (m)
g	gravitational acceleration (ms^{-2})
Gr	Grashof number, $g\beta qH^4/\upsilon^2 k$
H	height of the vertical sidewall (m)
h	convective heat transfer coefficient ($Wm^{-2}K^{-1}$)
h_i	height of the inflow openings (m)
h_o	height of the outflow openings (m)
k	thermal conductivity of the fluid ($Wm^{-1}K^{-1}$)
L	width of the enclosure (m)
Nu	Nusselt number
p	pressure (Nm^{-2})
P	non-dimensional pressure, $p/\rho u_{in}^2$
Pe	Peclet number, $Re{\cdot}Pr$
Pr	Prandtl number, υ/α
q	heat flux (Wm^{-2})
Ra	Rayleigh number, $Gr{\cdot}Pr$
Re	Reynolds number, $u_{in}H/\upsilon$
Ri	Richardson number, Gr/Re^2
T	temperature (K)
T_{in}	inlet temperature
T_H	heated wall temperature
u, v	velocity components (ms^{-1})
U, V	non-dimensional velocity components, u/u_{in}, v/u_{in}
$\vec{V}$	dimensionless velocity vector = $(U, V) = (u/u_{in}, v/u_{in})$
x, y	Cartesian coordinates (m)
X, Y	non-dimensional Cartesian coordinates, x/H, y/H
Greek symbols	
α	thermal diffusivity, $k/\rho C_p$ (m^2s^{-1})
β	thermal expansion coefficient (K^{-1})
ρ	density of the fluid (kgm^{-3})
υ	kinematic viscosity of the fluid (m^2s^{-1})
θ	non-dimensional temperature, $(T-T_{in})/(qH/k)$
Superscripts	
m	exponent of (Ri)
n	exponent of (Re)
Subscripts	
av	average
H	heated
in	inlet state
l	local quantity
max	maximum

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