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REYNOLDS AND PRANDTL NUMBERS EFFECTS ON MHD MIXED CONVECTION IN A LID-DRIVEN CAVITY ALONG WITH JOULE HEATING AND A CENTERED HEAT CONDUCTING CIRCULAR BLOCK

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ABSTRACT

The fluid flows and heat transfer induced by the combined effects of mechanically driven lid and buoyancy force within a rectangular cavity is investigated in this paper numerically. The horizontal walls of the enclosure are insulted while the right vertical wall is maintained at a uniform temperature higher than the left vertical wall. In addition, it contains a heat conducting horizontal circular block in its centre. The governing equations for the problem are first transformed into a non-dimensional form and the resulting nonlinear system of partial differential equations are solved by using the finite element formulation based on the Galerkin method of weighted residuals. The analysis is conducted by observing the variations of streamlines and isotherms for different Reynolds number and Prandtl number ranging from 50 to 200 and from 0.071 to 3.0 respectively. The results indicated that both the streamlines and isotherms strongly depend on the Reynolds number and Prandtl number. Moreover, the results of this investigation are also illustrated by the variations of average Nusselt number on the heated surface and average fluid temperature in the enclosure.

Keywords: Mixed convection, Galerkin method, Circular block and Enclosure.

NOMENCLUTURE

B_0	magnetic induction (Wb/m^2)
C_p	Specific heat of fluid at constant pressure
D	block of diameter
g	gravitational acceleration (ms^{-2})
Gr	Grashof number
h	convective heat transfer coefficient(Wm^{-2}K^{-1})
Ha	Hartmann number
J	Joule heating parameter
k_f	thermal conductivity of the fluid (Wm^{-1}K^{-1})
k_s	thermal conductivity of the solid (Wm^{-1}K^{-1})
K	solid fluid thermal conductivity ratio
L	length of the cavity (m)
Nu	average Nusselt number
p	dimensional pressure (Nm^{-2})
P	dimensionless pressure
Pr	Prandtl number
Ra	Rayleigh number
Re	Reynolds number
Ri	Richardson number
T	dimensional temperature(K)
ΔT	temperature difference (K)
u, v	dimensional velocity components (ms^{-1})
U, V	dimensionless velocity components
U_0	lid velocity
$\overline{V}$	cavity volume (m^3)
x, y	Cartesian coordinates (m)
X, Y	dimensionless cartesian coordinates

Greek symbols

α	thermal diffusivity (m^{2}s^{-1})
β	thermal expansion coefficient (K^{-1})
ν	kinematic viscosity (m^{2}s^{-1})
θ	dimensionless temperature
μ	dynamic viscosity (m^{2}s^{-1})
ρ	density of the fluid (kgm^{-3})
σ	fluid electrical conductivity (Ω^{-1}.m^{-1})
ψ	stream function

Subscripts

av	average
h	heated wall
c	cold wall
s	solid

1. INTRODUCTION

1.1 Literature Review

Flow and heat transfer phenomena in lid-driven enclosures caused by the conjugate effect of buoyancy and shear forces. Mixed convection flow and heat transfer in lid-driven enclosures have been receiving a considerable attention in the literature. Conjugate mixed convection heat transfer has possible applications in many engineering and natural processes. Such applications include cooling of electronic devices, lubrication technologies, drying technologies, food processing, and flow and heat transfer in solar ponds, thermal –hydraulics of nuclear reactors etc. Flow and heat transfer from obstructed enclosures are often encountered in many engineering applications to enhance heat transfer such as micro-electronic device, flat plate solar collectors, and flat-plate condensers in refrigerator etc. This kind of problems was investigated mostly for the case of natural convection in enclosures. Roychowdhury et al. (2002) analyzed the natural convective flow and heat transfer features for a heated cylinder kept in a square enclosure with different thermal boundary conditions. Dong and Li (2004) and Hasanuzzaman et al. (2007) studied conjugate of natural convection and conduction in a complicated enclosure. The authors investigated the influences of material character, geometrical shape and Rayleigh number on the heat transfer in overall concerned region and concluded that the flow and heat transfer increase with the increase of thermal conductivity in the solid region; both geometric shape and Rayleigh number affect the overall flow and heat transfer significantly. Hasanuzzaman et al. (2009) investigated the nature convection in closed cavity of refrigerator. Braga and Lemos (2005) numerically studied steady laminar natural convection within a square cavity filled with a fixed amount of conducting solid material consisting of either circular or square obstacles. The authors showed that the average Nusselt number for cylindrical rods is slightly lower than those for square rods. Lee and Ha (2005) investigated natural convection in a horizontal layer of fluid with a conducting body in the interior, using an accurate and efficient Chebyshev spectral collocation approach. Later on, the same authors Lee and Ha (2006) also studied natural convection in horizontal layer of fluid with heat generating conducting body in the interior. Kumar and Dalal (2006) studied natural convection around a tilled heated square cylinder kept in an enclosure in the range of $10^3 \leq Ra \leq 10^6$. The authors reported detailed flow and heat transfer features for two different thermal boundary conditions and found that the uniform wall temperature heating is quantitatively different from the uniform wall heat flux heating.

The cavity with moving lid has the most important application for these heat transfer mechanism, which is seen in cooling of electronic chips, solar energy collection and food industry etc. Numerical analysis of these kinds of systems can be found in many literatures. Moallemi and Jang (1992) investigated mixed convective flow in a bottom heated square lid-driven cavity. The authors studied the effect of Prandtl number on the flow and heat transfer process. They found that the effects of buoyancy are more pronounced for higher values of Prandtl number and also derived a correlation for the average Nusselt number in terms of the Prandtl number, Reynolds number and Richardson number. Aydin and Yang (2000) numerically studied mixed convection heat transfer in a two-dimensional square cavity having an aspect ratio of 1. In their configuration the isothermal sidewalls of the cavity were moving downwards with uniform velocity while the top wall was adiabatic. Oztop and Dagtekin (2004) investigated numerically steady state two-dimensional mixed convection problem in a vertical two-sided lid-driven differentially heated square cavity.

A combined free and forced convection flow of an electrically conducting fluid in cavities in the presence of a magnetic field is of special technical significance because of its frequent occurrence in many industrial applications such as geothermal reservoirs, cooling of nuclear reactors, thermal insulations and petroleum reservoirs. These types of problems also arise in electronic packages, microelectronic devices during their operations. Garandet et al. (1992) studied natural convection heat transfer in a rectangular enclosure with a transverse magnetic field. Rudraiah et al. (1995a) investigated the effect of surface tension on buoyancy driven flow of an electrically conducting fluid in a rectangular cavity in the presence of a vertical transverse magnetic field to see how this force damps hydrodynamic movements. At the same time, Rudraiah et al. (1995b) also studied the effect of a magnetic field on free convection in a rectangular enclosure. The problem of unsteady laminar combined forced and free convection flow and heat transfer of an electrically conducting and heat generating or absorbing fluid in a vertical lid-driven cavity in the presence of a magnetic field was formulated by Chamkha (2002). Recently, Rahman et al. (2009a) studied MHD mixed convection around a heat conducting horizontal circular cylinder placed at the center of a rectangular cavity along with joule heating. Very recent, the effect of a heat conducting horizontal circular cylinder on MHD mixed convection in a lid-driven cavity along with joule heating is investigated by Rahman et al. (2009b). The authors concluded the streamlines and isotherms strongly depend on the size and locations of the inner cylinder, but the thermal conductivity of the cylinder has significant effect only on the isothermal lines.

1.2 Objectives of the Present Study

In this paper, the effect of Reynolds and Prandtl number on MHD mixed convection in a lid-driven rectangular cavity along with a heat conducting circular block is to analyze using Finite Element technique. From the literature review it is clearly seen that no work has been paid on such research interest.

2. PHYSICAL DOMAIN

The configuration under study with the system of co-ordinate is sketched in Figure 1. Fluid flows and heat transfer modeled in a two-dimensional rectangular lid-driven cavity of length L with a heat conducting horizontal circular block of diameter D = 0.2, placed at the centre of the cavity. The left wall of the cavity is set to move upward on its own plane at a velocity U_0. Horizontal walls of the cavity are insulated while the left and right vertical walls are isothermal but the temperature of the right wall is higher than that of the left wall. The fluid permeated by a uniform external magnetic field B_0. The resulting convective flow is governed by the combined mechanism of driven (share and buoyancy) force and the electromagnetic retarding force. The magnetic Reynolds number is assumed to be small so that the induced magnetic field produced by the motion of the electrically conducting fluid is negligible compared to the applied magnetic field B_0. The density variation considered only in the body force term according to the Boussinesq approximation. In addition, joule heating is considered, but pressure work, radiation and viscous dissipation are supposed to be negligible. All solid boundaries are assumed to be rigid no-slip walls.

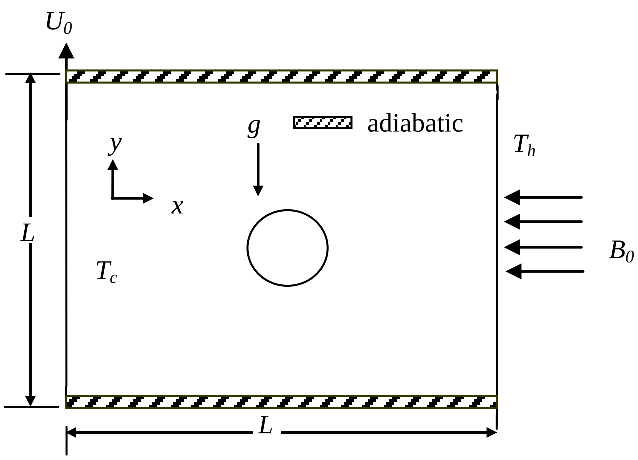

Figure 1 Schematic representation of the physical model with boundary conditions

2.1 Mathematical Modeling

The flow of enclosed fluid is considered to be tow-dimensional, steady, incompressible and laminar. In addition, the electrically conducting fluids are assumed to be Newtonian with all the fluid properties taken as except for the density variation with temperature for Boussinesq approximation. The governing equations for the system of enclosed fluid are the expression for the conservation of mass, momentum and energy transport at every point of the system within the limit of basic assumptions and the use of appropriate boundary conditions. Moreover, the equation of continuity and u-momentum equation remain unchanged, but the equation of v-momentum is modified from Maxwell's field equation and Ohm's law. On the other hand, the equation of energy is modified due to joule heating considered in the cavity.

The governing equations are non-dimensionalzed to yield

The continuity equation

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \qquad (1)$$

The momentum equations

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

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

The thermal energy transport equation

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

For the solid circular block

$$\frac{\partial^2 \theta_s}{\partial X^2}+\frac{\partial^2 \theta_s}{\partial Y^2}=0 \qquad (5)$$

The following non-dimensional variables are used in the above equations.

$$X=\frac{x}{L},\ Y=\frac{y}{L},\ U=\frac{u}{U_0},\ V=\frac{v}{U_0},\ P=\frac{p}{\rho {U_0}^2},$$

$$\theta=\frac{(T-T_c)}{(T_h-T_c)},\ \theta_s=\frac{(T_s-T_c)}{(T_h-T_c)}$$

The dimensionless parameters that have been appeared in the above equations are:

$$Re=U_0L/\upsilon,\ Gr=g\beta\Delta TL^3/\upsilon^2,\ Ri=Gr/Re^2,$$

$$Ha^2=\sigma B_0^2L^2/\mu,\ Pr=\nu/\alpha,\ J=\sigma B_0^2LU_0/\rho C_p\Delta T,$$

$$K=k_s/k_f$$

Here Re, Gr, Ri, Ha and Pr are the familiar Reynolds, Grashof, Richardson, Hartmann and Prandtl numbers respectively. We propose that J be called the Joule heating parameter. We also propose that K be called solid fluid thermal conductivity ratio.

2.2 Boundary Conditions

The dimensionless form of the boundary conditions, associated to the problem are:

$U=0, V=1, \theta=0$ on the left vertical wall

$U=0, V=0, \theta=1$ on the right vertical wall

$U=0,\ V=0$ on the solid surface

$U=0,\ V=0,\ \frac{\partial\theta}{\partial N}=0$ at the top and bottom walls

$\left(\frac{\partial\theta}{\partial N}\right)_{\text{fluid}}=K\left(\frac{\partial\theta_s}{\partial N}\right)_{\text{solid}}$ at the fluid-solid interface

Where N is the non-dimensional distances either along X or Y direction acting normal to the surface.

The average Nusselt number at the heated wall of the cavity based on the dimensionless quantities may be defined by $Nu = -\int_0^1 \frac{\partial \theta}{\partial X} dY$ and the average temperature of the fluid in the cavity is defined by $\theta_{av} = \int \theta \, d\bar{V} / \bar{V}$, where $\bar{V}$ is the cavity volume.

The dimensionless stream function is defined as

$$U = \frac{\partial \psi}{\partial Y}, V = -\frac{\partial \psi}{\partial X}.$$

2.3 Numerical Procedure

The governing equations together with the boundary conditions are solved numerically by a Galarkin finite element method. The method for analyzing mixed convection in an obstructed vented cavity in our recent previous study Rahman et al. (2009c) is employed in order to investigate the mixed convection in an obstructed lid-driven cavity with slight modification.

In order to ensure the grid-independence solutions, a series of calculation were conducted for different grid distributions. The details of grid independence test are available in Rahman et al. (2009b).

The accuracy of the numerical model was also checked by comparing the results from the present study with those obtained by Chamkha (2002). The comparison is well listed in Rahman et al. (2009a).

3. RESULTS AND DISSCUSSION

The main objective of this investigation is to analyze the effects of *Re* and *Pr* on the flow and heat transfer in a lid-driven cavity. The mixed convection phenomenon inside an obstructed lid-driven cavity is influenced by the Reynolds number *Re*, Prandtl number *Pr*, Richardson number *Ri*, Hartmann number *Ha*, Joule heating parameter *J*, solid fluid thermal conductivity ratio *K*. In order to focus on the effect of *Re* and *Pr* at the three convective regimes in the cavity, we assign *Ha* =10.0, *J* =1.0, *K* =5.0. In addition, the effect of cavity aspect ratio and Hartmann number on the flow and heat transfer characteristics have already been reported by Rahman et al. (2009a). Very Recent, the effect of the size, location and the thermal conductivity of the inner cylinder on the flow and heat transfer in the cavity have been recorded by Rahman et al. (2009b). The results are presented in terms of streamline and isotherm patterns at the three different regimes of flow, viz., pure forced convection, mixed convection and dominating natural convection with *Ri* = 0.0, 1.0 and 5.0 respectively. The variations of the average Nusselt number at the heated surface and average fluid temperature in the cavity are plotted for the different values of the parameters. In addition, the variations of the average Nusselt number at the heated surface are also decorated in tabular form.

3.1 Effect of Reynolds number

The effect of Reynolds number on the flow fields as streamlines in the cavity operating at three different values of *Ri*, while the values of *Pr* = 0.71 is keeping fixed presented in the Figure 2. As well known from the literature, the values of the Richardson number *Ri* is a measure of the importance of natural convection to forced convection. Left column of this Figure shows that the forced convection plays a dominant role; as a result the recirculation flow is mostly generated only by the moving lids at low *Ri* (*Ri* = 0.0) and all the values of *Re* considered, as expected which is due to the upward motion of the left wall. It is also observed that the orientation of the core in the recirculation cell changes as Reynolds number *Re* changes. Further at *Ri* = 1.0, two counter rotating cells are developed in the cavity for all the values of *Re*, which indicates both the buoyant force and the shear driven force are presented in the cavity. In this folder at low *Re* = 50 the clockwise cell, which is due to shear driven force occupies most of the part of the cavity and two small anticlockwise rotating cells due to buoyant force are developed at the bottom and top corner in the cavity near the right wall. But, dramatically, the anticlockwise rotating cell is reduced with increasing *Re*. Moreover, Bouncy dominated streamlines are observed at *Ri* = 5.0 for different values of *Re*.

The corresponding effect of the Reynolds number on thermal fields as isotherms at various values of *Ri* is shown in the Figure 3. From this figure, we can ascertain that for *Ri* = 0.0 and *Re* = 50, the isothermal lines near the hot wall are parallel to the heated surface and parabolic shape isotherms are seen at the left top corner in the cavity, which is similar to forced convection and conduction like distribution. It is also seen that isothermal lines start to turn back from the cold wall to the hot wall near the top wall at *Ri* = 0.0 and all values of *Re* due to the dominating influence of conduction and forced convection in the upper part of the cavity. Further at *Ri* = 1.0, both the buoyant force and the shear force are same order of magnitude, the isothermal lines are nearly parallel to the vertical walls in the cavity and parabolic shape isotherms at the left top corner in the cavity becomes negligible for the lower values of *Re* (*Re* = 50, 100) and parabolic shape isotherms are seen at the right top corner in the cavity for the higher values of *Re* (*Re* = 150, 200). Moreover, the convective distortion of isothermal lines start to appear at *Re* = 50 and *Ri* = 5.0. Next at *Ri* = 5.0 and *Re* = 100, it is seen that the isothermal lines turn back towards the left cold wall near the top of the cavity and a thermal boundary layer is developed near the left vertical (cold) wall due to the dominating influence of the convective current in the upper part of the cavity. In addition, at *Ri* = 5.0, the convective distortion in the isotherms become more and the thermal boundary layer near the cold wall becomes more concentrated with further increasing the values of Reynolds number due to the strong influence of the convective current.

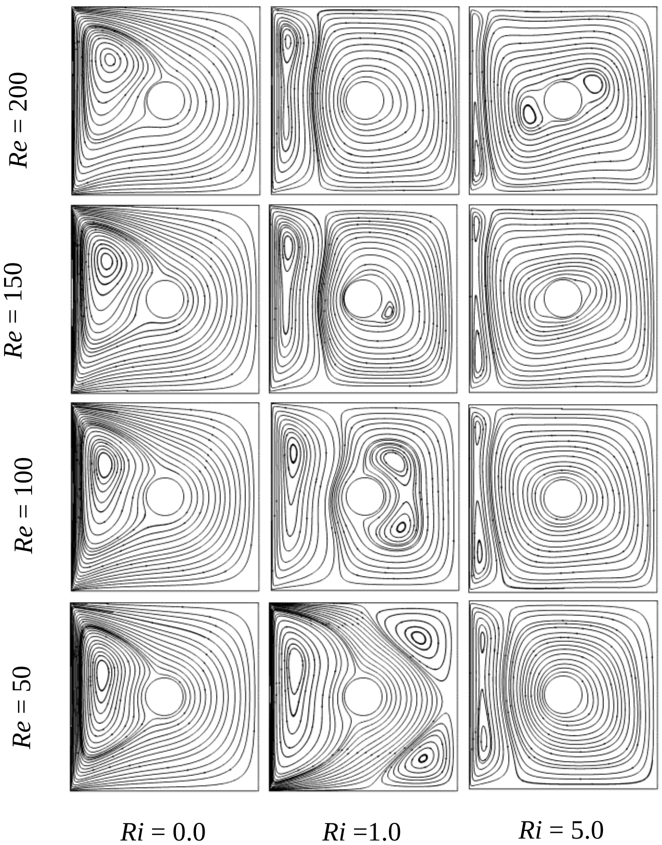

Figure 2 Streamlines for different values of *Re* and *Ri*, while *Pr* = 0.71.

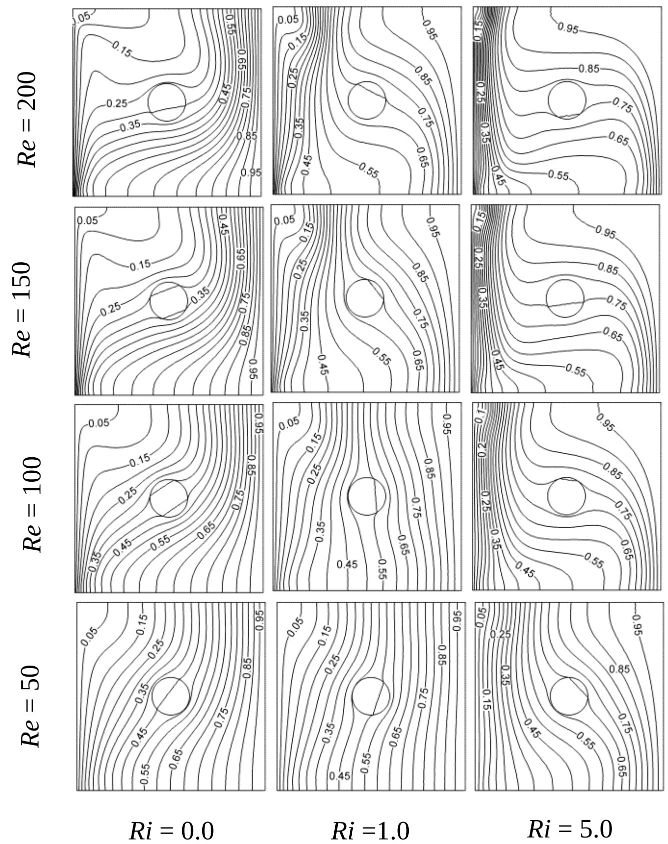

Figure 3 Isotherms for different values of *Re* and *Ri*, while *Pr* = 0.71

The effect of Reynolds number on average Nusselt number *Nu* at the heated surface and average fluid temperature θ_{av} in the cavity are displayed as a function of Richardson number at some particular Reynolds number in the Figure 4. It is observed that the average Nusselt number at the hot wall decreases very sharply in the forced convection dominated region and increases gradually in the free convection dominated region with increasing *Ri* for the higher values of Reynolds number *Re* (*Re* = 100, 150 and 200), but is different for the lowest value of Reynolds number *Re* (*Re* = 50). However, maximum values of *Nu* is found for the highest value of *Re* (*Re* = 200) at the aforesaid three convective regimes. In addition, the average fluid temperature θ_{av} in the cavity increases smoothly for higher values of *Re* (*Re* = 100, 150, 200) and increases slowly for the lowest value of *Re* (*Re* = 50) with increasing *Ri*. It is also added that the values of θ_{av} are found minimum for *Re* = 200 at the pure forced convection (*Ri* = 0.0) and for *Re* = 50 at the pure mixed convection (*Ri* = 1.0) and free convection dominated region. Finally, the quantitative differences of the values of *Nu* at different values of *Re* are documented in Table 1.

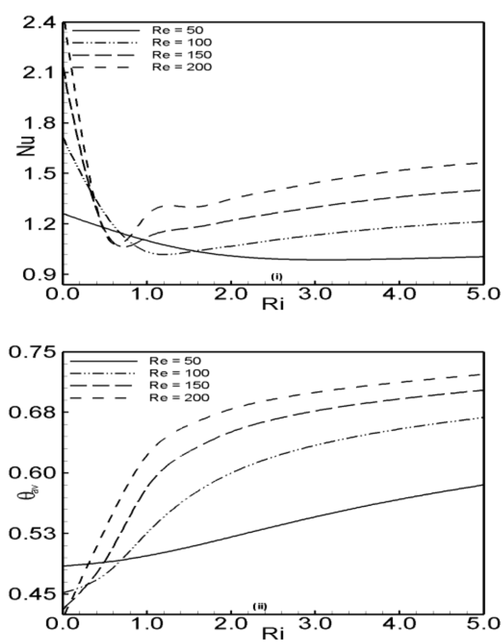

Figure 4 Effect of Reynolds number *Re* on (i) average Nusselt number and (ii) average fluid temperature while *Pr* = 0.71.

Table 1 Variation of average Nusselt number with Reynolds number.

Ri	Nu			
	Re = 50	*Re* = 100	*Re* = 150	*Re* = 200
0.0	1.259431	1.710726	2.113681	2.440418
1.0	1.086457	1.022651	1.120741	1.252180
2.0	0.992551	1.064352	1.220219	1.351091
3.0	0.973097	1.132416	1.303509	1.451683
4.0	0.982533	1.182924	1.367189	1.526451
5.0	0.997365	1.217358	1.410237	1.572461

3.2 Effect of Prandtl number

The effect of Prandtl number on the flow fields as streamlines in the cavity at three different values of *Ri* is shown in figure 5, while *Re* = 100.

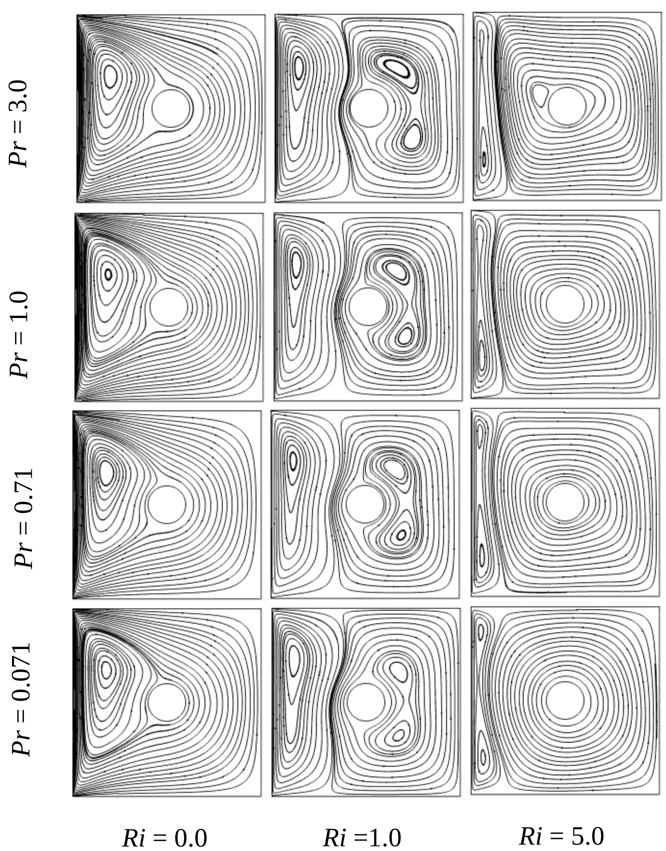

Figure 5 Streamlines for different values of *Pr* and *Ri*, while *Re* = 100

The flow fields for all values of *Pr* (*Pr* = 0.071, 0.71, 1.0 and 3.0) at low *Ri* (= 0.0) are found to be established due to the shear induced force by the moving lid only. Next at *Ri* = 1.0, the balance between the shear and buoyancy effect is manifested in the formation of two vortices inside the cavity. It is also seen that the shear effect produces the clockwise vortex, which is comparatively smaller than that of the two-cellular counter clockwise vortex produced by the buoyant force. As the Richardson number increases further to 5.0, the heat transfer is mostly by convection in the cavity, as a result the two-cellular counter clockwise vortex become uni-cellular and large enough and the clockwise vortex becomes shrink in size at each values of *Pr* considered. Besides, the flows represented by the streamlines are almost independent of the Prandtl number at each *Ri*.

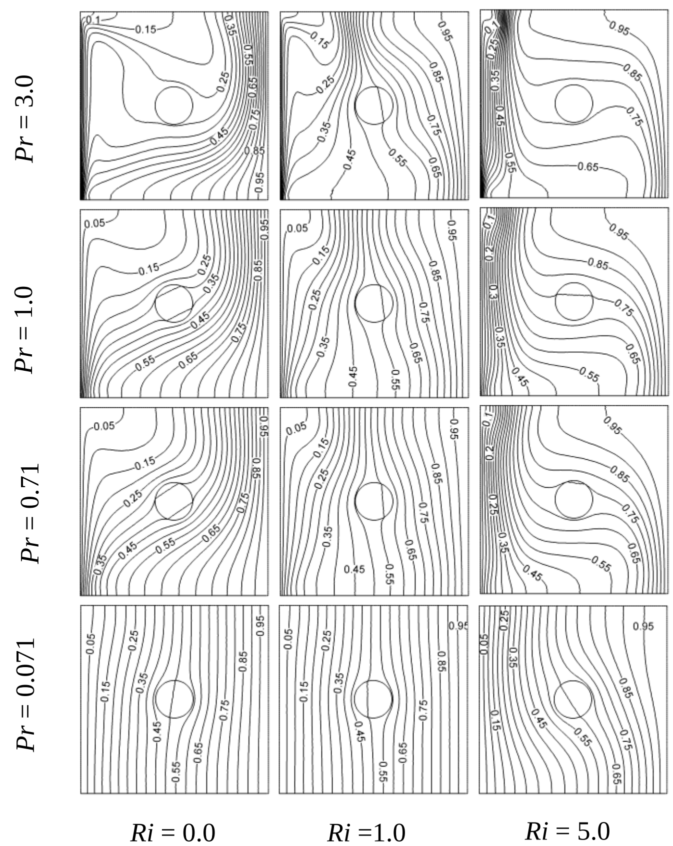

Figure 6 Isotherms for different values of *Pr* and *Ri*, while *Re* = 100.

The effect of Prandtl number on thermal characteristics as isotherms in the cavity at three different values of *Ri* are shown in figure 6, while *Re* = 100. The isotherms at very low *Pr* (*Pr* = 0.071) and lower *Ri* (*Ri* = 0.0, 1.0) become almost parallel to the vertical walls, resembling the conduction like heat transfer in the cavity. A closer examinations also show the isothermal lines are symmetric about the line *Y* = 0.5. As *Ri* increases to 5.0, the symmetry in isotherms become disappears in the cavity. But in this folder the degree of distortion from the conduction heat transfer is very noticeable and the isotherms become more packed near the left top surface in the cavity at each values of *Ri* and the higher Prandtl numbers (*Pr* = 0.71, 1.0 and

3.0). The bend in isothermal lines appears due to the high convective current inside the cavity. Figures 7 depict the variations of average Nusselt number *Nu* at the heated wall, average temperature θ_{av} of the fluid in the cavity at various values of *Pr* and *Ri*. It is shown that, for the lowest *Pr* the average Nusselt number decreases gradually with increasing *Ri* and for the higher *Pr* (*Pr* = 0.71 and 1.0) the values of *Nu* decreases with increasing *Ri* in the forced convection dominated region and increases gradually with *Ri* in the free convection dominated region. But for the highest *Pr* (*Pr* = 3.0), it is seen that the values of *Nu* decreases sharply with increasing *Ri* in the forced convection dominated region and increases smoothly up to $Ri \leq 1.8$, after then *Nu* is independent of *Ri*. In addition, maximum values of *Nu* are found for the highest value of *Pr* at all values of *Ri*. Moreover, the average fluid temperature θ_{av} in the cavity increase smoothly for higher values of *Pr* (*Pr* = 0.71, 1.0 and 3.0) and increases gradually for the lowest value of *Pr* (*Pr* = 0.071) with increasing *Ri*. On the other hand, minimum values of θ_{av} are found at the highest *Pr* (*Pr* = 3.0) in the forced convection dominated region and at the lowest value of *Pr* (*Pr* = 0.071) in the free convection dominated region. Lastly, the quantitative differences of the values of *Nu* at different values of *Pr* are tabulated in Table 2.

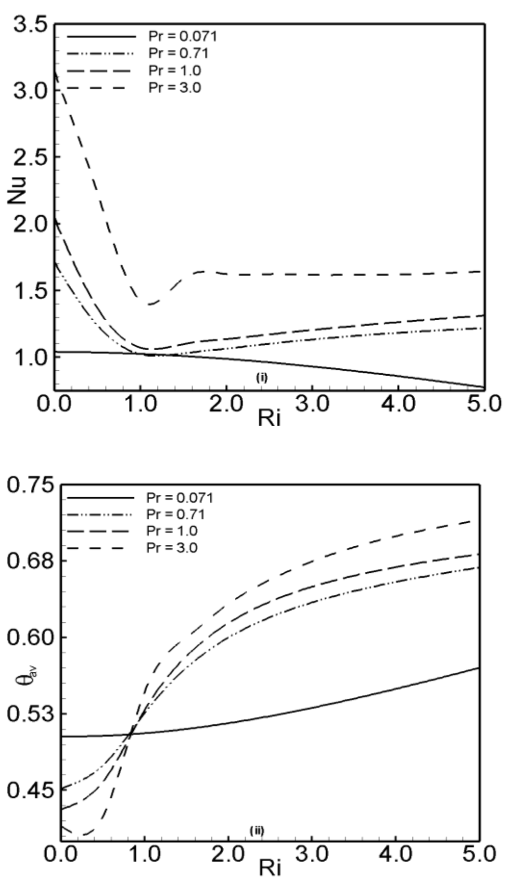

Figure 7 Effect of *Pr* on (i) average Nusselt number and (ii) average fluid temperature, while *Re* = 100.

Table 2 Variation of average Nusselt number with Prandtl number.

Ri	*Nu*			
	Pr = 0.071	*Pr* = 0.71	*Pr* = 1.0	*Pr* = 3.0
0.0	1.039148	1.710726	2.041835	3.136956
1.0	1.024186	1.022651	1.071852	1.422283
2.0	0.986682	1.064352	1.136079	1.623106
3.0	0.929720	1.132416	1.203577	1.616093
4.0	0.857280	1.182924	1.262789	1.619037
5.0	0.773427	1.217358	1.311545	1.642432

4. CONCLUSION

The present study investigates numerically the characteristics of a two dimensional MHD mixed-convection problem in a lid-driven square cavity with a heat conducting horizontal solid circular block. To simulate the flow and heat transfer in the cavity, Finite Element Method is implemented here. A detailed analysis for the distribution of streamlines, isotherms, average Nusselt number at the heated surface and average temperature of the fluid are carried out to investigate the effect of the Reynolds number and Prandtl number on the fluid flow and heat transfer in the cavity for different Richardson numbers in the range of $0.0 \leq Ri \leq 5.0$. The following conclusions are made:

- Reynolds number *Re* has a great significant effect on the streamlines and isotherms at the three convective regimes. Buoyancy-induced vortex in the streamlines increased and thermal layer near the cold surface become thin and concentrated with increasing *Re*. Reynolds number *Re* has also a great significant effect on the average Nusselt numbers, *Nu* and the average temperature, θ_{av} in the cavity.

- The influence of Prandtl number on the streamlines in the cavity is found insignificant for all the values of *Ri*, whereas the influence of *Pr* on the isotherms is remarkable for different values of *Ri*. In addition, for lower values of *Pr* the heat transfer is dominated by conduction and it become reduces with increasing *Pr*. The average Nusselt number is always superior for the larger values of *Pr*. The average temperature of the fluid in the cavity is inferior for *Pr* = 3.0 in the forced convection dominated region, also for *Pr* =0.071 in the free convection dominated region.

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