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A numerical study of mixed convection in a lid-driven porous square cavity with internal elliptic block and linearly heated side walls

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Abstract

The present work is a numerical investigation of mixed convection in a lid-driven porous square cavity with internal elliptic block and linearly heated side walls with Prandtl number of 0.7. The left and right walls of the cavity were linearly heated while the top and bottom walls were maintained at constant temperatures with the top surface being the cold wall and bottom surface being the hot wall. In the double-lid cavity configuration, the top wall moved to the right while the bottom wall moved to the left at the same constant speed. The magnetic field of strength B is applied parallel to x -axis. Finite element method was employed to solve the dimensionless governing equations of continuity, momentum and energy of the problem. Using our developed code, a parametric study was performed, and the effect of the Darcy number, Grashof number, and Reynolds number on the fluid flow and heat transfer inside the square enclosure were investigated. The isotherms are also almost symmetric at small Re with Gr ($Gr = 10^5$) and Da ($Da = 10^{-3}$) and natural convection is found to be dominant whereas the isotherms are compressed near the left and bottom walls at higher Re for linearly heated side walls. Results are presented in the form of streamlines and isotherms plots as well as the variation of the maximum temperature and Nusselt numbers under different conditions. The numerical results indicate the strong influence of the mentioned parameters on the flow structure and heat transfer as well as average Nusselt number. An optimum combination of the governing parameters would result in higher heat transfer.

Keywords: Mixed convection, square cavity, elliptic obstacle, finite element method and lid-driven cavity

1. Introduction

A mixed convection problem with lid-driven flows within an enclosure finds a wide range of applications in various fields of engineering and science, such as cooling of electronic equipments, heat exchangers, building ventilation, cooling processes, and heat exchangers. Heat transfer and fluid flow in porous media filled enclosures or channels have important engineering applications such as filtration, separation processes in chemical industries, solar collectors, heat exchangers, etc. Studies on thermal convection in porous media and state-of-the-art reviews are given by Ingham and Pop^[1], Bejan^[2], and Vafai^[3]. The literature detailed concerning the porous media in the books by J. N. Ready^[4], O. S. Zienkiewicz *et al.*^[5], T. K. Bask *et al.*^[6], M. Sathiyamoorthy^[7]. The modern technologies require through understanding of the pertinent processes involved in these fields.

1.1 Nomenclatures

Gr	Grashof number	X, Y	non-dimensional Cartesian coordinates
g	gravitational acceleration	x, y	Cartesian coordinates Greek symbols
L	length of the cavity	α	thermal diffusivity
Nu	Nusselt number	β	Volumetric coefficient of thermal expansion
P	dimensional pressure	ν	kinematic viscosity of the fluid
Pr	Prandtl number	θ	non-dimensional temperature
p	pressure	ρ	density of the fluid
Ra	Rayleigh number	ψ	non-dimensional stream function Subscripts
Re	Reynolds number	c	cold wall
Ri	Richardson number	h	hot wall
T	dimensional temperature		
U, V	non-dimensional velocity components		
u, v	velocity components		

Considerable research work has been reported in the literature on natural convection due to combined thermal and mass buoyancy forces. Unfortunately, pure natural or pure forced convection situations seldom arise in practice. The involvement of both natural and forced convection, referred as mixed convection, in porous media has been an important topic because of its wide range of application in engineering and science. The governing non-dimensional parameter for the description of flows are Grashof number (Gr), Reynolds number (Re), Prandtl number (Pr) and Darcy number (Da). Another important non-dimensional parameter, Richardson number characterizes mixed convection flow where Gr and Re represent the strength of the natural and forced convection flow effect, respectively. The detailed literature concerning convective flow in porous media is available in the books by Nield and Bejan [8], Ingham and Pop [9], Ingham [10] and Bejan *et al.* [11]. Oztop [12] investigated numerically combined convection heat transfer and fluid flow in a partially heated porous lid-driven cavity. Volume averaged equations governing unsteady, laminar, mixed convection flow in an enclosure filled with a Darcian fluid-saturated uniform porous medium in the presence of internal heat generation is investigated by Khanafer and Chamkha [13]. Chamkha [14] also studied the mixed convection flow along a vertical permeable plate embedded in a porous medium in the presence of transverse magnetic field and non-Darcy fully developed steady laminar mixed convection flow of an electrically conducting and heat generation/ absorption fluid in a porous medium channel. Mixed convection in porous cavity has also been analyzed in presence of internal heat generation. Khanafer and Chamka [15] analyzed laminar, mixed convection flow in a Darcian fluid-saturated porous cavity in the presence of internal heat generation and they found that the flow and the heat transfer inside the cavity were strong function of Richardson number (Ri). A recent investigation of Bask *et al.* [16] was based on mixed convection of fluid within a cavity in presence of uniform and non-uniform bottom heating while the top plate is maintained at uniform velocity.

On the basis of the literature review, it appears that very little work was reported on numerical investigation of mixed convection in a lid-driven porous square cavity with internal elliptic block and linearly heated side walls. Thus, the obtained numerical results of the present problem are presented graphically in terms of streamlines, isotherms, velocity, dimensionless temperature, local Nusselt number and average Nusselt number for different Grashof numbers, Darcy numbers and Reynolds numbers.

2. Problem Formulation

The physical model considered in the present study of mixed convection in a lid-driven porous square cavity with internal elliptic block and linearly heated side walls is shown in Fig. 1. The height and the width of the cavity are denoted by L . The heated wall denoted by T_h and cold wall denoted by T_c respectively.

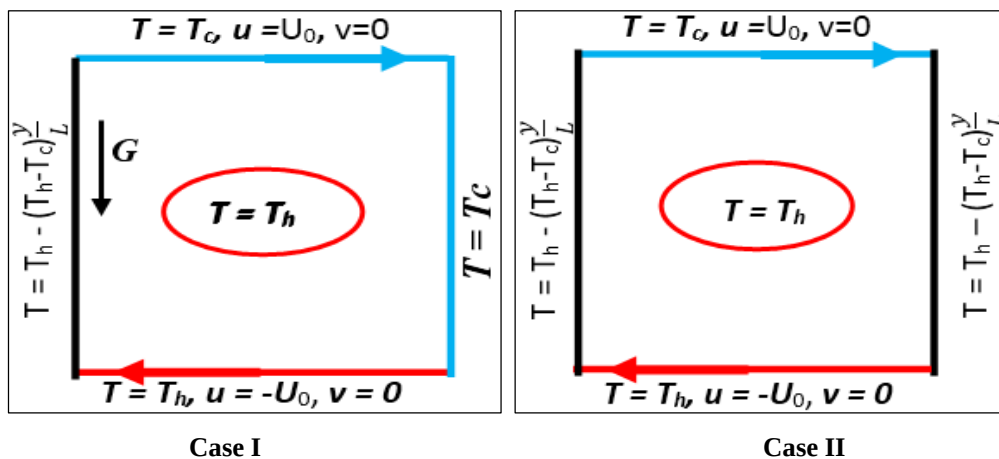


Fig 1: Schematic diagram of the physical system

The arrangement of the moving and linearly heated has great impact on the fluid flow structures in the enclosure as follows:

Case I. The left wall linearly heated, upper and right wall cold, lower wall and inside elliptic obstacle heated. The top lid is assumed to have a constant speed U_0 that slides in the positive direction of x -axis and lower lid is assumed to have a constant speed $-U_0$ that slides in the negative direction of x -axis.

Case II: The left and right wall linearly heated, upper wall cold, lower wall and inside elliptic obstacle heated. The top lid is assumed to have a constant speed U_0 that slides in the positive direction of x -axis and lower lid is assumed to have a constant speed $-U_0$ that slides in the negative direction of x -axis.

2.1 Governing Equations

The fluid is considered as incompressible, Newtonian and the flow is assumed to be laminar. The governing equations for steady two-dimensional natural convection flow in the porous cavity using conservation of mass, momentum and energy can be written as:

Continuity Equation

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0, \quad (1)$$

Momentum Equations

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \nu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) - \frac{\nu}{K} u, \quad (2)$$

$$u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial y} + \nu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) - \frac{\nu}{K} v + g\beta(T - T_c) \quad (3)$$

Energy Equations

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \alpha \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (4)$$

where x, y are the distance measured along the horizontal and vertical directions, u, v are the velocity components in the x and y directions, ρ is the density, p is the pressure, ν is the kinematic viscosity, K is the permeability of the porous medium, g is the acceleration due to gravity, α is the thermal diffusivity, β is the volume expansion coefficient, T is the temperature.

2.2 Boundary Conditions

The boundary conditions for the present problem are specified as follows:

At the cold top walls: $T = T_c, u = U_0, v = 0$

At the heated bottom wall: $T = T_h, u = -U_0, v = 0$

At the right walls: $u = v = 0, T = T_c$

Linearly heated left and right walls: $T = T_h - (T_h - T_c) y/L$

At the heated elliptic body: $u = v = 0, T = T_h$

2.3 Dimensional Analysis

The governing equations for steady two-dimensional mixed convection flow in a lid-driven porous square cavity using conservation of mass, momentum and energy can be written with the following dimensionless variables or numbers:

Continuity Equation

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0, \quad (5)$$

Momentum Equations

$$U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \frac{1}{Re} \left(\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} \right) - \frac{1}{ReDa} U, \quad (6)$$

$$U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \frac{1}{Re} \left(\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} \right) - \frac{1}{ReDa} V + \frac{Gr}{Re^2} \theta, \quad (7)$$

Energy Equations

$$U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} = \frac{1}{RePr} \left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} \right). \quad (8)$$

2.4 Transformed boundary conditions are:

At the cold top walls: $\theta = 0, U = 1, V = 0$

At the heated bottom wall: $\theta = 1, U = -1, V = 0$

Linearly heated left wall: $\theta = 1 - Y, U = 0, V = 0$

Linearly heated right wall: $\theta = 1 - Y$ or 0 , and $U = 0, V = 0$

At the heated elliptic body: $U = 0, V = 0, \theta = 1$

(9)

The following dimensionless variables and parameters have been used in Eqs. (5)-(8):

$$X = \frac{x}{L}, Y = \frac{y}{L}, U = \frac{u}{U_0}, V = \frac{v}{U_0}, \theta = \frac{T - T_c}{T_h - T_c}$$

$$P = \frac{p}{\rho U_0^2}, Pr = \frac{\nu}{\alpha}, Da = \frac{K}{L^2}, Re = \frac{U_0 L}{\nu}, Gr = \frac{g\beta(T_h - T_c)}{\nu^2} \quad (10)$$

Note that, thermal diffusivity (α) and kinematic viscosity (ν) in Eq. (8) are defined based on local thermal equilibrium between solid and fluid within a porous control volume D. A. Nield *et al.* [8] where $\alpha = \frac{k_{eff}}{\phi \rho_0 C_{pf}}$ and $\nu = \frac{\mu_f}{\rho_0}$. Here k_{eff} is the effective thermal conductivity of the porous matrix, ϕ is the porosity, C_{pf} is the specific heat of fluid, μ_f is the viscosity of fluid and $\rho_0 = \rho$ is the density of fluid at $\theta = 0$.

3. Grid Refinement Test

A grid independence test is reported with $Pr = 0.7, Re = 100, Da = 10^{-3}$ and $Gr = 10^5$ in order to decide the suitable grid size for the study. The extreme values of the mean Nusselt number (Nu) that relates to the heat transfer rate of the heated surface and the average

temperature (θ_{av}) of the fluid in the square cavity are used as sensitivity measures of the correctness of the solution. The addition of the quantities Nu and θ_{av} on the grid size are shown in Table

Nodes (elements)	3517 (6790)	658 (1212)	493 (957)
Nu	3.52087	3.52035	3.50118
θ_{av}	0.54298	0.54027	0.53855

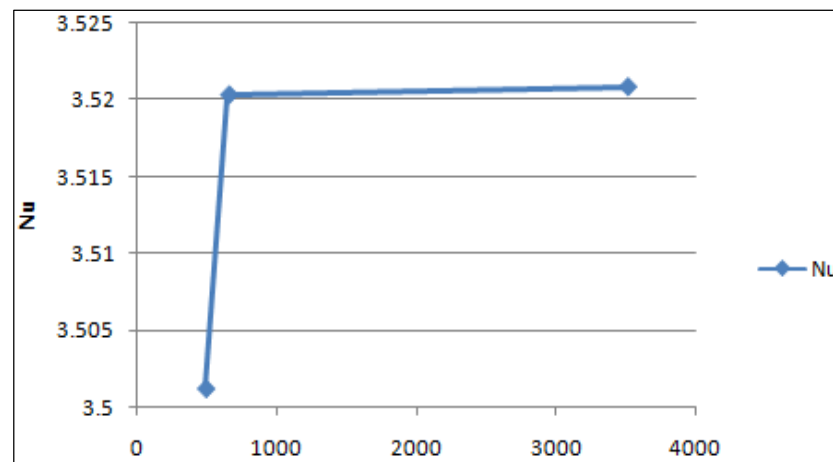


Fig 2: Effect of grid refinement test on average Nusselt number

4. Mesh Generation

In finite element method, the mesh generation is the technique to subdivide a domain into a set of sub-domains, called finite elements, control volume etc. The discrete locations are defined by the numerical grid, at which the variables are to be calculated. The computational domains with irregular geometries by a collection of finite elements make the method a valuable practical tool for the solution of boundary value problems arising the various fields of engineering. Fig. 3 displays the finite element mesh of the present physical domain.

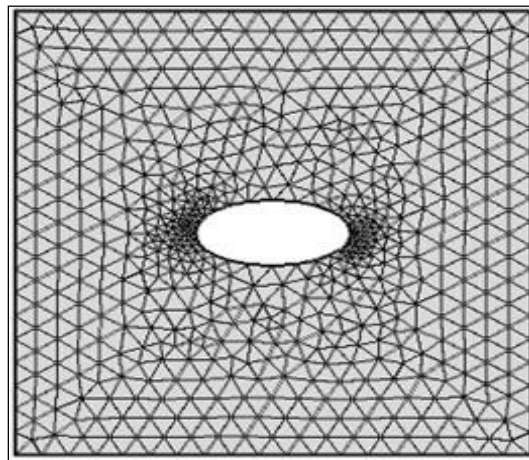


Fig 3: Mesh generation in a lid-driven porous square cavity with internal elliptic block

5. Results and Discussions

A numerical study has been performed a numerical study of mixed convection in a lid-driven porous square cavity with internal elliptic shape heated block and linearly heated side walls. The results are presented in terms of streamlines, isotherm, Local Nusselt number along the hot wall, dimensionless temperature, velocity profiles and the average Nusselt number of the hot wall.

5.1 Effect of Grashof number

Variation of streamlines and isotherms inside the cavity with different Grashof number and Darcy number are shown in Fig. 4 and 5, respectively. As can be observed from the Fig, the strength of the buoyancy inside the cavity is significant and more fluid rise from the centre of the cavity at $Gr = 10^{[3]}$. As Gr increases to $10^{[4]}$, the effect of natural convection increases and the strength of both primary and secondary circulations increase. For higher Grashof number increases to $10^{[5]}$, the strength of the buoyancy increases and two elliptic shape eyes inside the cavity and also flow strength increases in the right wall. Comment to both cases, the flowing flow due to moving lid becomes very strong due to high Grashof number. The natural convection becomes dominant as the lid-driven flow and the strength of both the circulations are increased. Due to uniform heating of the bottom wall and linearly heated side walls, fluid rise from the upper and lower walls of the cavity forming two almost symmetric rolls with clockwise rotations inside the cavity.

Conduction dominant heat transfer is observed from the isotherms in Fig. 5. We observed the isotherms line bending upper half of the square cavity for both cases. Comment to the both cases, isotherms lines are developed.

The local Nusselt number (Case I) along the bottom wall for different Grashof number with $Da = 10^{-3}$, $Re = 100$ and $Pr = 0.7$ of the cavity are shown in Fig. 6. Maximum shape curves are obtained here. For lower Grashof number the value of local Nusselt number smaller changed but higher Grashof number has larger change. The local Nusselt number (Case II) along the left wall for different Grashof number. It can be seen from the Fig. that the maximum and minimum value of local Nusselt number increasing with increasing the Grashof number.

Variations of the vertical velocity component along the bottom wall (Both cases) for different Grashof number with $Da = 10^{-3}$, $Re = 100$ and $Pr = 0.7$ of the cavity are shown in Fig. 6. From the Fig. it can be observed that the maximum and minimum value of velocity increases with increasing the Grashof number.

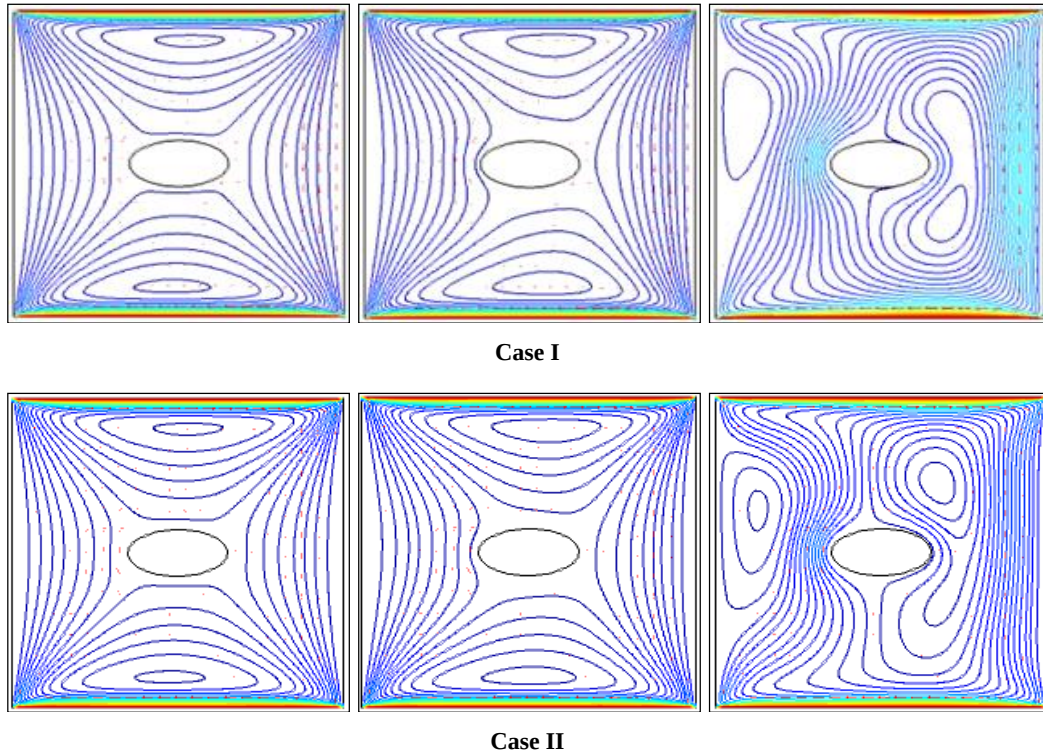


Fig 4: Illustrate the streamlines involving heated bottom moving wall, linearly heated left and right wall, upper cold moving wall and elliptic shape heated with $Pr = 0.7$, $Re = 100$ and $Da = 10^{-3}$ for $Gr = 10^3$ to 10^5 .

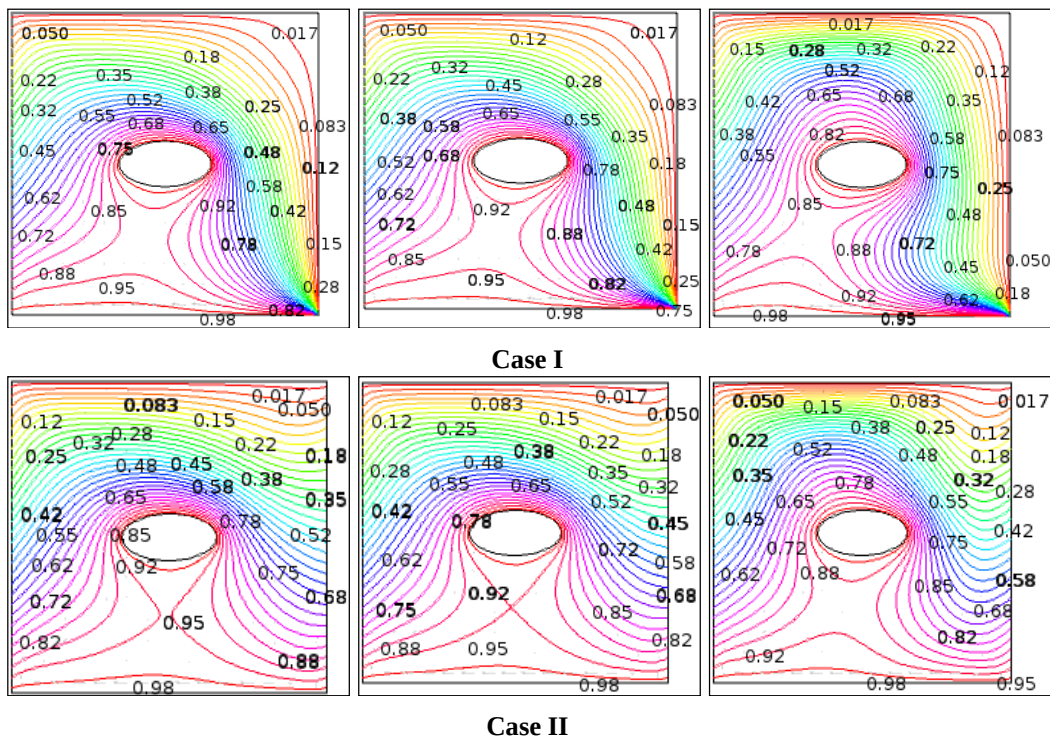


Fig 5: Illustrate the isotherms involving heated bottom moving wall, linearly heated left and right wall, upper cold moving wall and elliptic shape heated with $Pr = 0.7$, $Re = 100$ and $Da = 10^{-3}$ for $Gr = 10^3$ to 10^5 .

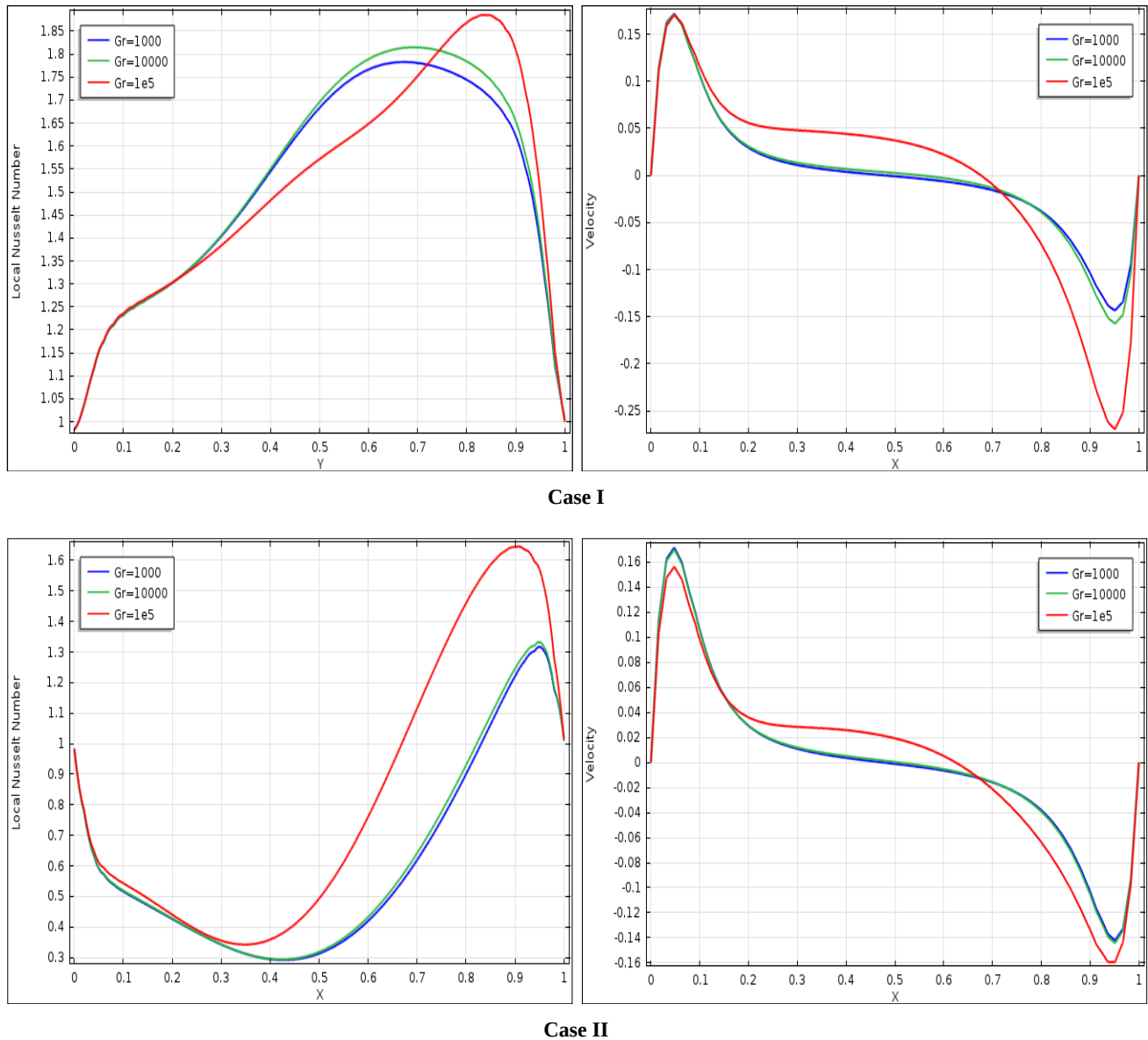


Fig 6: Variation of Local Nusselt number and Velocity of the square cavity with Grashof number $Gr = 10^3$ to 10^5 .

5.2 Effect of Darcy number

Variation of streamlines and isotherms of the lid-driven square cavity inside the elliptic shape block are shown in Fig. 7 and 8, respectively. As can be seen from the streamlines in the Fig. 7, a pair of counter- rotating elliptic shape eyes is formed in the left upper corner and right middle of the cavity for all Darcy numbers and Rayleigh numbers considered. Each cell ascends through the symmetry axis, this phenomenon means increasing Darcy number in the flow velocity increasing. Conduction dominant heat transfer is observed from the isotherms in Fig. 8, at Darcy and Rayleigh numbers. As can be seen from the isotherm lines are more bending of left and right walls of the square cavity. Darcy number increasing isotherms condense left and right wall which means increasing heat transfer through convection. Finally from the isotherms, it can observe that with increase in Darcy number, mixed convection is suppressed and heat transfer occurs mainly through conduction.

The local Nusselt number (Case I) along the bottom wall for different Darcy number with $Gr = 10^5$, $Re = 100$ and $Pr = 0.7$ of the cavity are shown in Fig. 9. Maximum shape curves are obtained here. For lower Darcy number the value of local Nusselt number smaller changed but higher Darcy number has larger change. The local Nusselt number (Case II) along the bottom wall for different Darcy number. It can be seen from the Fig. that the maximum and minimum value of local Nusselt number increasing with increasing the Darcy number.

Variations of the vertical velocity component along the bottom wall (Both cases) for different Darcy number with $Gr = 10^5$, $Re = 100$ and $Pr = 0.7$ of the cavity are shown in Fig. 9. From the Fig. it can be observed that the maximum and minimum value of velocity increases with increasing the Darcy number. For higher Darcy number the value of velocity has larger change.

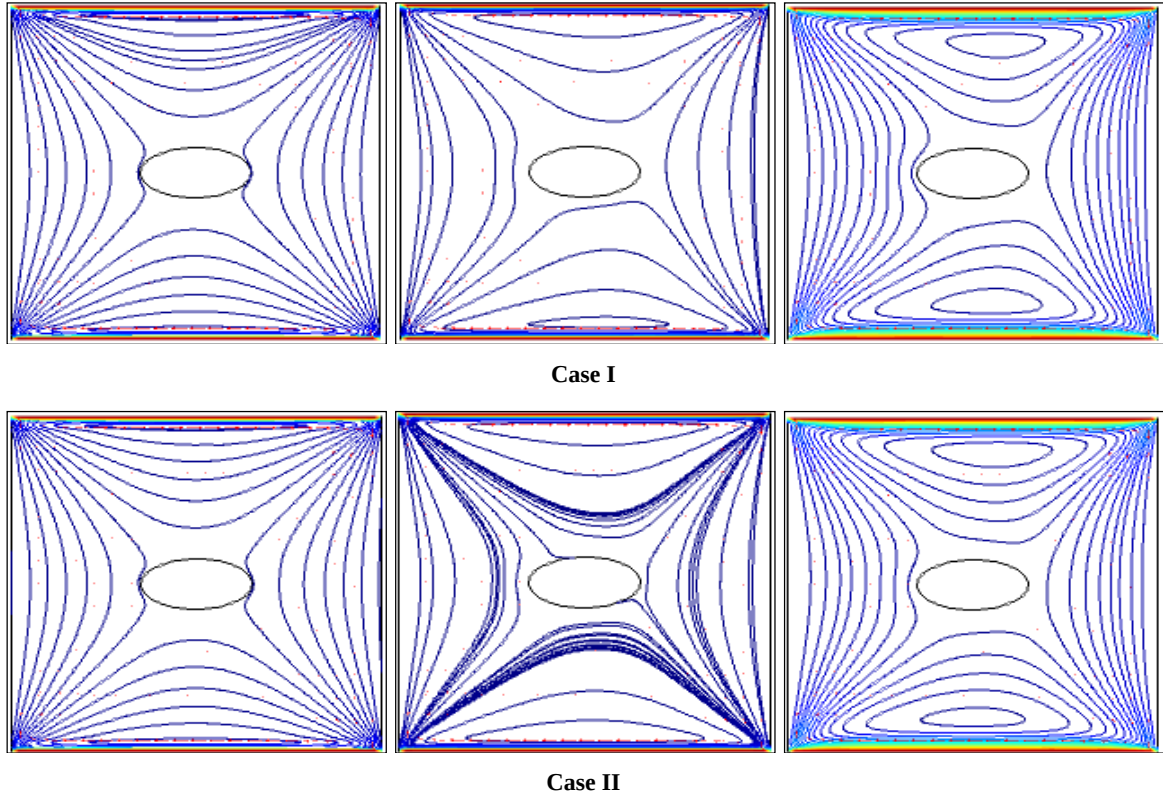


Fig 7: Streamlines and isotherms for linearly heated left walls and cold right wall with $Pr = 0.7$, $Re = 100$ and $Gr = 10^5$ for: (a) $Da = 10^{-5}$; (b) $Da = 10^{-4}$; (c) $Da = 10^{-3}$.

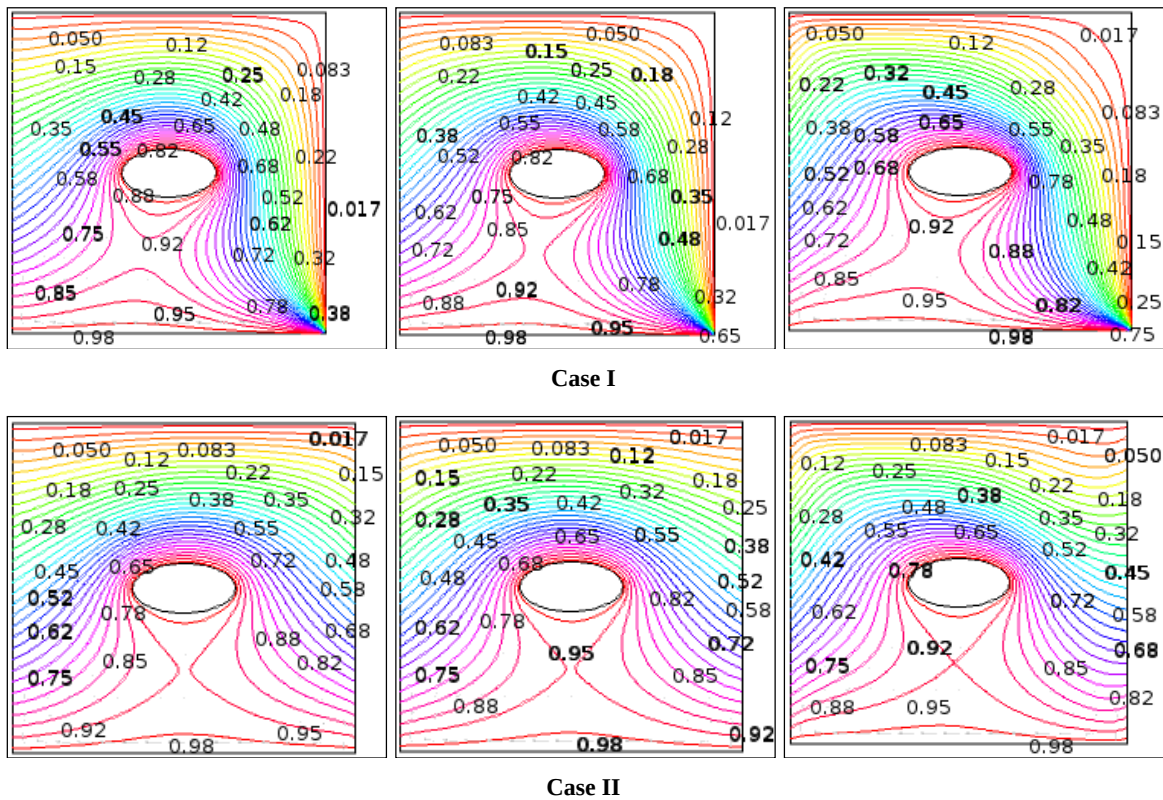
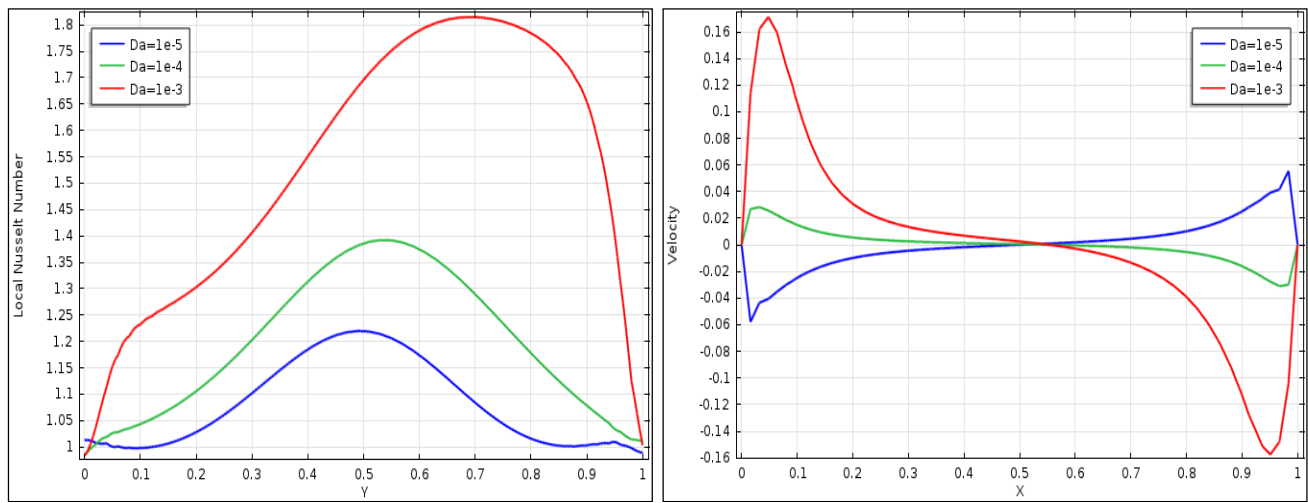
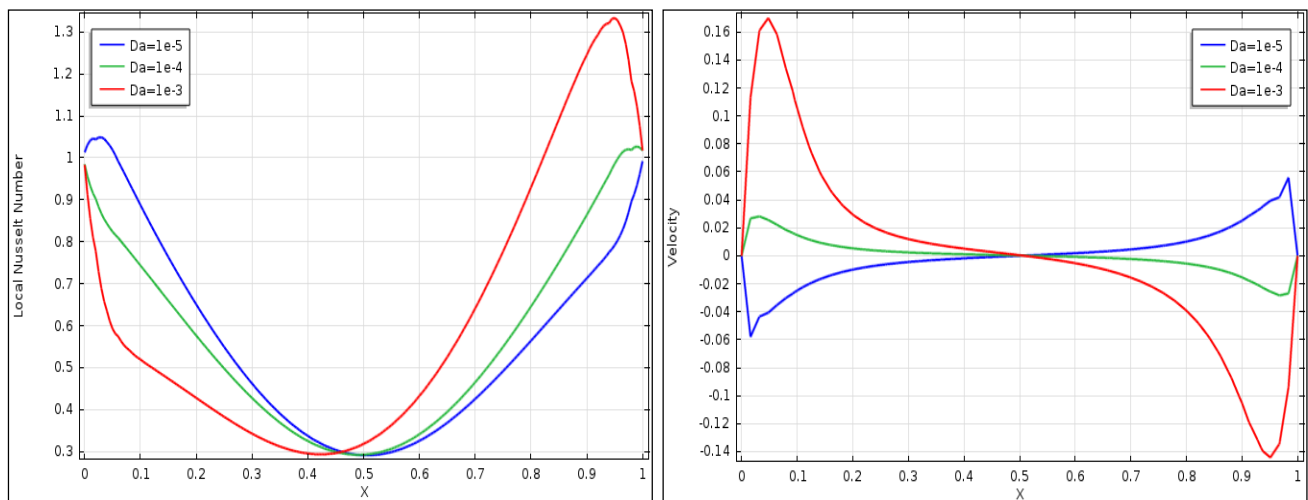


Fig 8: Isotherms for linearly heated left wall and cold right wall with $Pr = 0.7$, $Re = 100$ and $Gr = 10^5$ for: (a) $Da = 10^{-5}$; (b) $Da = 10^{-4}$; (c) $Da = 10^{-3}$.



Case I



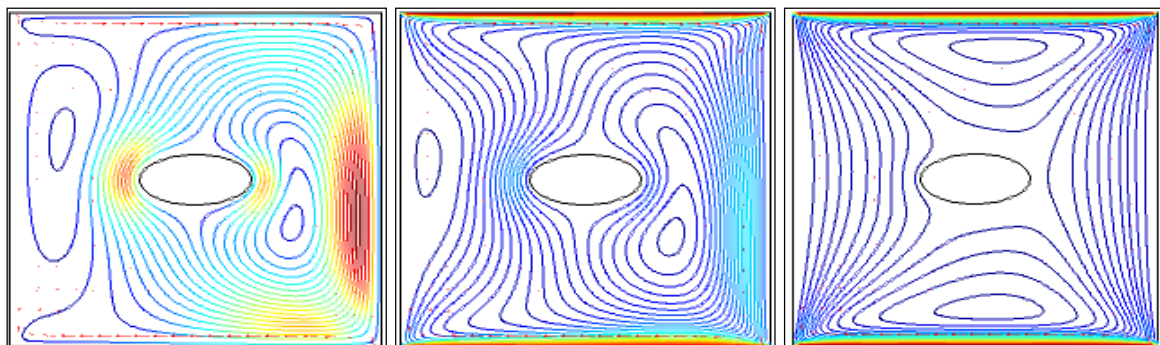
Case II

Fig 9: Variation of temperature gradient along the right wall of the square cavity with Darcy number

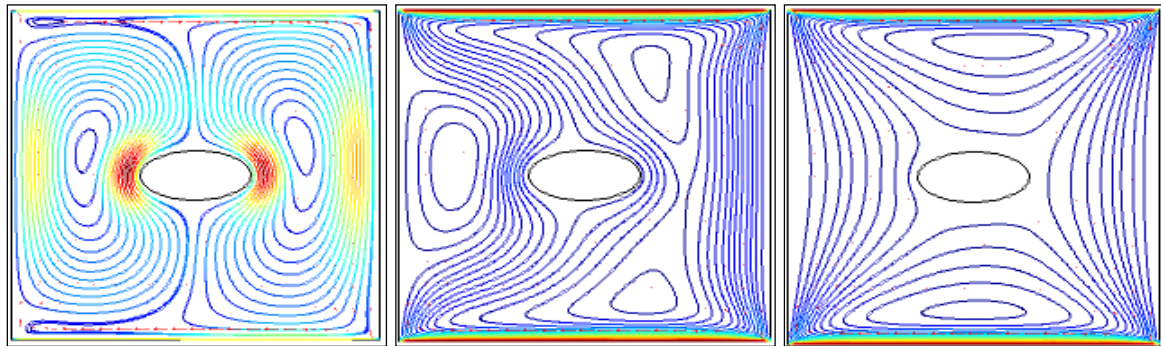
5.3 Effect of Reynolds number

The effect of Reynolds numbers on the flow fields in the cavity operating at three different values of Re ($= 1, 10, 100$), with $Pr = 0.7$, $Gr = 10^5$ and $Da = 10^{-3}$ are shown in Fig. in 10. It is observed that natural convection is dominant at low value of Re with $Pr = 0.7$ and two elliptic shape eyes are formed inside the cavity are shown in Fig. As Re increases to 10, due to increase of inertia effect, the forced convection gradually becomes dominant over natural convection. The right circulation cells are gradually deformed due to motion of upper lid and left circulation cell still denotes the dominant effect of natural convection. As Re increases to 10^2 , the secondary circulation almost disappears and primary circulation occupies most of the cavity due to enhanced forced convection. The isotherms are also found to be non-symmetric showing forced convection dominant heat transfer.

As Re increases to 10, for low value of $Re = 1$, the secondary circulation cells appear near the bottom wall. Both primary and secondary circulation cells are found to be symmetric for $Re = 1$. The isotherms also tend to compress towards left showing convection dominant effect. The dominance of forced convection leads to greater degree of thermal mixing near the upper portion of right half of the cavity. As Re increases to 10^2 , the primary circulation cell gets deformed towards the middle portion of the left wall and the secondary circulation cell near the bottom portion of the right wall tends to grow.

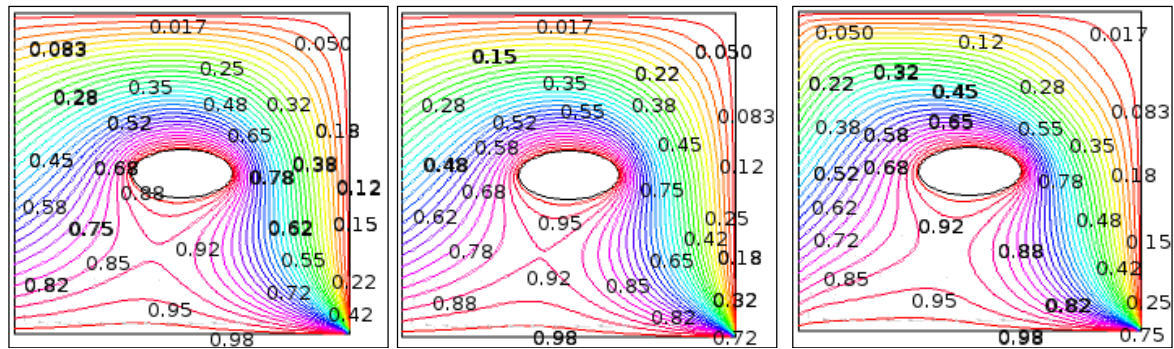


Case I

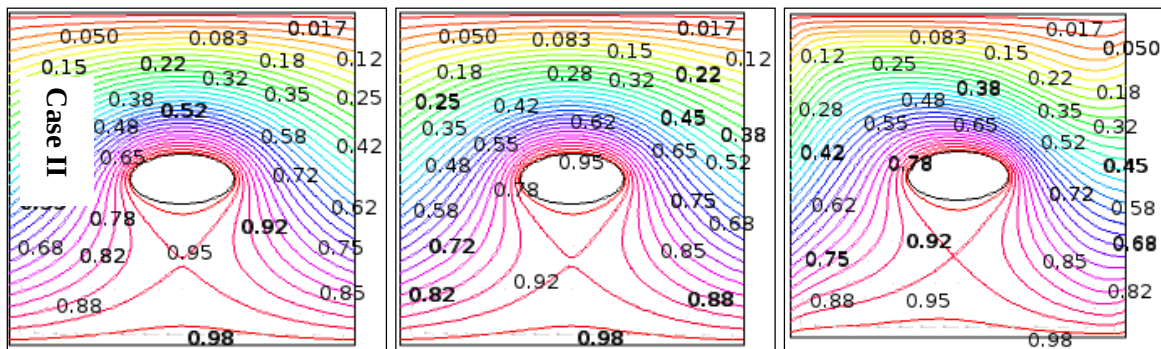


Case II

Fig 10: Streamlines and isotherms for linearly heated left wall and cold right wall with $Pr = 0.7$, $Gr = 10^4$ and $Da = 10^{-3}$ for: (a) $Re = 1$; (b) $Re = 10$; and (c) $Re = 100$

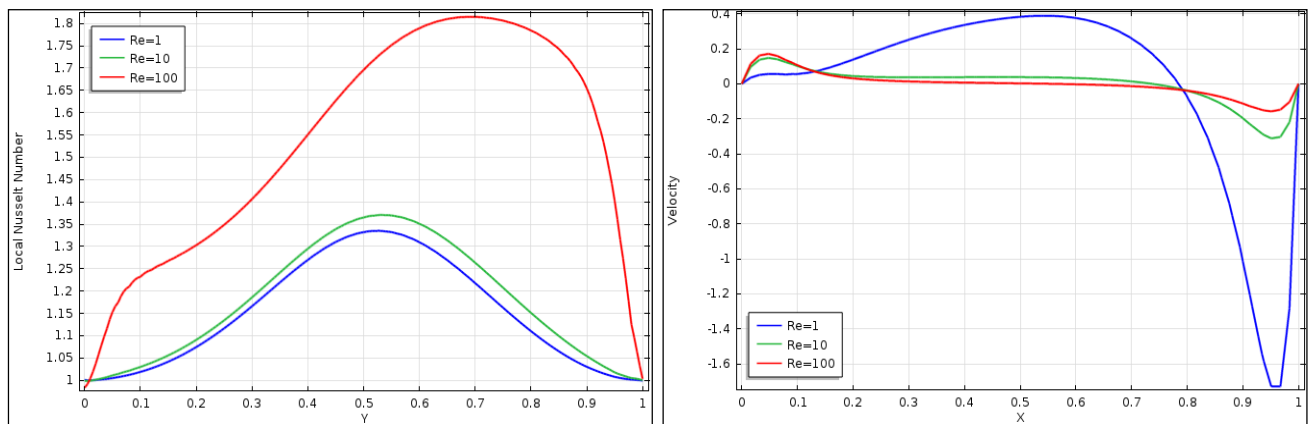


Case I



Case II

Fig 11: Isotherms for linearly heated left wall and cold right wall with $Pr = 0.7$, $Gr = 10^4$ and $Da = 10^{-3}$ for: (a) $Re = 1$; (b) $Re = 10$; and (c) $Re = 100$



Case I

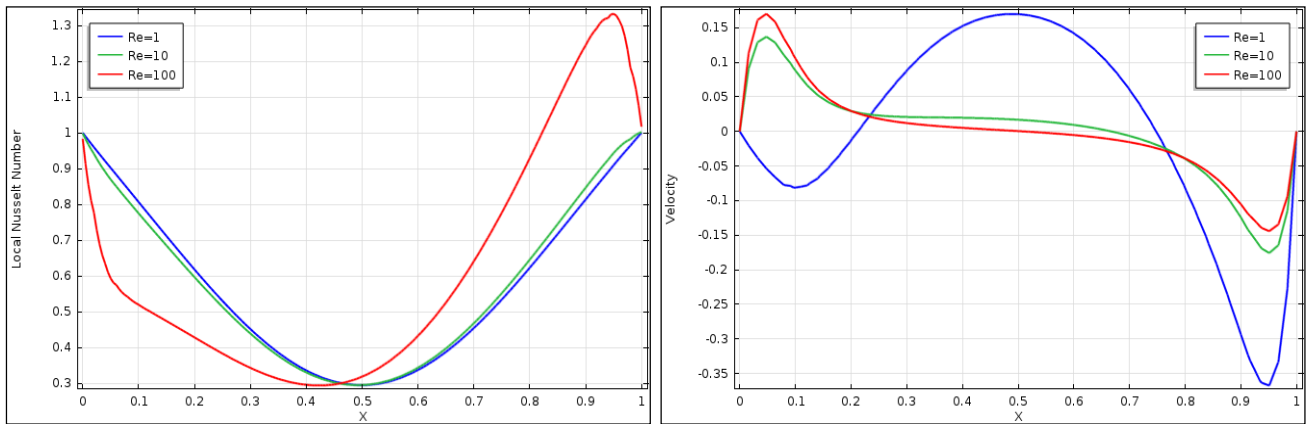
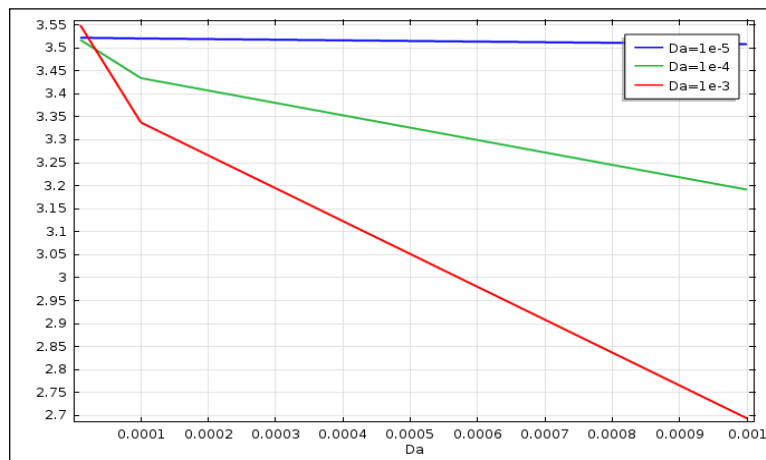


Fig 12: Variation of Local Nusselt number and dimensionless temperature along the right wall of the square cavity with Reynolds number

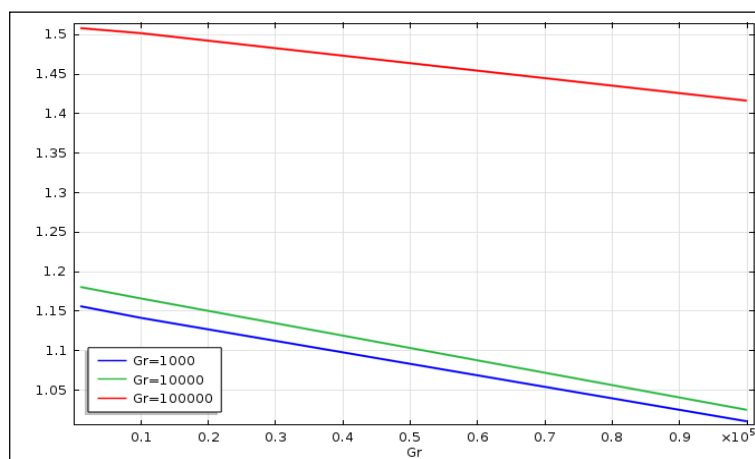
The local Nusselt number (Case I) along the bottom wall for different Reynolds number with $Gr = 10^5$, $Da = 10^{-3}$ and $Pr = 0.7$ of the cavity are shown in Fig. 12. Maximum and minimum shape curves are obtained here. For lower Reynolds number the value of local Nusselt number smaller changed but higher Reynolds number has larger change. The local Nusselt number (Case II) along the bottom wall for different Reynolds number. It can be seen from the Fig. that the minimum value of local Nusselt number decreasing with increasing the Reynolds number.

Variations of the vertical velocity component along the bottom wall (Both cases) for different Reynolds number with $Gr = 10^5$, $Da = 10^{-3}$ and $Pr = 0.7$ of the cavity are shown in Fig. 12. From the Fig. it can be observed that the maximum value of velocity increases with increasing the Reynolds number. For lower Reynolds number the value of velocity has larger change and higher Reynolds number has smaller changed.

In Fig. 13 present the average Nusselt number for different Darcy and different Grashof numbers respectively. It is observed that (Case I) the average Nusselt number decreases with increasing Darcy number. Also it can be seen (Case II) the average Nusselt number increases with increasing Grashof number. Validation of the code was done by comparing streamlines and isotherms without elliptic obstacle and with elliptic obstacle shown in Fig. 14(a) and 14(b). As seen from these Figures increases the rate of heat transfer. Therefore at high Reynolds numbers, a relatively stronger magnetic field to increase the rate of heat transfer.



Case I



Case II

Case I

	$Re = 1$	$Re = 10$	$Re = 10^2$
$Da = 10^{-5}$	3.52195	3.51743	3.54934
$Da = 10^{-4}$	3.52035	3.43408	3.33755
$Da = 10^{-3}$	3.50818	3.19154	2.69411

Case II

	$Gr = 10^3$	$Gr = 10^4$	$Gr = 10^5$
$Da = 10^{-5}$	1.15392	1.16117	1.2442
$Da = 10^{-4}$	1.13287	1.12903	1.09465
$Da = 10^{-3}$	1.000829	1.00199	1.24463

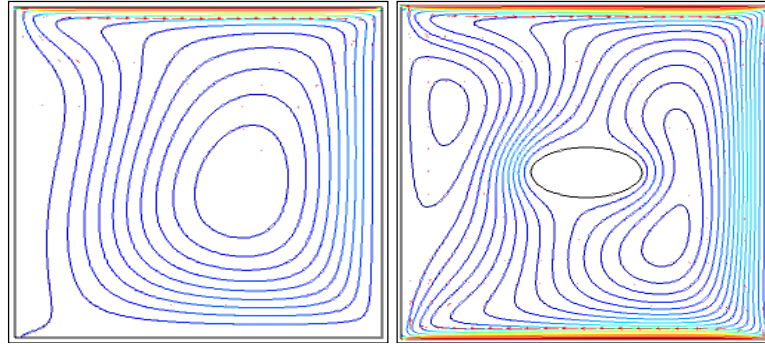
Streamlines

Fig 13: Variation of the average Nusselt number along the right wall of the square cavity with different Darcy numbers and Reynolds numbers (Case I), Grashof numbers and Darcy numbers (Case II).

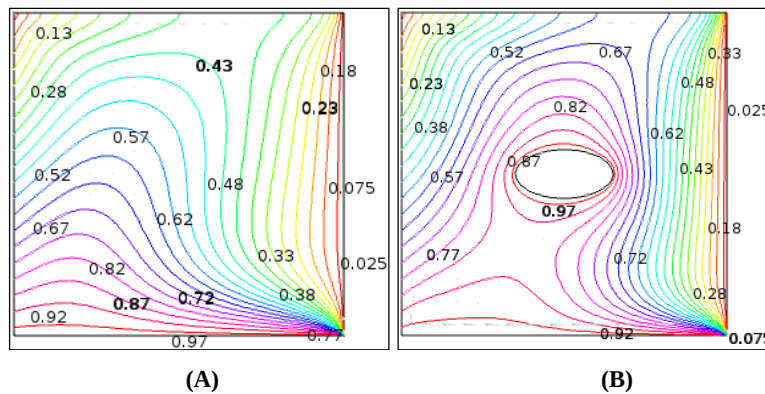
Isotherms

Fig 14: Streamline and isotherms for different Grashof number Present study (a) without obstacle (b) with elliptic obstacle

Nusselt number Table						
Gr	Without obstacle		With obstacle		% increase in heat transfer	
	$Re = 100$	$Re = 1$	$Re = 100$	$Re = 1$	$Re = 100$	$Re = 1$
1000	3.0133	2.9969	3.84999	3.86637	27	29
10000	2.99448	3.77787	3.85849	4.8715	28	28
100000	3.35433	8.11676	4.53785	9.41524	35	16

6. Conclusions

A numerical work has been done to simulate mixed convection in a lid-driven square cavity filled with porous medium with internal heated elliptic block with an electric conductive fluid with Prandtl number 0.7. Subsequently, a parametric study was performed and the effect of the Grashof number, Darcy number and Reynolds number on the fluid flow and heat transfer were investigated. The effect of varying the mentioned parameters on the distribution of streamlines, isotherms, average Nusselt number at the hot wall. For all cases considered, two counter rotating eddies were formed inside the cavity regardless of the Grashof numbers, Darcy numbers and Reynolds numbers. A heat generation parameter has a significant effect on streamlines during the mixed convection dominated regions. At the same time, it has a significant effect on thermal fields at the three convection regimes. The large values of Reynolds number lead to increase the lid-driven effect whereas the small values of Reynolds number lead to increase effect of presence of the heat source on the flow and heat characteristics. In addition, it was also observed that the isotherms are in general symmetric at smaller Pr irrespective of Da and Re at $Gr = 10^5$ for linearly heated left wall and cold right wall. Interesting results are obtained at higher Gr and Da with various regimes of Re . the isotherms are almost symmetric at small Re with higher Gr ($Gr =$

10^5) and $Da = 10^{-3}$ and mixed convection is found to be dominant whereas, forced convection leads to compressed isotherms near the left and bottom walls at higher Re for linearly heated side walls.

7. Acknowledgements

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8. References

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