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Research Paper

A Study on Flow & Temperature Field for Variation of Governing Parameters in an Open Cavity with Prismatic Body

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ABSTRACT: *A computational work has been carried out in the present study on flow and temperature field for variation of governing parameters in an open cavity with prismatic body by utilizing Finite element method. Fluid flow and heat transfer characterization is elaborated in detail by solving leading differential equations over wide ranges of forced convection parameter namely, Reynolds number, $50 \leq Re \leq 500$; magnetic force parameter Ha and mixed convection parameter Ri in the range of $0 \leq Ha \leq 150$ & $0.1 \leq Ri \leq 10$ respectively. Various results for flow pattern and temperature profile in the domain are exposed in terms of the streamlines, isotherms and also heat transfer rate at hot and cold wall in terms of the average Nusselt number is presented for considered parameters. The results indicate that the average Nusselt number at both the heated and cold surface as well as flow and temperature distribution of the fluid inside the cavity is strongly dependent on the configuration of the system under imposed boundary conditions. The average Nusselt number increases significantly with the mounting values of Re.*
KEYWORDS: *Flow and temperature field, Governing parameter, Open cavity, Prismatic body.*

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I. INTRODUCTION

Fluid flow pattern and temperature distribution for variation of Governing parameter in a vented cavity with obstacle has received great attention from researchers due to its many engineering and industrial applications. This type of problem is frequently experienced particularly in cooling of electronic devices, thermal insulation, ventilation of buildings, air conditioning, heat exchangers and chemical processing equipment. A good number of researchers are actively engaged in investigation on flow and temperature field for the effect of pertinent parameter in an open cavity with various types of obstacle.

An analysis of mixed convection in a differentially heated square cavity with moving lids was carried out by Abraham and Varghese [1]. Waheed [2] analyzed mixed convective heat transfer in rectangular enclosures driven by a continuously moving horizontal plate. Numerical study of natural convection in square cavity with inner bodies using finite element method was presented by Pinto et al. [3]. Mixed convection flow in a trapezoidal cavity with non-uniform temperature has been performed by Ibrahim and Hirpho [4], they pointed out that Hartman number has negative impact while Richardson number has a positive effect on the heat transfer rate. Elsherbiny et al. [5] analyzed heat transfer in inclined air rectangular cavities with two localized heat sources. Sererir et al. [6] studied optimal conditions of natural and mixed convection in a vented rectangular cavity with a sinusoidal heated wall along with a heated solid block. Combined convection in rectangular cavity has been carried out by Guo and Sharif [7] at various aspect ratios with moving isothermal sidewalls and constant flux heat source on the bottom wall. Natural convection with MHD in the localized heat sources of an inclined trapezoidal cavity filled with nano-fluid has been analyzed by Mansour et al. [8]. Timuralp and Altac [9] conducted fluid flow and heat transfer in a channel with an open cavity heated from bottom side. Shirvan et al. [10] investigated magnetic field effect on combined free and forced convection heat transfer in a ventilated square cavity.

Free convection flow analysis has been analyzed in a trapezoidal enclosure with a rectangular heated body in presence of external oriented magnetic field by Akter and Parvin [11]. Chattopadhyay et al. [12] studied on mixed convection in a double lid-driven sinusoidally heated porous cavity. Flow and heat transfer characteristics of an open cubic cavity with different inclinations was performed by Saxena and Singh [13]. Free convection heat transfer in a closed cavity with hot and cold tubes has been investigated by Zheng et al. [14]. Saieed et al. [15] carried out heat transfer enhancement in lid driven cavity. Zhu et al. [16] analyzed vortex dynamics and flow patterns in a two-dimensional oscillatory lid-driven rectangular cavity. The effect of prandtl number on natural convection in triangular enclosures with localized heating from below has been presented by Koca et al. [17]. Holtzman and Hill [18] focused on laminar natural convection in isosceles triangular enclosures heated from below and symmetrically cooled from above.

From the above writing review it is followed that open cavity having an inner prismatic block have not been deliberated yet. The purpose of the present work is to analyze the flow and thermal characteristics in an open cavity in presence of prism shaped block for the variation of some chosen parameters.

II. CONFIGURATION DETAILS

The studied geometry of the current work is shown in Figure-1 that consists of an open rectangular cavity with internal isolated prism shaped solid block. The bottom surface of the cavity maintains sinusoidal temperature, while the top wall is set as thermal insulation and rest two sides of the cavity were kept at a low temperature T_c. A uniform magnetic field of strength B_0 is imposed to the flat direction of right wall. All solid boundaries are considered as rigid no-slip walls that is velocity components u and v are set to be zero. The flow enters the cavity at a uniform velocity, u_i and the ambient temperature, T_i whereas the outgoing flow is assumed to have zero diffusion flux for all dependent variables.

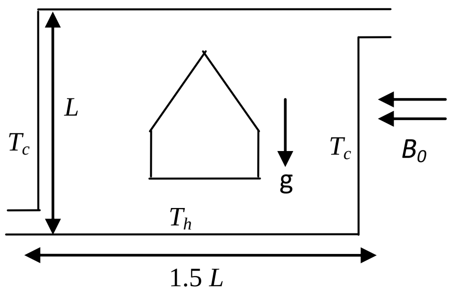


Figure 1: Studied Geometry

III. PROBLEM FORMULATION

The present problem consisting of the conservation of mass, momentum, and energy that can be expressed as non-dimensional form given below considering the working fluid as steady state, two-dimensional, laminar, compressible, Newtonian and electrically conducting with constant thermo-physical properties.

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

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

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

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

Here, $\mathrm{Re}=\frac{u_i L}{\upsilon}$ is the Reynolds number, $\mathrm{Pr}=\frac{\upsilon}{\alpha}$ is the Prandtl number, $Ha=B_0 L\sqrt{\frac{\sigma}{\mu}}$ is the Hartmann number, $Ri=\frac{Gr}{\mathrm{Re}^2}$ is the Richardson number.

The concerning boundary conditions in the dimensionless form are given below:

$U = 1, V = 0, \theta = 0$ at the inlet

$P = 0$ at the outlet: convective boundary condition (CBC)

$\theta = 1$ at the bottom heated surface

$\theta = 0$ at the left and right walls

$\frac{\partial \theta}{\partial N} = 0$ at the top wall

U = 0, V = 0 at all solid boundaries

$U = 0, V = 0, \frac{\partial \theta}{\partial N} = 0$ at the surface of the centered block

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

The average Nusselt number Nu at the hot wall is given by

$$Nu_{av} = -\int_0^1 \left(\frac{\partial \theta}{\partial Y} \right) dX$$

and the bulk average fluid temperature in the enclosure is defined as

$\theta_{av} = \int \theta \frac{d\overline{V}}{\overline{V}}$, where $\overline{V}$ is the cavity volume.

IV. MESH GENERATION AND VALIDATION

To ensure the accuracy of the results with the grid variations for the present model, various grid sized elements are taken into consideration. Heat transfer rate that is shown as average Nusselt number at the bottom heated surface of the cavity are examined for these selected elements. Small variations are detected among the results for different numbers of elements which is tracked by Table 1and selected the grid consisting of 33582 elements for calculating the average Nusselt number at left cooled, right cooled and bottom heated surface of the cavity.

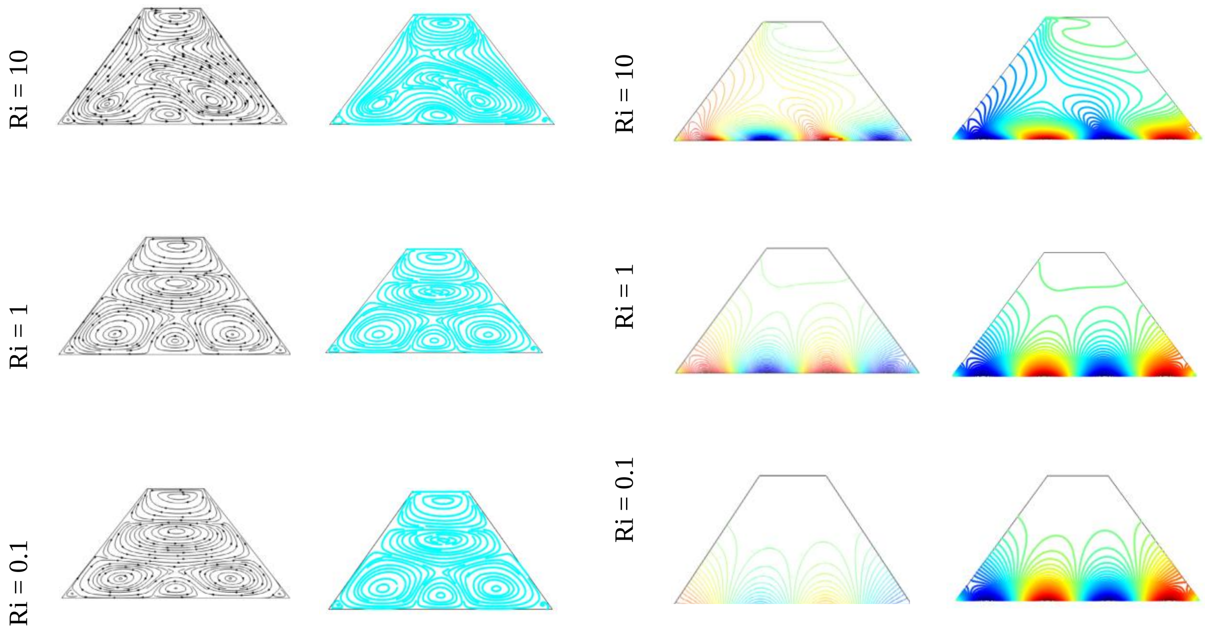
Table-1: Average Nusselt number at hot wall while Pr = 0.71, Re= 200, Ha = 50 and Ri = 10

No. of elements	2224	3266	5402	13212	33582	45790
Nu_{av}	7.6346	7.6847	7.7854	8.0223	8.2600	8.2583
Deviation	-	0.0501	0.1007	0.2369	0.2377	0.0017

To validate the code of the current work a computation is performed for comparison with the work of Ibrahim and Hirpho [4]; finite element analysis of mixed convection flow in a trapezoidal cavity with non-uniform temperature. Figure 2 reveals the relationships between the works of Ibrahim and Hirpho [4] and present with tremendous consistency in velocity and temperature profiles that are displayed as streamlines and isotherms.

V. RESULT AND DISCUSSION

The influence of Reynolds number, Hartman number and Richardson number in the domain of a ventilated rectangular cavity with prism shaped obstacle has been analyzed in this research. The streamlines, isotherms and average Nusselt number are exposed to explain the flow and thermal field structure of the problem for wide range of Reynolds number, Hartman number and Richardson number while Prandtl number is chosen as 0.71 which corresponds to air.



(b) Isotherms [4]   Isotherms (present)

(a) Streamlines [4]   Streamlines (present)

Figure 2: Comparison of (a) streamlines and (b) isotherms while Gr = 10^4, Pr = 6.2, m = 0.25, Ha=50 and Re = 100

The overall streamlines and isotherms arrangements are depicted in Figure 3 for the different values of Reynolds number varied from 50 to 500. From Figure 3(a) it is observed that at low Reynolds number Re = 50, a large rotating cell is revealed in the right side of the centered body. When Re = 100, right sided vortex enlarges and another small eddy is formed above the inlet of the enclosure. For the upper value of Re = 200 large vortex disappears and consequently left eddy reduces to a big vortex. It is noticed that at the highest value of Re = 500 there are three vortices form inside the cavity in where centered block is confined by the rotating cells.

In Figure 3(b), the resembling isotherms for different Reynolds number are displayed where it is noticed that for the lower value of Re = 50 the heat lines are non-linear that stretched the whole cavity and a small change is followed for higher values of Re = 100. When Re = 200, the isotherms are densed at the right wall of the cavity and heat lines encompass the obstacle. A drastically change is observed for Re = 500, boundary layer thickness increases and temperature profiles are very non-uniform.

The impact of Hartman number on overall streamlines and isotherms are exhibited in Figure 4 that varies from 0 to 150. It is followed from Figure 4(a) that at low Hartman number Ha = 0 the streamlines occupy the whole domain with a vortex located at the left side and above the inlet of the cavity. When Ha= 50, left sided vortex vanishes and streamlines capture that region with almost vertical and horizontal flow lines. There is no significant change is noticed for the rising values of Ha = 100 and Ha = 150.

The corresponding isotherms for different Hartman numbers are depicted in Figure 4(b). For the lower value Ha = 0 the temperature distribution seems to be non-linear that scattered the whole domain and a minor variation is observed for the rest three upper values of Hartman number Ha = 50, 100, 150. The isotherms are packed near the inlet and create a boundary layer in the vicinity of the right sided surface of the cavity.

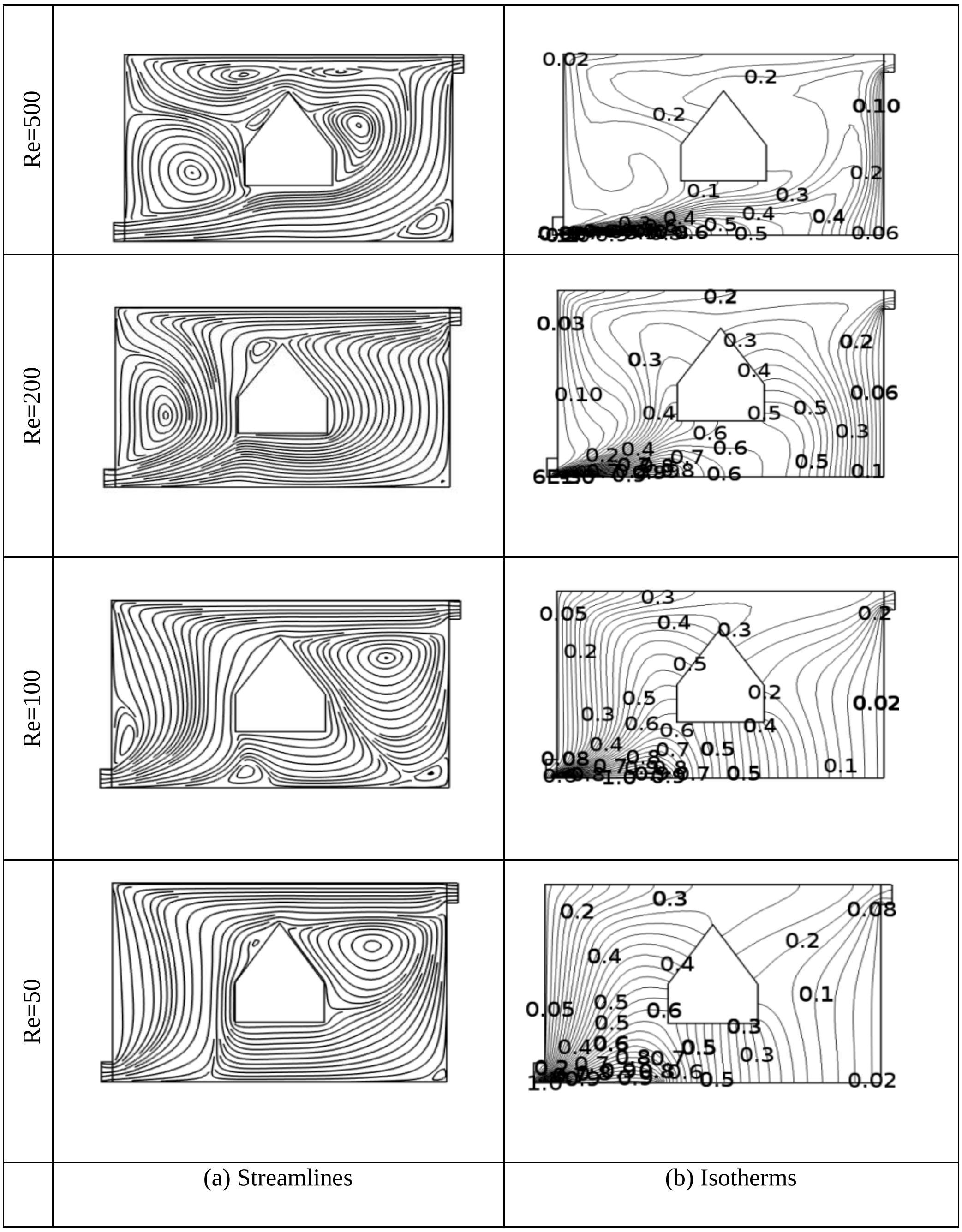


Figure 3: (a) streamlines and (b) isotherms for variation of Re while Pr = 0.71, Ha = 50 and Ri = 10

For the variation of Richardson number Ri (0.1, 0.5, 1, 10), velocity profile and temperature profile are displayed in terms of streamlines and isotherms in Figure 5. From Figure 5(a) it is seen that when Ri = 0.1 the streamlines absorb the whole cavity with very well distribution and a small eddy is marked at the left corner above the inlet. It is also noticed that for the rest three larger values of Ri, the flow patterns are almost identical.

The isotherms for different mixed convection parameter Ri are depicted in Figure 5(b). With a minor change heat lines are found elongated from both the left and right cold wall towards inner prism shaped adiabatic body for all of the considered values of Ri. Moreover heat lines are packed at heated bottom surface near the entry port and a thermal boundary layer is developed in the neighborhood of the right cooled wall of the cavity.

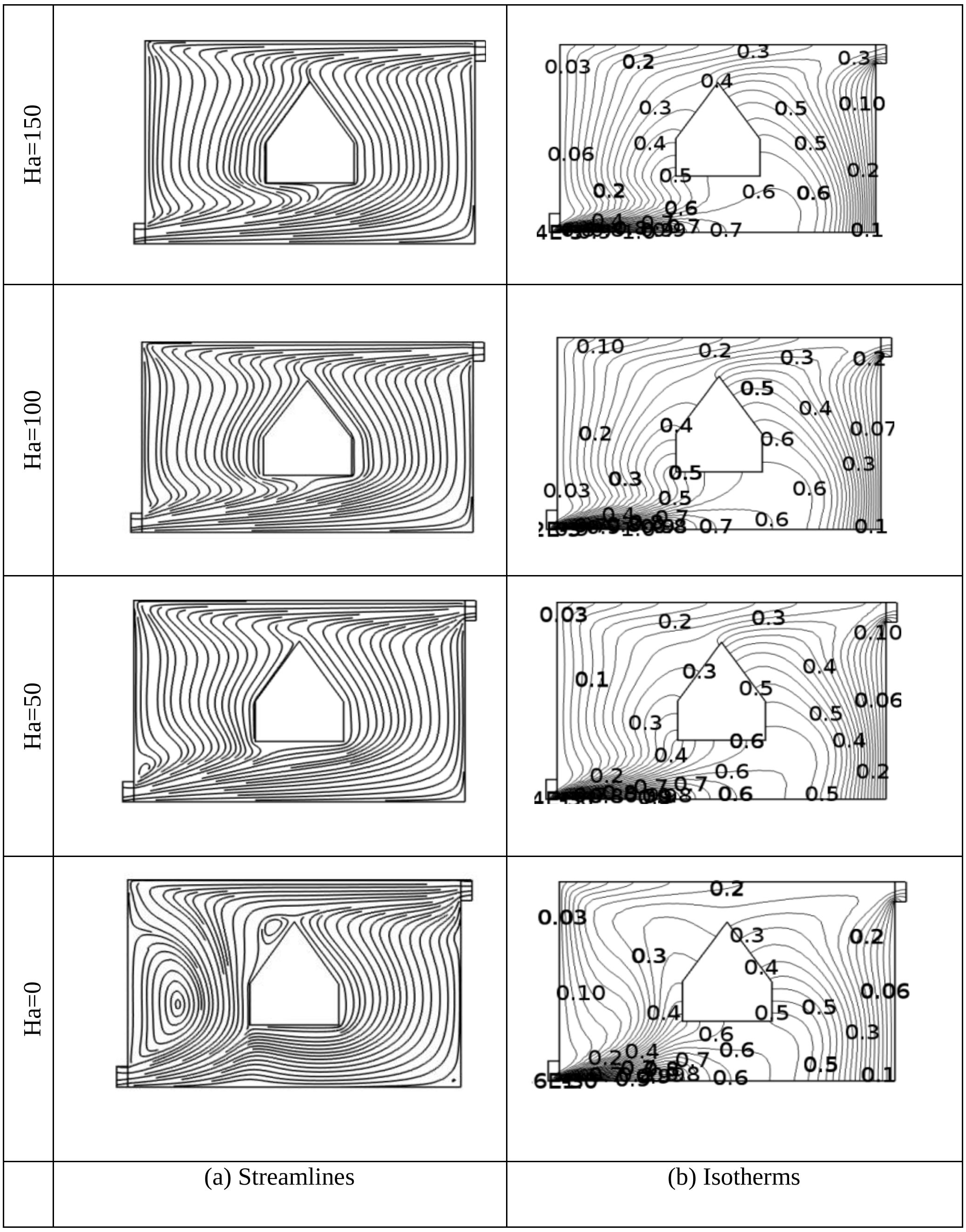


Figure 4: (a) streamlines and (b) isotherms for variation of Ha while Pr = 0.71, Re = 200 and Ri = 10

The rate of heat transfer at bottom heated surface, left and right cooled walls for different values of Reynolds number are shown in Table 2. From table it can be found that heat transfer rate at hot and right cool walls increases with the increasing values of Re, while it decreases at left cool wall.

Average Nusselt number at bottom hot wall, left and right cold walls for various values of Hartman number are tabulated in Table 3 and it is followed that heat transfer rate at hot and cold walls enhances for the upper values of Ha except for Ha = 50.

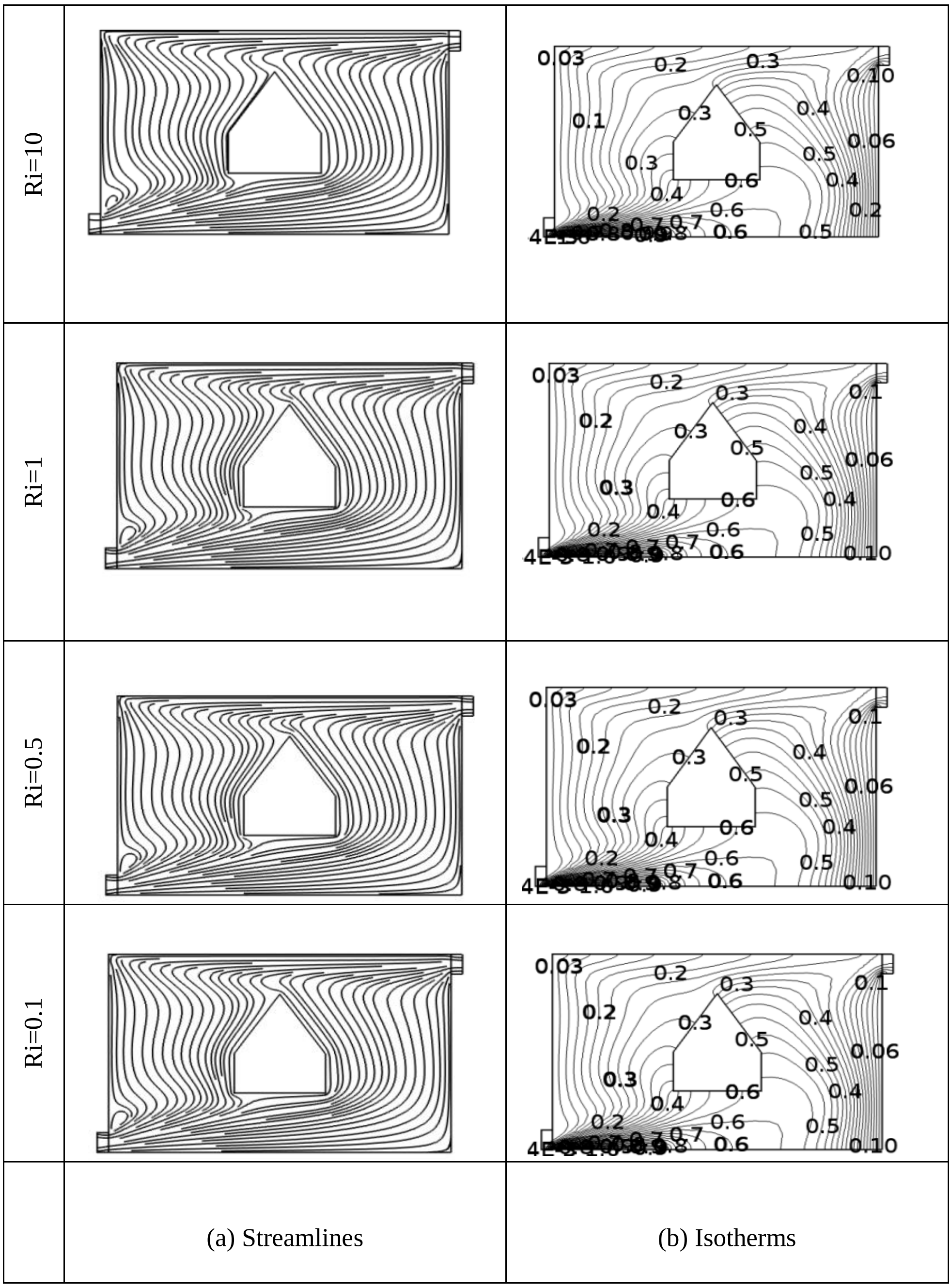


Figure 5: (a) streamlines and (b) isotherms for variation of Ri while Pr = 0.71, Ha = 50 and Re = 200

Table-2: Average Nusselt number for variation of Re while Pr = 0.71, Ha= 50 and Ri = 10

Re	$(Nu_{av})_{hot,}$	$(Nu_{av})_{R\ cool,}$	$(Nu_{av})_{L\ cool}$
50	5.6751	3.2822	1.1801
100	6.8154	3.3388	0.8815
200	8.2600	3.3722	0.5055
500	10.3900	3.4063	0.2408

Table-3: Average Nusselt number for variation of Ha while Pr = 0.71, Re= 200 and Ri = 10

Ha	$(Nu_{av})_{hot,}$	$(Nu_{av})_{R\ cool,}$	$(Nu_{av})_{L\ cool}$
0	8.5881	3.2634	0.5407
50	8.2600	3.3722	0.5055
100	8.7114	3.5196	0.5762
150	9.1313	3.6644	0.6517

Lastly, Table 4 gives heat transfer rate at heated and two cooled walls for distinct Richardson number. From here it can be noted that heat transfer rate at all of considered surfaces are negligible for the different values of Ri.

Table-4: Average Nusselt number for variation of Ri while Pr = 0.71, Re= 200 and Ha = 50

Ri	$(Nu_{av})_{hot,}$	$(Nu_{av})_{R\ cool,}$	$(Nu_{av})_{L\ cool}$
0.1	8.2601	3.3737	0.50553
0.5	8.2601	3.3737	0.50553
1	8.2601	3.3736	0.50553
10	8.2600	3.3722	0.50555

VI. CONCLUSION

The effects of Reynolds number, Hartman number and Richardson number on flow and temperature field have been studied in an open rectangular cavity with internal thermally isolated prismatic body. The findings in this work can be summarized as below:

-There is a considerable enhancement in heat transfer at hot wall due to larger values of Re and small variation is found at two cold side walls.

-As Ha increases average Nusselt number raises for both hot and cold wall but an exception is observed at Ha = 50.

-No significant change is noticed in heat transfer rate for all considering values of Ri.

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