Journal of Applied Science and Engineering, Vol. 20, No. 2, pp. 173–182 (2017) *DOI: 10.6180/jase.2017.20.2.05*

Effect of Prandtl Number on Natural Convection in a Rectangular Enclosure with Discrete Heaters

N. Nithyadevi, V. Divya* and M. Rajarathinam

Department of Mathematics, Bharathiar University, Coimbatore - 641 046, India

Abstract

The present numerical study investigates the laminar natural convection heat transfer and the effect of Prandtl number in a two dimensional rectangular enclosure with discrete heaters. Four different cases are considered based on the number of discrete heaters which is maintained at isothermal condition T_h ($T_h > T_c$). The right vertical wall is maintained at cold temperature T_c and the remaining all other walls are thermally insulated. The above schematic setup can be modeled into mathematical form and the governing non-dimensional equations are solved using Finite Volume Method with power-law scheme. SIMPLE algorithm is employed for the pressure-velocity coupled momentum equations. Numerical simulations are carried out to find the effect of different Prandtl numbers (0.054, 0.71, 1.4 and 7.0), internal heat generation parameter Ri ranging from 0.1 to 10.0 and the distribution of discrete heaters. Results are given in the form of streamlines, isotherms, the velocity profiles and average Nusselt number. It is found that the maximum heat transfer rate is achieved for the distribution of discrete heater and also for increasing values of Prandtl number.

Key Words: Discrete Heater, Natural Convection, Prandtl Number, SIMPLE Algorithm

1. Introduction

A rapid increase in the growth of electronic equipments needs an effective cooling to achieve the optimal performance. The mechanism of cooling in closed configurations by natural convection is preferable due to its low maintenance cost and high efficiency. Hence natural convection in rectangular enclosures has received much attention among researchers to consider different combinations of aspect ratios, location of discrete heaters on the walls and boundary conditions. The numerical and experimental investigation on natural convective heat transfer in a vertical rectangular enclosure with four discrete heaters for different aspect ratios was carried out [1]. The natural convective heat transfer of micropolar fluids in an enclosure with an isoflux discrete heater on one of the sidewalls was investigated and reported that the heat transfer rate was an increasing function of Rayleigh number [2]. Later, the numerical study of the same process in a square enclosure with a heat generating fluid was done [3]. The study of discrete isothermal bottom heating was investigated [4] to report the effect of aspect ratio of the horizontal rectangular enclosure. The nine different location of heater on the vertical rectangular enclosure which was partially active to describe the natural convection has been examined [5] and they found that the middle-middle active location of the heater was effective in transferring the heat. In a similar study [6], the maximum heat transfer rate was achieved when the thermally active hot and cold location was placed at the middle of the vertical walls.

The numerical investigation of the natural convection respectively in the rectangular and square enclosure with partial isothermal heating and cooling on the vertical walls was studied by the authors in [7] and [8]. They reported that the rate of heat transfer increased with increasing Rayleigh number and it reached the maximum value when AR = 1.0. Laminar natural convection of Newtonian fluids in rectangular enclosures with differentially heated side walls along with the influence of

*Corresponding author. E-mail: divya7291@gmail.com

boundary conditions has been analyzed [9]. A special attempt was made by placing two discrete sources at the bottom of the enclosure with the isothermally cooled vertical walls and adiabatic top wall [10]. The computation of the melting of PCM with natural convection inside a rectangular enclosure heated by discrete protruding heat sources was done numerically [11]. Both the numerical and analytical solution for natural convection in a cavity with volumetric heat generation was obtained in [12]. Very recently, the laminar steady mixed convection flow from a discrete heat source in a lid-driven square enclosure filled with water-based micropolar nanofluids was numerically examined [13]. They found that the average Nusselt number along the heat source decreased for increase in the heat source length while it was increased for the solid volume fraction. A recent numerical study of 3D natural convection heat transfer and fluid flow in a cubical cavity induced by a thermally active heater was described [14].

Some of the authors had developed the studies on natural convection problem to find the influence of various Prandtl number in a rectangular enclosure [15]. The steady laminar convection flow of two different nature namely uniform and non-uniform heating for the case of $0.7 \leq \Pr \leq 10$ was analyzed [16]. It is proposed that the average Nusselt number increased for both uniform and non-uniform heating. A numerical study was made to find the effects of Reynolds and Prandtl number on a mixed convection having heat generating solid circular block [17]. They concluded that the effect of Prandtl number was negligible on the streamlines and considerable influence was found on the isotherms for different values of Richardson number by analyzing the MHD mixed convection having heat conducting circular block placed in the middle of the cavity. A similar problem has been done for the case of rotating cylinder [18].

The authors [19] used various Prandtl numbers (0.1, 1, 10) in a rectangular enclosure with a fin attached to the vertical cold wall. An investigation on natural convection horizontal enclosure having a fixed adiabatic square body at the center and the fluid with three different Prandtl numbers 0.01, 0.7, 7.0 was made in [20]. They observed that the oscillatory thermal behavior was obtained for the case of Prandtl numbers 0.01 and 7.0 with high Rayleigh number. Recently the problem of natural convection for the purpose of finding the effects of Prandtl numbers on entropy generation, heat transfer and fluid flow in the presence of magnetic field was studied [21]. From the above reviews of literature, they had handled different Prandtl numbers and concluded that for increasing value of Prandtl number, the heat transfer rate increases.

Several theories have been explained by the researchers in order to explain the theory of discrete heaters but they have considered only the particular number of discrete heaters and no comparison was found between them. Hence the authors take this opportunity to investigate the effect of Prandtl number on a natural convection problem in a rectangular enclosure with discrete heaters on one of the vertical walls of the enclosure. Simulations are carried out for the parameters like internal heat generation Ri, Prandtl number Pr and distribution of discrete heaters.

2. Mathematical Formulation

Figure 1 (a–d) shows the schematic of the problem which considers a two dimensional rectangular enclosure of height H (2.0) and length L (1.0) filled with different fluids like liquid metal, air and water. Case-1 describes the enclosure with a single discrete heater placed at the middle of the left vertical wall Figure 1(a). In Case-2, the heater is divided into two discrete heaters Figure 1(b). In the same way, the heater is divided into three and four discrete heaters for the Case of 3 and 4 which is shown in Figure 1(c–d). The length of the heater is equal to the half of the height of the enclosure in each case. The thick solid lines represent the discrete heater and its position, which is maintained at constant temperature T_h while the opposite wall is maintained at cold temperature T_c and the remaining walls of the enclosure are thermally insulated. Four different Prandtl numbers 0.054, 0.71, 1.4, 7.0 are used in the present study. Thermophysical properties of the fluid are assumed to be constant, except the variation of density in the buoyancy term due to Boussinesq approximation. The governing dimensional equations of the conservation of mass, momentum and energy can be converted into the following dimensionless form:

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

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

$$\frac{\partial V}{\partial \tau}+U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\Pr\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)+Ra_E\Pr\theta \tag{3}$$

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

using the non-dimensional parameters,

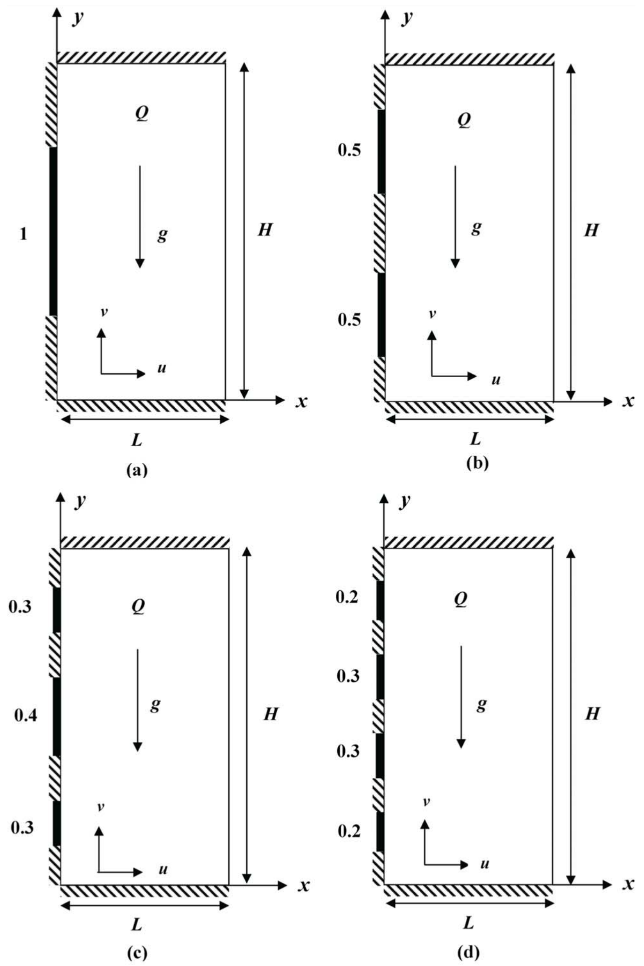

Figure 1. Schematic of the cavity with discrete heaters (a) Case-1, (b) Case-2, (c) Case-3, (d) Case-4.

$$(X,Y)=\frac{(x,y)}{L},\ (U,V)=\frac{(u,v)}{\alpha/L},\ \tau=\frac{\alpha T}{L^2},\ P=\frac{pL^2}{\rho\alpha^2},$$
$$\theta=\frac{T-T_c}{T_h-T_c},\ \Pr=\frac{\nu}{\alpha},\ Ra_I=g\beta\frac{QL^5}{\nu\alpha k}, \tag{5}$$
$$Ra_E=g\beta\frac{T_h-T_c}{\nu\alpha}L^3,\ Ri=\frac{Ra_I}{Ra_E}$$

where Ri is the internal heat generation parameter. The governing equations (1)–(4) are subjected to the following boundary conditions:

$$\begin{aligned}
&\tau=0: \quad U=V=0,\ \ \theta=\theta_c,\ \ 0\le X\le L,\ \ 0\le Y\le H,\\
&\tau>0: \quad U=V=0,\ \ \theta=\theta_c,\ \ X=L,\ \ 0\le Y\le H,\\
&\qquad U=V=0,\ \ \frac{\partial\theta}{\partial X}=0,\ \ Y=0,H,\ \ 0\le X\le L,\\
&\qquad U=V=0,\ \ \theta=\theta_h,\ \text{on heaters};\\
&\qquad \frac{\partial\theta}{\partial X}=0\ \ \text{elsewhere},\ X=0,\ 0\le Y\le H
\end{aligned} \tag{6}$$

The heat transfer coefficients in terms of local Nusselt number (Nu) is defined by

$$Nu=-\frac{\partial\theta}{\partial n} \tag{7}$$

where 'n' denotes the normal direction in a plane. The average Nusselt number at the hot and cold vertical walls are computed as

$$\overline{Nu_h}=\frac{1}{L}\int_L Nu_h dy,\quad \overline{Nu_c}=\frac{1}{A}\int_0^A Nu_c dy \tag{8}$$

The average Nusselt number is taken as the sum of all average Nusselt number of each heater.

3. Numerical Method

The governing equations (1)–(4) associated with the boundary conditions are numerically solved by Finite Volume method with Power Law scheme using uniform staggered grid arrangement. A Semi Implicit Method for Pressure-Linked Equation (SIMPLE) algorithm [22] is used to solve the pressure-velocity coupled equations. The obtained discretized equations are solved by TDMA (Tri-Diagonal Matrix Algorithm) line-by-line method. At each step, the solution is obtained by iterative technique and the steady state solution is established by the following convergence criteria:

$$\frac{\sum\left|\phi_{i,j}^{\,n+1}-\phi_{i,j}^{\,n}\right|}{\sum\left|\phi_{i,j}^{\,n+1}\right|}<\varepsilon \tag{9}$$

In the above equation ε is chosen of order 10^{-5}, n is any time level and ϕ represents θ, U or V. Grid independent test has been performed for various grid sizes from 22 × 42 to 102 × 202 for Case 1. Table 1 shows that, increasing the grid size from 82 × 162 to 92 × 182, there is no significant change in the average Nusselt number. Since large grid size leads to time consumption, so we have fixed the grid size as 82 × 162. In order to check the computing FORTRAN code of the present problem validation is carried out to ensure that the code is free from error. Benchmark solutions of Davis [23] and Mobedi et al. [24] have been taken into account. Table 2 shows the comparison of the present and previous results so that we can found the results are in good agreement. Based on the confidence of the results, the code is extended to this present study. In this study, all of the computations were performed at Ra_E = 10^5 on the CPU of Intel(R)

Table 1. Grid independent study for Case-1 at $Ri = 0.1$, Pr = 0.71, $Ra_E = 10^5$

Grid size	22 × 42	32 × 62	42 × 82	52 × 102	62 × 122	72 × 142	82 × 162	92 × 182
$\overline{Nu}$	7.24778	6.78569	6.69594	6.65568	6.67628	6.66057	6.59845	6.59635

Table 2. Comparison of the present results with the benchmark solutions of Davis (1983) and Mobedi et al. (2010)

Ra	Average Nusselt number		
	Davis	Mobedi et al.	Present study
10^3	1.118	1.113	1.115
10^4	2.243	2.237	2.247
10^5	4.519	4.510	4.547
10^6	8.800	8.833	8.978

Core(TM) (model name: i3-2120, CPU frequency: 3.30 GHz, memory size: 4 GB) at Bharathiar University. Typical CPU time is around 14,400 s for a productive run.

4. Results and Discussion

Natural convection in a rectangular enclosure with the presence of discrete heaters on the vertical wall is numerically simulated for several combinations of parameters like internal heat generation, Prandtl number and the distribution of discrete heaters. The qualitative study of results is expressed in the form of streamlines and isotherms while the quantitative study is represented through Nusselt number and velocity profiles.

Figure 2 and Figure 3 illustrates the flow field and thermal field for Case-3 with different *Ri*, Pr = 0.054 and 1.4 respectively. The streamlines demonstrates that the flow is rotating in anticlockwise direction indicated by the negative values. At the earlier stage, the center portion of the cavity is stagnant but for increasing values of *Ri* leads to the formation of bean shaped cell in the stagnant region. The large value of *Ri* provides the fluid flow to gain more energy from the internal Rayleigh number and also this enormous energy paved a way to form more cells inside the cavity. This is due to the minimum density hotter fluid moves very faster by forming a secondary cell on the top left corner of the cavity which tries to dominate the primary cell. It is ensured from values of

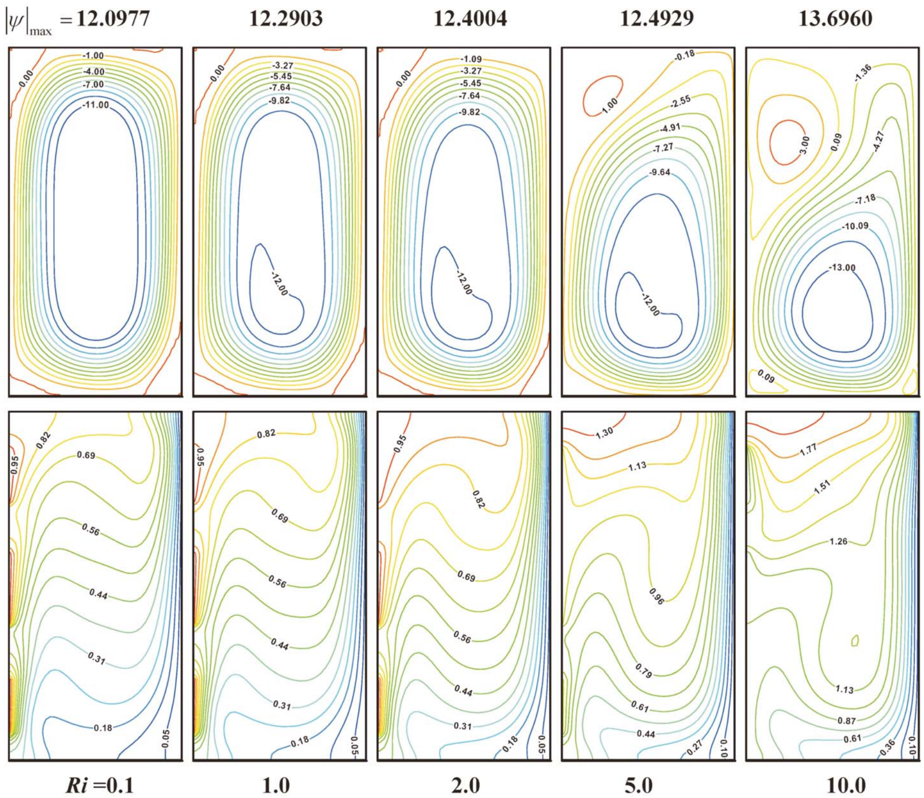

Figure 2. Streamlines and isotherms for different *Ri* with Pr = 0.054 for Case-3.

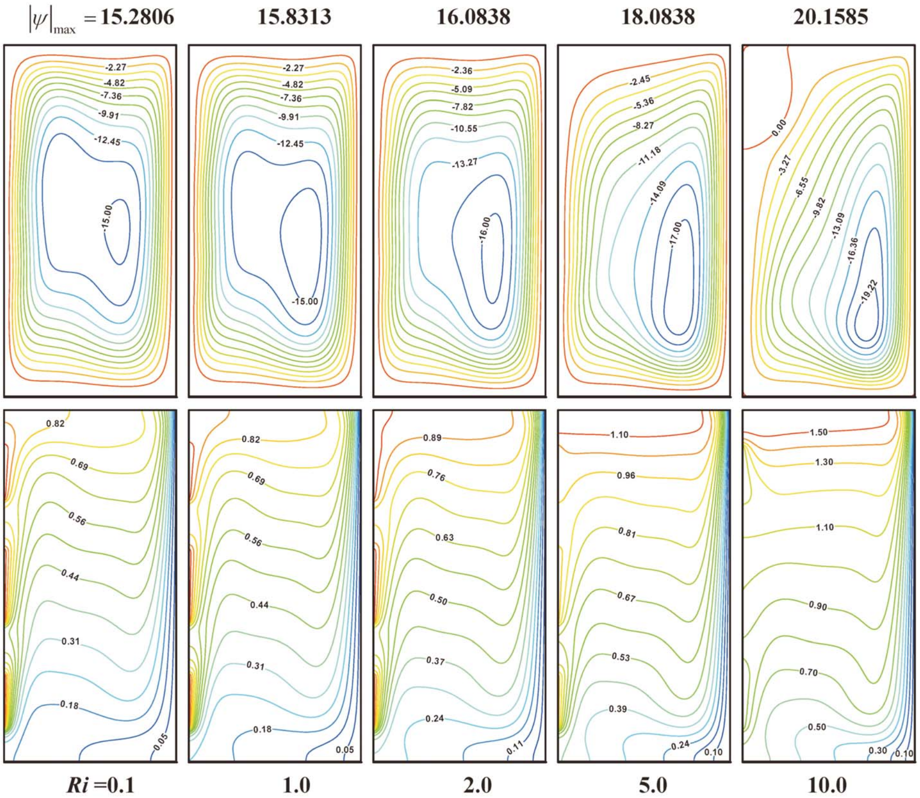

Figure 3. Streamlines and isotherms for different *Ri* with Pr = 1.4 for Case-3.

streamfunction that, increasing the value of *Ri* leads to increase the fluid motion.

Initially, the isotherms are crowded around the position of the discrete heaters and also we can see the natural convective flow of heat towards the cold wall from the discrete heaters. The internal temperature is maintained for low values of *Ri* indicated by the labels of the isotherms. For descending values of *Ri*, a significant change on the heat flow is observed since the internal Rayleigh number is high which leads to increase in the average fluid temperature dominating the temperature of the discrete heaters, hence the effect of distribution of heat from the discrete heater is entirely reduced. Due to buoyancy mechanism, the fluid with high temperature of the fluid is seen at the top portion of the cavity.

Now the discussion is extended to the natural convection of discrete heaters with the effect of Prandtl number on Case-3. When the value of Pr is fixed at 0.054, the new secondary cell is observed along with the elongated circular motion of the fluid flow but at Pr = 1.4, we can observe a significant change in the flow pattern of the streamlines. The domination of secondary cell is reduced because of the increase in the momentum diffusivity. The flow field exhibits a widened anticlockwise circular motion with high intensity along the active vertical walls which can be noticed from the streamfunction values. The isotherms in Figure 2 show high internal temperature throughout the whole cavity and it bends towards the bottom and finally it reaches the cold wall but in Figure 3 the temperature of the flow is maintained because of rapid convection of heat transfer inside the cavity. Hence we can conclude that increasing the Prandtl number of the fluid results in increasing both the fluid flow and heat transfer.

Figure 4 and Figure 5 represent the streamlines and isotherms for different Prandtl numbers on Case-1, Case-2 and Case-4. The streamlines in all the cases shows that the pattern of the fluid flow is significant mainly in the core of the cavity and the fluid circulation increases with increasing value of Prandtl number which we can identify through the values of streamfunction. When the Prandtl number increases, the fluid becomes heavier due to increase in momentum diffusivity hence there is formation of hydrodynamic boundary layer along the vertical walls. At low Prandtl numbers, the isotherms are more curved whereas for high values of Pr, it exhibits only the vertical flat isotherms. The stratification of isothermal layers is found along both the cold wall and

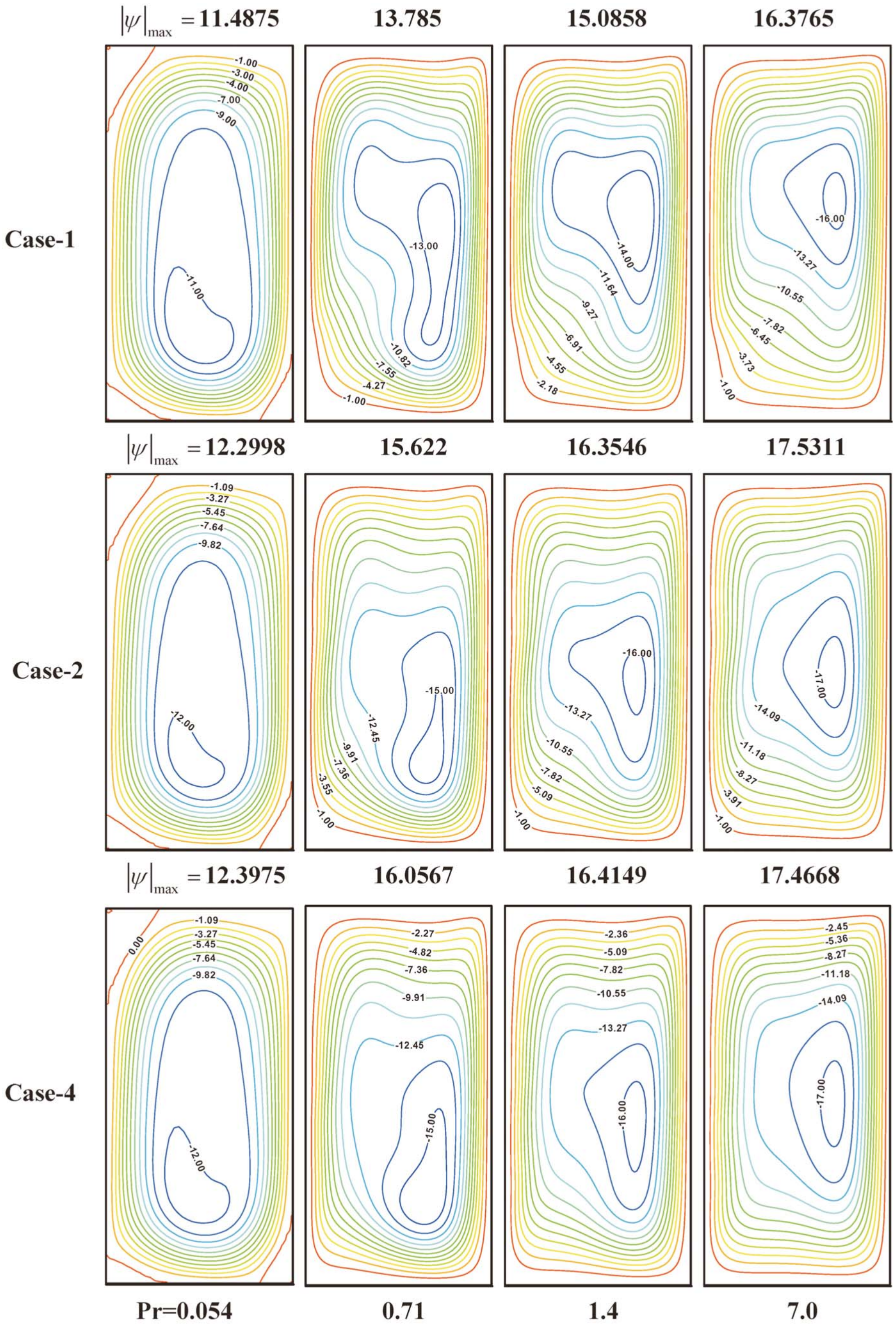

Figure 4. Streamlines for different Prandtl number with $Ri = 2.0$ for Case-1, 2 & 4.

around the discrete heater by forming a thermal boundary layer due to increase in Prandtl number hence resulting in increased rate of heat transfer. When the discrete heater is split into small heaters, the circulation of the heat flow increases and more heat is transferred from the heater to the cold wall.

Figure 6 illustrates the variation of average Nusselt number with internal heat generation parameter *Ri* for four different cases. The average Nusselt number for different Prandtl number in Case-1 and Case-2 is displayed through Figure 6(a). It is seen that, increasing values of *Ri* leads to decrease the average Nusselt number since the internal temperature is higher than the temperature on the discrete heaters, hence there is no transfer of heat from the discrete heater. At Pr = 0.054, the minimum rate of heat transfer is obtained because of low thermal diffusivity. The heat transfer rate increases with increase in Pr. Also, the negative value of Nusselt number indicates

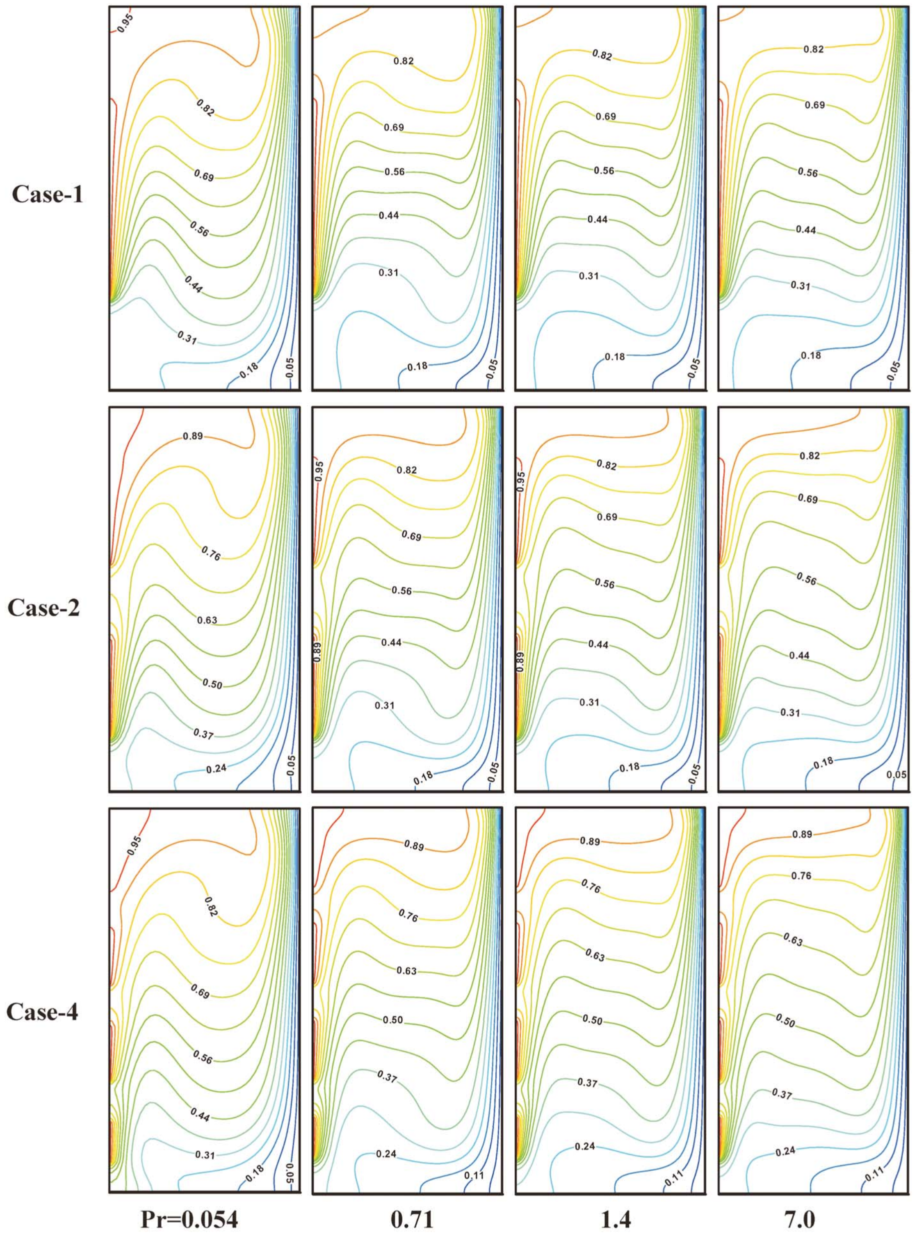

Figure 5. Isotherms for different Prandtl number with $Ri = 2.0$ for Case-1, 2 & 4.

the domination of internal temperature so that the direction of heat is towards the discrete heaters instead of away from it. Similarly, Figure 6(b) showed the quantitative values of average Nusselt number for Case-3 and Case-4. On comparing the values of Nusselt number in all the four cases, Case-4 has maximum heat transfer rate because of the distribution of the discrete heaters.

Figure 7 evaluates the value of average Nusselt number for Case-1 and Case-3 against various Prandtl number is plotted for different Ri. A slow and steady increase in average Nusselt number is observed with increase in Pr and the heat transfer rate is decreased with increase in the heat generation parameter Ri. Table 3 displays the average Nusselt number for different Prandtl numbers and four Cases have been considered with fixed $Ri = 2.0$. The rate of heat transfer increases with increase in both the Prandtl numbers and partitioning of the discrete heater.

The midheight horizontal and vertical velocity profile for different Prandtl number in Case-1 and Case-3 can be seen in Figures 8(a) and 8(b) respectively. Initially the velocity values are influenced towards the cold wall and the colder fluid due to high density moves only

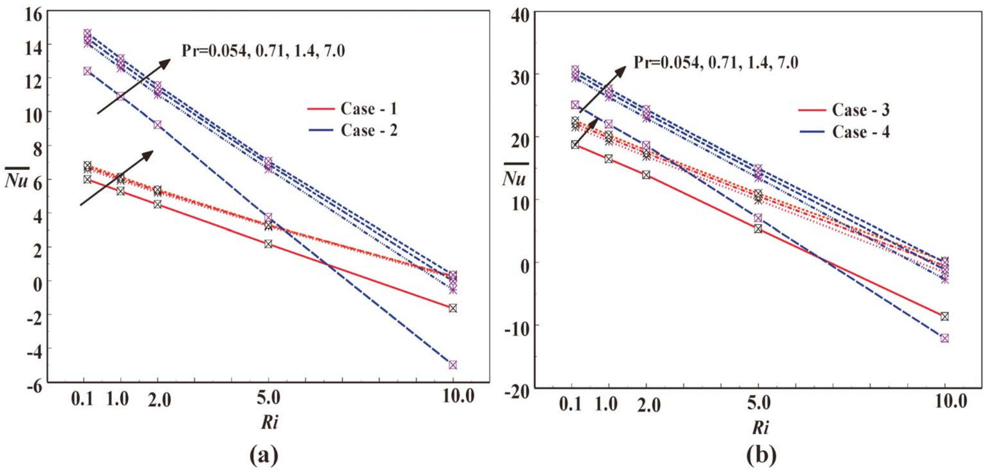

Figure 6. Variation of average Nusselt number with *Ri* (a) Case-1 & Case-2, (b) Case-3 & Case-4.

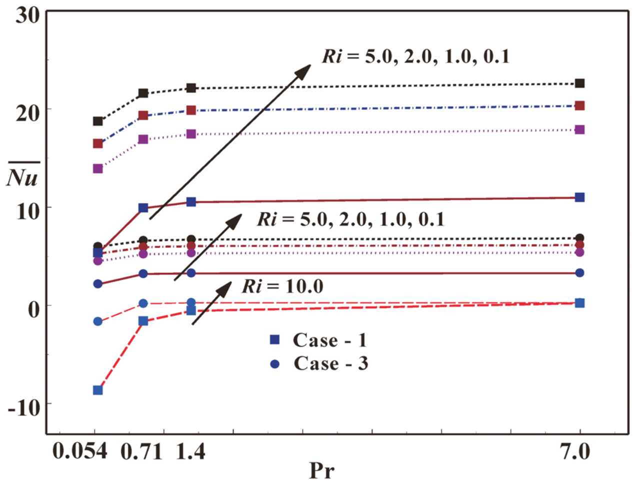

Figure 7. Variation of average Nusselt number with Pr for Case-1 & Case-3.

with minimum velocity. Also, the liquid metal with Pr = 0.054 attains the most negative and positive values when compared with the other three Prandtl numbers. In the middle of the enclosure, there are some oscillations and finally they have attained the positive values since the hotter fluid moves with greater velocities. On comparing Case-1 and Case-3, the largest velocity values are attained for Case-3. The vertical velocity reveals that the maximum velocity values are attained for all Pr's except Pr = 0.054. Except liquid metal, the speed of movement of all other fluids is similar which is ensured from the streamlines of the all the cases.

5. Conclusions

Numerical simulations have been performed for unsteady natural convection in a rectangular enclosure with the presence of discrete heater. The effect of discrete heater, Prandtl number and internal heat generation parameter have been analyzed through streamlines, isotherms, velocity profiles and heat transfer rates. From the above study, we can conclude the following remarks:

- Among the four different cases, Case-4 yields the better heat transfer rate than the other three cases. Hence, the distribution of discrete heater produces the maximum heat transfer rate and the fluid motion than other discrete heater.
- Increasing the values of internal heat generation parameter *Ri*, internal temperature exceeds the temperature of the discrete heater, hence the effect of discrete heater is entirely reduced.
- Increasing values of Prandtl number leads to increase the fluid flow and heat transfer thereby forming hydrodynamic and thermal boundary layers respectively.
- The heat transfer rate increases with increase in the distribution of discrete heater and Prandtl number but decreases with increase in the internal heat generation parameter.

Table 3. Comparison of four different Cases for four different Prandtl numbers and *Ri* = 2.0

Case	Pr = 0.054	0.71	1.4	7.0
1	4.5073	5.2097	5.3096	5.3824
2	9.2239	11.0120	11.3096	11.5522
3	13.9046	16.8711	17.4097	17.8594
4	18.6318	22.9930	23.7367	24.3224

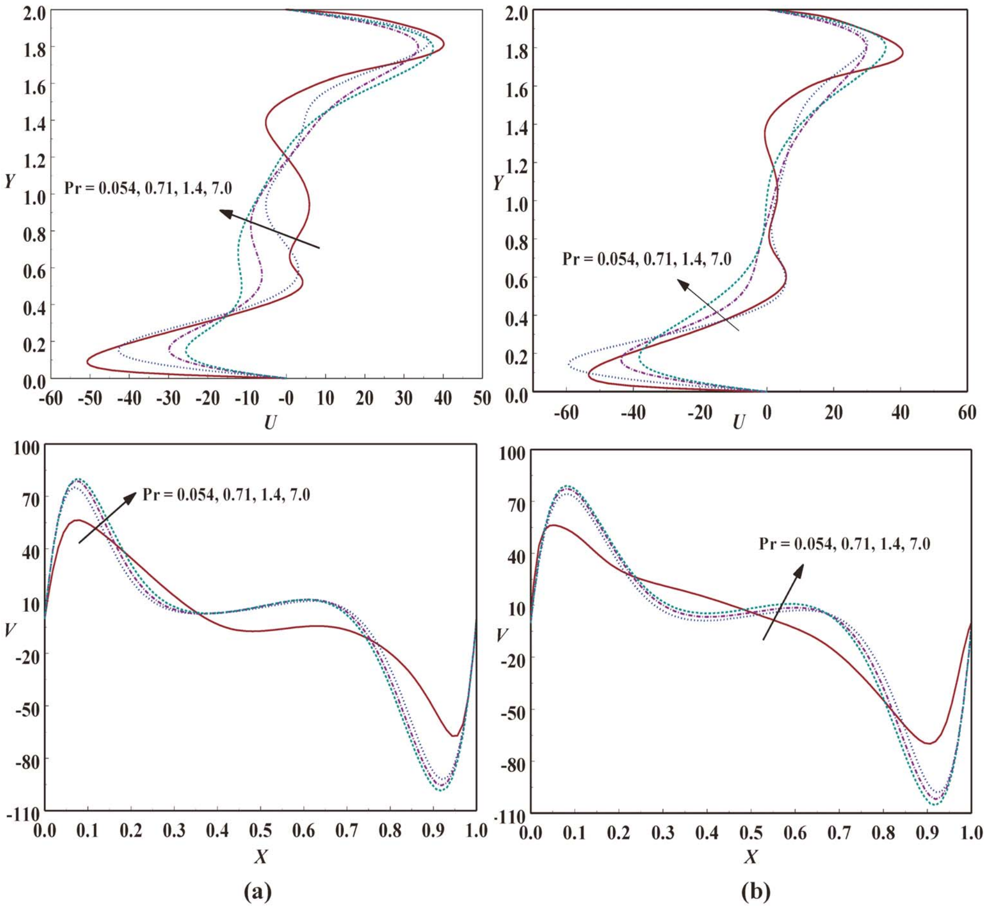

Figure 8. Midheight horizontal and vertical velocity against *Y*-axis and *X*-axis with *Ri* = 2.0 (a) Case-1, (b) Case-3.

References

[1] Ho, C. J. and Chang, J. Y., "A Study of Natural Convection Heat Transfer in a Vertical Rectangular Enclosure with Two-dimensional Discrete Heating: Effect of Aspect Ratio," *Int. J. Heat Mass Transfer*, Vol. 37, No. 6, pp. 917–925 (1994). doi: 10.1016/0017-9310(94)90217-8

[2] Aydin, O. and Pop, I., "Natural Convection from a Discrete Heater in Enclosures Filled with a Micropolar Fluid," *Int. J. of Engg. Science*, Vol. 43, No. 19–20, pp. 1409–1418 (2005). doi: 10.1016/j.ijengsci.2005.06.005

[3] Sivasankaran, S., "Buoyant Convection in a Cavity with Discrete Heat Sources and Internal Heat Generation," *Int. J. of Appl. Math. and Mech.*, Vol. 2, No. 2, pp. 63–74 (2006).

[4] Saha, G., Saha, S., Islam, M. Q. and Akhanda, M. A. R., "Natural Convection in Enclosure with Discrete Isothermal Heating from Below," *J. Naval Architecture and Marine Engg.*, Vol. 4, No. 1, pp. 1–13 (2007). doi: 10.3329/jname.v4i1.912

[5] Nithyadevi, N., Kandaswamy, P. and Lee, J., "Natural Convection in a Rectangular Cavity with Partially Active Side Walls," *Int. J. of Heat and Mass Transfer*, Vol. 50, No. 23–24, pp. 4688–4697 (2007). doi: 10.1016/j.ijheatmasstransfer.2007.03.050

[6] Kandaswamy, P., Nithyadevi, N. and Ng, C. O., "Natural Convection in Enclosures with Partially Thermally Active Side Walls Containing Internal Heat Sources," *Phys. of Fluids*, Vol. 20, pp. 097104–097123 (2008). doi: 10.1063/1.2981834

[7] Alam, P., Kumar, A., Kapoor, S. and Ansari, S. R., "Numerical Investigation of Natural Convection in a Rectangular Enclosure Due to Partial Heating and Cooling at Vertical Walls," *Commun. in Nonlinear Sci. Numer. Simulat.*, Vol. 17, No. 6, pp. 2403–2414 (2012). doi: 10.1016/j.cnsns.2011.09.004

[8] Falahat, A., "Effect of Aspect Ratio on Laminar Natural Convection in Partially Heated Enclosure," *Univer-*

sal J. Mech. Engg., Vol. 2, No. 1, pp. 28–33 (2014). doi: 10.13189/ujme.2014.020104

[9] Turan, O., Poole, R. J. and Chakraborty, N., "Influence of Boundary Conditions on Laminar Natural Convection in Rectangular Enclosures with Differentially Heated Side Walls," *Int. J. Heat and Fluid Flow*, Vol. 33, No. 1, pp. 131–146 (2012). doi: 10.1016/j.ijheatfluidflow.2011.10.009

[10] Zaman, F. S., Turja, T. S. and Molla, Md. M., "Buoyancy Driven Natural Convection in an Enclosure with Two Discrete Heating from Below," *Procedia Engg*., Vol. 56, pp. 104–111 (2013). doi: 10.1016/j.proeng.2013.03.095

[11] Qarnia, H. E., Draoui, A. and Lakhal, E. K., "Computation of Melting with Natural Convection Inside a Rectangular Enclosure Heated by Discrete Protruding Heat Sources," *Appl. Math. Modelling*, Vol. 37, No. 6, pp. 3968–3981 (2013). doi: 10.1016/j.apm.2012.08.021

[12] An, C., Vieira, C. B. and Su, J., "Integral Transform Solution of Natural Convection in a Square Cavity with Volumetric Heat Generation," *Braz. J. Chem. Eng*., Vol. 30, No. 4, pp. 883–896 (2013). doi: 10.1590/S0104-66322013000400020

[13] Ahmed, S. E., Mansour, M. A., Hussein, A. K. and Sivasankaran, S., "Mixed Convection from a Discrete Heat Source in Enclosures with Two Adjacent Moving Walls and Filled with Micropolar Nanofluids," *Int. J. Engg. Science and Technology*, Vol. 19, No. 1, pp. 364–376 (2016). doi: 10.1016/j.jestch.2015.08.005

[14] Purusothaman, A., Oztop, H. F., Nithyadevi, N. and Abu-Hamdeh, N. H., "3D Natural Convection in a Cubical Cavity with a Thermally Active Heater under the Presence of an External Magnetic Field," *Comp. & Fluids*, Vol. 128, pp. 30–40 (2016). doi: 10.1016/j.compfluid.2016.01.011

[15] Graebel, W. P., "The Influence of Prandtl Number on Free Convection in a Rectangular Cavity," *Int. J. Heat and Mass Transfer*, Vol. 24, No. 1, pp. 125–131 (1981). doi: 10.1016/0017-9310(81)90100-9

[16] Basak, T., Roy, S. and Balakrishnan, A. R., "Effects of Thermal Boundary Conditions on Natural Convection Flows within a Square Cavity," *Int. J. Heat and Mass Transfer*, Vol. 49, No. 23–24, pp. 4525–4535 (2006). doi: 10.1016/j.ijheatmasstransfer.2006.05.015

[17] Rahman, M. M., Parvin, S., Rahim, N. A., Islam, M. R., Saidur, R. and Hasanuzzaman, M., "Effects of Reynolds and Prandtl Number on Mixed Convection in a Ventilated Cavity with a Heat-generating Solid Circular Block," *Appl. Math. Modelling*, Vol. 36, No. 5, pp. 2056–2066 (2012). doi: 10.1016/j.apm.2011.08.014

[18] Urquiza, G., Castro, L., Garcia, J., Basurtio, M. and Bogarin, E., "Numerical Simulation on Mixed Convection in a Rotating Cylindrical Cavity: Influence of Prandtl Number," *Advances in Mech. Engg*., Vol. 2013, pp. 9507665–9507673 (2013). doi: 10.1155/2013/950765

[19] Jani, S., Mahmoodi, M. and Amini, M., "Natural Convection at Different Prandtl Numbers in Rectangular Cavities with a Fin on the Cold Wall," *J. Energy: Engg. and Management*, Vol. 2, No. 4, pp. 58–69 (2012).

[20] Lee, J. R. and Park, I. S., "Numerical Analysis for Prandtl Number Depending on Natural Convection in an Enclosure Having an Vertical Thermal Gradient with a Square Insulator Inside," *Nuclear Engg. Tech*., Vol. 44, No. 3, pp. 283–296 (2012). doi: 10.5516/NET.02.2011.027

[21] Bouabid, M., Hidouri, N., Magherbi, M., Eljery, A. and Brahim, A. B., "Irreversibility Investigation on MHD Natural Convection in a Square Cavity for Different Prandtl Numbers," *World Science Research Journals*, Vol. 2, No. 4, pp. 60–75 (2014).

[22] Patankar, S. V., *Numerical Heat Transfer and Fluid Flow*, Hemisphere/McGraw-Hill, Washington (1980).

[23] Davis, D. V., "Natural Convection of Air in a Square Cavity; a Bench Mark Numerical Solution," *Int. J. Numer. Methods in Fluids*, Vol. 3, No. 4, pp. 249–264 (1983). doi: 10.1002/fld.1650030305

[24] Mobedi, M., Ozkol, U. and Sunden, B., "Visualization of Diffusion and Convection Heat Transport in a Square Cavity with Natural Convection," *Int. J. Heat Mass Transfer*, Vol. 53, No. 1–3, pp. 99–109 (2010). doi: 10.1016/j.ijheatmasstransfer.2009.09.048

Manuscript Received: May 24, 2016
Accepted: Jan. 31, 2017