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Numerical Simulation for Conjugate Natural Convection in a Partially Heated Rectangular Porous Cavity

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Abstract: Natural convection heat transfer in a porous rectangular partially active heated wall is numerically investigated using finite element method. Three different cases of heating and cooling zone had been taken in the consideration along the vertical walls while the others are considered to be adiabatic. The governing equations are obtained by the applying of Darcy Model and Boussinesq approximation. Finite element method is used to solve the dimensionless governing equations with the specified boundary conditions. The investigated parameters in the present study are the modified Rayleigh number (10 • Ra • 10^3), aspect ratio (0.5 • A• 2), finite wall thickness (0.02 • D• = 0.5) and the thermal conductivity ratio (0.1• K_r • 10). The results are presented in terms of streamlines, isotherms and Nusselt number. The results indicate that as the aspect ratio, finite wall thickness increase, Nusselt number decrease. Also, as the modified Rayleigh number increases, the Nusselt number will increase. Case 1 and 2 gave approximately the same effects of heat transfer rate while case 3 give lower rate of heat transfer rate.

Key words: Porous media, natural convection, partially heated, conjugate problems, approximately, transfer rate

INTRODUCTION

An equally large body of research has emerged with respect to investigations of convective heat transfer in rectangular porous cavities and vertical surfaces with surface temperatures. The classical Darcy formulation and non-Darcy formulations have been used.

Recently, many investigations undertaken on natural convection in enclosures filled with porous media (Nithiarasu *et al*., 1997; Tanmay *et al*., 2006; Oztop, 2007; Sathiyamoorthy *et al*., 2007). On the other hand, the conjugate heat convection in enclosures filled with porous medium has evoked much interest by many researchers due to its various practical applications such as cold storage installations and high performance insulation for buildings. These areas have been thoroughly reviewed recently by Nield and Bejan (2006). Mbaye *et al*. (1993) had investigated analytical and numerical studies of natural convection in an inclined porous layer boarded by a finite-thickness wall. The horizontal walls were insulated while the vertical sidewalls kept at constant heat flux ($\bar{\bar{q}}$).Wen-Jeng and Hui-Chuan (1994) examined the heat transfer effects on conjugate natural convection in a porous enclosure surrounded by finite walls. The results showed that the overall heat transfer rate from the hot side to cold side of the enclosure decreased due to the heat conduction on the wall. Baytas *et al*. (2001) studied numerically, the effects of conjugate heat convection in a square porous cavity bounded by two horizontal conductive finite thickness walls and two vertical walls at different uniform temperatures. They concluded that for large values of the conductivity ratio, the flow characteristics were significantly influenced due to the strong coupling effects fluid-saturated porous medium and the solid walls. Al-Farhany and Turan (2012) performed a numerical analysis of natural convective heat transfer in an inclined porous rectangular using finite volume method for various inclination angles (0•••85) and aspect ratio (2•A•5). The results show that the average Nusselt and Sherwood numbers decrease when the aspect ratio increases, moreover they decrease when the angle of inclination increases. Saeid (2007) studied numerically the effects of conjugate natural convective heat transfer in a two-dimensional vertical porous enclosure. The horizontal walls were kept insulated while the outer surfaces of the vertical walls with a finite thickness were isothermal at different temperatures. The numerical work covered a wide range of governing parameters. It was found that the average Nusselt numbers increased when the Rayleigh number and the thermal conductivity increased while the Nusselt numbers decreased when the wall thickness

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increased. Al-Amiri *et al*. (2008) studied numerically, the conjugate natural convective heat transfer in a fluid-saturated porous enclosure. The porosity of the porous medium (•) were investigated as well as the effect of Darcy number.

Saleh and Hashim (2012) studied the effects of internal heat generation on the natural convection heat transfer in porous cavity. Rayleigh number, thermal conductivity ratio and the wall thickness were used as governing parameters. Their concluded were the maximum fluid temperature increase when the thickness of a solid wall decrease. Furthermore, it increase when the thermal conductivity ratio increase. Bhuvaneswari *et al*. (2011) investigated numerically the effect of aspect ratio in a porous enclosure with partially active walls. The Brinkman-Forchheimer extended Darcy Model have been used and the governing equations solved by using finite volume method with simple algorithm. Their results shows that the location of heating and cooling area influence significantly in the rate of heat transfer and the flow patterns in the enclosure. Numerical investigation of conjugate natural heat convection in a differentially heated square enclosure done by Roslan *et al*. (2014). A conductive polygon object used for various Rayleigh number. The results display that polygon object impact on the heat transfer rate. Hossain and Wilson (2002) solved using finite difference approach the natural convective fluid flow in a porous rectangular enclosure. All the walls assumed to be non-isothermal walls with effects of the internal heat generation. The researcher studied the effect of porosity, internal heat generation and the rate of heat transfer. Saleh *et al*. (2011) studied the effect of conduction wall from the bottom of a porous square cavity on the rate of heat transfer. COMSOL Multiphysics Software used to solve the non-dimensional governing equations. The results indicate that when the Rayleigh number and thermal conductivity ratio increasing in addition to decreasing the bottom wall thickness, the circulations of fluid flow and the rate of heat transfer rate increasing. An in-house code based on finite volume method with simple/simpler algorithm made by Al-Farhany and Turan (2011a, b) to study the heat convection in a square porous enclosure with finite wall thickness from two sides. A good correlation equation presented to predict average Nusselt number on the interface of left wall in terms of the governing parameters. More works that are complicated done by Al-Farhany and Turan (2011a, b) to investigate the effects of variable porosity. Revnic *et al*. (2009) used finite difference method to solve numerically free convection flow in a bidisperes porous medium in a square cavity. Numerical investigation of natural heat transfer in an inclined porous square enclosure with finite walls thickness from both sides by Ahmed *et al*. (2016). The left sidewall heated partially when the other walls insulated. The results indicate that the average Nusselt number increases when the inclination angle increases (0°• • • 45°). Also when • = 60°, the average Nusselt number decreases slightly while it decreases significantly when • = 90°.

The mean objective of this research is to investigate the partially heated conduction wall on the left sidewall of a porous enclosure in the rate of heat transfer rate. Finite different have been used to solve the dimensionless governing equations using Darcy Model. The considered parameters are modified Rayleigh number (Ra), non-dimensional finite wall thickness (D), Aspect ratios (A) and three different locations of active cooling and hot zones.

MATERIALS AND METHODS

Mathematical formulation: In this study, two-dimension natural convection on porous cavity have been investigated numerically with the effect of the partially heated conduction on vertical wall as shown in Fig. 1. The horizontal walls are adiabatic while three different cases have been investigations with different partially heated walls on the left side while the right walls are kept at constant different partially cold temperature. The Darcy Model has been used to solve the governing equations. The porous medium is assumed to be homogeneous, isotropic and saturated with an incompressible fluid. All the properties have been assumed to be constant except the density. The flow is driven by buoyancy effect due to temperature variations. The density variations are described by the Boussinesq approximation:

$$\rho=\rho\left[1-\beta_T(T-T_0)\right] \tag{1}$$

where, $•_T$ = -1/• (••/•T)$_P$ is the thermal expansion coefficient, the two-dimensional continuity, energy and momentum equations in the x and y directions equations for steady natural convection in porous cavity are:

Continuity equation:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \tag{2}$$

X-momentum equation:

$$U\frac{\partial U}{\partial X}+\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\Pr\left(\frac{\partial^2 U}{\partial x^2}+\frac{\partial^2 U}{\partial Y^2}\right)-\frac{\Pr}{Da}U \tag{3}$$

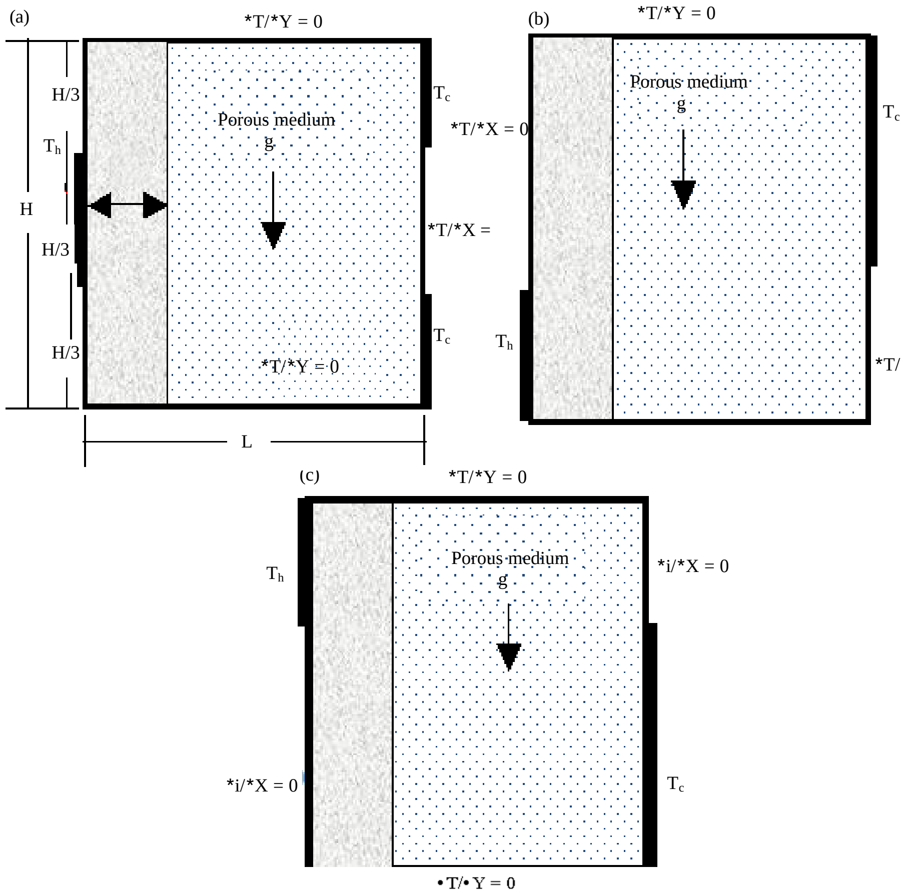


Fig. 1: Schematic diagram of the present problem

Y-momentum equation:

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+Pr\left(\frac{\partial^2 V}{\partial x^2}+\frac{\partial^2 V}{\partial Y^2}\right)-\frac{Pr}{Da}V+Ra.Pr.T \qquad (4)$$

Energy equation:

$$U\cdot\frac{\partial T}{\partial X}+V\cdot\frac{\partial T}{\partial Y}=\frac{\partial^2 T}{\partial X^2}+\frac{\partial^2 T}{\partial Y^2} \qquad (5)$$

and the energy equation at the walls:

$$\frac{\partial^2 T_w}{\partial X^2}+\frac{\partial^2 T_w}{\partial Y^2}=0 \qquad (6)$$

The heat transfer at the walls are defined as in the following:

$$Nu=\frac{1}{A}\int_0^A\left(-\frac{\partial T}{\partial X}\right)\cdot\partial Y \qquad (7)$$

The non-dimensional parameters can be seen by Al-Farhany and Turan (2010), Al-Farhany (2012). Equations 2-7 are solved using non-dimensional initial boundary conditions:

$$\text{at } X=D \quad U=V=0 \quad \frac{\partial T_w}{\partial X}=k_r\frac{\partial T}{\partial X} \qquad (8)$$

$$\text{at } X=1 \quad U=V=0 \quad T_c=0 \qquad (9)$$

$$\text{at } X=0 \quad T_h=1 \qquad (10)$$

$$\text{at } Y=0, A \quad U=V=0 \quad \frac{\partial T}{\partial Y}=0 \qquad (11)$$

Code validation: With the aim of check the validity of this program, the obtained results are compared with the results computed by Saeid (2007) for streamline and isotherm contours at (Ra = 1000, K_r = 1 and D = 0.02). Comparisons give a good agreement between the present results with Saeid results as shown in Fig. 2. In addition, Table 1 shows a great match in Nusselt number between previous results with the present results.

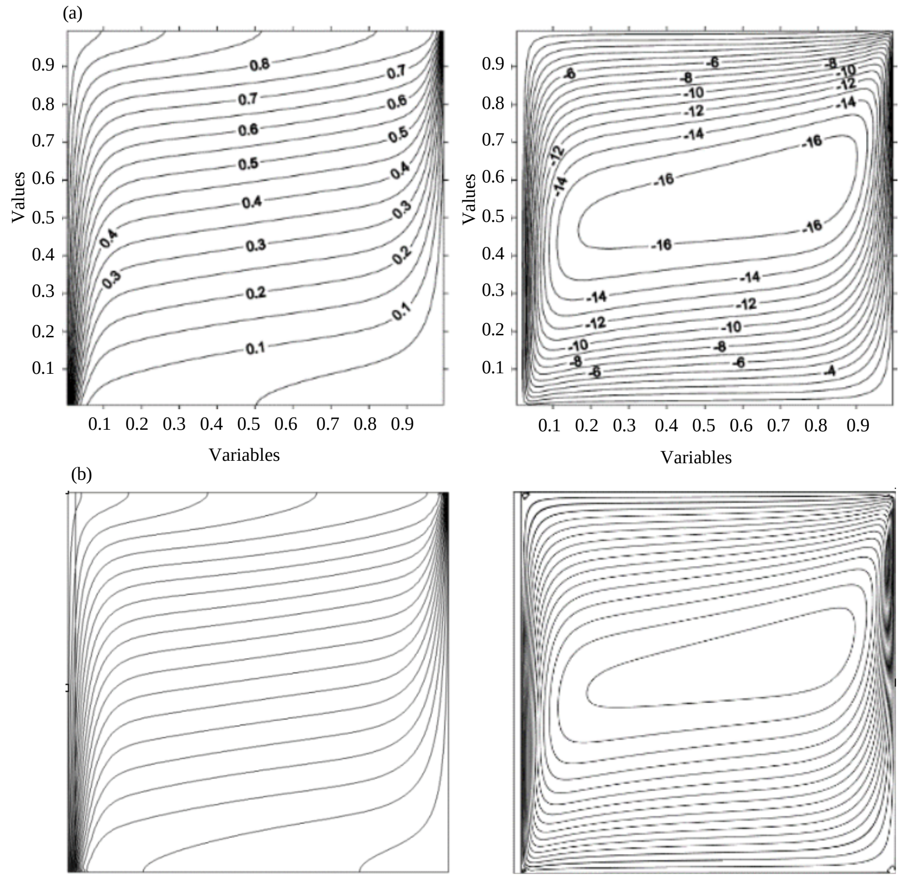


Fig. 2: Comparison of streamlines and isotherms contours between the present research and (Saeid, 2007) at Ra 1000, Kr = 1 and D = 0.02; a) Saeid work and b) Pret senior

Table 1: Comparison of average Nusselt number (Nu) with some previous results

Researchers	Ra = 10	Ra = 100
Beckermann *et al.* (1986)		3.11
Moya *et al.* (1987)	1.07	2.80
Nithiarasu *et al.* (1996)	1.08	3.02
Al-Farhany and Turan (2011)	1.08	3.13
Sameh *et al.* (2016)	1.09	3.01
Present research	1.08	3.12

RESULTS AND DISCUSSION

In this study, results are presented for different values of dimensionless parameters including, the modified Rayleigh number (10• Ra• 10^3), aspect ratio (0.5• A• 2), dimensionless wall thickness (0.02• D• 0.5) and the Thermal conductivity ratio (0.1• K_r• 10).

Effects of modified Rayleigh number: Figure 3 show the effects of modified Rayleigh number at A = 2, D = 0.1 and K_r = 1 in terms of streamlines and isotherms for Ra = 10 and Ra = 1000. It is observed that when modified Rayleigh number increases, the vortex strength increases for all cases. The effects on the streamline behaviors is very pronounced at high Rayleigh number. These influences on the vortex strength are proportional to buoyancy force increases due to increasing in heat transfer rates.

In case three for high modified Rayleigh number (Ra = 1000) two different vortex strength appears because of the heat sink location. The results also shows most of high isotherms line in the top half of the cavity when the bottom half kept at lowest isotherms (just above zero). Same effects shown in figures.

Figure 4 presents the effect of modified Rayleigh number on the local Nusselt number for the fluid phase and along the thermal active heated wall for the first case. As it is expected, the local Nusselt number increase as modified Rayleigh number increases. The physical reason behind this is that asmodified Rayleigh number increases, natural convection and buoyancy force increase. For example, at Ra = 10, Nu_s = 1.66 and Nu_f = 0.679 while when Ra = 1000 they increases to 5.95 and 2.20, respectively. In general, the local Nusselt number for the fluid phase increase near the vertical wall from bottom to top until be maximum near the middle of the cavity then it decrease to minimum value near the top left corner.

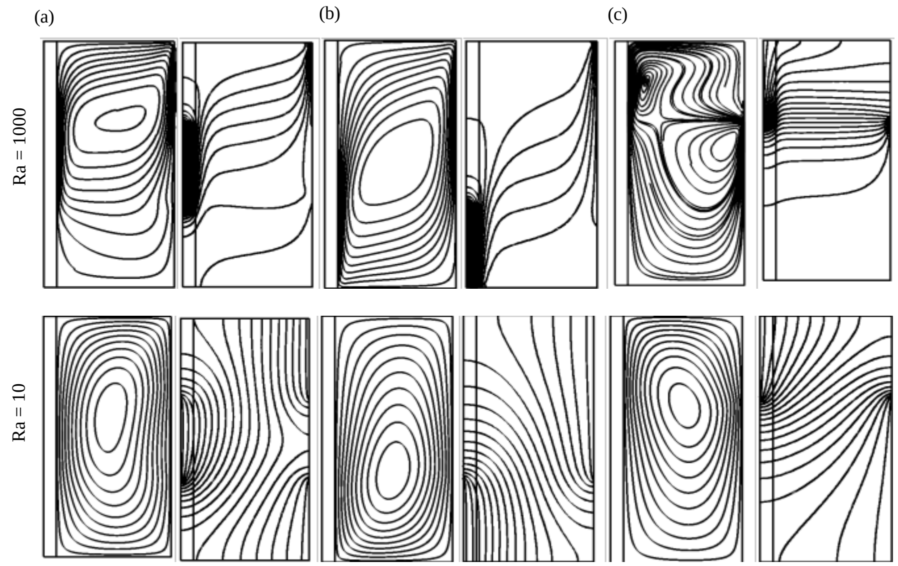


Fig.3: Streamlines and isotherms for different active thermally walls locations and modified Rayleigh number at A = 2, D = 0.1 and Kr = 1; a) Case 1; b) Case 2 and c) Case 3

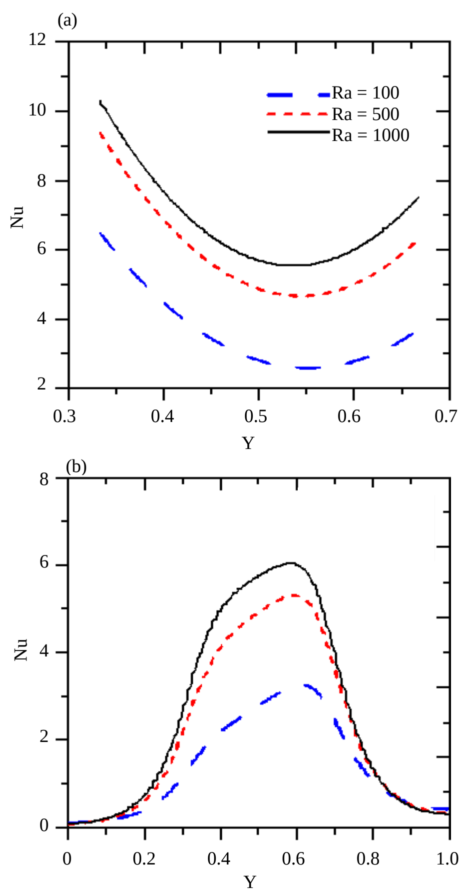
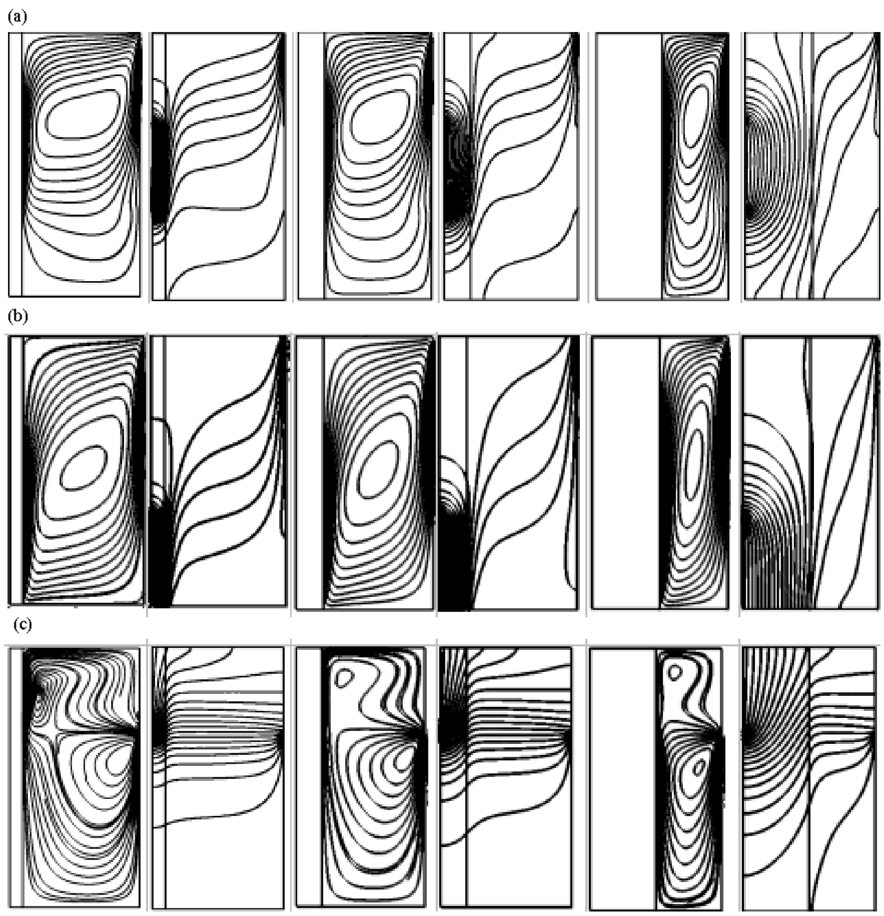


Fig. 4: Variation of local Nu_s (left) and local Nu_f (right) with different values of modified Rayleigh number for case 1, at A = 2, D = 0.1 and Kr = 1

Effect of dimensionless wall thickness: To show the effects of dimensionless wall thickness streamline and isotherms are presented in Fig. 5 for the three cases at Ra = 1000, Kr = 1 and A = 2. The flow circulation strength in the porous medium is higher with thin walls while it was lower with thick wall as shown in Fig. 5. The average Nusselt number for the fluid phase on the left wall porous interface increase when the wall thickness is decreased and that because the conduction heat transfer is dominant with increasing wall thickness as shown in Fig. 6. For example when D = 0.1, Nu_s = 5.95 and Nu_f = 2.20 while the increasing of dimensionless wall thickness to D = 0.5, Nu_s = 2.07 and Nu_f = 0.91. The local Nusselt numbers of fluid phase Nu_f at the lower and upper parts of the left sidewall increase when wall thickness (D) increases. On the other hand, Nu_f decrease when the wall thickness (D) increases in the mid-way of the wall.

As the dimensionless wall thickness increases, the temperature difference between the interface temperature and the cold boundary decreases which reduces the local Nusselt number at the mid-way of the wall.

Effect of thermal conductivity ratio: Figure 7 shows the effect of thermal conductivity ratio on heat transfer rate at Ra = 1000, D = 0.1 and A = 1. In this study, the thermal conductivity ratio (K_r) is defined as the ratio of the thermal conductivity of solid walls to the thermal conductivity of the fluid. Therefore, from this definition it can be obtained that when K_r is small, (K_r<1) the thermal conductivity of walls is small too. That effects will be increase when the thermal conductivity ratio increase specially when



D = 0.1 D = 0.2 D = 0.5

Fig. 5: Streamlines and isotherms for different finite wall thickness and active wall locations at Ra = 1000, Kr = 1 and A = 2; a) Case 1; b) Case 2 and c) Case 3

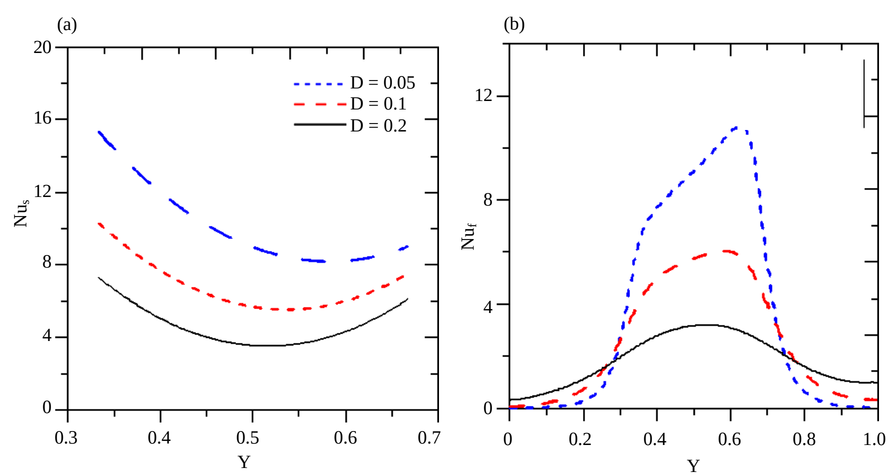


Fig. 6: Profile of local Nusselt number (Nu_s and Nu_f) with various finite wall thickness for case 1, at Ra = 1000, Kr = 1 and A = 2

(K_r = 10). Also, results shows that when the thermal resistance is high, the average Nusselt number at solid walls is high while the average Nusselt number of fluids near left solid wall is low. On the contrary when the

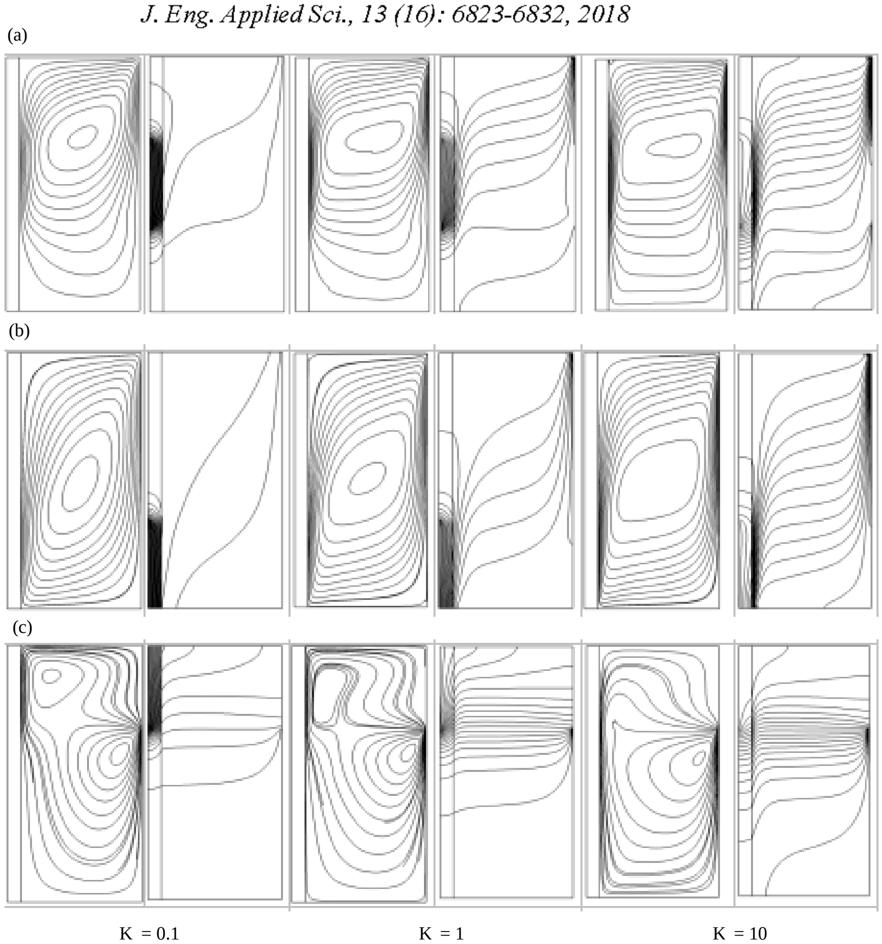


Fig. 7: Streamlines (left) and isotherms (right) for different thermal conductivity ratio and active wall locations at Ra = 1000, D = 0.1 and A = 2; a) Case 1; b) Case 2 and c) Case 3

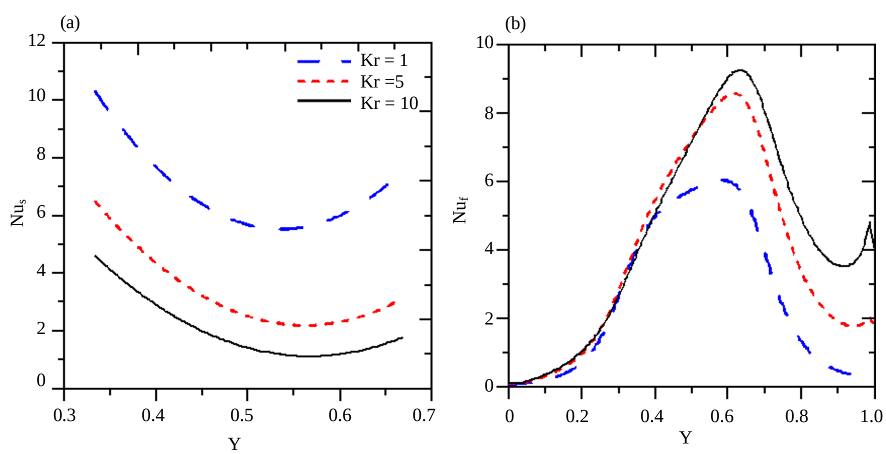


Fig. 8: Profile of local Nu_s (left) and local Nu_f (right) for different values of thermal conductivity ratio for case 1, at Ra = 1000, D = 0.1 and A = 1

thermal conductivity ratio (1<K_r• 10) increases, the thermal conductivity of solid walls increases while the average Nusselt number at fluid walls decreases. In addition, the results demonstrated that the heat transfer mechanism inside the enclosure is converted from conduction when K_r is small to convection when K_r is high as shown in Fig. 8.

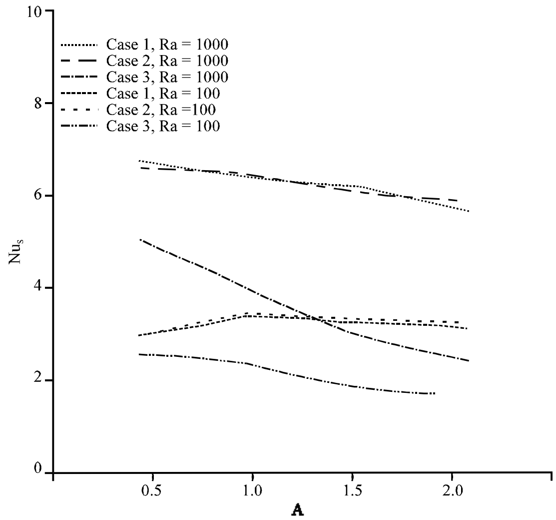


Fig. 9: Profile of average Nusselt number (Nu_s) vs. Aspect ratio (A) for various thermally active wall locations

Table 2: Variation of average Nusselt number (Nu) with different Ra for three cases at A = 2

Ra	Case 1	Case 2	Case 3
Nu_f			
10	0.68	0.47	0.45
100	1.26	1.19	0.61
1000	2.20	2.16	0.89
Nu_s			
10	1.67	1.25	1.14
100	3.25	3.38	1.27
1000	5.95	6.21	1.51

Effect of aspect ratio: Figure 9 illustrates the effect aspect ratio on average Nusselt number along the solid heated wall for various thermally active walls. Under Ra = 100, both case 1 and 2 give similar behavior of Nusselt number with aspect ratio. Where Nusselt number increases until A = 1 and then slightly decreases. While for Ra = 1000, Nusselt number decreases with the increasing of aspect ratio. Case 3 gives similar behavior with respect to aspect ratio for both mentioned values of Ra number.

Effect of thermally active walls: The main emphases of the present study are how the location of the active walls effects on heat transfer rate. Figure 10 displays the effect of three cases on average Nusselt number along the fluid phase for various modified Rayleigh number and aspect ratios at rate at K_r = 1 and D = 0.1. It can be noticed that the case 1 gives the higher rate of heat transfer while the case 3 is the lowest one. Figure 11 demonstrates the effect of average Nusselt number at Kr = 1, D = 0.1 and A = 0.5 with respect to modified Rayleigh number for different locations of thermally active walls where case 1 gives higher rate of heat transfer.

Tables 1-4 shows the variation of average Nusselt Number (Nu) with different Ra, D and K_r for the three cases at A = 2.

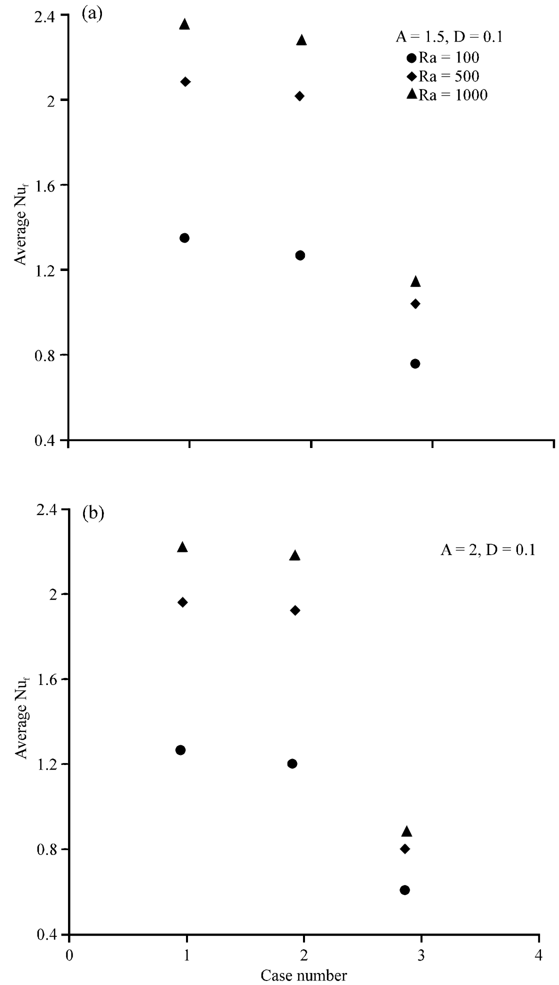


Fig. 10: Average Nusselt number (Nu_f) for different Ra, A and active wall positions for D = 0.1 and Kr = 1

Table 3: Variation of average Nusselt number (Nu) with different D for three cases at A = 2

D	Case 1	Case 2	Case 3
Nu_f			
0.1	2.20	2.16	0.85
0.2	1.95	1.37	0.73
0.5	0.90	0.74	0.51
Nu_s			
0.1	5.95	6.21	2.44
0.2	3.78	3.78	2.12
0.5	2.07	1.90	1.45

Table 4: Variation of average Nusselt number (Nu) with different Kr for three cases at A = 2, D = 0.1, Ra = 1000

K_r	Case 1	Case 2	Case 3
Nu_f			
0.1	1.87	1.75	1.39
1	2.20	2.16	0.85
10	3.19	3.40	0.98
Nu_s			
0.1	9.04	8.90	7.30
1	5.95	6.21	2.58
10	1.52	1.80	0.51

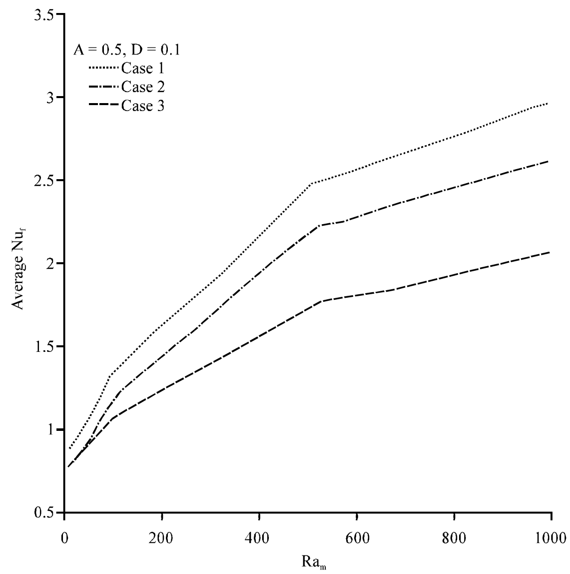


Fig. 11: Average Nusselt number (Nu_f) vs. modified Rayleigh number (Ra) for different locations for case 1, at Kr = 1, D = 0.1 and A = 0.5

CONCLUSION

Conjugate natural convection heat transfer in a partially heated porous rectangular have been investigated numerically. Three different cases of heating and cooling zone had been taken in the consideration along the vertical walls while the others are considered to be adiabatic. A wide range of non-dimensional parameters used including Ra, A, D and Kr. The Darcy Model with Boussinesq approximation have been used to solve the governing equations. The results shown that as A and D increasing, Nusselt number decreasing while it is increasing when Ra and Kr increasing. Same effects can be seen for case 1 and 2. Finally, the first case gives the higher rate of heat transfer while the case 3 is the lowest heat rate at same magnitude of modified Rayleigh number.

NOMENCLATURE

A = Aspect ratio
c_p = Specific heat at constant pressure
d = Dimensional wall thickness m^{-1}
D = Non-dimensional wall thickness D = d/L
Da = Darcy number
g = Gravitational acceleration, m s^{-2}
H = High of the enclosure, m
K = Permeability of the porous medium, m^2
k = Thermal conductivity, W m^{-1} K^{-1}
k_{eff} = Effective thermal conductivity of porous medium
k = thermal conductivity, W m^{-1} K^{-1}
k_r = Thermal conductivity ratio, $k_r = k_w/k_f$
L = length of the enclosure, m
Nu = Average Nusselt number
p = Pressure, kg m^{-1} s^{-2}
P = Non-dimensional pressure
Pr = Prandtl number, Pr = v/•
Ra = Modified Rayleigh number for porous medium
t = time, s
T = Non-dimensional temperature
u = Velocity components in x-direction, m s^{-1}
U = Dimensionless velocity components X-direction
v = Velocity components in y-direction, m s^{-1}
V = Dimensionless velocity components Y-direction
x = X coordinates, m
X = Non-dimensional X-coordinates
y = Y coordinates, m
Y = Non-dimensional Y-coordinates

Greek symbols:
• = Effective thermal diffusivity, m^2 s^{-1}
$•_T$ = Coefficient of thermal expansion, K^{-1}
• = Porosity of the porous media
v = Kinematic viscosity, m^2 s^{-1}
• = Density, kg m^{-3}
• = Ratio of specific heats

Subscripts:
c = Cold
eff = Effective
f = Fluid
h = Hot
s = Solid
w = Wall

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