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Study the Mixed Convection in a Cavity with Inclination Wall Subjected to Solar Heat Flux with Inner Strip

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Abstract: A numerical study is presented on natural convection heat transfer in inclined wall solar chimney. A 2-D plane, steady, incompressible, laminar flow field developed by natural convection inside a cavity with aspect ratio W/H (1.1, 1.2, 1.3 and 2.0) an inclined wall as a solar chimney at different inclination angles ranging from 20°, 40°, 60°, 80°, 90° for Raleigh number (3.2×10^7-3.2×10^8) and Richardson number at (0.5, 0.7, 1.4, 7) from and chimney glass wall thickness (6 mm) and to study the effect of using the forced convection heat transfer the Reynolds number was varied by 250, 750, 1500, 1800 in the laminar rang flows from a bottom (20 mm) opening and exits from anther bottom (20 mm) opining. Also to increase the distribution of the hot air inside the cavity using a baffle with varying its height ratio by $H_b = h_b/H = 0.125$, 0.25, 0.5 and 0.6 mm and increasing its distant ratio from the heated wall by $D_b = x_b/L = 0.1$, 0.4, 0.6 and 0.8 mm chimney is investigated numerically.

Key words: Cavity, induced flow, mixed natural convection, solar chimney, different inclination angles, investigated numerically

INTRODUCTION

The heat transfer by mixed convection coupled to the radiation in a rectangular cavity continues to be a fertile area of research, due to the interest of the phenomenon in many technological processes such as the design of solar collectors, thermal design of buildings, air conditioning and recently the cooling of electronic circuit boards, electronic enclosures and industrial furnaces. The basic nature of the problem interaction between the forced external air stream and the buoyancy-driven flow by the heat source may be found in many applications (Ambarita *et al*., 2006).

Presented a numerical study on a square cavity with differentially heated which is formed by horizontal adiabatic walls and vertical isothermal walls. The used two baffles which perfectly insulated were attached to the cavity horizontal walls. Their study performed parameters of Rayleigh number from 10^4-10^8, non-dimensional thin baffles length are 0.6, 0.7 and 0.8, non-dimensional baffle positions from 0.2-0.8. They showed that the two baffles trap some fluid in the cavity and affect the flow fields and the Nusselt number strongly depends on baffle positions (Elatar *et al*., 2016).

The effect of height partition on combined mixed convection and surface radiation in a vented rectangular cavity is study by Haghighi and Vafai (2014). They showed that the relative height of the partition, contributes to increase/decrease the radiative/convective heat transfer component at the level of the heated wall.

Nardini *et al*. (2015) investigated the differentially heated square cavity with two insulated horizontal baffles. Their results presented for Rayleigh number from 10^4 up to 10^7 and are in form of streamlines, isotherms as well as Nusselt number. They observed that the two baffles trap some fluid in the cavity and affect the flow fields. Also, the Nusselt number decreases with baffle length. Finally, they have shown that Nusselt number changes with baffles position.

Studied numerical the natural convection at Rayleigh number, R_a in the range of 10^5-10^8 inside right angled triangle with aspect ratio of A = 1.0. To investigate the thermal boundary layer adjacent to the heated inclined wall with an adiabatic fin attached to that surface. They showed that the thermal boundary layer adjacent to the inclined wall breaks initially in presence of adiabatic fin (Paul *et al*., 2012).

Study presented an analysis study on the capabilities heat transfer reduction in a differentially heated cavity with insulated top and bottom walls with a vertical or a horizontal adiabatic partial partition. They considered the effects of partition length and location for Rayleigh numbers from 10^3-10^6 and for aspect ratios from 1-4. Based on their results, vertical baffles more efficiently reduce heat transfer in most cases (Ghassemi *et al*., 2007).

The study of two-dimensional open top cavity with a conjugate natural convection with surface radiation. Their results indicate that the surface radiation changes the basic flow physics and enhances the radiative heat transfer as a result of which heat transfer by convection decreases (Belmiloud *et al*., 2017).

Moreover, the natural convection in a square enclosure filled with air was analyzed experimentally and numerically (Bahlaoui *et al*., 2011). The used two Plexiglas baffles are attached to the cavity's vertical walls. Their results are presented for Rayleigh numbers from 10^4-10^5. They showed that the different baffle lengths have a significant effect on the heat transfer and flow characteristics of the fluid. Also, the effect of baffle number on mixed convection within a ventilated cavity was presented (Bahlaoui *et al*., 2011).

The study investigate the numerically the effect of a single horizontal fin attached to the hot wall of a square enclosure on the laminar natural convection. The Rayleigh number was ranged between 10^3 and 10^6, fin Length (L) and position (H) were in the range between 0.125 and 0.875. They showed that the fin thickness has negligible effect on the average Nusselt number for all values of fin conductivity ratios (Belmiloud, 2015). Finally the influence of the baffle length and emissivity of the walls on heat transfer in a mixed convection coupled with radiation in a rectangular ventilated cavity was studied numerically (Singh and Singh, 2015). The baffle length is used = 0.3, 0.5 and 0.7. The left vertical wall is heated to a constant heat flux and the other walls are adiabatic. The Reynolds number R_e between 50 and 500 and the Grashof number fixed to 10^4. Their results showed that the heat transfer enhancement depends on the increase in emissivity. However, the increase in total Nusselt number Nu depends on the increase of the baffle length.

In the present numerical study the flow and heat transfer governing equation was solved by finite element method inside a 2D cavity at Raleigh number (3.2×10^7, 6.4×10^7, 1.2×10^8 and 3.2×10^8). Its right wall was heated by a solar source and titled with vertical axis by an angle of inclination by γ. It was using an immersed adiabatic baffle inside the cavity with different height and distance.

MATERIALS AN METHODS

Consider the motion of a viscous fluid within a semi-trapezoidal enclosure with vertical right angle left wall with dimension (H) and right wall tilted at inclination angle of (γ) with the Y-axis it was assumed to be isothermally heated by a constant solar heat flux (q'') as shown in Fig. 1. The adiabatic bottom wall has a one inlet and one outlet with Equal length of (E).

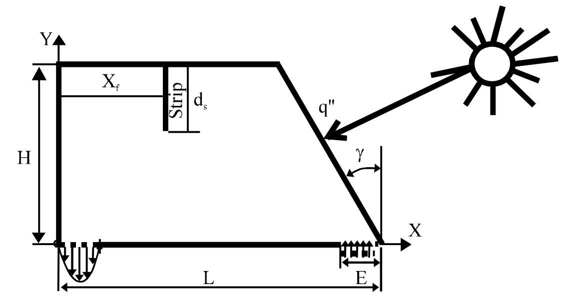

Fig. 1: Schematic diagram of the physical system

The fluid properties are assumed constant except for the density variation which is treated according to Boussinesq approximation. The present flow is considered steady, laminar, incompressible and two-dimensional. The viscous incompressible flow and the temperature distribution inside the enclosure are described by the Navier-Stokes and the energy equations, respectively (Hsu *et al*., 1997). Continuity Eq. 1:

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

X-component of momentum equation:

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

Y-component of momentum equation:

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+v\left[\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right]+g\beta(T\text{-}T_C) \quad (3)$$

The energy Eq. 4:

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

The non-dimensional variables used in the above equations are defined as:

$$X=\frac{x}{L},\ Y=\frac{y}{L},\ U=\frac{u}{u_i},\ V=\frac{v}{u_i},\ P=\frac{p}{\rho u_i^2},\ \theta=\frac{T\text{-}T_i}{T_h\text{-}T_i},\ \theta=\frac{T_s\text{-}T_i}{T_h\text{-}T_i}$$

And the parameters R_e, R_i, P_r and K are defined as:

$$R_e=\frac{u_i E}{v},\ R_i=\frac{g\beta(T\text{-}T_i)E}{u_i^2},\ P_r=\frac{v}{\alpha}$$

Using non-dimensional variables defined below, the non-dimensional forms of the governing equations of the present problem are obtained as follows. Dimensionless continuity (Eq. 5):

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \tag{5}$$

Dimensionless X-component of momentum (Eq. 6):

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\frac{1}{Re}\left[\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right] \tag{6}$$

Dimensionless Y-component of momentum (Eq. 7):

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\frac{1}{Re}\left[\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right]+Ri\theta \tag{7}$$

The energy (Eq. 8):

$$U\frac{\partial T}{\partial X}+V\frac{\partial T}{\partial Y}=\frac{1}{Re\,Pr}\left[\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right] \tag{8}$$

The appropriate dimensionless form of the boundary conditions summarized by Fig. 2 used to solve equations 1-5 inside the cavity is given as: At the inlet:

$$U=0,\ V=1,\ \theta=0$$

Convective Boundary Condition (CBC) at the outlet:

$$P=0$$

At all wall boundaries:

$$U=0,\ V=0,\ \frac{\partial p}{\partial n}=0$$

At the solar heated right titled wall:

$$\theta=1$$

At the left, top and bottom walls:

$$\left.\frac{\partial\theta}{\partial X}\right|_{X=0}=\left.\frac{\partial\theta}{\partial Y}\right|_{Y=1,0}$$

The average Nusselt number (Nu) at the hot wall is defined as:

$$Nu=\frac{1}{H}\int_0^{H\cos(\gamma)}\left.\frac{\partial\theta}{\partial X}\right|_{X=1}dY \tag{9}$$

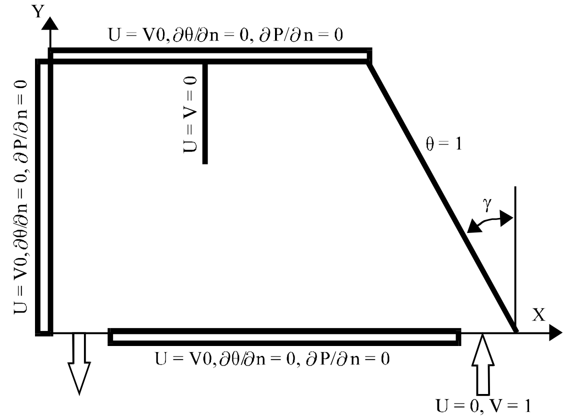

Fig. 2: Schematic diagram of the boundary conditions

And the bulk average temperature in the cavity is defined as:

$$\theta_{av}=\int\frac{1}{\bar{V}}\theta\, d\bar{V} \tag{10}$$

Numerical technique: In this numerical study the finite volume method was considered in order to discretization the above governing equations. The discretization equations is transformed the system of elliptic partial differential equations into a system of algebraic equations by using the implicit line by line Gause elimination scheme. A Fortran computer program is developed to attain the results using the pressure velocity coupling (Simplec algorithm) (De Vahl Davis, 1983). The relaxation factors used for velocity components, temperature and pressure are 0.3, 0.6 and 0.8, respectively. These relaxation factors have been adjusted for each case studied in order to accelerate convergence. Non uniform grid with refinements near the walls is used. The computational grids are staggered for the scalar variables and not staggered for the scalar one as presented in Fig. 3. The model is terminated when the mass, momentum and energy for each simulation evaluated over the course, residuals drop below 10^{-7}. Moreover, in these numerical simulations, the convergence criterion for temperature, pressure and velocity is Versteeg and Malalasekera (1995):

$$Error=\frac{\sum_{k=1}^{l}\sum_{i=1}^{m}\sum_{j=1}^{n}\left|\xi_{i,j,k}^{t+1}-\xi_{i,j,k}^{t}\right|}{\sum_{k=1}^{l}\sum_{i=1}^{m}\sum_{j=1}^{n}\left|\xi_{i,j,k}^{t+1}\right|}\le 10^{-7} \tag{11}$$

Grid independence test: Computations have carried out for three selected grid sizes (i.e., 150×200, 180×220, 200×250). Figure 4 shows the local Nusselt number distribution along the inclined heated wall subjected to a constant solar heat flux for 6.4×10^7. Results for the

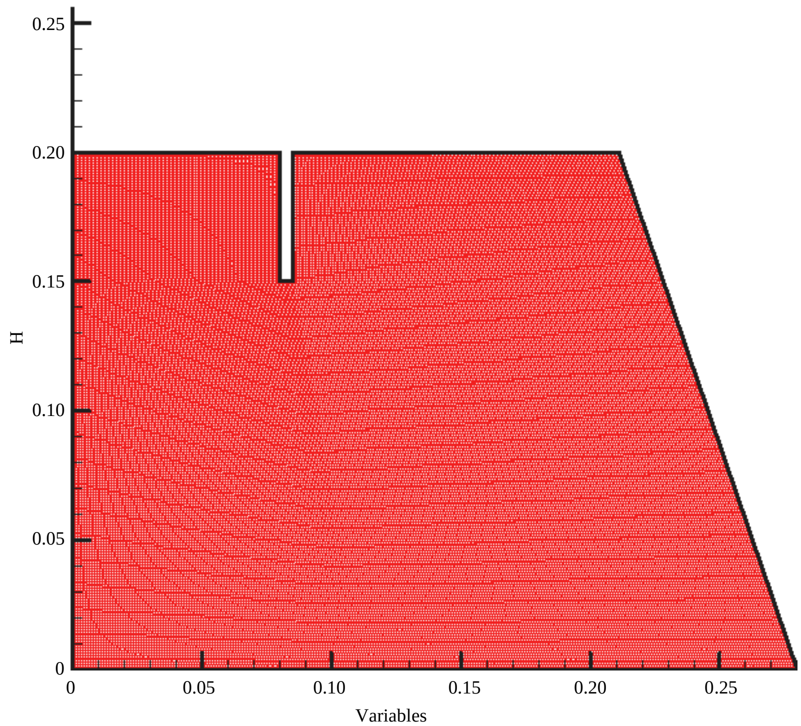

Fig. 3: Side view of the three dimensional cavity (A = 12)

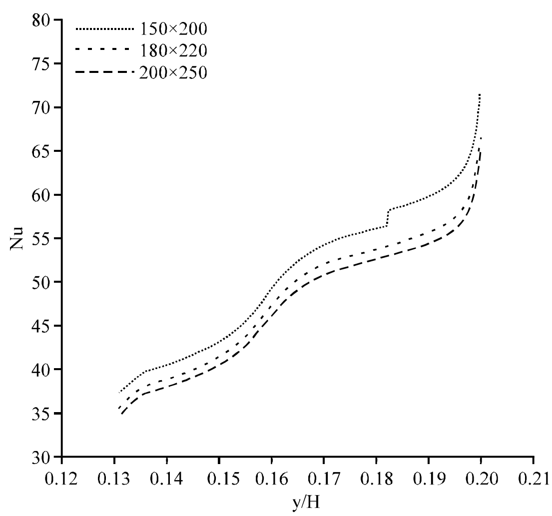

Fig. 4: Grid independence test for local Nusselt number for 6.4×10^7, L/H = 1.0, $x_b/L = 0.4$, $\theta° = 20$, $h_b/H = 0.5$

selected grid sizes show very good agreement with each other. Maximum grid (200×250) is presented throughout this study in order to obtain an optimum simulation accuracy.

RESULTS AND DISCUSSION

The present numerical results carried out for 2D mixed natural convection heat transfer and fluid flow in a partitioned porous triangular enclosure. The numerical study presented under the following parameter: cavity with aspect ratio (A_r = W/H = 1.1, 1.2, 1.3 and 2.0) an inclined wall as a solar chimney at different inclination angles ranging from (γ = 20°, 40°, 60°, 80° and 90°) for Raleigh number ($R_a = 3.2\times10^7$ to 3.2×10^8) and Richardson number at (R_i = 0.5, 0.7, 1.4 and 7) and to study the effect of using the forced convection heat transfer the Reynolds number was varied by 250, 750, 1500, 1800. The distribution of convective y-velocity, isotherm contours, local and average Nusselt numbers are obtained under different parameters such as, baffle height ratio by $H_b = h_b/H$ = 0.125, 0.25, 0.5 and 0.6 mm and increasing its distant ratio from the heated wall by $D_b = x_b/W$ = 0.1, 0.4, 0.6 and 0.8 mm.

The distribution of y-velocity and isotherm contours for different values of baffle height ratio by (H_b) are depicted in Fig. 5 through a-d. In this Fig. 5, the isotherm contours distribution at Rayleigh number $R_a = 1.2\times10^8$. In Fig. 5a, the isotherm lines near the baffles and in a part of the enclosure are seem to be parallel and aligned to the heated tilted wall and this represent a semi conduction regime in the cavity in the case of using baffle height ratio by (H_b = 0.125) and the hot air pushed toward the top wall and there is insufficient hot air distribution inside the cavity in this case. Also, it can be show that the velocity increased neat the tilted wall due to the increasing the current of the convective heat transfer near

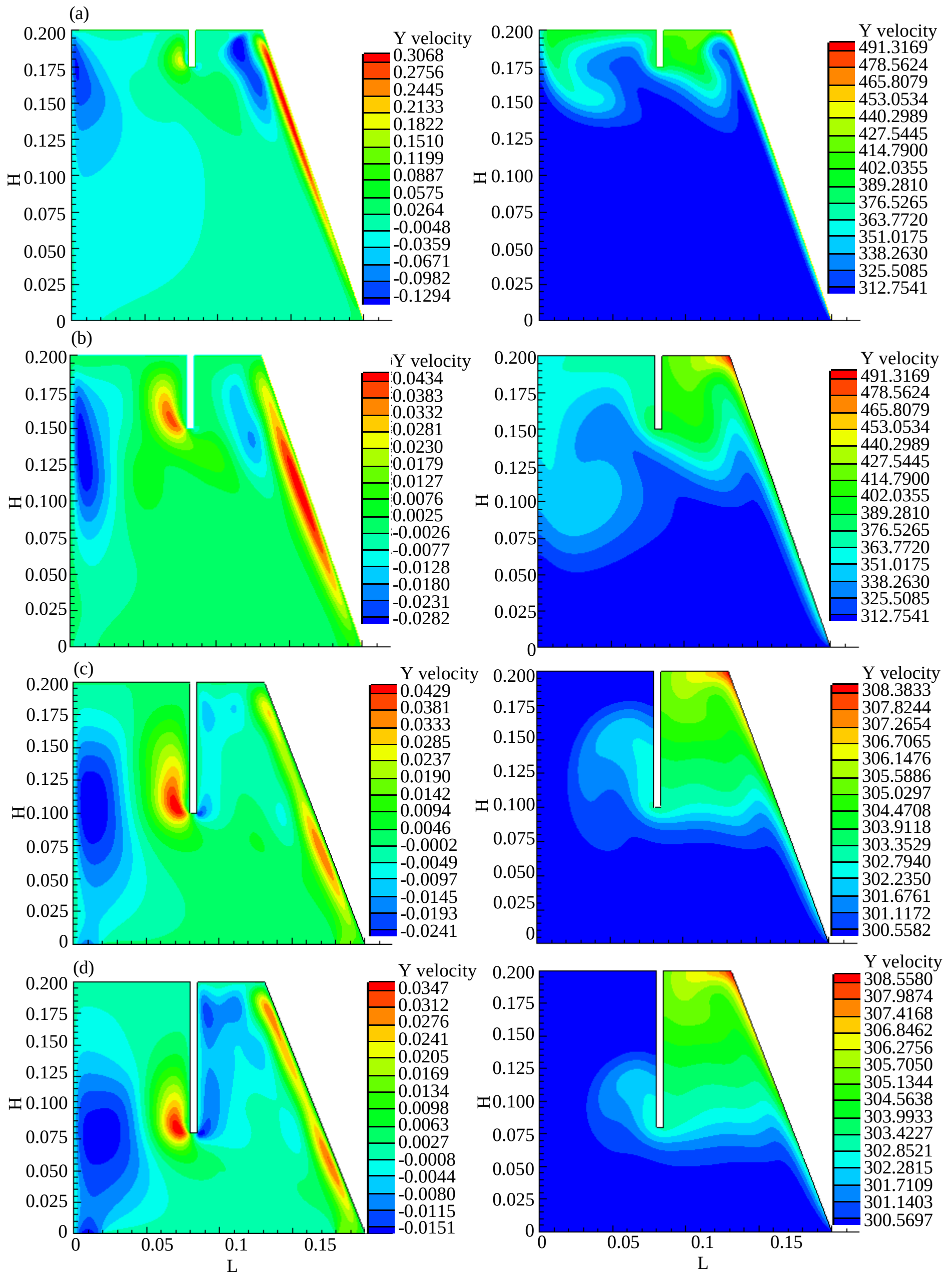

Fig. 5: Effect of baffle height on temperature and velocity at Ra = 1.2×10^8 and A_r = 1.0, D_b = 0.4, θ° = 20; a) H_b = 0.125; b) H_b = 0.25; c) H_b = 0.50; d) H_b = 0.60

the heated wall. When increasing the height ratio to (H_b = 0.25) can be observe that the temperature distribution will increased and fill the half space of the cavity this can conclude that the increasing baffle height ratio (H_b) will increase the temperature distribution in the cavity and it leads to create a y-velocity circulation cell behind the long baffle.

The effect of cavity Aspect ratio (Ar) on the temperatures lines and y-velocity was discussed ssby Fig. 6, show that the temperature lines circulation disappear gradually when increasing the aspect ratio from 1.2-1.6 and the temperature behind the baffle decreasing also. But the y-velocity lines value increasind in front the baffle as the aspect ratio increased. The effect baffle distance from the adiabatic wall (D_b) on the velocity and temperature lines presented in Fig. 7, it can be showed that the temperature lines value near the baffle increasing with increased baffle distance ratio (D_b) at R_a = 1.2×10^8,

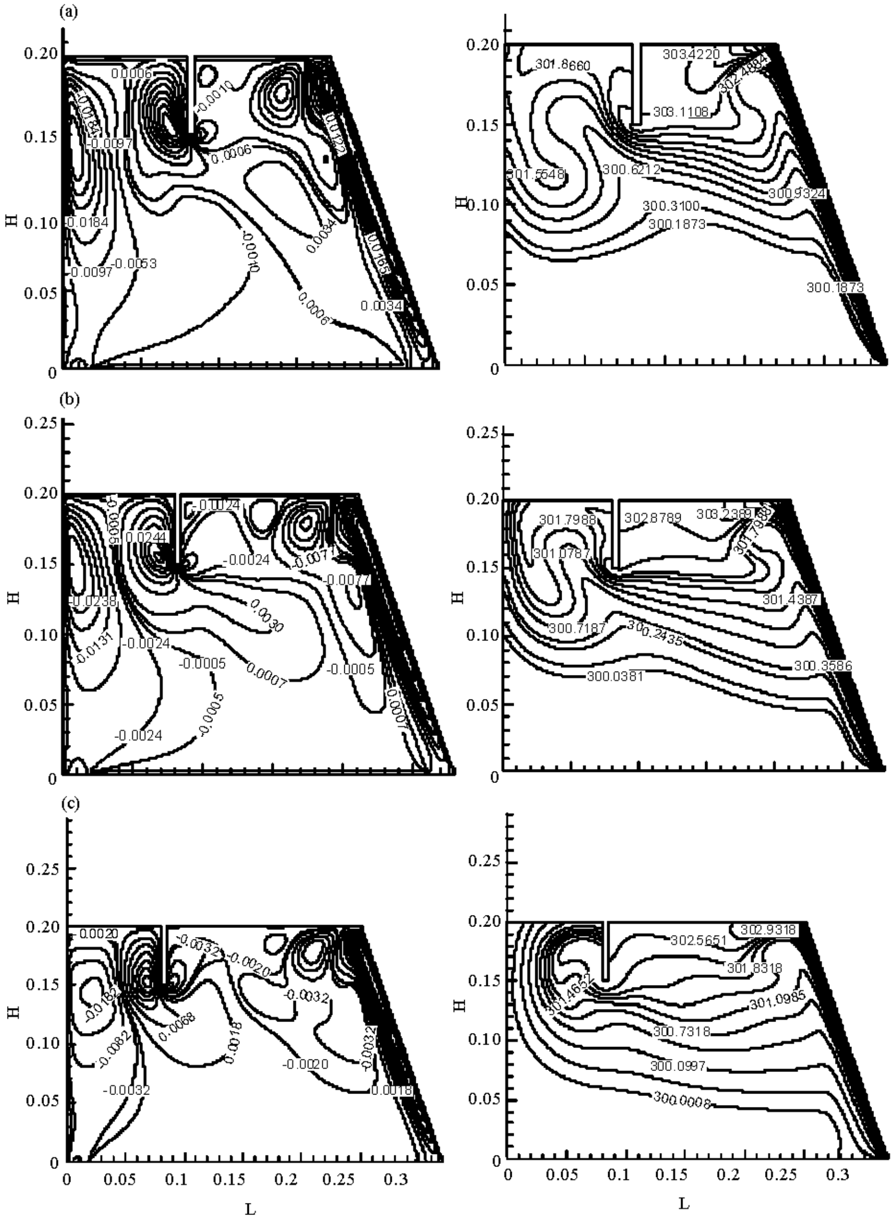

Fig. 6: Effect of cavity aspect ratio on temperature and velocity $R_a = 1.2\times10^8$, $H_b = 0.25$, $D_b = 0.4$, $\theta° = 20$; a) $A_r = 1.2$; b) $A_r = 1.2$; $A_r = 1.2$

Ar = 1.4, $H_b = 0.25$, $\theta° = 20$. And the temperature will move across the baffle as the baffle coming near the heated wall and it circulated behind the baffle. Movere the convective y-velocity spread behind the baffle dramatically when increasing the baffle distance (D_b) and created a cell zones behind it.

The y-velocity in the first column and temperatures lines plot for the variation of the enclosure shape, namely the inclination angle and the aspect ratio are shown in Fig. 8. The inclination angle is changed for a constant aspect ratio $A_r = 1.4$, $H_b = 0.25$, $D_b = 0.6$ with varied the inclination angle $\gamma = 20°$, $30°$, $40°$ and $50°$. The plots are taken for a constant Raleigh number of $R_a = 1.2\times10^8$, a Richardson number $R_i = 0.5$. If the heat source position remains constant, the temperatures lines will change significantly as the inclination angle increased and the timperachres become more and more stratified align the heated wall at ($\gamma = 20°$) as indicated in the figure. the velocity become more fluctuating in the region between the baffle and the adiabatic vertical wall as the inclination angle increased. Increasing the Raleigh number from (3.2×10^7-3.2×10^8) presented in Fig. 9 the results indicated

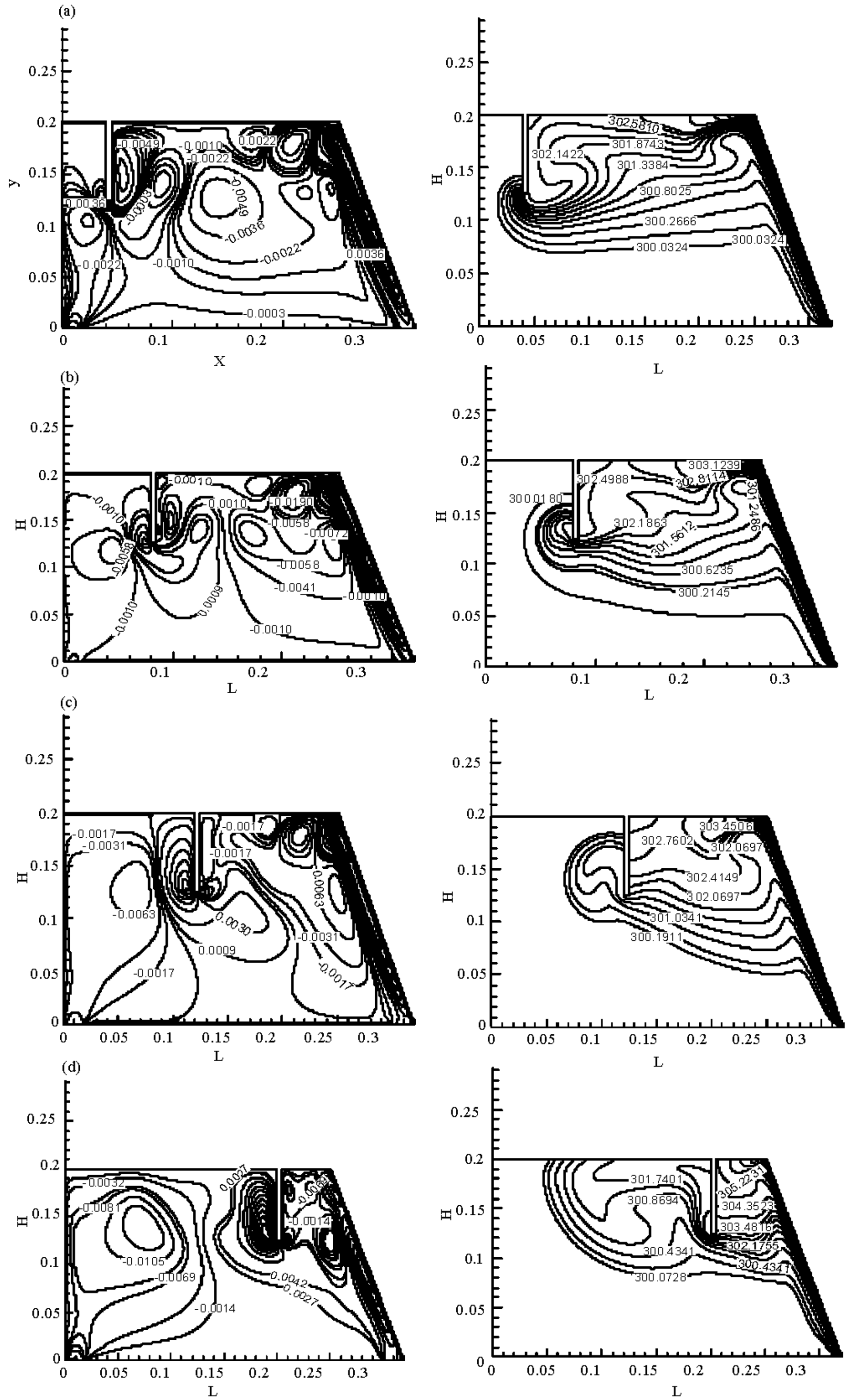

Fig. 7: Effect of baffle position on temperature and velocity at $R_a = 1.2\times10^8$, $A_r = 1.4$, $H_b = 0.25$, $\theta^\circ = 20$; a) $D_b = 0.1$; b) $D_b = 0.4$; c) $D_b = 0.6$; d) $D_b = 1.0$

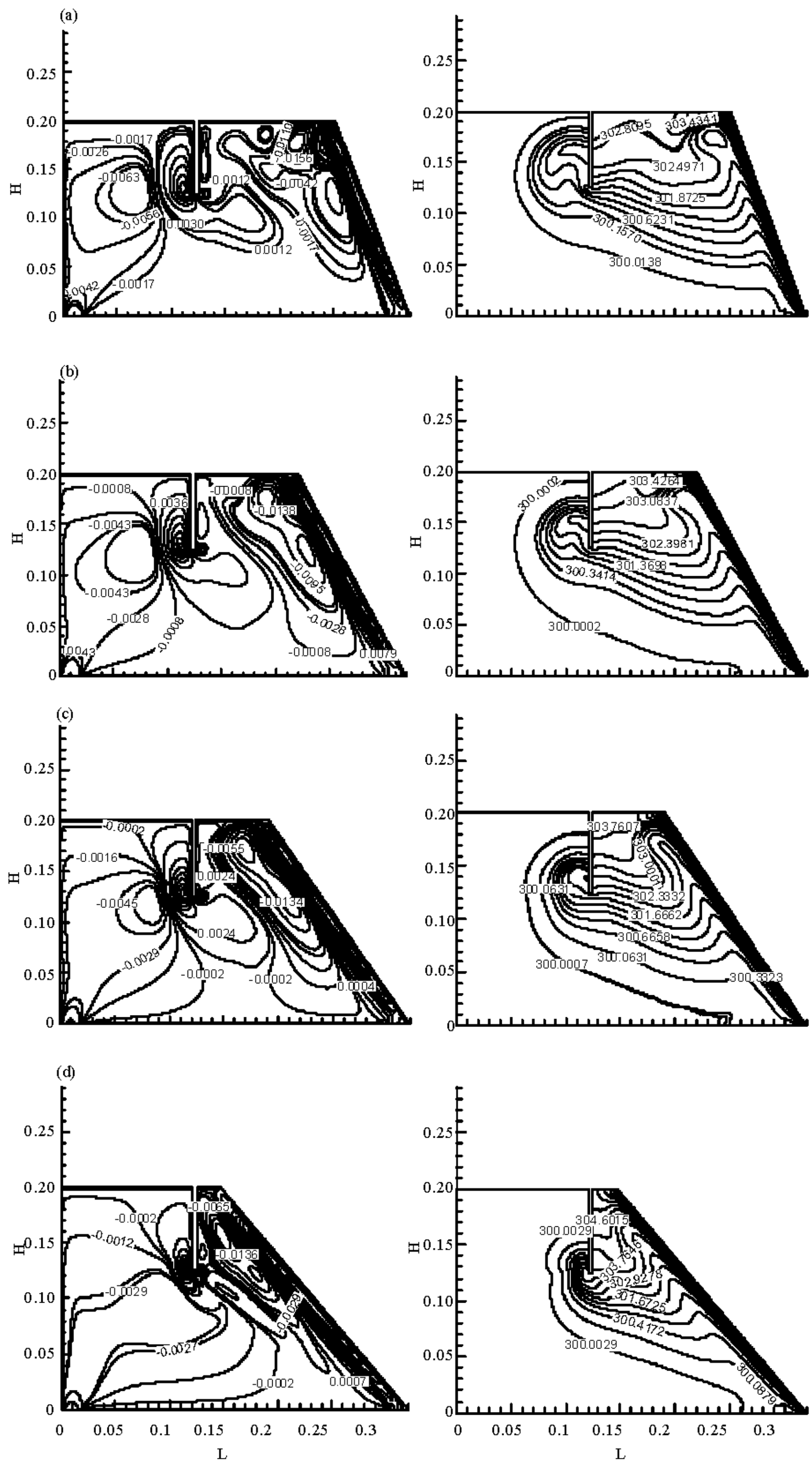

Fig. 8: Effect of wall inclination on temperature and velocity at $R_a = 1.2\times10^8$, $A_r = 1.4$, $H_b = 0.25$, $D_b = 0.6$; a) $\gamma° = 20$; b) $\theta° = 30$; c) $\theta° = 40$; d) $\theta° = 60$

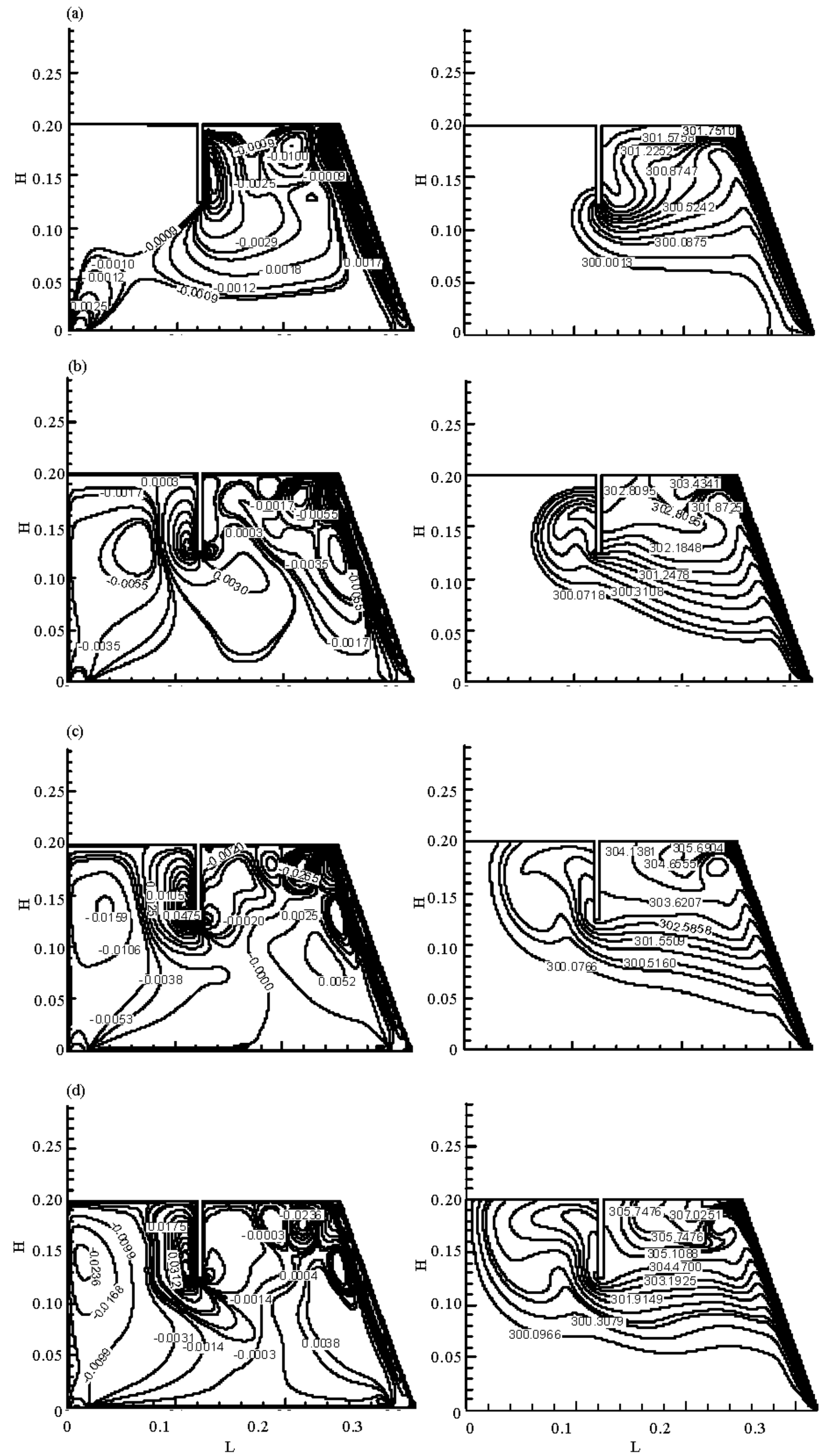

Fig. 9: Effect of Grashof-number on temperature and velocity $A_r = 1.4$, $H_b = 0.25$, $D_b = 0.6$, $\theta° = 20$; a) $R_a = 3.0\times10^7$; b) $R_a = 6.4\times10^7$; c) $R_a = 1.2\times10^8$; d) $R_a = 3.2\times10^8$

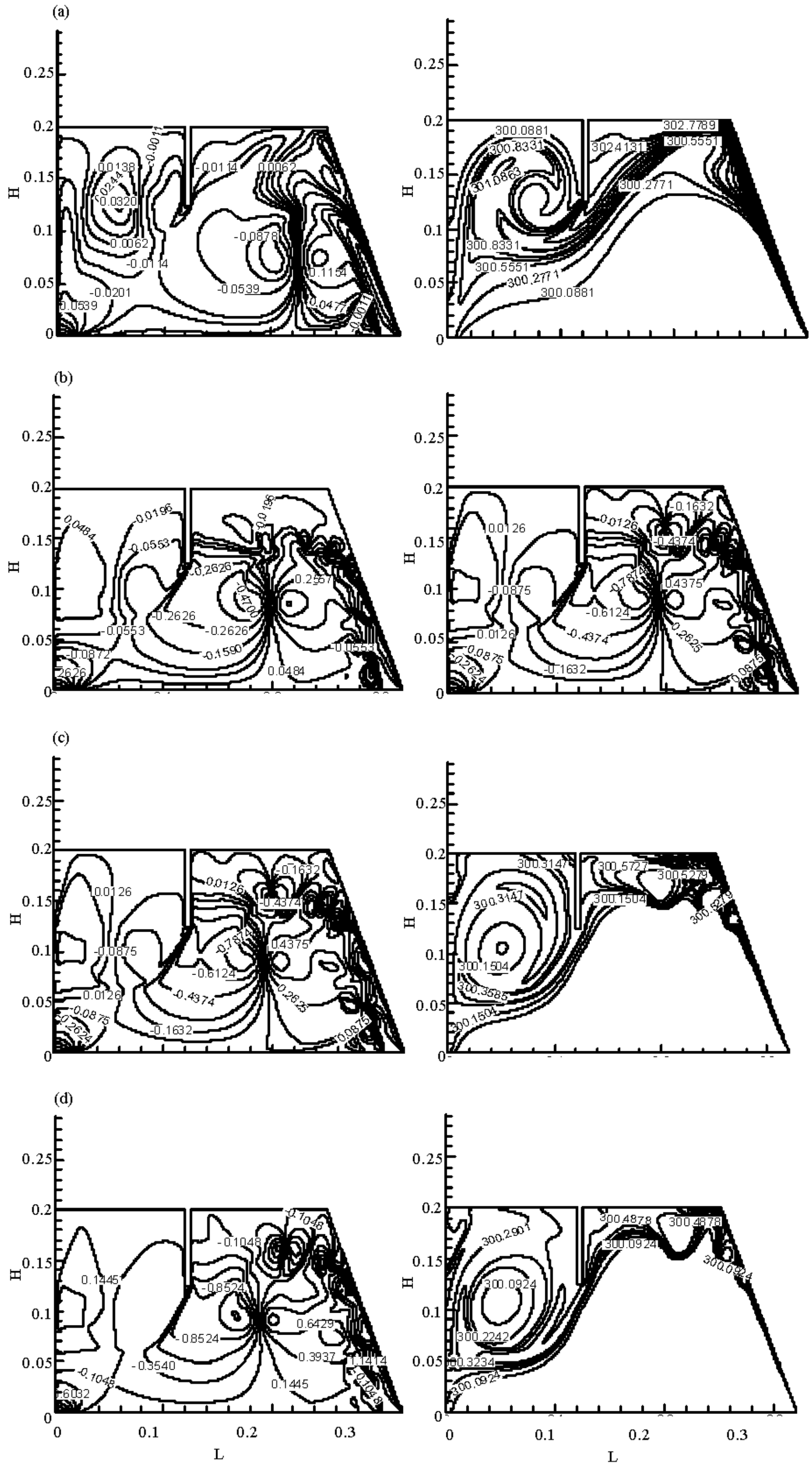

Fig. 10: Effect of Reynolds number on temperature and velocity, $R_a = 1.2\times10^8$ $A_r = 1.4$, $H_b = 0.25$, $D_b = 0.6$, $\theta° = 20$; a) $R_i = 7$; b) $R_i = 1.4$; c) $R_i = 0.7$; $R_i = 0.5$

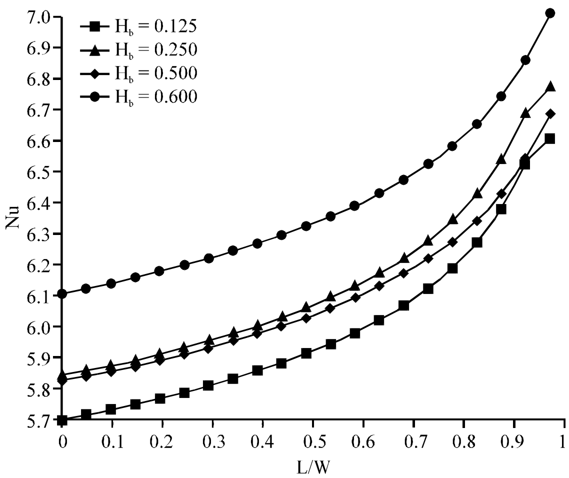

Fig. 11: Local Nusselt number in dependence of the dimensionless heat source length for different baffle height at 6.4×10^7, $A_r = 1.2$, $D_b = 0.6$, $\theta° = 20$

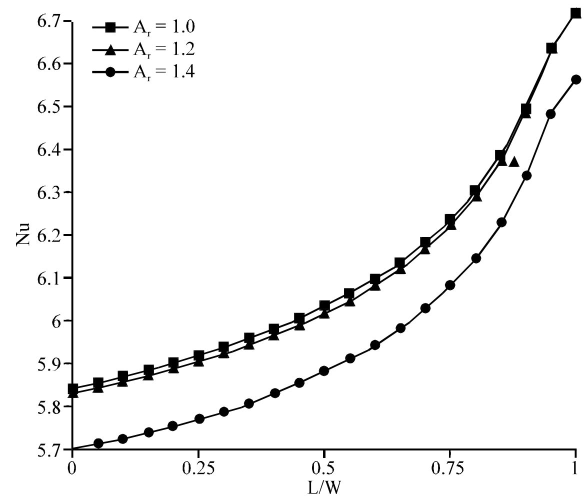

Fig. 12: Local Nusselt number in dependence of the dimensionless heat source length for cavity aspect ratio at $R_a = 6.4\times10^7$, $H_b = 0.25$, $D_b = 0.6$, $\theta° = 20$

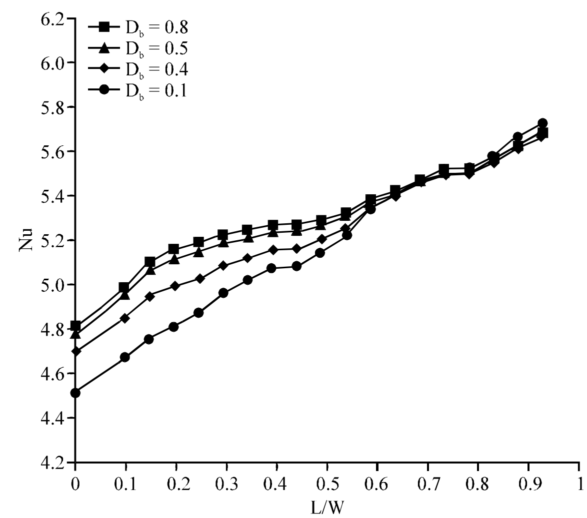

Fig. 13: Local Nusselt number in dependence of the dimensionless heat source length for different baffle height at 6.4×10^7, $A_r = 1.2$, $H_b = 0.25$, $\theta° = 20$

that the temperature increase in front of baffle in the region surrounding between the heated wall and the baffle as the Raleigh number increased. And the velocity distribution spread to the zone between the a baffle and the adiabatic vertical wall and make a high cell circulation flow in this region. The effect of grasheof and inlet Reynolds number assmbleded by Richardson number ($R_i = G_r/R_e$) and plotted in Fig. 10 show that the decreasing the Richardson number from $R_i = 7$ where the neutral convection dominate to $R_i = 1.4$ where the for mixed convection dominate to $R_i = 0.5$ when the forced convection will dominated, it showed that the temperatures distribution near the heated wall was decreased as the Richardson number decreasing but the temperature will circulated behind the baffle. Also, the velocity will increase behind the baffle as the Richardson number decreasing.

The local Nusselt number along the heated solar titled wall was plotted in Fig. 11 with different value of baffle height at fixed flow and heat flux condition. The figure shoed that the Nusselt number basically increased as the dimensionless distance increased from the bottom of the cavity at (L/W = 0) toward the top wall (L/W = 1) due to increasing the convective current aligen the heated wall, also the local Nusselt number increased by about 10% with increasing the baffle height ratio Hb from 0.125-0.6 at (L/W = 0.5) due to the action of the baffle height in the increasing the temperature distribyution near the heated wall.

Moreover Fig. 12 presented the variation of the local Nusselt number with the cavity aspect ratio, the results that the Nu increased with decreasing the Ar. And Fig. 13 plotted the effect of baffle distance ration on the local Nusselt number in the case of using the parameter of $R_a = 6.4\times10^7$, $H_b = 0.25$, $D_b = 0.6$, $\theta° = 20$ the results indicated that the increasing the D_b lead to increasing the Nu due to the pressure the air flow near the heated wall and then increased the temperatures of the film air. And the effect of heated wall Inclination angles and the Ra number on the Nusselt number presented in Fig. 14, the resulted showed that the Nusselt number increased with

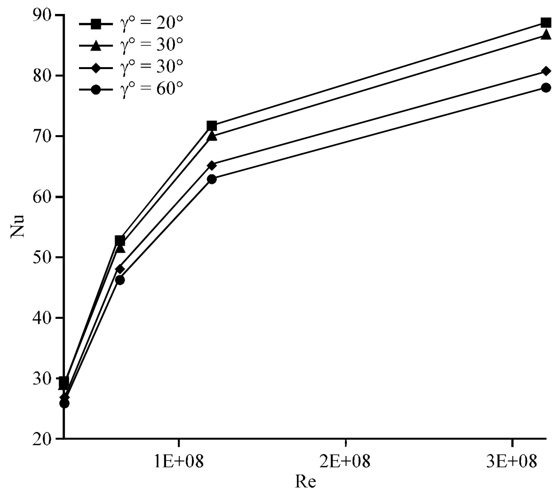

Fig. 14: Average Nusselt number in dependence of the Rayleigh number for different baffle angle of inclinations at A_r = 1.2, H_b = 0.25, D_b = 0.6, $\theta°$ = 20

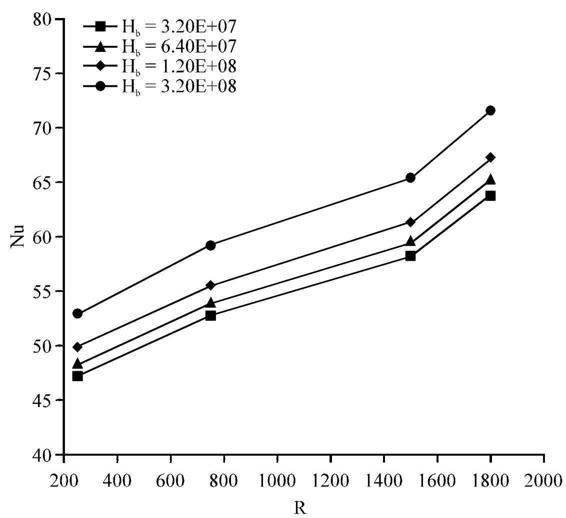

Fig. 15: Average Nusselt number in dependence of the Reynolds number for different baffle Rayleigh number at A_r = 1.2, H_b = 0.25, D_b = 0.6, $\theta°$ = 20

increasing the Ra the same angle but when decreasing the cavity angle of inclination from 50-20°C. It showed that the Nu began to increasing by about 15% at R_a = $1.2\times10^8$and Fig. 15 showed that the average Nu Increasing with increasing the inlet Re for the all value of Ra due to the convective heat transfer near the heated wall. Finally, the effect of baffle height H_b on the Nu plotted in Fig. 16, it showed that the Nu increased with increasing the H_b at the same condition.

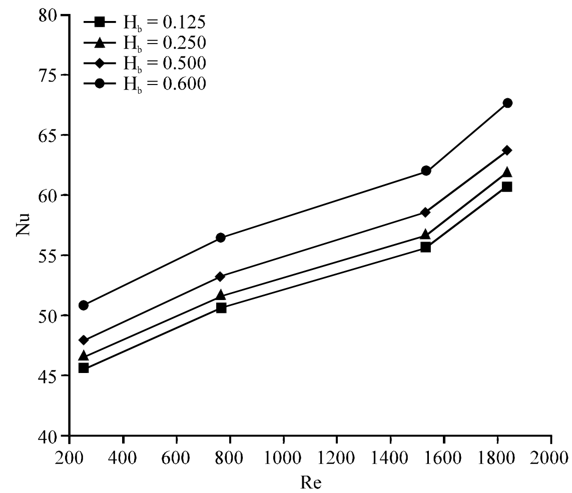

Fig. 16: Average Nusselt number in dependence of the Reynolds number for different baffle height at 6.4×10^7, A_r = 1.2, D_b = 0.6 , $\theta°$ = 20

CONCLUSION

A study on the mixed convection heat transfer of a 2D semi-trbizoidal cavity was presented numerically. The natural convection from a right titled wall subjected to a solar heated with assume constant heat flux. Th all cases. Remarkable improvement of the cavity flow and heat transfer can be achieved with the baffle height and its distance from the heated wall. However the increasing the baffle height and its distance, it is positioned in the mean flow the heat transfer is maximized. Although, changing the enclosure shape by inclination angle and aspect ratio in the way that the enclosure area is maximized but this lead to decreasing the overall Nusselt number. Therefore, the influence of the baffle height and distance ratios is much bigger than the influence of the enclosure inclination angle and aspect ratio on the Nusselt number.

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