Mixed convection flow and heat transfer in a double lid-driven cavity containing a heated square block in the center

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Abstract. In the present work, laminar mixed convection of a Newtonian fluid around a hot obstacle in a square cavity with moving vertical walls is studied numerically. The objective of this study is to analyze the effect of the Richardson number ($0 \leqslant \mathrm{Ri} \leqslant 10$) and Reynolds number ($50 \leqslant \mathrm{Re} \leqslant 500$) on both hydrodynamic and thermal characteristics around a hot obstacle in the enclosure. The analysis of the obtained results shows that the heat transfer is enhanced for high values of Richardson and Reynolds numbers.

1 Introduction

For many years, mixed convection flows are present in many transport processes in nature and in engineering devices. Examples of mixed convection flows can be found in heat exchangers, nuclear reactors, solar energy storage and refrigeration devices, etc. Some studies on mixed convection in a cavity can be encountered [1-2].
Convection in enclosures containing heating block has gained recent research significance as a means of heat transfer enhancement. One of the systematic numerical investigations of this problem was conducted by [3, 4]. Bhuiyan and Munshi [5] studied numerically the phenomena of mixed convection in a lid-driven porous square cavity with internal heat generating. Their conclusion showed that for higher values of the Grashof number and the Darcy number, the streamlines and isotherms are distributed strongly in the enclosed domain and the heat is transferred due to convection. Both Darcy and moving lid ordinations have a significant effect on the flow and thermal fields in the temperature.
The literature shows that the thermal convection in a square cavity with moving vertical walls and equipped with an isothermal obstacle has not been well studied. Motivated by previous works, the objective of the present study is to examine the influence of the Richardson number Ri and the Reynolds number Re on the hydrodynamic and thermal behavior of the flow within the cavity in the presence of this obstacle. The obtained results may have direct applications in industrial processes and technologies such as: furnaces, lubrication technologies, drying technologies and others.

2 Description of the problem

The considered two-dimensional model is illustrated in Figure 1 with boundary conditions and coordinates. The system consists of a double-lid-driven square enclosure. A hot block is placed in the center of the cavity. Moreover, the vertical walls of the cavity are mechanically lid-driven and considered to be at a constant temperature and uniform velocity in the different directions (upward and downward). Besides, the bottom wall is also maintained at a constant temperature but higher than that of vertical walls. The top wall is insulated

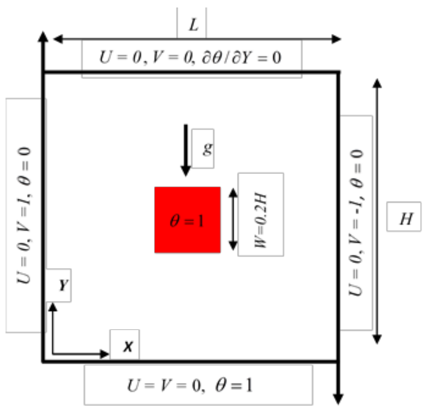

Fig. 1. Schematic of the enclosure.

3 Mathematical formulation

The system is considered to be a two-dimensional, steady-state, laminar and incompressible flow inside the enclosure. Newtonian and Boussinesq approximation is applied to a fluid with constant physical properties. Taking into account the above mentioned assumptions, the non-dimensional governing equations for the continuity, momentum and energy conservation are given as follows:

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$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \quad (1)$$

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

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

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

The local Nusselt number is evaluated along the vertical walls and is given by the following expression:

$$Nu=-\left(\frac{\partial \theta}{\partial X}\right)_{X=0,1} \quad (5)$$

The average Nusselt number is determined from:

$$Nu_{avg}=\int_0^1 Nu\cdot dY \quad (6)$$

4 Numerical modeling

Numerical solution of the governing equations (equations of continuity. momentum and energy) associated with the boundary conditions is done by a finite volume method proposed by Patankar [6] on a staggered grid. The SIMPLER algorithm is used to solve the pressure-velocity coupling. The grid size 300 x 300 is chosen for all of further computations.

5 Validation of the computer code

The validation of the present code is performed by comparing the local Nusselt number values obtained by the present code and those obtained by Chamkha and Abu-nada [7]. The comparison is represented in Table 1. As can be seen from Table 1 there is a good agreement for average Nusselt numbers obtained in the present study when compared to those of [7].

Table 1. The comparison of the local Nusselt number values with Chamkha and Abu-nada [7] at Ri = 0.001

Position Y	Chamkha and Abu-nada [7] Nu	This study Nu
0.0	24.615	24.634
0.2	10.612	10.620
0.4	5.529	5.533
0.6	3.513	3.516
0.8	2.517	2.528
1.0	0.679	0.684

6 Results and discussion

In this section, the different results concerning the effect of the Richardson number on the hydrodynamic and thermal structure of the flow under consideration are presented.

6.1. Thermal convection mode:

Figure 2 (a) shows the influence of the Richardson number on the flow structure (left current lines) and heat transfer (right isotherms) for a Reynolds number Re = 100.

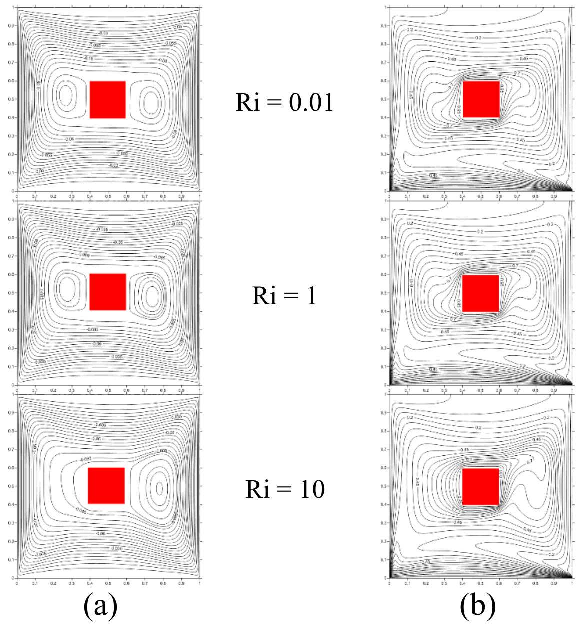

Fig. 2. Streamlines (a) and isotherms lines (b) at Re = 100 and Pr = 7.01

For low values of the Richardson number (Ri ≼ 1, dominant forced convection), the streamlines are very intense near the vertical walls indicating a very high fluid flow velocity in these zones. We also note the presence of a large recirculation cell that encompasses the obstacle and occupies the entire space of the cavity (turning clockwise). On both sides of the obstacle appear two other rotation cells. These secondary cells are rotated by the translational movement of the vertical walls.

For Ri = 1 (mixed convection), the buoyancy effect is of relatively comparable magnitude of the shear effect due to the translational movement of vertical walls. It is illustrated that the secondary cell near the right corner of the bloc is bigger and larger than the one in the left corner. The combination of the movement of the right wall of the cavity and buoyancy forces intensifies the secondary cell located on the right side of the obstacle. The bloc being hot, it will stop the descent of the fluid on the left side of the obstacle which reduces the size of the secondary cell in this zone.

For large values of Ri (Ri = 10, dominated natural convection), the buoyancy effect is dominant. Along the right wall of the cavity, the downward movement of this wall and that induced by natural convection are cooperating and move fluid in a downward movement with very high intensity. As this descent, the fluid becomes cold and arrives at the bottom of the cavity. It encounters the hot obstacle which causes its progressive warming provoking the increasing of the fluid masses because of the high intensity of the movement.

The visualization of the isotherms for different values of the Richardson number is shown in Figure 2(b). It is indicated from this figure that for the low values of the Richardson number (Ri = 0.01), isotherms tend to become stratified near the cold walls. It reflects a heat transfer by conduction. On the other hand, by increasing the intensity of the buoyancy forces (Ri = 10) a convective regime takes place in the regions situated between the hot blockage and the right wall of the enclosure. We observe that the thickness of the thermal limit layer decreases as the Richardson number increases.

6.1. Effect of Richardson number on the heat transfer:

To evaluate the heat transfer in the cavity, Figure 3 shows the variation of the average Nusselt number as function of the Richardson number at Re = 100. As it is obvious in this diagram, the heat transfer is enhanced at high values of the Richardson number.

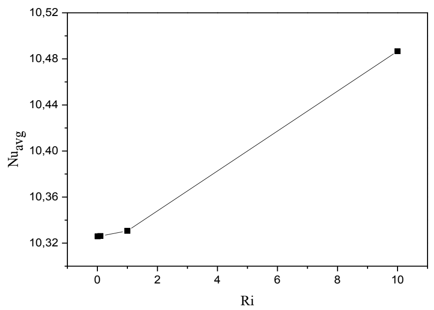

Fig. 3. Average Nusselt number versus Richardson number at Re = 100

Figure 4 delineates the variations of the vertical velocity component at the horizontal mid plane (Y = 0.5) along the bottom wall for various values of the Reynolds number are shown at Ri = 1. It is observed that the vertical velocity varies between two values 1 and -1 because of the no slip condition at the ascending left and descending right vertical walls. respectively. All the profiles have a bearing in the middle of the cavity. The velocity in this zone takes zero value. This zone corresponds to the location of the obstacle. The decrease of the Reynolds number induces the intensification of the flow between the ascending left wall and the left wall of the obstacle (movement of ascending wall is opposite to that of buoyancy forces). The variation of the Reynolds number has almost no effect at the right side of the cavity.

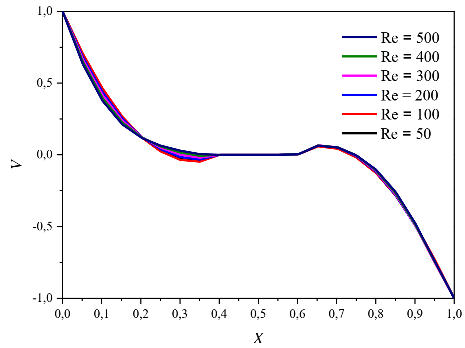

Fig. 4. Variations of the vertical velocity component at the horizontal mid plane (Y = 0.5) along the bottom wall for various values of the Reynolds number at Ri = 1.

6.2. Effect of Reynolds number on the heat transfer:

The effect of Reynolds number on the average Nusselt number inside the cavity is shown in Figure 5 for Ri = 1. From this figure, it is observed that the average Nusselt number increases with increasing the Reynolds number. Consequently, when the Reynolds number increases because of the intensification of the inertia forces (due to the movement of the vertical walls), it improves the heat exchange within the enclosure. This intensification is more important for low values of the Reynolds number.

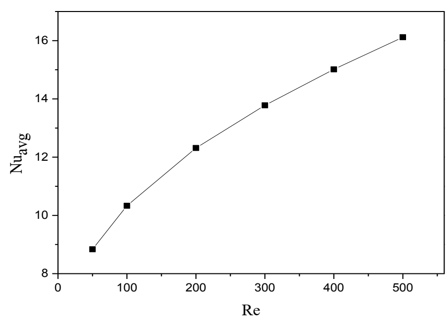

Fig. 5. Variation of the average Nusselt number according to the Reynolds number for Ri = 1.

7 Conclusion

A numerical study is performed to investigate the laminar mixed convection flow in a double-lid driven enclosure with built-in square heated block. The following conclusions may be drawn from the present investigations:

-The transition from one mode of convection to another modifies the structure of the flow and the heat transfer because of the competition between the inertial forces and the buoyancy forces.

-The intensification of the buoyancy forces characterized by the Richardson number and the inertia forces described by the Reynolds number are related to improve heat exchanges inside the cavity.

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