NATURAL CONVECTION WITHIN TRAPEZOIDAL CAVITY WITH TWO BAFFLES ON THE LOWER HORIZONTAL SURFACE

Adriano da Silva, adrianodasilva.ufsj@gmail.com
Universidade Federal de São João Del-Rei - UFSJ

Éliton Fontana, eliton_fontana@hotmail.com
Universidade Federal de Santa Catarina - UFSC

Francisco Marcondes, marcondes@ufc.br
Universidade Federal do Ceará - UFC

Viviana Cocco Mariani, viviana.mariani@pucpr.br
Pontifícia Universidade Católica do Paraná – PUCPR
Universidade Federal do Paraná – UFPR

Abstract. *This paper presents an investigation into natural convection in trapezoidal cavities with two baffles attached to the cavity's plane horizontal surface. It examines a cavity whose floor and upper inclined walls are both adiabatic while the vertical walls are isothermal. The right tall vertical wall is heated while the left short vertical wall is cooled (buoyancy opposing mode along the upper inclined surface of the cavity). The main differences of the present paper with relation the previous work in literature refer to the baffle position inside the cavity. Considering laminar condition and a two-dimensional system, steady state, and Boussinesq approximation computations are carried out to assess the effects of the baffle's height (H_b), Rayleigh number, $10^3 \leq Ra \leq 10^6$, and three Prandtl number values. To demonstrate the various effects, the results from several designed case studies are shown in terms of isotherms, streamlines, and local and average Nusselt numbers in order. Predictions reveal that the second baffle decreases the cavity's fluid flow and heat transfer. As the height of the baffle rises, the heat transfer drops drastically.*

Keywords: *trapezoidal cavity, natural convection, baffles, Nusselt number*

1. INTRODUCTION

The study of natural convection heat transfer has been and continues to be an area of interest of researchers from the standpoint of fundamental and applied research. The natural convection is applied in solar collectors, environmental engineering and electronic packaging. Many studies considering natural convection on plates, channels and enclosures with heat walls have been performed (De Vahl Davis, 1983; Oosthuizen, 2000). After investigations into square, rectangular, and triangular patterns, researchers began studying trapezoidal shapes. Iyican *et al.* (1980a, 1980b) investigate natural convection in an inclined trapezoidal cavity. Their cavity comprises a cylindrical cold surface parallel to a hot horizontal surface, and plane adiabatic sidewalls. Lam *et al.* (1989) report similar results for a trapezoidal cavity comprising two vertical adiabatic walls, a hot horizontal surface, and an upper inclined cold surface. Karyakin [4] shows transient results for natural convection in an isosceles trapezoidal cavity, where a single circulation region is found after reaching a steady-state regime. Karyakin (1989) found that the heat transfer rate increases with the wall's inclination angle. Kuyper and Hoogendoorn (1995) investigate the influence of the inclination angle and Rayleigh number on flow and average Nusselt number in trapezoidal enclosures.

Researchers, in recent years, have considered how heat transfer in cavities is affected when obstacles and fins are attached to the walls. Frederick (1989) investigates natural convection in inclined square cavities with a diathermal partition on the cold wall. Frederick observed heat transfer reductions of up to 45% greater than partition-less cavities. Scozia and Frederick (1991) report natural convection in a differentially heated rectangular cavity with multiple conducting fins on the cold wall. Facas (1993) reports numerical results for natural convection in a non-rectangular inclined cavity with baffles attached along the heated and cooled vertical walls at heights of 0.1W, 0.3W, and 0.5W,W being the cavity's width). Nag *et al.* (1993) study the effect of a horizontal thin partition placed on the hot wall of a differentially heated square cavity.

Tasnim and Collins (2004) attached a highly conductive thin baffle to a hot wall of a differentially heated square cavity. They concluded that the baffle increases the rate of heat transfer by as much as 31%. Boussaid et al. (2003) investigate heat transfer within a trapezoidal cavity heated at the bottom and cooled at the upper inclined surface. Moukalled and Acharya (1997, 2000) investigate natural convection in a trapezoidal cavity with partial dividers attached to the lower horizontal base or to the upper inclined surface of the cavity. Moukalled and Darwish (2003) study the natural convection in a partitioned trapezoidal cavity differentially heated from the sides. They found that heat transfer decreases by increasing the Prandtl number and height of the baffle. Moukalled and Acharya (1997, 2000, 2001) observed natural convection heat transfer in a trapezoidal cavity with partial dividers attached to the lower horizontal base, to the upper inclined surface of the cavity, or to both surfaces, respectively. Moukalled and Darwish

(2003) studied natural convection in a partitioned trapezoidal cavity with one baffle attached to the lower horizontal base. They found that heat transfer is decreased by increasing the Prandtl number and height of the baffle. Moukalled and Darwish (2004) investigate the natural convection in trapezoidal cavities with the baffle attached to the upper inclined surface. Moukalled and Darwish (2007) investigate the natural convection in trapezoidal cavities with two baffles, where one baffle was attached to the upper inclined surface and the other one was attached to the lower horizontal base.

This paper, also investigating natural convection in trapezoidal cavities, analyzes the effects of placing two baffles on the cavity's plane horizontal surface. However the main difference of the results presented in this paper in relation to previous papers refers to position of the baffles. As mentioned above most of the works used one baffle attached to the lower horizontal base or to the inclined upper surface. When two baffles were used in the investigations one baffle was attached to the upper inclined surface and the other one was attached to the horizontal surface. In the present investigation two baffles are attached to the lower horizontal base. Thus, to the best of our knowledge, such a placement has not yet been studied. We analyze in detail how the number and height (H_b) of adiabatic baffles (of finite thickness, $W_b = L/20$), affect heat transfer. We consider for air as the working fluid a range of Rayleigh numbers ($10^3 \le Ra \le 10^6$), and three Prandtl number values. The upper inclined and lower horizontal walls are insulated. With a constant temperature, the left and right vertical walls are alternately heated and cooled (uniformly). It was employed the Element based Finite-volume method to solve the nonlinear, coupled, partial differential equations for fluid flow and temperature fields. The results are shown in terms of isotherms, streamlines, and local and average Nusselt numbers.

2. PHYSICAL AND MATHEMATICAL MODEL

The physical system sketched in Fig. 1 consists of air confined to a two-dimensional trapezoidal cavity with two baffles in which the width of the cavity (L) is 4 times the height (H) of the shortest vertical wall. The inclination of the upper wall of the cavity is fixed at 15°. Baffles are placed at three heights ($H_{b1} = H_{b2} = H^*/3$, $2H^*/3$, and H^*), where H^* denotes the height of the cavity where the baffles are located. Two baffles with thickness ($W_b = L/20$) located in $L_{b1} = L/3$ and $L_{b2} = 2L/3$ are considered.

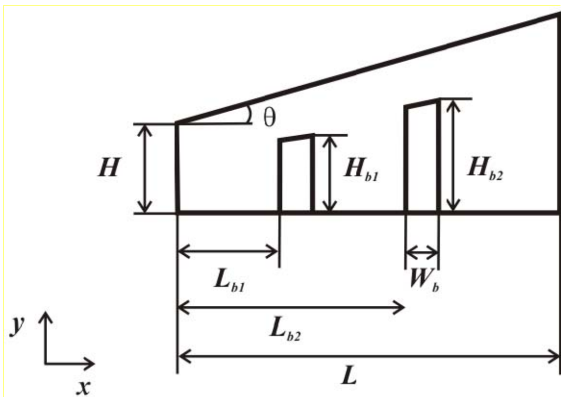

Figure 1. Computational domain for two baffles.

The buoyancy-driven air flow is conceived as two-dimensional and laminar in which the gravitational acceleration acts perpendicular to the insulated horizontal walls. The thermophysical properties of air are taken as temperature-invariant, except in the buoyancy force term where the Boussinesq approximation is applied. Accordingly, the flow and temperature fields are described by a system of conservation equations in Cartesian coordinates in dimension form such as

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0\,, \tag{1}$$

$$\frac{\partial (uu)}{\partial x} + \frac{\partial (vu)}{\partial y} = -\frac{1}{\rho}\frac{\partial p}{\partial x} + \frac{\partial}{\partial x}\left(\nu \frac{\partial u}{\partial x}\right) + \frac{\partial}{\partial y}\left(\nu \frac{\partial u}{\partial y}\right), \tag{2}$$

$$\frac{\partial (uv)}{\partial x} + \frac{\partial (vv)}{\partial y} = -\frac{1}{\rho}\frac{\partial p}{\partial y} + \frac{\partial}{\partial x}\left(\nu \frac{\partial v}{\partial x}\right) + \frac{\partial}{\partial y}\left(\nu \frac{\partial v}{\partial y}\right) + \beta (T_0 - T) g\,, \tag{3}$$

$$\frac{\partial(uT)}{\partial x}+\frac{\partial(vT)}{\partial y}=\frac{\partial}{\partial x}\left(\alpha\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(\alpha\frac{\partial T}{\partial y}\right). \tag{4}$$

where u (m/s) is the velocity in x-direction, v (m/s) is the velocity in y-direction, ρ (kg/m^3) is fluid density, ν (m^2/s) is kinematic viscosity, α (m^2/s) is thermal diffusivity ($\alpha = k/\rho c_p$), β (1/K) is the thermal expansion coefficient of air, T_0 (K) is the reference temperature, T (K) is temperature, and g (m/s^2) is gravitational acceleration.

Along the vertical wall, we consider two sets of boundary conditions. In the first, the following Dirichlet conditions are used:

$$T\,(x=0,y)=T_C, \tag{5}$$
$$T\,(x=L,y)=T_H. \tag{6}$$

In the second, Eqs. (5) and (6) are interchanged. That is, $T\,(x = 0, y) = T_H$ and $T\,(x = L, y) = T_C$. As shown in Eq. (7) through (10), we assume no-slip velocities on all the walls:

$$u(x=0,y)=v(x=0,y)=0, \tag{7}$$
$$u(x=L,y)=v(x=L,y)=0, \tag{8}$$
$$u(x,y=0)=v(x,y=0)=0, \tag{9}$$
$$u(x,y=H+xtg(\theta))=v(x,y=H+xtg(\theta))=0. \tag{10}$$

The bottom and top walls remain insulated.

$$\left.\frac{\partial T}{\partial y}\right)_{y=0}=0, \tag{11}$$
$$\left.\frac{\partial T}{\partial y}\right)_{y=H+xtg(\theta)}=0 \tag{12}$$

The energy balance at the baffle-air interface can be stated as:

$$-\frac{1}{Pr}(\hat{n}.\vec{\nabla}\theta)_i=-\frac{k_r}{Pr}(n.\vec{\nabla}\theta_b)_i, \tag{13}$$

where $\hat{n}$ is a unit vector normal to the baffle-air interface, the subscript i refers to the interface, and k_r is the ratio between the thermal conductivity of the baffle and the convective fluid. The Rayleigh number, for all results shown, is based on the shortest length of the vertical wall. Therefore the Rayleigh number is defined by:

$$Ra=g\beta(T_H-T_C)H^3/\nu\alpha. \tag{14}$$

The local and average Nusselt numbers along the hot and cold walls are defined by

$$Nu_y=-\left[\left.\frac{\partial T}{\partial x}\right)_{x=0,x=L}\right]\Big/(T_H-T_C) \tag{15}$$
$$\overline{Nu}=\frac{1}{H*}\int_o^L Nu_y\,dy \tag{16}$$

where H* denotes the height of either the hot or cold wall.

3. NUMERICAL PROCEDURE

The numerical procedure used to solve the governing equations for the present work is based on the finite volume technique suggested by Patankar (1980). However, in the commercial computational code (Ansys CFX, version 11.0) the conservation equations for mass, momentum are solved together using an element based finite volume method. The flow field is discretized into cells forming a staggered grid arrangement. The resulting discrete system of linear equations is solved using an algebraic multigrid methodology called the additive correction multigrid method. Global convergence is achieved when the sum of absolute normalized residual values of the different equations is sufficiently low (typically 10^{-6}).This convergence criterion is done in order to assure good convergence solutions.

Four different grid configurations (30×30, 60×60, 120×120, and 240×240) were used. Since the differences between the results obtained with 60×60, 120×120, and 240×240 grids were minor, we chose the 60×60 non-uniform grid for all the simulations presented in this work.

3.1 Numerical Validation

For the purpose of validation, the problem of natural convection of air in a trapezoidal cavity with one baffle inside has been solved for various Rayleigh numbers (based on the height). This problem has been reported previously by Moukalled and Darwish (2003). In Tabs. 1 and 2 present a comparison between average Nusselt numbers for $L_b = L/3$ and $2L/3$, respectively, and $H_b = H^*/3$, $2H^*/3$ and 0, and $Pr = 0.7$ for a non-uniform grid of 60×60 and numerical results of Moukalled and Darwish (2003). The results obtained in this study show good agreement with literature. Note that average Nusselt numbers increase with increasing of Rayleigh numbers and with decreasing baffle height.

Table 1. Comparison of average Nusselt numbers ($\overline{Nu}$) obtained for the buoyancy-opposing boundary condition along the cold left and hot right walls for $Pr = 0.7$ and $L_b = L/3$ with solution of Moukalled and Darwish (2003).

Ra	Moukalled and Darwish (2003)			Present study		
	$H_b = H^*/3$	$H_b = 2H^*/3$	$H_b = 0$	$H_b = H^*/3$	$H_b = 2H^*/3$	$H_b = 0$
10^3	0.5080	0.4790	0.6153	0.5030	0.4829	0.6175
10^4	1.4750	0.9880	1.9220	1.4517	1.0140	1.9223
10^5	3.6780	2.1456	4.4310	3.4042	2.3144	4.3409

Table 2. Comparison of average Nusselt numbers ($\overline{Nu}$) obtained for the buoyancy-opposing boundary condition along the cold left and hot right walls for $Pr = 0.7$ and $L_b = 2L/3$ with solution of Moukalled and Darwish (2003).

Ra	Moukalled and Darwish (2003)			Present study		
	$H_b = H^*/3$	$H_b = 2H^*/3$	$H_b = 0$	$H_b = H^*/3$	$H_b = 2H^*/3$	$H_b = 0$
10^3	0.5040	0.4640	0.6153	0.5037	0.4710	0.6175
10^4	1.5510	1.0720	1.9220	1.5619	1.1080	1.9223
10^5	3.5640	2.2940	4.4310	3.6236	2.4361	4.3409

4. RESULTS AND DISCUSSION

After validating the numerical results with the numerical results reported in literature, a wide range of relevant parameters such as Rayleigh numbers and baffle's height are analyzed in this study. Both baffle's height of $H^*/3$, $2H^*/3$, and H^*, Rayleigh number between 10^3 and 10^5, and Prandtl numbers of 0.7, 10 and 130 were chosen.

4.1 Streamlines and isotherms

Figs. 2-4 illustrate the stream function contours of the numerical results for various $Ra = 10^3 – 10^5$ and $Pr = 0.7$, 10 and 130 when the right vertical wall is heated while the left vertical wall is cooled. Fig. 2 shows the typical pattern of streamlines for different Rayleigh and Prandtl numbers for $H_{b1} = H_{b2} = H^*/3$. Thus the streamlines circulate in a counterclockwise direction, mainly owing to the presence of hot and cold walls in the right and left sides, respectively, showing a similar pattern irrespective of the Prandtl number used. It possesses internal single cells between the baffles and vertical walls. For $Ra = 10^5$, the two vortices between the baffles and close to the hot wall merge into one. Further increasing the Rayleigh joins the three internal vortices into a single cell.

In order to quantify the effect of Rayleigh and Prandtl numbers is presented in Fig. 3 the streamlines for $H_{b1} = H_{b2} = H^*/3$, for Rayleigh number in the range $10^3 \le Ra \le 10^6$ and $Pr = 0.7$, 10 and 130. Fixing the Rayleigh number the Prandtl number has no effect on flow as can be seen in Fig. 3. On the other hand fixing the Prandtl number and changing the Rayleigh number it is possible note that the pattern of the flow is drastically modified, observe that the pattern of streamlines is similar to each other, but the resistances in the flow for the first configuration, presented in Fig. 2, are amplified by the baffle's height that divide the cavity into three sub-cavities.

Fig. 4 shows the streamlines for Rayleigh number in the range $10^3 \le Ra \le 10^6$, $Pr = 0.7$, 10 and 130 for for $H_{b1} = H_{b2} = H^*$. Note that the pattern of the flow is not modified by changing the Prandtl numbers, however is modified by changing the Rayleigh numbers. Comparing the streamlines for Ra = 10^4 and H_{b1} = H_{b2} = H^* (Fig. 4) with those obtained for H_{b1} = H_{b2} = $2H^*/3$ (Fig. 3) is possible to note that the pattern of isotherms and streamlines is similar to each other, but the resistances in the flow for the first configuration (Fig. 4) are amplified by the baffles that divide the cavity into three sub-cavities.

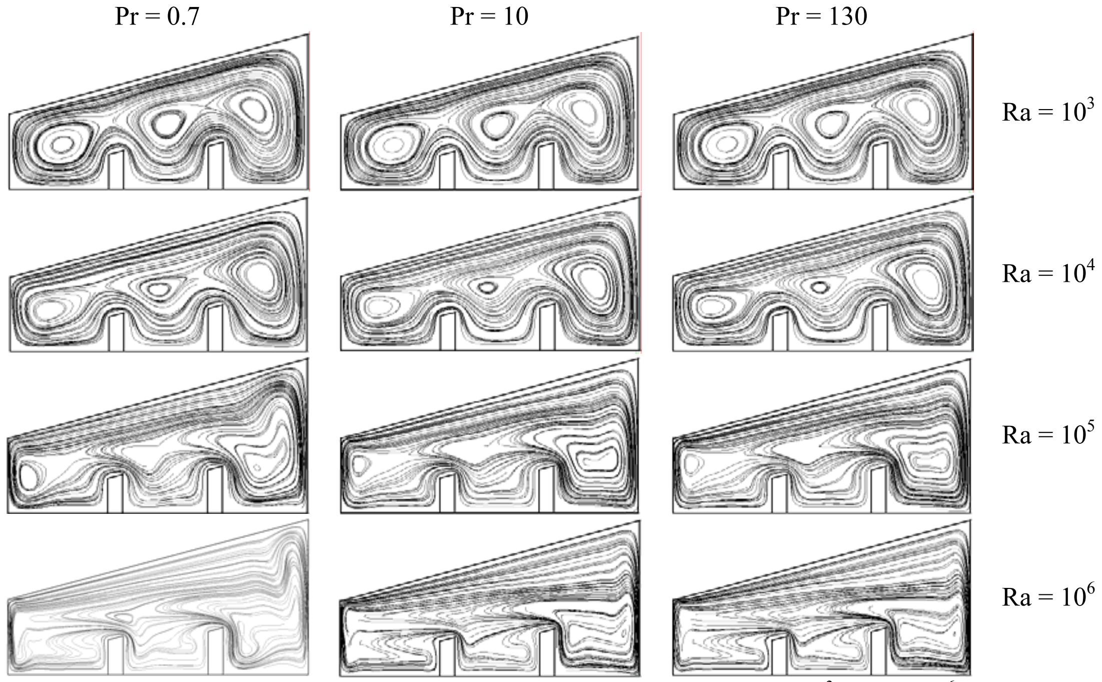

Figure 2. Streamlines (H_{b1} = H_{b2} = $H^*/3$): $0.7 \leq Pr \leq 130$ and $10^3 \leq Ra \leq 10^6$.

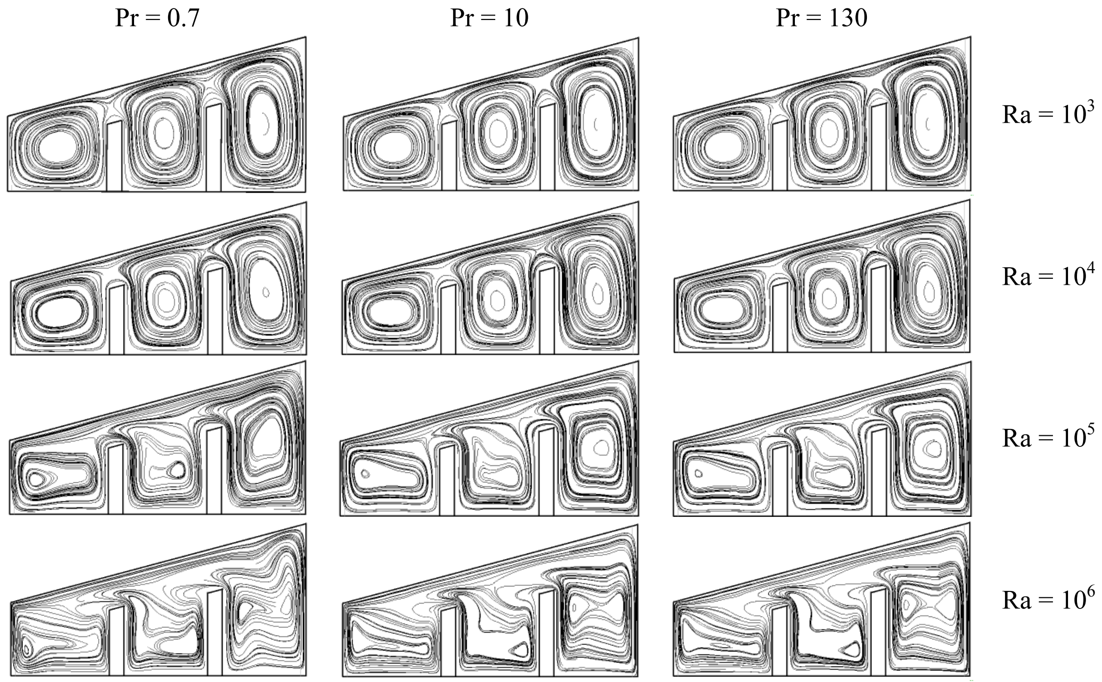

Figure 3. Streamlines (H_{b1} = H_{b2} = $2H^*/3$): $0.7 \leq Pr \leq 130$ and $10^3 \leq Ra \leq 10^6$.

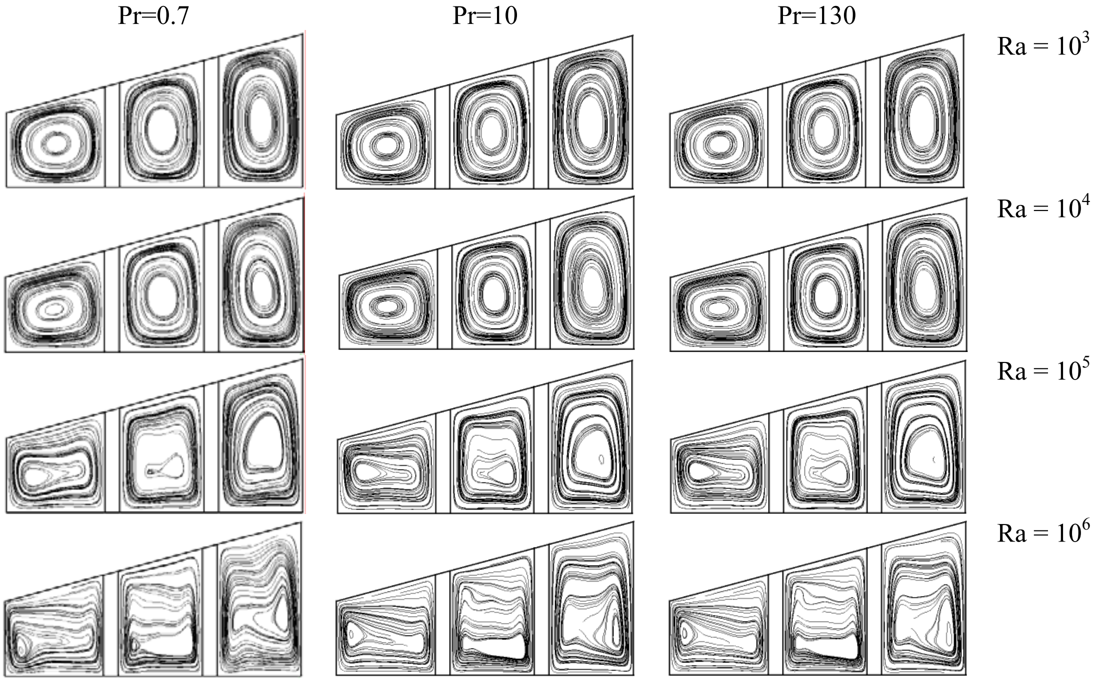

Figure 4. Streamlines ($H_{b1} = H_{b2} = H^*$): $0.7 \leq Pr \leq 130$ and $10^3 \leq Ra \leq 10^6$.

Figs. 5-7 show the isotherms for buoyancy opposing boundary condition for Rayleigh number in the range $10^3 \leq Ra \leq 10^6$, for different Prandtl numbers and $H_{b1} = H_{b2} = H^*/3$, $2H^*/3$ and H^*, respectively. From these figures is possible to infer that all patterns of isotherms are very similar to each other denoting that Prandtl number has a little effect in the natural convection. This effect was already observed for just one baffle by Moukalled and Darwish (2003). Also, we can see that the patterns of isotherms for $Ra = 10^3$, for all Pr numbers, varies almost linearly inside the cavity, indicating that conduction is dominant. Figs. 5-7 also reveal that the isotherms for low Rayleigh numbers are similar to the natural convection with just one baffle as observed in Moukalled and Darwish (2003).

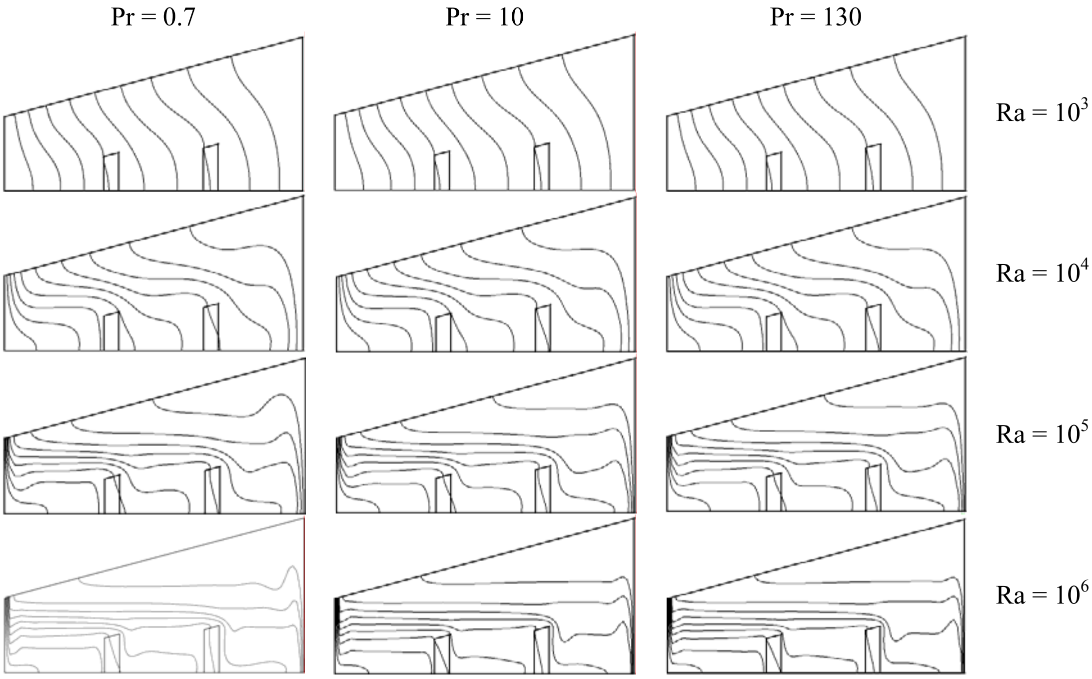

Figure 5. Isotherms ($H_{b1} = H_{b2} = H^*/3$): $0.7 \leq Pr \leq 130$ and $10^3 \leq Ra \leq 10^6$.

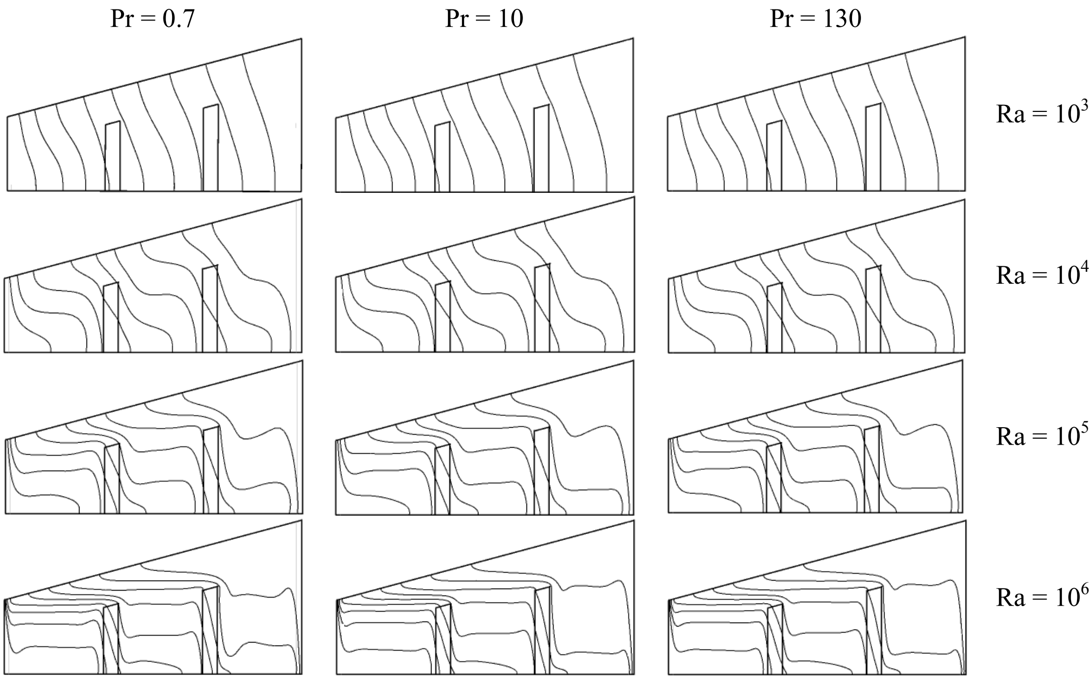

Figure 6. Isotherms ($H_{b1} = H_{b2} = 2H^*/3$): $0.7 \leq Pr \leq 130$ and $10^3 \leq Ra \leq 10^6$.

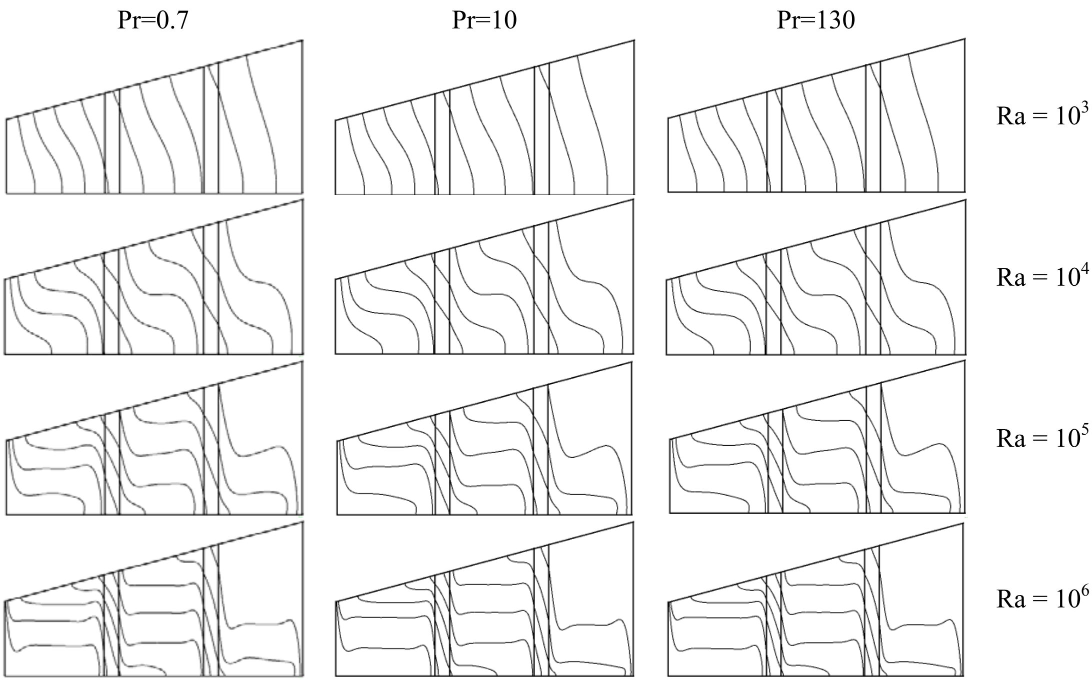

Figure 7. Isotherms ($H_{b1} = H_{b2} = H^*$): $0.7 \leq Pr \leq 130$ and $10^3 \leq Ra \leq 10^6$.

4.2 Local and average Nusselt numbers

In order to quantify the effect of height and number of baffles in the heat transfer, are presented now the local and average Nusselt numbers. Figs. 8 through 10 present the local Nusselt number along the hot and cold wall for Rayleigh number in the range $10^3 \leq Ra \leq 10^6$ and Prandtl number, $Pr = 10$ (because the local Nusselt number almost not change with the change in Prandtl number). As can be seen from these figures for small Rayleigh number the local Nusselt number is almost constant for all height of baffles investigated and Prandtl numbers indicating that the conduction is the dominant mode. Moreover, for every Prandtl number investigated the heat transfer is decreased when the height of the baffle is increased. We explain this by observing that the resistances to the flow and heat transfer are amplified when the height of the baffle is increased. Also, for every baffle height and Prandtl number investigated the heat transfer is increased when the Rayleigh number is increased, thus variation in the local Nusselt number curves is observed. This fact is in perfect agreement with the isotherms and streamlines presented in Figs. 2 through 7.

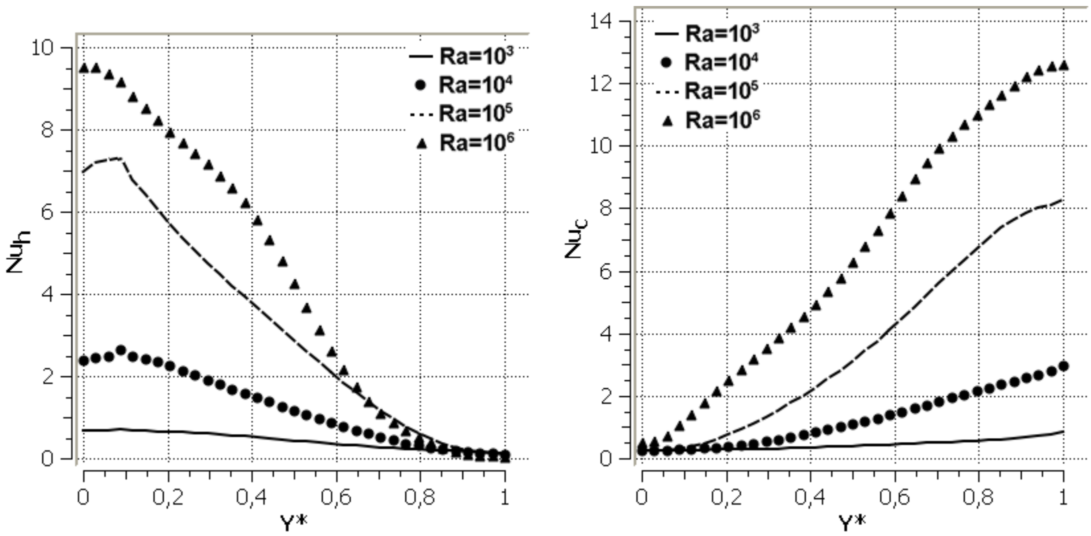

Figure 8. Local Nusselt number for Pr = 10 and H_{b1} = H_{b2} = $H^*/3$ in (a) hot and (b) cold wall.

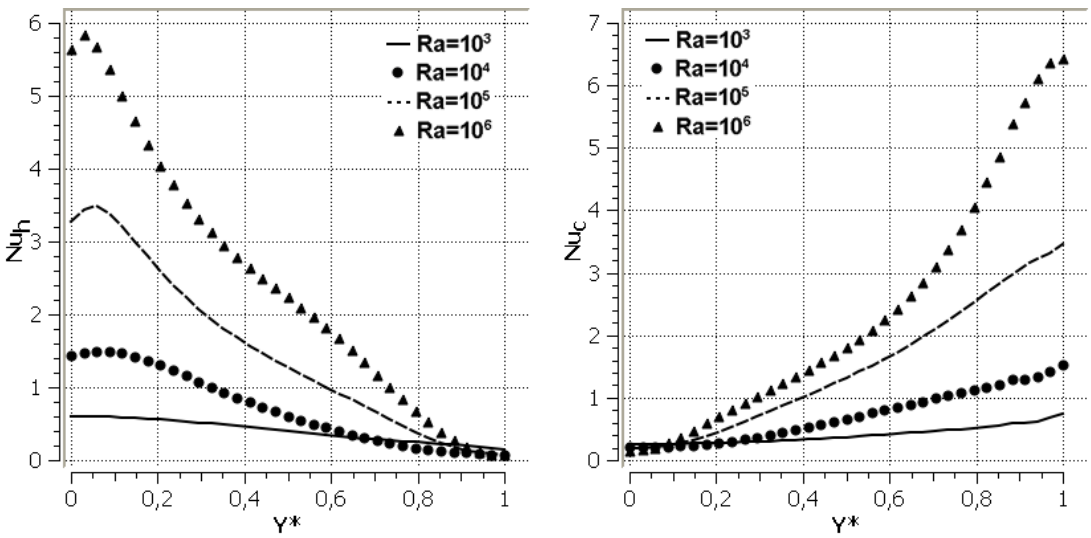

Figure 9. Local Nusselt number for Pr = 10 and H_{b1} = H_{b2} = $2H^*/3$ in (a) hot and (b) cold wall.

The average Nusselt number values for all configurations investigated in this paper and Prandtl numbers (0.7, 10 and 130) are listed in Tabs. 3 through 5, respectively. As previously discussed, it possible to observe from these tables that for a fixed Prandtl and Rayleigh number a significant reduction in heat transfer occurs when the baffle height is increased, main for the biggest Rayleigh numbers. For the smallest Rayleigh number the average Nusselt number tends to a constant value with increasing the baffle height and Prandtl number. For a given baffle height and Prandtl number, the total heat transfer increases with increasing Rayleigh values due to an increase in convection heat transfer.

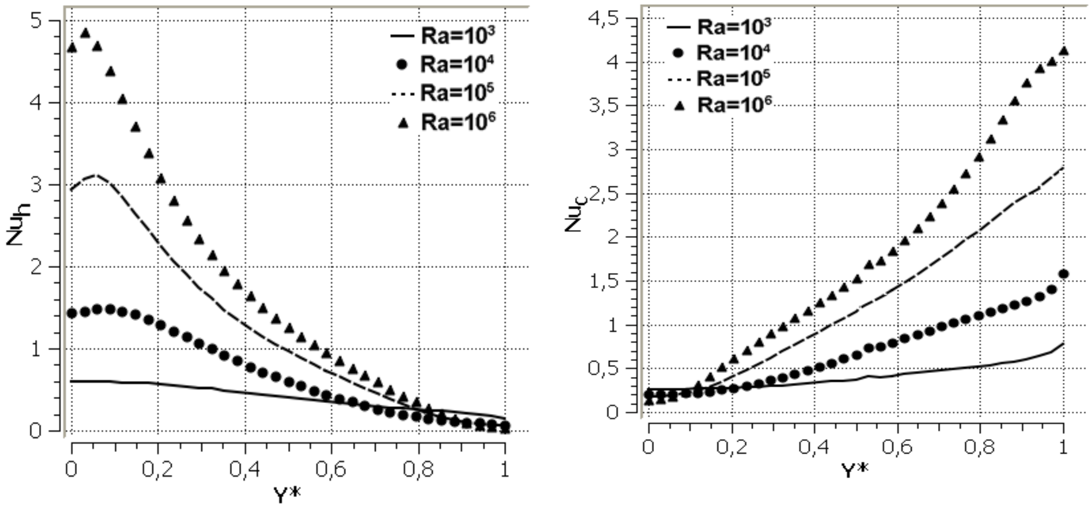

Figure 10. Local Nusselt number for Pr = 10 and $H_{b1} = H_{b2} = H^*$ in (a) hot and (b) cold wall.

Table 3. Average Nusselt number values ($\overline{Nu}$) for Pr = 0.7.

	$H^*/3$	$2H^*/3$	H^*
$Ra = 10^3$	0.4228	0.3937	0.4039
$Ra = 10^4$	1.2433	0.7035	0.6958
$Ra = 10^5$	3,3088	1,4852	1.2428
$Ra = 10^6$	6,2760	2,7402	1.9856

Table 4. Average Nusselt number values ($\overline{Nu}$) for Pr = 10.

	$H^*/3$	$2H^*/3$	H^*
$Ra = 10^3$	0.4227	0.3999	0.4055
$Ra = 10^4$	1.2539	0.7090	0.7020
$Ra = 10^5$	3.4234	1.5021	1.2624
$Ra = 10^6$	6.4913	2.8157	1.9870

Table 5. Average Nusselt number values ($\overline{Nu}$) for Pr = 130.

	$H^*/3$	$2H^*/3$	H^*
$Ra = 10^3$	0.4228	0.3999	0.4055
$Ra = 10^4$	1.2527	0.7090	0.7020
$Ra = 10^5$	3.4211	1.5023	1.2624
$Ra = 10^6$	6.4748	2.8194	1.9874

5. CONCLUSIONS

The present study investigates variations of streamlines, isotherms and local and average Nusselt numbers as a function of different baffle's height, Rayleigh and Prandtl numbers in partitioned trapezoidal cavities with two internal baffles. It is possible to infer that Prandtl number has a little effect in the natural convection. For $Ra = 10^3$ and all Prandtl numbers and baffle's height the isotherms inside the cavity indicate that conduction is dominant. The isotherms for low Rayleigh numbers are similar to the natural convection with just one baffle. For a given baffle height and Prandtl number, the total heat transfer increases with increasing Rayleigh number values due to an increase in convection heat transfer. Increasing the baffle's height the resistances in the flow and heat transfer are amplified by the baffles that divide the cavity into three sub-cavities.

6. ACKNOWLEDGEMENTS

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8. RESPONSIBILITY NOTICE

The authors Adriano da Silva, Éliton Fontana, Francisco Marcondes e Viviana Cocco Mariani are the only responsible for the printed material included in this paper.