A numerical study of natural convection in a cavity with two baffles attached to its vertical walls

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Abstract: -The purpose of this study is to investigate the effect of two insulated horizontal baffles placed at the walls of a differentially heated square cavity. The vertical walls are at different temperatures while the horizontal walls are adiabatic. In our formulation of governing equations, mass, momentum and the energy equations are applied to the cavity and the baffles. To solve the governing differential equations a finite volume code based on Pantenkar's simpler method is utilized. The Results are presented for Rayleigh number from 10^4 up to 10^7 and are in form of streamlines, isotherms as well as Nusselt number. It is observed that the two baffles trap some fluid in the cavity and affect the flow fields. Also, the Nusselt number increases as Rayleigh number increases and decreases with baffle length. Finally it is shown that Nusselt number changes with baffles position.

Key-words: -Natural Convection, Cavity, baffles, Nusselt Number

1 Introduction

The phenomenon of natural convection heat transfer plays an important role, both in nature and in engineering systems. Many investigations have been performed for cavities both theoretically and experimentally for a wide range of Ra (Rayleigh number) [1-2]. Natural convection in an air filled cavity with vertical walls that are heated and cooled while its horizontal walls are adiabatic has received a great consideration because many of the industrial applications employ this concept as a prototype. Noticeable examples include heating and ventilation of rooms, solar collector systems, and electronic cooling devices. In many applications, for some reasons, attaching baffle(s) to its vertical wall or to its horizontal wall(s) partitions the cavity. Recently studies of heat transfer and fluid flow characteristics of partitioned cavity have come under scrutiny both numerically and experimentally [3-11].

Bajorek and Llyod [3] studied experimentally a differentially heated air filled square cavity for various Ra from 1.25×10^5 to 2.16×10^6. The insulated baffle is attached to the horizontal walls at positions in the middle. Non-dimensional baffle length is 0.25. The effect of the baffle positions was not considered. It was found that the baffles significantly influence the heat transfer rate and the average Nusselt numbers reduced to approximately 15% compared to the non-partitioned cavity. Jetli et al [6] studied numerically a differentially heated air filled square partitioned cavity for various Ra from 10^4 to 3.55×10^5 with three different combinations of the baffle positions. Non-dimensional baffles length is fixed at 0.33. The results clearly demonstrate that the baffles positions have a significant effect on the heat transfer and flow characteristics of the fluid. For all baffles locations, the average Nusselt number is smaller than the corresponding value in a cavity without baffle. Ambarita studied numerically a differentially heated square cavity with two perfectly insulated baffles were attached to its horizontal walls at symmetric position [9]. A parametric study was carried out using following parameters: Rayleigh number from 10^4 to 10^8, non-dimensional thin baffles length were 0.6, 0.7, 0.8, non-dimensional baffle position were 0.2 to 0.8.

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2 Problem definition

The typical cavity with the boundaries and its coordinates system are depicted in Fig.1. In this study only the square cavity is considered, H denotes its height and width. The cavity is differentially heated, left and right walls are isothermal at T_h and T_c respectively (T_h>T_c) and horizontal walls are adiabatic.

Two thin baffles with non-dimensional length L_b, perfectly insulated, are attached to the left and right wall. The non-dimensional position of the left baffle from the bottom wall and right baffle from the top wall are the same and denoted by D_b.

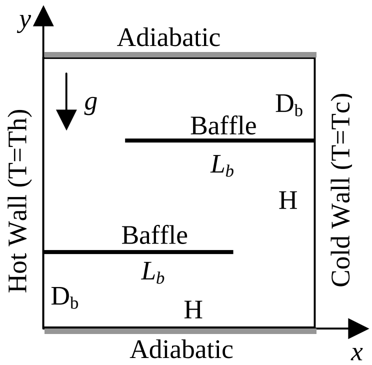

Fig.1 Schematic of the square cavity with thin insulated baffles attached to the vertical walls

The governing equations are converted into the non-dimensional form by defining the non-dimensional variables:

$$X=\frac{x}{H},\ Y=\frac{y}{H},\ U=\frac{uH}{\alpha},\ V=\frac{vH}{\alpha},\ P=\frac{pH^2}{\rho\alpha^2},\ \theta=\frac{T-T_c}{T_h-T_c} \quad (1)$$

Based on these non-dimensional variables, the governing equations are obtained as follows:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \quad (2)$$

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

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial X}+\Pr\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)+Ra\Pr\theta \quad (4)$$

$$U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right) \quad (5)$$

The boundary conditions are:

On the left wall: $U=V=0,\theta=1$ (6)

On the right wall: $U=V=0,\theta=0$ (7)

On the top and bottom walls: $U=V=0,\partial\theta/\partial Y=0$ (8)

On the baffles: $U=V=0,\partial\theta/\partial Y=0$ (9)

In order to compare total heat transfer rate, Nusselt number is used. The local and average Nusselt numbers are defined as follows:

$$Nu_y=-\frac{\left.\partial\theta/\partial X\right|_{X=0}}{(\theta_h-\theta_c)}\ ,\ \ \overline{Nu}=\int_0^1 Nu_y\,dY \quad (10)$$

3 Numerical procedure

The coupled governing equations are transformed into sets of algebraic equations using finite volume method and are solved by the SIMPLER algorithm [15]. The staggered grid system is used and the convective terms are handled by the power law scheme.

3.1 Validation of the numerical code

In order to make sure that the developed codes are free of error coding, a validation test is conducted. Calculations for an air filled cavity without baffle for Ra $=10^4$ to 10^7 are carried out and the results are shown in Table 1. The results of the previous publications for the same problem are also presented in the Table 1. Data from the table shows that the results of the code, even though there are some differences, do agree very well with the previous results. Based on this successful validation, the problem is solved by using the code.

Table 1 Comparison of the present result and the previous result

Reference	Average Nusselt numbers, $\overline{Nu}$			
	Ra=10^4	Ra=10^5	Ra=10^6	Ra=10^7
Hortmann [10]	2.4468	5.5231	8.8359	-
Collins[4]	2.244	4.5236	8.8554	-
Nag[11]	2.24	4.51	8.82	-
Shi[1]	2.247	4.532	8.893	16.935
Bilgen[8]	2.245	4.521	8.8	16.629
Ambarita[9]	2.228	4.514	8.804	16.52
Present	2.25	4.53	8.85	16.43

Flow and temperature fields for the square cavity without baffle are presented in Fig.2. It is observed that as natural convection strengthened, temperature contours show slight deviation from the pure conduction case with the isotherms becoming skewed. Under high Ra conditions, the degree of distortion from the pure conduction case is varying marked and the contour lines become almost horizontal lines around the center of the cavity; Fig.2(a). The rise of the fluid due to heating on the left wall and consequent falling of the fluid on the right wall creates a clockwise rotating vortex, referred to as the primary vortex; Fig.2(b).

Another feature of these streamlines patterns is that the streamlines become more packed next to the sidewall as the Ra increases. This suggests that the flow move faster as natural convection is intensified. The maximum absolute value of stream function can be viewed as a measure of the intensity of natural convection in the cavity. As the Ra increases, the maximum absolute value of the stream function increases.

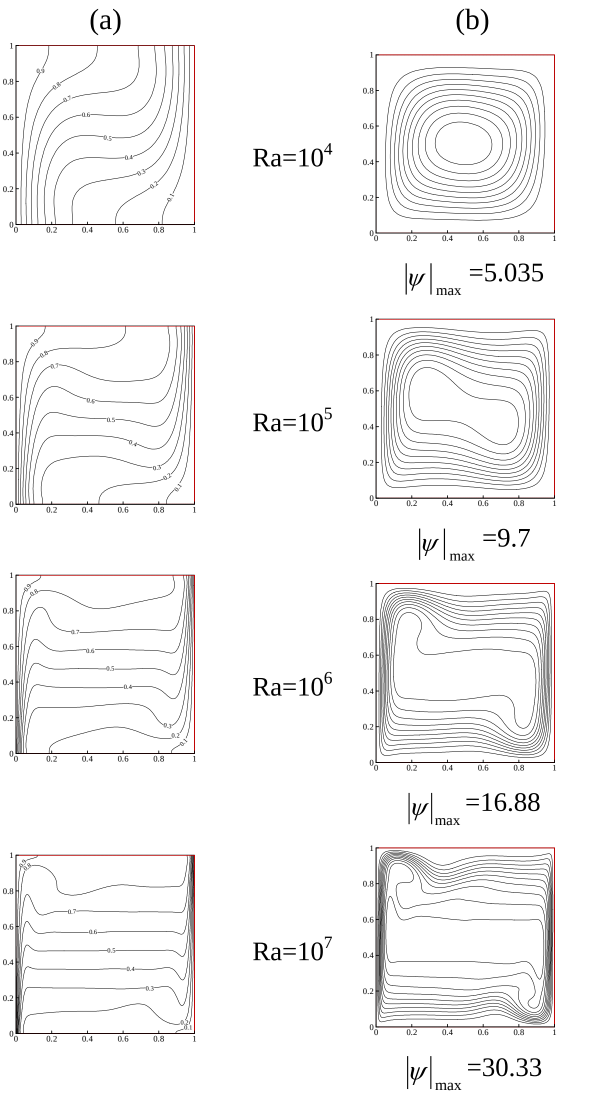

Fig.2 Isotherms (a) and Streamlines (b) of natural convection in a square cavity

4 Results and discussion

In order to understand the flow and temperature fields and heat transfer characteristics of the typical cavity a total of 64 cases are considered. To study the effects of baffle position and baffle length, the non-dimensional baffle positions D_b= 0.2, 0.4, 0.6 and 0.8 and the non-dimensional baffle lengths L_b=0.5, 0.6, 0.7 and 0.8 are considered. The fluid inside the cavity is dry air with Pr = 0.7 and Rayleigh number varied from 10^4 to 10^7.

4.1 Flow and temperature fields

Flow fields for the typical cases with non-dimensional baffle length L_b=0.6 are presented in Fig.3. The plots are arranged going from left to right with the ascending of Ra and from top to bottom with the ascending of D_b.

A glance at Fig.3 shows that the flow fields can be divided into two different patterns. The first pattern is the fluid circulates and creates a large primary vortex strangled by the baffles. The second pattern is fluid separated into two different vortexes. The two baffles create fluid trapping phenomena in the cavity. The first raw of Fig.3 is the flow fields for the case when D_b=0.2. At a low Rayleigh number Ra=10^4 there are two trapped fluids in the cavity, bottom trapped fluid and top trapped fluid. The bottom-trapped fluid exists between bottom baffle and bottom adiabatic wall and top trapped fluid exists between top baffle and top adiabatic wall. This is because at Ra=10^4 natural convection is too weak to make the trapped fluids moving and also the space of trapped fluids is limited due to a small D_b. The natural convection creates a primary vortex strangled by these two trapped fluids. By increasing Ra the primary vortex is strong enough to penetrate the two trapped fluid.

The second raw of Fig.3 is the flow field when D_b=0.4. The figure shows that even for a low Rayleigh number Ra=10^4 the primary vortex divide into two vortexes. This is because the space between each baffle and the nearest horizontal wall is larger compared to the corresponding cases when D_b=0.2.

In order to satisfy continuity, the circulation on the bottom side is separated from topside. At low Ra a weak vortex is formed between two baffles. By increasing Ra these vortexes will be strengthened.

The flow field for case with D_b = 0.6 are shown in the third column of the fig3. At Ra=10^4 there are two vortexes in the cavity and a trapped fluid exists between the two baffles. The fourth raw of Fig.3 is the flow fields for the case when D_b=0.8. In this case by increasing space between two baffles and reducing the space between baffles and adiabatic walls, two separated vortexes will be merged and again primary vortex between baffles and two-trapped fluid will be formed. At a low Ra (Ra=10^4) there are two trapped fluids in the cavity, bottom trapped fluid and top trapped fluid. The bottom-trapped fluid exists between bottom baffle and bottom adiabatic wall and top trapped fluid exists between top baffle and top adiabatic wall. This is because at Ra=10^4 natural convection is too weak to make the trapped fluids moving. By increasing Ra the primary vortex tends to penetrate the trapped fluid and trapped vortex will be strengthened.

Temperature fields when L_b=0.6 are presented in Fig.4.

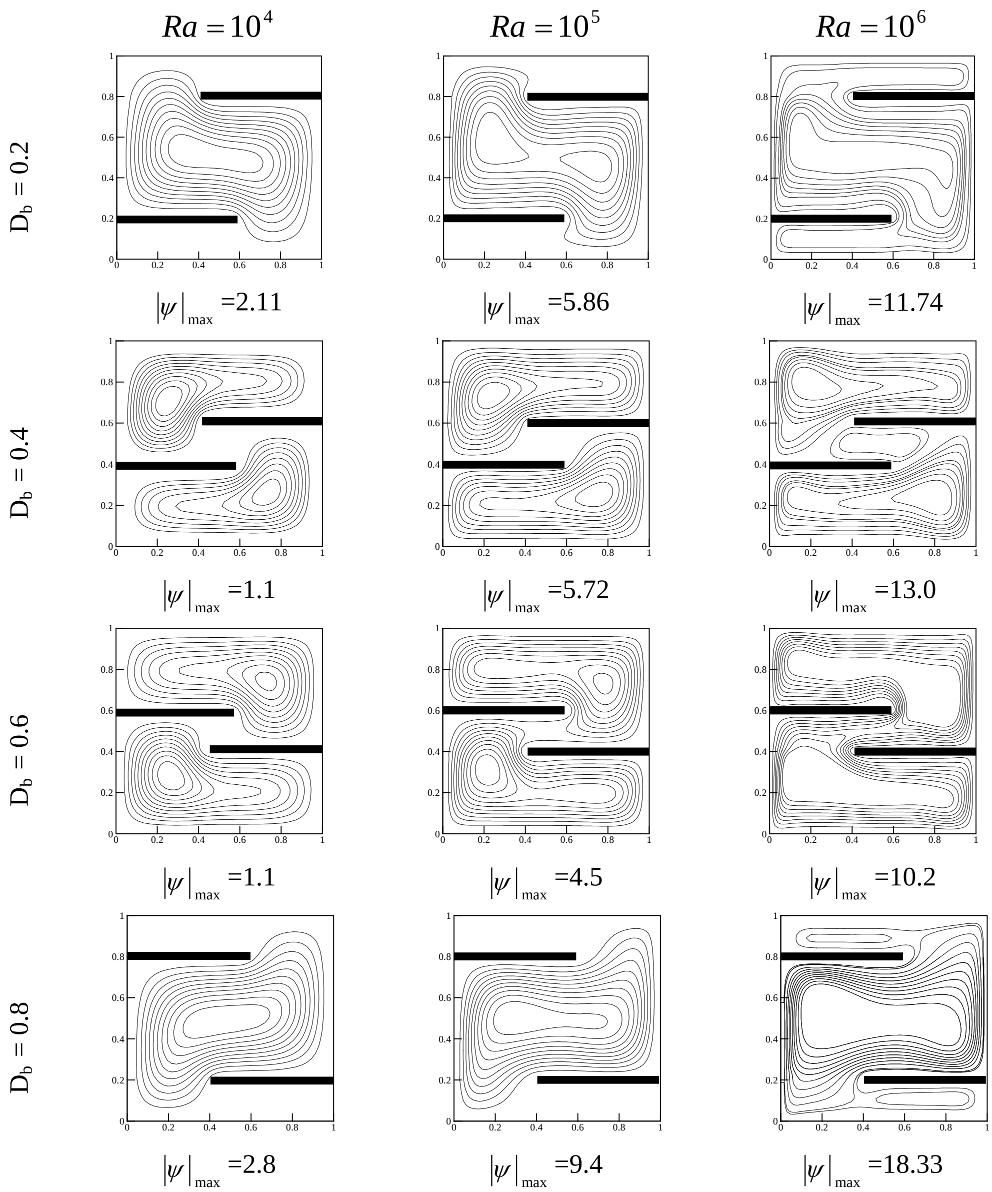

Fig.3 Streamlines for L_b =0.6 and various Ra and D_b

The plots are arranged going from left to right with the ascending of Ra and from top to bottom with the ascending of D_b. Contour level increments for each case are kept constant at 0.1. Indeed in comparison to the square cavity without baffles the appearance of the temperature fields is strongly modified due to the presence of the two insulated baffles.

The first raw of Fig.4 is for D_b = 0.2. At Ra=10^4 when the natural convection is weak and in the region where fluid is trapped the conduction heat transfer only occurred. In the region where natural convection takes place the isotherms show deviations from the pure conduction case with the contour lines becoming skewed. In the trapped fluid areas heat transfer is inactive due to presence of the insulated baffles and stagnant fluids. At Ra=10^5 the isotherms in the top left and bottom right of the cavity become more skewed but almost vertical between the two baffles and insulated walls. This is because natural convection is more vigorous in the top left and bottom right of the cavity but becomes weak between the two baffles and insulated walls. For Ra=10^6, primary vortex is strong enough to penetrate the two trapped fluid then Isotherms in these regions are skewed and convection heat transfer is appearance.

The second raw of Fig.4 is the temperature fields for cases when baffle positions D_b = 0.4. The figure shows that in the regions between baffles and adiabatic walls the isotherms are skewed and conduction heat transfer is less compared to the corresponding cases when D_b=0.2.

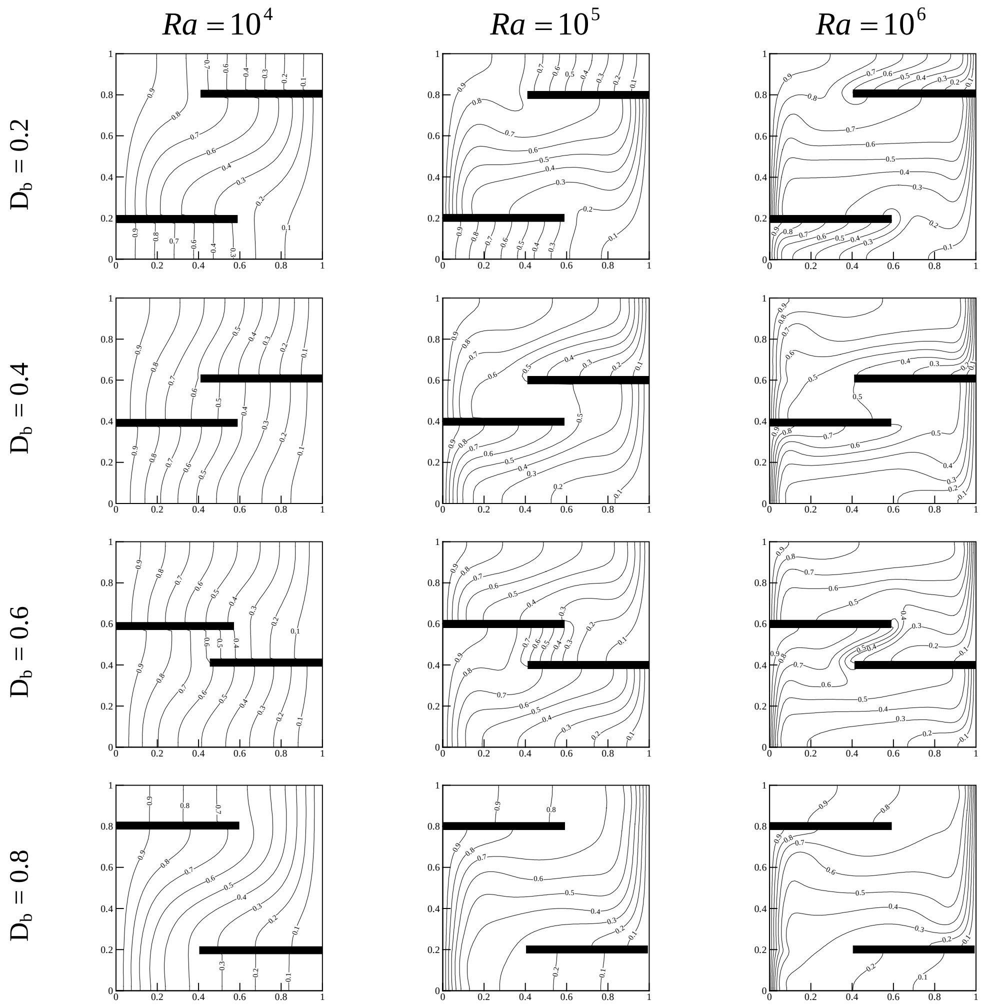

Fig.4 Isotherms for L_b =0.6 and various Ra and D_b

By increasing the Ra the isotherms are more skewed in these regions and convection heat transfer is clearer. The third raw is for case when D_b =0.6. At Ra=10^4 where the flow is still separated into two vortexes and a trapped fluid exists between the two baffles the isotherms are skewed on the vortex areas but almost vertical in the trapped fluid area. At Ra=10^5 the isotherms are more skewed in the vortex areas but less vertical in the trapped fluid area since intensity of the natural Convection increases. At Ra=10^6 two vortexes have strengthened.

Further there is no trapped fluid between baffles. The forth raw of Fig.4 are the temperature fields for cases when the baffle positions D_b = 0.8. The figure shows that the Isotherms lines in the regions between baffles and adiabatic walls are vertical and only conduction heat transfer occurred there. And by increasing the Rayleigh number isotherms are skewed and convection heat transfer is clearer.

4.2 Heat transfer

In order to evaluate how the presence of the two baffles affects the heat transfer rate through the cavity, the average Nusselt number will be discussed. The average Nusselt numbers for all cases as a function of Rayleigh number and for various parameters, L_b=0.5, L_b=0.6, and L_b=0.7, L_b=0.8 are presented in Table 2.

Generally, as expected $\overline{Nu}$ is an increasing function of Ra and a decreasing function of L_b. Also the Nusselt number is baffle position dependent. As the Ra increases the maximum absolute value of the stream function increases. This means the intensity of natural convection in the cavity increases as well as the temperature gradient near the isotherm walls. Table 2 shows the case with L_b =0.6 and For D_b = 0.2, $\overline{Nu}$ is decreased by 32% at Ra=10^4 and is 12% at Ra=10^7, the average is 22%. For D_b =0.4, Nu is decreased by 50% at Ra=10^4 and is 9% at Ra=10^7, the average is 29.5%. These values suggest that the case with D_b =0.4 blocks the heat transfer rate more effectively than the case with D_b =0.2.For D_b=0.6, $\overline{Nu}$ is decreased by 47% at Ra=10^4, 13% at Ra=10^7, the average is 30%. For D_b = 0.8, $\overline{Nu}$ is decreased by 35% at Ra=10^4 and is 14% at Ra=10^7, the average is 24.5%. The results clearly demonstrate that the baffles positions have a significant effect on the heat transfer and flow characteristics of the fluid. For all baffles locations, the average Nusselt number is smaller than the corresponding value in a cavity without baffle.

Table 2 Nusselt number for various Ra, L_b and D_b

	Ra=10^4	Ra=10^5	Ra=10^6	Ra=10^7
D_b=0.2	1.615	3.39	7.12	14.66
D_b=0.4	1.18	3.35	8.12	15.31
D_b=0.6	1.22	3.21	7.8	14.29
D_b=0.8	1.66	3.81	7.62	14.25
L_b=0.5				
	Ra=10^4	Ra=10^5	Ra=10^6	Ra=10^7
D_b=0.2	1.53	3.33	6.96	14.4
D_b=0.4	1.11	3.07	8.04	15.24
D_b=0.6	1.19	3.19	7.63	14.15
D_b=0.8	1.46	3.66	7.37	14.11
L_b=0.6				
	Ra=10^4	Ra=10^5	Ra=10^6	Ra=10^7
D_b=0.2	1.49	3.21	6.8	14.25
D_b=0.4	1.07	2.71	7.9	15.15
D_b=0.6	1.15	2.98	7.4	13.78
D_b=0.8	1.34	3.54	7.26	14.07
L_b=0.7				
	Ra=10^4	Ra=10^5	Ra=10^6	Ra=10^7
D_b=0.2	1.40	3.11	6.64	13.94
D_b=0.4	1.05	2.55	7.7	15.12
D_b=0.6	1.12	2.89	7.3	13.5
D_b=0.8	1.26	3.06	7.14	14.02
L_b=0.8				

5 Conclusion

Heat transfer by natural convection in a differentially heated square cavity with two thin insulated baffles has been numerically studied. The cavity is performed by vertical isothermal walls and adiabatic horizontal walls. Two thin insulated baffles are attached to its vertical walls at symmetric positions. In order to understand the flow and temperature fields and heat transfer characteristics of the typical cavity a total of 64 cases are considered. To study the effects of the baffle position, the non-dimensional baffle positions D_b= 0.2, 0.4, 0.6 and 0.8 are considered. To study the effects of baffle length, the non-dimensional baffle lengths L_b=0.5, 0.6, 0.7 and 0.8 are considered and Ra varies from 10^4 to 10^7. Two different flow field patterns are observed. The first pattern is flow fields with two different vortexes separated by a trapped fluid between the baffles and the second pattern is flow field with a primary vortex strangled by two trapped fluids. The flow field pattern is non-dimensional baffle positions and Rayleigh numbers dependent. It is also observed that the Nusselt number increases as Rayleigh number increases and decreases with baffle length

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