A Numerical Study of Natural Convection in a Tilted Cavity with Two Baffles Attached to its Isothermal Walls

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Abstract: -The purpose of this study is to investigate the effect of inclination angle on flow field and heat transfer in a differentially heated square cavity with two insulated baffles attached to its isothermal walls. The isothermal walls are at different temperatures. The walls that make an angle φ with the horizontal are adiabatic. In our formulation of governing differential 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 various Rayleigh number in form of streamlines, isotherms as well as Nusselt number. It is observed that for all baffle lengths and baffle positions when $\varphi = 90^o$ the Nusselt number is almost minimum. In addition the Nusselt number decreases as baffle length increases, generally. Finally it is shown that Nusselt number changes with baffles position.

Key-words: Natural Convection, inclined 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 $1.25x10^5$ to $2.16x10^6$. 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 $3.55x10^5$ 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.

Ghassemi et al. [10] studied the effect of two insulated horizontal baffles on flow field and heat transfer in a cavity. It was found that 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 depends on dimensional baffle positions and Rayleigh numbers. It is also observed that the Nusselt number increases as Rayleigh number increases and decreases with baffle length.

The effect of inclination angle on flow field and heat transfer in a tilted cavity with two baffles on isothermal walls has not been reported yet; therefore in this study we investigate this subject.

2 Problem definition

The typical cavity with the boundaries and its coordinates system are depicted in Fig.1. In this study tilted square cavity is considered, φ denotes the inclination angle. The cavity is differentially heated. The isothermal walls are at T_h and T_c (T_h>T_c) and the adiabatic walls making an angle φ with the horizontal.

Two thin baffles with non-dimensional length L_b, perfectly insulated, are attached to the left and right walls. 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.

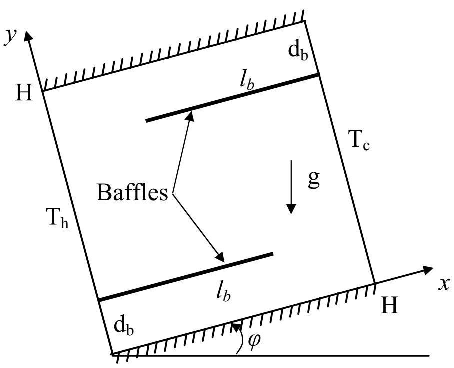

Fig.1 Schematic of the square inclined cavity with thin insulated baffles attached to the isothermal walls

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

$$X=\frac{x}{H}\quad,\quad Y=\frac{y}{H}\quad,\quad L_b=\frac{l_b}{H}\quad,\quad D_b=\frac{d_b}{H}$$

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

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

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

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\Pr\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right)+Ra\Pr\theta g\sin\varphi\qquad(3)$$

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

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

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:

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

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 [11]	2.4468	5.5231	8.8359	-
Collins[4]	2.244	4.5236	8.8554	-
Nag[12]	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.

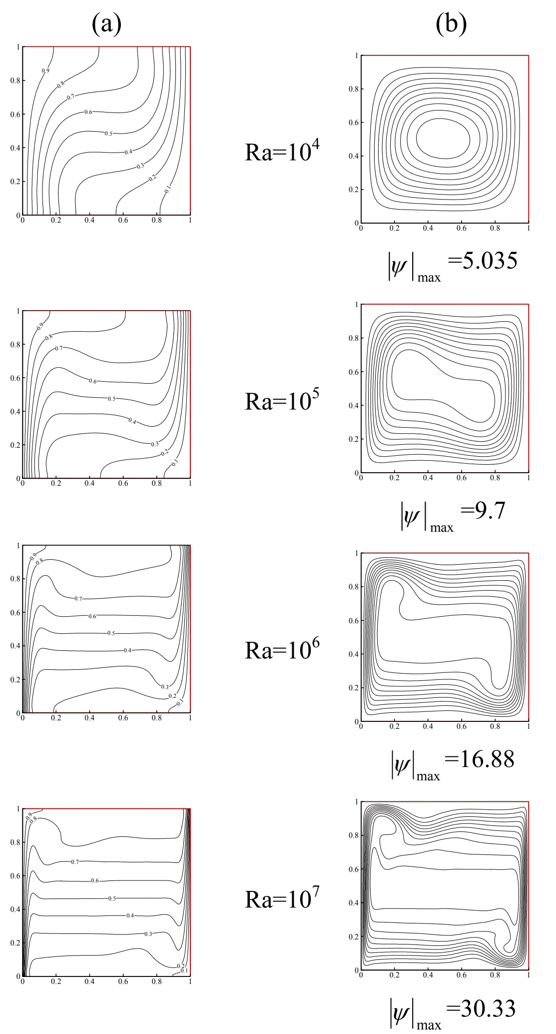

Fig.2 Isotherms (a) and Streamlines (b) of natural convection in a square cavity

4 Results and discussion

In order to understand the flow and temperature fields and heat transfer characteristics of the typical cavity 120 cases are considered. To study the effects of inclination angle, baffle position, baffle length, the inclination angle φ=0^o, 45^o, 90^o, 135^o and 180^o, the non-dimensional baffle positions D_b= 0.2, 0.4 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 10^6.

4.1 Flow and temperature fields

Stream lines and isotherms for the typical cases with non-dimensional baffle length $L_{b=}$0.6 are presented in Fig.3. As shown by Fig.3 for Ra = 10^4, D_b = 0.2 and all angles there are two trapped fluids between each baffles and adiabatic walls. This is because at Ra=10^4 natural convection is too weak to make the trapped fluids moving. In addition the space of trapped fluids is limited due to a small baffle position $D_{b.}$ Between two baffles a weak convection occurs which tends to create a primary vortex that is strangled by trapped fluids. At inclination more than 90^o the direction of air circulation changes from clockwise to counterclockwise.

For the case D_b=0.4, it is observed that two primary vortexes between each baffle and adiabatic wall are formed. It seems that the strength of convection in the region between each baffle and adiabatic wall increases as the dimension of this region increases and it tends to form these vortexes. At inclination angle equal 90^o the direction of air circulation changes from clockwise to counterclockwise.

At Ra=10^5 and all angles, for the case D_b=0.2, in the regions between baffle and adiabatic wall the fluid is trapped and conduction heat transfer occurs toward to the cold wall. At this Rayleigh, formed vortex between two baffles is stronger than the case Ra=10^4 and by increasing the inclination angle, before reaching the 90^o, the direction of air circulation changes. At $\varphi = 90^o$ it is observed that four vortexes form in opposite direction. For D_b=0.4, between baffle and adiabatic wall there is a vortex and by increasing φ, before reaching 90^o, the direction of circulation of vortex changes.

At Ra=10^6 and all angles, for the case D_b=0.2, the strength of primary vortex increases so that it penetrate into the region between baffle and adiabatic wall. By increasing φ, the vortex has tendency to change its direction, so at $\varphi = 90^o$, two circulating cells can be find out and in the angles more than 90^o, the vortexes combine together again.

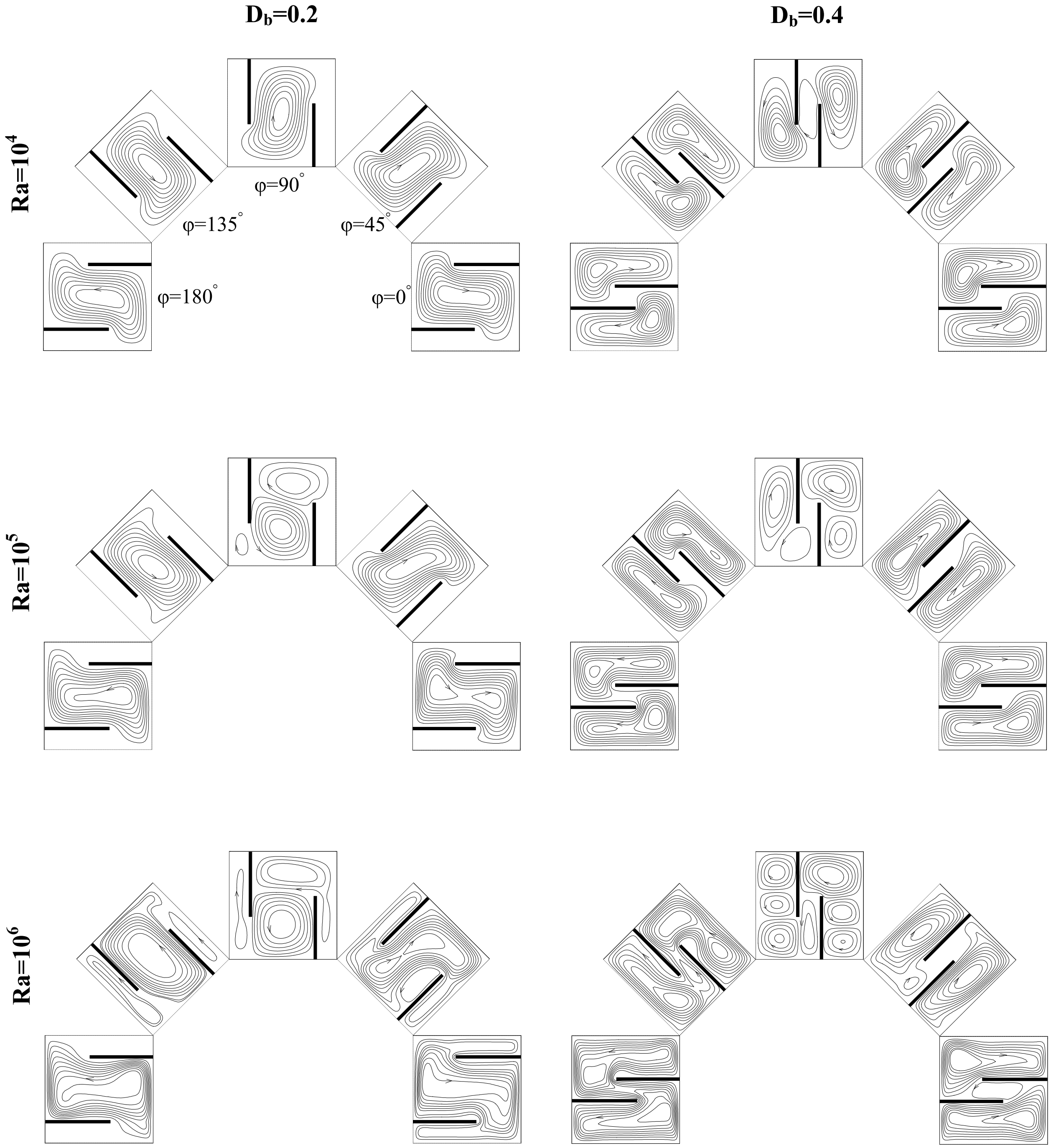

Fig.3 Streamlines in cavity for L_b =0.6 and various Ra, D_b and φ

For the case D_b=.4, the strength of convection increases between baffles and adiabatic walls but it decreases between two baffles. By increasing φ, the vortex has tendency to change its direction. It is observed that at $\varphi = 90^o$ several vortexes form between baffles and adiabatic walls and by increasing φ these vortexes combine together again. Isotherms when L_b=0.6 are presented in Fig.4. Contour level increments for each case are kept constant at 0.1.

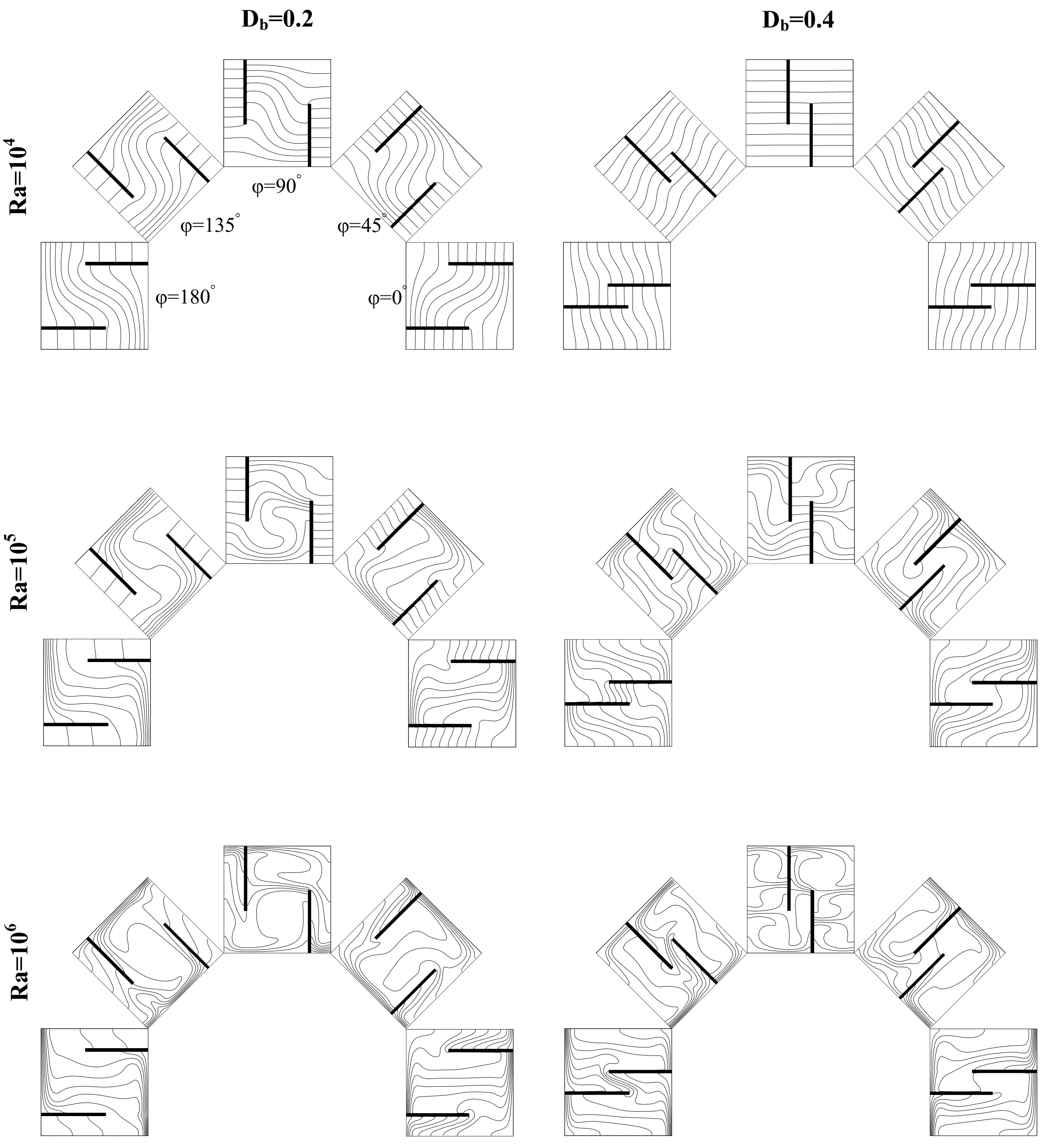

Fig.4 Isotherms in cavity for L_b =0.6 and various Ra, D_b and φ

At Ra=10^4 and all angles, for the case D_b=0.2, the natural convection is weak and in the region where fluid is trapped, heat transfer occurs only by conduction. 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.

For the case D_b = 0.4, the regions between each baffle and adiabatic wall the isotherms are skewed and convection heat transfer is more compared to the corresponding cases when D_b=0.2. In this case, the heat transfer mechanism between the hot and cold walls is nearly by conduction, so that at $\alpha = 90^o$ the isotherms are horizontal and pure conduction occurs.

At Ra=10^5 and all angles, for the case D_b=0.2, the isotherms between baffles become more skewed but almost straight between the baffles and insulated walls. This is because natural convection is more vigorous in the region between baffles and but becomes weak between baffle and insulated wall.

For the case D_b = 0.4, in the regions between each baffle and adiabatic wall the isotherms are more skewed and convection heat transfer is more compared to the corresponding cases when D_b=0.2. It is observed that at $\varphi \geq 90^o$ between two baffles, the heat transfer occurs nearly by conduction.

At Ra=10^6 and all angles the convective heat transfer is dominant in the all region of cavity .it is seen that for the case D_b = 0.2 in the region between each baffle and adiabatic wall the isotherms are skewed.

4.2 Heat transfer

In order to evaluate how the inclination angle and the baffles affect 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 inclination angle, length and position of baffles are presented in figure 5. As it is expected by increasing the Rayleigh number the intensity of natural convection increases that it tends to increase the Nusselt number for all cases. Also it is observed that, generally, the Nusselt number decreases as baffle length increases

At Ra=10^4, the Nusselt number is minimum at $\varphi = 90^o$ for all baffle lengths and for the case D_b=0.4, heat transfer is less than the case D_b=0.2 so that at $\varphi = 90^o$ (D_b=0.4) heat transfer occurs almost by conduction.

For the cases Ra=10^5 and D_b=0.2, It is observed that for all baffle length the minimum and maximum of $\overline{Nu}$ occur at $\varphi = 90^o$ and $\varphi = 180^o$, respectively.

For the cases Ra=10^5 and D_b=0.4, when L_b=0.7, 0.8 and L_b=0.5, 0.6 minimum heat transfer occurs at $\varphi = 90^o$ and $\varphi = 180^o$, respectively.

For the cases Ra=10^6, it is seen that for all baffle length the Nusselt number is minimum at $\varphi = 90^o$ and also position of baffles nearly doesn't affect on heat transfer at angles $\varphi = 135^o, 180^o$.

The effect inclination angle on heat transfer at high Rayleigh numbers is more considerable

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.

5 Conclusion

Heat transfer by natural convection in a differentially heated square tilted cavity with two thin insulated baffles has been numerically studied. The adiabatic walls making an angle φ with the horizontal and other walls are isothermal. Two thin insulated baffles are attached to the isothermal walls at symmetric positions. In order to understand the flow and temperature fields and heat transfer characteristics of the typical cavity 120 cases are considered. To study the effects of inclination angle, baffle position and baffle length, the inclination angle φ=0°, 45°, 90°, 135° and 180°, the non-dimensional baffle positions D_b= 0.2, 0.4 and 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 10^6.

The numerical simulation reveals the following conclusions:

1- For all baffles locations, the average Nusselt number is smaller than the corresponding value in a cavity without baffle.

2- The baffles positions have a significant effect on the heat transfer and flow characteristics of the fluid

3- The Nusselt number is almost minimum at $\varphi = 90^o$ for all baffle lengths and positions

4- Generally, the Nusselt number decreases as baffle length increases

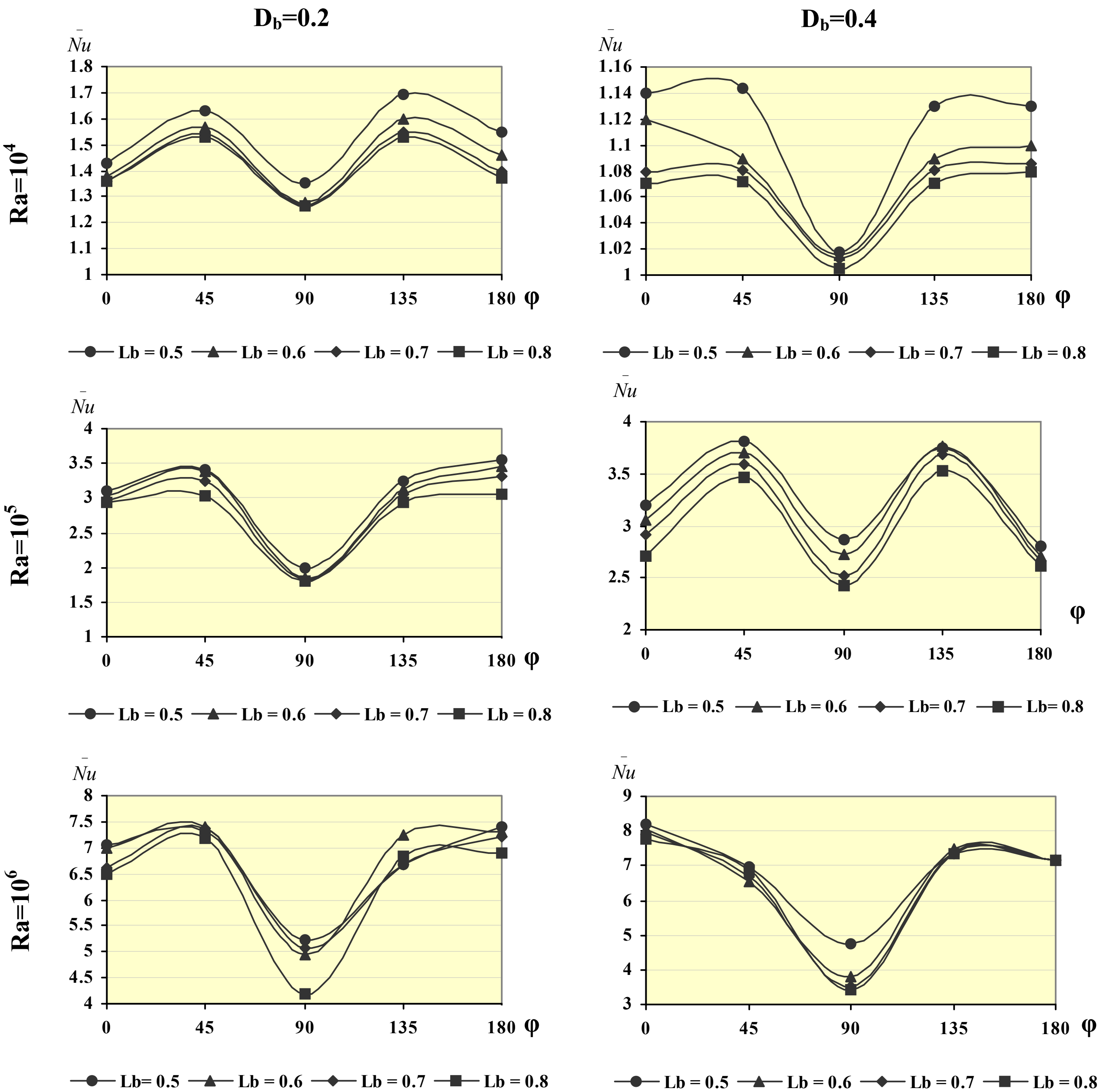

Fig.5 Variations of Nusselt Number vs inclination angel for various L_b, Ra, and D_b

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