Research Article

NUMERICAL STUDY OF NATURAL CONVECTION IN A FINNED RECTANGULAR ENCLOSURE HEATED FROM BELOW AND COOLED FROM ABOVE *1

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ABSTRACT

Laminar natural convection in an air filled enclosures heated from below and cooled from above is studied numerically. Fins are attached to the hot wall in order to study the effect of fins number on the heat transfer. The working fluid media is air with Prandtl number of 0.71 and Rayleigh number ranging from 10^4 to 10^6. The coupled equations of continuity; momentum and energy are solved by a finite volume method. The SIMPLE algorithm is used to solve iteratively the pressure velocities coupling. The numerical investigations in this analysis are made over a wide range of parameters, cavity aspect ratio, Rayleigh number and number of fins. The effect of these parameters was evaluated. Results are presented graphically in the form of streamlines, isotherms and also with temperature profiles and average Nusselt numbers. The heat transfer increases with the increase of the aspect ratio and Rayleigh number and decreases with the increase of fins number.

Keywords: *Natural convection; Rayleigh-Bénard convection; steady regime; fin; Rayleigh Number;Nusselt Number.*

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1.INTRODUCTION

Studies of natural convection in confined cavities have for several years been the subject of several studies, because of its involvement in many natural phenomena and industrial applications. The Rayleigh Bénard convectioncan be encountered in many physical applications as the rooms heating in buildings or the cooling of electronic components.

Natural convection heat transfer in cavities has been a subject for various experimental and numerical studies found in the literature [1–3]. Numerical simulation of two-dimensional Rayleigh–Bénard convection in an enclosure is investigated by Ouertatani et al [4], a Benchmark solution are proposed for Rayleigh numbers ranging from 10^3 to 10^6. Some streamlines and isotherms are presented to analyze the natural convection flow patterns set up by the buoyancy force.

Also, a two-dimensional steady-state simulation of laminar natural convection in square enclosures with differentially heated horizontal walls with the bottom wall at higher temperature have been conducted by Turan et al [5] for yield-stress fluids obeying the Bingham model. Heat and momentum transport are investigated for Rayleigh number in the range 10^3–10^5 and a Prandtl number range of 0.1–100. The mean Nusselt number is found to increase with increasing values of Rayleigh number for both Newtonian and Bingham fluids.

Natural convection heat transfer inside a vertical rectangular enclosure with four two-dimensional discrete flush-mounted heaters is investigated numerically and experimentally by Ho et al [6] to expose the influence of aspectratio of the enclosure. Their numerical results reveal that the increase of the aspect ratio leads to substantial degradation of convective dissipation from the discrete heaters.

Different Modes of Rayleigh–Bénard Instability in Two- and Three-Dimensional Rectangular Enclosures are described by Gelfgat et al [7], The results of the parametric calculations are presented as neutral curves showing the dependence of the critical Rayleigh number on the aspect ratio of the cavity.

Raji et al [8] showed numerical results of natural convection within an air-filled square cavity with its horizontal walls submitted to different heating models. The temperature of the bottom horizontal surface (hot temperature) is maintained constant, while that of the opposite surface (cold temperature) is varied sinusoidally with time. The remaining vertical walls are considered adiabatic. In comparison with the constant heating conditions, it isfound that the variable cooling temperature could lead to an extreme change in the flow structure and the corresponding heat transfer. This leads to a resonance phenomenon characterized by an important increase in heat transfer by about 46.1% compared to the case of a constant cold temperature boundary condition.

Steady natural convection in an enclosure heated from below and symmetrically cooled from the sides is studied numerically by Ganzarolli et al [9], using a stream function-vorticity formulation. The Rayleigh number based on the cavity height is varied from 10^3 to 10^7. Values of 0.7 and 7.0 for the Prandtl number are considered.The aspect ratio (length to height of the enclosure) is varied from 1 to 9. Boundary conditions are uniform wall temperature and uniform heat flux. Numerical values of the Nusselt number as a function of the Rayleigh number are reported, and the Prandtl number is found to have little influence on the Nusselt number. Further, Laminar and turbulent Rayleigh–Benard flows in a perfectly conducting cubical cavity were numerically simulated by Pallares et al[10] , Complete numerical simulations of laminar flows were conducted in the range of Rayleigh numbers $7*10^3 < Ra < 10^5$. . The large-eddy simulation (LES) technique was used for the simulations at two high Rayleigh numbers ($Ra = 10^6$ and 10^8).

Natural convection in enclosure with discrete isothermal heating from below is studied by Goutamet al[11]. They found that the effect of enclosure aspect ratio on the average Nusselt number of the discrete heaters tends to improve with the increase of the Grashof number.

Abdul Rasih et al [12] studied Natural Convection in Polar Enclosure Heated from Below and Cooled from Above, Simulations are performed for several values of both the length in radial direction with angle direction ratio of the enclosure, and the Rayleigh number based on the angle of enclosure in the range of 10^2 to 10^6. Their results show that the increase in both aspect ratio and Rayleigh number gives an effect on flow configuration of the enclosure. Otherwise, Mirabedin et al [13] performed a two-dimensional numerical simulation to study natural convection in circular enclosures filled with water considering different central angles. They showed that decreasingcentral angle of the cavity increases averaged Nusselt number in a cavity heated from below.

In addition to the above-mentioned previous studies, comprehensive investigations of the natural convection flow in Rayleigh Bénard enclosures with fins on the hot wall have been reported in the currentliterature. The effects of the number, material and position of the fins on the natural convection flow in the cavity have been remunerated much attention. In most of these studies, the thickness of the fin is considered to be sufficiently small in comparison with the length fin.

Zilic et al [14] examined the augmentation of classic Rayleigh-Bénard convection by the addition of periodically-spaced fins attached to the heated, lower plate. The respective impacts of the fin size, the fin spacing and the thermal conductivity of the fin material are observed through numerical simulations, they found that for veryshort fins, an enhancement of heat transfer is possible for the range of conditions examined.

Rayleigh-Bénard convection driven by the temperature difference of horizontal top and bottom surfaces ofa finned square cavity filled with liquid gallium for Pr=0.024, is studied numerically by Selamet et al [15] for $Ra=10^5$ and 3×105. Steady or unsteady, they illustrated the cellular flow structures and temperature patterns along with evolution of heat transfer rates (Nusselt number). The effect of fin length and placement on flow regime and heat transfer is established, they concluded that Short fins play a stabilizing role for $Ra=3\times10^5$.

Also, natural convection in a square cavity with a thin partition for linearly heated side walls is analyzed by Sathiyamoorthy et al [16]. The purpose of their paper is to optimize the heat transfer rate in square cavity by attaching fin at the bottom wall. They found that attaching fin reduces heat transfer rate inthe cavity.

Pathak et al [17] studied numerically natural Convection in Rectangular Enclosure with Heated Finned Base; the enclosure is heated from bottom wall and is cooled from the opposite top wall while the other walls of the enclosure are assumed to be adiabatic. They observed that Nu increases with increasing the number of fins until it reaches a maximum at certain fin spacing and with further increasing the number of fins, Nu starts to decrease. The heat transfer rate also increases with increasing the Rayleigh number.

Karki et al [18] made a Comparative study on air, water and nanofluids based Rayleigh–Benard natural convection using lattice Boltzmann method. One part of study shows the deviation in onset of critical Rayleigh number for air is 1.58%. The other part indicates dimensionless heat transfer, fluid flow and total irreversibility decrease with the increase in volume fraction of nanoparticles in the base fluid. However, Haragus et al [19] studieda bifurcation of symmetric domain walls for the Rayleigh–Bénard convection problem; they prove the existence of domain walls for the Bénard–Rayleigh convection problem. Their approach relies upon a spatial dynamics formulation of the hydrodynamic problem, a center manifold reduction, and a normal forms analysis of an eight-dimensional reduced system.

The purpose of this study is to simulate the natural convection in a Rayleigh-Benard heated cavity with finson the hot wall and different number of fins. The side walls are assumed to be adiabatic and the flat top and bottom walls are considered as differentially heated. The thermal and flow behavior and heat transfer characteristics have been studied for various Rayleigh number and aspect ratio. The working fluid media is air with Prandtl number of and Rayleigh number ranging from 10^3 to 10^6.

2.ANALYSIS AND MODELING

The numerical model considered is shown schematically in Figure 1. It is a rectangular cavity filled with airwith a fins placed on the bottom wall, surface heated from below in two dimensions with a side equal to L which represents the length of the cavity and a height H, aspect ratio is A = L / H=2 , the height (h) of the fin is H/3.The Prandtl number is equal to 0.71. The two vertical walls are maintained adiabatic, while the horizontal walls are kept isothermal (the lower hot wall Th, and the upper cold wall, Tc). The numerical study was carried out with the CFD code [24] using a two-dimensional model and the Boussinesq approximation for air. The investigation concerns cases with Rayleigh numbers ranging from 10^4 to 10^6, the effect of the fins is observed.

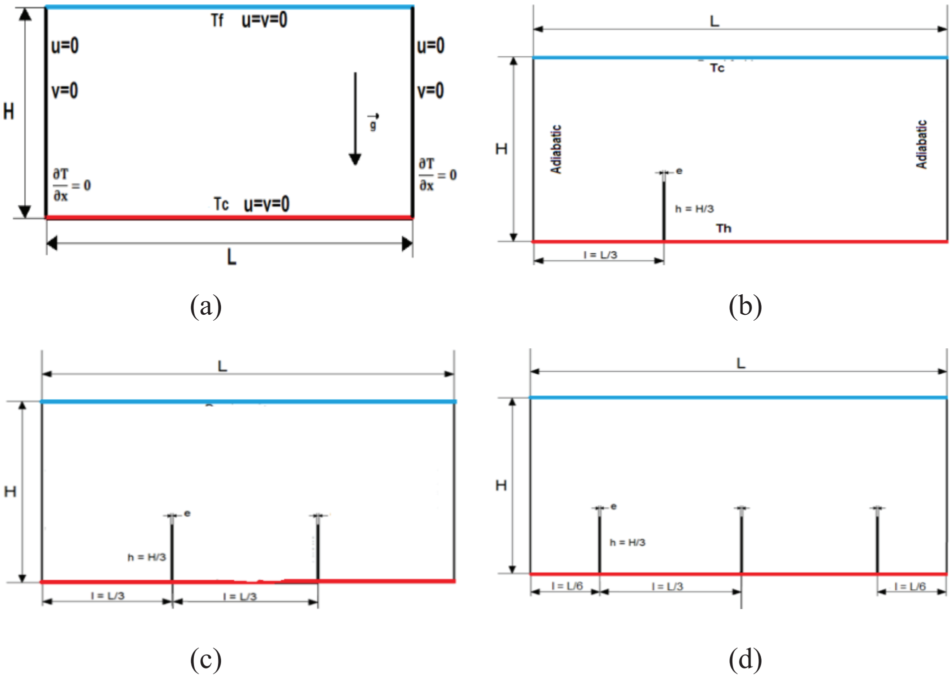


Figure 1. physical domain and boundary conditions (a) without fin, (b) 1 fin, (c) 2 fins, (d) 3fins.

The natural convection in the cavity as depicted in Figure 1 is described by the differential equations expressing conservation of mass, momentum and energy. The Boussinesq approximation is implemented for thefluid properties to relate density changes to temperature change and to couple in this way the temperature field to theflow field. The governing equations for Rayleigh Bénard convection can be expressed as follows:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \tag{1}$$

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

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial y}+\Pr\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)+Ra.\Pr.\theta \tag{3}$$

$$U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right) \tag{4}$$

$$Ra=\frac{g\beta L^3\Delta T}{\eta\alpha} \quad \text{and} \quad \Pr=\frac{\eta}{\rho} \tag{5}$$

$$X=\frac{x}{L},\quad Y=\frac{y}{L},\quad U=\frac{uL}{\alpha},\quad V=\frac{vL}{\alpha}$$

$$P=\frac{pL^2}{\rho\alpha^2},\quad \theta=\frac{T-T_0}{\Delta T},\quad \Delta T=T_h-T_0$$

The average Nusselt number is defined as follows:

$$Nu=-\int_0^x \left.\frac{\partial\theta}{\partial y}\right|_{y=0} dx \tag{6}$$

The boundary conditions are no slip for all walls and for energy equation; the horizontal walls have been maintained at differentially heated condition while the other walls are considered as adiabatic. The boundary conditions and the flow domain are shown in figure 2. It can be written mathematically as non-dimensional form:

Right surface $\quad U=V=0,\ \frac{\partial\theta}{\partial Y}=0$ (7a)

Left surface $\quad U=V=0,\ \frac{\partial\theta}{\partial Y}=0$ (7b)

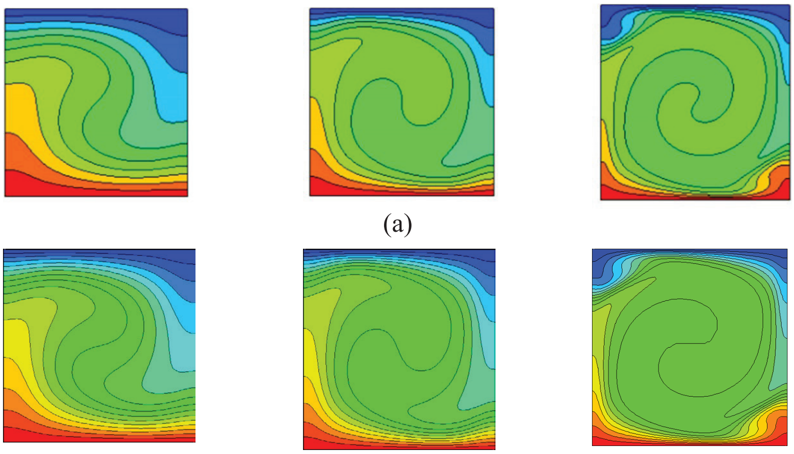
Top surface $\quad U=V=0,\ \theta=-0.5$ (7c)

Bottom surface $\quad U=V=0,\ \theta=0.5$ (7d)

3.NUMERICAL METHOD AND MODEL VALIDATION

The adimensional governing equations with their boundary conditions are solved using the finite volume method. The SIMPLE (semi-implicit method for pressure linked equations) algorithm is used to determine the pressure field, while the QUICK scheme is used to discretize the convection terms in the momentum and energy equations. The program was used to generate the grid of the simulated domain.

In order to verify the accuracy of the simulation results obtained with the CFD code, a validation was made, taking into account the numerical studies of Ouertatani et al [4]. The same boundary conditions were used: the fluid is air, of the case where the aspect ratio is 1, for different Rayleigh number, where the lower wall completely heated, the upper wall is cooled and the other walls are considered adiabatic.



(a)

(b)

Ra = 10^4 Ra = 10^5 Ra = 10^6

Figure 2. Comparison of isotherm contours for different Rayleigh number and A = 1Ouertatani et al [4], (b) present result

Table 1. Comparison of mean Nusselt number for A=1 and different Grashof number

Rayleigh number	Present study	[4]
10^4	2. 1463	2. 1581
10^5	3.7303	3.9103
10^6	6.5134	6.3092

For Rayleigh numbers from 10^4 to 10^6, the comparison of the isotherms with the numerical results of Ouertatani et al [4] (Figure 2) show an excellent agreement. This allows to validate our numerical simulation procedure. Also, a comparison of the averaged Nusselt numbers Nu_m (Table 1), the results are in quite agreement with those of [4].

The grid system for the computational domain is created using structured quadrilateral cells. A grid independency test was carried out, four sets of grids 50×25, 100×50,200×100and 200×100were employed; the case with 200×100grids (Figure 3) was used for taking both the accuracy and convergence rate into account.

To choose the best mesh which allows to obtain the most exact results possible, we studied the influence of the size and the distribution of the nodes on the average Nusselt number (Figure 4) for Rayleigh number varies between 10^4 to 10^6 (Table.2).

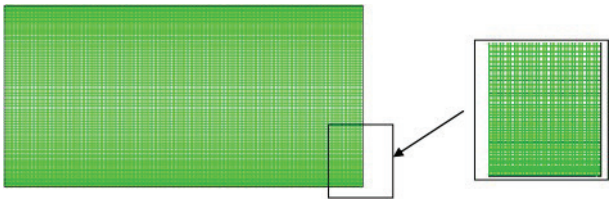


Figure 3. Grid Structure used in the computations(A=2)

Table 2. Mean Nusselt number for different grids and different Rayleigh number Ra

Nu_m

Grid numbers in X-Y	$Ra=10^4$	$Ra=10^5$	$Ra=10^6$
50×25	2,2618	3,56918	6,14002
100×50	1,89941	3,5376	6,0982
200×100	1,5648	3,1832	6,0229
300×150	1,5734	3,185	6,0233

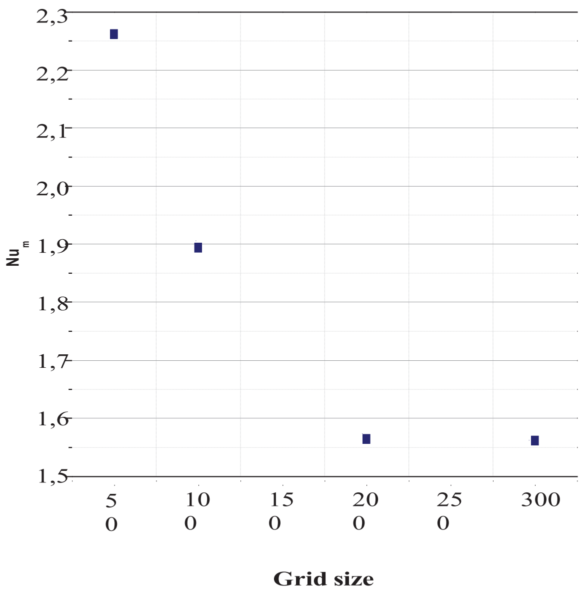


Figure 4. Variation of mean Nusselt for different grids and Rayleigh number Ra $=10^5$

4.RESULTS AND DISCUSSION

Figure 5 shows the streamlines for Rayleigh numbers Ra $=10^4$ to 10^6. For Ra = 10^4, the values of the stream function increase and the streamlines change to square rolls. For higher values of the Rayleigh number Ra = 10^5 and Ra = 10^6 the streamlines change to become transverse rolls. These rolls are still developing to become square rolls but with the narrowing of their centres. The increase in the Rayleigh number therefore reflects an intensification of natural convection.

The isotherms contours are shown in Figure 6 for different values of the Rayleigh number. For small Rayleigh, the isothermal lines are parallel to the two horizontal plates, the temperature distribution is simply decreasing from the hot plate to the cold plate; the heat transfer is essentially by conduction. These isothermal lines begin to change and deform slightly for Ra = 10^4, this reflects a growing Rayleigh-Bénardconvection. Heat transfer is dominated by a convective regime. For high values of the Rayleigh number (Ra =

10^5 and Ra = 10^6), the isothermal lines deform to become perpendicular to the plates in the central part and become horizontal in the vicinity of the plates. It is also noted that these lines are narrowing at the level of thesolid walls, which reflects a very intense transfer in these regions. The development of Rayleigh-Bénard's instability is clearly noted by the visible development of hot temperature plumes that attempt to brush a path upwards and therefore the cold particles move downwards.

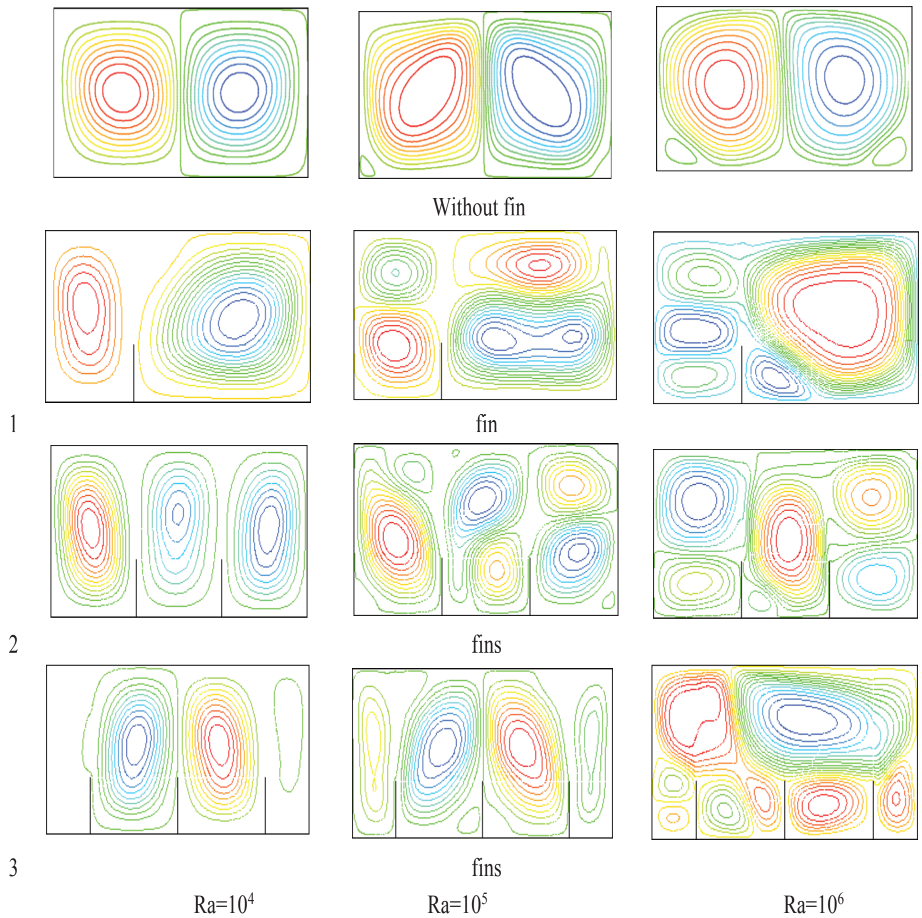


Figure 5. Streamlines for different Rayleigh number.

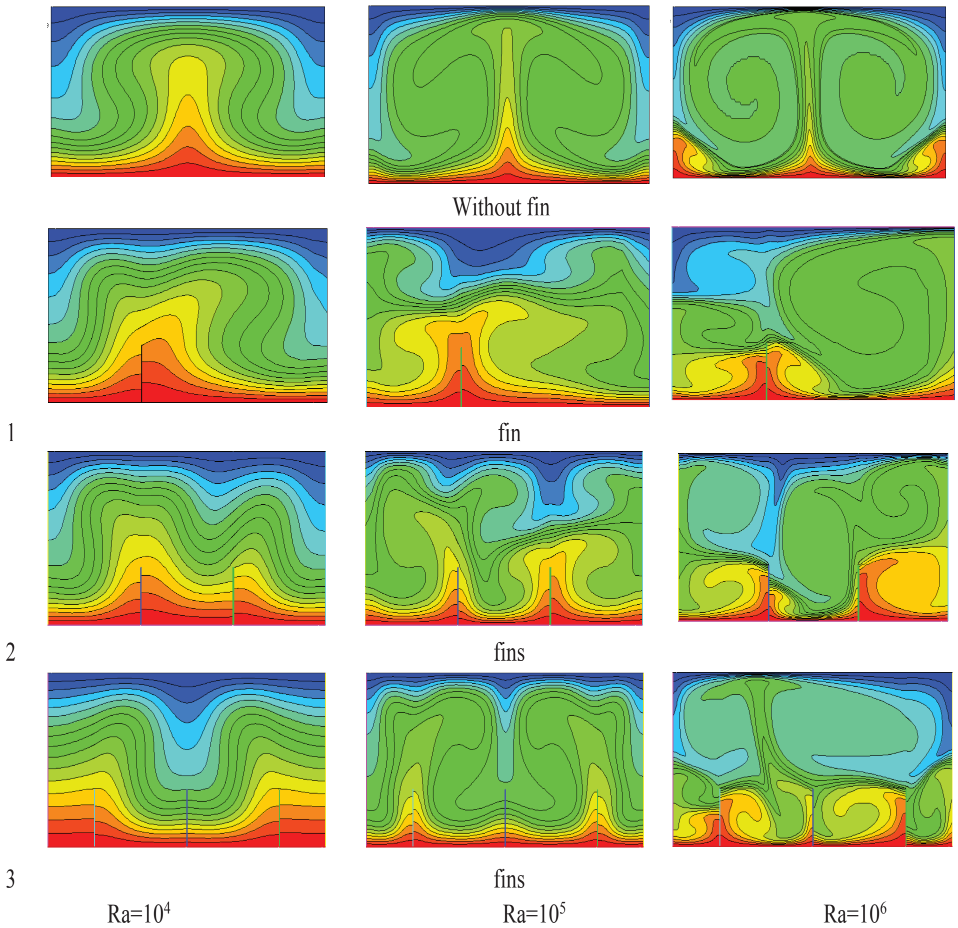


Figure 6. Isotherms contour for different Rayleigh number.

As shown in Figures 5 and 6, the number of the fins has a large impact on the characteristics of the temperature wave and streamlines. The variations of the characteristics of the isotherms are associated with the variations of fins number. The fin is an important parameter, when the number of fins increases the temperature alsoincreased.

Figure 7 shows the local Nusselt number distributions at the hot wall for different fins number and differentRayleigh number; it is observed that the addition of fins has an influence on the local Nusselt number and on theheat transfer. Because of the symmetry of boundary conditions, the center of the heated part becomes a zone of minimal heat flow since it is at a maximum temperature; this implies a minimal local Nusselt number (Figure 7 (a))

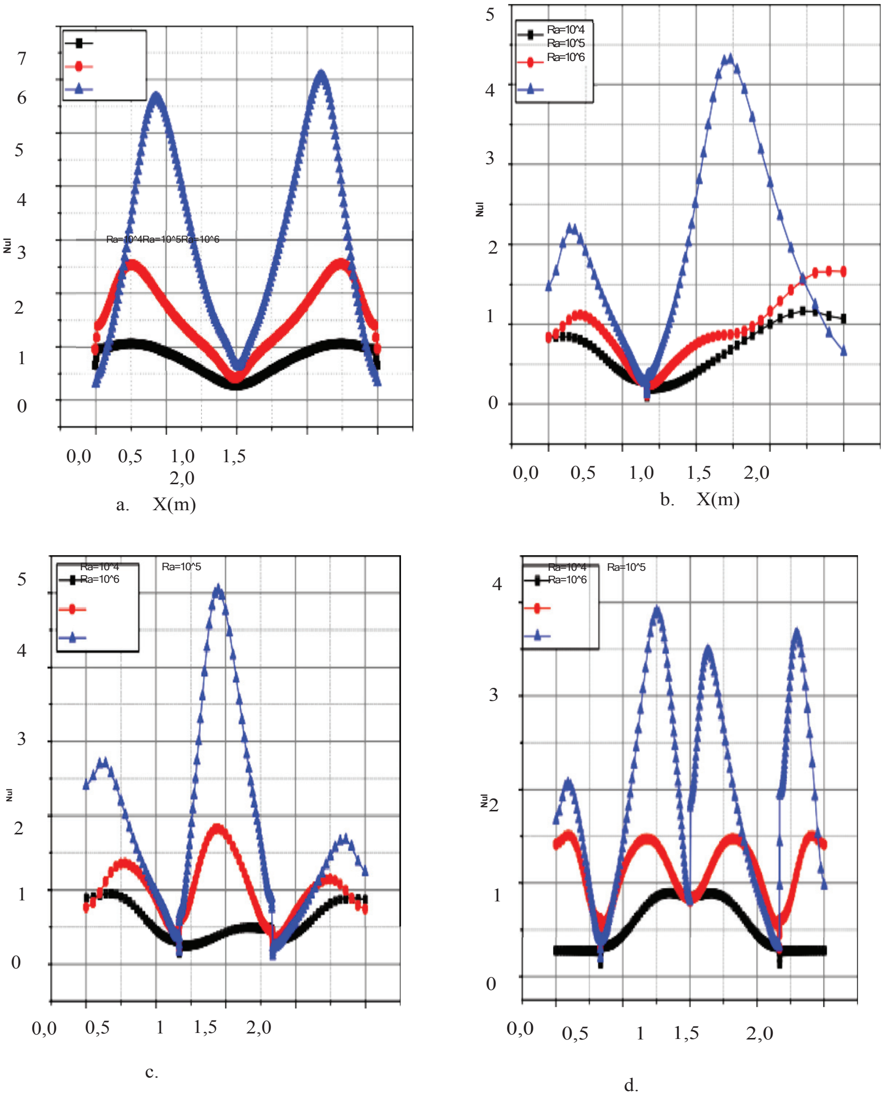


Figure 7. Local Nusselt number along the hot wall for different Rayleigh number and fins number. (a) Without fin, (b) 1 fin, (c) 2 fins, (d) 3 fins.

The distribution of the mean Nusselt number for different fins number is shown in Figure 8 for the Rayleigh number Ra = 10^5. It is shown that the mean Nusselt number increases when the Rayleigh number is increased. It is clearly seen that there is an influence of fins number on the average Nusselt number; the trend of the curve decreases significantly with an increase in the fins number.

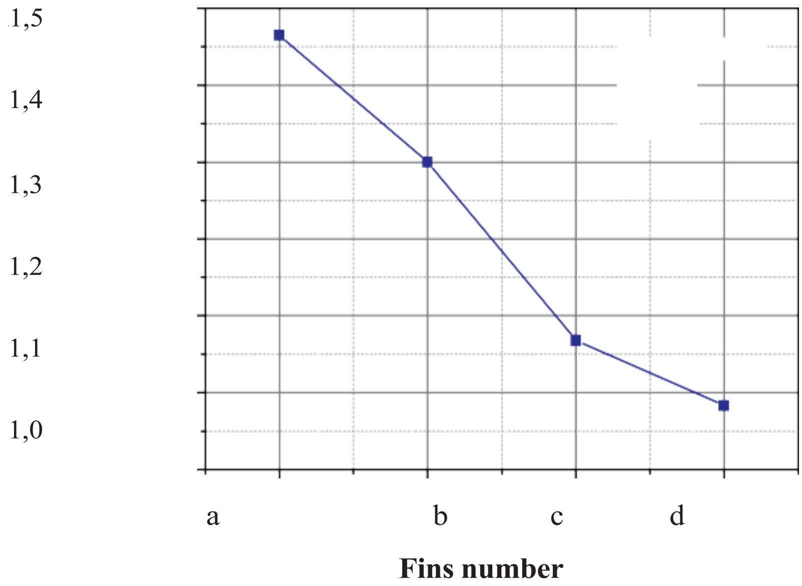


Figure 8. Mean Nusselt number along the hot wall versus fins number

The heat transfer through the finned hot wall is notably enhanced so the presence of fins on the hot wall shows a promising result for the enhancement of the heat transfer through a Rayleigh Bénard cavity. Thus, the mean Nusselt number is much smaller near the fins, although oscillatory downstream of the fins.

5.CONCLUSION

Laminar natural convection within air filled Rayleigh-Bénard enclosures which is heated from below and cooled from above is studied numerically. Fins are attached to the hot wall in order to study the effect of fins number on the heat transfer. The numerical investigations in this analysis are made over a wide range of parameters, Rayleigh number and number of fins. The effect of these parameters was evaluated. Results are presented graphically in the form of streamlines, isotherms and also with temperature profiles average Nusselt numbers and local Nusselt number. The heat transfer rate is found to decrease with an increase in the fins number, and increase with an increase of Rayleigh number.

The results obtained show that for a low Rayleigh number the dominance of the heat transfer mode is by conduction. Beyond this value, the convection dominates and appears more clearly for Ra = 10^5. It has been concluded that the heat transfer rate increases with the increase of the Rayleigh number and decreases with the increase of fins number, which shows that natural convection is very sensitive to the variation of Rayleigh number and the number of fins.

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