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A Numerical Study of Natural Convection in a Square Enclosure with Non-Uniformly Heated Bottom Wall and Square Shape Heated Block

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Abstract : *The present work is aimed to study has been carried out of natural convection in a square enclosure with non-uniformly heated bottom wall and square shape heated block with different Prandtl numbers of 0.71, 1.0 and 1.5 has been investigated numerically. The horizontal bottom wall of the square cavity was non-uniformly heated and inner square shape heated block kept at T_h while the side walls of the cavity were maintained at a cold temperature T_c with $T_h>T_c$ and upper wall is adiabatic. Finite element method was employed to solve the dimensionless governing equations of continuity, momentum and energy of the problem. Using the developed code, a parametric study was performed, and the effects of the Rayleigh number and the different Prandtl number on the fluid flow and heat transfer inside the square enclosure were investigated. The obtained results showed that temperature distribution and flow pattern inside the square enclosure depended on both strength of the magnetic field and Rayleigh number. For all cases two counter rotating eddies were formed inside the square enclosure. The magnetic field is decreased with the intensity of natural convection and flow velocity. Also it was found that for higher Rayleigh numbers a relatively stronger field was needed to decrease the heat transfer through natural convection.*

Keywords: *Natural convection, square enclosure, magnetic field, square shape heated block.*

Nomenclatures:

B_0	Strength of the magnetic field	Greek symbols	
g	gravitational acceleration	α	thermal diffusivity
L	length of the cavity	β	Volumetric coefficient of thermal expansion
Nu	Nusselt number		
P	dimensional pressure	ν	kinematic viscosity of the fluid
p	pressure	θ	non-dimensional temperature
Pr	Prandtl number	ρ	density of the fluid
Ra	Rayleigh number	ψ	non-dimensional stream function
T	dimensional temperature		
u, v	velocity components	Subscripts	
U, V	non-dimensional velocity components	c	cold wall
x, y	Cartesian coordinates	h	hot wall
X, Y	non-dimensional Cartesian coordinates		

I. Introduction

Natural convection have industrial applications in which the heat and mass transfer occur concurrently as a result of combined buoyancy effects of thermal and species diffusion. Now a day, natural convection is a subject of central importance in the present technology development and all kinds of engineering applications. Historically, some of the major discoveries in natural convection have helped shape the course of development and important fields for the application of the cooling of power electronics, solar collectors, solar stills, attic spaces, etc.

A number of studies have been conducted to investigate the flow and heat transfer characteristics in closed cavities in the past. Kuhen et al. [1] an experimental and theoretical study of natural convection in the annulus between horizontal concentric cylinders. S. V. Patankar [2] numerical methods for heat transfer and fluid flow, New York, Hemisphere. Acharya [3] studied natural convection in an inclined enclosure containing internal energy sources and cooled from below. Chadwick et al. [4] presented natural convection in a discretely heated enclose for single and multiple heater configurations experimentally and analytically. Aydin et al. [5] investigated natural convection of air in a two-dimensional rectangular enclosure with localized heating from below and symmetrical cooling from the side walls. Fusegi et al. [6] performed a numerical study on natural convection in a square cavity by using a high-resolution finite difference method. The authors considered differentially heated vertical walls and uniform internal heat generation in the cavity. Ganzarolli et al. [7] investigated the natural convection in rectangular enclosures heated from below and cooled from the sides. Natural convection heat transfer in rectangular cavities heated from the bottom had been investigated by Hasaaoui et al. [8]. Turkoglu et al. [9] made a numerical study using control volume approach for the effect of heater and cooler locations on natural convection on cavities. The authors indicated that for a given cooler position, average Nusselt number increases as the heater is moved closer to the bottom horizontal wall. Aydin et al. [10] numerically investigated natural convection of air in a two-dimensional rectangular enclosure with localized heating from below and symmetrical cooling from the side walls. Asan et al.[11] studied the steady-state, laminar, two-dimensional natural convection in an annulus between two isothermal concentric square ducts. Roy chowdhury et al. [12] analyzed natural convective flow and heat transfer features for a heated cylinder kept in a square enclosure with different thermal boundary conditions. Dong et al. [13] studied the conjugate of natural convection and conduction in a complicated enclosure. Their investigations showed the influences of material character, geometrical shape and Rayleigh number (Ra) on the heat transfer in the overall concerned region and concluded that the flow and heat transfer increase with the increase of thermal conductivity in the solid region; the overall flow and heat transfer were greatly affected by both geometric shape and Rayleigh number. Braga et al. [14] numerically studied steady laminar natural convection within a square cavity filled with a fixed amount of conducting solid material consisting of either circular or square obstacles. It was found that the average Nusselt number for cylindrical rods was slightly lower than those for square rods. De et al. [15] performed a simulation of natural convection around a tilted heated square cylinder kept in an enclosure within the range of $10^3 \leq Ra \leq 10^6$. They reported detailed flow and heat transfer features for two different thermal boundary conditions and obtained that the uniform wall temperature heating was quantitatively different form the uniform wall heat flux heating. Varol et al. [16] solved the problem of two-dimensional natural convection in a porous media filled triangular enclosure with a square body. Kahveci et al. [17] studied natural convection in a partitioned vertical enclosure heated with a uniform heat flux. Basak et al. [18] studied and solved the finite element analysis of natural convection flow in a isosceles triangular enclosure due to uniform and non-uniform heating at the side walls. Varol et al. [19] numerically investigated natural convection in right-angle porous trapezoidal enclosure with partially cooled from inclined wall. Kent, EF. [20] presented numerical analysis of laminar natural convection in isosceles triangular enclosures for cold base and hot inclined walls. Yui et al. [21] made a work on unsteady natural convection heat transfer from a heated horizontal circular cylinder to it air-filled coaxial triangular enclosure. Rahman et al. [22] offered a numerical model for the simulation of double-diffusive natural convection in a right-angled triangular solar collector. Mahmoodi et al. [23] studied numerically magneto hydrodynamic free convection heat transfer in a square enclosure heated from side and cooled from the ceiling. Jani et al. [24] numerically investigated magneto hydrodynamic free convection in a square cavity heated from below and cooled from other walls. Bhuiyan et al. [25] offered a numerical method for the simulation magnetohydrodynamic Free Convection in a square cavity with semicircular heated block. Nader et al. [26] investigate natural convection of Water-Based Nanofluids in a Square Enclosure with non-uniform heating of the bottom wall. Pirmohammadi et al. [27] shows that the effect of a magnetic field on buoyancy-driven convection in differentially heated square cavity.

On the basis of the literature review, it appears that no work was reported on the natural convection flow in a square enclosure with non-uniformly heated bottom wall and square shape heated block with internal heat generation. The obtained numerical results are presented graphically in terms of streamlines, isotherms, local Nusselt number and average Nusselt number and Rayleigh numbers.

II. Physical Configuration

The physical model considered in the present study of natural convection in a square enclosure with non-uniformly heated bottom wall and square shape heated block is shown in Fig.1 The height and the width of the cavity are denoted by *L*. Inside the square cavity at a constant high temperature T_h, while, the two vertical walls are maintained at cold temperature T_c and the upper wall is adiabatic. The bottom wall is non- uniformly heated wall. The magnetic field of strength B_0 is applied parallel to *x*- axis. The square cavity is filled with an electric conductive fluid with Pr = 0.71, 1.0 and Pr =1.5 that is considered Newtonian and incompressible.

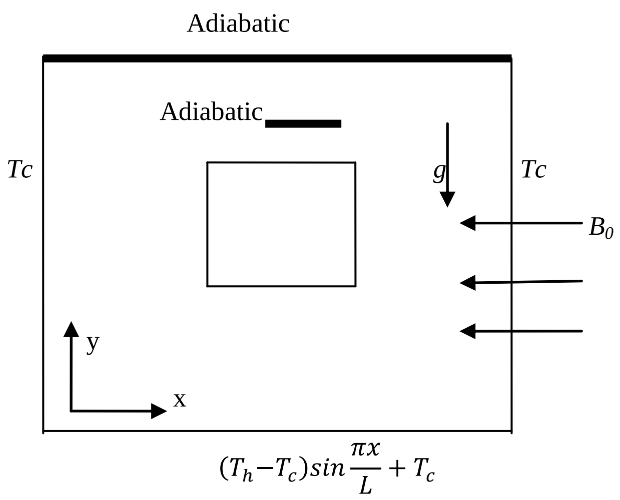


Fig.1 Schematic view of the cavity with boundary conditions considered in the present study

III. Mathematical Formulation

The assumption uncouples the Navier-Stokes equations from Maxwell's equations. No electric field is present and the Hall effect is neglected. The thermo-physical properties of the fluid are assumed to be constant. Except the density variation in the buoyancy term which is treated according to Boussinesq approximations while viscous dissipation effects are considered negligible. In general, the cavity fluid is assumed to be Newtonian and incompressible, steady, laminar, natural convection flow.

With above mentioned assumptions, the governing equations for conservations of mass, momentum and energy can be written as

Mass conservation equation:

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

Momentum conservation equations:

$$\rho\left(\frac{\partial u}{\partial x} + \frac{\partial u}{\partial y}\right) = -\frac{\partial p}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \tag{2}$$

$$\rho\left(\frac{\partial v}{\partial x} + \frac{\partial v}{\partial y}\right) = -\frac{\partial p}{\partial y} + \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) + \rho g \beta (T - T_c) - \sigma B_0^2 v \tag{3}$$

Energy conservation equations:

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \alpha\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \tag{4}$$

where x and y are the distances measured along the horizontal and vertical directions respectively; u and v are the velocity components in the x and y directions respectively; T denotes the fluid temperature, p is the pressure, ρ is the fluid density, g is the gravitational acceleration, β is the volumetric coefficient of thermal expansion, B_0 is the magnitude of magnetic field, α is the thermal diffusivity.

The above equations are non-dimensionalzed by using the following dimensionless quantities

$$X = \frac{x}{L}, Y = \frac{y}{L}, \ U = \frac{uL}{\alpha}, V = \frac{vL}{\alpha}, P = \frac{pL^2}{\rho\alpha^2}, \theta = \frac{(T - T_c)}{(T_h - T_c)} \tag{5}$$

where $\upsilon = \frac{\mu}{\rho}$ is the reference kinematic viscosity and θ is the non-dimensional temperature. After substitution of dimensionless variable we get the non- dimensional governing equations are:

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \tag{6}$$

$$U\frac{\partial U}{\partial X} + V\frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \Pr\left(\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2}\right) \tag{7}$$

$$U\frac{\partial V}{\partial X} + V\frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \Pr\left(\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2}\right) + Ra\Pr\theta - Ha^2\Pr V \tag{8}$$

$$U\frac{\partial \theta}{\partial X} + V\frac{\partial \theta}{\partial Y} = \left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2}\right) \tag{9}$$

where *Ra*, *Pr* and *Ha* are the Rayleigh, Prandtl and Hartman numbers and are defined as:

$$Ra = \frac{\beta(T_h - T_c)L^3}{\alpha\nu}, \Pr = \frac{\nu}{\alpha}, Ha = B_0 L\sqrt{\frac{\sigma}{\rho\upsilon}} \tag{10}$$

The effect of magnetic field into the momentum equation is introduced through the Lorentz force term, $\vec{J} \times \vec{B}$ that is reduced to a systematically damping factor $-\sigma B_0 v^2$.
To computation of the rate of heat transfer, Nusselt number along the hot wall of the enclosure is used that is as follows:

$$Nu_{local} = -\left.\frac{\partial\theta}{\partial Y}\right|_{Y=0} \quad Nu_{local} = -\left.\frac{\partial\theta}{\partial X}\right|_{X=0} \tag{11}$$

The average Nusselt number of the hot bottom wall of the square cavity is obtained as follows:

$$Nu = \int_0^1 Nu_{local}\,dX \tag{12}$$

The non- dimensional boundary conditions for solving the governing (6- 9) are:

On the bottom wall of the square cavity: U= 0, V= 0, θ = sin(πX)
On the square shape heated block walls: U= 0, V= 0, θ = 1
On the cold walls: U= 0, V= 0, θ = 0
The non-dimensional stream function is defined as $U = \frac{\partial\psi}{\partial Y}, V = -\frac{\partial\psi}{\partial X}$ (13)

IV. Numerical Technique

The nonlinear governing partial differential equations, i.e., mass, momentum and energy equations are transferred into a system of integral equations by using the Galerkin weighted residual finite-element method. The integration involved in each term of these equations is performed with the aid Gauss quadrature method. The nonlinear algebraic equations so obtained are modified by imposition of boundary conditions. These modified nonlinear equations are transferred into linear algebraic equations with the aid of Newton's method. Lastly, Triangular factorization method is applied for solving those linear equations.

4.1. Program Validation and Comparison with Previous Work

In order to check on the accuracy of the numerical technique employed for the solution of the problem considered in the present study, the code is validated with Pirmohammadi et al. [27] shows that the effect of a magnetic field on buoyancy-driven convection in differentially heated square cavity, the two vertical walls are maintained at cold temperature T_c and the upper wall is adiabatic. Streamlines and isotherms are plotted in Fig. 2. Showing good agreement.

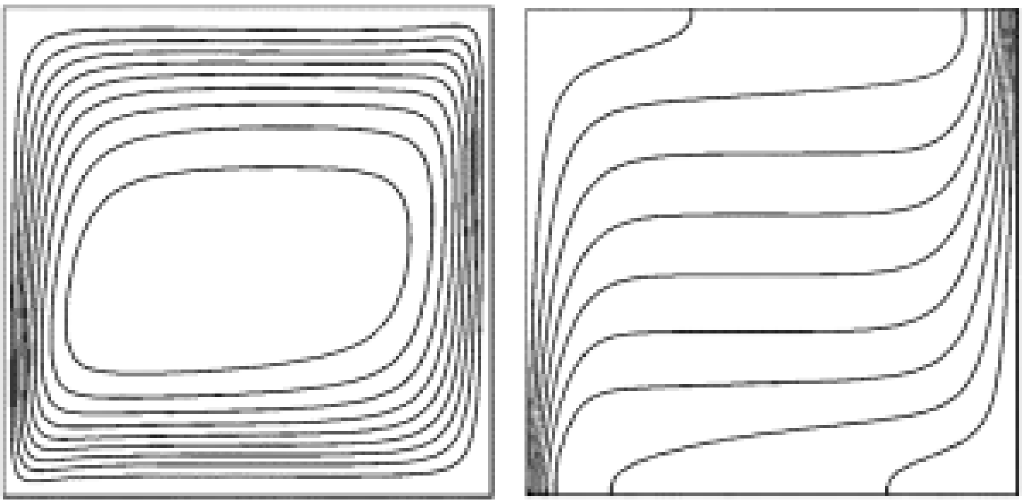


Fig. 2(a). Pirmohammadi et al.[27] Obtained streamlines and Isotherms for Ra = 10^6, Pr = 0.7 33 and Ha=50.

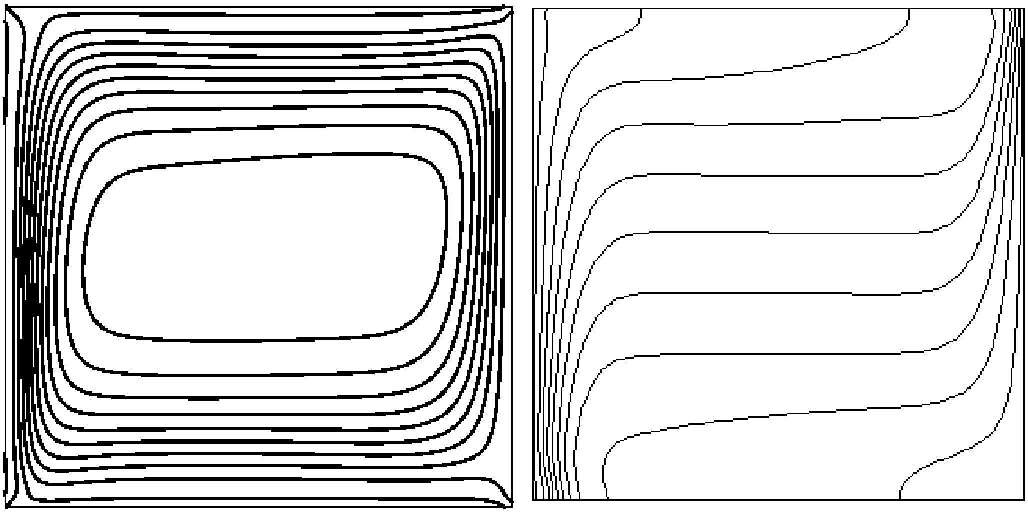


Fig. 2(b). Present: Obtained streamlines and Isotherms for Ra = 10^6, Pr = 0.71 and Ha = 50

V. Result and Discussion

A numerical work was performed by using finite element method to natural convection in a square enclosure with non-uniformly heated bottom wall and square shape heated block. The non- dimensional significant parameter different Prandtl numbers of 0.71, 1.0 and 1.5 has been investigated numerically. The results have been obtained for the Rayleigh number ranging from 10^3 to 10^6, Hartmann number of 50. The results are presented in terms of streamlines, isotherms inside the square cavity, and the vertical velocity component along the bottom wall of the square cavity, the local Nusselt number along bottom wall and left wall of the square cavity and the average Nusselt number along the bottom wall of the square cavity. Variations of streamlines and isotherms inside the square cavity with Rayleigh number are presented in Fig. 3.1, 3.2, 3.3 and 4.1, 4.2, 4.3 respectively. We observed from the figures with existence centerline of the cavity, the flow and temperature fields are symmetrical about this line. As can be seen from the streamlines in the Fig. 3.1, 3.2, 3.3 a pair of counter-rotating eddies are formed in the left and right half of the cavity for all Rayleigh numbers considered. Each cell ascend through the symmetry axis, then faces the upper wall and moves horizontally and finally descends along the corresponding cold side wall.

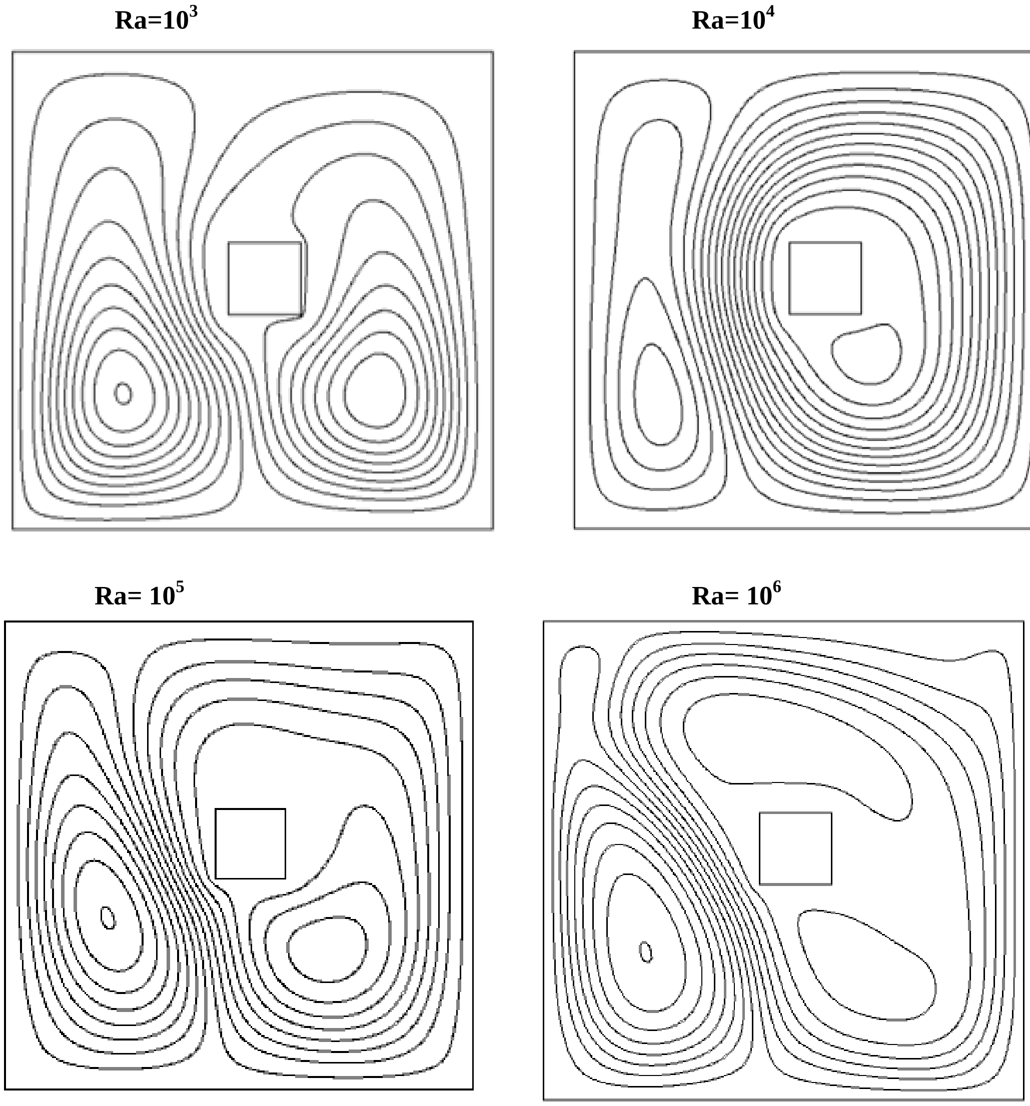
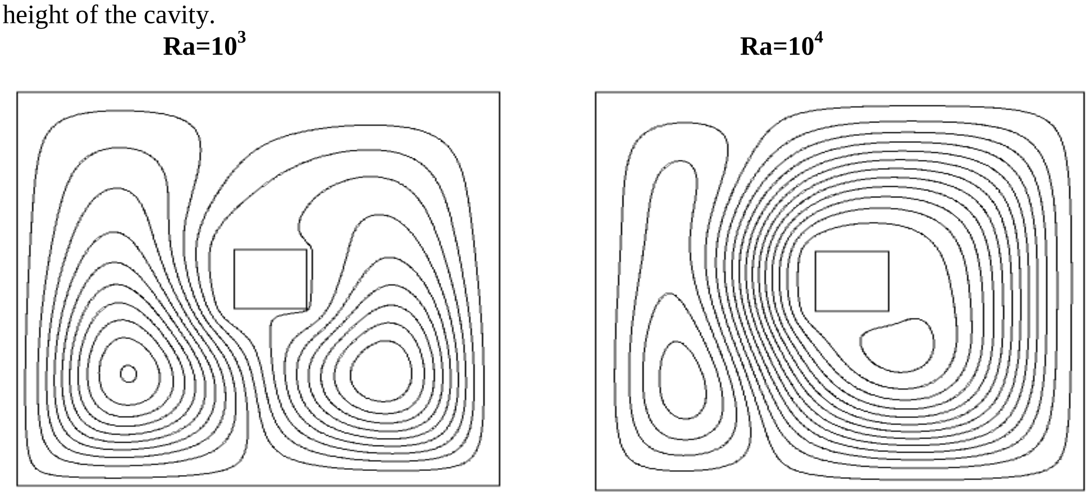


Fig. 3.1 Streamlines at different Rayleigh numbers, Prandtl number of 0.71

At Ra= 10^3 and in the absence of the magnetic field, the maximum absolute value of the stream function increases; there is a circulating flow in the cavity with central streamline into an elliptic shape of eddies are in the lower and upper half of the cavity. For different Prandtl number and Ra increasing, so flow strength increasing. With increase in Rayleigh number and buoyancy force, the elliptic shape of the eddies move upward insofar as at Ra= 10^6, locate in the upper half of the cavity. At Ra= 10^6 the elliptic shape are elongated along the height of the cavity.



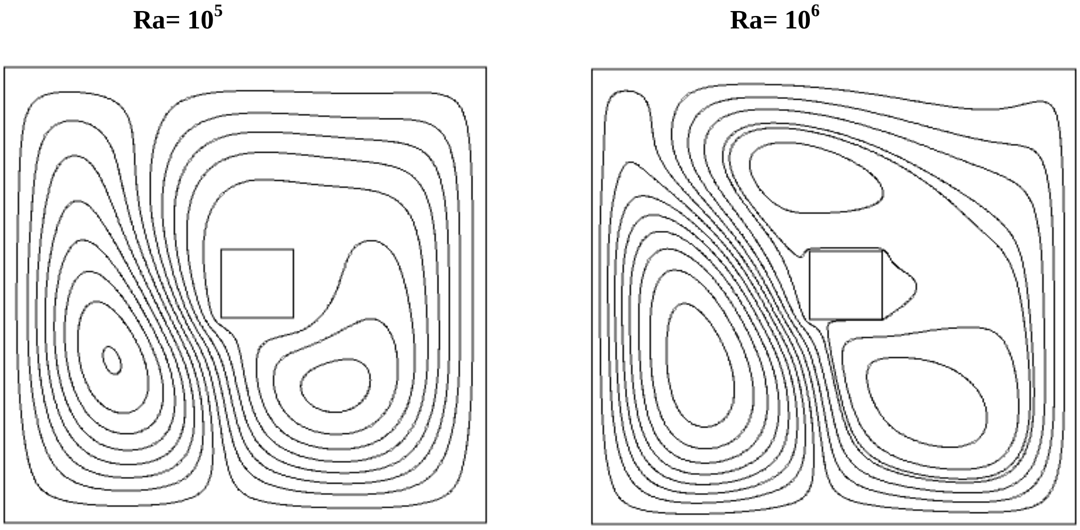


Fig. 3.2 Streamlines at different Rayleigh numbers, Prandtl number of 1.0

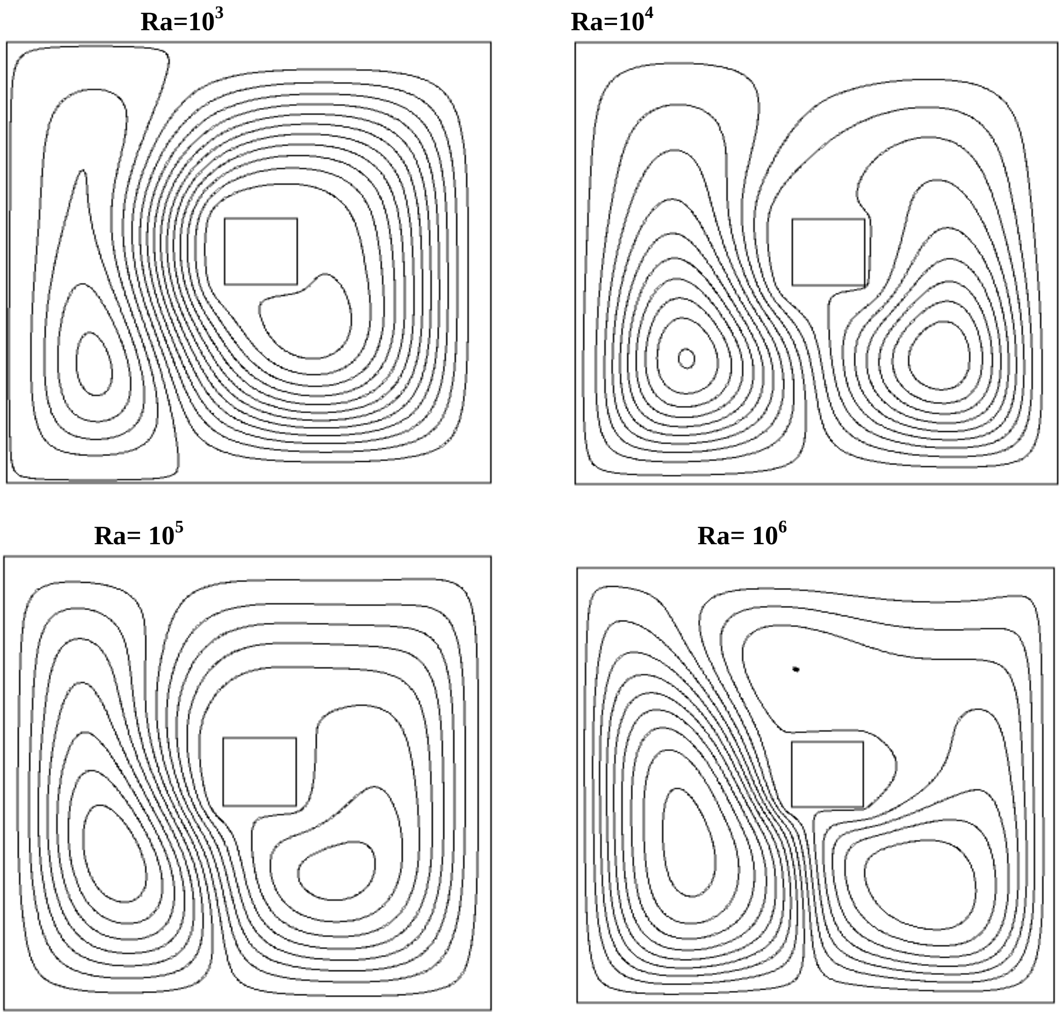


Fig. 3.3 Streamlines at different Rayleigh numbers, Prandtl number of 1.5

Conduction dominant heat transfer is observed form the isotherms in Fig. 4 at Ra= 10^3 to 10^6. With increase in Rayleigh number, the isotherms are condensed next to the side wall which means increasing heat transfer through convection. Formation of thermal boundary layers can be found from the isotherms at Ra= 10^3 and 10^6. The cavity was heated at the left and right wall are cooled while the rest of the boundaries were insulated. Moreover, the inner body is kept isothermal.

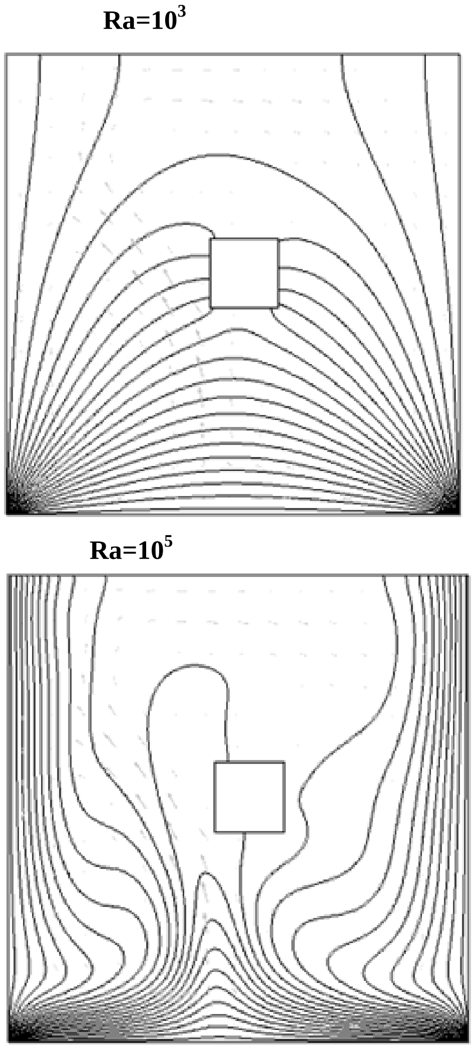
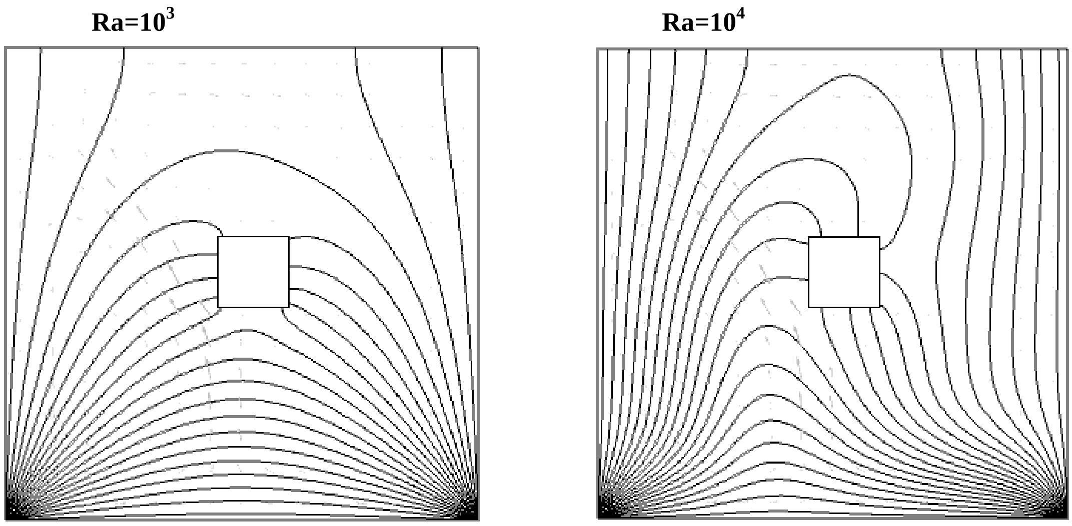


Fig. 4.1 Isotherms at different Rayleigh numbers, Prandtl number of 0.71



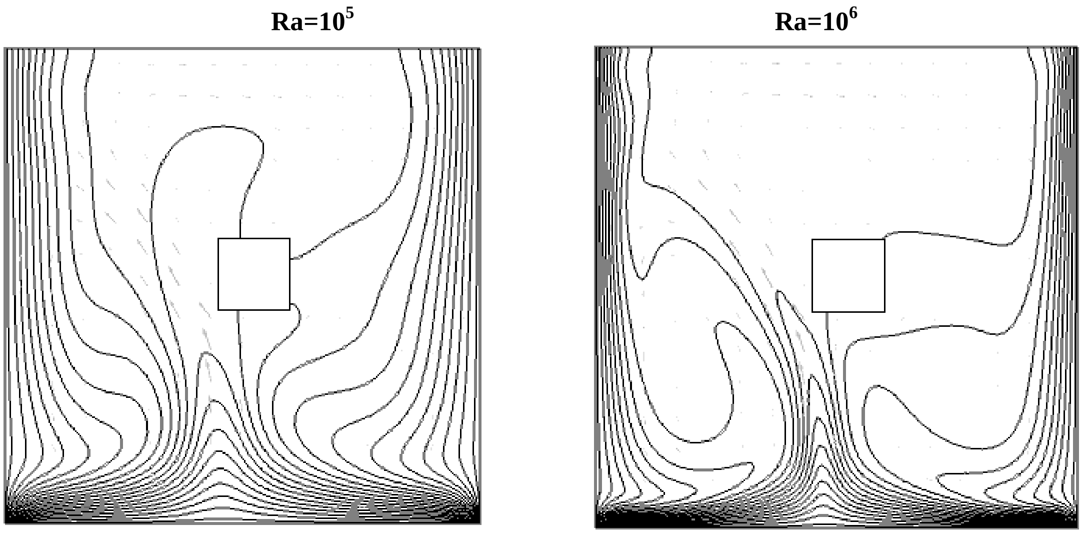


Fig. 4.2 Isotherms at different Rayleigh numbers, Prandtl number of 1.0

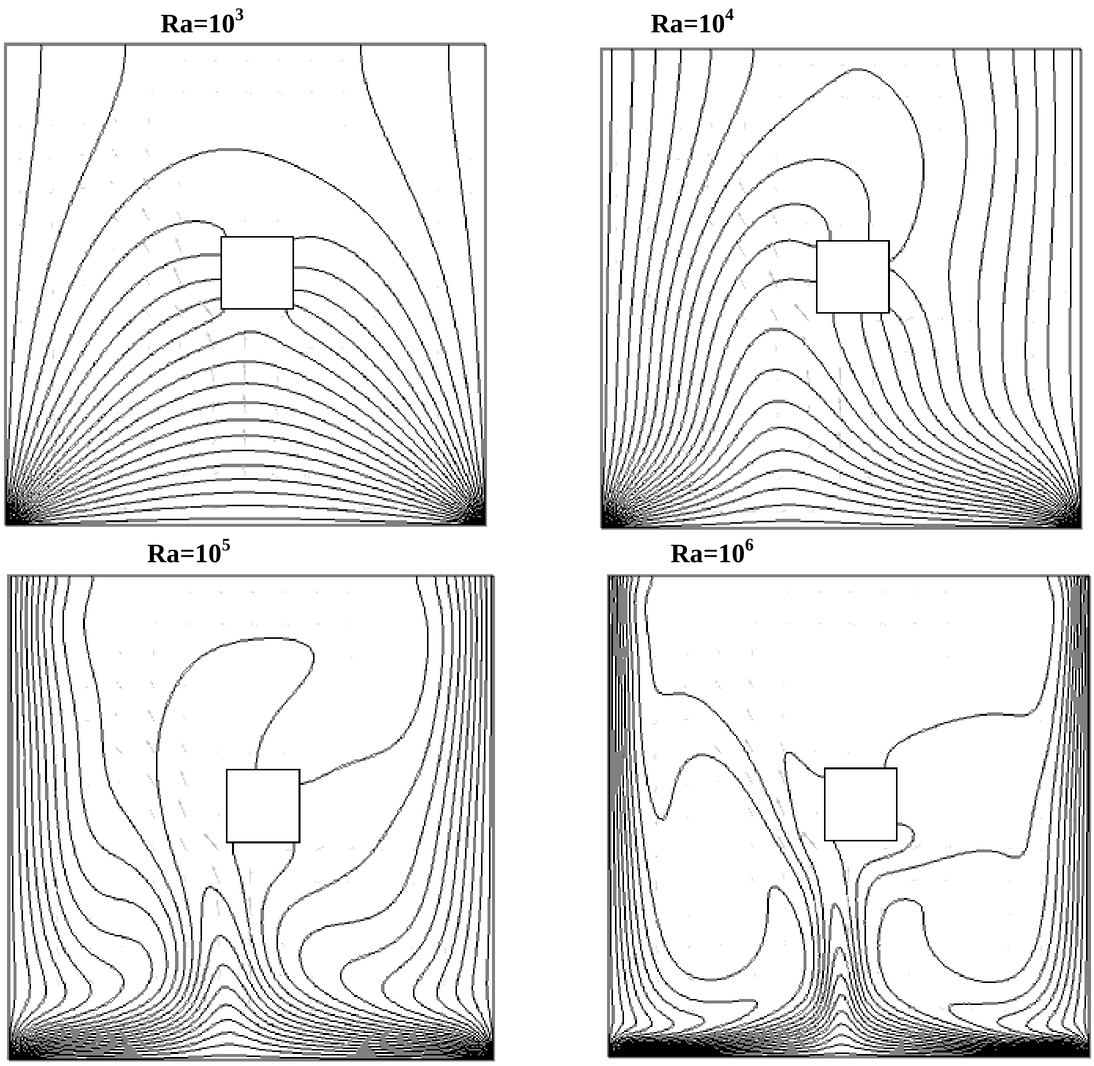
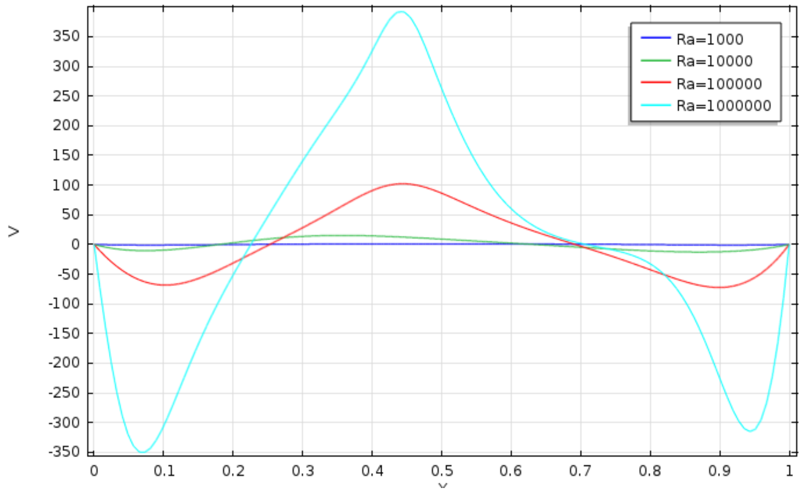
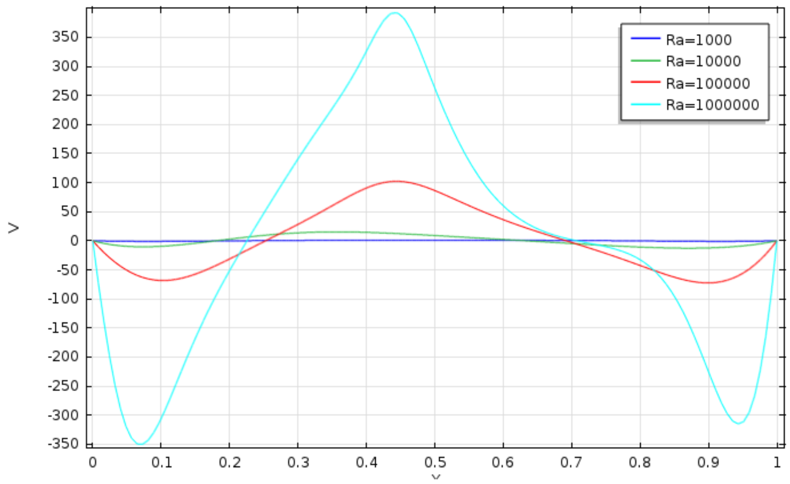


Fig. 4.3 Isotherms at different Rayleigh numbers, Prandtl number of 1.5

Pr = 0.71



Pr = 1.0



Pr = 1.5

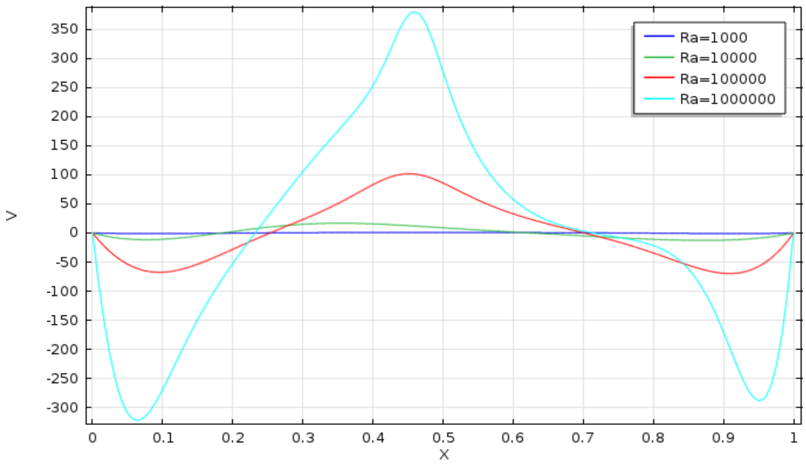


Fig. 5. Variation of the vertical velocity component along the bottom wall of the square cavity with Rayleigh number at different Prandtl number of 0.71, 1.0 and 1.5

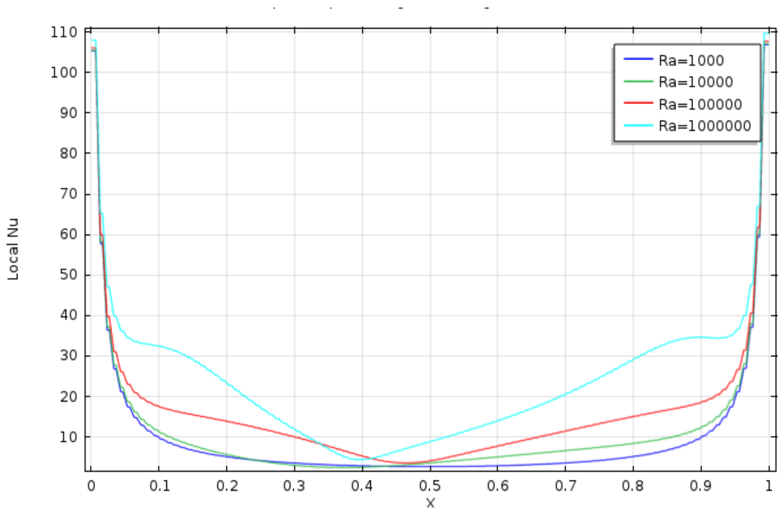
For increasing the Prandtl number (increasing the strength of the magnetic field) we found stream lines the elliptic shape of eddies move upward and close to the upper corners of the cavity. This phenomenon means decrease in the flow velocity with increasing Prandtl number. From the isotherms it is observed that with increase in Prandtl number, free convection is suppressed and heat transfer occurs mainly through convection.

At Pr = 0.71and for Ra= 10^3 and 10^5, the isotherms illustrate a pure conduction heat transfer, while at Ra= 10^6 a combination of conduction and weak free convection observed from the corresponded isotherms. As a result it can be said that with increase in the buoyancy force via increase in Rayleigh number, to decrease free convection, a stronger magnetic field is needed compared to the lower Rayleigh numbers.

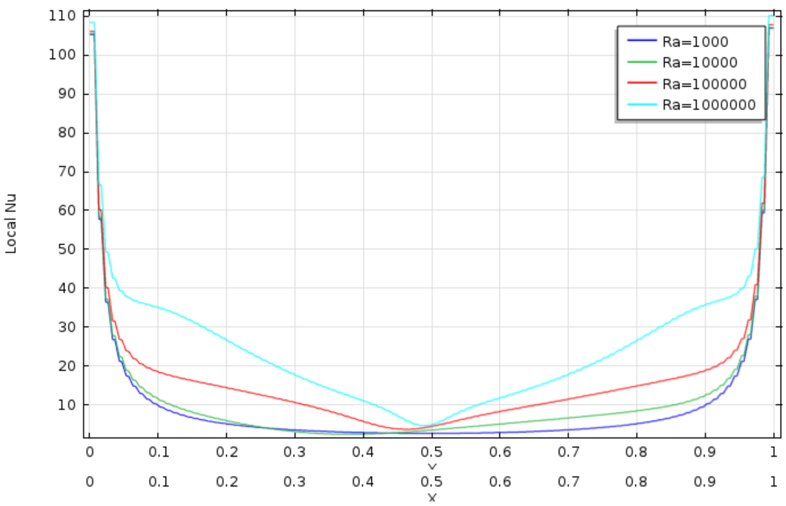
Effects of the magnetic field on the flow and temperature fields at Ra= 10^6 are completely different from the previous results at lower Rayleigh numbers. At Pr= 1.0 the thickness of the thermal boundary layers decreases. At Pr = 1.5 the isotherms are parallel with the horizontal walls in the vicinity of these walls, while in the major portion of the cavity the isotherms are parallel with the side walls. From the streamlines it is observed that at Pr = 0.71, the centrally located elliptic shape of the eddies move downward and their shape converts from ellipse to a right triangle. At Pr = 1.0 and Pr = 1.5 the shape of the core of eddies converts to isosceles triangle and locates close to the hot bottom wall.

Variations of the vertical velocity component along the bottom wall of the square cavity with the Rayleigh number and for Pr = 0.71, Pr = 1.0 and Pr = 1.5 are shown in Fig. 5. It can be seen from the figure that the absolute value of maximum and minimum value of velocity increases with increasing the Rayleigh number (increasing the buoyant force).

Pr = 0.71



Pr = 1.0



Pr = 1.5

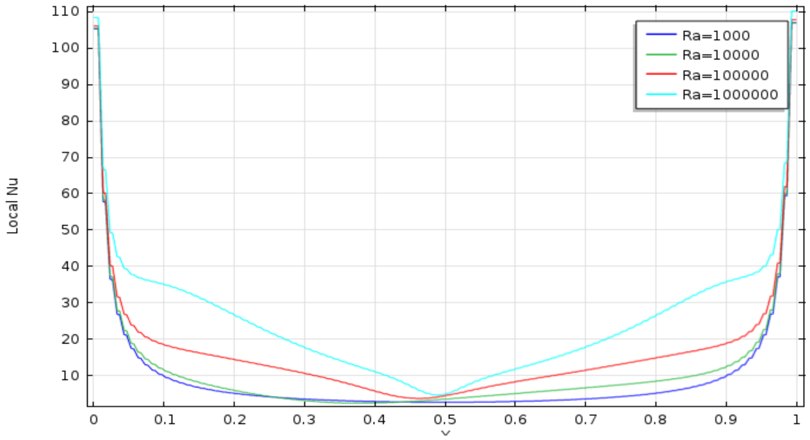
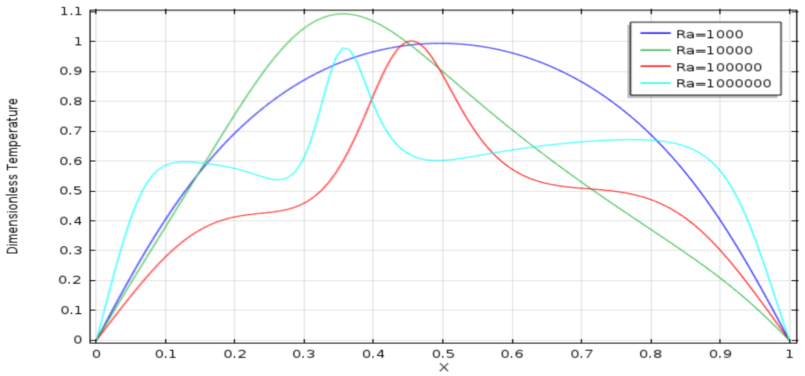


Fig. 6. Variation of the local Nusselt number along the bottom wall of the square cavity with Rayleigh number at different Prandtl number of 0.71, 1.0 and 1.5.

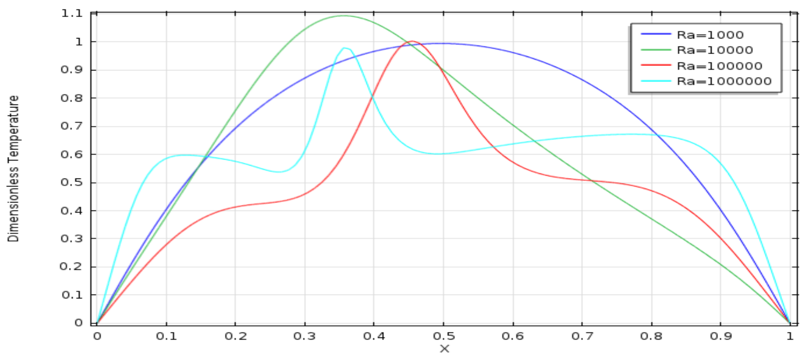
Variations of the local Nusselt number along the bottom wall of the square cavity with the Rayleigh number at different Prandtl number of 0.71, 1.0 and 1.5are shown in Fig. 6. Owing to the symmetry in thermal boundary conditions, the local Nusselt number is symmetrical with respect to the vertical midline of the cavity. It can be seen from the figure that the local Nusselt number increases with the Rayleigh number in major portion of the hot wall. In the middle of the bottom wall the local Nusselt number equals to zero and does not change significantly with increase in the Rayleigh number.

Variations of the local Nusselt number along the left wall of the square cavity with the Rayleigh number at different Prandtl number of 0.71, 1.0 and 1.5 are shown in Fig. 6. Owing to the symmetry in thermal boundary conditions, the local Nusselt number is symmetrical with respect to the left wall of the cavity. It can be seen from the figure that the local Nusselt number increases with the Rayleigh number in major portion of the hot wall. In the middle of the left wall the local Nusselt number equals to zero and does not change significantly with increase in the Rayleigh number.

Pr = 0.71



Pr = 1.0



Pr = 1.5

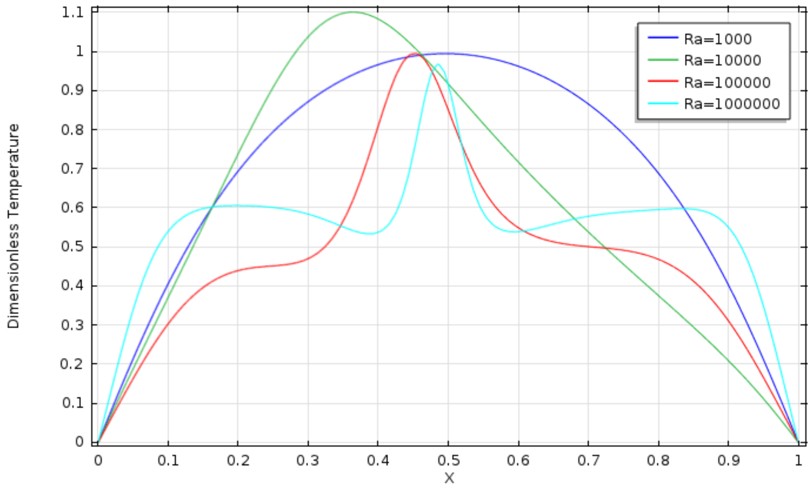


Fig. 7. Variation of the dimensionless temperature along the bottom wall of the square cavity with Rayleigh number at different Prandtl number of 0.71, 1.0 and 1.5

Variation of the dimensionless temperature along the bottom wall of the square cavity with Rayleigh number at different Prandtl number of 0.71, 1.0 and 1.5 are shown in Fig. 7. For moderate and high Rayleigh number (Ra= 10^3, 10^4 ,10^5 and 10^6) and for a weak magnetic field strength there is a temperature of the absolute value of maximum and minimum value of temperature increases with increasing the Rayleigh number (increasing the buoyant force). This is because at high Ra and low Pa, convection is dominant heat transfer mechanism. Heat transfer by convection alters the temperature distribution to such an extent that temperature gradients in the centre are close to zero. From the stream line pattern we see that there is a strong upward flow near the cold wall and downward flow at the hot wall.

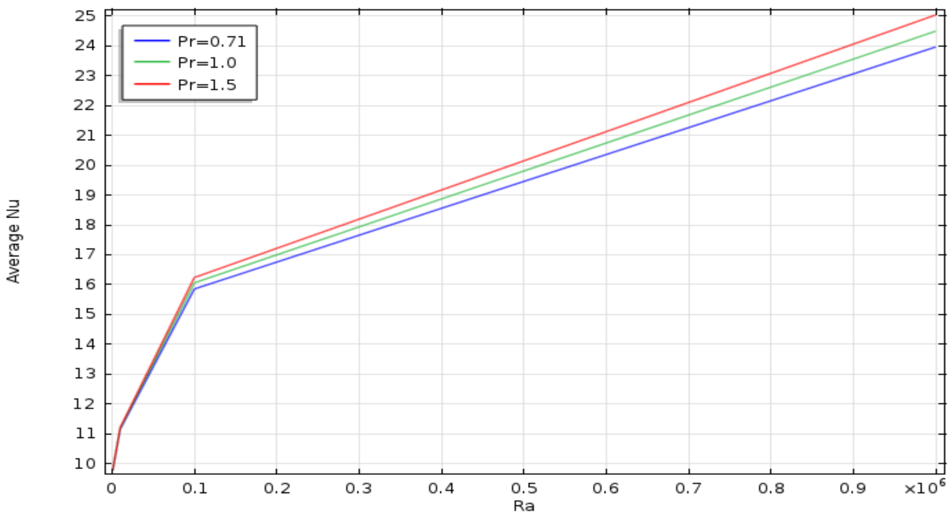


Fig. 8. Variation of the average Nusselt number along the bottom wall of the square cavity with Rayleigh numbers at different Prandtl number of 0.71, 1.0 and 1.5.

Plot of the average Nusselt number of the nonuniformly heated bottom wall as a function of Rayleigh number at different Prandtl numbers is shown in Fig. 8. For a fixed Rayleigh number, with increase in the Prandtl number the flow velocity decreases, the free convection is suppressed and finally the rate of heat transfer decreases. At Pr = 0.71, 1.0 and 1.5 the average Nusselt number is equal for Ra = 10^3, 10^4 , 10^5 and 10^6 a higher heat transfer rate occurs. At a constant Prandtl number, with increase in Rayleigh number the buoyancy force increases and the heat transfer is enhanced. Therefore at high Rayleigh numbers, a relatively stronger magnetic field is needed to decrease the rate of heat transfer.

IV. Conclusion

A numerical work has been done to simulate natural convection in a square enclosure with non-uniformly heated bottom wall and triangular shape heated block, cooled from other walls filled with an electric conductive fluid with different Prandtl number of 0.71, 1,0, 1.5 was studied numerically. Finite element method was to solve governing equations for a heat generation parameters, Rayleigh numbers and Prandtl numbers. Very good agreements were observed between Rayleigh numbers at different Prandtl numbers. Subsequently, a parametric study was performed and the effects of the Rayleigh number and the Prandtl number on the fluid flow and heat transfer were investigated. For all cases considered, two counter rotating eddies were formed inside the cavity regardless the Rayleigh and the Prandtl number. The obtained results showed that the heat transfer mechanisms, temperature distribution and the flow characteristics inside the cavity depended strongly upon both the strength of the magnetic field and the Rayleigh number. Moreover it was observed that, for low Rayleigh numbers, by increase in the Prandtl number, natural convection is suppressed and heat transfer occurs through conduction mainly.

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