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Numerical Investigation of Effect Partially Corrugated Heated Wall on Behaviour Natural Convection Heat Transfer in A Square Enclosure

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ABSTRACT

For fluid flow and heat transfer with engineering industrial applications important methods called natural convection are used. The current numerical study focuses on the effect of changing the heating positions of the corrugated left wall on heat transfer by natural convection with laminar flow in a closed square two-dimensional cavity filled with air as a working fluid. The code of a commercial program (COMSOL Multiphysics 6.0) was used to solve the mathematical equations governing fluid flow represented by continuity, momentum, and conservation of thermal energy. The right wall is exposed to a cold temperature, the other walls are thermally insulated. The effect of partial heating of the left corrugated wall and the Rayleigh number on the process of heat transfer by free convection was studied. At different Rayleigh numbers, flow field patterns and temperature distribution lines appear. The average Nusselt number, the average air temperature, and the flow field are slightly affected by low Rayleigh numbers, while with high Rayleigh numbers, these parameters are significantly and noticeably affected and behave differently by changing the heating sites. Numerical results show negligible effects of wall heating sites at low Rayleigh numbers and significant effects at high Rayleigh numbers. Also, the Nusselt number increases progressively with increasing Rayleigh number, so the heat transfer process gets improved.

1. Introduction

Natural convection is a very important topic due to its many applications in many fields of nature, and although there is no forced velocity for this type of load, natural convection currents are generated inside the fluid that induces it to flow as a result of the effects of buoyancy force or the so-called flotation force. As we see in many devices that include multiple methods of heat transfer, through which the heat transfer rates or the cost of operation are affected, the natural load plays an important role in the design or performance of the device, which is much preferred over the forced load. Due to various industrial engineering applications such as solar collectors, nuclear reactor design, building insulation, twin panel windows, effective of energy building etc., so in space with different configurations, the process of heat transfer by natural convection has wide areas for those applications. In recent years, the rate of heat transfer by free convection has been verified numerically and experimentally by many researchers [1-3]. Debayan Das et al. [4]

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presented a review study on enhancing heat transfer by free convection in a space of various configurations, including triangular, square, circular, and trapezoidal, as well as with curved or corrugated walls, some of which are filled with liquid working media and the other porous, in addition, the study dealt with reviewing the space filled with nanofluid under different thermal boundary conditions. Abderrahmane Baïri et al. [5] reviewed a study of heat transfer by natural convection in cavities of various configurations for its importance in engineering industrial applications. The boundary conditions affecting the change of flow and heat transfer properties, as well as the temperature distribution locations, the nature of the liquid used, and the radiation properties, were verified. Experimental and numerical studies have been used to solve the rational problem of heat transfer inside cavities with different ranges of Rayleigh numbers and with laminar and turbulent flows. Many large-scale flows occur in the environment as a result of buoyancy force, where the fluid flow becomes turbulent to exceed the Rayleigh number of critical values, which represents the ratio of buoyancy force to viscosity force [6-8]. In cavities during convection heat transfer, the contribution of the thermal radiation process is one of the most important topics. Several different parameters such as the temperature of the walls, the geometry of the cavity, the thermophysical properties of the internal medium, and the emission of thermal radiation by the surface are all reliable. As a result, the huge number of computational resources required by thermal radiation can be neglected [9-11]. He presented a numerical analysis of turbulent natural convection heat transfer inside a cavity with a cubic configuration. Parameters were investigated for counting, including the boundary layer and the effect of radiation. Four calculation circumstances have been analyzed: non-radiation, gas radiation, surface radiation, and combined radiation. This allows for the separation of the radiation impacts of the gas and surface radiations. The numerical results of the study showed the generation of shear stress for turbulent flow occurs by surface radiation inside the cavity also includes the first total heat transfer convection and the second radiation. P. Ternik and R. Rudolf [13] presented a numerical analysis to optimize the process of heat transfer by natural convection in a square cavity for the Rayleigh number range ($10^{\text{٣}}$-$10^{\text{٥}}$). Nanomaterials with a range of volumetric fractions (0-0.1) of which (Au, Al_2O_3, and Cu) were included. The Ansys Fluent program was used to solve the equations governing the flow inside the cavity. Numerical results showed an increase in the heat transfer rate by increasing the Rayleigh number and volumetric fractions of nanomaterials. Soheil Soleimani et al. [14] presented a numerical study using the finite element method to investigate heat transfer by free convection within a semi-annular space. The inner and outer walls are heated and cold, respectively, while the other walls are thermally insulated. Several parameters were studied numerically, including the Rayleigh number and volumetric fractions of nanomaterials added with distilled water to increase heating, as well as changing the angle of the cavity. The findings show that for any Rayleigh number, there is an ideal angle of turn where the average Nusselt number is at its highest. Furthermore, streamlines, isotherms, and the maximum and lowest values of the local Nusselt number are all significantly impacted by the spin angle. Qasim S. Mehdi and Khudheyer S. Mushatet [15] presented a numerical investigation of heat transfer by turbulent free natural convection and fluid flow within a partially heated square space. The effect of changing the angle of inclination (0°-180°) was studied for the range of Rayleigh numbers (10^8-10^{16}). The finite volume method was used to solve the Navier-Stokes equations and also the thermal energy equation. To address the effect of fluid turbulence the K-epsilon model was used. The numerical results showed that as the Rayleigh number increases, the heat transfer rate gradually increases and decreases with increasing tilt angle. The vortices within the space flatten and unite strongly when the number of Rayleigh increases. Akeel Mohammed et al. [16] experimentally studied the transfer of air heat around a heated circular cylinder immersed inside a triangular cavity inclined at an angle. The dimensions of the triangle were changed by three different values (0.2,0.25, and 0.3m), thickness (0.001 m), and depth (0.5 m). Several parameters have been studied for their influence on the heat transfer process by free convection, including the Rayleigh number in the range ($5x\ 10^6$-$2.5x10^8$) and the radius of the cylinder. Experimental results indicated heat transfer rates gradually increase with increasing Rayleigh number. Hicham Salhi et al. [17] reviewed a numerical analysis to improve the natural convective heat transfer within a square cavity with two-dimensional steady-state laminar flow. Hybrid nanomaterials were mixed as a technology for the optimization process of heat transfer (Nusselt number) with the working fluid (water). The effect of several parameters including the Rayleigh number, the type of nanomaterial and the configurations of the heat source were studied numerically using a commercial program (Ansys Fluent). The results showed that as the volume fractions of the nanomaterial increase, the number of Nusselt gradually increases.

The current numerical simulation focuses on the investigation of laminar heat transfer by the method of free convection inside the cavity of the configuration square heated with three different for the left corrugated wall and highlighted with a cold temperature for the right wall and the upper and lower walls are thermally insulated. The study of the Reynolds number range ($1.3x10^5$ to $4.55x10^5$) was conducted using a commercial program for solving the governing equations of the flow based on the finite element method.

2. Description of Physical Problem

Figure (1) presents the schematic diagram of the mathematical model represented by a space of equal height to length (square). There are two walls exposed to different temperatures and the other two upper and lower are thermally insulated. In the original location of the space without the angle of inclination, the left wall is highlighted with a hot temperature in three different locations (bottom, middle, and top) while the right wall is highlighted with a cold temperature. Due to the difference between the temperatures of the hot and cold walls of the square space, Heat Flow is generated by natural laminar convection. At different Rayleigh numbers the geometry of the square space is defined and to examine the effects of the velocity and temperature fields the direction can be changed and controlled. The effect of a change in three different states of the location of the corrugated hot left wall on the natural convection was studied as shown in Figure (2).

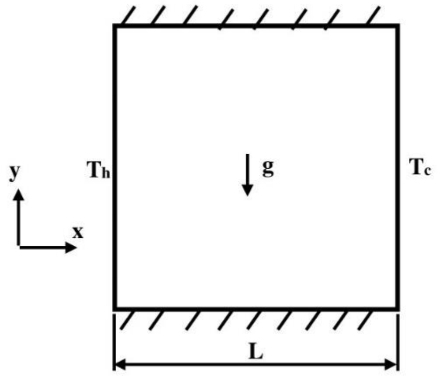


Figure 1. Schematic diagram of a square cavity with boundary condition

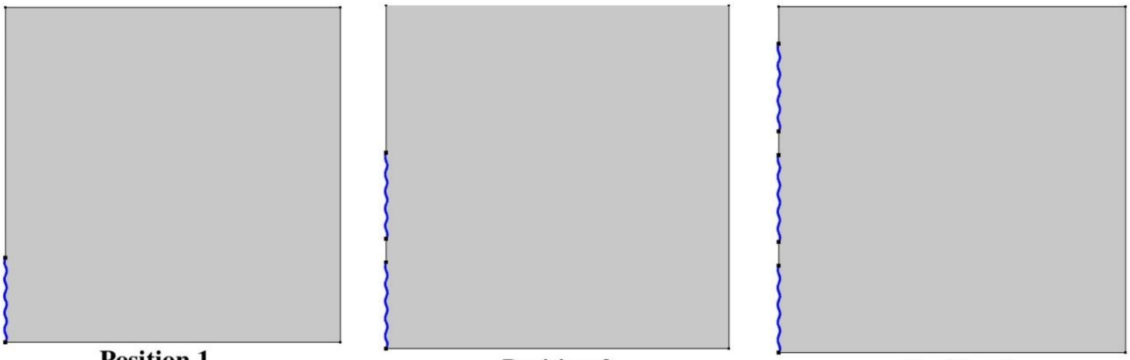


Figure 2. Three different locations of the corrugated hot wall

3. Governing Equation and Boundary Conditions of Numerical Model

The flow is imposed for laminar, incompressible, constant-state, and two-dimensional square space. In buoyancy force, the flow is considered to be Newtonian with constant properties except density. Both the effects of viscous dissipation and thermal radiation, as well as internal heat generation, are neglected with the use of the Boussinesq approximation. The governing equations of the flow can be expressed in their non-dimensional form based on the previously mentioned assumptions as follows [18-20]:

Continuity equation:

$$\frac{\partial \hat{u}}{\partial \hat{x}} + \frac{\partial \hat{v}}{\partial \hat{y}} = 0 \quad (1)$$

Equation of momentum in the x direction:

$$\hat{u}\frac{\partial \hat{u}}{\partial \hat{x}} + \hat{v}\frac{\partial \hat{u}}{\partial \hat{y}} = -\frac{\partial \hat{p}}{\partial \hat{x}} + \left(\frac{\partial^2 \hat{u}}{\partial \hat{x}^2} + \frac{\partial^2 \hat{u}}{\partial \hat{y}^2}\right) \quad (2)$$

Equation of momentum in the y direction:

$$\hat{u}\frac{\partial \hat{v}}{\partial \hat{x}} + \hat{v}\frac{\partial \hat{v}}{\partial \hat{y}} = -\frac{\partial \hat{p}}{\partial \hat{x}} + \left(\frac{\partial^2 \hat{v}}{\partial \hat{x}^2} + \frac{\partial^2 \hat{v}}{\partial \hat{y}^2}\right) + \left(\frac{Ra}{\Pr}\theta\right) \quad (3)$$

Heat energy equation:

$$\hat{u}\frac{\partial \theta}{\partial \hat{x}} + \hat{v}\frac{\partial \theta}{\partial \hat{y}} = \frac{Ra}{\Pr}\left(\frac{\partial^2 \theta}{\partial \hat{x}^2} + \frac{\partial^2 \theta}{\partial \hat{y}^2}\right) \quad (4)$$

The set of dimensionless coordinates in the above governing mathematical equations can be expressed as follows:

$$\hat{x} = \frac{x}{L} \; and \; \hat{y} = \frac{y}{L} \quad (5)$$

$$\hat{u} = \frac{uL}{\lambda} \; and \; \hat{v} = \frac{vL}{\lambda} \quad (6)$$

$$\hat{p} = \frac{\bar{p}L^2}{\rho\lambda^2}, \; \bar{p} = p + \rho_c g y \quad (7)$$

$$\theta = \frac{T - T_c}{T_h - T_c} \quad (8)$$

$$Ra = \frac{g\beta L^3 (T_h - T_c)}{\lambda\alpha} \quad (9)$$

$$\Pr = \frac{\lambda}{\alpha} \quad (10)$$

$$\beta = -\left(\frac{1}{\rho}\right)\left(\frac{\partial \rho}{\partial T}\right)_p \quad (11)$$

$$Nu_x = \frac{-\partial \theta}{\partial \hat{y}} \quad (12)$$

The boundary conditions of all walls square enclosure are $\hat{u} = \hat{v} = 0$ for hot and cold walls

$(\theta = 1 and \theta = 0)$ respectively, hence for thermally insulation walls $\frac{\partial \theta}{\partial \hat{x}} = 0$ or $\frac{\partial \theta}{\partial \hat{y}} = 0$.

4. Method of Numerical Model

For the analysis and display of materials, temperature distribution, and natural convection heat transfer inside the square cavity, the commercial Multiphysics program COMSOL 6.0 was used, which is based on the finite element method of solving all the governing equations of flow. The finite element method provides a high level of versatility based on the components used for foundation work and separation of spaces. For solving computational fluid dynamics problems, the choice of the optimal mesh for the computational domain is necessary. It requires the use of many cells of the grid type to achieve the best high accuracy of the numerical solution. To create the numerical domain of the two-dimensional model in Cartesian coordinates, the number (27041) was used for the selected mesh elements, as shown in Figure (3).

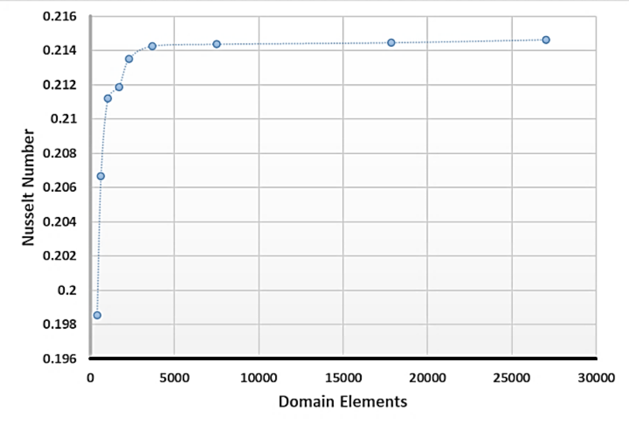


Figure 3. Variation of Average Nu with the number of elements for the computational domain

5. Results and Discussions

The characteristics of airflow and heat transfer by natural convection inside the cavity of the square were examined for different Rayleigh numbers with changing locations of the corrugated hot wall. Figure (4) indicates the change in the free convective heat transfer rate by the Nusselt number against the change of the heating positions of the left wall of the cavity at different ranges of the Rayleigh number. It can be observed that the Nusselt number gradually increases with increasing Rayleigh numbers and also with increasing the area of the heated wall due to the increase in the strength of the buoyancy forces of the high Rayleigh numbers. When using three heating sites along the left wall, the heat transfer rate improved compared to the first and second sites.

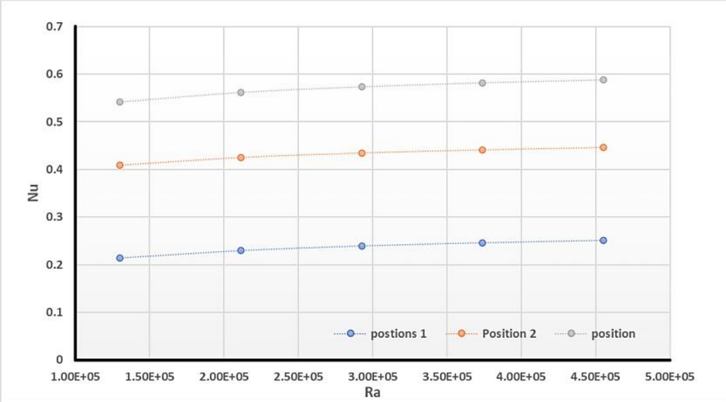


Figure 4. variation of Nu with Ra for different positions of the corrugated heated wall.

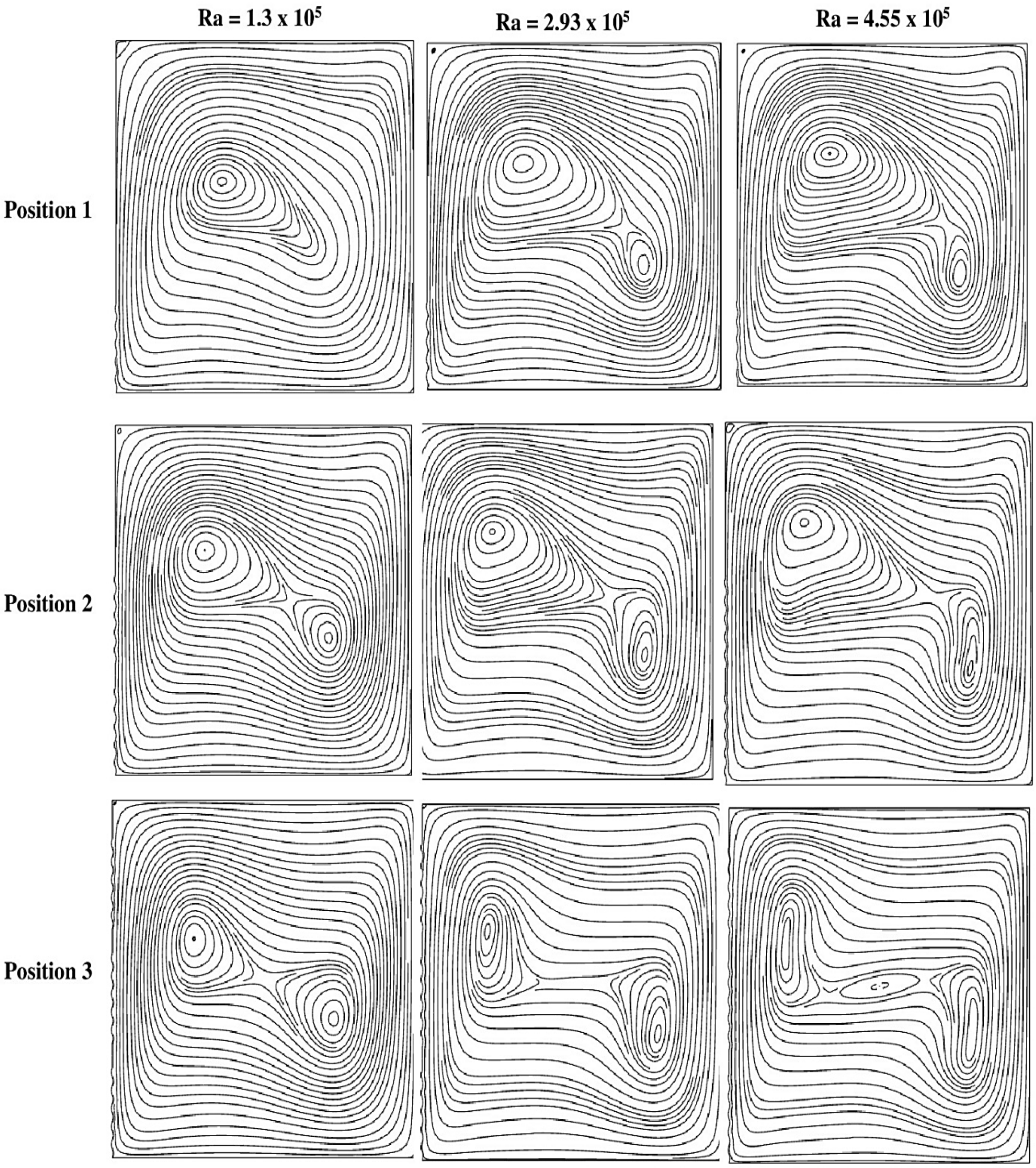
Figure (5) shows some numerical results for the air flow fields inside the square cavity from the simplification of the lines. At low Rayleigh numbers, it can be observed that the flow lines do not change significantly when changing the location of the corrugated hot wall, while with an increase in the Rayleigh number, the nature of the flow lines of the air is affected by the change, moreover, as a result of the predominance of the viscosity force over the buoyancy force at a low Rayleigh number, the pattern of rotation inside the cavity is weak. The apparent peak shifts toward the corners of the hot and cold walls as the recirculation patterns get more intense and the cells take on asymmetrical forms. The buoyancy force has also been observed to increase as the Rayleigh number increases and becomes more powerful. The corners of the hot and cold walls of the cavity are the place where the closest air flow patterns move. The flow lines of the air inside the cavity increase and converge by changing the positions of the hot wall.

Figure (6) indicates equal temperature lines inside the square cavity, where for all the changes observed in the locations of the hot corrugated wall, the temperature lines tend to be parallel to the walls exposed to heat and cold, and also perpendicular to the thermally insulated walls. At low Rayleigh numbers, the conduction heat transfer process is the predominant method for this case, where temperature fronts penetrate deeply into the body of the liquid from the side of the corrugated hot wall. Where the nature of the temperature flow lines does not change significantly. Increasing the range of Rayleigh numbers causes an increase in the concentration of temperature line patterns near the wall with a cold temperature, this is an indicator of the control of convective heat transfer against heat transfer through conduction. Therefore, the most affected part is the hot corrugated wall. By increasing the Rayleigh numbers changing the heating positions of the corrugated wall and making it along the total length, the concentration of air heat flow lines near the cold wall increases.

6. Conclusions

A numerical investigation is done on the influence of a partially heated side wall on laminar natural convection heat transfer in a square cavity at different Rayleigh numbers. The flow and temperature field findings exhibit distinct behaviors at different heated wall and Rayleigh number locations. No noticeable changes in the flow field of the airflow lines inside the cavity at low Rayleigh numbers. Moreover, as a result of the increase in buoyancy force, different configurations of flow lines are observed at high Rayleigh numbers.

The intensity of the flow field either stays constant at low Rayleigh numbers or increases at high Rayleigh numbers in proportion to the position of the heated side wall. Similar contours at low Rayleigh numbers and unique contours at high Rayleigh numbers are indicated by the behavior of temperature lines at different positions along the heated wall. In general, the average temperature in the square enclosure rises as the position heated wall climbs to its maximum and then falls at all Rayleigh numbers. When changing the heating positions of the corrugated wall the temperature variation appears along the central line of the square cavity with Rayleigh number as well as for the local Nusselt number appears along the wall exposed to a cold temperature. Finally, the average Nusselt number gradually increases by increasing the Rayleigh number also with increasing positions of the left hot wall of the square cavity.



-Figure (5): Flow field of three variable positions of the heated wall cavity at different ranges of the Rayleigh number

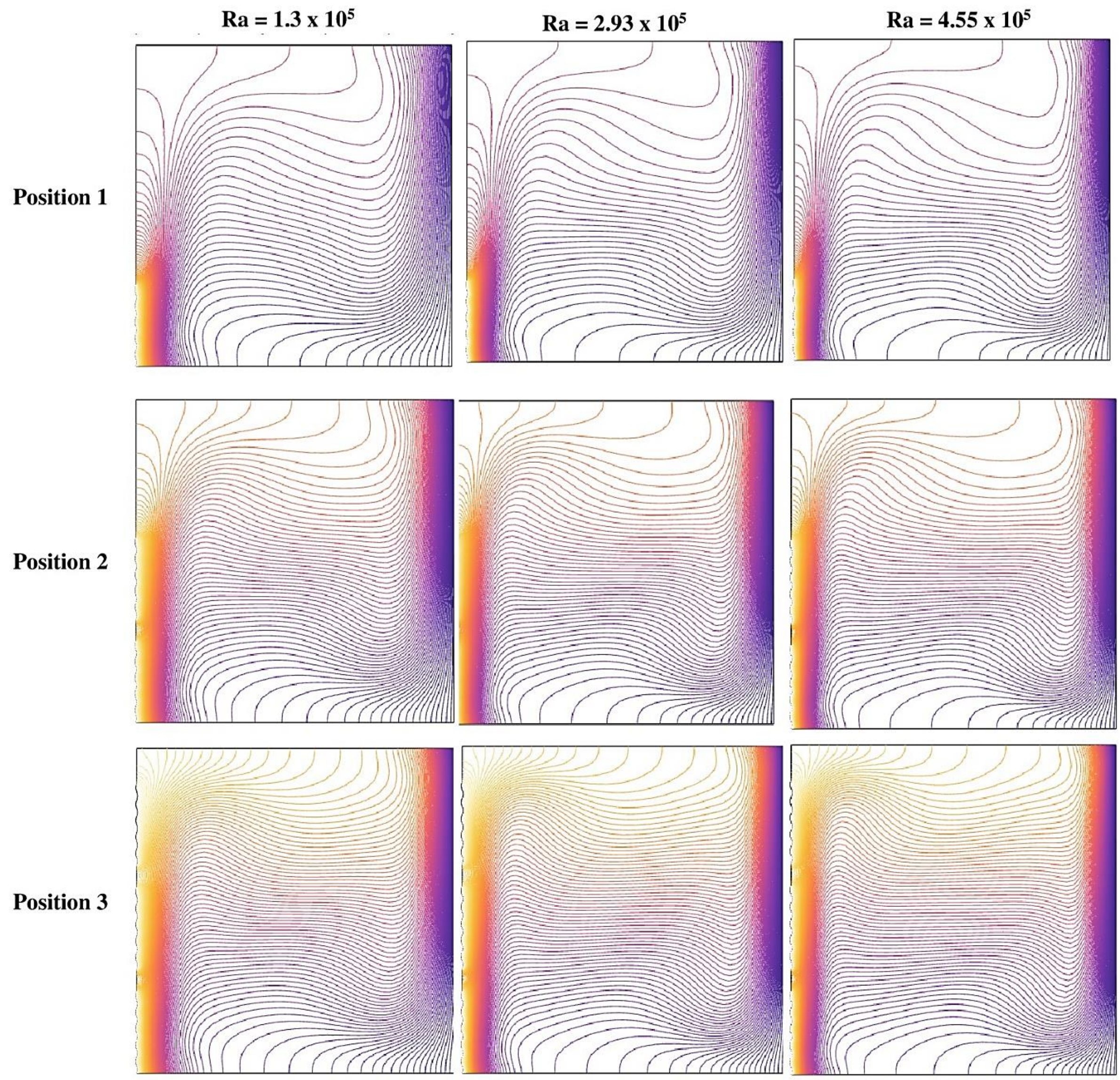


Figure (6): Temperature contour of three variable positions of the heated wall cavity at different ranges of the Rayleigh number

Nomenclature

G	Gravity acceleration ($m.s^{-2}$)
L	Square cavity length (m)
Nu	Nusselt number (dimensionless)
P	Fluid pressure ($N.m^{-2}$)
Pr	Prandtl number (dimensionless)
Ra	Rayleigh number (dimensionless)
T	Fluid temperature (K)
U	x-direction fluid velocity ($m.s^{-1}$)
V	y-direction fluid velocity ($m.s^{-1}$)
x and y	Component of fluid velocity (dimensionless)

Greek Symbols

θ	Dimensionless temperature difference
Α	Fluid thermal diffusivity ($m^2.s^{-1}$)
β	Fluid thermal expansion coefficient (1/K)
Λ	Fluid kinematic viscosity ($m^2.s^{-1}$)
Ρ	Fluid density ($kg.m^{-3}$)

Subscript

C	Cold
H	Hot
X	local

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