Natural Convection in a Square Enclosure with Partitions

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Abstract *– In this paper numerical computation of the free convection in a square enclosure with partitions is investigated using control volume based computational procedure. The cavity is bounded by uniformly heating left vertical surface, right cold wall and the top and lower horizontal walls are adiabatic. The numerical simulations used in the current study yields persistent performance over a extensive range of parameters, Ra ($10^3 \leq Ra \leq 10^6$, Stature of partition $0 \leq H \leq 0.4$ and $Pr = 0.7$) with respect to constant heating and cold vertical walls. The reality inside the cavity with centrally situated and vertically altered partitions are also investigated through temperature contours, stream function pattern, local and average Nusselt number. It is noticed that the overall heat transfer increases with increase of Ra. The conduction is dominated for a Rayleigh number up to 5×10^3 and increases further with increase of stature of partitions vertically. It is noticed that when stature of centrally located partition increased beyond a stature of 0.3H, the heat transfer decreased drastically for all Rayleigh numbers. The correlations between Rayleigh number and average Nusselt number are dealt in the form of power law.*

Keywords - Free convection; Heat transfer; Square cavity; Partitioned cavity; Rayleigh number

I. INTRODUCTION

The analysis of flow of fluid and transfer of heat has occupied the centre stage in many industrial and engineering applications. However, from one decade, awareness on free convection in a closed cavity with baffles is increased very rapidly. The computations of heat and flow of fluid play an vital role in the field of engineering such as the card or any projections on mother board of a computer, the placement of electronic components inside the electronic cabinet, the plastic coated baffles in a refrigerator, and wooden partitions in an air conditioned rooms. Protruded instruments used for intrusive measurement can also considered as insulated partitions. The flow and thermal field is caused due to density difference. The internal buoyancy flows are more complex, due to the essential coupling among transport properties of thermal and flow fields. A brief review of the pertinent literature is conferred.

The experimental and computational studies on natural convection in rectangular enclosures with or without partitions are given. Computations have been carried out with or without openings of the cavities [1, 2]. In these studies, the investigations have been performed for Rayleigh number ranging from 10^4 - 10^{11} and $Pr = 0.7$. The turbulent boundary layer formations occur near the ceiling and vertical walls. Guo and Sharief [3] have computed, heat transfer caused by mixed convection in a 2D rectangular enclosure with uniform heat flux from discretely heated bottom surface with vertically operating isothermal side walls. For their study, the lengths of heat source and cavity aspect ratio are varied. It is identified that the average Nusselt number rises as the heat source is moving close to isothermal side walls. The mixed convective transfer of heat and fluid flow in a shallow cavity with series of heat generating blocks for ranges of Reynolds and Grashoff numbers and block-to-fluid thermal conductivity ratios is performed by Bhoite [4]. It is noticed that the recirculation region occurs in the core for $Re > 600$. Also, they observed that the cause of buoyancy force is insignificant.

Mahapatra et al. [5], have conducted a numerical simulations on mixed convection with radiation effect. The work concedes that radiation has substantial bearing on free convection and it is managed to flow turnaround in the middle of the enclosure. After thorough literature, the authors feel to outline a few prior research works defining the phenomenon of natural convection in partitioned cavities. Numerical simulations on laminar free convection in a 2D square enclosure filled by air with a vertical baffle of finite thickness carried out by Mezrhab and Bchir [6]. The computational model is bounded by adiabatic horizontal end surfaces, and vertical surfaces at different temperature. Calculations including shadow effects have been carried out using finite volume formed computational procedure. It is noticed that there is no significant modifications on transfer of heat through the enclosure, especially at large Rayleigh numbers. Belgen [7] has carried out numerical simulations on laminar and turbulent free convection in a cavity including baffles with vertical surfaces of which isothermal cooling and horizontal end surfaces are adiabatic.

Nomenclature

g	Gravity acceleration, m s^{-2}	X	dimensionless length along x co-ordinate
k	thermal conductivity, W m^{-1} K^{-1}	Y	dimensionless length along y distance
H	stature of the square cavity, m	*Greek symbols*	
L	length of the square cavity, m	α	thermal diffusivity, m^2 s^{-1}
Nu	local Nusselt number	β	volume expansion co-efficient, K^{-1}
$\overline{Nu}$	average Nusselt number	ε	Position of the partition, m
P	dimension less pressure	θ	dimensionless temperature
Pr	Prandtl number	γ	kinematic viscosity, m^2 s^{-1}
Ra	Rayleigh number	ρ	density, kg m^{-3}
t	thickness of the partition, m	ψ	stream function
T	temperature, K	*Subscripts*	
U	dimensionless velocity in x - direction	c	cold wall
V	dimensionless velocity in y - direction	h	hot wall

It is noticed that heat transfer reduces, when more than one partition, cavity aspect ratio made short and position of location of partition is further away from the heating wall. Work on numerical simulations of opposing mixed convection in variously heated square cavity with partitions is reported by Mahapatra et al. [8]. Simulation are performed by varying Richardson number (Ri = 1 and 0.1), partition height 0.1 to 0.55 and for Ra = 10^4 only. It is found that when partition stature greater than 0.3H, the transfer of heat in opposing mixed convection case is higher than that of free convection. Recently, Ghazian et al. [9] conducted experimental studies on free convection in a enclosure with partitions at various angles. It is noticed that at each Rayleigh number, the optimum angle of inclination of partition exists which minimizes the rate of overall heat transfer. The effect of partition on thermo magnetic free convection and entropy generation in inclined enclosure with porous media is numerically simulated by Heidary et al. [10]. It has been pointed out that magnetic field, rotation of cavity and partitions can control the transfer of heat and entropy generation. However, the free convection in a square enclosure with partitions in the past, the open literature available on simple partitions placed inside enclosure is limited to low Rayleigh numbers only. The numerical simulations on laminar natural convection in a square enclosure for offset of partitions, position and height of the partitions are not reported in the open literature.

The main aim of the present numerical investigation is to study the shielding effect of baffles located at centre and offset from the adiabatic horizontal surfaces of a square enclosure. The height ($0.1H \geq h \geq 0.4H$) and location ($0.3 \geq \varepsilon \geq 0.7$) of the partition is varied by keeping thickness of partition constant (see Fig. 1). The simulations are carried out for different Ra ($10^3 \geq Ra \geq 10^6$), Pr = 0.7 for horizontal adiabatic walls and isothermal vertical walls.

II. Mathematical Formulations

Fig. 1 illustrates the schematic representation of the physical model. The computations are carried out with the assumptions such as the flow of fluid is 2 dimensional, incompressible, steady and laminar. The reference temperature is kept constant for the thermal properties of the fluid so as to couple the field of isotherms to fluid flow regions. However, the Boussinesq approximation is entreated for the of the momentum equation.

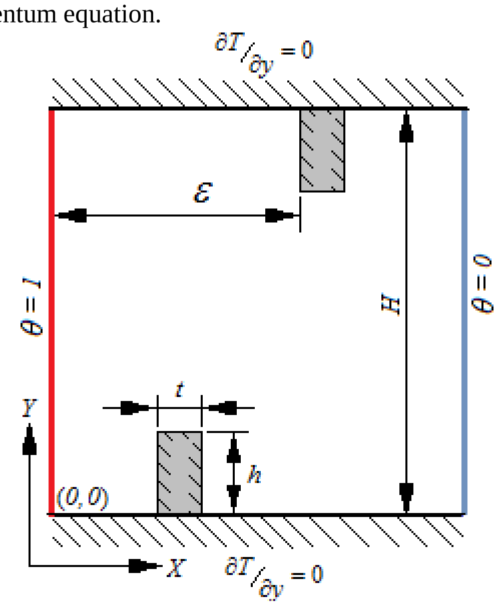

Fig. 1: Computational domain.

In order to write governing equations in the non-dimensional form, the transformations of the variables have been carried out as given by;

$$\left.\begin{aligned} & X=\frac{x}{H},\ Y=\frac{y}{H},\ U=\frac{uH}{\alpha},\ V=\frac{vH}{\alpha}, \\ & P=\frac{pH^2}{\rho\alpha^2},\ \Pr=\frac{\gamma}{\alpha},\ \theta=\frac{T-T_C}{T_h-T_C} \\ & Ra=\frac{g\beta(T_h-T_C)H^3\Pr}{\gamma^2} \end{aligned}\right\} \quad (1)$$

Where, X and Y are the non-dimensional coordinates with regards to x and y - axes respectively; the dimensionless velocities in X and Y co-ordinates are U and V respectively; the dimensionless pressure is P; Pr and Ra are the Prandtl number and Rayleigh number, respectively.

The free convective flow of fluid and heat transfer inside the partitioned enclosure is demonstrated using the continuity, momentum and energy equations are given below;

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

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

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

$$U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2} \tag{5}$$

The control volume based computational procedure is used to work out governing equations (2 to 5) exposed to the boundary conditions (shown in the computational domain) built in ANSYS FLUENT software [11]. The solution domain is discretized in to finite number of quadrilateral cells. The numerical solution is obtained by integrating the governing equations over each cell (control volume).

The mathematical model shown in Fig. 1 having the following barriers;

$$\left.\begin{array}{l} u(x,0)=u(x,H)=u(0,y)=u(H,y)=0 \\ v(x,0)=v(x,H)=v(0,y)=v(H,y)=0 \\ \dfrac{\partial T}{\partial y}(x,0)=\dfrac{\partial T}{\partial y}(x,H)=0 \\ T(0,y)=T_h,\ T(H,y)=T_C \end{array}\right\} \tag{6}$$

III. Valuation of Stream Function Contours and Rate of Heat Transfer

A. Stream Function

The components of velocity U and V are used to find the streamfunction ψ which is display the motion of the fluid inside the control volume. The correlation among the velocity components and streamfunction dealt by Batchelor [12] are given below:

$$U=\frac{\partial \psi}{\partial Y} \text{ and } V=\frac{\partial \psi}{\partial X} \tag{7}$$

Which turnout a single correlation

$$\frac{\partial^2 \psi}{\partial X^2}+\frac{\partial^2 \psi}{\partial Y^2}=\frac{\partial U}{\partial Y}-\frac{\partial V}{\partial X}. \tag{8}$$

The stream function defined in equation (8) is used, the positive and negative sign of ψ is represented by counter-clockwise circulation and clockwise circulations.

B. Local Heat Transfer Rate

The local heat transfer rate is obtained from the gradients of heat transfer;

$$Nu=-\frac{\partial \theta}{\partial n} \tag{9}$$

Where, 'n' is the normal to a plane from which the local Nusselt number is extracted. The overall transfer of heat for heating wall is achieved by;

$$\overline{Nu}=\int_0^H Nu\ dY \tag{10}$$

The trapezoidal rule is enforced to integrate the above equation.

IV. Grid Test

The governing expressions are discreted by finite volume procedure provided in ANYSS FLUENT software. In order to justify the predictive compatibility and accuracy of the present methodology, the numerical computations on free convective heat removal in a trapezoidal cavity with partitions carried out by Moukalled and Darwish [13] is selected. In this study, the physical domain has a length 4 times the tallness (H_S) of the shorter vertical surface, the inclined adiabatic top surface is 15^0, vertical baffles of three different stature (H_b= H_S/3, 2H_S/3 and H_S), two baffle position ($L_b = L/3$, 2L/3) and thickness of the baffle $W_b = L/20$. For convergence study, a baffle height $H_b = 2H_S/3$, location of the baffle $L_b = L/3$ and $Ra = 10^6$ is preferred.

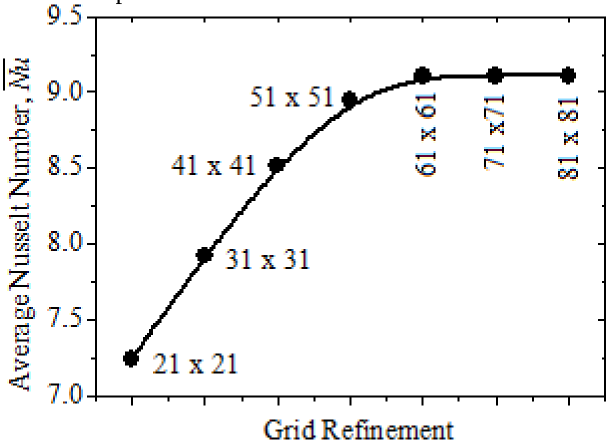

Fig. 2: Grid convergence for $Ra = 10^6$

The different sizes of grid sizes ranging from 21 x 21, 31 x 31, 41 x 41, 51 x 51, 61 x 61, 71 x 71 and 81 x 81 are used for the computation. The variations of average Nusselt numbers are demonstrated (in Fig. 2) for the specified configuration of the cavity. It is observed that $\overline{Nu}$ for 21 x 21 is about 7.28 and is rising as the grid is fine and reaches to a stable value for grid sizes of 61 x 61. Further, this value is almost constant for next sizes 71 x 71 and 81 x 81. Due to this, a grid of 61 x 61 is used for further computations.

Table 1 Comparison of average Nusselt number with the values of [13] for a baffle height, $H_b = 2H_S/3$, Location $L_b = L/3$ and $Ra = 10^6$.

Ra	$\overline{Nu}$ Published [13]	$\overline{Nu}$ Present	% error
10^4	0.503	0.504	0.198
10^4	1.131	1.133	0.176
10^5	3.557	3.548	0.224
10^6	9.106	9.091	0.219

The computations of $\overline{Nu}$ for Rayleigh numbers, Ra ($10^3 \leq Ra \leq 10^6$) with present code has been computed with baffle height, $H_b = 2H_S/3$ and Location of the baffle, $L_b = L/3$. IN the table 1, the obtained values have been compared with that of Moukalled and Darwish [13]. It is seen that there is good agreement with maximum error incurred between two are less than 1%.

V. Results and Discussions

The effect of baffle height and its location on natural convective phenomenon inside the partitioned square cavity by differentially heated vertical walls is carried out by control volume based numerical procedure. The thickness of the adiabatic partition (i.e. $t = 0.1H$) is maintained constant. The temperature patterns, stream function contours, local and overall heat transfer are used for the analysis.

A. *Effect of Rayleigh Number*

The free convective phenomenon within the differentially heated square cavity is demonstrated by *Ra*, partition height (*h*) and location of the partition (ϵ).

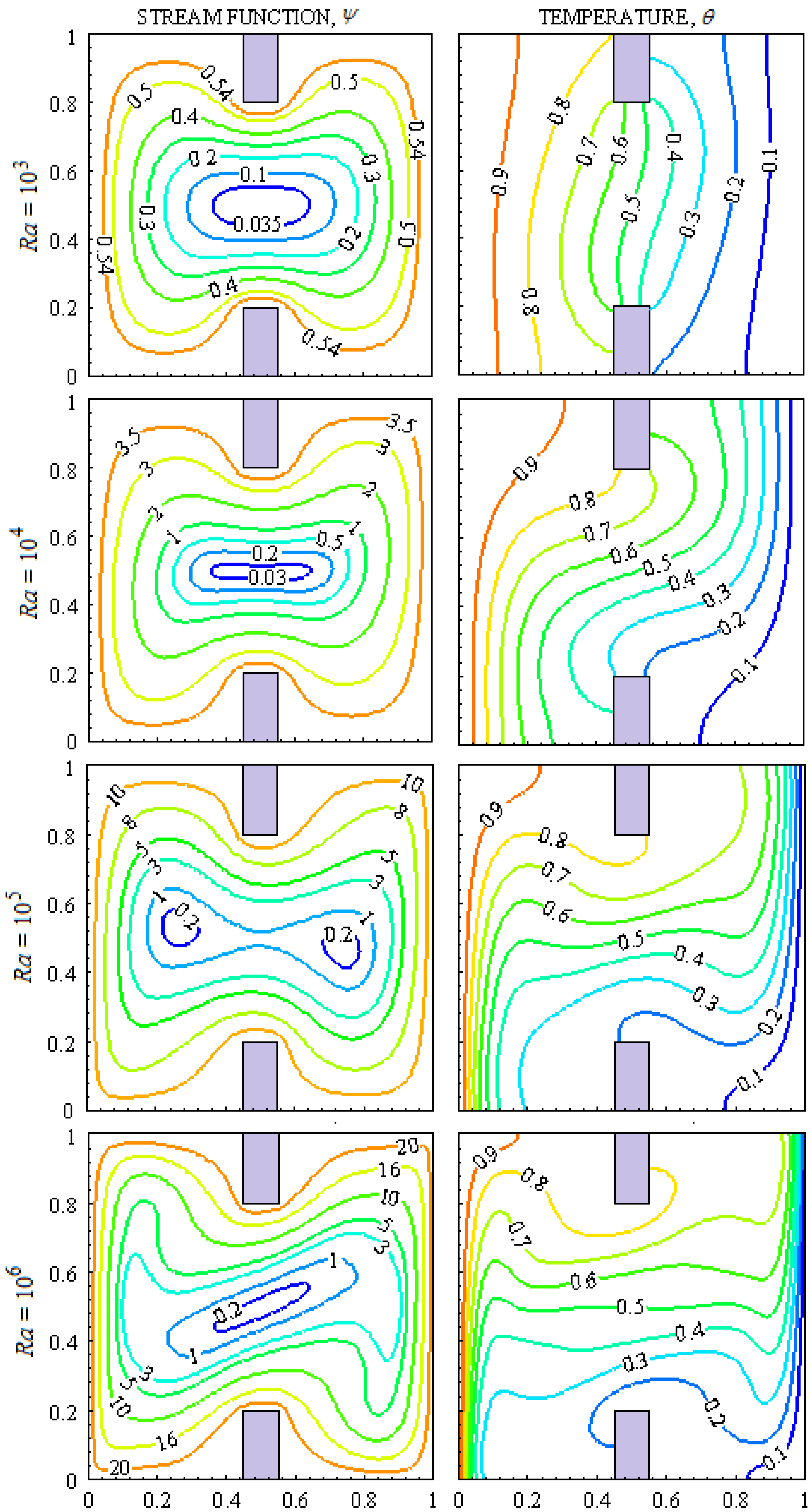

Fig. 3: Contour plots for different Rayleigh number of case 1.

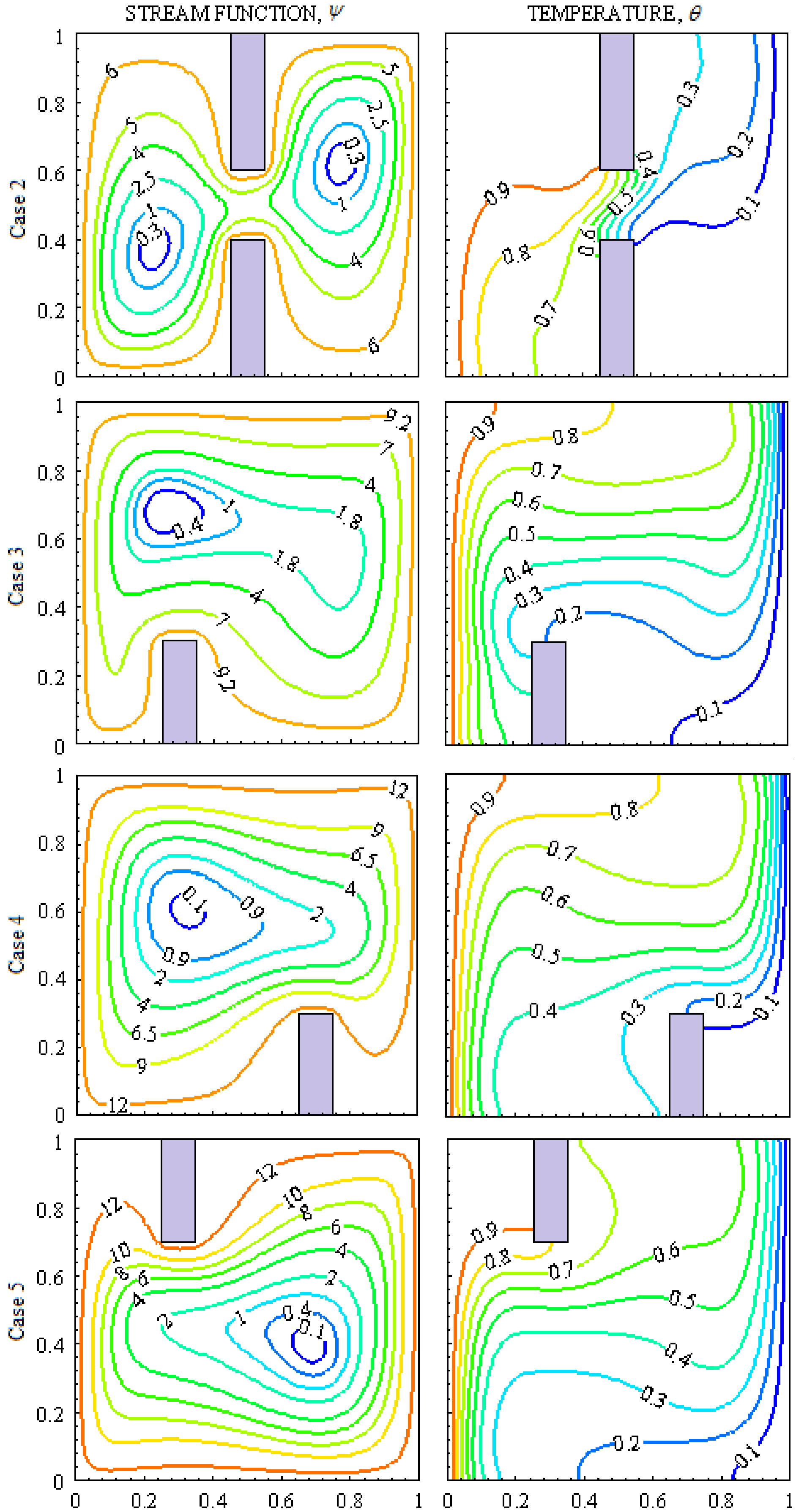

Fig. 4: Contour plots for cases 2 to 5 with Rayleigh number Ra = 10^5.

In the first case, the two partitions of height h = 0.2H are placed at adiabatic wall symmetrically with vertical centre line as shown in Fig. 3. The curves of stream function and isotherms of computational solutions for various Rayleigh numbers ranging from Ra = 10^3 to 10^6 and Pr = 0.7, when the left vertical wall is heated uniformly for case 1 is illustrate in

Fig. 3. As expected, due to uniform heating and cooling of vertical walls, the fluid starts to rise up from bottom end to top end of the heating surfaces and flow downward direction along the constantly cooled right vertical surfaces. The presence of partitions at the top reduces the magnitude of the flow of fluid and deflects in the downward direction. Again, fluid flows upward as soon as it comes to lower end of the partition and then reaches to the upper end of the cold surface. Due to the existence of partition at the lower side of the cavity, the cold fluid coming out of the cold wall moves up and down before it reaches to hot wall forming rolls with clockwise rotation. At $Ra = 10^3$, the values of the flow field is very small and the transfer of heat occurs essentially by conduction dominated mode. Due to symmetry in partitions, the flow fields are symmetry with respect to horizontal and vertical symmetry lines with hither magnitudes at the outer one and stagnant ($\psi = 0$) at core. The core starts deformed horizontally, due to the existence of partitions at the top and lower ends of the cavity.

The isotherm contours which are very close to the hot surfaces ($\theta = 0.9$) and cold surface ($\theta = 0.1$) are almost parallel to the vertical wall. However, the isotherm contours, θ ($0.8 \geq \theta \geq 0.2$) are smooth curves and surround the partitioned walls. The isotherm contours (which are not shown) are not varying with Rayleigh number up to $Ra < 5 \times 10^3$. The circulations become stronger at the extreme outside the core of centre of rotations and consequently, the isotherm contour of $\theta = 0.1$ is smooth curve with top portion shifting towards the partition and lower portion near the hot wall. The upper part of isotherms with values $\theta \geq 0.5$ is deformed towards cold wall. As Rayleigh number rises to 10^5, the convection current caused by buoyancy force within the cavity, the values of the flow of fluid increases. The central core is bifurcated into two eyes and is placed at the centre of the two halves of the cavity. Consequently, due to the presence of partition at the bottom, isotherm contours $\theta \leq 0.7$ distorting against the top side of the cold surface. In contrast, the lower partition causes $\theta \geq 0.2$ deformed towards the lower portion of the hot wall. At $Ra = 10^6$, the convection current is more dominated and hence the values of stream functions are doubled to that of $Ra = 10^5$. The stratification of the isotherm contours is more and more and covers 80% of the vertical walls. The central portion of the contours is almost horizontal. The portion of stream function contours placed in the left half of the cavity starts deforming upward direction and same thing happening downward in the right 50% of the cavity. However, due to higher stratification rate of isotherm contours near the vertical walls and strong convective mode of heat transfer the central core reunite and starts bulging inclined with left end lower and upper at the other.

B. Effect of Partition Stature

In the second case, height of the partitions is doubled (i.e. $h = 0.4H$) as compared to first case. The fluid flow and isotherm contours are predicted for case 2 is shown in Fig. 4. The 20% of the height of the cavity is remains open at the centre of the vertical symmetry. The warm fluid rising along the hot wall pushes the fluid stagnant adjacent to left side of the top corner of the top partition to downward. On contrary, the stagnant fluid at the bottom portion of the right 50% of the enclosure is pushed by downstream of the cold fluid. Consequently, it is observed from the isotherm contours that the stagnation of the hot fluid ($\theta \geq 0.9$) rises to 40% length of the left top portion of the enclosure and cold fluid is stagnant in the right lower end of the enclosure. The upward stream of the condensed fluid pushes up the hot fluid entered in to the right portion of the cavity and hot fluid pushing the cold fluid coming from right at the entry in the down ward direction. The nuclei is split in to two and placed one at the lower left side of enclosure and other at the top-right half of the enclosure.

C. Effect of Position of Partition

The effect of different position of a single partition (Case 3, 4 and 5) is shown in Fig. 4. In the 3rd case, a partition of height $h = 0.2H$ is placed at a distance, $\epsilon = 0.3L$ at the bottom horizontal wall. It is seen from the fluid flow contours for $Ra = 10^5$ that the nuclei of rotation of the fluid is almost a height of $0.7H$ and exactly above the partition. The temperature contours are smooth curves, the contour starts from $\theta = 0.7$ to $\theta = 0.3$ spreads the entire cavity. Due to partition near the warmed surface, the temperature contour near the hot surface is almost straight line up to 80% of the height of the enclosure and contours adjacent to cold wall are straight upto 50 to 60% from the top. The partition is shifted from 0.3L to 0.7L (case 4) in the bottom adiabatic wall (near cold wall). The velocity of the cold fluid which comes in contact at the bottom side of the hot surface is more. Hence, the magnitude of the stream function is increased by 30% as compared to case 3. Fig. 4 shows the position of the partition, stream function curves and temperature contours. The eye of the fluid flow is shifting right bottom corners as compared to case 3 and 4. However, except the deflection of the stream contours near the partition, the magnitude and behavior of the stream functions are same in the cases of 4 and 5. The location of the partition near the warmed and cold surfaces, the stratification rate of temperature contours of case 4 are opposite in nature as compared to case 5.

D. Heat Transfer Rate; Local and Overall Heat Transfer Rate

Figs. 5 and 6 show the variations in local heat transfer across the hot vertical surface for distinct Rayleigh numbers and different positions and height of partitions respectively.

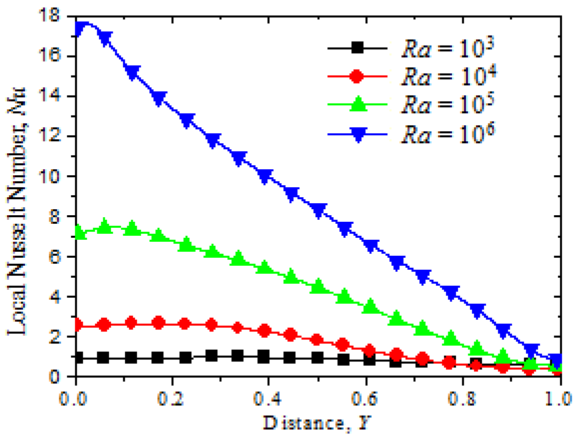

Fig. 5: Variations in local Nu along the hot surface for different Ra of case 1.

The effect of Ra on local heat transfer along the warm surface for case 1 is plotted in Fig. 5. At $Ra = 10^3$, the transfer of local heat is primarily by conduction dominated mode. In contrast, the magnitudes of stream function curves are very small and the stratification of temperature contours are around the partitions. Hence, the local heat transfer is almost straight line. The significant of convection occurs at $Ra \geq 5 \times 10^3$. The magnitude of the fluid flow is very small near the corners. The increasing and decreasing behavior is noticed at the beginning and end of the local heat transfer along the warmed and chilled surfaces.

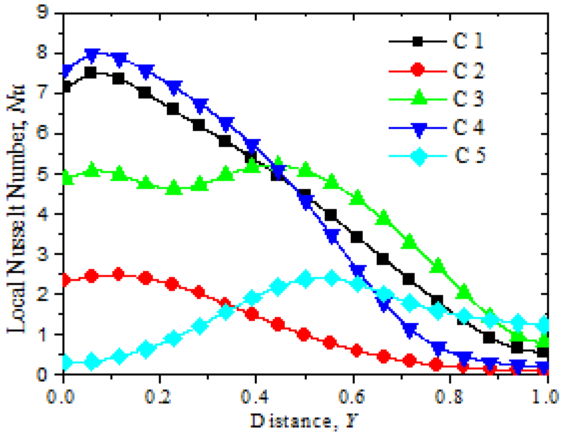

Fig. 6: Variations of local *Nu* along the warm surface for different cases at $Ra = 10^5$.

The Nusselt number increases with increase of *Ra*, The temperature contours are concentrating at left bottom corner and pushes upward along the vertical hot surface and it is downward in the right side cold wall. At higher *Ra*, the local Nusselt numbers are almost inclined straight lines except ends.

The variations in transfer of local heat along the warmed surface for different positions and height of the partitions along the adiabatic walls at $Ra = 10^5$ is illustrated in Fig. 6. In case 1, the local number increases and decreases and it is higher for 40% of length of heating wall. Further, it starts decreasing to lower values at the upper corner to the values that of case 2. It is noticed that the local heat transfer decline with increase of partition height either from top or bottom or from both and decrease with decreasing location of partition ϵ.

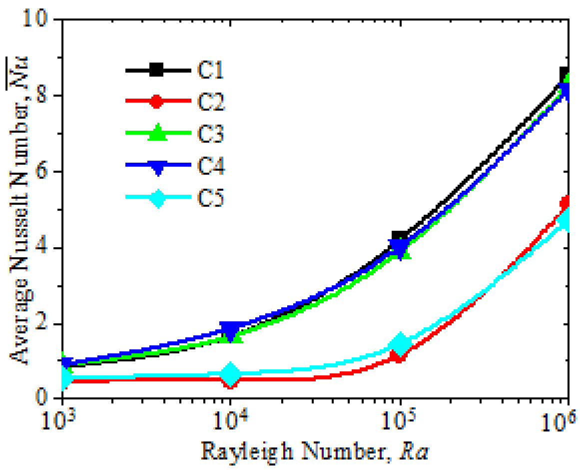

Fig. 7: Fluctuations of average Nusselt number with Rayleigh numbers along the warmed surface for various statures of partitions.

The effect of location and stature of the baffles on overall heat transfer rate are illustrated in Fig. 7, where the deviations of average Nusselt numbers are plotted against logarithmic *Ra* for heating wall. It is noticed from the plot that the overall heat transfer rate monotonically increases with increase of *Ra*. It is seen that overall heat transfer rate remains same up to $Ra \geq 5 \times 10^3$ for cases 1, 3 and 4. However, this is extended up to $Ra = 10^4$ for the case 4 and 5. As the partition height increase from 0.2*H* to 0.4*H* and length of location ϵ decreased to 0.3*L*. The overall heat transfer is higher at all the points for cases 1, 3 and 4 compared to case 2 and 5. The log-log linear path is developed with more than 25 points in the convection dominated regions. The least square is adopted with overall percentage error less than 1%. The power law equations are developed for convection dominated regimes for all the cases considered for the computations and tabulated in the table 2.

Table 2 The Power law correlations with regression coefficient for a square cavity with partitions.

Case	*Ra* range	Correlation	R^2
C1	10^3 - 10^6	$Nu = 0.072\ Ra^{0.347}$	0.999
C2	10^4 - 10^6	$Nu = 0.005\ Ra^{0.492}$	0.999
C3	10^3 - 10^6	$Nu = 0.082\ Ra^{0.332}$	0.998
C4	10^3 - 10^6	$Nu = 0.094\ Ra^{0.324}$	0.999
C5	10^3 - 10^6	$Nu = 0.094\ Ra^{0.324}$	0.999

Closing Remarks

The prime investigation of the present work is to study the effect of Rayleigh number, position and stature of the partitions on natural convection with in a square cavity. The following conclusions have been noted from the present computations.

a) The lower values of local Nusselt number are achieved along the hot surface for case 2 and 5, as compared to the rest of the cases.
b) The average Nusselt number rises monotonically with increase of *Ra*.
c) The conduction domination mode is observed upto $Ra \geq 5 \times 10^3$ for cases 1, 3 and 4, whereas for case 2 and 5 extended upto $Ra = 10^{4.}$
d) In comparison, the overall transfer of heat is maximum at all the points for cases 1, 3 and 4, that of cases 2 and 5.
e) It is noticed that the overall transfer of heat reduces with increase of partition height and decrease in the length of location of the partition.

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