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Convection in a Tilted Square Enclosure with Various Boundary Conditions and Having Heat Generating Solid Body at its Center

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In this study free convection flow and heat transfer of a fluid inside a tilted square enclosure having heat conducting and generating solid body positioned in the center of the enclosure with various thermal boundary conditions has been investigated numerically. The governing equations are transformed into non-dimensional form and the resulting partial differential equations are solved by Finite Volume Method applying power-law scheme using SIMPLE algorithm with Under-Relaxation technique. The parameters leading the problem are the aspect ratio, thermal conductivity ratio, temperature difference ratio and the angle of inclination. The effect of different thermal boundary conditions on streamlines and isotherms as well as on the rate of heat transfer on all walls of the enclosure are presented graphically.

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1. Introduction

The buoyancy-driven flow in a square enclosure with differentially heated walls is one of the least pursued areas in finite volume methods. Buoyancy-induced heat transfer in enclosed spaces has been extensively studied in the past, the study

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of heat generation in moving fluids is important from the view point of several physical considerations such as those dealing with chemical reactions and those concerned with dissociating fluids. Possible heat generation effects may alter the temperature distribution and, therefore, the particle deposition rate. Also it plays a fundamental role in variety of practical problems such as fire and combustion modelling, nuclear reactor cores, ventilation of rooms, solar energy collection, crystal growth in liquids and convective heat transfer associated with boilers, the case of the cooling of electronic assemblies, in which a large number of high power dissipating components are often packaged in modular enclosures such that space and external cooling sources are minimal. Numerical simulation of natural convection

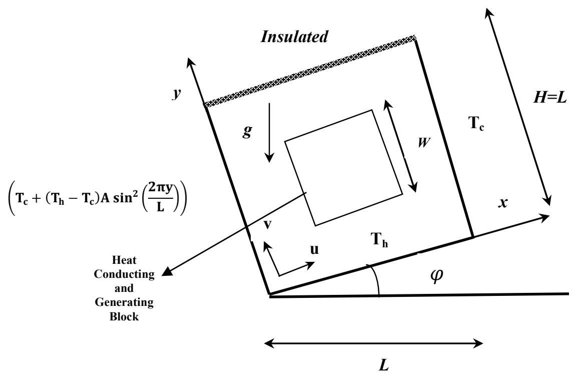

Figure 1. Schematic of the Problem

heat transfer inside enclosure was initially done by Davis (1983) which was qualitative benchmark problem of buoyancy-driven flow in an enclosed domain. Mariani and Moura Belo (2006) survayed the numerical studies of natural convection in a square enclosure.

Existence of obstacles within an enclosure was one of the interesting topic for researchers in the past two decades. One of the earliest investigations of an enclosure with obstacles was performed by House et al. (1990), they considered the influence of a centered conducting body on natural convection within an enclosure.It was observed that the case with a thermal conductivity ratio of less than unity (one) enhances the heat transfer.

Oh et al. (1997) investigated the steady natural convection in an enclosure with a heat generating conducting body numerically. They focused the effect of Reyleigh number and temperature difference across an enclosure on the fluid flow and heat transfer. Natural convective flow and heat transfer features around a heated body in an square enclosure was numerically analyzed by Roychowdhury et al. (2002).

Jeng et al. (2008) presented an experimental and numerical study of the transient natural convection due to mass transfer in inclined enclosures and suggested that the fluid concentration and streamlines vary with the angle of inclination. Aminossadati and Ghasemi (2005) numerically studied the flow and temperature fields in an inclined enclosure simulating an inclined electronic device. They resulteded that placing an enclosure at different orientations significantly affect the rate of heat transfer. Tilting the enclosure considerably affected the flow and temperature fields as well as the heat transfer characteristics of a partitioned enclosure was found by Ben-Nakhi and Chamkha (2006). Ghassemiet al.(2007) examined the effect of two insulated horizontal baffles on the flow field and heat transfer in an enclosure. It was suggested that placing baffles on the vertical walls generally causes flow and thermal modification occurs due to the blockage effect of the baffles. Also the baffles cause heat transfer reduction and convection suppression relative to the undivided enclosure at the same Rayleigh number. Kahveci (2007) studied natural convection in an enclosure partitioned with a finite thickness partition and found that the average Nusselt number decreases with an increase of the distance between the hot wall and the partition, also the thickness of the partition has a minor effect on natural convection.

Aydin and Yang (2000) numerically simulated the natural convection of air in a vertical square cavity with localized isothermal heating from below and symmetrical cooling from sidewalls. Through their investigation, average Nusselt number at the heated part of the bottom wall was shown to increase with increasing Rayleigh number as well as with increasing length of the heat source. Ganzarolli and Milanez (1995) performed the numerical study of steady natural convection in rectangular enclosures heated from below and symmetrically cooled from the sides. The size of the cavity was varied and the heat source, which spanned the entire bottom wall, was either isothermal or at constant heat flux condition.

Anderson and Lauiat (1986) studied the natural convection in a vertical square cavity heated from bottom and cooled from one side. Convection in a similar configuration where the bottom wall of the rectangular enclosure was partially heated with cooling from one side was observed by November and Nansteel (1986).

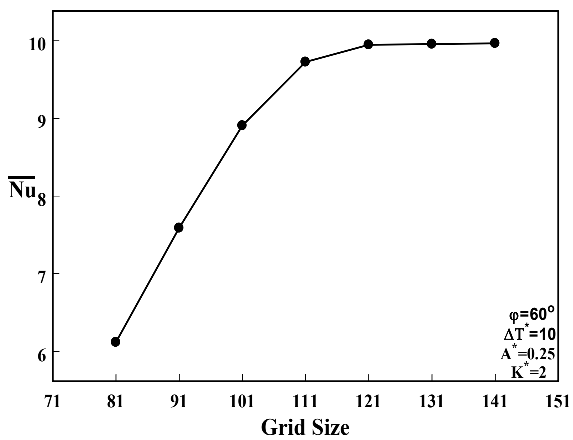

Figure 2. Grid independent test for $K^* = 2$, $\phi = 60^o$, $A^* = 0.25$ and $\Delta T^* = 10$

Natural convection around a tilted heated square cylinder kept in an enclosure with three aspects ratio has been studied by Arnab et al.(2006). They concluded that thermal stratification, flow pattern and overall heat transfer are modified against aspect ratio. Numerical investigation on laminar convective flows in a differentially heated square enclosure with a heat-conducting cylinder at its center was analyzed by Jami et al.(2006). They studied that for a constant Rayleigh number, the average Nusselt number at the hot and cold walls varied linearly with temperature-difference ratio.

Braga (2005) analyzed steady laminar natural convection within a square cavity filled with a fixed volume of conducting solid material consisting of either circular or square obstacles. He found that the average Nusselt number for cylindrical rods was slightly lower than those for square rods. Ortega (1996) experimentally investigated on natural convection air cooling of a discrete source on a conducting board in a shallow horizontal enclosure. They resulted that the heat transfer coefficients have a distinct behavior at high aspect ratio in which the dominant length scales were related to the source.

Hussain and Hussein (2010) numerically investigated the laminar steady natural convection where a uniform heat source applied on the inner circular cylinder enclosed in a square enclosure assumed with all boundaries are isothermal .They concluded that with increasing Rayleigh number, the average and local Nusselt number values increased with different up and downwards locations of the inner cylinder, due to the little influence of thermal convection.

Paramane and Sharma (2009) examined the convection heat transfer across a circular cylinder rotating with a constant non-dimensional rotation rate. They found that, the average Nusselt number decreased with increasing rotation rate and increased with increasing Reynolds number. It was also observed that, the heat transfer suppression due to rotation increased with increasing Reynolds number and rotation rate. Numerical Study of Laminar Natural Convection in Inclined Rectangular Enclosures of Various Aspect Ratios are done by Rahman and Sharif (2003).

Apart from this , present work is about the fluid flow and heat transfer in an square enclosure having heat generating body with four different thermal wall boundary conditions.

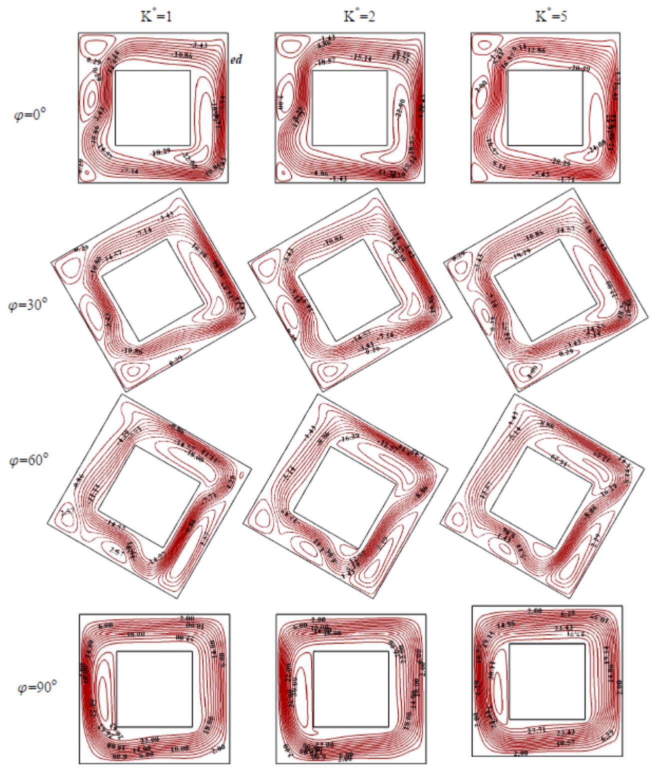

Figure 3. Streamlines for various K^* and ϕ with $A^* = 0.25$ and $\Delta T^* = 4$

2. Mathematical formulation

In the present study, we considered steady, laminar, incompressible, free convection flow in a two-dimensional tilted square enclosure of length L, which is filled with a Bousinessq fluid. A solid body of width W is placed at its center that generates heat. The geometry and the coordinate system with boundary conditions are schematically shown in Fig. 1. Both the bottom and right side walls of the tilted enclosure are maintained at constant temperature T_h and T_c ($T_h > T_c$) respectively. Left wall is sinusoidally heated $T_c + (T_h - T_c)Asin^2(2\pi y/L)$ and the top wall of

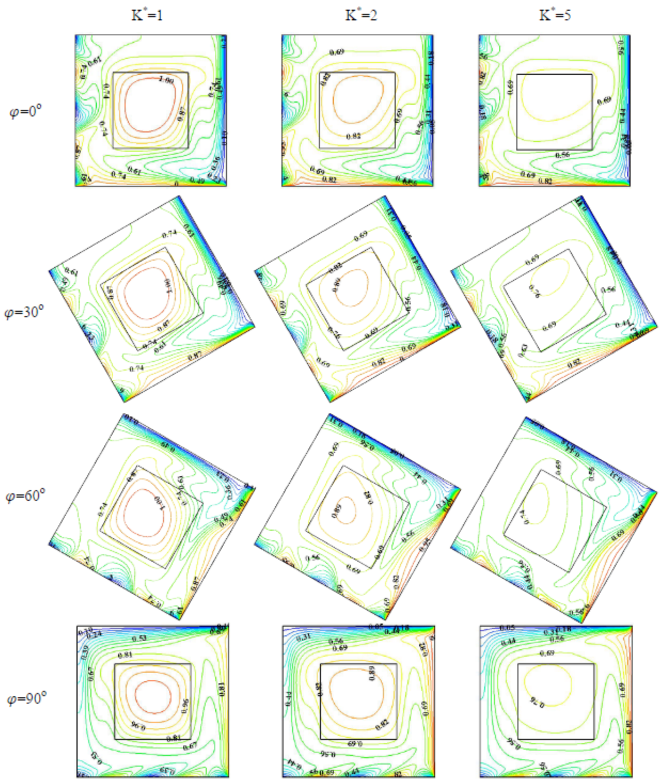

Figure 4. Isotherms for various K^* and ϕ with $A^* = 0.25$ and $\Delta T^* = 4$

the enclosure is adiabatic. Gravity acts normal to y-axis. The following mass, momentum and energy equations are obtained in dimensional form for steady natural convection process:

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

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{\rho}\frac{\partial p}{\partial x} + \nu\nabla^2 u + g\beta(T - T_c)sin\phi \tag{2}$$

$$u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} = -\frac{1}{\rho}\frac{\partial p}{\partial y} + \nu\nabla^2 v + g\beta(T - T_c)cos\phi \tag{3}$$

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \alpha\nabla^2 T \tag{4}$$

Energy equation for the solid body is,

$$K_s\frac{\partial^2 T_s}{\partial x^2} + K_s\frac{\partial^2 T_s}{\partial y^2} + \dot{q} = 0. \tag{5}$$

Dimensional form of boundary conditions are:

$$x = 0, \quad 0 \leqslant y \leqslant L: \quad u = v = 0, \quad T = T_c + (T_h - T_c)A\sin^2(\frac{2\pi y}{L}),$$
$$x = L, \quad 0 \leqslant y \leqslant L: \quad u = v = 0, \quad T = T_c,$$
$$y = 0, \quad 0 \leqslant x \leqslant L: \quad u = v = 0, \quad T = T_h,$$
$$y = L, \quad 0 \leqslant x \leqslant L: \quad u = v = 0, \quad \frac{\partial T}{\partial y} = 0.$$

At the fluid-solid interface:

$$K_f\left(\frac{\partial T}{\partial n}\right)_{fluid} = K_s\left(\frac{\partial T_s}{\partial n}\right)_{solid},$$

where n is the vector acting normal to the boundary. The above equations and boundary condition can be non-dimensionalized using the following non-dimensional variables and parameters.

$(X, Y) = \frac{(x,y)}{L}$, $(U, V) = \frac{(u,v)}{\alpha/L}$, $P = \frac{pL^2}{\rho\alpha^2}$, $\theta = \frac{T-T_c}{T_h-T_c}$, $\theta_s = \frac{T_s-T_c}{T_h-T_c}$, $(T_h > T_c)$.

$Ra = \dfrac{g\beta(T_h - T_c)L^3}{\alpha\nu}$ Rayleigh number, $Pr = \dfrac{\nu}{\alpha} = 0.7$ Prandtl number, $K^* = \dfrac{K_s}{K_f}$ solid fluid thermal conductivity ratio, $\Delta T^* = \dfrac{(\frac{\dot{q}W^2}{K_f})}{(T_h - T_c)}$ temperature difference ratio and $A^* = \dfrac{W^2}{L^2}$ aspect ratio.

Once the numerical values of the temperature function may obtained, the rate of heat transfer in terms of the Nusselt number for all walls of the enclosure were calculated using:

$$\overline{Nu}_{x=0} = \overline{Nu}_{x=L} = -\int_0^L \left(\frac{\partial T}{\partial x}\right)_{x=0 \ or L} dy,$$
$$\overline{Nu}_{y=0} = -\int_0^L \left(\frac{\partial T}{\partial y}\right)_{y=0} dx.$$

3. Method of Solution

A finite volume method based on SIMPLE algorithm as proposed by Patankar (1980) is used to solve numerically the non-dimensional conservation equations for mass, momentum and energy with appropriate boundary conditions. Power-law scheme is utilized to approximate the convection and diffusion terms. The algebraic system resulting from numerical discretization was calculated using tridiagonal matrix algorithm (TDMA) applied in a line passing through every control volumes in the computational domain. The solution procedure is iterated until the following convergence criterion is satisfied

$$\left|\frac{\chi_{i,j}^{n+1} - \chi_{i,j}^{n}}{\chi_{i,j}^{n+1}}\right| < 10^{-6}.$$

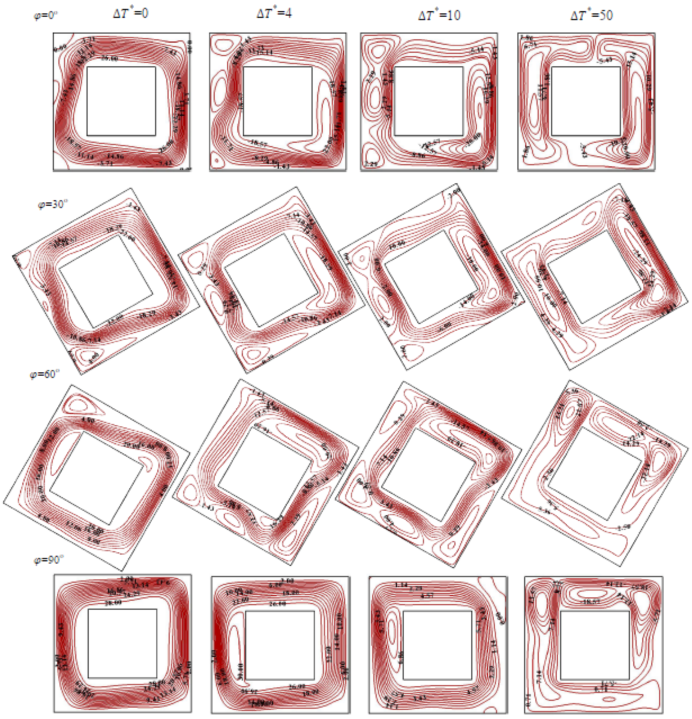

Figure 5. Streamlines for various ΔT^* and ϕ with $A^* = 0.25$ and $K^* = 2$

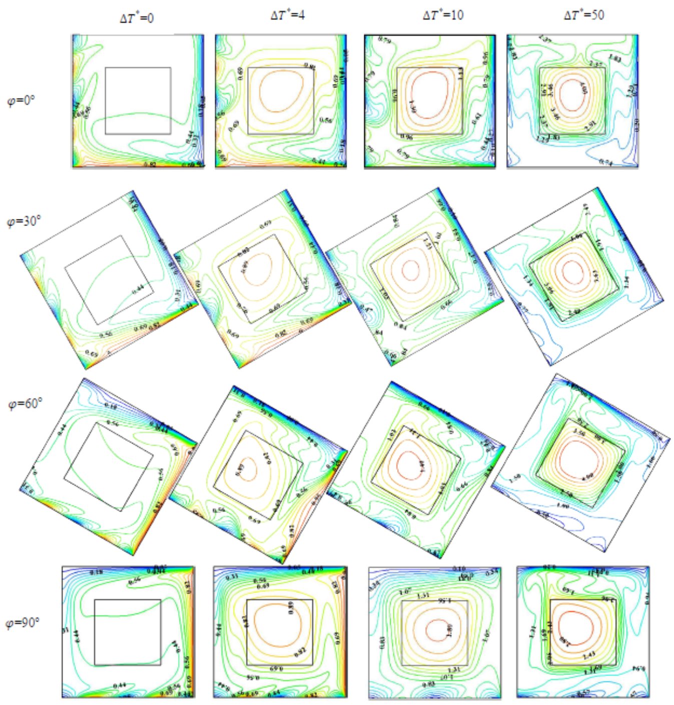

Figure 6. Isotherms for various ΔT^* and ϕ with $A^* = 0.25$ and $K^* = 2$

3.1 *Code Validation*

The developed computational model is validated with previous study and good agreements are achieved proving the ability of this method to simulate fluid flow and heat transfer for simple and complex problems in engineering and applied sciences. Table 1. shows the comparative study of present model with those of

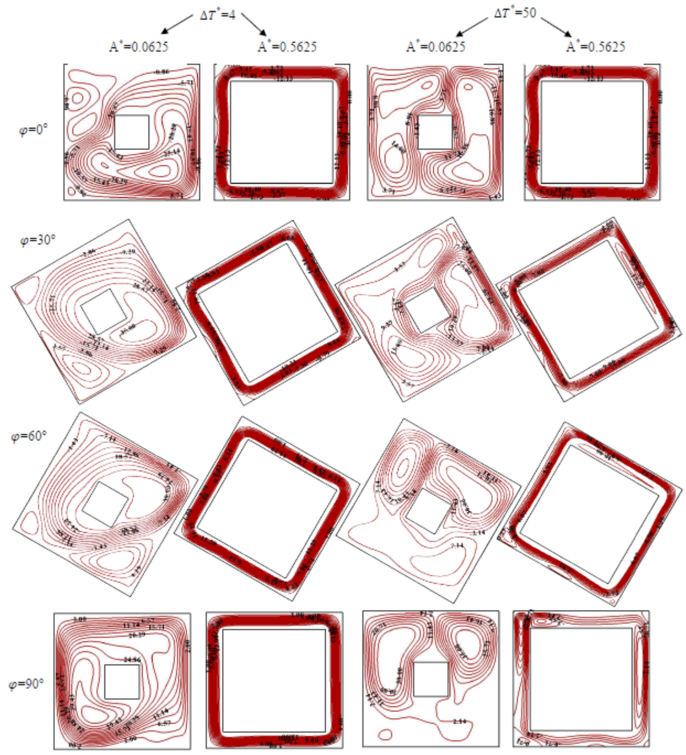

Figure 7. Streamlines for various A^*(0.0625 and 0.5625) and ϕ with$K^* = 2$, ΔT^* (4 and 50)

Davis (1983), House et al. (1990) and Oh et al. (1997).

Table 1.Comparison table for Nusselt number.

Ra	Solid Body	Davis (1983)	House et al. (1990)	Oh et al. (1997)	Pre sent
10^3	*no*	1.118	$--$	1.119	1.117
10^5	*no*	4.519	$--$	4.565	4.550
10^5	$yes(A^* = 0.25, K^* = 0.2)$	$--$	4.624	4.626	4.651
10^5	$yes(A^* = 0.25, K^* = 5.0)$	$--$	4.325	4.327	4.457

3.2 *Grid Independence Test*

A staggered grid system was followed to verify grid independence, numerical procedure was carried out for seven different mesh sizes, namely; 81X81, 91X91, 101X101, 111X111, 121X121, 131X131 and 141X141. Average Nu of the hot wall is obtained for each grid size. As can be observed in Fig. 2, an 131X131 uniform grid size, yielded the required accuracy and was hence applied for all simulation exercises in this work as presented in the following section.

4. Results and Discussion

Figure 3 and 4 shows the streamlines and isotherms for various values of $K^*$and ϕ with fixed $A^* = 0.25$ and $\Delta T^* = 4$. For $\phi = 0^o$ a large anticlockwise rotating circular eddy is found which covers the solid body.

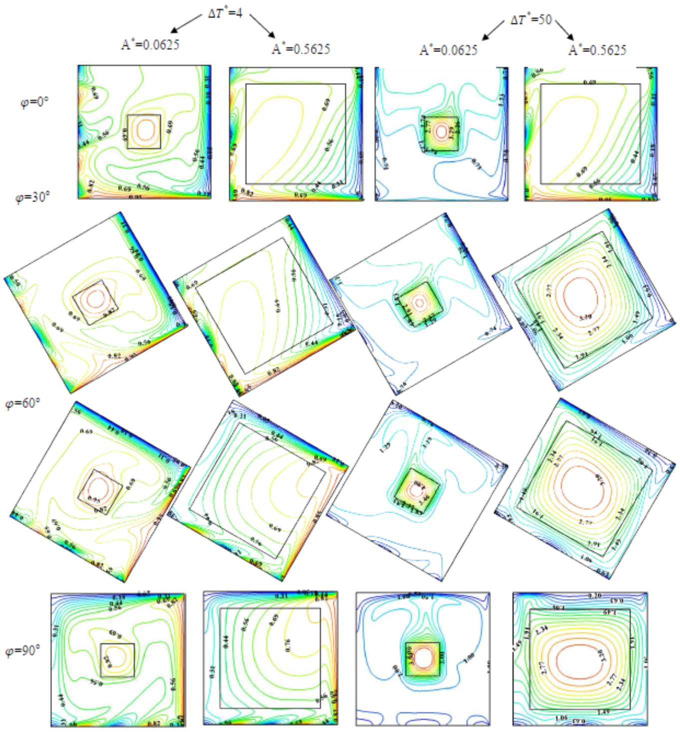

Figure 8. Isotherms for various A^*(0.0625 and 0.5625) and ϕ with$K^* = 2$, ΔT^* (4 and 50)

A secondary eddy near the right bottom corner of the solid body is seen because of the temperature imbalance between the solid body, right cold wall and bottom hot wall. The clockwise rotating secondary eddies also appear near the left wall is due to the thermal non-equilibrium between the non-uniform heating of the left wall and the solid body. When the enclosure is tilted to 30^o, the secondary eddy near the left bottom wall of the enclosure begins to grow, growth of this eddy suppresses the large circular primary eddy and finally at 90^o the primary eddy is merged with all secondary eddies to form a single large clockwise rotating eddy. Only significant changes were noticed on increasing K^*. As the increase in thermal conductivity ratio leads to suppress the rate of heat transfer.

The effect of angle of inclination with respect to various temperature difference ratio (ΔT^*) were illustrated through Fig. 5 and 6. When $\Delta T^* = 0$, the heat transfer takes place only due to convection. The isothermal lines are crowded near the enclosure walls and the core region is almost empty. As ΔT^* was 4 and above, the heat emerged by the solid body slowly begins to disturb the convection current and conduction mode of heat transfer begins to takes place inside the enclosure. On increasing ΔT^*, the heat transmitted by the solid body is greater than the temperature of the fluid inside the enclosure. It is seen that flow pattern is distracted due to this thermal variation. Finally the single circular eddy is subdivided vertically into two eddies which separately covers the right and left zone of the enclosure. As $\Delta T^* = 4$ a week secondary circulating eddy is found near the left wall of the enclosure. After on increasing ΔT^* leads to the increase of circulation strength all over the enclosure. On the other hand, crooked isothermal lines were observed in the fluid region for high value of ΔT^*.

It is also found that isotherms get departs from the bottom wall and inactive top region of the enclosure becomes active at $\Delta T^* = 50$ for all ϕ. Red isothermal lines at the solid region get concentrated on increasing ΔT^* showing the increase in heat generation. The solid body temperature is higher than all the wall temperature of

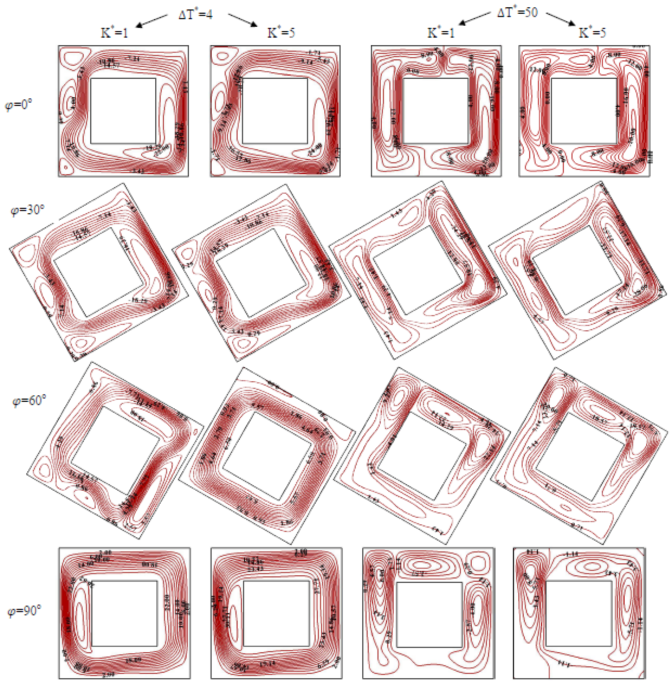

Figure 9. Streamlines for different K^*(1 and 5), ΔT^*(4 and 50) and ϕ with A^*=0.25

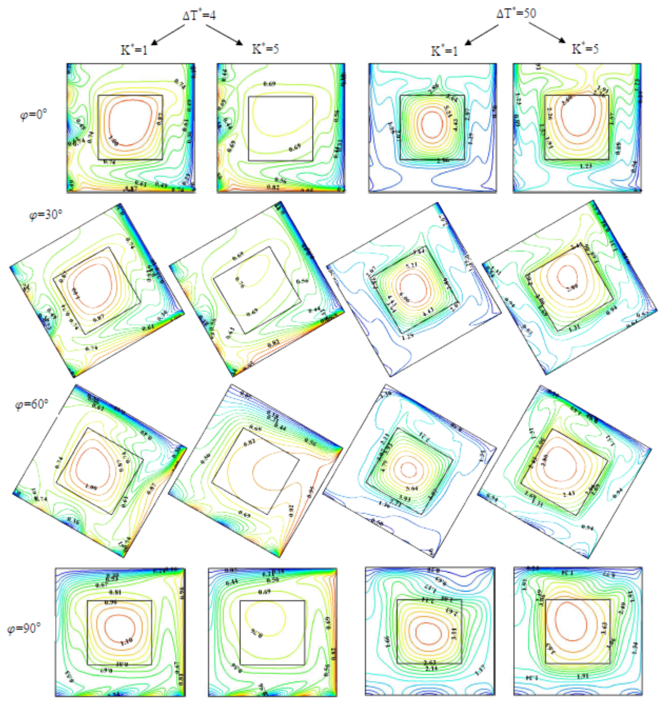

Figure 10. Isotherms for different K^*(1 and 5), ΔT^*(4 and 50) and ϕ with A^*=0.25

the enclosure (left, right and bottom). Therefore all the walls receive heat from the solid body.

Figure 7 and 8 demonstrate the effect of the angle of inclination for $A^* = 0.0625$ and 0.5625 with $\Delta T^* = 4$ and 50 respectively. When $A^* = 0.0625$, fluid get transported all over the enclosure. Because of the heat generating property, the heated solid body surface heats the adjacent fluid layer. This in turn progresses the buoy-

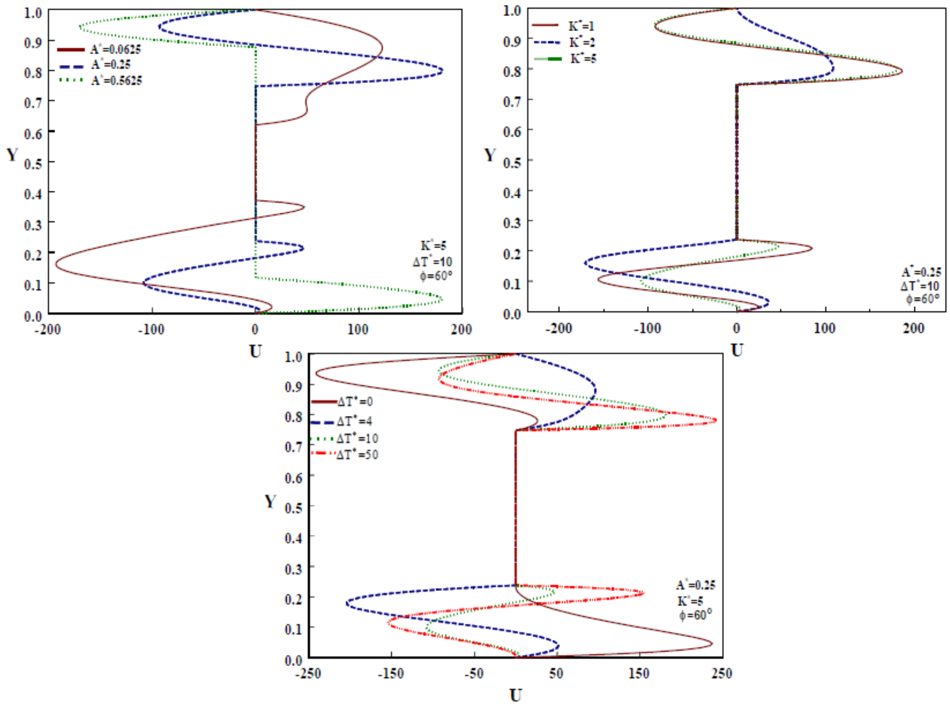

Figure 11. Horizontal mid-height velocity profile for different A^*, K^* and ΔT^*

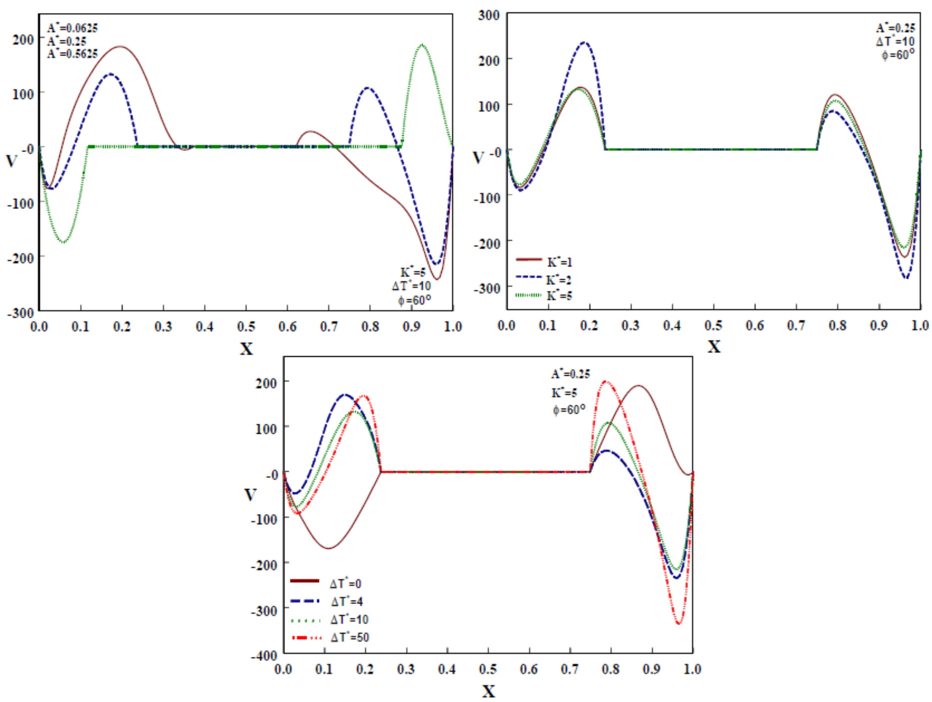

Figure 12. Vertical mid-height velocity profile for different A^*, K^* and ΔT^*

ancy force around the solid body. But a temperature gradient takes place at the lower right corner of the solid body. Thus a short circulating eddy was found at the corner. On increasing the value of ϕ, this corner eddy gets suppressed and was fetched to top left corner. But for $\Delta T^* = 50$, maximum fluid circulation inside the enclosure is found for $\phi = 0^o$ but the gradual increase of ϕ tends to make the

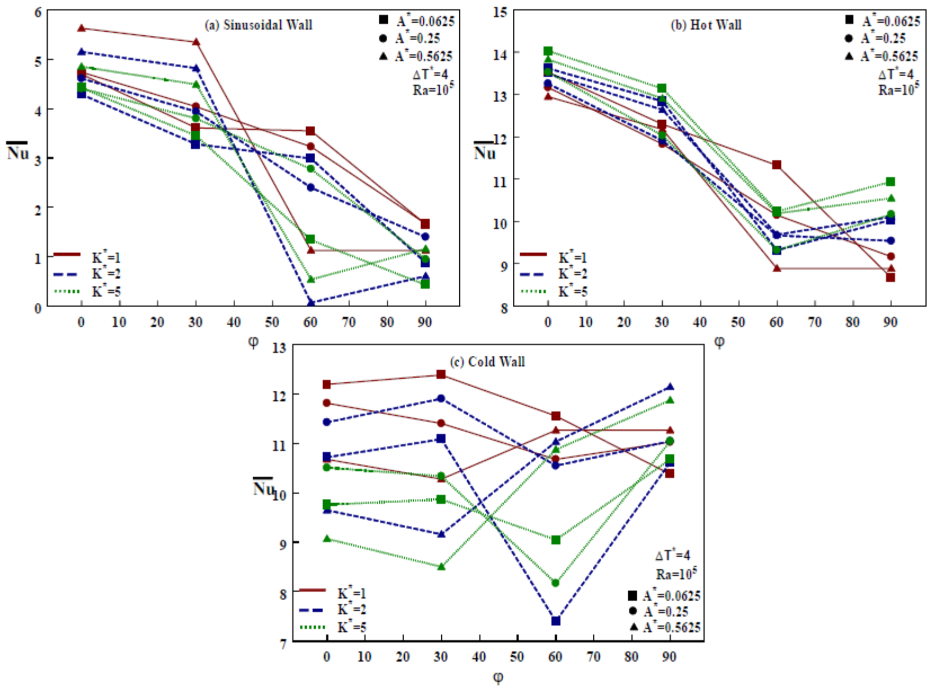

Figure 13. Nusselt number Vs ϕ for different A^* and K^* at various thermal boundaries

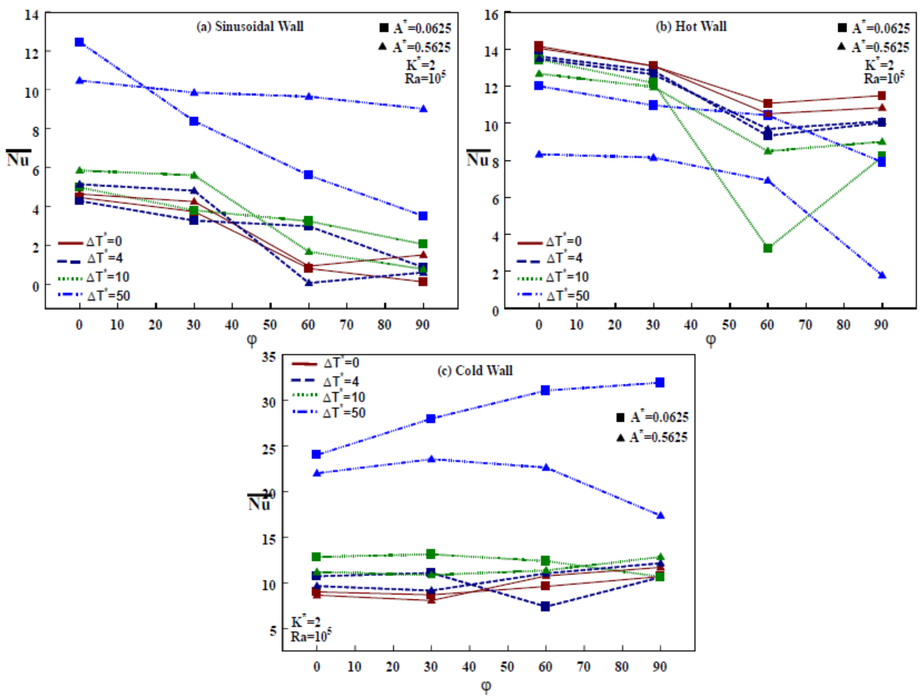

Figure 14. Nusselt number Vs ϕ for different ΔT^* and A^* at various thermal boundaries

left part of the enclosure to be stagnant. Unoccupied isothermal line pattern was found. Moreover, for $A^* = 0.5625$, major part of the enclosure is engaged by the solid body, the unavailability of the free space guide the heat transfer through the conduction. For $\Delta T^* = 4$, convection dominated flow regime is observed, where as conduction dominated heat transfer is found for $\Delta T^* = 50$.

Likewise the effect of angle of inclination with respect to temperature difference ratio and the thermal conductivity ratio was shown in Fig. 9 and 10. Similar effect

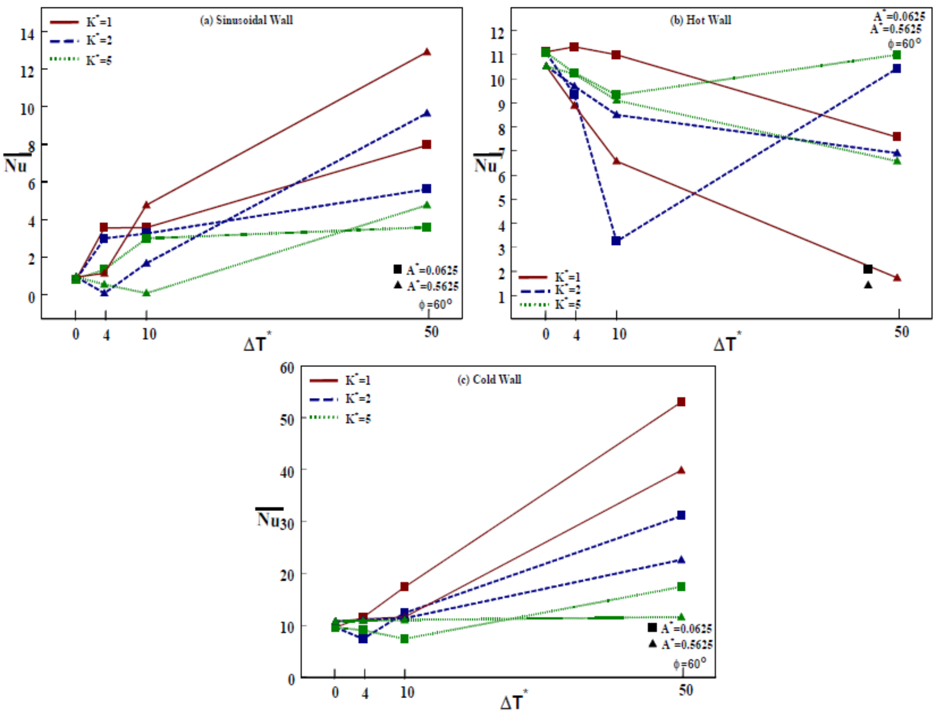

Figure 15. Nusselt number Vs ΔT^* for different A^* and K^* at various thermal boundaries

were observed as in Fig. 7 and 8. Particularly on increasing K^*, the streamlines and isothermal lines get strength and shows the convection mode of heat transfer. For $\Delta T^* = 50$, the enhancement of heat transfer is attained through pure conduction. Figure 11 and 12 shows the horizontal and vertical mid-height velocity profile for varying values of A^*, K^* and ΔT^*. Velocity profile gets more oscillated only for $A^* = 0.0625$ and the oscillation get decreased for increasing A^*. Only significant effect is observed for the increasing value of K^*. As in the variation of ΔT^*, the oscillation is minimum for $\Delta T^* = 0, 4$ and it is increased for $\Delta T^* = 10$. The maximum oscillation is noticed on varying ΔT^* to 50. Figure 13 illustrate the effect of angle of inclination against Nusselt number for different K^* at three different walls of the enclosure. At all the walls, the Nusselt number graph decline for the increasing value of ϕ upto 60^o but after 60^o in most of the cases the Nusselt number get increased. Particularly in sinusoidally heated wall, only for $A^* = 0.5625$ it is noticed. In constantly heated hot wall and cold wall, increasing Nusselt number graph was observed after $\phi = 60^o$. Maximum Nusselt number was attained in the hot wall for $K^* = 5$ and $A^* = 0.0625$, while the minimum is observed at sinusoidal heated wall for $K^* = 2$ and $A^* = 0.05625$.

The same effect was observed in Fig. 14. On increasing ϕ with respect to various values of ΔT^*. In particular the only exceptional graph is for $A^* = 0.0625$ and $\Delta T^* = 50$. An increasing manner of Nusselt number graph is noticed for all values of ϕ and in this particular graph only very maximum Nusselt number is obtained. Figure 15 shows the effect of temperature difference ratio against Nusselt number. In high heat generation condition, a higher temperature of the solid body is found inside the enclosure, this tends to reduce the temperature gradient near the hot wall. This also reduces the rate of heat transfer in the hot wall. But in sinusoidal hot wall and cold wall, the graph get increased for increasing ΔT^*.

5. Conclusion

The natural convection in a tilted square enclosure with a heat conducting and generating solid square body located in the middle was investigated using an accurate and efficient control volume approach. Graphical results in term of streamlines and isotherms are strongly affected by the pertinent parameters aspect ratio, thermal conductivity ratio, temperature difference ratio and the angle of inclination.

- In most of the cases increasing value of the angle of inclination leads to the decreases in rate of heat transfer upto 60^o. After 60^o it get increased. The only exceptional case in which the increasing rate of heat transfer is observed in the case of $A^* = 0.0625$, $\Delta T^* = 50$ and $K^* = 2$.
- Comparing all those investigated cases, the maximum heat transfer was attained in the case of $A^* = 0.0625$, $\Delta T^* = 50$, $\phi = 60^o$ and $K^* = 1$.
- Only significant changes were noticed on increasing K^*.
- For $\Delta T^* = 4, 10$ convection dominated flow regime is observed, the enhancement of heat transfer is attained through pure conduction for $\Delta T^* = 50$. Increasing value of A^* progresses in decreasing fluid circulation.

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