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MIXED CONVECTION WITH ISOFLUX HEATING FROM BOTTOM IN A TILTED LID-DRIVEN TRAPEZOIDAL ENCLOSURE

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ABSTRACT

A numerical study has been performed on laminar fluid flow and heat transfer characteristics in a titled lid-driven trapezoidal enclosure with isoflux heating from below. In this study, the top wall is considered adiabatic that is moving in its own plane at a constant speed, the inclined sidewalls are maintained at a constant cold temperature, and an isoflux heat source is provided at the bottom surface that is also moving in the opposite direction of the moving top wall. The enclosure is assumed to be filled with a Bousinessq fluid with a Prandtl number of 0.71. The influence of the Richardson number, Ri and sidewall inclination angle (Ø) of the trapezoid is emphasized here considering different base angles. The results are shown in terms of streamline plots and isotherm contours within the enclosure for Ri ranging from 0.1 to 10 and sidewall inclination angles ranging from 15° to 45° in which base wall tilt angles to be 0°, 15°, and 30°. Air is considered as a working fluid keeping the Reynolds number fixed at 100. The numerical results indicate that both the temperature and velocity profiles depend strongly on the governing parameters. Heat transfer performance in terms of average Nusselt number is found to be improved with the increase of the Richardson number and sidewall inclination angle. Effect of inclination angle on heat transfer characteristics becomes significant at higher Richardson number.

Keywords: Mixed Convection, Lid-Driven Trapezoid, Finite Element Method, Isoflux.

1. INTRODUCTION

There have been many investigations in the past on mixed convective flow in lid-driven cavities. Aydin and Yang [1] numerically studied mixed convection heat transfer in a two-dimensional square cavity having an aspect ratio of 1. Migeona *et al.* [2] investigate the problem of the time-dependent laminar incompressible flow motion within parallelepipedic cavities in which one wall moves with uniform velocity after an impulsive start using a particle-streak and a dyeemission techniques. Chen and Cheng [3] experimentally and numerically studied mixed convection and flow pattern in a lid-driven arc shape cavity. Later, Chen and Cheng [4] numerically studied the effects of cavity shape on flow and heat transfer characteristics of the lid-driven cavity flows. Sarif [5] studied laminar mixed convection in shallow inclined driven cavities with hot moving lid on top and cooled from bottom. Amiri *et al.* [6] examined the effect of sinusoidal wavy bottom surface on mixed convection heat transfer in a lid-driven cavity. Kumar [7] experimentally investigated natural convective heat transfer in trapezoidal enclosure. Natural convection in tilted parallelepipedic cavities for large Rayleigh numbers was studied numerically and experimentally by Baïri *et al*. [8]. Kuyper and Hoogendoorn [9] investigated laminar natural convection flow in trapezoidal enclosures to study the influence of the inclination angle on the flow and also the dependence of the average Nusselt number on the Rayleigh number. The present study shows the effects of mixed convection in a tilted trapezoidal enclosure with isoflux heat source, in which, almost no previous work has been occurred including isoflux heating from below with top and bottom wall movement.

2. PROBLEM FORMULATION

A trapezoidal enclosure of length L and height H is considered in Fig. 1 having the sidewalls inclined at an angle Ø with the y-axis. The top and bottom wall of the enclosure is allowed to move in its own plane at a constant speed U_0. The base wall is considered to be tilted at ψ = 0°, 15°, and 30° respectively with the horizontal x-axis.

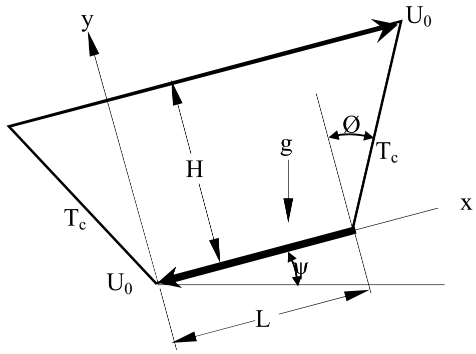


Fig 1. Schematic diagram of the physical system.

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \quad (1)$$

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\frac{1}{Re}\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right)+\left(Ri\sin\psi\right)\theta \quad (2)$$

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\frac{1}{Re}\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)+\left(Ri\cos\psi\right)\theta \quad (3)$$

$$U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=\frac{1}{Re\,Pr}\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right) \quad (4)$$

The Richadson number (Ri), Renolds number (Re), Grashof number (Gr) and Prandtl number (Pr) are given by

$$Ri=\frac{Gr}{Re^2},\ Re=\frac{U_0 L}{\nu},\ Gr=\frac{g\beta\Delta T L^3}{\nu^3},\ Pr=\frac{\nu}{\psi} \quad (5)$$

The dimensionless parameters in the equations above are defined as follow:

$$X=\frac{x}{L},\ Y=\frac{y}{L},\ V=\frac{v}{U_0},\ U=\frac{u}{U_0},\ P=\frac{p}{\rho U_0^2},\ \theta=\frac{T-T_c}{\Delta T},\ \Delta T=\frac{q' L^2}{k} \quad (6)$$

where, ρ, β, ν, ψ, g, and c_p are the fluid density, coefficient of volumetric expansion, kinematic viscosity, thermal diffusivity, gravitational acceleration and specific heat, respectively. The average Nusselt number at the moving wall of the enclosure may be expressed as

$$Nu=\int_0^1 \frac{1}{\theta_S(X)}dX$$

3. RESULTS AND DISCUSSIONS

The numerical procedure used to solve the governing equations for the present work is the combined finite element method. The application of this technique is well documented Zienkiewicz and Taylor [10]. Since the validation against experimental data is not possible in the present case, the computational code is validated with the results obtained for natural convection flows in a trapezoidal enclosure as mentioned by Natarajan *et al.* [11]. It has been observed that the present numerical solution is almost in complete agreement with the aforementioned one in terms of the isotherm plot. Therefore, it can be decided that the current code can be used to predict the flow and thermal field for the present problem accurately. However, almost similar experimental results for square straight enclosure with discrete bottom heating obtained by Corvaro and Paroncini [12]. The agreement is found to be excellent which validates the present computations and lend us confidence for the use of the present code.

3.1 Effect Of Richardson Number On The Flow And Heat Transfer

The Richardson number is defined as the ratio of Gr/Re^2, which provides a measure of the importance of buoyancy-driven natural convection relative to lid-driven trapezoidal enclosure force convection. It reflects a dominant conduction mode as $Ri \geq 1$, while it resembles similar driven enclosure flow behavior for a nonstratified fluid if $Ri \leq 1$ [13]. In the present work, the Richardson number (Ri) which determines the order of influence of both natural convection and forced convection, has been varied within a range of 0.1 to 10 for fixed Re = 100.

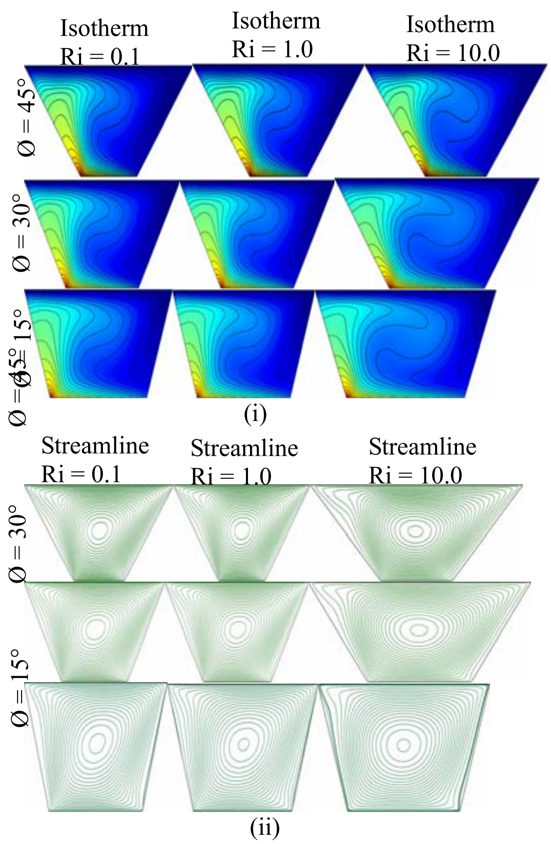


Fig 2. Effect of Richardson number on Isotherms and Streamline plots for different sidewall inclination angles at base wall tilt angle, ψ = 0°.

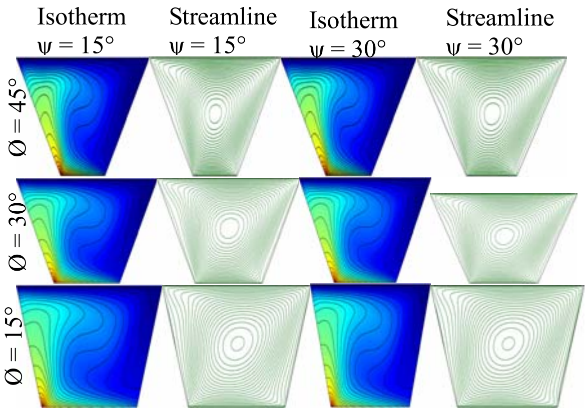
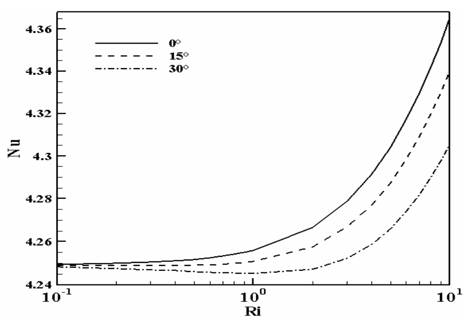
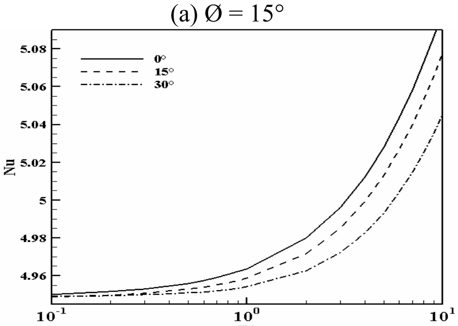


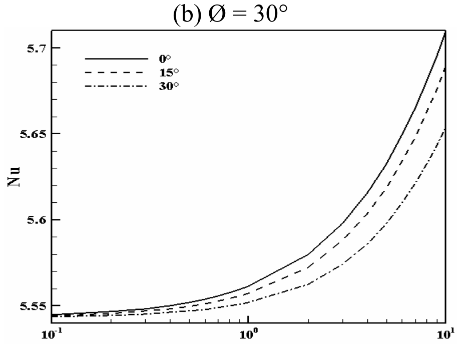
Fig 3. Effect of sidewall inclination angles on Isotherm and streamline patterns for Ri=1 at base wall tilt angle, ψ = 15° and ψ = 30°.



(a) Ø = 15°



(b) Ø = 30°



(c) Ø = 45°

Fig 4. Variation of Nusselt number for different Richardson numbers and base wall tilt angles at sidewall inclination angles, (a) Ø=15°, (b) Ø=30°, (c) Ø=45°.

As shown in Fig 2, the flow and temperature fields change with the variations of *Ri*. The effects of mixed convection have been analyzed through isotherm and streamline patterns in Fig 2 and Fig 3.

Streamlines for Ø=15^0 and ψ =0° shown in Fig 2 indicate that a large vortex is created rotating in the clockwise direction. It shows that heat is carried out along with the motion of the moving both lower and upper walls and the intensity of the flow is increased at the middle region of the enclosure with the help of increased buoyancy forces for higher Richardson number. Flow starts from the bottom moving wall to the left and it goes upward until the upper region is reached. As soon as it reaches the upper region, it starts to move in the right direction and turns to the lower region as it reaches the right corner of the top wall. The flow along the upper and the bottom walls is almost similar for different Richardson number. And for Ri=1 and 10, the streamlines are almost same. The difference is that a series of downward and upward curvature from the top and bottom center decreases with the increase in Richardson number which helps the more heat transfer rapidly. Also it is observed that the center of the main large vortex is shifted towards the center of the enclosure. Streamlines for inclination angle, Ø=30° & 45° and base angle, ψ=0° shown in Fig. 2 indicate that a series of downward and upward curvature from the top and bottom center decreases more with the increase in Richardson number which helps the heat transfer more rapidly than streamlines for Ø=15° and ψ=0° which is explained before. Also it is observed that at Ø=30° & 45° and ψ=0° the center of the main large vortex is shifted towards the center of the enclosure more accurately. Besides, streamlines for Ri=1 at inclination angle, Ø=15°, 30°, and 45° and base angle, ψ =15°, 30° shown in Fig. 3. In Fig. 3, almost same patterns of streamlines are found and a series of downward and upward curvature from the top and bottom center decreases less with the increase in Richardson number which helps the heat transfer less rapidly respectively comparing to the streamlines for Ø=15°, 30°, 45° and ψ=0° in Fig 2.

Some significant changes in the isotherms are also observed for the different values of the Richardson number (Ri) in Fig 2. The isotherms also reveal the similar scenario of the heat transfer characteristic within the enclosure. The isotherms get stronger near the hot wall with the increasing value of the Richardson number (Ri). The isotherms are at first more uniform in nature. With the increase of Richardson number (Ri), the non-linearity of the isotherms increases at a higher value resulting the higher plume formation and a distorted plume is formed indicating better convective heat transfer for Ri=10. When Ri=1, the buoyancy and shear effects are in comparable level and with the further increase of Ri leads to dominant natural convection over mixed convection. However, for lower Ri, where dominant forced convection existed, shows less heat transfer as evident in Fig 4. However, it should be noted that this convective region is decreasing with increasing Ri. In case of Ri≤1, the heat is transfer due to natural convection while if Ri≥1 then the heat transfer mode is force convection. Mixed convection occurs when Ri=1.

Due to inclined thermal line the overall heat transfer is higher.

3.2 Effect Of Base Angle And Inclination Angle On The Flow And Heat Transfer

The streamlines and isotherms are shown for the inclination angle, Ø = 15°, 30°, 45° and Ri=1 in case of base angle, ψ = 0°, 15°, 30° in Fig 2 and Fig 3. From the observation of these figures, it is implied that the patterns of the streamline and isotherm are almost same in all three configurations but resulting heat transfer enhancement with the increase in inclination angle is different as appeared in Fig 4. In addition from Fig 4, with the increase of inclination angle, average Nusselt number (Nu) increases to a great extent. From Fig. 4, it is clearly observed that at Ri=1 (mixed convection condition); the average Nu is maximum for base angle 0° in all three inclination angle conditions like 15°, 30°, and 45°. Besides, it can be said that if the base angle is maximum, then the heat transfer rate becomes minimum. At base angle 0°, it has found the best heat transfer rate comparing to the other base angles (15° and 30°) at each inclination angle condition.

4. CONCLUSION

In this study, the results of a numerical study of shear and buoyancy induced flow and heat transfer in a two-dimensional trapezoidal enclosure with localized heating from below and cooling from side walls are presented. The main parameters of interest are mixed convection parameters, Richardson number (Ri) and the inclination angle (Ø) with the different base angles (ψ). With the increasing value of Richardson number in the mixed convection regime shows that the behavior of progressively dominant natural convection, which has been analyzed in this study. It has also been shown that for a particular inclination angle of the enclosure, the value of average Nusselt number increases with the increase of Richardson number and this will be found with the minimum value of base angle (ψ=0°). Again it is found that the maximum value of the average Nusselt number can be attained at the maximum value of the inclination angle (Ø= 45°) and the maximum value of average Nusselt number will be found at the minimum value of the base angle (ψ =0°). So an analysis of mixed convection is good enough to know the condition of heat transfer in free and force convection.

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