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CFD Analysis of Heat Transfer and Flow Characteristics in A 3D Cubic Enclosure

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Abstract: *Flow arising "naturally" from the effect of density difference, resulting from temperature or concentration difference in a body force field such as gravity, the process is termed as natural convection. There has been growing interest in buoyancy-induced flows and the associated heat and mass transfer over the past three decades, because of the importance of these flows in many different areas such as cooling of electronic equipment, pollution, materials processing, energy systems and safety in thermal processes. Steady state laminar natural convection in a cubic enclosure with a cold vertical wall and two square heaters with constant temperature on the opposite wall is studied numerically. The enclosure is fitted with various liquids. Three-dimensional Navier Stokes equations are solved by employing SIMPLE algorithm. Computations are performed for a range of Rayleigh number from 10^4 to 10^7 while enclosure aspect ratio varies from 0.1 to 1.25. The effects of Rayleigh number, enclosure aspect ratio, and Prandtl number on heat transfer characteristics are studied in detail. The results show that the flow field is very complex and heat transfer from the two heaters is not the same. The effect of Prandtl number is negligible in the range 5 to 100 with other parameters kept constant. This allows the use of liquids such as water for studying other dielectric liquids, provided the flow geometry and other non-dimensional parameters are similar. The overall Nusselt number increased markedly with Rayleigh Number. It is also affected by enclosure aspect ratio.*

Keywords: *Temperature, Aspect ratio, Computation, Three Dimensional, Rayleigh number***.**

I. INTRODUCTION

Steady-state laminar natural convection in a cubic enclosure with a cold vertical wall and two hot squares heaters with constant temperature on opposite wall was studied numerically by Y.L.He, W.W.Yang and W.Q.Tao [1]. The enclosure is filled with liquid various liquids and three dimensional numerical analysis was carried and computations were performed for a range of Rayleigh numbers with various aspect ratios. The effect of Rayleigh number, enclosure aspect ratio and Prandtl number on heat transfer characteristics were studied in detail. The results show that the flow field is very complex and heat transfer from the two heaters is not the same. The effects of Prandtl number are negligible with other parameters kept constant. This allows the use of liquids such as water for studying other dielectric liquids, provided the flow geometry and other non dimensional parameters are similar. The overall Nusselt number increases markedly with Rayleigh number and was also affected by enclosure aspect ratio.

Natural convection in a cubical enclosure with an internal isolated heated vertically plate was investigated both experimentally and numerically by W. Yang and W.Q.Tao [2]. The internal plate was suspended under lower surface of the enclosure top wall and electrically heated. The six enclosure walls were at a lower constant temperature. The plate average Nusselt number and the air temperature in the enclosure symmetry plane were experimentally determined in the range of $8x10^4$ to $5x10^5$. Numerical simulations of laminar natural convection in the same configuration were performed. The agreement between the test data and the numerically predicted values is reasonably good, with a maximum deviation in plate Nusselt number of 12.3% and that in air temperature of 9.4%. Detailed temperature and velocity distribution in four cross sections were presented for the case of $Ra=6.57x10^5$. In the low Rayleigh number region, the plate average Nusselt number is quite close to that of the vertical plate situated in infinite space. The difference between the average Nusselt numbers of a vertical plate in a confined space and infinite space gradually becomes large with increasing Rayleigh number. It is revealed that the three dimensional effect i.e. the flow in the Z direction is quite significant in the region around the end vertical plate, while for the plane normal to the plate and near the symmetry plane, the flow pattern is quite similar to that of the two dimensional results.

II. PROBLEM STATEMENT

The Steady state 3D numerical simulation was carried out inside a cavity by varying the aspect ratio, Rayleigh Number and the Prandtl number. The flow and heat transfer characteristics are functions of geometry and fluid properties. Varying the aspect ratio varied the geometry. The various aspects ratios considered for the analysis are0.1, 0.25, 0.5, 1, and 1.25. In the analysis the fluid properties are varied in such a way the Prandtl number takes values of 5, 50 and 100. The body forces are increased or the viscous forces are reduced to attain a Rayleigh number of 10^4, 10^5, 10^6 and 10^7. The effect of the same on the flow analysis was determined numerically by using a commercial CFD code "FLUENT".

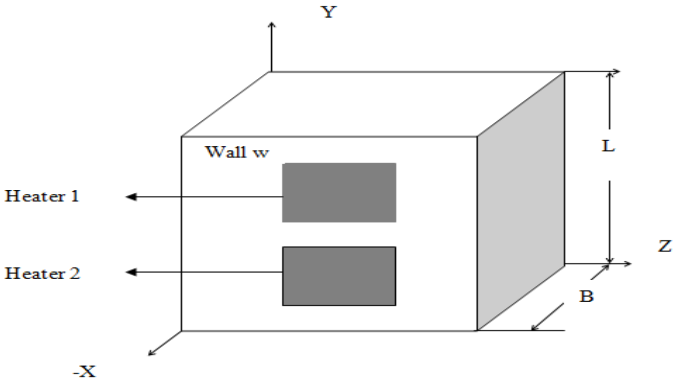


Fig.1: **Boundary Condition for the problem.**

Two heaters are maintained at a higher temperature on one side of the wall of the enclosure. The rest of the wall w on which heaters are installed is completely insulated. The opposite wall to the heaters is maintained at a low temperature i.e. cold wall. The rest of the four walls are completely insulated. No slip boundary condition is considered.

1. At X=0: θ=1, for the isothermal region on the wall containing heaters.
2. At X=0; $\frac{\partial \theta}{\partial X} = 0$ for the rest of the region on the wall containing heaters.
3. At X=1: θ=0, for the wall opposite to the wall containing heaters.
4. At Y=0 and Y=1 $\frac{\partial \theta}{\partial Y} = 0$.
5. At Z=0 and Z=1 $\frac{\partial \theta}{\partial Z} = 0$.

III. GRID GENERATION

Before proceeding further, it is necessary to ascertain the reliability and accuracy of the present numerical model. A grid independence test was carried out, and the results were compared. Three sets of grid, 42×42×42, 50×50×50 and 62×62×62 were employed; the case with 50×50×50 grids was used for taking both the accuracy and convergence rate into account. Also quantitative comparisons were made with the results of various grid setting and it was observed that there were quite negligible variations in the results obtained after increasing the grid size beyond 50×50×50 hence it was treated as an optimum range for obtaining reliable results. Fig.2 shows detailed view of Grid for aspect ratio of 1.25.

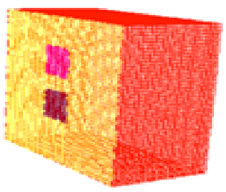
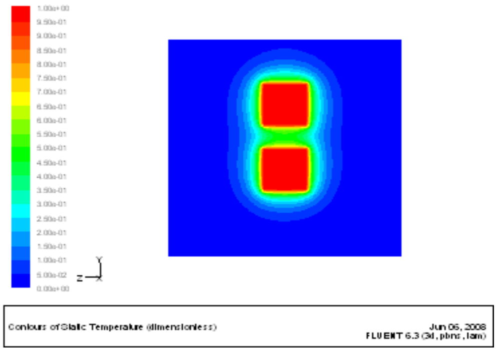


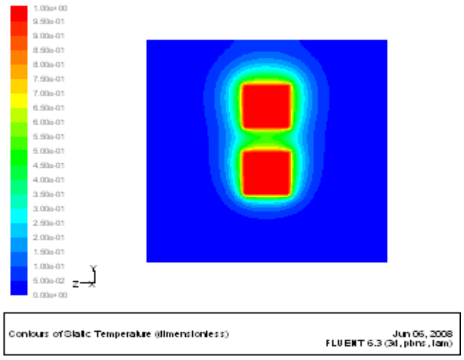
Fig.2

IV. RESULTS & DISCUSSION

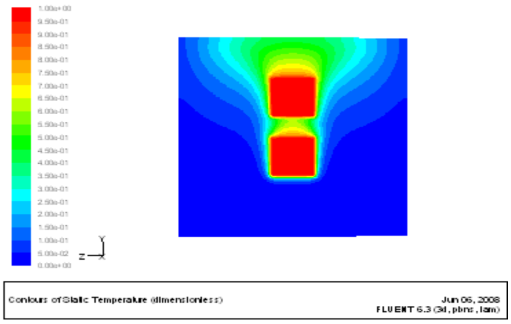
A. RESULTS FOR ASPECT RATIO=0.1 AND PRANDTL NUMBER=5.



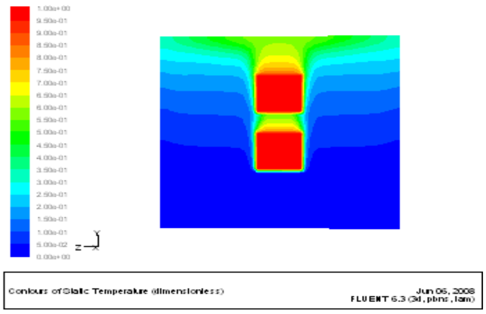
Ra=10^4



Ra=10^5

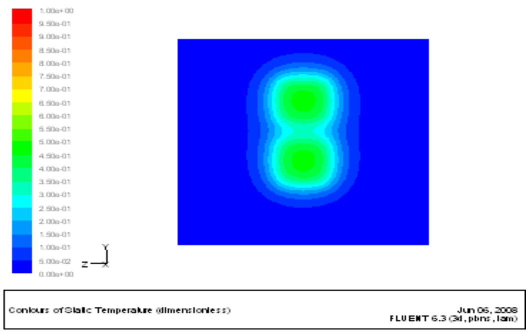


Ra=10^6

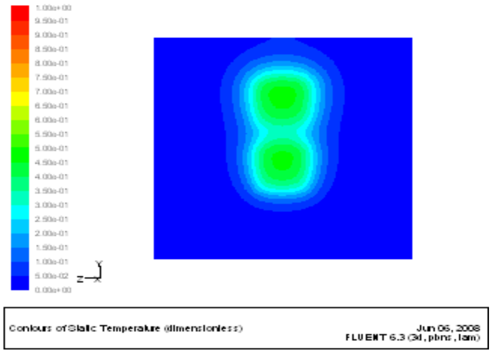


Ra=10^7

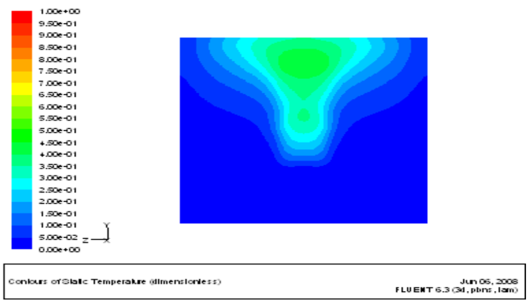
Fig. 3:-Temperature distributions on the plate of the enclosure containing the heaters at various Rayleigh numbers of 10^4, 10^5, 10^6 and 10^7 and **aspect ratio of 0.1**.



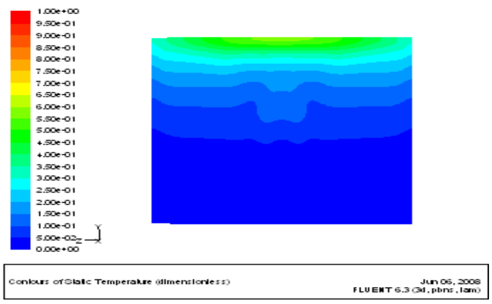
Ra=10^4



Ra=10^5

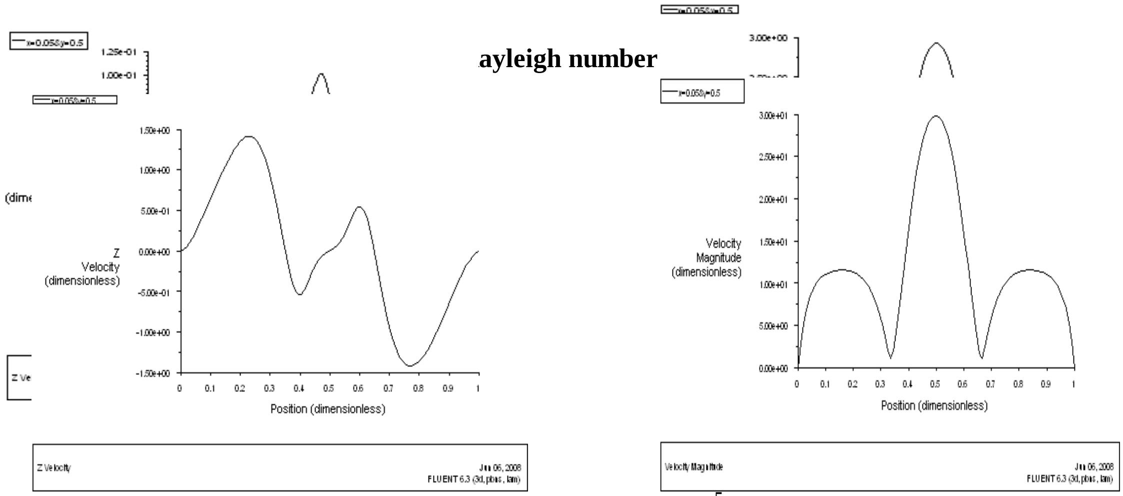


Ra=10^6



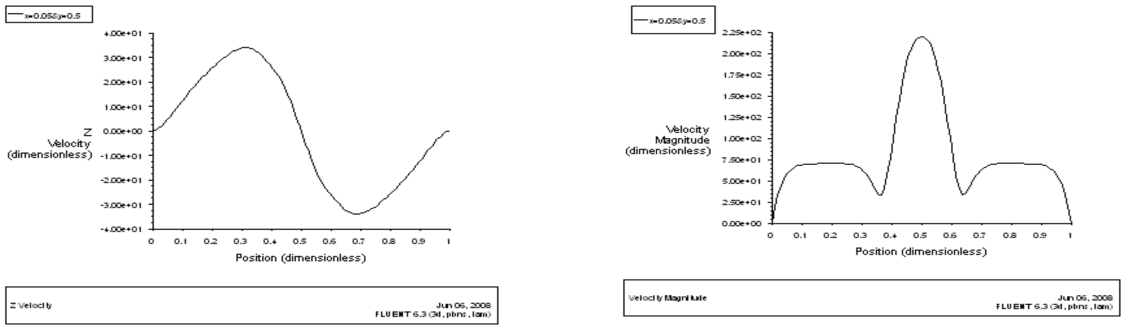
Ra=10^7

Fig. 4:-Temperature distributions at X=0.05 from the surface of enclosure containing the heaters for various Rayleigh numbers of 10^4, 10^5, 10^6 and 10^7 and **aspect ratio of 0.1.**

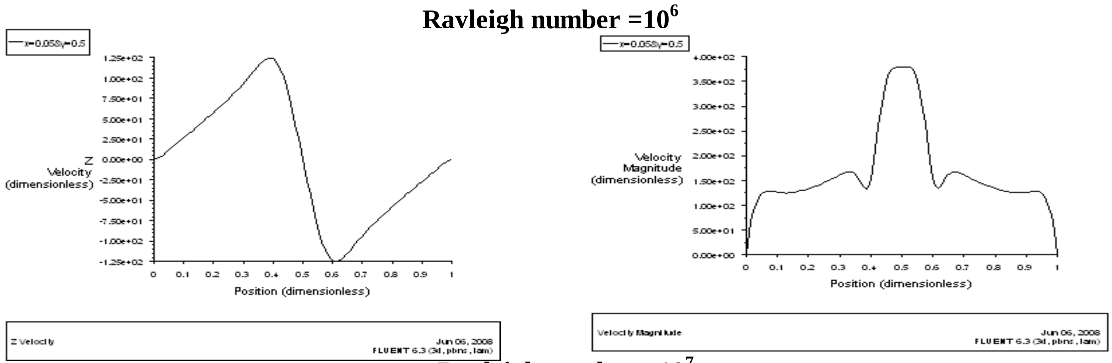


Rayleigh number =10^5

Fig.5:-Velocity distribution at X=0.05 from the surface of enclosure containing the heaters for Ra=10^4 and Ra=10^5 for **aspect ratio of 0.1** and Pr=5.



Ravleigh number =10^6



Rayleigh number =10^7

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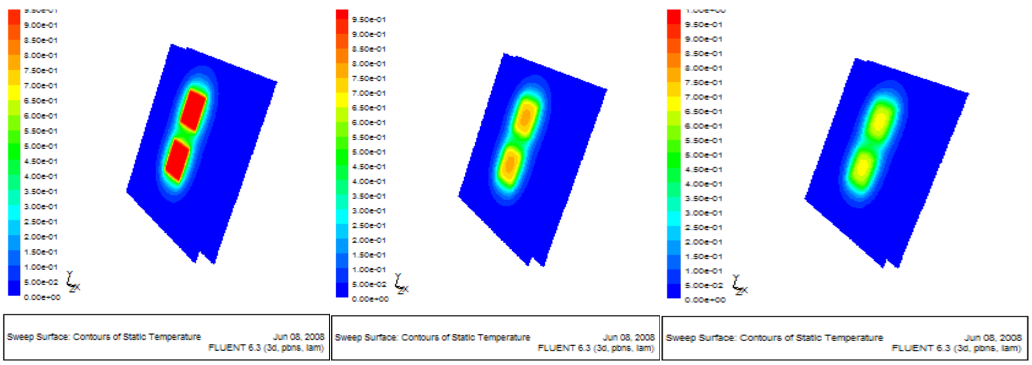
Fig.6: **-** Velocity distribution at X=0.05 from the surface of enclosure containing the heaters at Rayleigh numbers of 10^6 and10^7 for **aspect ratio 0.1** and Pr=5.



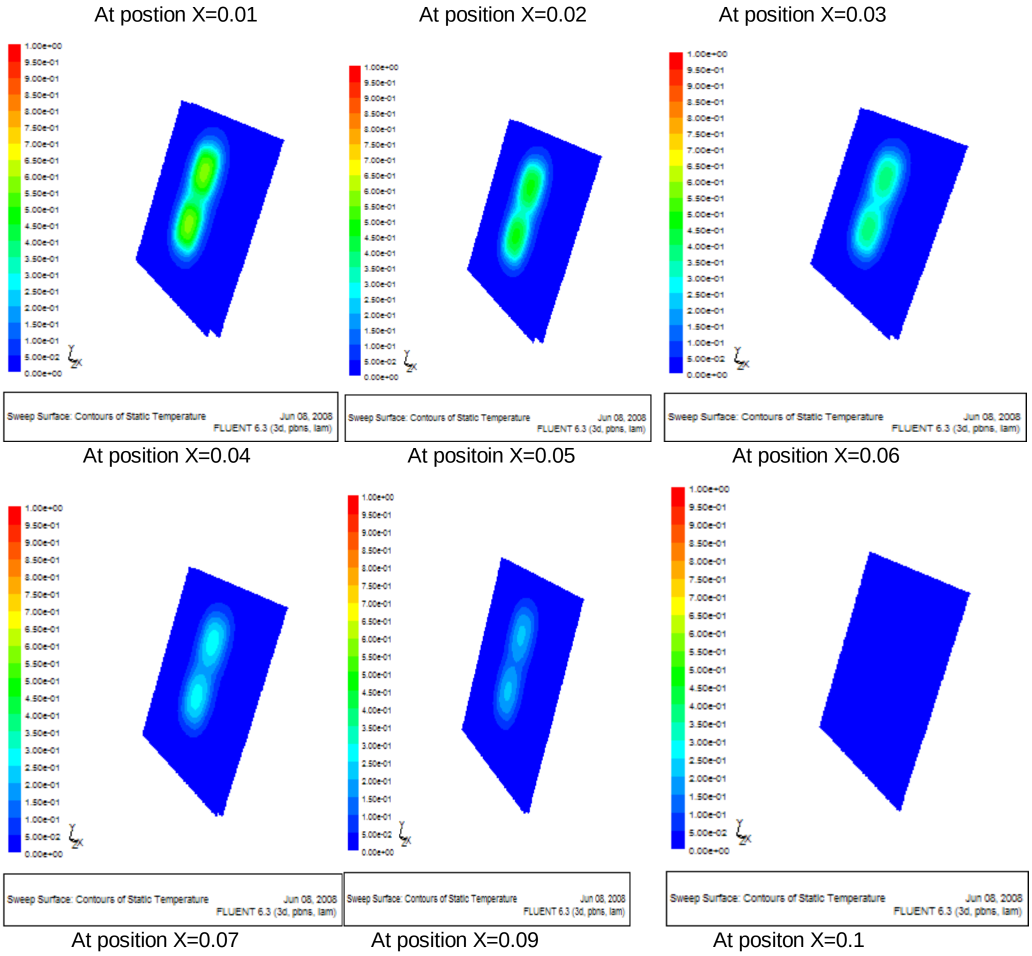


Fig.7:-The above fig shows the animated views of the flow of heat along the x- direction for an enclosure with **aspect ratio of 0.1**.

When the aspect ratio is 0.1, the temperature gradients on the plate containing the heaters are less (Fig. 3). This shows the effect of flow fields on the temperature variations is negligible. When the flow area is restricted as in the case of aspect ratio 0.1, the velocity fields are very small and less convection mode of heat transfer occurs inside the cavity. But as the Rayleigh number is increased the flow picks up and the temperature gradients near the heater picks up. As we move away from the plate containing the heaters, the temperature gradients decreases (Fig. 4 & Fig. 7) and most of the fluid inside the cavity is at rest as there is no density gradient. We can observe slight increase in the temperature gradients with increase in Rayleigh Number. From the temperature variation contour plots it is clear that the temperature gradients are more on heater 2 than Heater 1. Relatively the Nusselt number is greater on Heater 2.

Table 1:-Variation of Nusselt number with Rayleigh number for Aspect Ratio 0.1 for Heater 1 & Heater 2.

Aspect Ratio(0.1)	Nusselt number on Heater 1	Nusselt number on Heater 2
Ra=10^4	25.12	25.31
Ra=10^5	24.7	26.75
Ra=10^6	30.08	42.88
Ra=10^7	54.94	76.84

The Z component of velocities (Fig. 5) is very small as the temperature gradients are very small. This is the direction along which we have the density gradients. The magnitude of the velocities are also very small, viewed at X=0.05 and Y=0.5 line. The maximum velocity attained is 4 dimensionless units for a Rayleigh Number of 10^6(Fig. 6). 1 units of dimensionless velocity corresponds to approximately 10^{-6}m/sec, which is very small. Similarly, For aspect ratio of 0.25, the variation in the temperature distribution is more or less the same.

For low Rayleigh numbers the gradients are very small across the heaters and increases as the Rayleigh number is increased. Temperature variations at X=0.05 shows that the gradients are more confined to the upper part of the cavity and the lower part of the cavity have zero temperature gradients. This means the velocities are very low below the cavity. The variation of Nusselt number for different Rayleigh number is given below.

TABLE 2:-Variation of Nusselt number with Rayleigh number for Aspect Ratio 0.25 for Heater 1 & Heater 2.

Aspect Ratio(0.25)	Nusselt Number on Heater 1	Nusselt Number on Heater 2
$Ra=10^4$	3.91	4.42
$Ra=10^5$	4.55	6.32
$Ra=10^6$	7.89	10.98
$Ra=10^7$	14.14	19.78

Though the Nusselt number is increasing as Rayleigh number is increased, when compared to the aspect ratio of 0.1, the heat transfer rates are very small.

Also, for aspect ratio of 0.5, except for Rayleigh number of 10^7, the temperature gradients are less around the heater, so we cannot find much variation in the Nusselt number as compared with aspect ratio of 0.25. There is a slight decrease in the Nusselt number when compared with aspect ratio of 0.25.

TABLE 3:-Variation of Nusselt number with Rayleigh number for Aspect Ratio of 0.5 for Heater 1 & Heater 2.

Aspect Ratio(0.5)	Nusselt number on Heater 1	Nusselt number on Heater 2
$Ra=10^4$	3.03	4.08
$Ra=10^5$	4.57	6.39
$Ra=10^6$	7.80	10.82
$Ra=10^7$	14.01	19.5

Further, for aspect ratio of 1.0, drops in the heat transfer rates were observed. The temperature variations are more or less similar with the gradients getting increased with increase in Rayleigh number.

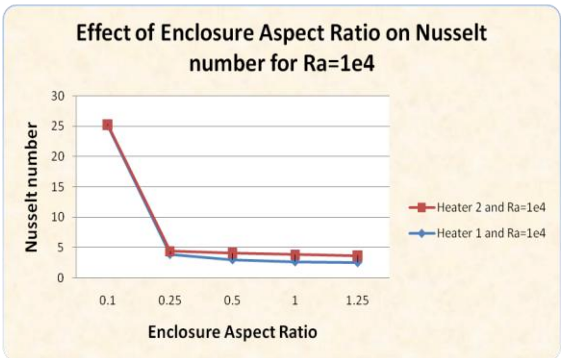
TABLE 4:-Variation of Nusselt number with Rayleigh number for Aspect Ratio 1 for Heater 1 & Heater 2.

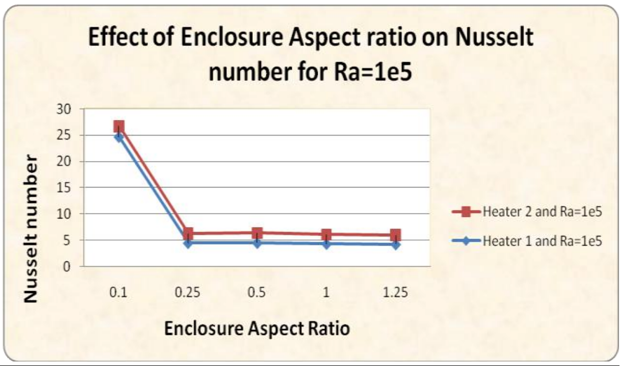
Aspect Ratio(1.0)	Nusselt number on Heater 1	Nusselt number on Heater 2
$Ra=10^4$	2.69	3.83
$Ra=10^5$	4.38	6.11
$Ra=10^6$	7.62	10.57
$Ra=10^7$	13.79	19.25

Nusselt Number variation for aspect ratio of 1.25.

TABLE 5:-Variation of Nusselt number with Rayleigh number for Aspect Ratio 1.25 for Heater 1 & Heater 2.

Aspect Ratio(1.25)	Nusselt number on Heater 1	Nusselt number on Heater 2
$Ra=10^4$	2.56	3.65
$Ra=10^5$	4.30	6.00
$Ra=10^6$	7.54	10.45
$Ra=10^7$	13.45	18.77





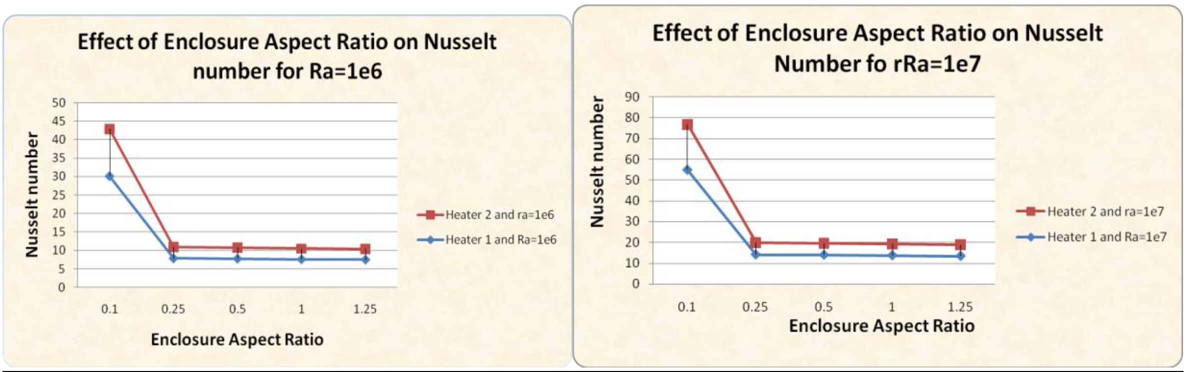


Fig 8:-The effect of enclosure aspect ratio on Nusselt number for heater 1 and heater 2 at Ra=10^4, 10^5, 10^6 and 10^7.

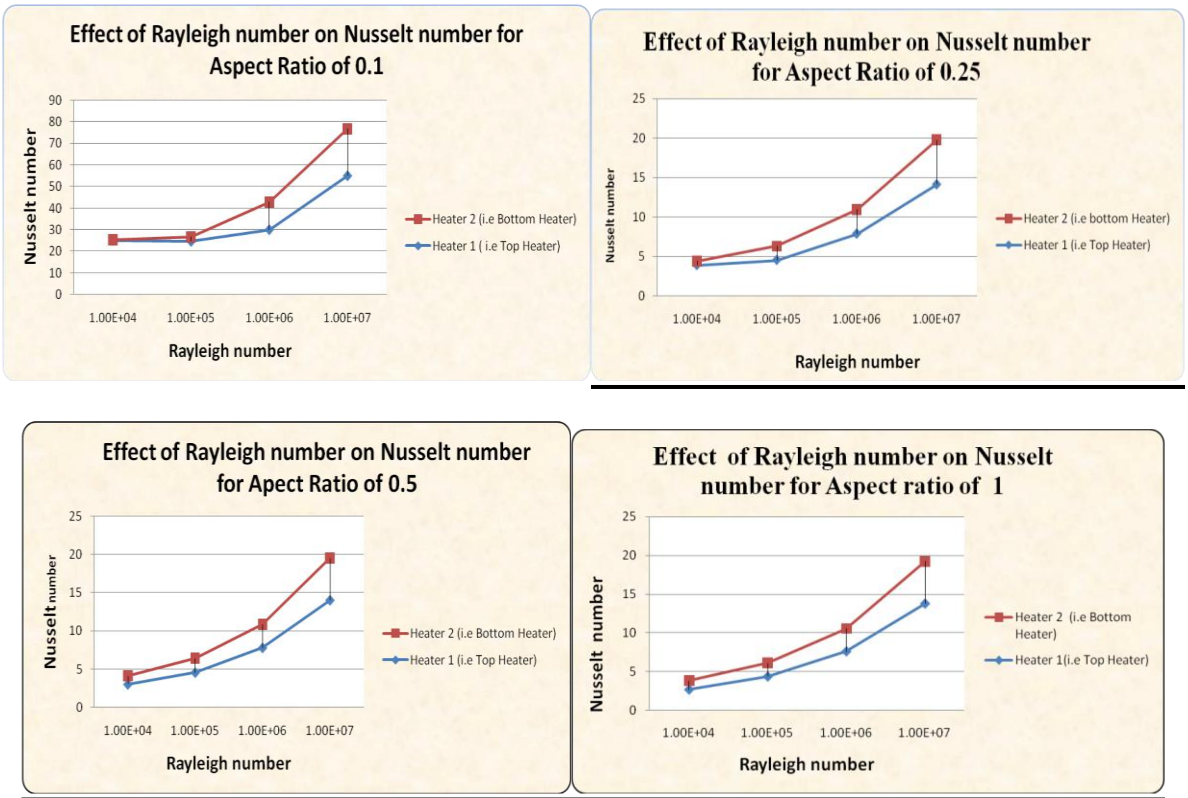
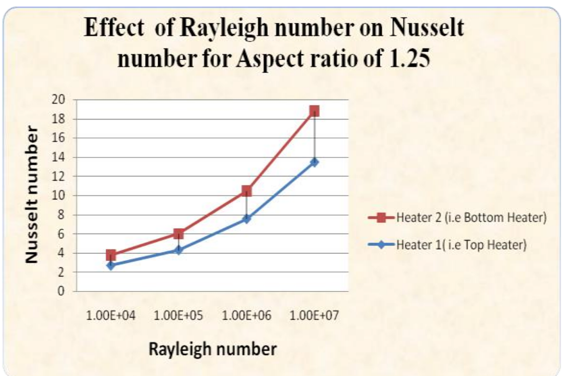


Fig 9:-Effect of Rayleigh number on Nusselt number for Heater1 and Heater 2 for Aspect Ratio of 0.25,0.5, 1 and 1.25.



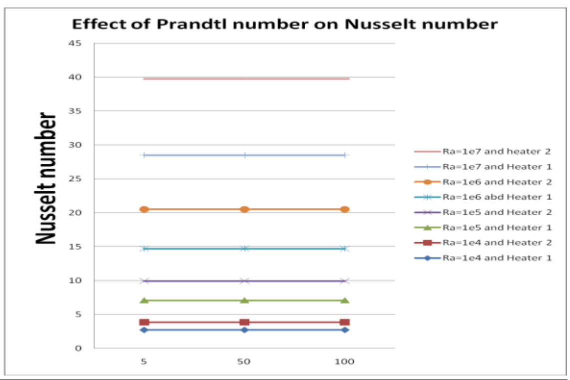


Fig 10:-Effect of Prandtl number on Nusselt number for Heater1&Heater 2.

V. CONCLUSION

The following are the conclusions derived from the numerical investigation carried out inside the cubical enclosure using FLUENT solver.

1. For all the aspect ratios considered for analysis, there was a gradual increase in the heat transfer rates, i.e. Nusselt number, with increase in Rayleigh number (Fig 9).
2. With increase in the aspect ratio from 0.25 to 1.25 there is a gradual reduction of heat transfer rates. Heat transfer rates reduced by 8-22% with increase in the aspect ratio. But there is a drastic reduction in heat transfer rates when the aspect ratio was varied from 0.1 to 0.25. Heat transfer rates reduced by 84%. (Fig 8).
3. From the velocity counters it is very clear that the location of the maximum z component of velocity varied unevenly which shows complex flow phenomena inside the enclosure.
4. With variation of Prandtl number from 1 to 100, the effect on flow and heat transfer characteristics is negligible (Fig. 8).
5. Nusselt number on heater 1 is less than that of heater 2 irrespective of aspect ratio, Prandtl and Rayleigh number (Fig 8) & (Fig 9).

Future Work

This analysis can be carried by including fins inside the cavity, which control the flow phenomena thereby effects the heat transfer rates. Aspect ratio of the heaters can be varied and analysis can be carried to find the optimum location at which the maximum heat transfer rates exists.

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