ANALYSIS OF NATURAL CONVECTION HEAT TRANSFER BY LAMINAR FLOW IN A SQUARE ENCLOSURE

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ABSTRACT: The study of a laminar natural convection in a square enclosure with conjugate boundary condition is done numerically. The top wall and right side wall of the enclosure are considered to have some finite thickness. The problem is solved using finite volume method and the multi-block method is used for meshing the domain. Various cases are considered by varying the parameters like characteristics Reynolds number, conductivity ratio and the wall thickness to analyses their effect on the heat transfer and flow characteristics. Reynolds number is taken equal to 103, 5×103and 104 to constrain the flow as laminar in enclosure. The result shows some significant dependence on Reynolds number in the flow and temperature field inside the domain. An effect of insulation near the top wall is also studied and compared with the case of without insulation by varying the same parameters. It is to be noted that the average temperature inside the enclosure is reduced considerably due to the insulation.

Keywords: conjugate heat transfer, heat transfer in enclosure, laminar, natural convection, multi-block method

I. INTRODUCTION

The study of the heat transfer in the enclosure has been prime topic for the researchers because of its wide range of application in the engineering and practical life. It finds its application from the cooling of nuclear reactor to the heat removal of the micro- electronic components and also for designing of the thermal conditioning of the room, thermal designing of the commercial buildings, cryogenic storage, furnace and many other. Besides its wide range of application, the simultaneous consideration of the con- duction in wall and convection in fluid flow, remains an interesting field of investigation over the past decades.

Numerous articles could be found in the literature for studying the natural convection in the enclosure [1–16]. There are many boundary condition that could be considered but more complex and practically applicable domain are found to be less studied in literature. Most of the already studied domain are either bottom heating or side heating. But in this study we have considered the practically applicable boundary condition along with conjugate heat transfer.

Buildings and Energy

Energy has always been an important topic of discussion amongst the researchers. During any development program of any technology energy consumption has always been given significant importance. As accounted by Cole and Kernan [17] and Ramesh et al

[18] a major part of the total energy consumed in a life time of any building is operating energy. This involves the maintenance and also most importantly the energy involved in maintaining the building in comforting condition both thermal and visual. The build- ing energy analysis tool used prior to start of any project aims to reduce this operating energy.

It is important to understand that this reduction of energy consumption should be achieved while not at the cost of the reduction in performance. So the building must provide comfortable environment as compared to its outside environment. Earlier stud- ies have proved the fact that the occupant quickly responds to any discomfort to regain their comfort, however this may adversely affect the energy consumption. Therefore precise prediction of thermal comfort is very important while designing the building to maintain lower energy consumption as stated by Nicol[19]

II. REVIEW OF LITERATURE

The chronicle of analytical consideration and the recognition of the importance of study- ing the convection heat transfer in the enclosure goes back to the 1954 [3]. Batchelor had foreseen its application in building thermally efficient rooms before studying this problem. But the recent development of nuclear reactor, electronic component and many such equipment have increase the need for their in depth study. After Batchelor's work many researchers have explored this field. The work of Davis [6, 20] has been fol- lowed for many numerical investigations. Many of this simple problem is further stud- ied for observing the effects of the various factors such as the inclined domain as done by Kuyper et al.[11] and Aydin et al [2], the effect of heat source inside the domain was studied by Frederick and Berbakow, Kuznetsov and Sheremet[8, 12], whilst the investi- gation on turbulent natural convection was done by Kuznetsov and Sheremet [14] and experiment was done by Betts and Bokhari [21] and the effect of radiation along with other modes of heat transfer was studied by Balaji and Venkateshan[4], Akiyama and Chong[1] and Ayachi et al. [22].

For in-depth understanding of this problem the comprehensive experiment was car- ried out and studied by Yin et al. [23] and Kim and Viskanta [24] with conducting wall. Kim and Viskanta [24] considered the square domain whilst Yin et al. [23] considered the tall rectangular cavity to study the heat transfer phenomenon. The latter experiment was designed for studying the effect of large aspect ratio. The low Reynolds number tur- bulence study of this natural convection in square cavity was studied by Henkes et al. [10].

Several in depth, up to-date discussion of heat transfer analysis inside the enclosure could be found in the literature related to convective heat transfer [9] and [25].

Du and Bilgen [7] studied the effect of coupling of the conduction in solid wall and convection in fluid flow for the various parameters such as the conductivity ratio, as- pect ratio, Rayleigh number and solid wall thickness. They considered the simple two dimensional domain with the constant heat flux applied to the solid vertical wall and the opposite wall is assumed to be insulated. The remaining two horizontal wall were considered to be at lower temperature. Their results stated that for the thinner wall the input parameters of low Rayleigh number, high aspect ratio and high conductivity ratio causes the heat transfer by the conduction to be more dominant.
This work of the natural convection for a cavity could be studied and applied for the specific practical problem such as room cooling as did by Horikiri et al. [28, 29]. They have studied the natural and forced convection heat transfer for a room with a heat source and wall with some finite thickness for ventilated 2D domain and then extended their study for the ventilated 3D domain with heat source. They have provided interre- lation between the heat source arrangement, the effect of wall thickness, the evaluation of the thermal comfort level and also the energy consumption.

Deftnition of the problem
When we talk about the enclosure here it covers a very broad area from a warehouse to small cold storage room where a heat is to be preserved. The current study was moti- vated from the fact that although most of the researchers are working to improve the efficiency of the cooling devices, very few have taken interest to develop thermally op- timised enclosure or environment numerically which can reduce the load of such de- vices and thus reducing the energy consumption. The non-dimensionalisation allows us to apply the obtained result to any scale and depends only on the non dimensional number. Also the consideration of conjugate heat transfer allows us to understand its effect and gives us physical resemblance of the actual system. The effect of the differ- ent parameters, with and without insulation, are studied in the present work. There comparison is undertaken in present study and result gives us the insight the clear idea about the parametric range to be considered while designing any small cold storage or a commercial building.

Objectives and methodology
The main objective of this work was to study the effect of the various parameters on the heat transfer and flow characteristics of the air trapped inside of the enclosure. The CFD model is developed for this problem and a nondimensionalised study was carried out so that the model could be applied for various scale. The finding of the current work will also present a reference for further developing of CFD model for enclosure with conjugate heat transfer.

III. METHODOLOGY

Initially a simple square enclosure is considered filled with air with given boundary conditions. Then an insulation is provided just below the top wall to study its effect. All the four boundaries are having different and practical boundary conditions as shown in figures 2.1 . The two of the walls, top wall and the right wall, are assumed to have some finite thickness to impose the conjugate boundary conditions. The bottom wall is assumed to be thermally insulated whilst the top wall is assumed to have constant hot temperature. The left wall is maintained isothermally at lower temperature and the right wall is having convective interaction with air at environmental conditions. Three different cases were considered by varying the thickness of both of the boundary walls simultaneously and for each case the Reynolds number and the conductivity of the solid wall are varied to study the effect of these two parameters on the flow and temperature field of fluid in enclosure. The insulation is provided at the height of 0.9 of the length of the side and the thickness of the insulation is as small as 0.01.

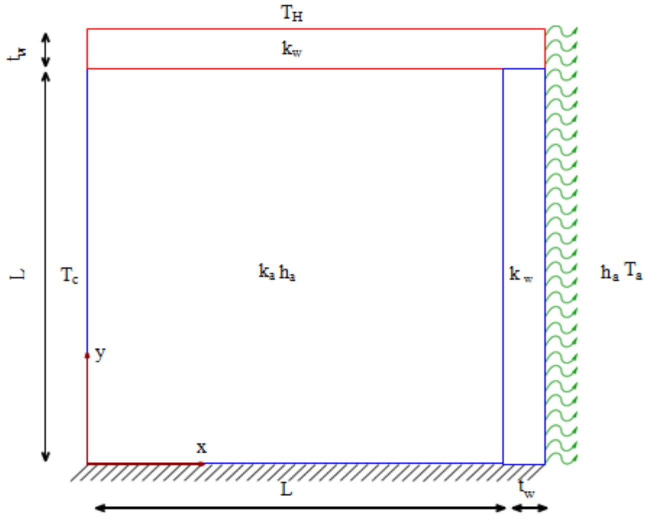


Figure 2.1: Boundary conditions of domain

Governing differentialequation
The governing equations for two dimensional, steady state flow are presented as continuity equation,

$$\frac{\partial}{\partial x}(\rho u)+\frac{\partial}{\partial y}(\rho v)=0$$

u-momentum equation,

$$\frac{\partial(\rho u u)}{\partial x}+\frac{\partial(\rho v u)}{\partial y}=\frac{\partial}{\partial x}\left(\mu\frac{\partial u}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu\frac{\partial u}{\partial y}\right)-\frac{\partial p}{\partial x}$$

v-momentum, equation,

$$\frac{\partial(\rho v u)}{\partial x}+\frac{\partial(\rho v v)}{\partial y}=\frac{\partial}{\partial x}\left(\mu\frac{\partial v}{\partial x}\right)+\frac{\partial}{\partial y}\left(\mu\frac{\partial v}{\partial y}\right)-\frac{\partial p}{\partial y}+\rho g\beta(T-T_C)$$

energy equation,

$$\rho c_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)=\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)$$

Assumptions
The domain is assumed to be two dimensional with laminar natural convection mode of heat transfer. The velocity components along the x and y

directions are u and v respectively. All the properties of the fluid in the enclosure are assumed to be constant. And, the density is taken as a constant except the buoyancy term which is governed by the Boussinesq approximation. Also, heat transfer by radiation is assumed to be negligible as compared to the other modes of heat transfer and so radiation terms are neglected. The conductivity of the insulation material is taken same as that of the air.

The above equations are non diamensionalised by following dimensionless parameter,

$$U=\frac{u}{U_0},\quad V=\frac{v}{U_0},\quad X=\frac{x}{L},\quad Y=\frac{y}{L}$$

$$K_r=\frac{k_s}{k_f},\quad P=\frac{p}{\rho U_0^2},\quad \theta=\frac{T-T_C}{T_H-T_C},\quad \Pr=\frac{v}{\alpha}$$

$$K^*=\frac{k}{k_f},\quad \rho^*=\frac{\rho}{\rho_f},\quad c_p^*=\frac{c_p}{c_{pf}}$$

The x and y are normalized with respect to the maximum length of the domain L and the velocity is scaled by U_0, obtained by setting the Richardson number as unity, where U

$$U_0=\sqrt{g\beta l(T_H-T_C)}.$$

The non-dimensionalised governing equations can be given as :

Continuity Equation:

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0$$

X-momentum equation:

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

Y-momentum equation:

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

Energy equation

$$\rho^* c_p^*\left(U\frac{\partial\theta}{\partial X}+V\frac{\partial\theta}{\partial Y}\right)=\left(\frac{K^*}{Re.Pr}\right)\left(\frac{\partial^2\theta}{\partial X^2}+\frac{\partial^2\theta}{\partial Y^2}\right)$$

It should be noted here that the momentum equations are solved only for the fluid region.

Boundary conditions

All the four boundaries of the square domain are maintained at the different boundary conditions. The left boundary is maintained at the lower temperature, i.e. cold wall. The top wall is considered to be maintained at the high temperature in the whole domain. A convective heat loss is considered through the right wall which is also considered having some finite thickness. The bottom of the enclosure is assumed to be perfectly insulated. The figures show the boundary conditions considered in the case of with insulation and without insulation respectively. The u and v are assumed as zero along the boundary wall due to the no slip condition.

Numerical Analysis

The method which is most commonly followed in the past is based on the Patankar's [27] method. This method is based on the assumption of the high viscosity value for the solid region when conjugate heat transfer is to be considered. So, the whole domain can be considered as one and with such assumption the velocity term can be neglected in the solid zone and simultaneous solution of the equation is possible. Thus, the energy equation is solved for both the zones but the momentum equation was solved for fluid region. And few other methods developed in the recent past involves the immersed boundary condition and ghost node method.

Grid In-dependency Test

The non-uniform grid is used for discrediting the computational domain. The domain is divided into the blocks and gridding of each individual block is carried out. There are three blocks in total where block 1 is fluid zone and remaining two blocks are solid zones. The grids are kept finer near the wall, where more disturbances are expected to happen and coarser at the center for all of the blocks. Figure shows the grid used for carrying out the computations. A multi block grid system having three blocks of 200×200, 40×200, and 240×40 was found to be sufficient to resolve the details of flow and temperature fields inside the domain.

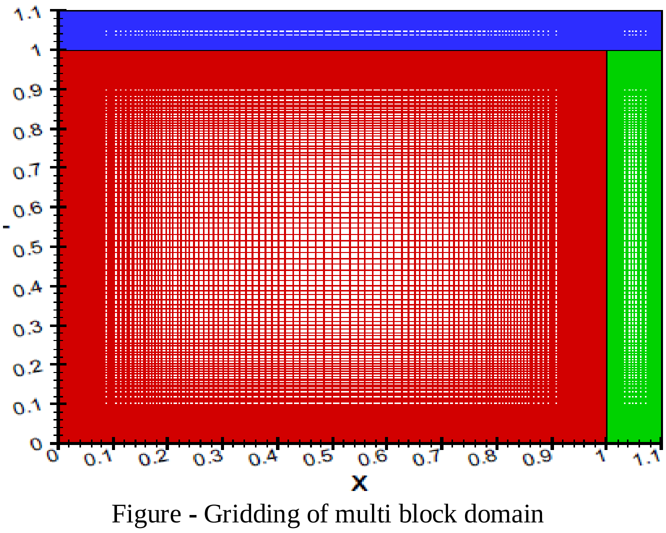


Figure - Gridding of multi block domain

IV. RESULT & DISCUSSION

The results show some interesting facts and phenomenon which were unexpected at the early stage of our investigation. Simulations for various parameters were carried with the same computer program and effects of various parameters on the result are presented. They are shown in tabular form in the following part and an effort to correlate them are made. Reynolds number is considered so as to have the laminar flow inside the enclosure. Another parameter which could not be neglected for the problem speci- fied here is conductivity of the solid wall. Thus, the effect of the solid wall conductivity is also studied and it is varied from 0.25 to 2 W/m-K. Also the effect of the insulation pro- vided at the top is compared with the results of the simulation of the enclosure without insulation.

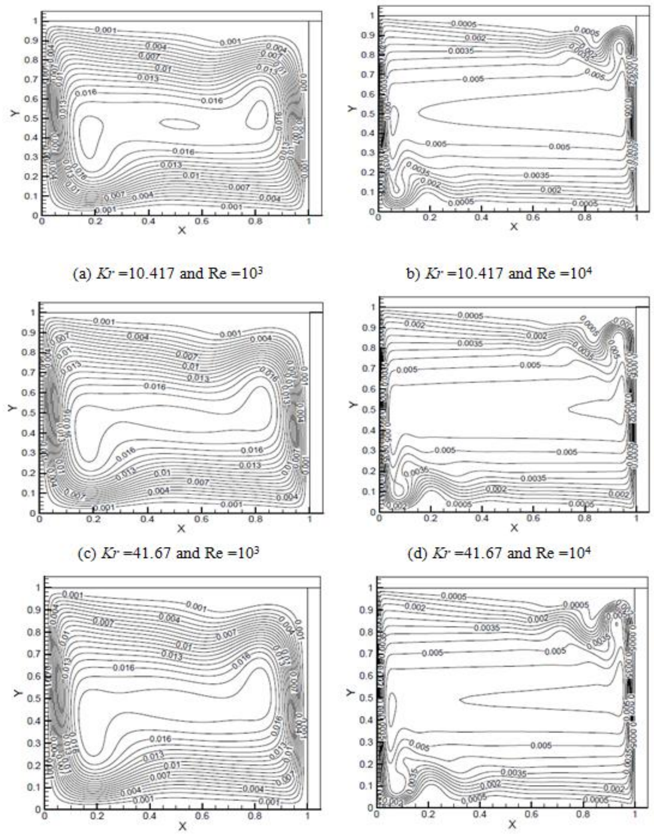
K	0.25	0.5	0.75	1.00	1.25	1.5	1.75	2	2.39
K	10.417	20.83	31.25	41.67	52.08	62.5	72.917	83.33	100

Table : Conductivity ratio

Heat Transfer Characteristics

Isothermal plots shown in figures show the variation of temperature field inside the enclosure for various Reynolds numbers, thicknesses and conductivity ratios.

The variation of temperature inside wall is more noticeable only for the combination of lower conductivity ratio and higher Reynolds number for both of the thicknesses as seen in figures. This temperature distribution changes with increasing conductivity ratio and almost a constant temperature is observed in the side wall. An- other noticeable fact is that the average temperature of the side wall is decreasing with the increase in the Reynolds number, signifying the lower temperature zone or more cooler area inside the enclosure. The atmospheric temperature is assumed to be 0.7 on non-dimensional scale; this explains the perpendicular intersection of the isotherm line of 0.7 with right side wall.



(a) Kr =10.417 and Re =10^3 b) Kr =10.417 and Re =10^4

(c) Kr =41.67 and Re =10^3 (d) Kr =41.67 and Re =10^4

(e) Kr =100 and Re =10^3 (f) Kr =100 and Re =10^4

Figure : streamline plots for t=0.05

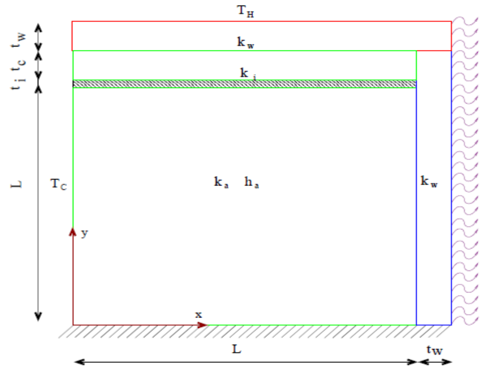
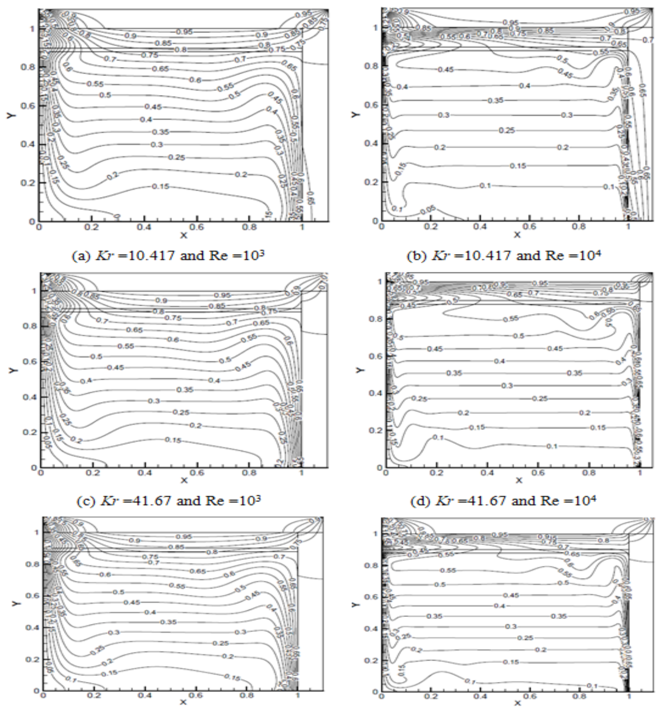


Figure : Boundary conditions of domain with insulation

More interesting observations can be drawn for the average heat loss across side and top wall for various conductivity ratios. As noted in the Tables represent- ing that $Q_{si\ de}$ increases for the initial increase in the conductivity ratio for the thinner wall but it is interesting to note that the further increase of conductivity ratio causes the Q to decrease, this variation is clearer for the higher Reynolds number , contrary to that of the thick wall.

Effect of Insulation near the top wall

To study the effect of insulation, a low thermal conductivity wall is provided at a non dimensional distance of 0.1 from the top. The results are compared and discussed in the following section. Due to the insulation most of the heat is constrained in the cavity above the insulation. This allows the enclosure to maintain at a lower temperature, i.e cooler environment can be maintained within the enclosure. Insulation causes disturbance to the flow near the hot wall and also due to low conductivity of the insulation the heat is trapped in the cavity above it. This can be noted in Table where the comparison can be made for the average temperature for both the cases.



(a) Kr =10.417 and Re =10^3 (b) Kr =10.417 and Re =10^4

(c) Kr =41.67 and Re =10^3 (d) Kr =41.67 and Re =10^4

(e) Kr =100 and Re =10^3 (f) Kr =100 and Re =10^4

Figure : Isothermplots for t=0.1 (with insulation wall)

CASE ⇒	With Insulation									Without Insulation		
$K_r \Downarrow$	θ^e_{avg}			θ^c_{avg}			θ^o_{avg}			θ_{avg}		
$Re \Rightarrow$	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4	10^3	5×10^3	10^4
10.417	0.357	0.292	0.251	0.802	0.715	0.644	0.402	0.335	0.292	0.38	0.325	0.288
20.83	0.367	0.318	0.289	0.812	0.735	0.675	0.412	0.361	0.328	0.391	0.349	0.323
31.25	0.371	0.329	0.304	0.815	0.743	0.686	0.416	0.371	0.343	0.395	0.359	0.338
41.67	0.373	0.335	0.313	0.817	0.747	0.692	0.418	0.377	0.352	0.397	0.364	0.346
52.08	0.374	0.339	0.319	0.818	0.749	0.695	0.419	0.380	0.357	0.399	0.367	0.351
62.5	0.375	0.341	0.323	0.819	0.751	0.698	0.421	0.383	0.361	0.3999	0.37	0.355
72.917	0.376	0.343	0.326	0.82	0.753	0.7	0.421	0.385	0.364	0.401	0.372	0.358
83.33	0.377	0.345	0.329	0.821	0.754	0.701	0.422	0.386	0.367	0.401	0.373	0.36
100	0.378	0.346	0.332	0.821	0.755	0.703	0.423	0.388	0.369	0.402	0.374	0.362

Table : Average temperature of the enclosure for both the cases and wall thickness t = 0.1

V. CONCLUSIONS

Numerical simulation are carried out for steady two dimensional laminar natural convection flow of air in a square enclosure and the results are discussed in the preceding section. various parameters are studied includes Reynolds number, wall thickness, and conductance ratio to investigate the behaviors on the temperature and flow field. From the above outlined result following conclusion is drawn:

1. The thickness of the wall of the enclosure should not be more as the result shows the average temperature is maintained constant for various thickness for high Reynolds number and high conductance ratio, so the average temperature can be maintained in the enclosure by using optimum thickness of the wall.
2. The effect of boundary layer is observed in the high Reynolds number as compared with the low Reynolds number as the driving force increases with the Reynolds number.
3. But as the Reynolds number increases the disturbances could be observed in the enclosure. The perturbations are observed in the streamlines plots at bottom left and top right corner of the enclosure suggesting more effective zone to carry any operation there.
4. Due to installation of insulation, lower temperature could be maintained in the enclosure. Also the heat at the top cavity could be used for other applications like storage or drying purposes. Insulation provided at the given height helps to reduce the temperature by almost 10% which is significant value of decrease in today's competitive world as it will decrease the load on cooling devices.

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