Numerical Analysis of Natural Convection in Rectangular Enclosure with Heated Finned Base Plate

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Abstract
In this paper, steady laminar natural convection heat transfer in 3-D horizontal narrow rectangular enclosure, with heated finned base plate is studied numerically using FLUENT 6.3. The variable parameters used in this study are fin spacing (S/H = 0.875-1.75) and fin height (L/H = 0.25-0.75). The enclosure is heated from bottom wall and is cooled from the opposite top wall while the other walls of the enclosure are assumed to be adiabatic. 3-D steady state continuity, Navier-Stokes and energy equations using Boussinesq approximation are solved. For each case Rayleigh number range ranging from $10^4$ to $3 \times 10^5$ is used. This paper presents the effect of fin height and fin spacing on the fin effectiveness and heat transfer in enclosure. Flow field characteristics in the form of velocity vectors are presented for different cases.

Keywords
CFD, Heat Transfer, Fluid Flow, Natural Convection, Horizontal Enclosure

I. Introduction
Natural convection phenomena in enclosures have become one of the major topics of interest in research due to its applications involved in various engineering applications. Buoyancy driven flows have many applications in thermal engineering since passive cooling of electronic components by natural convection is the least expensive, quietest and most reliable method of heat rejection alternatives. Among these applications involving enclosures are solar energy systems, nuclear reactors and electronic packages of computer components.

Heat transfer and fluid flow characteristics in closed enclosures with differently heated walls mainly depends upon the enclosure orientation, horizontal or vertical. Under certain circumstances, electronic components are packaged within sealed enclosures, while one or more of the walls are cooled. The main source of heat within the medium is electronic components or boards situated in various configurations. In the design of electronic packages, there are strong incentives to mount as much electronic components as possible in a given enclosure. This leads to high power generation density and this may raise the temperature of the packages above the allowable limit [1]. Heat transfer rate from the packages must be maximized to overcome this problem. Using finned surfaces is the most common technique for maximizing heat transfer rate. Fins orientation and geometry of fins array are the main parameters which affects the enhancement ratio of heat transfer.

In literature, Numerical and/or experimental studies involving enclosures with fins or pins are very rare. Numerical experiments are conducted by Arquis and Rady [2] to investigate natural convection heat transfer and fluid flow characteristics from a horizontal fluid layer with finned bottom surface. For a limited low range of Rayleigh number, the effects of fin height, fin spacing and Rayleigh number on fin surface effectiveness have been studied. An experimental investigation of natural convection heat transfer and fluid flow in horizontal and vertical narrow enclosures with heated rectangular finned base plate for different fin spacing and fin lengths were studied by S. A. Nada [1]. He reported that Nusselt number and finned surface effectiveness increased with increasing the fin length. He also reported that the increasing the Ra for any fin geometry increases the Nusselt number it means the heat transfer is enhanced using finned base plates. 2-D Numerical analysis of natural convection in a differentially heated square cavity whose vertical walls have a finite conductivity and a thickness was studied by Raji et al. [3]. His numerical simulations of natural convection in a cavity having walls of finite width have shown the possibility to reduce significantly the heat transfer by using appropriate isolation techniques. The effects of spacing on cooling of heated electronic components and of the removal of heat input in one of the components were determined. R. L. Frederick and S. G. Moraga [4] numerically investigated 3D natural convection in a cubical enclosure with a fin attached to the hot wall and air as fluid. The fin was horizontally attached to the hot wall, and the Rayleigh numbers ranged from $10^3$ to $10^5$ is studied. They concluded that for the above Rayleigh range a fin of partial width is more effective in promoting heat transfer than a fin of full width. A numerical simulation of conjugate turbulent natural convection air cooling of three heated ceramic components, mounted on a vertical adiabatic channel was studied by Bessaih and Kadja [5]. A numerical study has been carried out by E. Bilgen [6] in differentially heated square cavities, which are formed by horizontal adiabatic walls and vertical isothermal walls. He concluded that the heat transfer may be suppressed up to 38% by choosing appropriate thermal and geometrical fin parameters. An experimental investigation to study the effects of vertical fins on heat transfer rate in a horizontal fluid layer in a finite extent was carried out by Inada et al. [7]. For a single value of fin height and for limited range of Rayleigh number and fin spacing, the heat transfer rates have been reported by them. A numerical investigation of steady state natural convection heat transfer in a longitudinally short rectangular fin array on a horizontal base was studied by Mobedi and H. Yücel [8]. The study was limited to Rayleigh number rage ranging from 120 to 39000. The fin length and fin height were varied from 2 to 20 and 0.25 to 7-fin spacing, respectively. The mechanisms of the flows are discussed and flow patterns are plotted by them.

To the best of our knowledge, the detailed three dimensional computational analyses on heat transfer and fluid flow characteristics within a finned horizontal fluid layer is not available in the literature. Therefore, the present work aims to present a comprehensive numerical investigation of heat transfer and fluid flow characteristics in a horizontal fluid layer of narrow closed enclosures with rectangular finned base plate.

Mathematical Formulation:
Model used for present study is a horizontal narrow enclosure with internal dimensions of a x b x H = 280mm x 180mm x 40mm (length, breadth and height). Fin thickness (t=1.5mm) is taken constant for all setups the varying parameter for current study is fin length (10-30mm) and fin spacing (3-7 fins). Dimensionless fin length (L/H=0.25-0.75) and fin spacing (S/H=0.875-1.75) are taken for present study. Fluid inside the enclosure is taken as air ($Pr=0.71$). Results are obtained for each fin array geometry at eight different Rayleigh numbers ranging from $1 \times 10^4$ to $3 \times 10^5$. 

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The enclosure is heated from bottom wall and is cooled from the opposite top wall while the other walls of the enclosure are assumed to be adiabatic. The temperature difference between the hot and cold surfaces supports the buoyancy driven flow inside the enclosure. The temperature difference in this study is assumed to be small enough so that the Boussinesq approximation is valid. The fin array geometry of the model with coordinate system used is shown in fig. 1.

Fig. 1: Schematic of Enclosure With Finned Base Plate Arrangement (for L/H=0.75)

The steady state equations [9] for a Newtonian fluid are given as:
1. Continuity equation
\[ \nabla \cdot \mathbf{V} = 0 \]  
2. Momentum equation
\[ (\nabla \cdot \mathbf{V}) \mathbf{V} = -\frac{1}{\rho} \nabla P + \nu \nabla^2 \mathbf{V} + g \beta (T - T_0) \mathbf{j} \]  
3. Energy equation
\[ (\nabla \cdot \mathbf{V}) T = \alpha \nabla^2 T \]  
Where, \( \mathbf{V} \) is the velocity, \( \rho \) is the density, \( \mathbf{j} \) is the unit vector in \( y \)-direction, \( \nu \) is the viscosity, \( \alpha \) is the thermal diffusivity, \( g \) is gravitational acceleration, \( \beta \) is the thermal expansion coefficient, \( P \) is the pressure and \( T \) is the temperature. The properties of air (\( Pr=0.71 \)) are assumed to be constant for this study and are obtained at 300K temperature. The Rayleigh numbers based on the enclosure height is defined as
\[ Ra = \frac{g \beta (T_h - T_c) H^3}{\alpha \nu} \]  
Where, \( T_h \) and \( T_c \) are the constant temperature of the hot and cold wall respectively.

The mean Nusselt number for the enclosure is calculated as
\[ Nu = \frac{\bar{h} H}{k} \]  
Where, \( \bar{h} \) is the average heat transfer coefficient.

Boundary conditions at walls are taken as:
(1). Velocity boundary conditions:
No slip condition at all walls (\( V=0 \))
(2). Temperature boundary condition:
Bottom wall: \( T_h \)
Top wall: \( T_c \)
Other walls: zero heat flux (\( q'' = 0 \))

C. Numerical Solution
In the present study, 3-D FLUENT version is used and for most of the numerical simulations. The continuity, Navier-stokes and the energy equations are solved using the commercially available code FLUENT® 6.3 [10]. The flow field in the computational domain is solved using SIMPLE algorithm with the second order upwind scheme both for momentum and energy equations. In the present study under-relaxation factors are set to the default values in FLUENT 6.3. Discretization used for pressure term is PRESTO. The absolute convergence criteria of the continuity, x-momentum, y-momentum and z-momentum equations is adopted as 10-3 while the convergence criteria for energy equation is adopted as 10-6. The iteration will continue until the sum of residuals drops below the convergence criterion.

A comparison of the average Nusselt number obtained in this study for different Rayleigh number ranges with the Nusselt number obtained with correlation given by Nada [1] is given in Table 1 for validation. The variation in the numerical results with the experimental results of Nada can be attributed as the numerical simplification of the present case and the varying geometry. It is observed that present solutions is having good agreement with those are obtained by the correlation.

Table 1. Validation of model(L/H=0.75, S/H=0.875)

<table>
<thead>
<tr>
<th>Rayleigh No.</th>
<th>Nusselt Number Correlation by Nada [1]</th>
<th>Present Study</th>
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</thead>
<tbody>
<tr>
<td>1.4x10^5</td>
<td>9.12</td>
<td>9.21</td>
</tr>
<tr>
<td>2.9x10^5</td>
<td>13.91</td>
<td>14.05</td>
</tr>
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</table>

D. Results and Discussion

1. Effect of Fin Length
For a vertical enclosure the dependence of the Nusselt number (\( Nu \)) on fin length (\( L/H \)) for different values of Rayleigh number (\( Ra \)) ranging from 104 to 3x105 is shown in fig. 2. \( Nu \) is plotted against \( Ra \) for different values of \( L/H \) as a parameter. It can be clearly seen that at any Rayleigh number the effect of increasing \( L/H \) increases \( Nu \). This increase in \( Nu \) with Increasing \( L/H \) can be attributed to the increase of heat transfer surface area with increasing \( L/H \). Possibility of formation of separate convection cell between two adjacent fins increases due to increase in \( L/H \) for a vertical enclosure and this leads in an enhancement of heat transfer rate (fig. 4).
2. Effect of Fin Spacing
The dependence of the Nusselt number (Nu) on fin spacing (S/H) for different values of fin lengths (L/H) and at the entire range of Rayleigh number (Ra) is shown in fig. 3. It can be seen that Nu increases with increasing the number of fins (i.e., decreasing S/H) until it reaches a maximum at a certain (S/H) and with further increasing the number of fins, Nu starts to decrease. For optimum performance of fin-arrays in practical applications the conditions at which Nu is maximum is important. As shown in the figures, the maximum Nu occurs at S/H = 1.17 (5 fin case) for present case. This variation of Nu with S/H can be attributed to the dependence of the heat transfer rate on the heat transfer surface area and increase or decrease of the flow intensity with insertion of fins.

C. Flow Patterns Inside the Enclosure at Ra=1.40×10⁵
The formation of circulation cell at Rayleigh numbers $2.45\times10^5$ is shown for four different fin array configurations in Fig.4. On x-y plane at $z=0.1$ the flow is captured in the form of velocity vectors.
colored by velocity magnitude. Colormap showing the velocity magnitude of fluid inside enclosure is also shown. Maximum and minimum intensity of velocity of air inside the enclosure can also be seen clearly from Colormap. It can be clearly seen that the increase in fin length leads to formation of separate circulation cells and the insertion of fin leads to increase in the number of convection cells. The increase in Rayleigh number increases the magnitude of air velocity for flow patterns obtained, thereby increasing the amount of heat to be transferred from the cold wall.

Fig. 4: Velocity Vector Colored by Velocity Magnitude at Ra=2.45x10⁵: (a) L/H=0.25, S/H=0.875; (b) L/H=0.50, S/H=0.875; (c) L/H=0.75, S/H=0.875; (d) L/H=0.75, S/H=1.17; (e) L/H=0.75, S/H=1.75.

E. Conclusion
In present numerical study of natural convection of air in a differentially heated horizontal rectangular enclosure with finned base plate, the variation of average Nusselt number with Rayleigh number, for different fin length and spacing was investigated. The Nu is a strong function of Rayleigh number, fin length and fin spacing. It can be concluded from the present study that for any fin configuration, the increase in fin length always leads to increase in heat transfer rate for any Rayleigh number range. It is observed that Nu increases with increasing the number of fins (i.e., decreasing S/H) until it reaches a maximum at a certain (S/H) and with further increasing the number of fins, Nu starts to decrease. The heat transfer rate also increases with increasing the Rayleigh number. The effect of increasing fin length leads to formation of separate circulation cells for as shown in flow pattern and also the insertion of fins leads to increase in number of convection cells.

References