Heat Transfer Enhancement Study for Electronic Cooling Application

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Abstract

With the advancement of technology, cooling demands has gone up. In order to minimize cost, it is required to find out means to enhance heat dissipation rate without much design modification of electronic component. Here, analysis is carried out to optimize heat transfer in mixed convection regime to ensure minimum pumping power and channel size modifications. A vertical enclosure with a hot plate is considered and the flow is varied to obtain forced and mixed convection. The flow is considered to be laminar with Reynolds number below 2500 and Rayleigh number below. The orientation of the electronic component is also varied to determine the best possible design. Study was conducted by analysing the heat transfer coefficient of a hot plate using air flow. By comparing hot plate at different orientations, it is observed that the heat transfer coefficient value is high when the plate is inclined at an angle of 30 for both mixed and forced convection. It’s found out the power consumption is less in case of mixed convection. So by considering a limiting region were the heat transfer rate is similar for mixed and forced convection, we can choose mixed convection mode as the optimized convection mode.

Keywords: Heat transfer coefficient, mixed convection, Vertical enclosure, pumping power

I. INTRODUCTION

Electronic equipment has made its way into practically every aspect of modern life, from toys and appliances to high-power computers. Electronic components are potential sites for excessive heating, since the current flow through a resistance is accompanied by heat generation. Unless proper cooling mechanism is used, high rates of heat generation result in high operating temperatures for electronic equipment, which affects its safety and reliability during heavy load conditions and failure of current cooling mechanism. Therefore, thermal control has become increasingly important in the design and operation of electronic equipment’s. Also continued miniaturization of the electronic devices from micro to Nano scale has resulted in a dramatic increase in the amount of heat to be dissipated and there is a need for efficient and effective cooling methods.

The traditional method to dissipate heat from electronic components was forced convection using a fan with a heat sink directly. In the past, the method used for solving the high heat capacity of electronic components has been to install a heat sink with a fan directly on the heat source, removing the heat through forced convection. Secondly, the electronic industries thermal design tends to be an afterthought of the design process only if the prototype raises any thermal issues, and – thirdly, the limit of pushing the use of air cooling with heat sink and fan is expected to be reached in the coming years. Therefore thermal management is a key enabling technology in the development of advance electronics. It is a necessary part of any competitive power density environment. Though the new tool and technologies are employed for cooling, there is no remarkable change in the constraints and design requirements.
The conventional techniques generally used for cooling of electronic devices are conduction cooling, natural convection and radiation cooling, forced air cooling, liquid cooling and immersion cooling. But electronic devices are observed to fail under prolonged use at high temperatures. Possible causes of failure are diffusion in semiconductor materials, chemical reactions, and creep in the bonding materials, etc. The failure rate of electronic devices increases almost exponentially with the operating temperature. The cooler the electronic device operates, the more reliable it is.

With the advancement of technology, cooling demands has gone up. In order to minimize cost, it is required to find out means to enhance heat dissipation rate without much design modification of electronic component. Various alternatives are being suggested or effective heat removal from these devices. Electronic components placed in enclosures such as a TV or DVD player are cooled by natural convection by providing a sufficient number of vents on the case to enable the cool air to enter and the heated air to leave the case freely. The combination of the advantages of natural convection along with the forced convection would be an efficient alternative to provide an effective heat transfer. Figure 1.1 shows the cooling of electronic circuits by mixed convection. Considering the pressure loss and power requirement, mixed convection may have an advantage over the forced convection. The alteration in the orientation of these circuit boards will also have impact on the heat transfer characteristics.

II. LITERATURE REVIEW

The traditional method to dissipate heat from electronic components was forced convection using a fan with a heat sink directly. In the past, the method used for solving the high heat capacity of electronic components has been to install a heat sink with a fan directly on the heat source, removing the heat through forced convection. Secondly, the electronic industries thermal design tends to be an afterthought of the design process only if the prototype raises any thermal issues, and – thirdly, the limit of pushing the use of air cooling with heat sink and fan is expected to be reached in the coming years. Therefore thermal management is a key enabling technology in the development of advance electronics. It is a necessary part of any competitive power density environment. Though the new tool and technologies are employed for cooling, there is no remarkable change in the constraints and design requirements.

The failure rates of most electronic components depend on their temperatures. For example, this dependence may involve the maximum temperature, a temperature threshold or the rate of change. Thus, the performance and reliability of boards with electronic devices may depend essentially on the way these devices interact thermally and therefore on their placement. In order to investigate the reliability of such systems, a model is required which accurately describes the evolution of temperature in the system.

Sudo et al. [1][2] carried out experimental investigation of the forced convection, free convection and combined convection on water flows through vertical rectangular channels of gap sizes of 18, 6 and 2.5 mm. They defined the conditions when forced convective heat transfer is dominant and when free convection heat transfer is dominant as well as the region of combined convection. Many experiments and analyses have been so far reported on the combined convective heat transfer characteristics.

Combined forced and natural convection, especially in turbulent flow, is not fully explored. Until recently, the methods most often used to discriminate between forced, mixed and natural convections have been to refer to the classical regime map.
suggested by Metais and Eckert, or to rely on the more classical rule proposed by McAdams: one calculates the heat transfer coefficient from both forced-convection and natural convection relations and then uses the larger value. Shitsman [3] compared heat transfer data for upward and downward flows of water in a heated tube at supercritical pressures and reported that, in the case of upward flow, the temperature distribution along the tube sometimes showed a local temperature rise due to the local impairment of heat transfer, while, for downward flow, heat transfer was stable and better than the upward flow under the same flow rate and the same heat flux.

Hwang et al. [4] conducted an experiment to investigate the convective heat transfer characteristics in fully developed laminar flows of water flowing through a circular pipe with a constant heat flux. Bodoia and Osterle [5] presented the first numerical simulation on natural convection in a two dimensions isothermal vertical channel. Zamora and Kaiser [6] carried out a numerical study on an asymmetrically heated two dimension vertical channel. They covered a large range of configurations with isoflux and isothermal conditions. Many experiments and analyses have been so far reported on the combined convective heat transfer characteristics. This numerical study is carried out in order to understand the forced and mixed convective heat transfer characteristics on a hot plate placed in a vertical rectangular channel.

III. NUMERICAL FORMULATION

A. Problem Definition:

The problem deals with the cooling of an electronic component inside a cabin. So here, a vertical plate is considered as the hot surface. The hot plate is kept inside a duct with insulated walls. In order to analyze the properties of the hot plate at different conditions, a 2-D model of the system is considered as in figure 3.1.1.

![Fig. 3.1.1: Schematic diagram of the enclosure](image)

The properties are assumed to be uniform in the three directions. So here, the various ways by which the heat transfer rate can be enhanced is studied by using a 2-D model. The air flow is varied to obtain mixed and forced convection conditions. The flow is assumed to be laminar with Reynolds’ number ranging below 2500 and Rayleigh number below $10^4$.

B. Geometric Modeling:

The geometric modeling of the defined problem is done using ANSYS Fluent Design Modular. The modeled geometry along with the mesh is shown in figure 3.2.1.
C. Governing Equations and Boundary Conditions

The fluid flow and heat transfer characteristics inside the enclosure is analysed using ANSYS FLUENT 14.0. The governing equations that is the continuity, momentum and energy equations are,

\[ \nabla \cdot \mathbf{v} = 0 \]  
\[ \rho \frac{dv}{dt} = -\nabla p + \mu \nabla^2 \mathbf{v} \]  
\[ \rho C_p \left[ u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right] = k \left[ \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right] + \mu \]  
\[ \mu \phi = \mu \left( \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)^2 + 2 \left( \frac{\partial u}{\partial x} \right)^2 + \left( \frac{\partial v}{\partial y} \right)^2 \]

The flow is considered to be laminar by choosing a particular range of values of Reynold’s number and Rayleigh number from the classical regime map shown in figure 3.3.1. Considering both forced and mixed convection regions, range of Reynold’s number is fixed below 2500 and Rayleigh number below $10^4$.

D. Grid Independence Study:

Grid independence study is done to ensure that the solution is independent on the grid size. Coarse and fine mesh are analysed to obtain similar results which helps in reducing the computational time. In the present study, numerical analysis is performed for natural convection from a vertical enclosure (vertical channel) of dimension 1cmx15cm as shown in figure 3.4.1. A two dimensional, laminar, viscous model is solved using FLUENT which is based on finite volume formulation. Boussinesq approximation is employed and the effect of radiation is neglected.
The boundary conditions used were as follows:

**E. Boundary Conditions:**

- **Inlet:** Velocity, \( V = \text{Constant} \)
- **Hot plate:** \( T = T_s, \) constant temperature
- \( \frac{\partial T}{\partial x} = 0, \) constant heat flux
- **Wall:** \( \frac{\partial T}{\partial y} = 0, V = 0, \) Adiabatic
- **Outlet:** pressure outlet

During the grid independence study, the number of mesh elements was first chosen as 4000 and was further increased by a ratio of 1:2 to obtain values for grid sizes 8000, 16000 and 32000. For selecting an appropriate mesh the average value of Nusselt number and heat transfer coefficient obtained for each grid size were considered as shown in the table 3.4.1. Also the variation of Nusselt number with length is plotted for different mesh sizes as shown in the figure 3.4.2.

<table>
<thead>
<tr>
<th>GRID SIZE</th>
<th>NUSSELT NUMBER</th>
<th>HEAT TRANSFER COEFFICIENT</th>
</tr>
</thead>
<tbody>
<tr>
<td>4000</td>
<td>1.104</td>
<td>0.614</td>
</tr>
<tr>
<td>8000</td>
<td>1.179</td>
<td>0.655</td>
</tr>
<tr>
<td>16000</td>
<td>1.250</td>
<td>0.695</td>
</tr>
<tr>
<td>32000</td>
<td>1.323</td>
<td>0.735</td>
</tr>
</tbody>
</table>

From table 3.4.1, it can be seen that there is not a considerable variation in the Nusselt number obtained for meshes 16000 and 32000. From figure 1 also it can be seen that there is not a considerable difference. Therefore mesh 16000 can be adopted thereby reducing computational time.

The numerical mesh is validated using the experimental results of Hiroaki Tanaka, Shigeo Maruyama and Shunichi Hatano. They conducted an experiment on a vertical pipe with constant heat flux. The experiment was performed for a Grashof number 4.5x10^5 and a Nusselt number of 11.4. The value of Nusselt number obtained for the present work is 16.35, i.e. a difference of
about 30%. This difference could be because of the fact that, in case of experimental setup for natural convection, it is very difficult to achieve steady state. Hence the value obtained may not have been taken at steady state. Thus the present numerical result show satisfactory agreement with the experimental data.

IV. RESULTS AND DISCUSSION

Heat transfer study was conducted for two conditions of the hot plate. In case 1, the study is done at constant temperature. In Case 2, the study is done at constant heat flux.

For each case, the velocity of air is varied to obtain mixed and forced convection mode. The effect of varying the orientation of geometry is analyzed thereafter. The flow is considered to be laminar and 2 dimensional. The air entering the inlet absorbs heat from the hot plate and is delivered at the outlet. The heat transfer for various cases is analyzed and is compared.

A. CASE 1: Constant Temperature:

1) Forced Convection:

In this case, the velocity of air is such that forced convection is predominant at the surface of the hot plate.

Fig. 4.1: Temperature profile for constant temperature forced convection

The orientation of the hot plate section is varied and the heat transfer characteristics are studied. The heat transfer coefficient values are plotted against the length of the hot plate for angles 0°, 50°, 150°, 300° and 400°.

Fig. 4.2: Velocity profile for constant temperature forced convection
From the velocity profile, it is observed that the velocity of air at the hot plate is maximum in case when the hot plate is inclined at 30°. This enables an efficient heat transfer since the air hot air is removed at a faster rate.

The figure shows the comparison of heat transfer coefficient for different geometries along the length of the hot plate for forced convection. From the figure, it is found that the heat transfer is more in the case when the hot plate is kept at an angle of 30°. This is due to the increased velocity of air over the plates.

2) Mixed Convection:
Here, the velocity of air is varied such that both forced convection and natural convection predominantly occur at the surface of the hot plate. The geometry of the hot plate section is varied and the heat transfer characteristics are studied. The heat transfer coefficient values are plotted against the length of the hot plate for angles 0°, 5°, 15°, 30° and 45°.

Fig. 4.3: Comparison of heat transfer coefficient for constant temperature forced convection

Fig. 4.4: Temperature profile for constant temperature mixed convection
The figure shows the comparison of heat transfer coefficient for different geometries along the length of the hot plate for mixed convection. From the figure, it is found that the heat transfer is more in the case when the hot plate is kept at an angle of 30°.

**CASE 2: Constant Heat Flux:**

1) **Forced Convection:**

In this case, the velocity of air is such that forced convection is predominant at the surface of the hot plate. The geometry of the hot plate section is varied and the heat transfer characteristics are studied. The heat transfer coefficient values are plotted against the length of the hot plate for angles 0°, 5°, 15°, 30°, and 40°.
The figure shows the comparison of heat transfer coefficient for different geometries along the length of the hot plate for forced convection. From the figure, it is found that the heat transfer is more in the case when the hot plate is kept at an angle of 30°.

2) Mixed Convection:
Here, the velocity of air is varied such that both forced and natural convection predominantly occur at the surface of the hot plate. The geometry of the hot plate section is varied and the heat transfer characteristics are studied. The heat transfer coefficient values are plotted against the length of the plate for angles 0°, 5°, 15°, 30°, and 40°.

Fig. 4.8: Velocity profile for constant heat flux forced convection

Fig. 4.9: Comparison of heat transfer coefficient for constant heat flux forced convection

Fig. 4.10: Temperature profile for constant heat flux mixed convection
The figure shows the comparison of heat transfer coefficient for different geometries along the length of the hot plate for mixed convection. From the figure, it is found that the heat transfer is more in the case when the hot plate is kept at an angle of 30°.

**C. Optimization of Orientation of Geometry:**

In order to optimize the orientation of geometry to maximize the rate of heat transfer, different orientations were considered. The velocity of flow was varied to obtain forced and mixed convection mode for both constant temperature and constant heat flux conditions. The heat transfer coefficient values for different orientations were compared for each of the cases separately. It was observed that the maximum heat transfer values were obtained when the hot plate was inclined at 30° in all cases. So the optimized orientation of the hot plate is chosen as 30°.

**D. Comparison of Mixed and Forced Convection:**

In the above all cases we have seen that the heat transfer is found to be maximum when the hot plate is inclined at an angle of 30°. So we have selected the optimized orientation to be 30°. Here we compare the heat transfer coefficient values of mixed and forced convection mode for both constant temperature and constant heat flux conditions at 30° inclination.
From these graphs it can be observed that the heat transfer coefficient value is higher for forced and mixed convection at 30° inclination. Mixed convection involves less power consumption than forced convection but for enhancing heat transfer, forced convection should be used. So there is a possibility that if a limiting region is considered where forced and mixed convection gives similar heat transfer rate, the optimized regime can be chosen as the mixed convection regime.

V. CONCLUSIONS

This study investigated the methods to enhance heat transfer in electronic devices by focusing on the design and orientation of electronic circuit boards and also the effects of both forced and mixed convection modes. For this study the 2D model of a vertical rectangular duct was considered with a hot plate placed inside the insulated walls. The velocity of air flow was varied to obtain mixed and forced convection modes. The flow is assumed to be laminar with Reynolds number value below 2500 and Rayleigh number below $10^4$. The heat transfer coefficient values for forced and mixed convection modes were compared and plotted along the length of the hot plate for constant temperature and constant heat flux conditions. Temperature and velocity profiles were also obtained.

Thereafter the orientation of the hot plate was varied from 0° to 40° and the heat transfer values for all cases were compared. It was observed that, for all these cases maximum heat transfer occurred when the hot plate was inclined at 30°. So the optimized orientation was chosen to be 30°. The heat transfer coefficient values for force and mixed convection were compared for this optimized orientation and it was found that the heat transfer rate is more in case of forced convection. But the power consumption is less in case of mixed convection. So by considering a limiting region were the heat transfer rate is similar for mixed and forced convection, we can choose mixed convection mode as the optimized convection mode.
REFERENCES


