**Experimental investigation of natural convection heat transfer of the fin arrangement on a computer heat sink**

Mahdi Fahiminia¹*, Mohammad Mahdi Naserian, Hamid Reza Goshayeshi and H. M. Heravi

Department of Mechanical Engineering, Mashhad Branch, Islamic Azad University, Mashhad, Iran.

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Needs for buoyancy driven ventilation appear in a variety of engineering applications, ranging from cooling of electronic components to cooling of nuclear reactor fuel elements. Hence, it becomes crucial to optimize the natural convection heat transfer as far as possible. If the fins are properly designed, they become very attractive for these applications since they offer an economical, trouble-free solution to the problem. This paper reports an experimental and numerical study of natural convection heat transfer from rectangular fin arrays on a vertical base. An experimental set-up was constructed and calibrated; six sets of fin-arrays and a base plate without fins were tested in atmosphere. A correlation was presented relating the convection heat transfer rate of fin arrays relative to that for base plate without fins with the relevant non-dimensional parameters. In order to improve the natural convection heat transfer rate, air gap was used for the optimum fin spacing.

Key words: Natural convection, experimental optimization, air gap, rectangular fin arrays.

**INTRODUCTION**

Since the rapid development of electronic technology, electronic appliances and devices now are always in our daily life. Under the condition of multifunction, high clock speed, shrinking package size, and higher power dissipations, the heat flux per unit area increased dramatically over the past few years. Besides, the working temperature of the electronic components may exceed the desired temperature level. Thus, the effective removals of heat dissipations and maintaining the die at a safe operating temperature have played an important role in insuring a reliable operation of electronic components.

There are many methods in electronics cooling, such as airflow cooling, heat pipe, etc. Conventional electronics cooling normally used airflow with heat sink showing superiority in terms of unit price, weight and reliability. Therefore, the most common way to enhance the air-cooling is through the utilization of airflow on a heat sink. In order to design an effective heat sink, some criterion such as a large heat transfer rate, a low pressure drop, an easier manufacturing, a simpler structure, a reasonable cost and so on should be considered.

Elenbass (1942) presented the article on heat dissipation of parallel plates by natural convection. He concluded that in the limit of small gap width, Nusselt number is proportional to Rayleigh number. Van de pol and Tierney (1973) studied the free convective Nusselt number for U-shaped channels. They developed a mathematical relationship for laminar flow heat transfer from constant temperature U-shaped channels. The experimental results for the optimal fin spacing of rectangular fins protruding from a vertical rectangular base are collated from Leung et al (1986), Güvenç and Yüncü (2001) and Yazıcıoğlu and Yüncüare (2007), analyzed in such a way that the separate roles of fin height, fin spacing and fin base to ambient temperature difference on convective heat transfer rate from the fin...
arrays are demonstrated. Then, data are rearranged to be presented in terms of the non-dimensional parameters as suggested by the intersection of asymptotes method proposed in Bejan and Scibubba (1992). The equations of the curves, which are provided to fit non-dimensional optimal fin spacing and maximum convective heat transfer rate, are obtained by least square regression.

When the literature is surveyed, a number of scholars have examined the airflow impingement on a heat sink in geometries, materials, and thermal resistance extensively. They carried out the correlation equations for optimum fin spacing and the maximum thermal conductance. The results revealed that the best performance occurred when the distance of two fins was between 6 and 9 mm, and when the Grashof number was between 3,100,000 and 4,600,000. Results for the average heat transfer coefficient were correlated as a function of the fin spacing, fluid properties and geometric parameters of the heat sinks. In addition, the numerical predictions were compared with the experimental results obtained in an earlier study. They concluded that the numerical analysis in a previous study was adequate for appraising the mean heat transfer rate in air impingement for situations of thermal management of electronics. Furthermore, optimization studies were discussed to quantify the effects of changing the fin length and the fin cross sectional area on the cooling performance. The numerical results illustrated a complex- pressure gradient inside the fin array and a greater pressure gradient improved mixing and heat transfer. Moreover, it revealed that the influence of fin width is more obviously than the fin height. Duan and Muzychka (2006) performed the experimental investigation of the thermal performance with four heat sinks of various fin spacing, fin heights. They developed a heat transfer model to predict the thermal performance of air-cooled plate fin heat sinks for design purposes.

Moreover, some scholars have interests in altering the fin shape to enhance the thermal and hydraulic performance. Lorenzini and Moretti (2007) analyzed the Y-shape fins and examined the geometries by varying the angle between the two arms of the Y and proposed new shape for the fins. Guglielmini et al. (1987) performed the experimental and numerical study of heat sinks with horizontal fin arrays. This study discussed the thermal performance on the orientation of the fin array and the tilting of the crests of the plate fins. Shah et al. (2002) have presented a study of a new and unique high-performance air-cooled impingement heat sink. They demonstrated the results of a numerical analysis of the performance of an impingement heat sink designed for use single unit. Their study extended the previous work by investigating the effect of removal of a fin material from the end fins, the total number of fins. Increasing the number of fins indicates a small drop in temperatures, accompanied by a significant pressure rise. The aforementioned references show the necessity of studying the optimization of vertical fins on vertical base plate. Then in this paper, the computational fluid dynamics (CFD) simulations and experimental study are carried out on vertical base plate.

EXPERIMENTAL APPARATUS

An experimental set-up was constructed and calibrated. Six sets of fin-arrays and a base plate without fins were tested in atmosphere. Fin spacing was varied from 3.9 to 18.6 mm. The base-to-ambient temperature difference was also varied independently and systemically with the power supply to heater ranging from 8 W to 15 W. Fins length and fins thickness was fixed at 29.2 and 1 mm, respectively. The experimental program was conducted to clearly delineate the separate roles of fin spacing and base-to-ambient temperature difference. It was found that for a given base-to-ambient temperature difference the convection heat transfer rate from fin-arrays takes on a maximum value as a function of fin spacing. For a given base-to-ambient temperature difference the enhancement of the convection heat transfer rate of fin-arrays relative to that for base plate without fins is strongly dependent on the fin spacing to fin height ratio and number of fins. Experimental investigation of notched fin array indicated improvement to the tune of more than 20%. The present paper attempts to establish a match between the experimental results and the results obtained by using CFD software. Both, the flow patterns as well as the trend of heat transfer coefficient are found to be within 5% range.

The setup consists of various instruments for measuring the ambient temperature, rear surface of the main heater temperature, the temperature of the front surface of guard heater the temperature of the air (as close as possible to fin configuration), close to fin configuration, and the input power of both heaters. Homogeneous and constant heating is ensured by using a thermo foil laminar heater that is attached to the rear of the base plate in fin configuration. Special attention was paid for the insulation of the rear surface of the main heater. To choose a suitable insulation, we know that only the thickness (L) and the thermal conductivity (k) of insulation are important for the conduction of heat through it. Hence, the heat loss through plate can be eliminated by letting either $L \to \infty$ or $k \to 0$. Since a plate of $L \to \infty$ or $k \to 0$ is physically impossible, the foregoing insulation may never be accomplished in the absolute sense. However, the larger the thickness or the smaller the thermal conductivity, the better the insulation will be if the heat loss through plate is desired to be completely eliminated, the use of another heater becomes necessary. The second heater, often referred to as the guard heater, is an important experimental tool for the control of heat transfer (Arpaci, 1966).

Then, by properly adjusting the power supply to the guard heater, all internal energy generated in the main heater may be transferred through other surfaces of it. Therefore, the gap between the rear surface of the main heater and the front surface of guard heater is filled by epoxy and hardener. The temperature of the surfaces of epoxy plate, which are in contact with heaters, is adjusted at the same degree, which almost eliminates the heat loss from the rear surface of main heater. The lateral surfaces of both heaters and the rear surface of the guard heater are insulated by rock wool. The experiments are conducted with constant temperature in air circumference excluding air currents. Figure 1 shows the power circuit. The electrical power was supplied through a regulated alternating current (AC) power supply. The input power of the main heater could be selected precisely by feeding the power to a variable transformer (variac). The voltage drop and the current flow were monitored by a voltmeter-ammeter combination; the supplied power is calculated by multiplying the voltage drop and the current through the heater.

The temperatures beside the heat sink are presented by temperature patterns. For given $Y_{exp}$ locations, we extract temperature profiles, expressed as temperature vs the distance
perpendicular to the heat sink of the measure \( T = f(Z_{exp}) \). Figure 2 shows temperature profile beside the vertical plate. Equations for these curves are obtained by polynomial fitting.

The calculated slope, which is deduced from curve equation, allows us to calculate local heat transfer coefficient, \( h_{y} \). Therefore, the local Nusselt number is deduced from Equation 1.

\[
\text{Nu}_{y} = \frac{h_{y} Y_{exp}}{k} = \frac{-k}{k(T_{i} - T_{e})} \left( \frac{Y_{exp}}{Z_{0} - Z_{exp}} \right) = \frac{-4(T_{e} - T_{i} - 3T_{e})}{2\Delta T_{exp}(T_{i} - T_{e})} \quad (1)
\]

Where \( (Z_{exp}) \) represents the space step in perpendicular direction to the plate, and \( T_{i} \) (i=1, 2, 3) is the successive temperature which is measured along a profile, beside the heat sink, along a profile. A schematic view of experimental setup and experimental apparatus is presented in Figure 3.

Figure 4 shows the configuration of the six types of the heat sinks. The material of heat sink is selected as aluminum alloy 6061 and has a thermal conductivity value of 168. The areas of the both heaters are 80 X 59.8 mm. The accuracy of experimental results depends upon the accuracy of the individual measuring instruments and the manufacturing accuracy of the test section. Based on the error theory, the total uncertainty, \( U \) comprises of uncertainties of many experimental parameters, which influence on the experiment. For a value of \( M \), whose results depend on uncorrelated input estimates \( x_{1}, x_{2}, \ldots, x_{N} \), the standard measurement uncertainty is obtained by appropriately combining the standard uncertainties of these input estimates. The combined standard uncertainties of the estimate \( M \) denoted by \( U \) is calculated (Moffat, 1988).

Where \( f \) is the function of \( M \) in terms of input, estimates \( x_{1}, x_{2}, \ldots, x_{N} \) and \( U(x) \) is a standard input uncertainty. Table 1 lists the experimental values of the directly measured parameters and the associated uncertainties. Thus, the experimental uncertainties of the convection heat transfer coefficient were estimated to be 3.5%.

The denotations and dimensions of the aluminum heat sinks with different uniform fin designs in this study are depicted in Table 2. In Table 2, the geometric parameters of Type-a (vertical plate) are the same as the one of the Brodowicz studies (Brodowicz, 1968). In addition, the designs from Type-b to Type-f are extended from the Type-a. The effects of the Nusselt number and type of heat sink design on the junction temperature and thermal performance were examined. The parameters used in this study include two Rayleigh numbers (\( Ra_{L} = 2.13 \times 10^{6} \) and \( Ra_{L} = 3.37 \times 10^{6} \)). Moreover, we compare five designs with the vertical plate design.

Mathematical model

The basic geometry of fin configuration that is studied in this article, and is fabricated by wire cut machine tool, is shown in Figure 4. The numerical simulation was conducted using fluent V6.3, a
commercially available computational fluid dynamics (CFD) code based on the finite volume method. The convergence of the computational solution is determined based on residuals for the continuity and energy equations. Solid modeling, computational grid generation and meshing were done using Gambit Preprocessor 2.2.30 Software. In Gambit session, the base plate, the fin configurations and heaters were defined as the walls; all other remaining boundaries were defined as pressure outlet.

The grid dependence was investigated by varying the number of grid points from 22,680 to 285,714. We selected 65,016 grid points. Governing equations are solved using a finite volume approach. The convective terms are discretized using the power-law scheme, whereas for diffusive terms the central difference is employed. Coupling between the velocity and pressure is made with simple algorithm. The resultant system of discretized linear algebraic equations is solved with an alternating direction implicit scheme. The geometry of coordinate system which is used in this study and velocity boundary layer is shown in Figure 5.

The governing equations in this study are as follows:

Continuity equation:

\[
\frac{\partial (\rho u)}{\partial x} + \frac{\partial (\rho v)}{\partial y} + \frac{\partial (\rho w)}{\partial z} = 0
\]  

(2)

X-Momentum equation:

\[
\frac{\partial (\rho u^2)}{\partial x} + \frac{\partial (\rho uv)}{\partial y} + \frac{\partial (\rho uw)}{\partial z} = -\frac{\partial p}{\partial x} + \mu \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right)
\]  

(3)

Y-Momentum equation:

\[
\frac{\partial (\rho u) \frac{\partial v}{\partial y} + \frac{\partial (\rho v^2)}{\partial y} + \frac{\partial (\rho vw)}{\partial z} = -\frac{\partial p}{\partial y} + \mu \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) + g \left( \rho - \rho_o \right)
\]  

(4)

Z-Momentum equation:

\[
\frac{\partial (\rho w)}{\partial x} + \frac{\partial (\rho wv)}{\partial y} + \frac{\partial (\rho w^2)}{\partial z} = -\frac{\partial p}{\partial z} + \mu \left( \frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right)
\]  

(5)
Table 1. Measuring instruments range and accuracy.

<table>
<thead>
<tr>
<th>Description</th>
<th>Model</th>
<th>Range</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature of ambient</td>
<td>K type</td>
<td>-200 to 1000</td>
<td>± 0.1 C</td>
</tr>
<tr>
<td>Temperature of surface</td>
<td>K type</td>
<td>-200 to 1000</td>
<td>± 0.1 C</td>
</tr>
<tr>
<td>Voltage measurement</td>
<td>Variac</td>
<td>0 to 400</td>
<td>± 0.1 V</td>
</tr>
<tr>
<td>Ampere-meter</td>
<td>Multimeter</td>
<td>0 to 20</td>
<td>± 0.1 A</td>
</tr>
<tr>
<td>Length measurement</td>
<td>Caliper</td>
<td>6</td>
<td>± 0.001 m</td>
</tr>
</tbody>
</table>

Table 2. Dimensions of the Fin configurations.

<table>
<thead>
<tr>
<th>Fin shape</th>
<th>Fin height H(mm)</th>
<th>Fin spacing (mm)</th>
<th>Number fin (n)</th>
<th>Gap 10 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type-a</td>
<td>N.A</td>
<td>N.A</td>
<td>N.A</td>
<td>N.A</td>
</tr>
<tr>
<td>Type-b</td>
<td>29.2</td>
<td>3.9</td>
<td>13</td>
<td>N.A</td>
</tr>
<tr>
<td>Type-c</td>
<td>29.2</td>
<td>7.4</td>
<td>8</td>
<td>N.A</td>
</tr>
<tr>
<td>Type-d</td>
<td>29.2</td>
<td>8.8</td>
<td>7</td>
<td>N.A</td>
</tr>
<tr>
<td>Type-e</td>
<td>29.2</td>
<td>18.6</td>
<td>4</td>
<td>N.A</td>
</tr>
<tr>
<td>Type-f</td>
<td>29.2</td>
<td>7.4</td>
<td>8</td>
<td>Available</td>
</tr>
</tbody>
</table>

Energy equation:

\[
\frac{\partial (\rho u T)}{\partial x} + \frac{\partial (\rho v T)}{\partial y} + \frac{\partial (\rho w T)}{\partial z} = K \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right)
\]  

(6)

The density of air was calculated from the ideal gas law,

\[
\rho = \frac{P_{atm}}{(R_{\gamma} / M_a) \gamma T}
\]

(7)

For the numerical analysis, the following assumptions were imposed. The flow was steady, laminar, and three-dimensional; a side from density, the properties of the fluid was independent of temperature. Air density was calculated by treating air as an ideal gas and radiation heat transfer was negligible.

The Grashof number is defined as,

\[
Gr = \frac{g \beta TL^3}{\nu^2}
\]

(8)

Where L denotes the fin length.

The average convection heat transfer coefficient \( \bar{h} \) is calculated by,

\[
\bar{h} = \frac{Q}{A_h (T_{b} - T_{a})}
\]

(9)

The average Nusselt number is calculated by,

\[
\bar{Nu} = \frac{\bar{h} L}{k_s}
\]

(10)

Where \( k_s \) is evaluated at the film temperature \( T_{film} \).

Following the definition, the coefficient of enhancement (COE) is defined to quantify the improvement in heat transfer rates due to the difference types of the heat sink fins. This is expressed as:

\[
COE = \frac{Nu_{new}}{Nu_{vertical plate}}
\]

(11)

The thermal resistance of the heat sink is calculated by,

\[
R_{th} = \frac{T_{a} - T_{b}}{Q}
\]

(12)

RESULTS

The numerical and experimental results are validated. In this study, to verify the numerical and experimental results, the results which are gained from this study for laminar air flow over a vertical plate with length 0.08 m was plotted with Churchill and Chu (1995), Equation 13, and McAdams (1954), Equation 14, correlations, as displayed in Figure 6. These results reveal that except for the McAdams’ correlation, which is a very rough correlation, the experimental and numerical results are to the same extent in good agreement the results from literature. The average error between experimental and numerical results is become 6.0%.

\[
Nu = 0.68 + \frac{0.67(Ra)^{1/2}}{1 + \left( \frac{0.492}{Pr} \right)^{1/7}}
\]

(13)
In general, there are many testing processes for heat sinks, which must be introduced in an effort to obtain the thermal and hydraulic performance of heat sinks. If we take advantage of the numerical simulation to obtain some probable optimal design parameters before running experiments, the cost and research time can be reduced. In this paper, the experimental study and numerical simulation of natural convection heat transfer from rectangular fin arrays on a vertical base will be investigated. In order to improve the natural convection heat transfer rate, air gap was used for the optimum fin spacing. Finally, for getting better conclusions, all the types were compared with vertical flat plate without fins. As seen in Figure 7, the area decreases as the fin shape changes from Type-b to Type-e and heat transfer rate among the new designs (Type-b to Type-f) appeared to be minimal in Type-e and maximal in Type-f. While Type-b has the maximum surface among considered models, it has lower heat transfer rate than optimal type. Although Type-f has not the largest area, it has the highest heat transfer rate.

Figure 8 depicts the trend of thermal performance for heat sinks. This figure shows that the temperature difference increases when the fin shape changes from Type-c to Type-e and the COE increases as the temperature difference increases. As the fin shape changes from Type-c to Type-e, the temperature difference increases from 95.13 K to 146.31 K at Ra = 2,130,000 to 3,370,000. Generally speaking, the COE increases as the temperature difference decreases. The effects of the fin design are more obvious at low Prandtl numbers. Although the influences of design parameters of heat sink considered for comparison of the temperature difference and the COE of heat sink are not so significant, the numerical approach has proven for the evaluation of performance.

The effect of the number of fins on the thermal resistance and heat transfer coefficient is shown in Figure 9. The average heat transfer coefficient decreased as the number of fins increased, since the flow rate of the cooler air entering the spaces between the fins decreased and the air was heated more quickly on account of the reduced space between fins. However, when the number of fins was less than 8, the thermal resistance of the heat sink decreased with increasing n, since the effect of the increased heat transfer surface area was larger than the

\[
\text{Nu} = 0.59 (Ra)^{1/4} \quad 10^4 < Ra < 10^9
\]

In general, the experimental and numerical results agree well. Figure 6 shows the comparison of results from various experiment and numerical simulations. Figure 7 compares heat transfer rate and area different type of heat sink.
Figure 9. The effect of the number of fins (power=8 W).

The effect of the decreased heat transfer coefficient. When the number of fins was greater than 8, the thermal resistance of the heat sink increased with increasing $n$, since the heat transfer coefficient was very small. Consequently, there exists an optimum number of fins that gives the minimum thermal resistance.

Figure 10 illustrates the effect of the heat flux applied to the heat sink base. The decrease in thermal resistance due to increasing heat flux resulted in a greater rising air velocity, which in turn increased the flow rate of the cooler air entering from outside. Accordingly, the average heat transfer coefficient increased almost linearly, thanks to the enhanced effect of natural convection.

Based on experimental data that is gathered, the variation of heat transfer coefficient of fin configurations versus dimensionless parameter of fin spacing ($S/L$) is shown in Figure 11. At a given difference between base and ambient temperature, the natural convection heat transfer coefficient of a fin array increases with respect to the increase in fin spacing and reaches a maximum. In this case study, the equation for optimum fin spacing as:

$$S_{opt} = 3.25(\rho)^{1/2} \left( \frac{\mu L K_a}{\beta g(T_b - T_a) c_p} \right)^{1/3}$$

(15)

The modified Rayleigh number is defined as:

$$Ra' = Gr \frac{S}{L} = \frac{g \beta (T_b - T_a) L^3 c_p S}{\mu K_a}$$

(16)

Rational curve fitting of this data, (Power function) Equation 17 results in the determination of the correlation of natural convection heat transfer coefficient versus $S/L$, Equation 18.

$$y = A \times |X - X_c|^\beta + Y_0$$

(17)

$$Nu = 0.58(\rho L)^{1/2} \left( \frac{\beta g(T_b - T_a) Sc}{\mu K_a} \right)^{1/3}$$

(18)

As shown in Figure 12, by comparing the obtained result and the results, which are gained from previous study (Kraus and BarCohen, 1983; Simons, 2002) are validated. The experimental and numerical results are to the same extent in good agreement with the results from literature. At low Rayleigh number, by increasing the Rayleigh number, slope of the curve decreases and development rate of the Nusselt number increased.

Creating air crack in heat sinks causes a good air current between fins and decreases the temperature difference between fin surface and environment. Also we should not choose a small distance between fins for they will become a barrier against Fluor current on the sheet and decrease the coefficient of heat transfer but when the fins are apart from each other there is less limitation for the currents on the fins, and by increasing the number of fins, this limitation will decreases and has the maximum coefficient of heat transfer that without decreasing cross section, has the least limitation for the current on the sheet. Figures 13 and 14 show contour of temperature and contour of free streamlines in the gap between the fins for optimum heat sink, which was chosen among studied cases. High rate of air flowing between the fins could be seen clearly.

Figure 15 shows heat transfer coefficient contour of heat sink with optimum fin spacing. To dig out the reason why by increasing the number of fins, more than optimum number, natural convection heat transfer decreases, the heat transfer coefficient and temperature distributions are investigated for type $g$. According to the figure, for each fin spacing, the thermal boundary layer is formed in the vicinity of lateral faces of the pieces and as it goes distant from pieces, this boundary layer vanishes. However, due
Figure 11. Variation of heat transfer coefficient of fin configurations with S/L.

Figure 12. Variation nusselt number by modified Rayleigh number.

Figure 13. Variation of temperature contour of the heat sink.
to the boundary layers interferences, the boundary layers would slowly dissipate in the air-flow that, in turn, would result in an extremely thick boundary layer. The thicker the boundary layer, the less the heat transfer coefficient will be.

**Conclusion**

The comparisons between numerical simulation and experimental data of the heat sink performance have been conducted. The effects of the fin thickness, inter-fin gap, and fin shape on the heat transfer and pressure drop have been studied. The study showed that the pressure drop can be decreased by cutting the fins in the central impingement zone without sacrificing heat transfer rates. The effects of the shape of the heat sink fins, particularly near the center of the heat sink were examined. It is found that removal of fin material from the central region of the heat sink enhances the thermal as well as the hydraulic performance of the sink.

**REFERENCES**


