Enhancement of Heat Transfer from Plate Fin Heat Sinks

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ABSTRACT

Heat Sinks are an extremely useful component used to lower the maximum temperature of various electronic devices during operation so as to increase their thermal efficiency and performance. Fins constitute an important and integral component of sinks. It is a passive cooling technique. Plate fin heat sinks are used in varied applications owing to its low manufacturing cost, ease of manufacture and its economical way to dissipate unwanted heat. Steady state natural convection is experimentally investigated for 06 sets of vertically mounted fin heat sinks. Aluminum is used because of its high conductivity. Length and fin thickness is kept constant at 200 mm and 05 mm respectively. Fin height is successively increased as 10mm, 30mm, 50mm. For the second set length is taken as 100mm. Height is varied in the range 05mm, 15mm and 25mm. Aspect Ratio for above sets is thus 0.05, 0.15 and 0.25 respectively. Effect of varying height, heat input and aspect ratio, keeping length constant is investigated on heat transfer through the sinks.

Keywords — Heat Sinks, Fins, Natural Convection, Aspect Ratio.

I. INTRODUCTION

Heat sink is an object that absorbs and dissipates heat from another object by thermal contact. Heat sinks find use in wide range of applications, providing efficient heat dissipation. Some applications include cooling of electronic devices, lasers, heat engines and refrigeration. Thermal energy transfer between two objects rapidly brings first object into thermal equilibrium with second lowering temperature of first object thus accomplishing heat sinks role as a cooling device. If temperature limits exceed a certain value in above applications it may even lead to total system failure. Therefore different methods are used by engineering systems so as to minimize this overheating problem as much as possible. Fins form one of the easiest and cheapest ways to dissipate this heat. Owing to low production costs and high effectiveness, rectangular fins are the most popular fin type. Vertical orientation of fins is widely used from a combination of horizontal and vertical orientation as it is highly effective. Heat dissipation from fins to surroundings takes place by two modes convection and radiation. As aluminium is used for fins, it has low emissivity value hence radiation heat transfer value is low. So convection heat transfer is dominant mode of heat transfer from fins. Heat transfer coefficient and surface area of fins are two important parameters on which rate of heat dissipation depends. Fluid can be forced to flow over fins by fans and in this manner heat transfer coefficient ‘h’ can be increased. But it is costly and requires more volume to include fan. Surface area can be increased by addition of more fins. But distance between adjacent fins reduces due to addition of more fins. This may result in offering resistance to air flow thereby reducing heat transfer coefficient.

II. EXPERIMENTATION

Main components include channel, concrete block, base plate, fin array, plate heater and power mains. Aerated concrete block 250x200x100 mm, is fixed on frame which ensures only one dimensional heat dissipation. A removable acrylic sheet is placed on front surface to replace fin arrays. Base thickness of arrays is kept as 05mm. The heater consists of nichrome wire wound around...
thin mica plate and mica sheet. Rating of heater plate is 250W and 230 V, AC. 0.05 mm depth is provided on aerated concrete block to include heater plate. Extruded surface is kept over fin array for fitting to aerated concrete block. Concrete block has in-built 04 bolts to tighten fin array over heater plate. Thus air gap between fin array and heater plate is considered to be negligible. A concrete block has high insulation quality and high temperature resistance. 

\[ K = 0.15 \text{ W/m} \text{k} \]

**TABLE I**

<table>
<thead>
<tr>
<th>Sr.No</th>
<th>Component</th>
<th>Dimensions(mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Frame Channel</td>
<td>550x550x550</td>
</tr>
<tr>
<td>2</td>
<td>Heater</td>
<td>180×200×5</td>
</tr>
<tr>
<td>3</td>
<td>Concrete</td>
<td>250x200x100</td>
</tr>
</tbody>
</table>

Fin configurations are produced on D.R.O milling. The fin arrays are produced from rectangular bars with dimensions 180x200x60 mm. Fins are integral with base plate of thickness 05mm and fin thickness is kept constant at 05mm.

**TABLE II**

<table>
<thead>
<tr>
<th>Block No.</th>
<th>Fin Length (mm)</th>
<th>Fin Width (mm)</th>
<th>Fin Thickness (mm)</th>
<th>Fin Height (mm)</th>
<th>No. of Fins</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>100</td>
<td>180</td>
<td>05</td>
<td>05</td>
<td>13</td>
</tr>
<tr>
<td>2</td>
<td>100</td>
<td>180</td>
<td>05</td>
<td>15</td>
<td>13</td>
</tr>
<tr>
<td>3</td>
<td>100</td>
<td>180</td>
<td>05</td>
<td>25</td>
<td>13</td>
</tr>
<tr>
<td>4</td>
<td>200</td>
<td>180</td>
<td>05</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td>5</td>
<td>200</td>
<td>180</td>
<td>05</td>
<td>30</td>
<td>10</td>
</tr>
<tr>
<td>6</td>
<td>200</td>
<td>180</td>
<td>05</td>
<td>50</td>
<td>10</td>
</tr>
</tbody>
</table>
### III. EQUATIONS

1. Heat Loss by Convection, $Q_c$:
   
   $$ Q_c = (\text{heater input}) - (\text{heat loss by radiation, } Q_r) $$

   $$ Q_r = \varepsilon \sigma A (T_{avg}^4 - T_{amb}^4) $$

2. Average Heat Transfer Coefficient,
   
   $$ h_{avg} = \frac{Q_c}{A \Delta T} $$

3. Mean Film Temperature:
   
   $$ T_f = \frac{T_{avg} + T_{amb}}{2} $$

4. Experimental Nusselt Number:
   
   $$ N_u_{avg} = \left( \frac{h_{avg} L}{K} \right) $$

   $$ Ra = \frac{g \beta L (T_{avg} - T_{amb})}{\mu} $$

### IV. EXPERIMENTAL PROCEDURE

Heat inputs can be adjusted by a dimmer stat. The temperature of heat sink at different locations and ambient temperatures are recorded at time interval of 30 minutes till steady state is reached. Generally it takes around 02 hrs to attain steady state condition. Temperature variation of around 0.5°C is taken for steady state approximation. Six thermocouples are used. Five of them are attached to the base and one is kept suspended inside the channel to record ambient temperature.

Parameters used for study are as follows:

- Heat Input (Qin) : 20W, 40W, 60W, 80W, 100W
- Base Temperature : $T_1, T_2, T_3, T_4, T_5$
- Ambient Temperature : $T_6$

### OBSERVATIONS

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Q</th>
<th>$T_1$</th>
<th>$T_2$</th>
<th>$T_3$</th>
<th>$T_4$</th>
<th>$T_5$</th>
<th>$T_6$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>20</td>
<td>44.4</td>
<td>48.8</td>
<td>39.4</td>
<td>40.6</td>
<td>46.9</td>
<td>27</td>
</tr>
<tr>
<td>2</td>
<td>40</td>
<td>59.8</td>
<td>59.2</td>
<td>54.1</td>
<td>59.1</td>
<td>55.1</td>
<td>27.2</td>
</tr>
<tr>
<td>3</td>
<td>60</td>
<td>77.9</td>
<td>94.1</td>
<td>54.8</td>
<td>59.8</td>
<td>96.3</td>
<td>28.2</td>
</tr>
<tr>
<td>4</td>
<td>80</td>
<td>99.2</td>
<td>112</td>
<td>77.4</td>
<td>94.3</td>
<td>105</td>
<td>28.4</td>
</tr>
<tr>
<td>5</td>
<td>100</td>
<td>109.3</td>
<td>139.6</td>
<td>90.2</td>
<td>96.4</td>
<td>129.6</td>
<td>29</td>
</tr>
</tbody>
</table>

### V. EXPERIMENTAL CORRELATIONS

Empirical relations are used to validate the vertical orientation model. Following correlations are considered:

1. McAdam’s correlation:
   
   $$ Nu = \frac{0.59 \times Ra^{0.25}}{\varepsilon} $$

2. Churchill and Chu’s first correlation:
\[ Nu = \left[ 0.825 + \frac{0.387 \times R a^{1/6}}{1 + \left( \frac{0.492}{Pr} \right)^{9/16}} \right]^{2} \]

(3) Churchill and Chu’s Second correlation:

\[ Nu = 0.68 + \frac{0.67 \times R a^{1/4}}{1 + \left( \frac{0.492}{Pr} \right)^{9/16}} \]

(4) Churchill and Usagi’s correlation:

\[ Nu = \frac{0.67 \times R a^{1/4}}{1 + \left( \frac{0.492}{Pr} \right)^{9/16}} \]

VI. RESULT ANALYSIS

A. Variation of Nusselt Number with Different Power Input

Figure 5-10 shows variation of Nusselt No. with different power input for fin arrays. Value of Nu number from experimentation is close to that of from existing equations. Three sets are plotted for 10, 30, 50 mm height and length of 200mm. Also remaining three sets are for 100 mm length and 05mm, 15mm, 25mm height. We observe that as the power input increases the Nusselt number increases.

B. Variation of Heat Transfer Coefficient with Different Power Input for Fin arrays

Figure 11-12 shows variation of heat transfer coefficient with different power input for various fin arrays. As power input increases ‘h’ increases for all arrays. The convective heat transfer coefficient is more for a smaller fin array i.e. smaller height fin array with constant length for given power input. This is because h not only depends on
area but temperature difference between fin and surrounding air.

**Fig.11** L=200mm and H=10, 30,50mm

**Fig.12** L=100mm and H=05, 15,25mm

Variation of heat transfer coefficient with input power

C. Variation of Convective Heat Transfer Rate with Fin Height for Different Power Input

From the figure 13-14 it can be observed that for every power input and fin length combination, the convection heat transfer rate from the fin array increases with the increase in the fin height. With an increase in fin height, the total heat dissipation area also increases. Since the convection heat transfer rate directly related to the surface area in contact with air, increasing fin height increases the total heat dissipation.

**TABLE X**

<table>
<thead>
<tr>
<th>Sr.No</th>
<th>Qc (10005)</th>
<th>Qc (10015)</th>
<th>Qc (10025)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>17.096</td>
<td>17.1922</td>
<td>17.9272</td>
</tr>
<tr>
<td>2</td>
<td>34.512</td>
<td>35.2922</td>
<td>35.312</td>
</tr>
<tr>
<td>3</td>
<td>50.269</td>
<td>51.8712</td>
<td>54.722</td>
</tr>
<tr>
<td>4</td>
<td>64.5717</td>
<td>66.941</td>
<td>67.123</td>
</tr>
<tr>
<td>5</td>
<td>79.8438</td>
<td>83.554</td>
<td>85.9687</td>
</tr>
</tbody>
</table>

**TABLE XI**

<table>
<thead>
<tr>
<th>P (W)</th>
<th>Qc (20010)</th>
<th>Qc (20015)</th>
<th>Qc (20025)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>16.5288</td>
<td>17.9141</td>
<td>18.11255</td>
</tr>
<tr>
<td>2</td>
<td>34.203</td>
<td>35.8017</td>
<td>37.18277</td>
</tr>
<tr>
<td>3</td>
<td>52.9891</td>
<td>53.2447</td>
<td>55.9191</td>
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<tr>
<td>4</td>
<td>67.35508</td>
<td>69.8387</td>
<td>73.8493</td>
</tr>
<tr>
<td>5</td>
<td>84.9419</td>
<td>84.7548</td>
<td>92.7947</td>
</tr>
</tbody>
</table>

**VILFLOW VISUALAZATION**

Computational fluid dynamics CFD is the useful to visualize the flow. In this section the variation of temperature and velocity of the flow with different parameters of heat sink is displayed. Since there are many fin arrays and their combinations with different heat inputs, it is not possible to show variation of flow speed and temperature for all. Therefore, only one fin configuration is selected to represent every visualization figure.

**Fig.13** L=200mm and H=10, 30,50mm

**Fig.14** L=100mm and H=05, 15,25mm Variation of Convective Heat transfer with power input

**Fig.15** Fin Arrays Before Meshing
The boundary layer effects had been captured in the simulation as evident from the velocity gradients near the fin surfaces in the above image. Also, the channel width between the fins results in high velocity also can be observed in the above image.
The thermal boundary layers surrounding the fins can be observed in the above image. Also, the thermal gradient was observed only the zones near the fins and the remaining locations have no change in temperatures. This was expected as the heat was supplied from the base plate and to the fins only.

High Nusselt number at the bottom section of the Fins indicates the higher heat transfer rate at the bottom than the top surfaces. The air that approaches the fin bottom is at the ambient temperature of 25 C. Due to the higher temperature on Fin surface, the heat transfer takes place and the air temperature increases. As the air moves upwards due to the density difference, the chances of heat removal from Fin’ top surfaces reduces since the air temperature is already high. This had been observed in all configurations.

VIII. CONCLUSION

- Natural convective heat transfer depends on fin height.
- Convective heat transfer rate from fins increases with an increase in height of fin arrays.
- Experimental Nusselt Number is quite close to the value obtained from correlation
- With an increase in heat input and temperature difference, natural convective heat transfer also increases.
- With an increase in heat input, heat transfer by radiation mode also correspondingly increases.
- Convective heat transfer increases with aspect ratio for given power input.

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REFERENCES

[12] Moghtada Mobedi, Bengt Sunden, Natural convection heat transfer from a thermal heat source located in a


[14] Barak Yazicioğlu and Hafit Yüncü, a correlation for optimum fin spacing of vertically-based rectangular fin arrays subjected to natural convection heat transfer, J. of Thermal Science and Technology ISSN 1300-3615 29, 1, 99-105, 2009


