

Thermal Analysis of Vertical Plate Fin Heat Sink

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Abstract: Heat Sinks are widely used to remove the heat from the components which are generating heat during their functioning. Overheating causes malfunctioning of the components as well as it is responsible for reducing their life. Free convection is very common way of heat transfer from the heat sink considering power requirement, pressure drop and cost of the forced convection. This paper presents the thermal analysis of vertical plate fin heat sink by theoretical and experimental method at variable heat input. The results are obtained by taking experimental observations and are validated with already existing correlations suggested by various researchers in the literature.

Keywords: Heat Sink, Natural Convection, Fin Array

1 Introduction

Fins are used when present surface area is not sufficient for the removal of generated heat in the components with available temperature difference between surface and surrounding. In various electronic systems such as relays, mother boards of personal computers, audio systems, LED lamps etc. as shown in Fig. 1, heat is generated during their working. Thermal management of these components is very essential in order to avoid their failure as well as to improve their life span. One of the effective techniques of cooling the electronics components to equip heat sink on their surface to increase their effective surface area and thereby to increase the heat dissipation rate. Heat sinks are the array of number of fins arranged together in order to increase the surface area available for heat transfer. In majority of applications passive cooling is preferred considering its low cost and trouble free service. Though air has low specific heat, low conductivity but because of its free availability, low cost and ease of maintenance still it is preferable choice in most of the cases. While using the heat sink the shape of the fin, its orientation, material, size and fin spacing plays very important role in transferring the heat. Various researchers studied the air flow in natural convection over vertical surface of the fin plate and proposed various correlations.



Fig. 1 Heat Sinks

2 Literature Survey

Hong and Chung [1] carried out experimental and numerical analysis of vertical plate fin array in natural convection heat transfer. They found out optimum spacing for maximum heat transfer rate and further studied the effect of varying Prandtl number on thermal performance. Chang et al. [2] examined numerically the thermal performance of vertical fin array with and without dimples. They concluded that average value of Nusselt number (Nu) is the strong function of Rayleigh number (Ra). They tested four configurations of dimpled fin array and found out the best dimple arrangement for higher heat transfer rate. Awasarmol and Pise [3] experimentally studied rectangular fin array with and without perforation effect at different angles of orientation. They concluded that perforations on the fin surface increases heat transfer rate with advantage of reduction in weight. In their study they found out optimum perforation diameter. Naserian et al. [4] studied the effect of v type fin array on a vertical base experimentally and numerically. Results for plate fins are validated with Churchill and Chu and McAdam correlations. Haghghi et al. [5] carried out experimental investigation of vertical fin array and compared the results with combination of plate fins and pin fins arranged between two consecutive plate fins of heat sink. The results show that combination of plate and pin fin have lower thermal resistance and higher heat transfer compared to only plate fins heat sinks. Heat transfer enhancement of new-designed heat sinks is about 10-41.6% higher, compared to normal pin-fins. Effect of fin spacing on the thermal resistance was also studied.

3 Objective

In the present work the vertical symmetrical heat sink is manufactured and tested in natural convection mode with various heat inputs in order to study the effect of heat input on the thermal performance of the heat sink. The results obtained experimentally are validated with already existing correlations. Many of the researchers validated their

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experimental results by using correlations applicable for vertical plate where characteristic dimension is the vertical length of the plate. But in the present work validation is done with correlations suggested for vertical rectangular fin array where characteristic dimension is the fin spacing. Finally, thermal performance is measured in terms of convective heat transfer coefficient.

4 Experimental set up

The dimensions for the fin for experimental work are width 150 mm, height 200 mm and thickness 2 mm. Spacing between the fins is created by spacer block of width 50 mm, height 200 mm and thickness 16.5 mm. the thickness of the block serves as fin spacing. All fins and spacer blocks are made up of aluminium material. Spacer blocks and fin plates are having three central holes of 18 mm diameter along middle line to insert three stainless steel cartridge type heater rods. Two small holes are also provided on top and bottom sides of fin to insert tie rods to held complete assembly of fins and spacer blocks together as shown in figure. The overall size of the vertical fin array is 150 mm width, 200 mm height and 150 mm breadth. After assembling the fin array system (FAS), the effective length of the fin i.e. base to tip distance is measured as 50 mm.

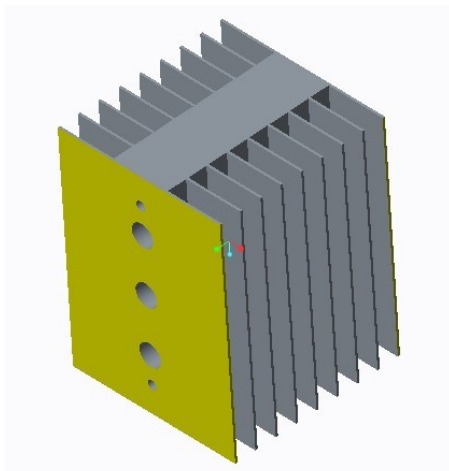


Fig. 2 Vertical Plate Fin Heat sink

The assembly is covered by bakelite plates from both the sides in order to reduce side heat loss. Bakelite plate size is equal to size of fin plate. Number of fins is taken as 9 and spacer blocks are 8. K type thermocouples with accuracy of 0.1°C are used to measure the temperatures at different locations of vertical heat sink. Total 6 no. of thermocouples are arranged on the spacer blocks and 6 no. of thermocouples are arranged on the fin plate surface. Two thermocouples are arranged on side bakelite covers and one thermocouple is in the enclosure to record ambient temperature. The arrangement of all thermocouples is as shown in Fig. 3.

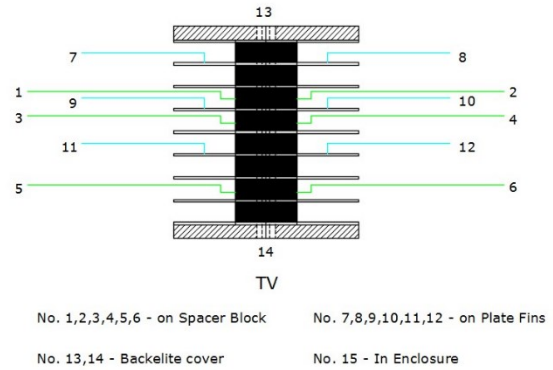


Fig. 3 Top view of the heat sink with Thermocouples positioning

In order to provide undisturbed surrounding, a rectangular box type enclosure from plywood is manufactured which is surrounding the arrangement. The total assembly of finned system is held vertical in hanging position with the help of metal wire supports so that natural convection current of hot air will setup around the fin surface.

The control panel is designed with arrangement of dimmerstat, display of heater wattage, temperature indicator with selector switch and mains switch. The power input connections to the heater is given through the control panel. All thermocouples are connected to control panel for displaying temperature on indicator.

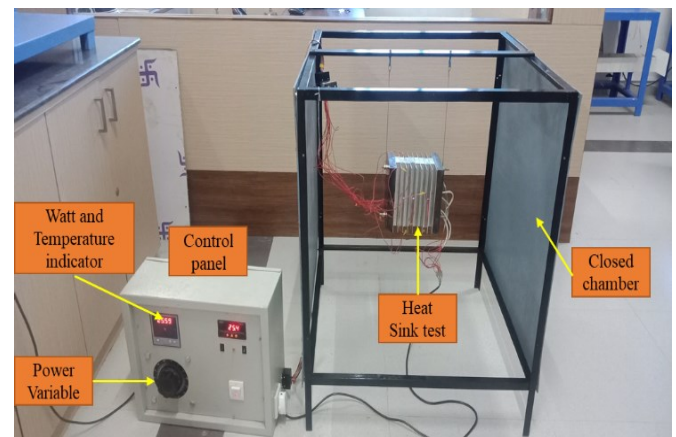


Fig. 4 Experimental Setup

5 Data reduction

- a) Temperature of finned surface is found out by taking average of all thermocouple reading mounted on base plate and fin surface.
- b) Temperature of surrounding is measured by thermocouple reading mounted in surrounding inside the enclosure.
- c) Area is calculated for the finned surface.
- d) The variable heat input is generator with the help of dimmer and heat supply is determined by wattage reading given by wattmeter.
- e) The various heat losses are calculated including side convective and radiation loss.
- f) By using Newton's law of cooling, the convective heat transfer coefficient is determined.

6 Validation of Experimental Setup and Procedure

6.1 Experimental Method:

1. Area calculation of system of fin array (FAS) for experimental analysis:

A_b = Area of base

A_f = Area of Fins

A = Total area considered for analysis = $A_b + A_f$

s = Fin spacing i.e. thickness of spacer block

H = Fin height i.e. vertical length of FAS

W = Width of FAS

B = Breadth of FAS

t = Thickness of fin

l = Fin length i.e. base to tip distance

$A_b = (8 \times s \times H) \times 2$

$$= (8 \times 0.0165 \times 0.2) \times 2 = 0.0528 \text{ m}^2$$

$A_f = (8 \times l \times H) \times 2 \times 2 + (9 \times t \times H) \times 2$

$$= (8 \times 0.05 \times 0.2) \times 2 \times 2 + (9 \times 0.002 \times 0.2) \times 2$$

$$= 0.3272 \text{ m}^2$$

$A = 0.0528 + 0.3272 = 0.38 \text{ m}^2$

2. Average surface temperature of FAS (T_s)

$$T_s = \frac{T_1 + T_2 + T_3 + T_4 + T_5 + T_6 + T_7 + T_8 + T_9 + T_{10} + T_{11} + T_{12}}{12}$$

$$T_s = 41.4 \text{ }^\circ\text{C} = 314.4 \text{ K}$$

3. Surrounding temperature (T_∞)

$$T_\infty = T_{15} = 28.3 \text{ }^\circ\text{C} = 301.3 \text{ K}$$

4. Radiation heat loss (Q_r)

$$Q_r = \sigma \cdot A \cdot \epsilon \cdot (T_s^4 - T_\infty^4)$$

Assuming emissivity of commercial aluminum surface (ϵ) = 0.09

$$Q_r = 5.67 \times 10^{-8} \times 0.38 \times 0.09 \times (314.4^4 - 301.3^4)$$

$$= 2.96 \text{ W}$$

5. Heater Input (Q_H)

Q_H = Wattmeter Reading = 25 W

6. Effective heat transfer rate for convection (Q)

$$Q = Q_H - Q_r$$

$$Q = 25 - 2.96 = 22.04 \text{ W}$$

7. Convective heat transfer coefficient (h)

Newton's law of cooling is given by mathematical expression:

$$Q = h \times A \times (T_s - T_\infty)$$

$$h = \frac{Q}{A \times (T_s - T_\infty)}$$

$$h = \frac{22.04}{0.38 \times (41.4 - 28.3)} = 4.42 \text{ W/m}^2\text{ }^\circ\text{C}$$

6.2 Theoretical Method

After calculating the experimental value of heat transfer coefficient, the value is compared with already existing correlations for vertical plate available in literature.

6.2.1 Correlations for vertical plate fin array

The correlations used for comparing the results are

- a) McAdams correlation

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

- b) Churchill and Chu's first relation (for laminar and turbulent flows)

$$Nu = \left[0.825 + \frac{0.387(Ra)^{1/6}}{\left[1 + \left(\frac{0.492}{Pr} \right)^{9/16} \right]^{8/27}} \right]^2 \text{ for } 10^{-1} < Ra < 10^{12} \text{ ----- [2]}$$

- c) Bar-Cohen Correlation

$$Nu = \frac{h \cdot s}{k} = \left[\frac{576}{\left(\frac{Ra_s \cdot s}{H} \right)^2} + \frac{2.873}{\left(\frac{Ra_s \cdot s}{H} \right)^{0.5}} \right]^{-0.5} \text{ ----- [3]}$$

- d) Elenbaas Correlation

$$Nu = \frac{h \cdot s}{k} = \frac{1}{24} \frac{Ra_s \cdot s}{H} \left[1 - \exp \left(\frac{-35}{\frac{Ra_s \cdot s}{H}} \right) \right]^3 \text{ ----- [4]}$$

6.2.2 Calculations

1. $\Delta T = T_s - T_\infty = 41.4 - 28.3 = 13.1 \text{ }^\circ\text{C}$

2. Mean film temperature (T_{mf})

$$T_{mf} = \frac{T_s + T_\infty}{2} = \frac{41.4 + 28.3}{2} = 34.85 \text{ }^\circ\text{C} = 307.85 \text{ K}$$

3. Coefficient of thermal expansion (β)

$$\beta = \frac{1}{T_{mf}} = \frac{1}{305.67} = 0.00324 \text{ K}^{-1}$$

4. Properties of fluid (air) at mean film temperature (Calculated from property table of air)

a. Kinematic viscosity (ν) = $1.65 \times 10^{-5} \text{ m}^2/\text{s}$

b. Prandtl Number (Pr) = 0.70

c. Thermal conductivity (k) = $0.0267 \text{ W/m}^\circ\text{C}$

5. Grashoff Number (Gr)

$$Gr = \frac{\beta \cdot g \cdot \Delta T \cdot L_c^3}{\nu^2}$$

$$Gr = \frac{0.00324 \times 9.81 \times 13.1 \times 0.2^3}{(1.65 \times 10^{-5})^2} = 1.23 \times 10^7$$

6. Rayleigh Number (Ra)

$$Ra = Gr \cdot Pr$$

$$Ra = 1.23 \times 10^7 \times 0.70 = 8.62 \times 10^6$$

7. Nusselt Number (Nu)

By using correlations given by equation 1,2,3 and 4 the Nusselt number values are obtained.

8. Convective heat transfer coefficient (h)

$$h = \frac{Nu \cdot k}{L_c}$$

Table 1 Values of h by various correlations for 25 W

Correlation	Mc Adam	Churchill and Chu's - first	Bar-Cohen	Elenbaas	Expt.
h ($\text{W/m}^2\text{ }^\circ\text{C}$)	4.27	3.99	4.21	4.23	4.42

Similarly, calculations are extended for 50, 75, 100 and 125 W and experimental and theoretical results are plotted on the graph in order to justify the agreement between them.

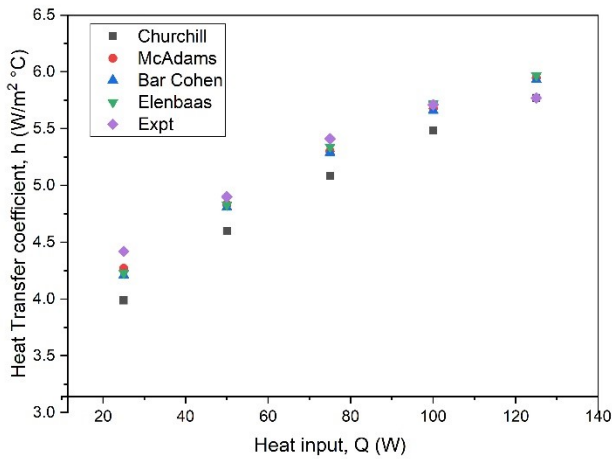


Fig. 5 Validation of the Experimental setup

The plot examinations indicate that the values of convective heat transfer coefficient (h) obtained from experimental observations are in good agreement with already existing correlations with relative deviation between 3-10 %. These results prove the validity of the experimental setup and experimental procedure.

7 Result of Plane Fins

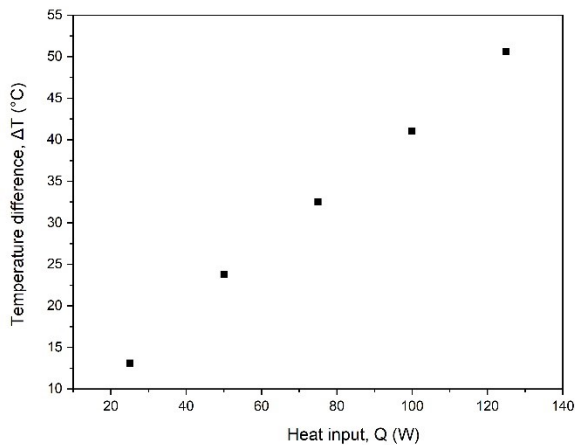


Fig. 6 Plot of temperature difference (TD) vs heat input (Q)

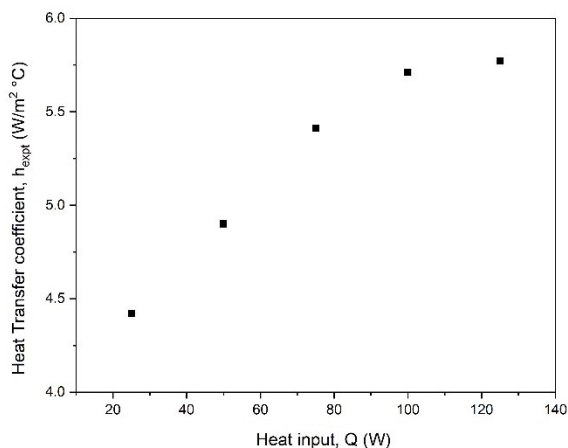


Fig. 7 Plot of heat transfer coefficient (h) vs heat input (Q)

It is clear from the Fig. 6 and Fig. 7 that heat input supplied at the base of FAS provides driving potential to create natural convection currents of air through the space available between two consecutive fins. With increase in heat input the temperature difference between fin surface and surrounding goes on increasing. The convection current becomes dominant with increase in heat input results into more degree of buoyancy force and higher convective heat transfer coefficient.

An uncertainty analysis was done to investigate the error in experimental setup which is a sum of individual uncertainty of experimental parameter used in the analysis. The total uncertainty found in the experimental setup and analysis is below 2%.

Conclusion

1. The experimental results of convective heat transfer coefficient are in good agreement (Maximum deviation less than 10%) with already existing correlations in the literature.
2. Thermal performance measured in terms of convective heat transfer coefficient for vertical heat sink is the function of heat input. As the heat input increases, the natural convection current augments as a result of density difference.

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