# EXPERIMENTAL ANALYSIS OF PARALLEL PLATE AND CROSSCUT PIN FIN HEAT SINKS FOR ELECTRONIC COOLING APPLICATIONS

by

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*Experimental investigation of parallel plate fin and the crosscut pin fin heat sinks where the heating element placed asymmetrically is performed. Theoretical calculations were done and compared with the experimental results. A comparative study was made based on their efficiencies, heat transfer coefficient, and the thermal performance. From the experimental results it was found that the average heat transfer coefficient of parallel plate fins is higher than that of crosscut pin fins with many perforations. However the performance efficiency of both the crosscut pin fins and parallel plate fins is similar. A hybrid approach was employed to significantly optimize the distance between the fan and heat sink for parallel plate and crosscut pin fins. Parallel plate heat sink with an average heat transfer coefficient of 46 W/m²K placed at an optimum fan distance of 40-60 mm is selected as the suitable choice for the micro-electronic cooling when the heating element is placed asymmetrically.*

Key words: *parallel plate, crosscut, asymmetric heating, heat sink, fan distance*

## Introduction

Advancements in semiconductor technology have led to the significant increase in power densities encountered in microelectronic equipment [1]. As the amount of heat that needs to be dissipated from electronic devices constantly increases, the thermal management becomes a more and more important element of electronic product design. Both the performance reliability and life expectancy of electronic equipment are inversely related to the component temperature of the equipment. Therefore, long life and reliable performance of a component may be

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achieved by effectively controlling the device operating temperature within the limits set by the device design engineers. With the increase in heat dissipation from the electronic devices and the reduction in overall form factors, it became an essential practice to optimize heat sink designs with least trade-offs in material and manufacturing costs.

A study of heat sink fin technologies has given information towards important design criteria for practical cooling of electronic components. Significant work has been carried out by various researchers in the thermal analysis of heat sink design. Ellison [2], Kraus *et al*. [3] have presented the fundamentals of heat transfer and hydrodynamics characteristics of heat sinks including the fin efficiency, forced convective correlations, applications in heat sinks, *etc*. Sasaki [4] optimized, with a criterion of fin to channel thickness ratio of unity, the dimensions of water cooled micro-channels at a given pressure. Azar *et al*. [5] reported a method of design optimization and presented contour plots showing the thermal performance of an air cooled narrow channel heat sink in terms of fin thickness and channel spacing parameters and employed Poiseuille's equation in relating the channel flow velocity to the pressure drop, and the optimization method was presented, assuming the pressure drop across the heat sink is known. An analytical method of optimizing forced convection heat sinks was proposed by Knight *et al.* [6, 7] for fully developed flow in closed finned channels. They presented normalized non-dimensional thermal resistances as a function of the number of channels, again for a fixed pressure drop. Wirtz *et al*. [8] investigated experimentally the effect of flow bypass on the performance of longitudinal fin heat sinks and devised a set of expressions for determining the optimum fin density for different fin geometry and flow conditions.

Keyes [9] analytically examined the fin and channel dimensions to provide optimum cooling under various forced convection cooling conditions. Bartilson [10] investigated, using both experimental and numerical techniques, air jet impingement cooling on a rectangular pin fin heat sink. Various shapes of longitudinal straight fin heat sinks were experimentally examined, and the thermal performance measurements were compared with existing correlations [11]. Seri Lee [12] observed that the actual convection flow velocity through fins is usually unknown to designers, yet, is one of the parameters that greatly affect the overall thermal performance of the heat sink and developed a simple method of determining the fin flow velocity and the development of the overall model of the heat sink. Different types of heat sink are examined, and their relative performances are presented. The analytical simulation model is validated by comparing the results with the experimental data, and sample cases are presented with discussions on the parametric behavior and optimization of bi-directional heat sinks with the heaters placed symmetrically.

Christopher *et al*. [13] studied on elliptical pin fin heat sink. Comparative thermal tests have been carried out using aluminum heat sinks with extruded fins, crosscut rectangular fins in low air flow environments. Besides the thermal measurements, the effect of airflow bypass characteristics in opened duct configuration was investigated. The testing described in the study incorporates several possible performance factors into two terms, flow bypass and overall thermal resistance. These simplified terms represent a combination of several factors, such as material conductivity, lateral fin conduction, boundary layer formation, effective surface area, and pressure drop. The crosscut heat sink offers ease of production assembly where misalignment of heat sink with respect to the direction of the air flow will result in failure.

Vollaro *et al*. [14] and Culham *et al*. [15] studied the optimization of the parallel plate heat sink and attempted to define general rules for optimizing it. Park *et al*. [16] proposed the progressive quadratic response surface model to obtain the optimal values of design variables for a plate fin type heat sink Park *et al*. [17, 18] performed an investigation of numerical shape optimization for high performance of a heat sink with pin fins. Yu *et al*. [19] developed a plate pin fin heat sink and compared its performance with a plate fin heat sink. Chiang *et al*. [20] developed the procedure of response surface methodology for finding the optimal values of designing parameters of a pin fin type heat sink under constraints of mass and space limitation to achieve the high cooling efficiency.

From the above literatures, it can be concluded that significant work needs to be carried out in optimizing the heat sink design. Plate fin and pin fin heat sinks are commonly used heat sinks for electronic cooling applications. This makes the selection of heat sink a difficult task for a particular application. In the present work, crosscut pin fin heat sink is developed and its performance is compared with the parallel plate heat sink in micro electronics cooling. In most of the electronic components the heat duty involved is high and also the heater is placed symmetrically about the axis, thereby leading to space constraints. An effort is made in the experiment by placing the heating element asymmetrically for the study of heat transfer characteristics and the thermal performance. The fan distance is also varied in the experimental work to find the optimum distance for maximum efficiency for both parallel plate and crosscut pin fin heat sinks.

## Construction

The dimensions of parallel plate fine heat sink component are 57 mm length and 49 mm breadth, rectangle shaped structure with a height of 15 mm. The base of the fin is of 5 mm thick. The extruded surface (fin) is of 10 mm height. The thickness of each fin is 2 mm. In the parallel plate structure the fin extends as a single structure along the length of 57 mm. There are 14 numbers of fins in this structure to dissipate the heat produced. The graphical representation of the parallel plate heat sink used in this experimental study is shown in fig. 1.

The crosscut pin fins provide a weight reduction while maintaining surface area comparable to that of parallel plate fin heat sink. The dimensions of this component are the same as parallel plate fin heat sink. In crosscut structure the fin extends only for 2 mm along the length of the structure. In this model there are 224 numbers of fins used to dissipate the heat. The graphical representation of the crosscut pin fin heat sink is shown in fig. 2. Both the heat sink fins are made of aluminum.

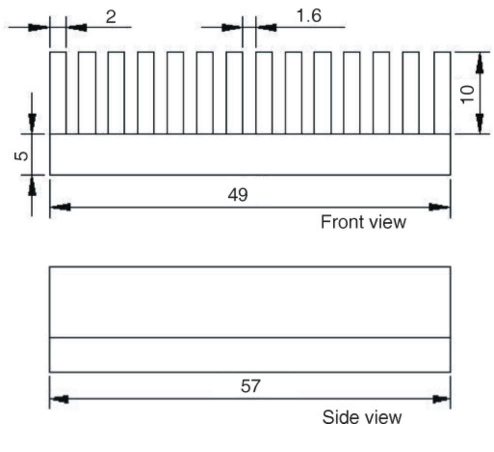


**Figure 1. Front and side view of parallel plate fin**

The casing plate is made up of copper material, and has the following dimensions: 57 mm length, 49 mm breadth, and 10 mm thickness. On the front right side of the plate, on the 49 mm side, there is a 22 5 mm rectangular slot, which is offset at a distance of 2.5 mm from the top and the bottom of this plate and at a distance of 3 mm from the front right side of the casing. This slot is provided for inserting the heater. The base plate is placed on an insulating material where half of the base plate *i. e*. 5 mm gets seated into the insulating material. The insulating ma-

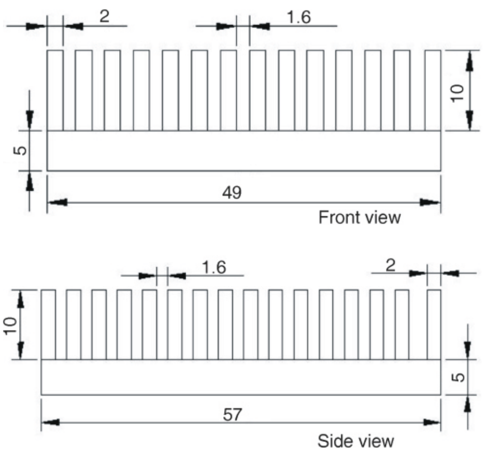


**Figure 2. Front and side view of crosscut pin fin**

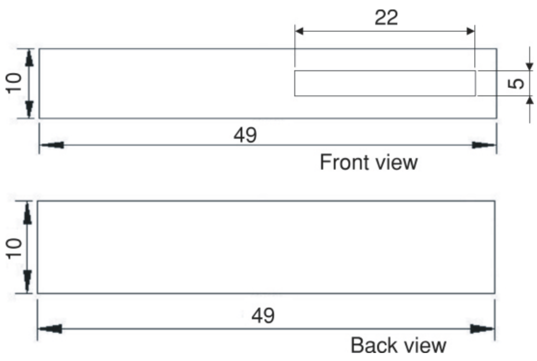


**Figure 3. Front and back view of casing**

terial provides also exactly to seat half of the base plate into it. The graphical representation of the casing plate is shown in fig. 3.

The J- type thermocouples are fixed at nineteen different nodes on the surface of the heat sinks and the copper casing where the temperature is to be measured. The thermal grease is applied on the top surface of the copper casing. The fin arrangement is placed on this grease surface. The thermal grease serves as a link between the copper casing and the fin. This helps in better conduction of heat from the casing to the fin. This also removes the air gap between the two surfaces which acts as an insulator.

## III Testing scheme

The experiment was done by varying the fan distance ranging from 0 to 60 mm and the heat input of 20 to 60 W at a room temperature of 297 K. The fan arrangement is such that it can slide on the fan stand pole and can be kept at a specified distance. The fan distance from the heat sink was varied from 0 to 60 mm, respectively. The power to the fan is given separately and this is maintained constant. The heat is supplied with the help of heater. The data acquisition unit serves as an inter link between the hardware and software where the nineteen thermocouples leads are fixed at the respective poles. The resultant input is the display of each thermocouple temperature in the DARWIN software interface.

### *Experimental setup*

The heater is placed in the slot provided in the copper casing. The copper casing is placed in the slot on the insulating base plate. The base plate is provided to minimize the heat loss by radiation. The fin is arranged on top of the copper casing. To avoid the heat transfer loss between the copper casing and the fin arrangement thermal grease is used. This serves as an inter-link between the two surfaces. The power supply to the heater is given using an auto-transformer. The power supply to the fan is given separately; using the DC regulated supply. The power supply to the fan is maintained constant throughout the experiment. The fan distance is varied by sliding the fan and fixing it at different position *i. e.* 20, 40, and 60 mm. The fin arrangement carries the thermocouple arrangement at nineteen different nodes. The free ends of the thermocouple leads are fitted to the data acquisition hardware unit which is linked to the computer by a software package. The data acquisition unit software helps to view the temperature varying with respect to time. After a period of 15-25 minutes, it is observed that the temperature of all the nineteen nodes corresponding to the nineteen thermocouples moving parallel to each other. This indicates that the experiment which is conducted for a specific fan distance has reached the steady-state condition. The experiment is repeated by varying the fan distance and the power supply to the heater. Heat dissipation in the heat sinks is primarily based on conduction and convection. Hence the radiation effects are neglected in the present investigation.

### *Flow diagram*

The flow diagram shown in fig. 4 represents the various stages of the experiment from the initial power supply to the final display (output) in the computer (monitor). The power supply is given to the constant voltage regulator and other supply is given to the computer. This power from the constant voltage regulator is distributed to two instruments namely; the autotransformer and the DC regulated power supply. The auto transformer supplies the power to the heater. The DC regulated power supply supplies the power to the fan. The thermocouples which are fixed on the fin casing arrangement are connected to the data acquisition unit in their corresponding slots. The data acquisition unit has two wires. Among them one is connected to the power supply for its working and the other is connected to the computer for receiving and sending the signals.

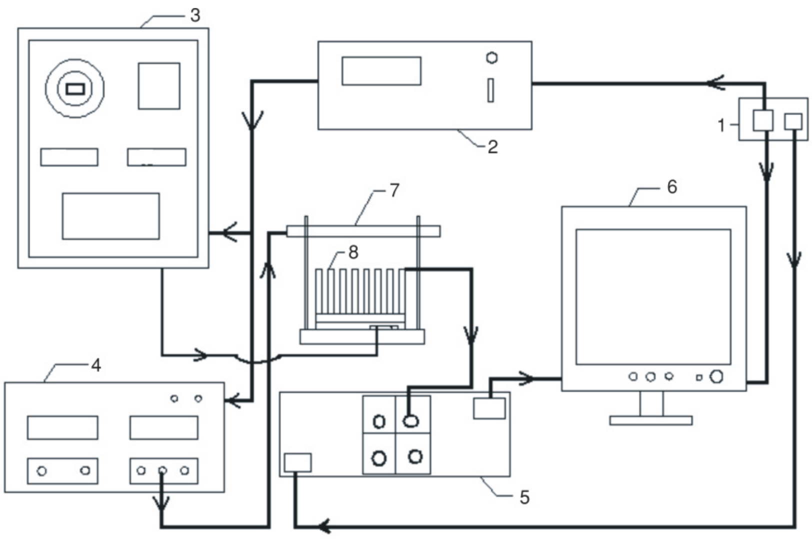


**Figure 4. Flow diagram for the experimental setup**
*(1) – power supply, (2) – constant voltage regulator, (3) – auto transformer, (4) – DC regulated power supply, (5) – data asquisition unit, (6) – computer, (7) – fan, (8) – heat sink*

## Results and discussion

### *Theoretical predictions*

To calculate the heat transfer coefficient ($h$), the operating condition is assumed to be in steady-state. Because of symmetry of fins the flow area between the two adjacent fins are considered. The theoretical heat transfer coefficient for various heat inputs is calculated using the correlations from Heat transfer data book [21] and the results are tabulated in tab. 1.

**Table 1. Theoretical heat transfer coefficient for various heat inputs**

| Heat input [W] | Theoretical heat transfer coefficient [$Wm^{-2}K^{-1}$] | |
|---|---|---|
| | Parallel plate fins | Crosscut fins |
| 20 | 28.13 | 31.82 |
| 30 | 37.69 | 42.63 |
| 40 | 46.66 | 52.78 |
| 50 | 55.23 | 62.48 |
| 60 | 63.02 | 71.79 |
| Average | 46.15 | 52.3 |

### *Simulation modeling*

A simulation model was created using ANSYS for understanding the manner in which the heat is distributed in the heat sinks and also to obtain the temperatures at various nodes which were useful for calculating other parameters.

The inputs given to the ANSYS are the thermal conductivities of aluminum and copper, the room temperature as 297 K, the heat flux at the insulating areas as zero, and the heat generation rate by the heater for different heat inputs.

### *Experimental heat transfer coefficient*

The temperature at 19 nodes of the experimental setup is acquired using the data acquisition unit. The ANSYS software analysis is performed for various heat inputs by varying the heat transfer coefficient ($h$) values from (45-75 W/m²K) in multiples of "5".The temperatures at the nineteen nodes are noted down using the select-node option present in the ANSYS software, so that these nodes are at the same location as that of thermocouples fixed in the surface of the experimental components. To minimize the error due to the minor deviations in the readings obtained, the standard deviation (SD) is calculated using the formula , SD = $e^2/(n-1)$ where $e$ is the difference between experimental values and ANSYS values at corresponding nodes, n is the number of nodes.

**Table 2. Experimental heat transfer coefficient for parallel plate fin heat sink**

| Heat input [W] | Experimental heat transfer coefficient [$Wm^{-2}K^{-1}$] | | | |
|---|---|---|---|---|
| | Fan distance 0 mm | Fan distance 20 mm | Fan distance 40 mm | Fan distance 60 mm |
| 20 | 52 | 61 | 45 | 44 |
| 30 | 75 | 56 | 53 | 42 |
| 40 | 65 | 64 | 51 | 48 |
| 50 | 57 | 61 | 49 | 49 |
| 60 | 75 | 56 | 49 | 47 |
| Average | 64.8 | 59.6 | 49.4 | 46 |

The calculation is repeated for various ANSYS $h$ values but with the same experimental values. The heat transfer coefficient is calculated separately at all the points in which the measurements are taken. The average of all the standard deviation is taken, which is done for a specific experimental values with different ANSYS $h$ values. It is plotted on the y-axis and the various ANSYS heat transfer coefficient values are plotted on the x-axis. A curve is obtained and the x-value where the error is minimum is taken as the experimental heat transfer coefficient. The experimental heat transfer coefficient of for various heat inputs and fan distances found using this method for parallel plate fin and crosscut pin fin heat sink are tabulated in tabs. 2 and 3, respectively.

For the results, it can be found that the average experimental heat transfer coefficient of crosscut pin fin heat sink is lesser than the parallel plate heat sink at all the fan distances. Despite having a much lesser weight than the parallel plate heat sinks, the use of crosscut pin fin heat sinks does not improve the heat transfer performance if the heating element is placed asymmetrically.

**Table 3. Experimental heat transfer coefficient for crosscut pin fin heat sink**

| Heat input [W] | Experimental heat transfer coefficient [$Wm^{-2}K^{-1}$] | | | |
|---|---|---|---|---|
| | Fan distance 0 mm | Fan distance 20 mm | Fan distance 40 mm | Fan distance 60 mm |
| 20 | 75.5 | 69.5 | 69 | 69 |
| 30 | 56 | 46 | 34 | 31 |
| 40 | 53.5 | 50 | 42.5 | 37 |
| 50 | 58 | 53 | 45 | 34 |
| 60 | 59 | 54 | 46 | 36 |
| Average | 60.4 | 54.5 | 47.3 | 41.4 |

### *The efficiency of heatsinks*

The efficiency of the parallel plate and crosscut pin fin heat sinks for various heat inputs and fan distance are tabulated in tabs. 4 and 5, respectively. The overall efficiency of the parallel plate fin and crosscut pin fin heat sinks are 12.77% and 12.8%, respectively. The efficiency is calculated using the expression

$$\eta = Q_{\text{actual}} / Q_{\text{theoretical}} \quad (1)$$

**Table 4. Efficiency of the parallel plate fin heat sink for various heat input and fan distance**

| Heat input [W] | Efficiency of the parallel plate fin heat sink [%] | | | |
|---|---|---|---|---|
| | Fan distance 0 mm | Fan distance 20 mm | Fan distance 40 mm | Fan distance 60 mm |
| 20 | 10.53 | 10.29 | 11.83 | 13.62 |
| 30 | 12.03 | 11.16 | 12.72 | 16.06 |
| 40 | 11.58 | 11.96 | 12.69 | 16.26 |
| 50 | 10.59 | 11.54 | 12.47 | 16.39 |
| 60 | 12.69 | 11.87 | 12.65 | 16.51 |

**Table 5. Efficiency of the crosscut pin fin heat sink for various heat input and fan distance**

| Heat input [W] | Efficiency of the parallel plate fin heat sink [%] | | | |
|---|---|---|---|---|
| | Fan distance 0 mm | Fan distance 20 mm | Fan distance 40 mm | Fan distance 60 mm |
| 20 | 9.15 | 10.24 | 11.61 | 15.03 |
| 30 | 11.13 | 10.7 | 12.36 | 16.05 |
| 40 | 11.35 | 11.65 | 12.94 | 15.42 |
| 50 | 12.05 | 11.99 | 13.29 | 16.84 |
| 60 | 12.23 | 11.85 | 13.09 | 17.1 |

As the heat input and fan distance increases the performance efficiency of both the heat sink increases. The efficiency is found to be maximum when the fan is placed at a distance of 60 mm from the heat sink and when the heat input is 60 W for both type of heat sinks investigated. Hence considering the average heat transfer coefficient and the performance efficiency of both the heat sinks, selection of parallel plate heat sink with an optimum distance of 40 to 60 mm is the suitable choice for this particular application of asymmetric heating.

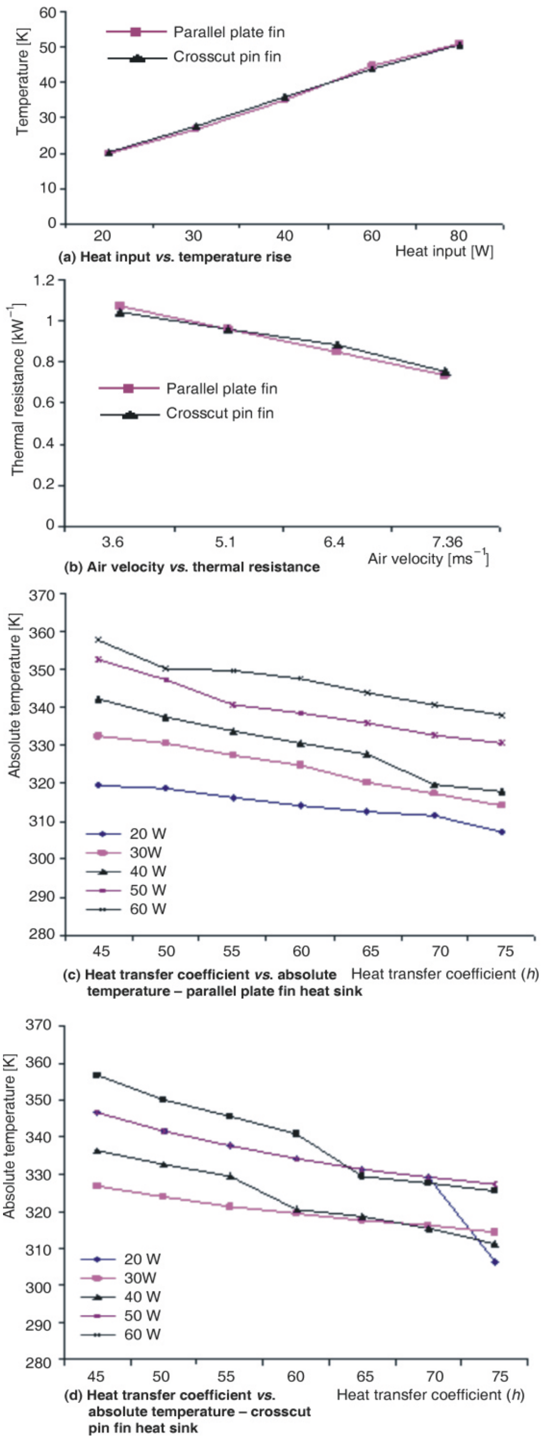


**Figure 5**

*Thermal performance graph*

Thermal performance graphs, fig. 5 (a-d), provide the information about the thermal performance of the heat sink with respect to the heat supplied, temperature rise, air velocity, and thermal resistance.

For both types of heat sinks the temperature rise increases as the heat input increases in a linear fashion. Lesser the fan distance from the fin arrangement more air flow rate is present and higher is the velocity of air and lesser the thermal resistance. As the fan distance increases the air flow rate decreases and the air velocity decreases with respect to the heat sink position which causes the increase in thermal resistance. At a constant heat input the absolute temperature of the heat sink decreases for increase in the heat transfer coefficient.

*Characteristic curve*

The characteristic curves fig. 6(a) and (b) helps to acquire information based on three parameters, *i. e.* fan distance, base temperature, and heat supplied. If any of the two parameters are only available the third parameter can found effectively.

**Conclusions**

The performance of parallel plate and crosscut pin fin heat sinks are investigated in the present work when the heating element is placed asymmetrically. The heat transfer coefficient for various heat inputs and for different fan distance for both the heat sinks is calculated. The performance efficiency of both types of fins remains similar. Both the heat sinks show high performance at higher velocities. A hybrid approach was employed to significantly optimize the distance between the fan and heat sink for parallel plate and crosscut pin fins, and based on the experimental results it was found that 40-60 mm fan distance is the optimum fan distance for getting maximum efficiency. From the experimental results it is found that the average heat transfer coefficient of parallel plate fins is higher than that of the crosscut pin fin heat sinks at all fan distance. Hence from the experimental results, it is concluded that the use of parallel plate heat sink at an optimum fan distance of 40-60 mm is the suitable choice to achieve good heat dissipation and maximum fin efficiency for an asymmetrically placed heating element. Finally the characteristic curve is drawn which is an effective tool to obtain the base temperature of the heat sink corresponding to the fan distance and heat input. Further study is needed to quantify the effect of heat-flux, the air flow by-pass factor and the overall thermal resistance for improving the efficiency of the heat sinks.

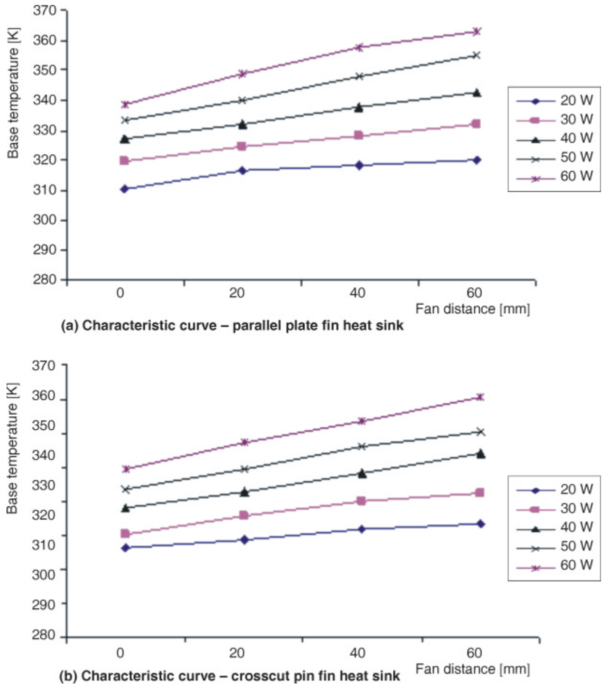


**Figure 6**

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