

# How Air Velocity Affects

## The Thermal Performance of Heat Sinks: A Comparison of Straight Fin, Pin Fin and maxiFLOW™ Architectures

### Introduction

A device's temperature affects its operational performance and lifetime. To achieve a desired device temperature, the heat dissipated by the device must be transferred along some path to the environment [1]. The most common method for transferring this heat is by finned metal devices, otherwise known as heat sinks.

Resistance to heat transfer is called thermal resistance. The thermal resistance of a heat sink decreases with more heat transfer area. However, because device and equipment sizes are decreasing, heat sink sizes are also growing smaller. On the other hand, device heat dissipation is increasing. Therefore, designing a heat transfer path in a limited space that minimizes thermal resistance is critical to the effective design of electronic equipment.

This article discusses the effects of air flow velocity on the experimentally determined thermal resistance of different heat sink designs. To be able to compare these designs, we need to first review basic heat transfer theory as applied to heat sinks. Previously published work is discussed, along with

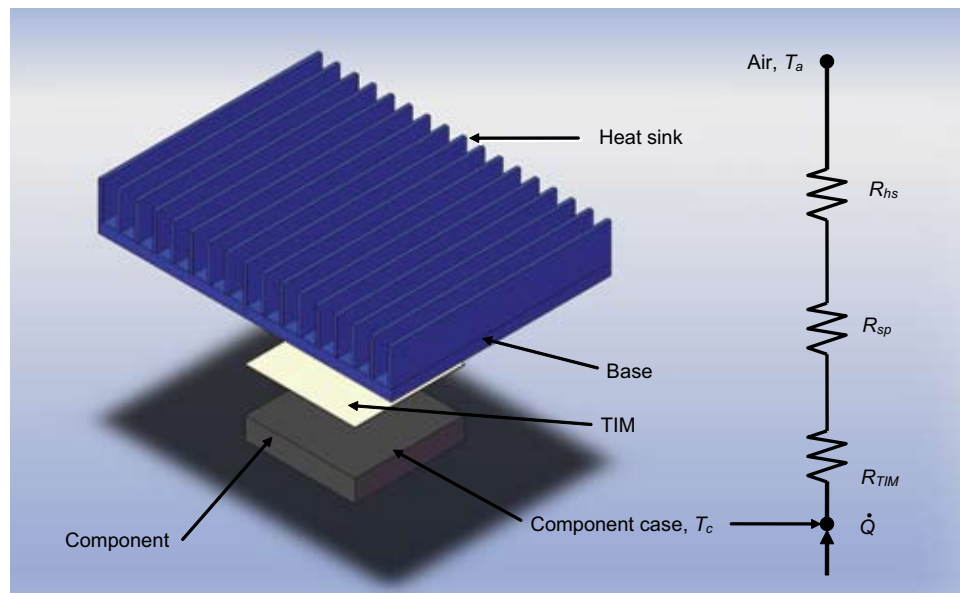


Figure 1. Exploded View of a Straight Fin Heat Sink with Corresponding Thermal Resistance Diagram.

heat sink selection criteria.

The heat transfer rate of a heat sink,  $\dot{Q}$ , depends on the difference between the component case temperature,  $T_c$ , and the air temperature,  $T_a$ , along with the total thermal resistance,  $R_t$ . This relationship is shown in Equation 1. For a basic heat sink design, as shown in Figure 1, the total thermal resistance depends on the sum of the heat sink resistance,  $R_{hs}$ , the spreading resistance in the heat sink base,  $R_{sp}$ , and the thermal interface resistance from the component to the heat sink base,

$\dot{Q} = (T_{hot} - T_{cold})/R_t = (T_c - T_a)/R_t$ , as shown in Equation 2.

$$\dot{Q} = (T_{hot} - T_{cold})/R_t = (T_c - T_a)/R_t \quad (1)$$

where

$$R_t = R_{TIM} + R_{sp} + R_{hs} \quad (2)$$

Therefore, to compare different heat sink designs, the thermal interface resistance,  $R_{TIM}$ , and the spreading resistance,  $R_{sp}$ , was similar among the heat sinks tested.

For this study, the same thermal interface material (TIM) was used with all heat sinks. This minimized the difference in the thermal interface resistance,  $R_{TIM}$ , between heat sink tests. As is normal, the spreading resistance of a heat sink's base,  $R_{sp}$ , increased with decreasing base thickness and conductivity. It also increased with an increasing difference in the heat sink base area and the heat dissipation area [2].

Lasance and Eggink have provided a method to rank a heat sink for certain applications [3]. This measurement is based on extracting the average heat transfer coefficients from time-dependant temperature curves as a function of velocity and bypass. The measured effective heat transfer coefficient is then scaled by mass, volume, weight or height. This provides several performance metrics to give designers a novel way to rank heat sinks in conditions that resemble an application. Heat sinks with heights between 5 and 20 mm were tested [3]. In addition to varying heights, the heat sink base thickness varied from 1.2 to 3 mm. The base width and length varied from 42.2 to 49.8 mm. The ratio between the width and length of the heat sink also varied, i.e. both square and rectangular base shapes were used. ATS maxiFLOW™ and pin fin heat sinks were tested, with the maxiFLOW™ series showing the lowest thermal resistance [3].

Forghan, et.al. also discuss the thermal resistance of various heat sink designs [4]. They conducted their tests based

on the steady state temperature of the heat sinks, unlike Lasance and Eggink, whose test are based on transient data. The maxiFLOW™ heat sinks were found to have the lowest thermal resistance, especially for air flow velocity below 2 m/s [4].

The heat sinks tests in this article were conducted in a wind tunnel and the data points were taken at steady state values. The heat sinks were selected based on the following heat sink selection criteria:

1. The same thermal interface material was used for all the heat sink

- tests.
2. The base thickness, width, and length were the same for all the heat sinks tested.
3. The same size heat source area was used for the different heat sink tests.
4. The heat sinks had the same fin height.

### About the Heat Sinks Tested

Heat sinks with 42 x 42 mm and with 27 x 27 mm base nominal dimensions were selected, as they are common sizes used in electronics cooling. Fin types included straight fin, pin fin, elliptical fin and ATS' maxiFLOW, as

Table 1. Types of Heat Sinks Tested, With Their Dimensions and Mass.

#	Heat Sink Type	L [mm]	w [mm]	H [mm]	Part No.	Mass, m (g)
1	Cylindrical Pin Fin (CPF)	42	42	24	CPF4224	32
2	Elliptical Pin Fin (EPF)	42.5	42.5	33	EPF4233	36
3	Cylindrical Pin Fin (CPF)	25	25	25	CPF2525	12
4	maxiFLOW™	42.5	42.5	17.5	MF4217	28
5	maxiFLOW™	25	25	17.5	MF2517	12
6	Elliptical Pin Fin (EPF)	25	25	33	EPF2533	12
7	Straight Fin (SF)	42.5	42.5	17.5	SF4217	18
8	Straight Fin (SF)	27	27	17.5	SF2517	10
9	Elliptical Pin Fin (EPF)	42	42	17	EPF4217	24
10	Elliptical Pin Fin (EPF)	25	25	17	EPF2517	18

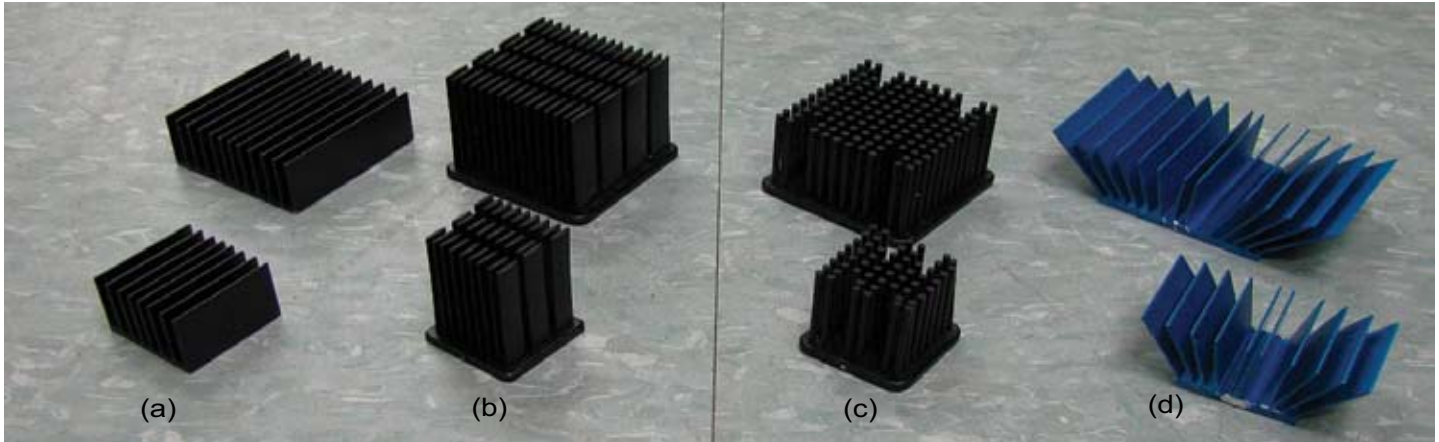


Figure 2. Various Heat Sinks Types Tested: Straight Fin (a), Elliptical Pin Fin (b), Cylindrical Pin Fin (c), and ATS' maxiFLOW™ (d).

shown in Table 1 along with dimensions, part numbers and mass. Figure 2 shows some of the selected heat sinks.

For testing purposes, a thermocouple hole was drilled into the center of the base of each heat sink, on the side perpendicular to the air flow direction in which each was tested. The position of a thermocouple hole can be seen in Figure 3.

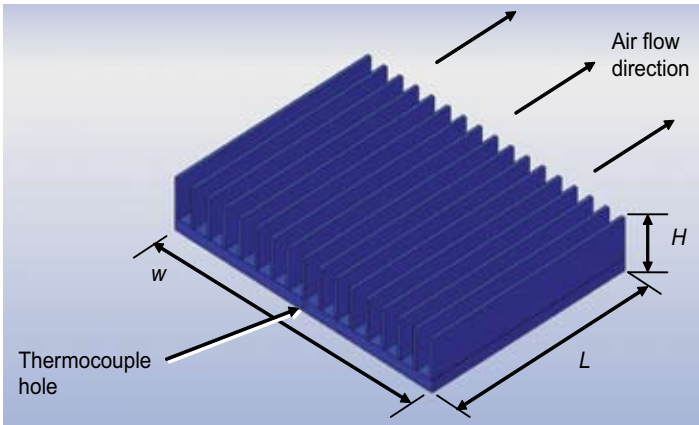


Figure 3. Illustration of a Straight Fin Heat Sink.

### Experimental Set-Up of the Heat Sinks in a Wind Tunnel

The natural and forced convection thermal resistances of each heat sink were determined in an ATS CWT-106™ open loop wind tunnel managed by a WTC-100 wind tunnel controller (Figures 4, 5, 6). Heater pads of 25 x 25 mm were placed in the center of the heat sinks to provide the heat sources. The heat sinks were mounted in the wind tunnel test section. Air flow and temperature were measured 15 cm

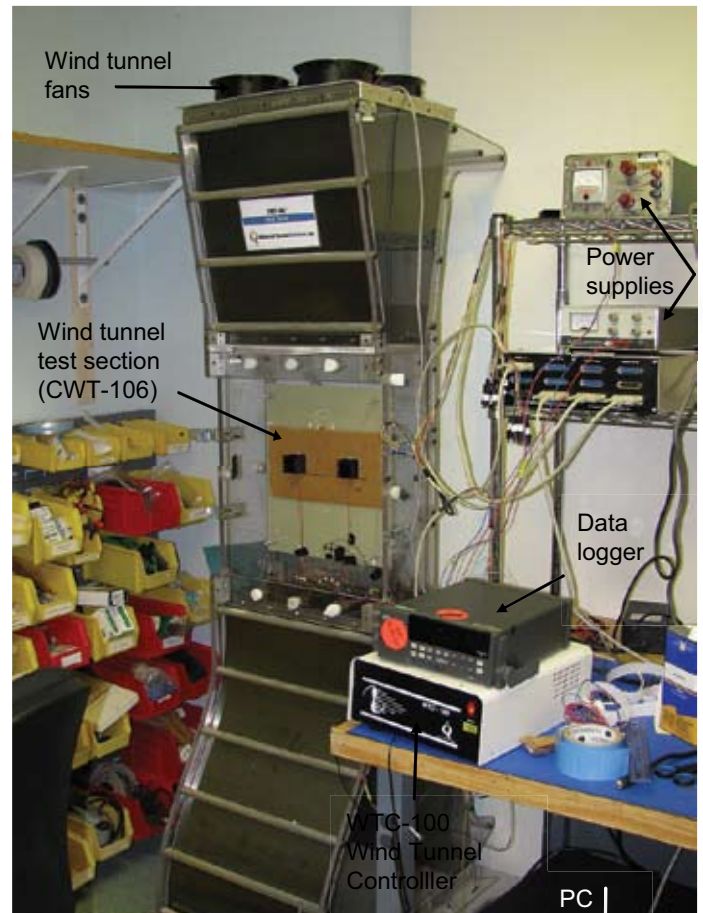


Figure 4. The ATS CWT-106 Open Loop Wind Tunnel and WTC-100 Wind Tunnel Controller.

upstream of the heat sink. Thermal paste was applied before the thermocouple was inserted into the hole.



### Test Procedure

After being set-up for testing as described in Section 2, heat dissipated by the heater pad was transferred to the air by natural convection. The heat sink base temperature and inlet air temperature were monitored until steady temperature state occurred. At this time, the heat sink base and inlet air temperatures were recorded, along with the voltage supplied to each heater pad.

For forced convection, steady state values were recorded at a range of air flow velocities. Power input for the 42 mm heat sinks was 12 W, while for the 27 mm heat sinks the power ranged from 6 to 9 W.

### Processed Results

The heat sinks were compared based on their thermal resistances. The heat transfer rate from heat sink base to air is given by Equation 3 (below). From here, the heat sink's thermal resistance is calculated using Equation 4. This assumes that the heat transferred by the sink to the air is equal to the power input to the heat source. After applying Equation 4 to the experimental data, the results are plotted in Figures 7 and 8.

$$\dot{Q} = (T_b - T_a)/R \quad (3)$$

$$R = (T_b - T_a)/P \quad (4)$$

Figure 7 shows that heat sink thermal resistance decreases for natural convection. The maxiFLOW™ heat sinks had the lowest thermal resistance for natural convection of the sinks tested. The 33 mm elliptical pin fin heat sinks

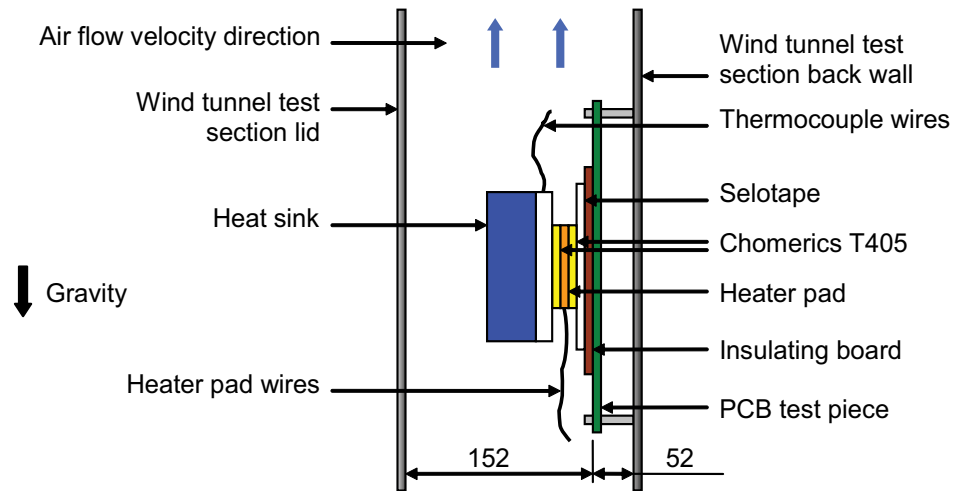


Figure 5. Illustration of the Experiment Set-up Inside the Wind Tunnel Test Section.

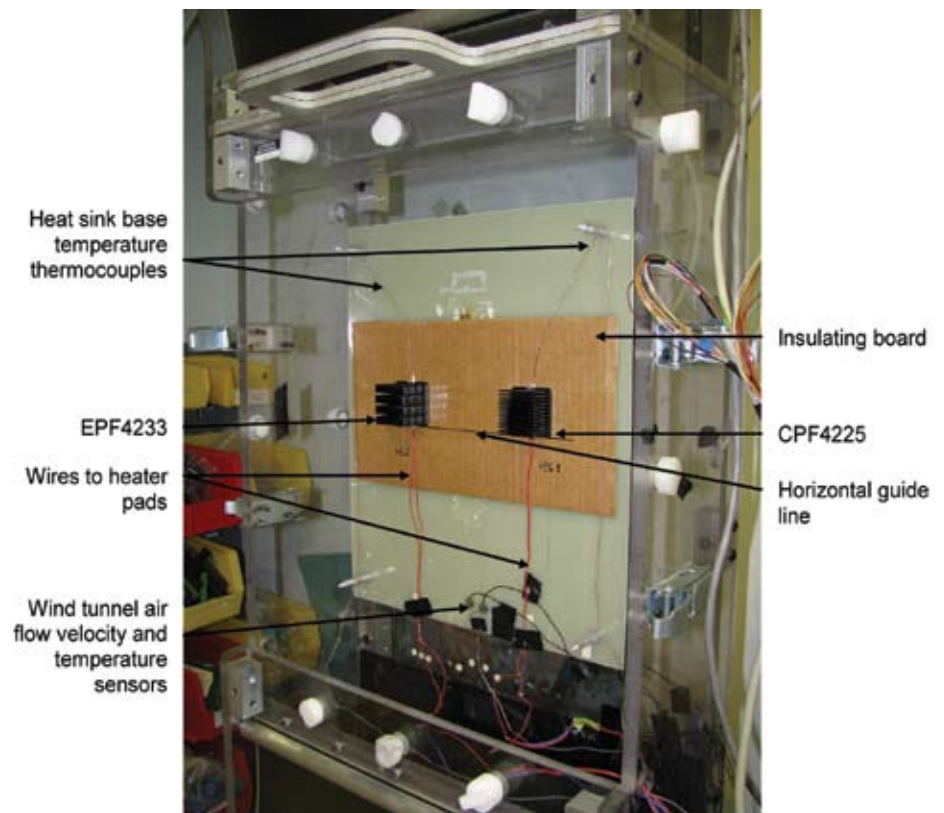


Figure 6. The Wind Tunnel Test Section. Heat Sinks EPF4233 and CPF4225 are Mounted Side by Side.

had a higher thermal resistance than the maxiFLOW heat sink, and the 17 mm elliptical pin fins had the highest thermal resistance. The 25 mm cylin-

drical pin fin heat sinks had about the same thermal resistance as the straight fin heat sinks. If it had been possible to cut the cylindrical pin fin heat sinks to

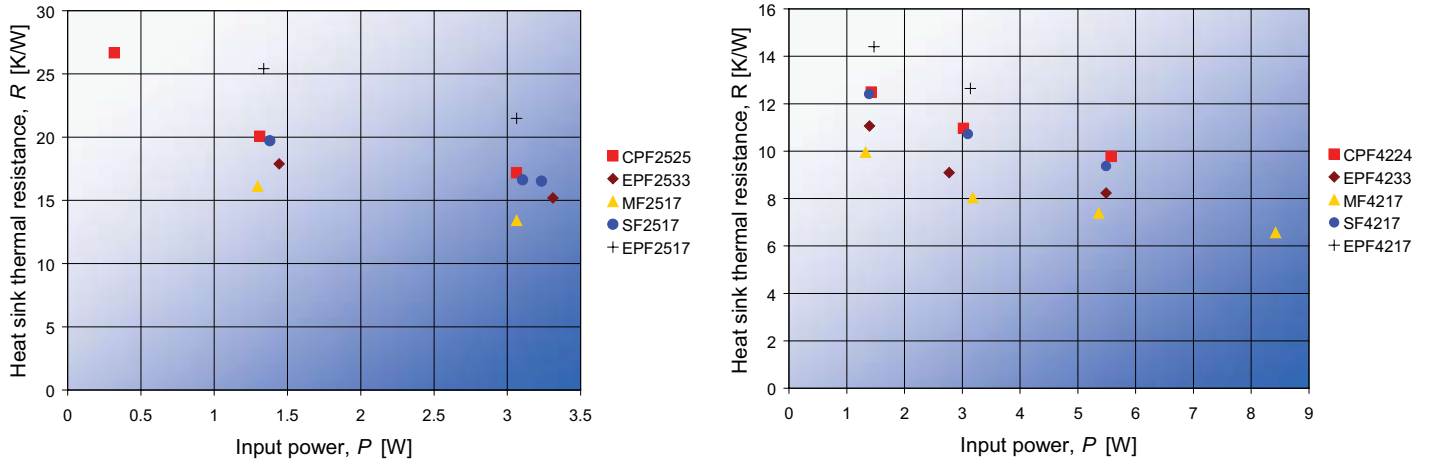


Figure 7. Thermal Resistance of 27 mm (a) and 42 mm (b) Heat Sinks for Natural Convection, Plotted Against Input Power.

a height of 17 mm, they would have had a higher thermal resistance than the straight fin heat sinks.

For forced convection, Figure 8 shows that the thermal resistance of all the heat sinks tested decreased with increased air flow rate. A 33 mm tall EPF had the lowest thermal resistance.

However, once they were cut to a height of 17 mm, they had the highest thermal resistance. This was similar to the natural convection tests. After the 33 mm EPF, the maxi-FLOW heat sinks had the best thermal resistance. The 42 mm straight fin and cylindrical pin fin heat sinks had similar thermal resistances.



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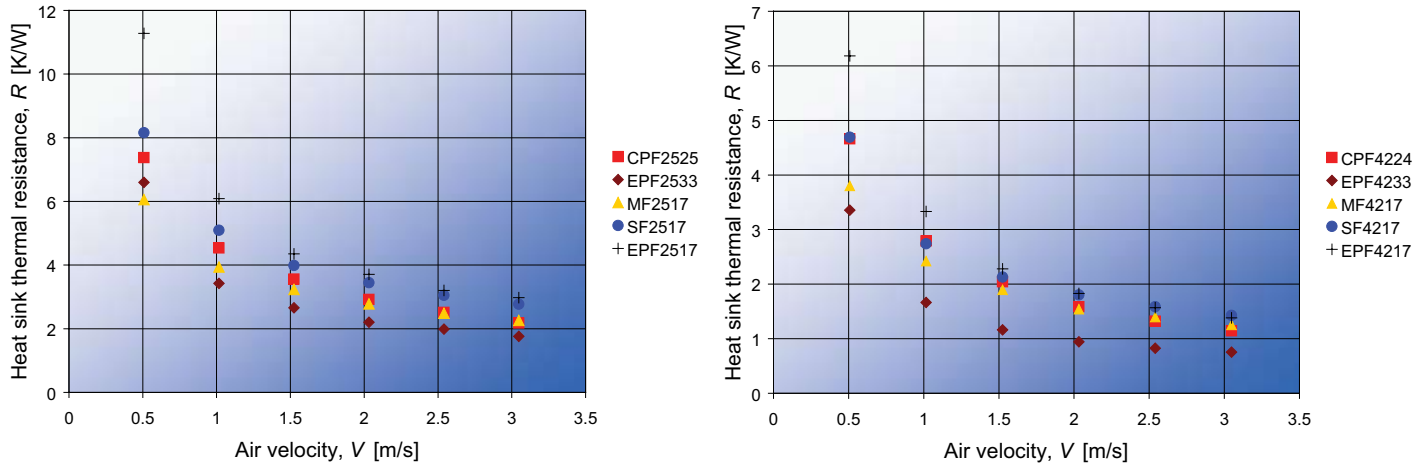


Figure 8. Thermal Resistance of 27 mm (a) and 42 mm (b) Heat Sinks for Forced Convection, Plotted Against Air Flow Velocity.

A criterion for the heat sink selection was that the sinks be the same height. However, the cylindrical pin fin heat sinks were 25 mm high. A possible scaling method is to calculate the heat transfer coefficient,  $h$ , for a heat sink. The heat transfer coefficient was then used to calculate the new thermal resistance of a shorter heat sink of the same type. To calculate the heat transfer coefficient, it was assumed that the heat sink base temperature measured was the fin base temperature and that the fin had a thermal efficiency of 100%, i.e. the fin tip had the same tem-

perature as the fin base. For the heat sinks tested, the thermal efficiency was in the order of 98%. Therefore, it was valid to assume that the thermal efficiency of the fin was 100%. Modifying Equation 3 resulted in Equation 7. The heat transfer coefficient was then calculated using Equation 8.

$$\dot{Q} = (T_b - T_i)/R = hA_{HT} (T_b - T_i) \quad (7)$$

$$h = \frac{1}{RA_{HT}} \quad (8)$$

Each heat sink was first measured and the heat transfer area,  $A_{HT}$ , was cal-

culated. This area was then used in Equation 8. Because elliptical pin fins were tested at two heights, their results were used to verify that the method used to scale the heat sinks was valid. The results for the natural convection tests appear in Figure 9. They show that there was not a good relationship between the heat sinks of the same base dimensions. For a specific heat sink base size, the heat transfer coefficient for the elliptical pin fin heat sinks differed by 25%. This is shown in Figure 9 (b).

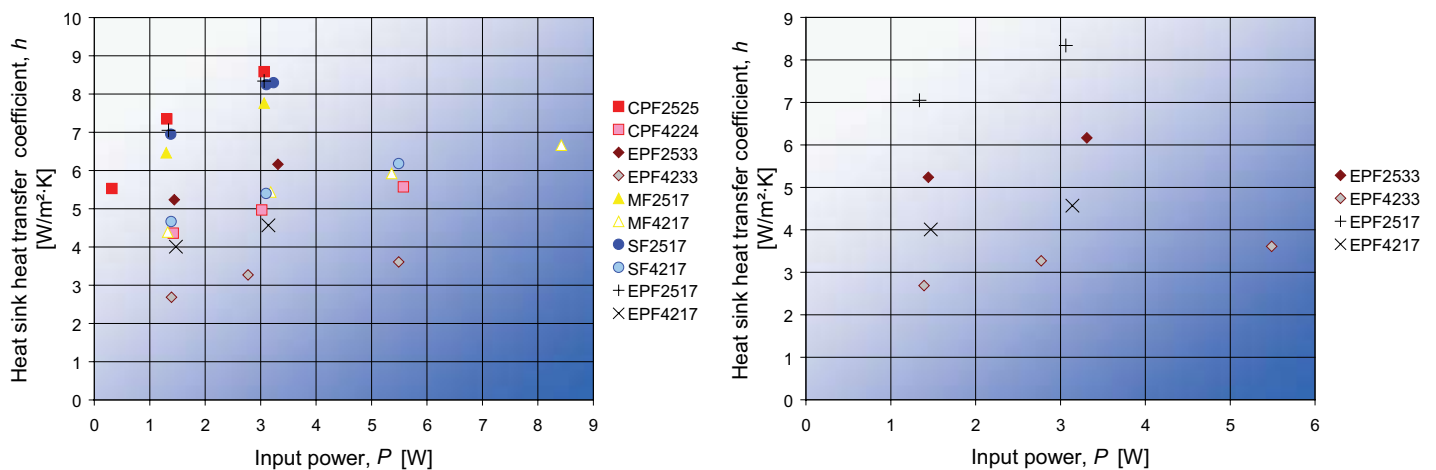


Figure 9. Heat Transfer Coefficients for Natural convection.

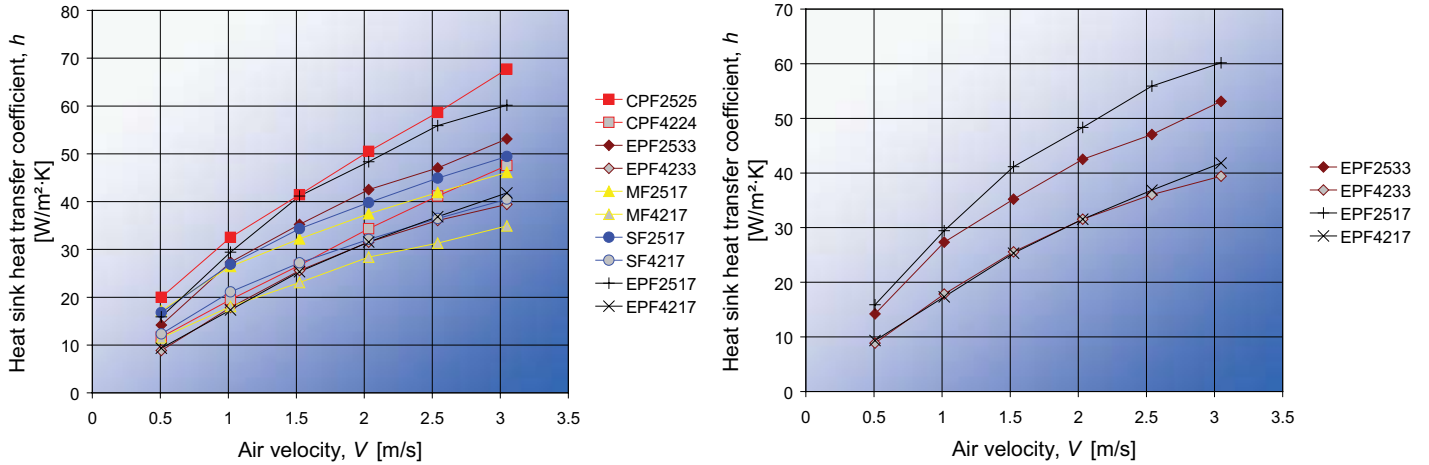


Figure 10. Overall Heat Transfer Coefficients for Forced Convection.

Figure 10 shows the heat transfer coefficient data for the forced convection tests. Figure 10 (b) shows that the heat transfer coefficient for elliptical pin fin heat sinks with 42 mm base dimensions differed by an average of 1.6%. For a 25 mm base, the difference was on average 13.7%. This indicates that the heat transfer coefficient can be used to scale a heat sink. However, this may only be used for heat sinks of the same type and heat sink base area. Using the heat transfer coefficient calculated for the cylindrical pin fin heat

sink, the thermal resistance of a 17 mm high heat sink was estimated. These values are plotted in Figure 11 with the thermal resistance values of the other 17 mm high heat sinks. Figure 11 shows that the maxiFLOW heat sinks had the best thermal resistance for 17 mm high heat sinks, followed by the straight fin heat sinks. The elliptical and cylindrical pin fin heat sinks had the highest thermal resistance of the sinks tested. Figure 11 also shows that the maxiFLOW heat sink were especially effective at air flow velocities below 1.5 to 2 m/s.

Having shown that it was not possible to scale the heat sinks for natural convection, Figure 12 shows the thermal resistance of the 17 mm high heat sink for natural convection.

**Summary**

By documenting the experimental performance of various types of heat sinks, it was found that for the same base dimensions and fin height, maxiFLOW heat sinks had the lowest thermal resistance. Straight fin heat sinks had a higher thermal resistance than

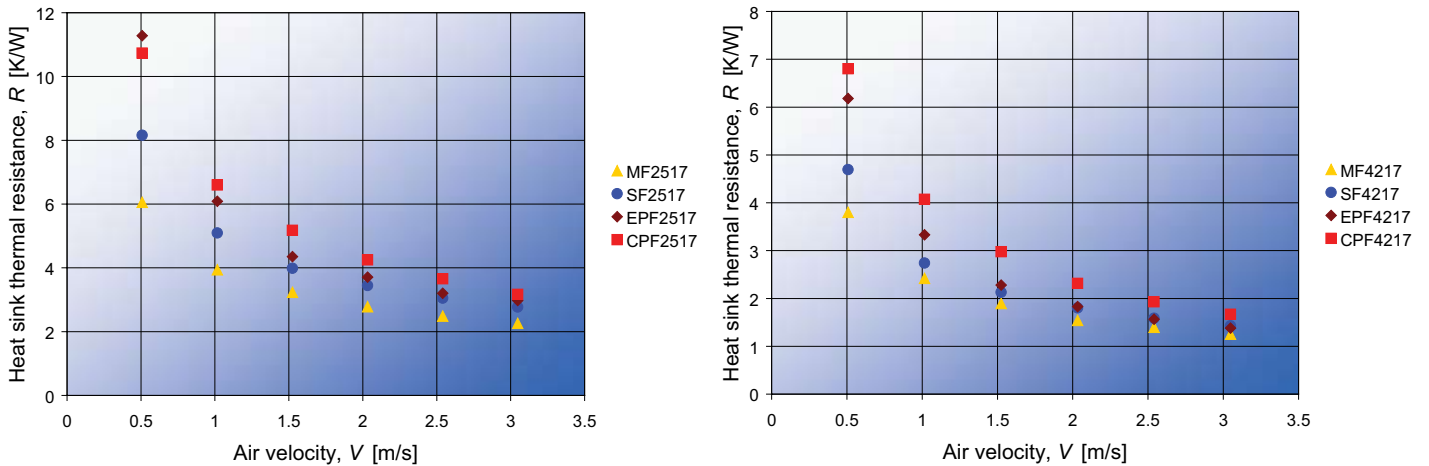


Figure 11. Heat Sink Thermal Resistance for 17 mm High Heat Sinks for Forced Convection. The Values of the Cylindrical Pin Fin Heat Sinks Are Based on the Overall Heat Transfer Coefficient Values of the Taller Heat Sink.



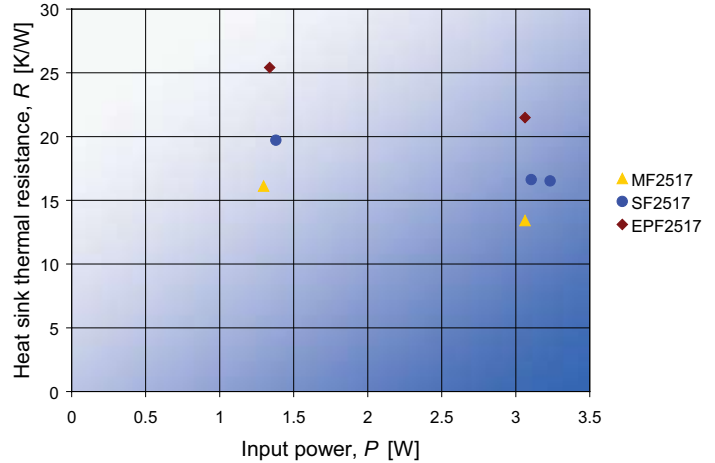
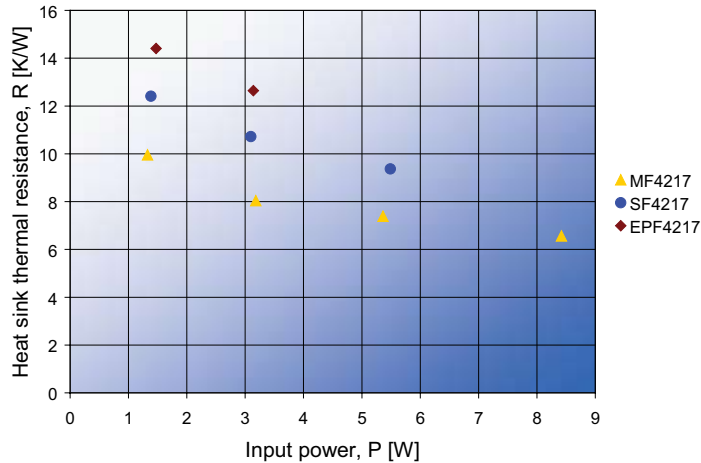


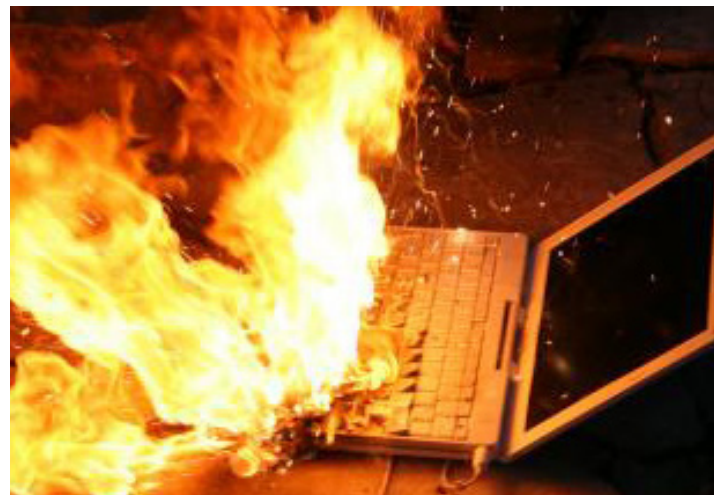
Figure 12. Thermal Resistance of 17 mm High Heat Sink, For the 27 mm (a) and 42 mm (b) Heat Sinks For Natural Convection, Plotted Against Input Power.

the maxiFLOW™ heat sinks. Cylindrical and elliptical pin fin heat sinks had the worst thermal resistance of the heat sinks tested.

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