University of California Riverside

ME170B Experimental Techniques: Lab 2

Forced Convection

Group A5

Elijah Perez | Soham Saha | Alex Pham

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Abstract:

The purpose of this experiment was to study convective heat transfer by observing the ability of various extended surfaces (fins) by comparing their ability to transfer heat to an airflow via convection. By comparing the effects of four different fin arrangements (no fin, single pin fin, multi-pin fin, vertical plate) the effectiveness of each arrangement was calculated. It was hypothesized that the multi-pin fin would be the most efficient arrangement, and that the convective heat transfer coefficient would be linear with the fan speed. The experiment was conducted by fastening the base of the fin to a heater and securing the heater such that the fins were exposed to an airflow controlled by a mini turbine. Using thermocouples, temperature measurements were taken at the base of the fin, along the fin, and at the ambient airflow before and after interacting with the fin. It was found that the convective heat transfer coefficient for the flat plate varied from h = [819 1231 1419 1682] W/k m^2, linearly with fluid velocity. It was also found that the multi pin fin arrangement was the most efficient. By comparing different fin arraignments this experiment allowed for the study of forced convection, providing interesting insights into the analysis of extended surfaces and fins.

Introduction:

Achieving heat transfer through the use of extended surfaces, or fins, is critical to many engineering applications. Industry applications range from electronics to aerospace, essentially anywhere where heat transfer is desired to keep the temperatures of critical components within safe ranges during operations. The working principle of the fin is that the addition of a fin to a plane wall increases the surface area, allowing for a greater rate of convective heat transfer. The parameters affecting a fin's function include its geometry, length, and the thermal conductivity of the fin's material. In industry, fins are designed and manufactured in several geometric arrangements, with the most common options being vertical flat plates, and pin fin arrangements.

A familiar example of a fin's use is a computer heat sink, which has the function of conducting heat away from the computer's CPU (central processing unit). For the heat sink to achieve its function, it must maintain a lower temperature than the CPU. This can be achieved through forced convection. The heat sink has fins protruding from its surface, which are exposed to an ambient airflow at a lower temperature. By increasing the surface area of the heat sink, the rate of convective heat transfer is increased as given by Newton's Law of Cooling, and the heat sink is able to maintain a temperature lower than the CPU. Fins on CPU heat sinks come in different arrangements, and so comparing the effectiveness and efficiencies of various fin arrangements under various convection conditions is critical to understanding and improving the design of these devices.

For this laboratory experiment, four fin arrangements are compared (no fin, single pin fin, multi pin fin, vertical plate), under a set heating load and under four different ambient air velocities. Temperature measurements were taken for the base of the fin, along the length of the fin along with the ambient airflow before and after interacting with the fins. The hypothesis of the experiment was that the multi-pin fin arrangement would be the most efficient, as it had the highest surface area of all four arrangements. It is also hypothesized that the convective heat transfer coefficient *h*, would be linearly proportional to air velocity.

Theory and Derivations:

Convection is the transfer of heat between a solid and the adjacent fluid that is in motion. This lab specifically focuses on forced convection which occurs when fluid flow is caused by external means such as a fan. Heat transfer through convection can be theoretically expressed as the following, where h is the heat transfer coefficient, A is the surface area of the body , T is the surface temperature, and T_inf is the ambient temperature.

$$q = hA(T_s - T_\infty) \tag{1}$$

The heat transfer coefficient is most often considered as an average for the body; however, it can vary as a function of the flow speed through the Nusselt number, Nu. For forced convection, the Nusselt number is generally a function of both Reynolds and Prandtl numbers. The Nusselt number will vary for fin arrangements. The Prandtl number for air can be assumed to be of value: 0.7.

For Cylindrical pins:

$$Nu = 0.683 \cdot Re^{0.466} Pr^{\frac{1}{3}} \tag{2}$$

For multi-cylindrical pins:

$$Nu = 0.4 \cdot Re^{0.6} Pr^{\frac{1}{3}} \cdot (\frac{P_r}{P_{rs}})^{\frac{1}{4}} \tag{3}$$

For single Plate:

$$Nu = 0.644 \cdot Re^{\frac{1}{2}} Pr^{\frac{1}{3}} \tag{4}$$

The heat transfer coefficients for any of the three configurations done in this experiment can be found by using Equation 5.

$$h = Nu\frac{k}{L} \tag{5}$$

Where L is the length of the fin and k is the specified thermal conductivity of the material.

Heat transfer for a fin can be determined to be from the following equation, where P is the perimeter, k is the thermal conductivity, As is the surface area, and θb is the temperature difference between the base temperature and the ambient temperature.

$$q_{fin} = M \cdot \frac{sinh(mL) + \frac{h}{mk}cosh(mL)}{cosh(mL) + \frac{h}{mk}sinh(mL)} \tag{6}$$

Where:

$$M = \sqrt{hPkA_c\theta_b} \tag{7}$$

$$\theta_b = T - T_b \tag{8}$$

$$m = \sqrt{\frac{hP}{KA_c}} \tag{9}$$

We can also express the temperature distribution across the fin as:

$$\frac{\theta}{\theta_b} = \frac{cosh\ m(L-x) + (\frac{h}{mk})sinh\ m(L-x)}{cosh\ mL + (\frac{h}{mk})sinh\ mL} \tag{10}$$

or we can rewrite the expression as:

$$T(x) = (T_b - T_\infty) \cdot \frac{cosh\ m(L-x) + (\frac{h}{mk})sinh\ m(L-x)}{cosh\ mL + (\frac{h}{mk})sinh\ mL} + T_\infty \tag{11}$$

Although fins are used to increase the heat transfer from a surface by increasing the effective surface area, the fin also contains a conduction resistance to heat transfer. Therefore, it is not assured that the heat transfer rate will increase through the use of fins. Fin effectiveness can be shown as the ratio of fin heat transfer to heat transfer rate in the absence of fin.

$$\epsilon = \frac{q_{fin}}{hA_c\theta_b} \tag{12}$$

Another fin thermal performance measurement is the fin efficiency. The fin efficiency is the ratio of the fin heat transfer over the maximum possible heat transfer of the fin, where Af is the surface area of the fin and m is determined by Equation 9.

Experimental:

$$\mu_{fin} = \frac{q_{fin}}{q_{max}} = \frac{q_{fin}}{hA_f\theta_b} \tag{13}$$

Theoretical:

$$\mu_{fin} = \frac{tanh\ mL_c}{mL_c} \tag{14}$$

Methods:

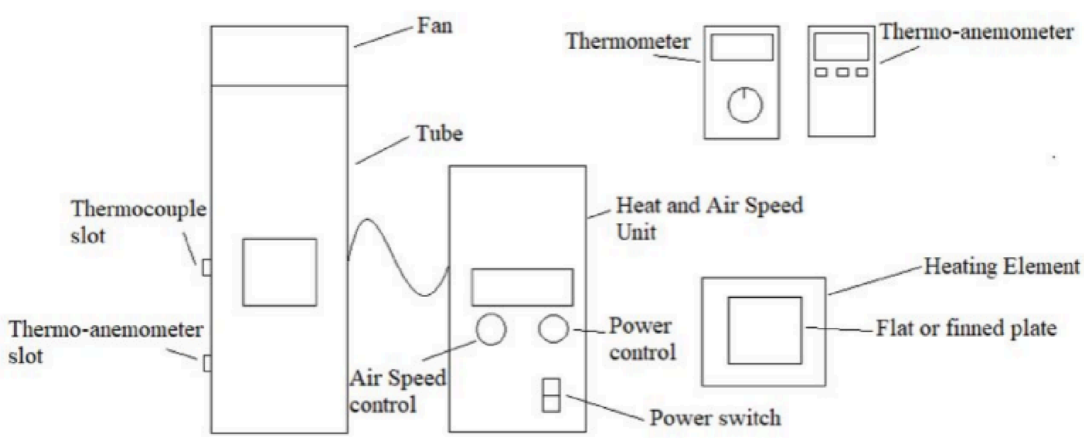

Figure 1: Schematic of forced convection apparatus with main components shown

Several trials with varying air velocities were performed in order to obtain temperature readings for four different types of 6068 aluminum elements. No calibrations are required. Apply a thin layer of conductive paste to the underside of the flat plate, and mount and screw the plate onto the heating element. Once secured, connect to the power source and turn on the power source. Set power to 60 W, allow the block to heat up, and reach a stable temperature. Turn on the fan to speed setting 1, wait for the temperature to stabilize, and record the temperature. Alternate the wind velocity to speed settings 2-3 and record the temperatures. Replace the flat plate with a single pin or cluster of pins or flat fin. Turn on the fan and alternate speed as stated above. Utilizing the three temperature probes, record the temperature at the base, middle, and tip. Make sure the probe is in contact with the pin/finned surface. Repeat the steps listed above for the remaining pin/finned elements; record the data. Once the experiment has been completed, turn off the power source and allow the apparatus to cool down.

To analyze the data, use Equation 5 to calculate the heat transfer coefficients for each fin arrangement. First determine the relationship between heat transfer coefficient and fluid velocity, then use Equation 12 to calculate the fin effectiveness and Equation 13 to calculate the fin efficiencies for each arrangement. Matlab scripts were utilized for the data analysis in this experiment.

Results:

Equation 5 was used to calculate the heat transfer coefficient for the flat plate, single pin, multiple pins, and vertical plate. The heat transfer coefficient for the flat plate varied from h = [819 1231 1419 1682] W/k m^2 as the velocity increased. This linear relationship between heat transfer coefficient and fluid velocity can be clearly seen in Figure 2.

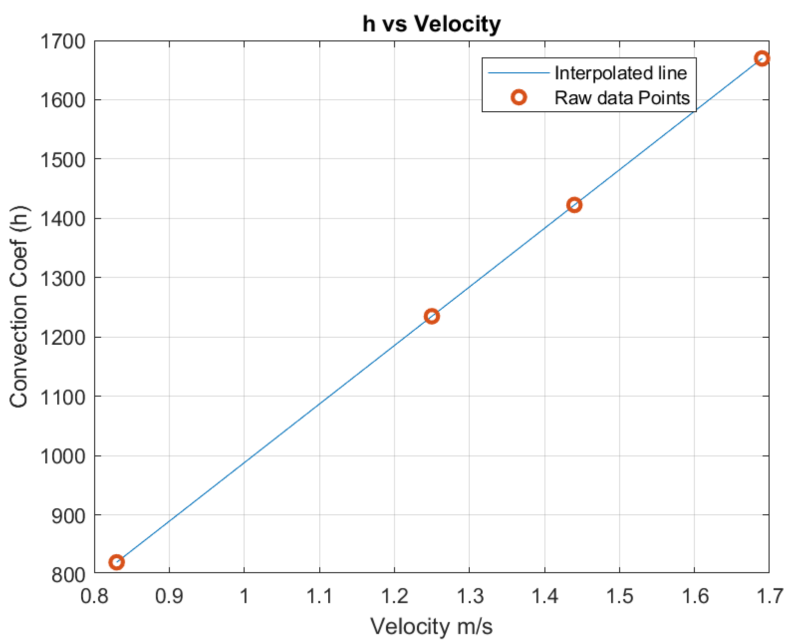

Figure 2: Heat Transfer Coefficient at different velocities for a flat plate

We observed that the temperature across the fins dropped as a function of distance as expected. The results for each fin case can be seen in Figures 3,4 and 5. The efficiency of the multiple pins and vertical plate were calculated using equation 13, they were found to be 0.35 and 0.101. The effectiveness of those fins was also calculated using Equation 12 the results are shown: 0.47 and 0.78. The temperature distribution for the pin and plate-fin were calculated using Equation 6 they were found to be 285 k and 295 k. Comparing these temperatures to the experimental data we can see from Figures 3 and 5 for plate and single pin fin the temperature distribution has a difference of 275 k for both fins.

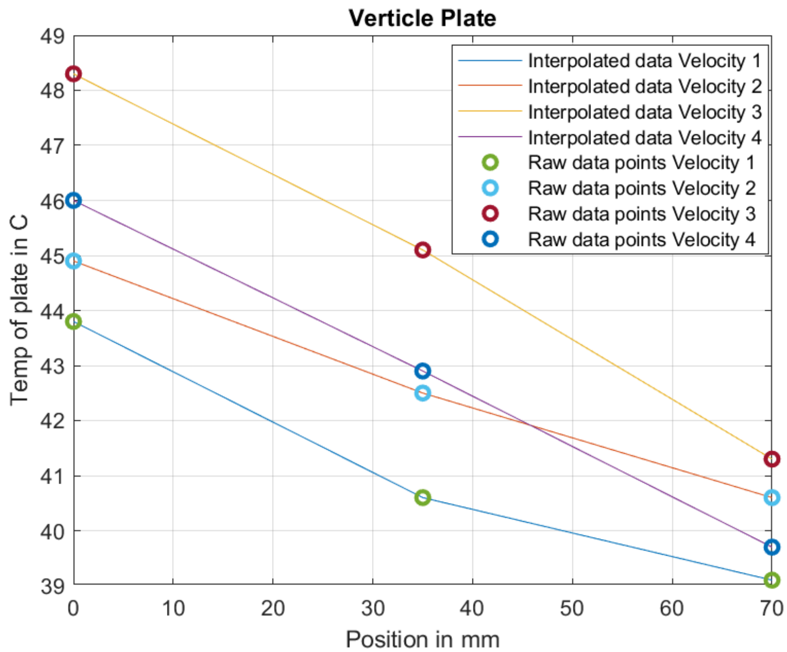

Figure 3: Temperature distribution across the vertical plate

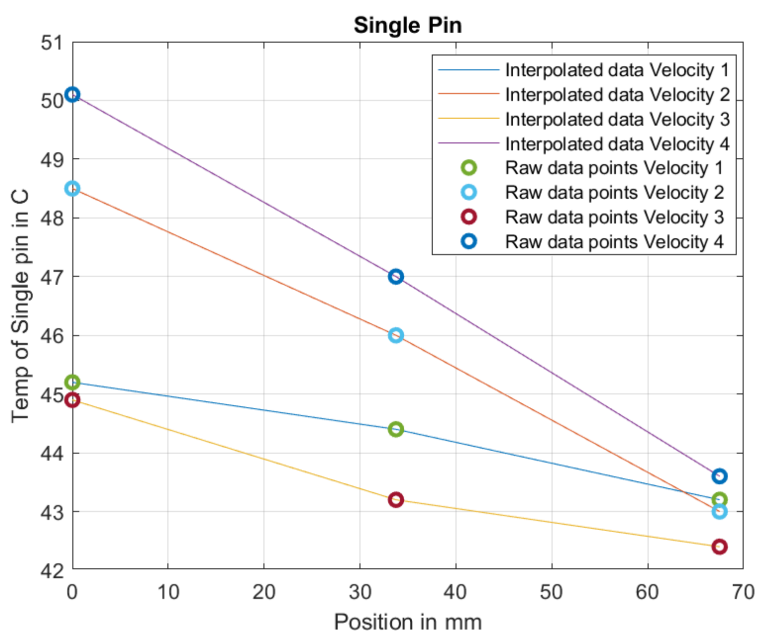

Figure 4: Temperature distribution across the single-pin fin

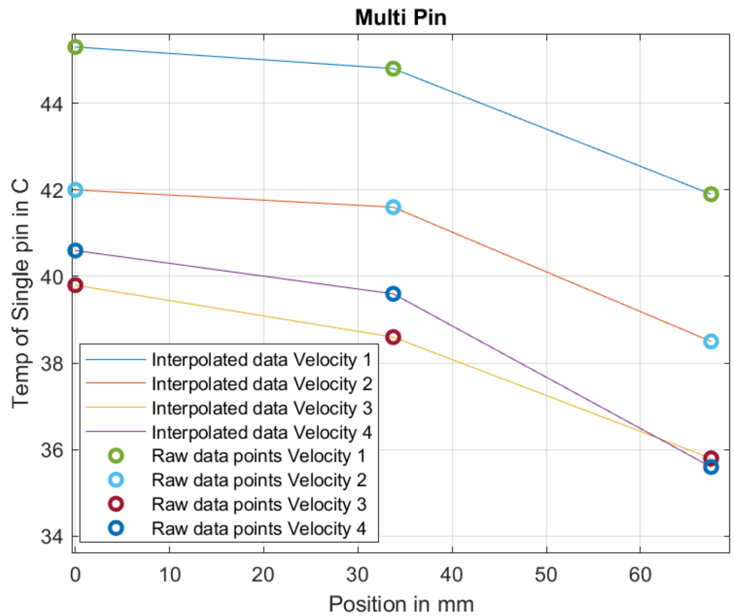

Figure 5: Temperature distribution across the multi-pin fin

In Figure 6, we can see that our experimental data is off from the theoretical temperature distribution, which could be caused by the minimal data set available. Having only three temperature distributions makes it difficult to determine the type of relationship between the variables. If there were more temperature data points to be observed it would be easier to say whether our experimental data resembles the theoretical temperature distribution. We also observed that on average the temperature at the end of the fin was higher than predicted using Equation 11. It is believed that this could be the result of uncertainty in heat transfer coefficients that were measured experimentally.

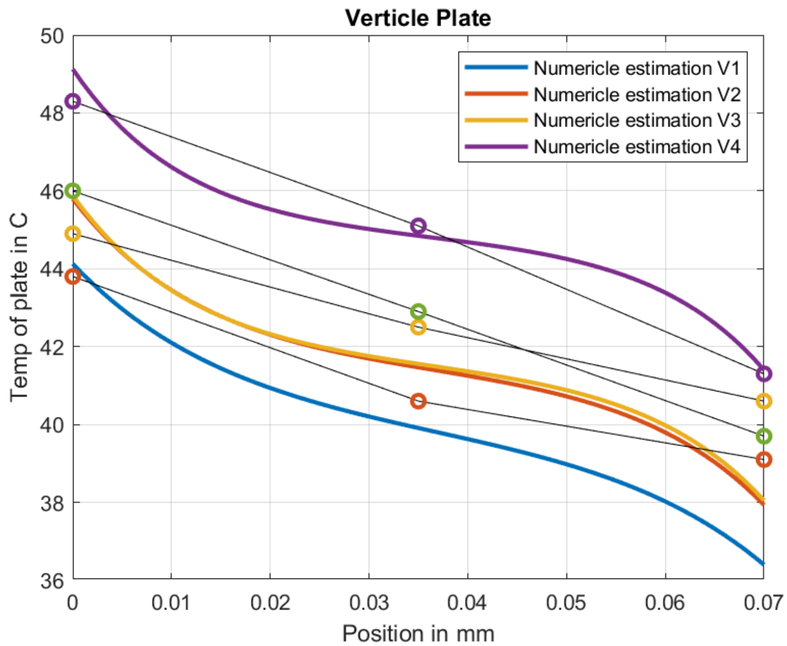

Figure 6: Theoretical Temperature distribution across the vertical late

Discussion:

The results of the experiment are significant as they suggest that the convective heat transfer coefficient is indeed linearly proportional to the fluid velocity, as predicted by the hypothesis. Additionally, the results also suggest that the multiple fin arrangement had the highest efficiency, which was also predicted by the hypothesis. This renders the initial hypothesis valid, as the results are consistent with the predictions that the convective heat transfer rate would be linearly proportional with velocity The implication of the results are that the rate of forced convection heat transfer depends heavily on the velocity of the fluid. This intuitively makes sense, but the results of this experiment solidify the intuition. The other major implication of the results are that fin efficiency and fin effectiveness are not necessarily directly correlated. This was seen when the efficiency of the multi-pin arrangement was higher than that of the vertical plate, with the vertical plate however having a higher fin effectiveness.

While this experiment allowed for interesting findings it is critical to mention the sources of errors and uncertainties in this experiment along with the limitations. The primary source of error and uncertainty was in the arrangement, calibration, and measurement fluctuations of the thermocouples. There are several reasons for this. Firstly, the thermocouples measuring the base and fin temperatures had to be pressed against the surface to properly read the temperature. The amount of force applied to achieve this varied with each measurement as it was done manually. Additionally, the thermocouple arrangements for the first half of the experiment were different than that of the second half as the experiment was conducted over the course of two laboratory sessions. It was also noted that individual thermocouple measurements contained fluctuations and discrepancies when measuring the same temperature, bringing into question the calibration of the thermocouples and creating another source of uncertainty in measurement. Along with the source of error and uncertainty, this experiment also had several limitations. The primary limitation of the experiment was the assumption that temperature distribution across the individual pins on the multi-pin fin arraignment was equal, as the temperature distribution was measured solely for the center-most pin. It was also assumed that the wattage consumed by the heating device was entirely transferred to the base of the fin. While thermal paste was applied to the base and the heater, it is not possible to ensure 100% contact and conduction. Additionally, it was noted that when fan speeds were adjusted, the wattage meter would fluctuate, indicating the power reading was for the operation of both the fan and the heater.

Conclusion:

It is concluded that the convective heat transfer coefficient indeed increases linearly with fan speed (fluid velocity) as expected. It was also concluded that the multi-pin plate and the flat plate were inconsistent with the expected results as they generated h values that were almost what we expected. It's believed that the slight errors in the final result could have been due to the thermocouples not reading accurate temperatures. There was also large uncertainty with their reading especially since they were not making direct contact with the fins consistently. If this experiment were to be performed again, a lower heat setting would be selected to allow thermal equilibrium to be achieved more rapidly and to allow more steady results. For future tests, the heating element and forced convection apparatus should be tightly snug to allow for tighter airflow and accurate temperature readings. Empirical results also showed that the rectangular fin is more effective than the single-pin fin. Despite the error, the results still show that installing a fin system will increase heat transfer. This helps to validate why most electronic manufacturers utilize a fin configuration when installing heat sinks as opposed to cylindrical fins and simple plate heatsinks. Another major recommendation would be to have longer fins with more thermocouples to see a more detailed temperature distribution.

References:

[1] Incropera, F.P., and DeWitt, D.P., Fundamentals of Heat and Mass Transfer, 5th edition, John Wiley & Sons, New York, 2000.
[2] Convective Heat Transfer. (n.d.). Retrieved from The Engineering ToolBox: https://www.engineeringtoolbox.com/convective-heat-transfer-d_430.html

Statement of Contributions:

Elijah Perez: Experimental Design, Data Analysis, Lab Report (Theory & Derivations, Methods, Results, Conclusion)
Soham Saha: Experimental Design, Data Collection, Lab Report (Abstract, Introduction, Discussion
Alex Pham: Experimental Design, Data Collection, Measurement of Materials

Appendix:

Table 1: Total surface Area for various shapes

Shape	Area formula	Measurements (mm)	Area (m^2)
Flat Plate (base)	Base area	101.56 * 109.29	0.0110994924 m^2
Vertical Plate	width*height + edge *height + base area	101.65* 69.99 + 2.30*69.99 + 101.56 * 109.29	0.0183749529 m^2
Single Pin	πDL + base	π*12.66*67.05 + 101.56 * 109.29	0.0137662427488
Multi-Pin	17*πDL + base	π*12.66*67.05* 17 + 101.56 * 109.29	0.0564342483292

Table 2: Heated Flat Plate at (101.3+-.1 W) \| A = 0.0110994924

	Mass Flow rate m'	**T ambient entering**	**T ambient exiting**	**T base**	**ΔT ambient**	**Q =m' Cp $\Delta T_{ambient}$ (W)**	**H = Q / A ΔT**
Fan Speed 1 = 0.83m/s	0.00905594816 kg/s	20.4	23.86	64+/-0.5	3.46	31.49024853	819.967938249
Fan Speed 2 = 1.25m/s	0.01363847615 kg/s	20.5	24.4	59.7	3.9	53.45600726 W	1234.89147379
Fan speed 3 = 1.44m/s	0.01571152452 kg/s	20.4	24.5	56.8	4.1	64.73933678 W	1422.59497754
Fan speed 4 = 1.69	0.01843921975 kg/s	20.5	24.53	52.4	4.03	74.68160587 W	1669.57327243

Table 3: Vertical Plate Fin (104.5 W)

	T ambient entering	T Ambient exiting	T Base	T middle	T tip
Fan Speed 1 0.89 m/s	19.8	26.5	43.8	40.6	39.1
Fan Speed 2 1.29 m/s	19.4	26.2	44.9	42.5	40.6
Fan speed 3 1.49	19.5	26.2	48.3	45.1	41.3
Fan speed 4	19.8	26.2	46	42.9	39.7

Table 4: Single Pin fin (101.3 W)

	Mass Flow Rate m*	T ambient entering	T Ambient exiting	T Base	T middle	T tip
Fan Speed 1 = 0.90 m/s	0.00981970282	20.3	32.366	45.2	44.4	43.2
Fan Speed 2 = 1.28 m/s	0.01396579957	20.3	23.333	48.5	46.0	43.0
Fan speed 3 = 1.52 m/s	0.016584387	20.3	24.3	44.9	43.2	42.4
Fan speed 4 = 1.74m/s	0.0189847588	20.4	23.466	50.1	45	43.6

Table 5: Multi Pin fin (*assumes temp distribution all 17 fins is equal) 102.5W

	T ambient entering	T Ambient exiting	T Base	T middle	T tip

Fan Speed 1 0.8 m/s	20.9	35.1	45.3	44.8	41.9
Fan Speed 2 1.20 m/s	20.8	31.4	42.0	41.6	38.5
Fan speed 3 1.40m/s	20.6	29.9	39.8	38.6	35.8
Fan speed 4 1.61	20.4	28.8	40.6	38.6	35.6