

# Final Report

## Team 13

### Designing & Testing a Thermal Management System for a SiC PV Converter



#### Members:

Melanie Gonzalez	mtg12c
Leslie Dunn	lvd12b
James Hutchinson	jrh12c
Tianna Lentino	tnl13
Colleen Kidder	cmk13

#### Faculty Advisor/s

Dr. Hui Li

Dr. Juan Ordonez

#### Sponsors

The Center for Advanced Power Systems (CAPS)

PowerAmerica – U.S. Department of Energy

#### Instructors

Dr. Chiang Shih

Dr. Jerris Hooker

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## ABSTRACT

The goal of this project, sponsored by Power America, was to design a cost-effective, lightweight, thermal cooling system for a SiC PV converter being developed by researchers at CAPS. Their goal for this converter was to have the highest power density available on the market. However, the original heatsink that was dissipating heat from the power modules in the converter was cool during operation and accounted for 1/3 of the weight of the system, proving it to be overdesigned. By reducing the weight of the thermal cooling system, Team 13 helped CAPS increase the power density from 2.5 kW/kg to 6.54 kW/kg. The objective of designing an improved heatsink design was verified using three different approaches: theoretical analysis, experimental testing, and thermal simulations using COMSOL Multiphysics software. A heat source emulator was developed for physical testing of both plate fin and pin fin configurations in order to safely test without using the SiC power modules of the converter. The two designs were compared and an optimized heatsink was designed after simulations, calculations, and testing were verified with one another. The final solution was to have four pin-fin heatsinks that housed two power modules each. Each heatsink had a thermal resistance of about 0.08 K/W, and weighed 211 grams. The total weight of all four cooling systems, including the weights of fans and screws, was 1.72 kg. This optimized cooling system had a 71% reduction in weight from the original one, and was designed to keep the power modules well under their maximum operating temperatures of 120C.

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On a final note, Team 13 gives thanks to Dr. Shih and Dr. Hooker for providing feedback and guidance necessary for a successful project.

# 1. Introduction

In silicon carbide photovoltaic converters, it is necessary to manage the thermal by-product of power electronic devices to prevent failure of the system. One of the most common methods to remove heat from the system is to use a heatsink and fan combination. Typically, a heatsink will be near to 30% of the overall system weight, significantly impacting the size of the converters. These heatsinks are usually not optimized to fit the specific power module and tend to be overdesigned which translates to wasted material, as well as an increase in the weight, size, and cost of the overall system. Team 13 has produced an optimal heatsink design as well as a heatsink selection guide to assist researchers at the Center for Advanced Power Systems (CAPS) in obtaining their goal of having the highest rated power density for a PV converter worldwide. This project focused on studying forced convective Aluminum heatsinks with two different fin designs: cylindrical pin fins and rectangular plate fins. These heatsink types were analyzed using thermal simulations, theoretical calculations, and experimental testing, which were verified with one another. Through this analysis, the team decided to pursue a pin fin heatsink design that would house two SiC power electronic devices. Optimization was achieved for a pin fin heatsink, and a new lightweight fan was selected.

## 2. Project Definition

### 2.1 Scope

#### 2.2.1 SiC in Power Electronics

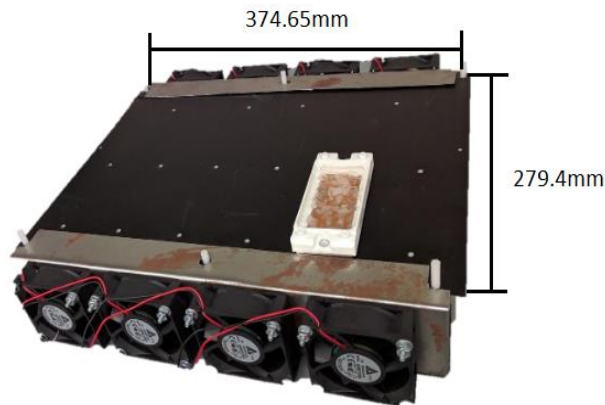
Silicon Carbide (SiC) switching devices are wide bandgap semiconductors that are the future of semiconductor devices. SiC devices can cut power losses in half compared to its counterpart silicon through higher efficiencies. This is because they switch at higher frequencies and operate at higher temperatures voltage. Wide bandgap devices are more efficient but also more expensive than the popular silicon choice. Applications of wide bandgap power electronics can impact small electrical devices such as computer chargers, which can be made smaller and more efficient, and large solar farms and wind turbines, which can be connected to the grid efficiently. Many industry leaders, including PowerAmerica, wish to make SiC a viable, cost-effective option for power electronic device manufacturers. To do so, research in converters that incorporate SiC switching devices must be done in order to lower the overall system cost. Researchers at the Center for Advanced Power Systems (CAPS) are helping to lead this initiative, developing both 50kW and 100kW power converters with SiC power electronic devices.

#### 2.2.2 CAPS PV Converter

Researchers at CAPS are developing both 50kW and 100kW photovoltaic converters using SiC switching devices. The converters would be used in conjunction with a solar array to convert DC power generated by the solar panels into AC power to be distributed into the electrical grid. With the use of the new SiC power electronic technology, the converters are highly efficient. The PV converter developed at CAPS has a very high power density of 2.5kW/kg. Researchers at CAPS want to have the highest power density converter because it would mean that their converter outputs the largest amount of power with the smallest amount of mass. Furthermore, having a small system is desirable because it makes installations easier.

Researchers at CAPS know that they could further increase the converter's power density because they realized that the thermal cooling system was extremely heavy and overdesigned. It was oversized, weighing 6.45kg and the bottom of the fins remained cool during operation. The thermal cooling system consisted of a plate fin heatsink with eight fans fixed to the sides, and

housed eight power modules. It had dimensions of 37.5cm X 27.9cm X 8cm and is shown in Figure 1 with one of the power modules on top.



*Figure 1: Original Heatsink Used by CAPS*

Optimizing the cooling system for this PV converter would significantly reduce the systems weight and in turn increase the power density of the PV converter.

## 2.2 Problem Statement

Usually, heatsinks and fans are used to remove the heat generated by electrical devices. However, these heatsinks have a flaw in their design – they are rarely optimized for the specific application and this can significantly impact its overall design. Not optimizing a heatsink can result in a much larger, more expensive, and a heavier thermal management system. This project proposes a way to improve the thermal management system for the SiC photovoltaic converter being developed by CAPS to make the system cost effective, smaller, and lighter.

## 2.3 Customer Needs Statement

The customer (researchers at CAPS) developed a heat sink for an electrical converter capable of cooling eight SiC power modules operating at a max temperature of 150°C. The thermal management system used a rectangular plate fin type design with 8 fans oriented horizontally along each side. This design, when installed into an electrical converter contributed 32.5% of the total weight and took up a large amount of space. It was also over-designed as the bottom of the fins on the heat sink remained cool throughout all operation.

“The current heat sink system is over designed and takes up too much space and is too heavy once installed under the electrical converter.”

After getting customer input, Team 13 developed a House of Quality (HOQ) to narrow down the most important components of the project. It is included in Appendix B of this report.

## 2.4 Goals & Objectives

Researchers at CAPS required a cost effective, lightweight thermal management system to be designed and rigorously verified for a PV converter. They had hopes of doubling the power density from 2.5kW/kg to 5kW/kg when assigning Team 13 with the task of optimizing the heatsink used for cooling the power modules. Prior to optimization, Team 13 considered two common heatsink designs: rectangular plate fin heatsinks, and cylindrical pin fin heatsinks. The team used 3 methods to approach comparing and selecting a design, including theoretical analysis, thermal simulations using COMSOL Multiphysics software, and experimental testing. After applying the three methods, an optimized bi-modular pin fin heatsink was designed with an appropriate fan selected. Team 13 also created a heatsink selection guide to provide insight in selecting appropriate heatsinks for further applications. The guide is included in section 5 of this report.

As provided by the sponsor, Figure 2 shows a high-level diagram of the responsibilities for both electrical engineering students and mechanical engineering students. Each subset of students performed their respective research to complete their portion of the overall project. After the students of the electrical and mechanical engineering discipline completed these individual tasks and responsibilities, the two disciplines combined their work for analysis of the final result. Collaboration between the two disciplines was crucial to ensure the team was headed in a cohesive direction.

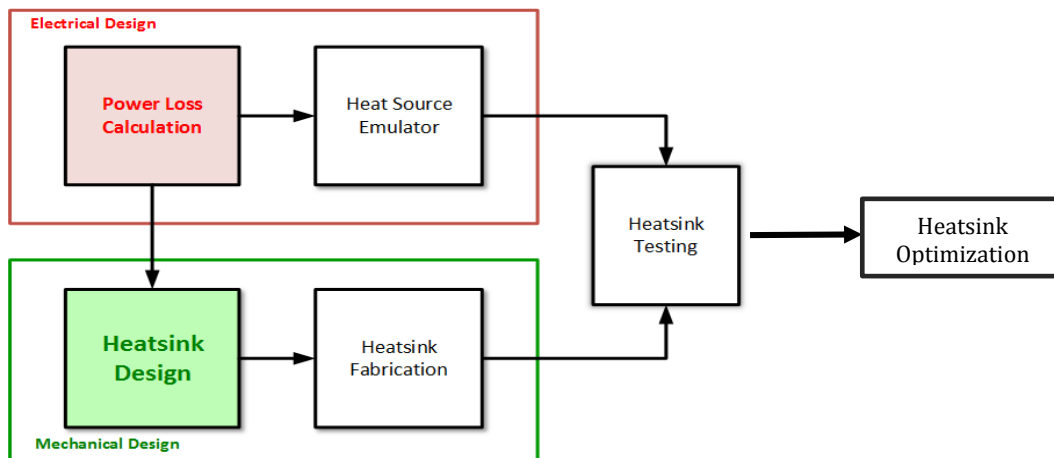


Figure 2: Responsibilities of EE and ME Students (Image Courtesy of Thierry Kayiranga)

## 2.5 Constraints

The following constraints were defined to guide the project to the finalized build. These constraints were slightly adjusted at the beginning of December when new information arose and the team better understood the scope of the project. The constraints are as follows:

- Heatsink must be made from Aluminum alloy (due to low cost and high thermal properties)
- Heatsink must weigh less than 6.5 kg
- System must prevent eight power modules from exceeding 120°C (30 degrees below failure point)
- Must reduce size of current heatsink design
- Heatsink must have a maximum thermal resistance below 0.792 K/W
- Each power module is assumed a maximum power loss of 100W

## 3. Methodology

Team 13's approach to finding a lightweight thermal cooling system for a PV Converter was to consider two common heatsink types: Plate pin and pin fin heatsinks. The team utilized 3 methods to analyze these heatsinks, including experimental testing, thermal simulations using COMSOL Multiphysics software, and thermal resistance calculations. These approaches are discussed as follows, with a comparison of their results. Additionally, a comparison between both plate and pin fin heatsinks types is made, and the optimal heatsink procedure is discussed.

### 3.1 Concept Generation

#### 3.1.1 Bi-modular design

The original heatsink housed all eight power modules on its baseplate. Their total footprint area accounted for roughly 38% of the available area on the heatsink baseplate. The fins for it were over 7cm tall, and were close to room temperature during operation. This gave team 13 reason to believe that the heatsink had excess material. To eliminate this unneeded material, the team decided to implement a bi-modular heatsink design. This concept called for four separate heatsinks, each housing two power modules and would eliminate the unused area, decreasing unnecessary weight. The baseplate size would be large enough to house two modules with at least 1.5cm distance separating them. With this kind of arrangement, the modules would take up 10.8cm x 10.7cm, which would be the absolute minimum footprint of the baseplate. Slightly larger baseplate sizes were analyzed to ensure tolerancing and adequate module attachment.

Another important aspect of the bi-modular design was its potential for an improvement in overall heat transfer. Due to the smaller size, the channels for airflow would be reduced. This ensured that a constant flow of cool air supplied by the fans would quickly enter and exit through the heatsink. The longer channels in the previous heatsink had the potential to increase the air temperature as the air would have taken a longer time to travel throughout the entry and exit points. This increase in air temperature had negative effects on the air flow characteristics and the overall heat transfer. By choosing a bi-modular concept, Team 13 could ensure that unnecessary weight and undesired air flow was kept to a minimum. All design concepts were made from Aluminum

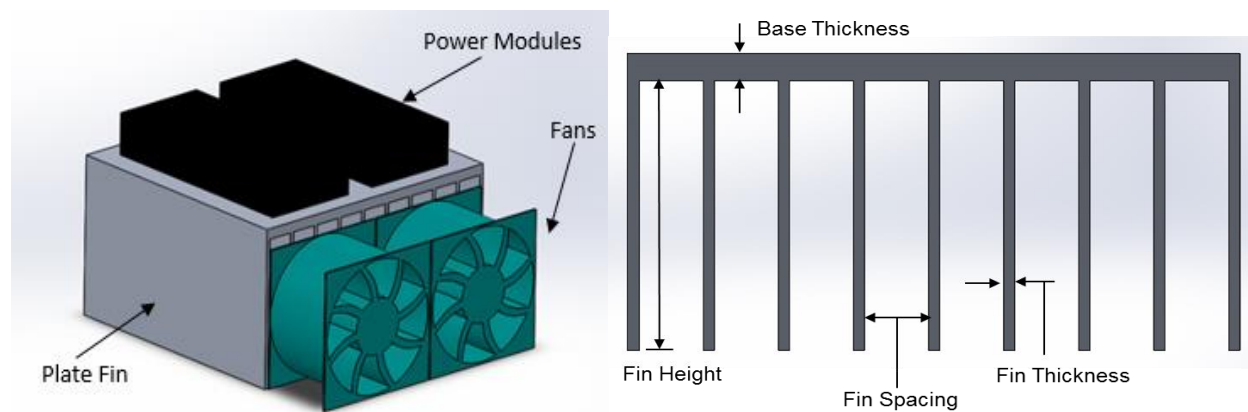


T-5 6063 due to its high thermal conductivity, low density, light weight, high thermal conductivity, low cost, and manufacturability.

Once it was decided that a bi-module heatsink design would be implemented, Team 13 had to decide whether to pursue a pin or a plate fin heatsink type. A heatsink of each type was selected for experimental study. Researchers at CAPS had a plate fin heatsink available that Team 13 could cut down to the desired size, and a pin fin heatsink was ordered from a manufacturer.

### 3.1.2 Plate Fin Heatsink

The plate fin heatsink that was selected for detailed study had nine rectangular fins and a length, width, and height of 127mm x 127mm x 69.2mm. The baseplate was 6 mm thick while the fins each had a thickness of 2.5 mm. Two power modules were placed on top of the heatsink baseplate. The plate fin was cooled by two fans fixed to the side of the heatsink that each had a maximum flow rate of 1.73 m<sup>3</sup>/min. The air flowed over the fins in the lateral direction. The total weight of the plate fin heatsink with the fans was 0.954 kg. The configuration of the plate fin heatsink with its important geometric features is shown in Figure 3.



*Figure 3: Plate Fin Configuration*

The plate fin design contained two main advantages, one being that the air flow acted in the laminar regime making the analytical process less complicated than that of the pin fin. However, a slight decrease in heat transfer was expected due to the laminar airflow. The manufacturability of the design was the second advantage as plate fin heat sinks are cheaper and more customizable as compared to their pin fin counterparts.

### 3.1.3 Pin Fin Heatsink

The pin fin heatsink that was chosen for study consisted of 313 small circular fins and an overall length, width, and height of 113.7mm x 113.7mm x 17.8mm. The pin fin was a staggered design in which the spacing between the rows of fins alternate, rather than being equidistant. The baseplate had a thickness of 4.7 mm, and the pins each had a diameter of 3.2 mm. The inline rows of pins were spaced 9 mm apart while the alternating staggered rows were spaced 4.5 mm from the adjacent inline rows. The heatsink had one fan, approximately the size of its baseplate, fixed over the tops of the pins which allows air to flow in the axial direction of the pins. It should be noted that other fan orientations were considered, but research proved them to be obsolete. Impingement cooling (with the fan mounted to the bottom) is the best way to provide uniform cooling across the baseplate. The max flow rate of the fan that was used was 3.03 m<sup>3</sup>/min. The overall weight of the heatsink and fan was 0.553 kg. Two power modules were to be placed along the topside of heatsink. The setup of the pin fin and its significant geometric features are shown in Figure 4.

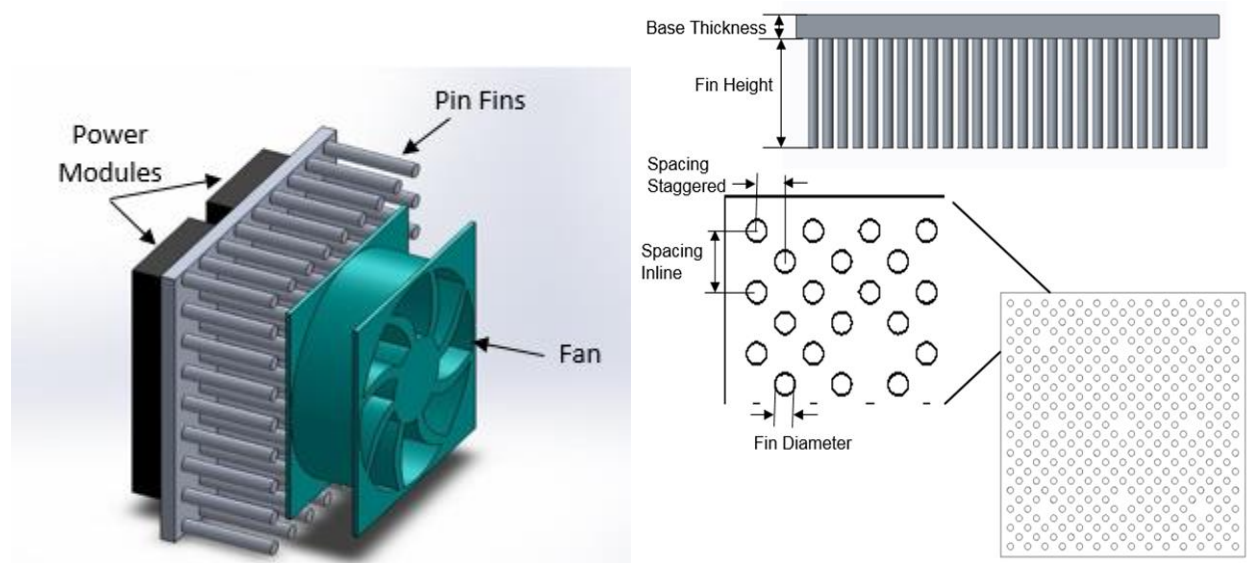
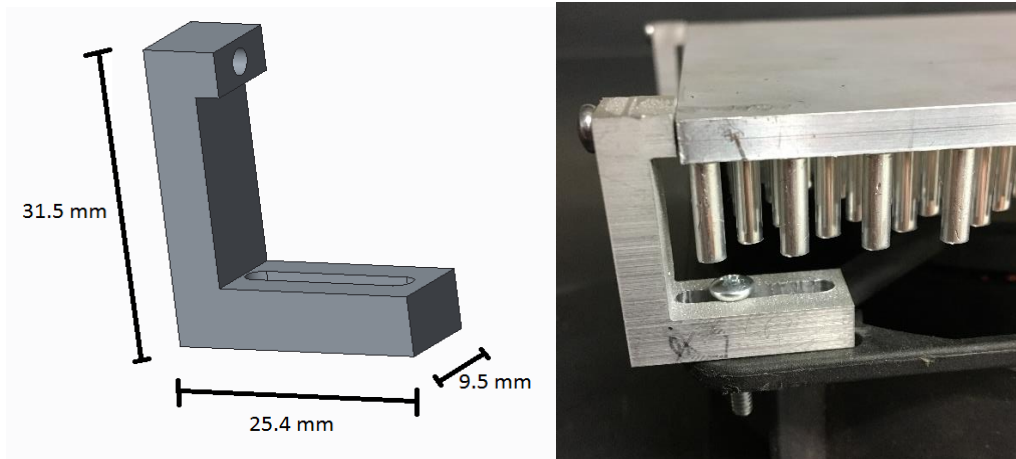


Figure 4: Pin Fin Configuration

Due to the fan being mounted axially along the base, a larger dimensioned fan was needed to cover the area of the pins, which caused the weight of the system to increase. The fan needed to be mounted in a manner where any fasteners, screws or bolts do not impinge on the pins or change the overall geometry of the heat sink as well. Team 13 developed an L-bracket to connect the fan

to the heatsink, which is shown in Figure 5. The fan was secured with four of these connectors, which were positioned near each of the corners of the heatsink. The longer end of the brackets were screwed into the sides of the heatsink while the shorter end fixed the fan in place with screws and nuts.



*Figure 5: Pin Fin Connector L-Bracket*

Overall, the pin fin design was expected to have better thermal properties than the plate fin design. This was because the cylindrical fins exposed more surface area per volume than the plate fin, thus enhancing the convective cooling.

The pin and plate fin heatsinks listed in this section were physically tested. Simulations and calculations were computed for many more variations of these types of heatsinks but for simplicity, the analytical results discussed in the following sections will be limited to the heatsinks that were just introduced.

## 3.2 Experimental Testing

### 3.2.1 Emulated Heat Source

To test the potential heatsink design and not risk any damage to the power modules used in the SiC PV converter, a heat source emulator needed to be constructed. Although power modules were assumed to have a 100W power loss that physically manifested as heat generated, it was more crucial to perform test for a variety of power dissipations to help fully characterize the heatsink design rather than only test using 100W loss. To emulate this loss, a heat source was constructed using high power resistors, each capable of consuming no more than 100W.

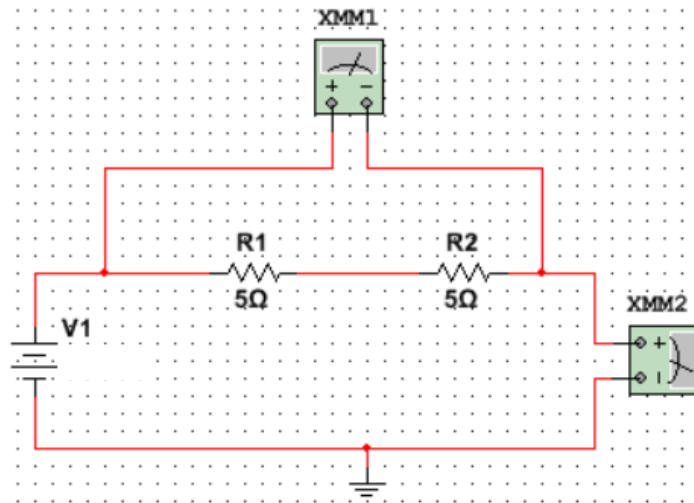


Figure 6: Circuit Diagram for 1 Emulator

Team 13 acquired ten  $5\ \Omega$ , 50W resistors to use for the heat source. To achieve near a 100W power loss per emulator, at least two of these resistors needed to be used. Figure 6 shows the circuit diagram of this configuration to reach the desired power loss. With two 5 ohm resistors in series, and an input of approximately 30 V, the desired 100W of loss could be achieved.

To accurately model the thermal property of the power modules, a copper base plate for the emulator was used to spread the heat generated by the resistors more evenly. This was done because it was known that the SiC switches inside each power module casing are evenly spaced, providing uniform heat dissipation through the bottom metal-plate of the power modules. Researchers at CAPS also confirmed that the power modules could be assumed to have uniform distribution of heat. In the emulator, the copper plate used for spreading was less than an eighth of an inch thick and had dimensions of 46mm by 108mm which was the same footprint as the power modules. To secure each heat source to the heatsink for testing, screws were placed through the resistors and copper plate corners and one screw in the middle of the copper plate. To enhance thermal conductivity, a layer of thermal grease was used between the resistors and the copper plate, and another layer between the copper plate and the heatsink. The copper plate was later removed in testing due to minimal effect in the thermal results. Removing this copper plate produced a higher heat density for the emulators than the power modules, creating a small over-estimation of the heat dissipation per area.

### 3.2.2 Testing Set-Up

The purpose of testing the emulated heat source with a heatsink was to determine the heatsink's thermal resistance. This was determined by measuring the junction temperature between the baseplate and the heat source emulator using Equation 1 below.

$$\frac{T_j - T_a}{P_d} = R_h \quad \text{Equation 1}$$

In Equation 1  $T_j$  is the junction temperature,  $T_a$  is the ambient temperature,  $P_d$  is the power dissipated, and  $R_h$  is the desired heatsink thermal resistance.

The testing set-up was as follows: two heat source emulators were attached to the baseplate of a plate or pin fin heatsink spaced at least 1.5cm apart. The heatsink had a fan fixed either on the side (plate fin) or the bottom (pin fin) of the heatsink as shown in Figure 7. With the fan(s) running near their maximum fan speeds, and the proper voltages and currents supplied to the resistors, steady state temperature readings were taken in 5 locations on the heatsink baseplate. These temperature readings were then averaged, and used in conjunction with Equation 1 to find the thermal resistance of the heatsink. For a more detailed explanation of the testing protocol and experimental set-up, see the Operation Manual provided in Section 4.3.

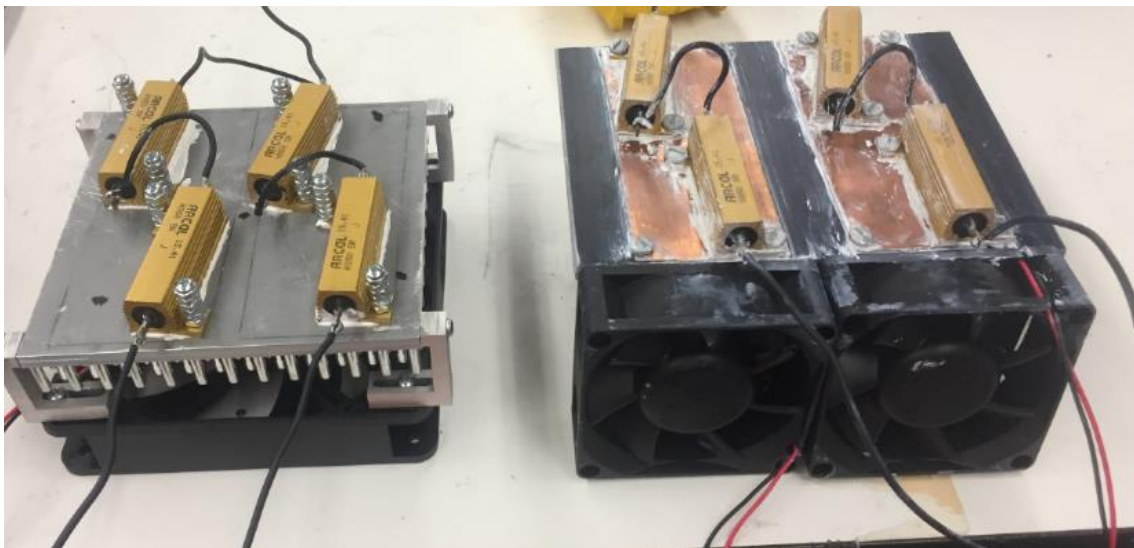


Figure 7: Testing Set-Up for Pin Fin Heatsink (left) and Plate Fin Heatsink (right)

### 3.2.3 Test Results

During testing, Team 13 wanted to test the necessity and effectiveness of the copper plate so the plates were removed and the test was re-performed. The test was only conducted on the

plate fin because the pin fin heatsink had not been acquired at that time. The results are tabulated in Tables 1 and 2. It's important to point out that the results from with and without the copper plate showed that the plate did not provide much significant impact on heating the baseplate evenly. This was determined from the similar results between the two designs for the average temperature. Because of this observation, the copper plate was deemed unnecessary and consequently not used during pin-fin heatsink testing.

As stated in the previous section, a single emulator would mimic the heat generated from a single power module. Since Team 13 decided on a bi-module design, two emulators needed to be used for each heatsink test. The total power dissipated, as listed in table 1, 2, and 3, is the summation of the power dissipated from each emulator. Therefore, to determine the power dissipated by a single emulator, the total would simply be divided by a factor of 2. This means that a range from 0 to 90W was tested for each emulator. Since testing may be time consuming, Team 13 decided not to test the emulators at their max power dissipation rating. Testing at the max power rating could cause early failure of the device and possibly endanger personnel and team members. Another reason Team 13 did not test at maximum power, is because the power supplies that were readily available in the lab were not capable of achieving voltage high enough to supply to the circuit. The test results for varied voltages on both plate and pin fin heatsinks are shown as follows.

#### Plate Fin:

*Table 1: Plate Fin Experimental Results with Copper Plate Used in Heat Emulator*

Total Power Dissipated (W)	Average Temperature (°C)	Equivalent Thermal Resistance (K/W)
0	22.36	N/A
30	28.04	0.189
60	30.86	0.142
90	33.22	0.121
120	36.4	0.112
150	42.08	0.131
180	42.5	0.112

*Table 2: Plate Fin Experimental Results without Copper Plate Used in Heat Emulator*

Total Power Dissipated (W)	Average Temperature (°C)	Equivalent Thermal Resistance (K/W)
0	22.9	N/A
30	28.16	0.175
60	29.22	0.105
90	34.6	0.130
120	37.2	0.119
150	39.6	0.111
180	41.4	0.103

Pin Fin:*Table 3: Pin Fin Experimental Results without Copper Plate Used in Heat Emulator*

Total Power Dissipated (W)	Average Temperature (°C)	Equivalent Thermal Resistance (K/W)
0	22.6	N/A
30	27.02	0.147
60	29.12	0.109
90	31.72	0.101
120	37.86	0.127
150	38.72	0.107
180	41.42	0.105

### 3.3 Thermal Simulations

#### 3.3.1 Simulation Set-Up

Thermal simulations were carried out using COMSOL Multiphysics software. The purpose was twofold: to provide verification of the theoretical analysis and to more efficiently assess

heatsink designs than with experimental testing. A heatsink tutorial created by COMSOL was an essential resource for setting up the simulations [1]. Both plate and pin fin heatsink geometries were constructed within COMSOL. Two rectangular boxes were positioned on each heatsink baseplate to represent the power modules. A box was added around each heatsink that was used to represent the inlet and outlet of the fan airflow. Material properties were added to the appropriate geometric boundaries. The heatsink material was set as Al 6063, the power module boxes were made copper in order to reflect the original testing setup with the copper plate, and the outer box/heatsink channels were designated as air.

The initial conditions of the system were added including an ambient air temperature of 23.25°C and atmospheric pressure. Boundary conditions for heat transfer and laminar flow were implemented. Both heat transfer through solids and fluids had to be specified. The power module boxes were assigned as the heat source with the power dissipated set to 120W. The correct boundaries for the air inlet and outlet were specified for each heatsink, and the assumed volumetric flow rate was defined. The mesh was then built, which the accuracy of a simulation generally depends on how refined the mesh is. The tradeoff of an increased mesh density is increased run time and file size. The computers utilized for this project did not have enough memory to compute a “fine” mesh for a simulation of this complexity. However, using a coarser mesh density generated fairly accurate results in a reasonable amount of time, approximately 30 minutes per simulation.

### 3.3.2 Simulation Results

The results of the thermal simulations for both plate and pin fin heatsinks are shown in Figure 8. The heatsink design both performed well, resulting in temperatures far below the maximum safe temperature of 120°C. The maximum baseplate temperature was approximately 38°C for plate fin and 33°C for pin fin. Plate and pin designs had very similar thermal performances; however, the pin fin weighed significantly less. The various COMSOL simulations that were completed also demonstrated that the airflow from the fan was a dominant factor in the cooling of the heatsink.



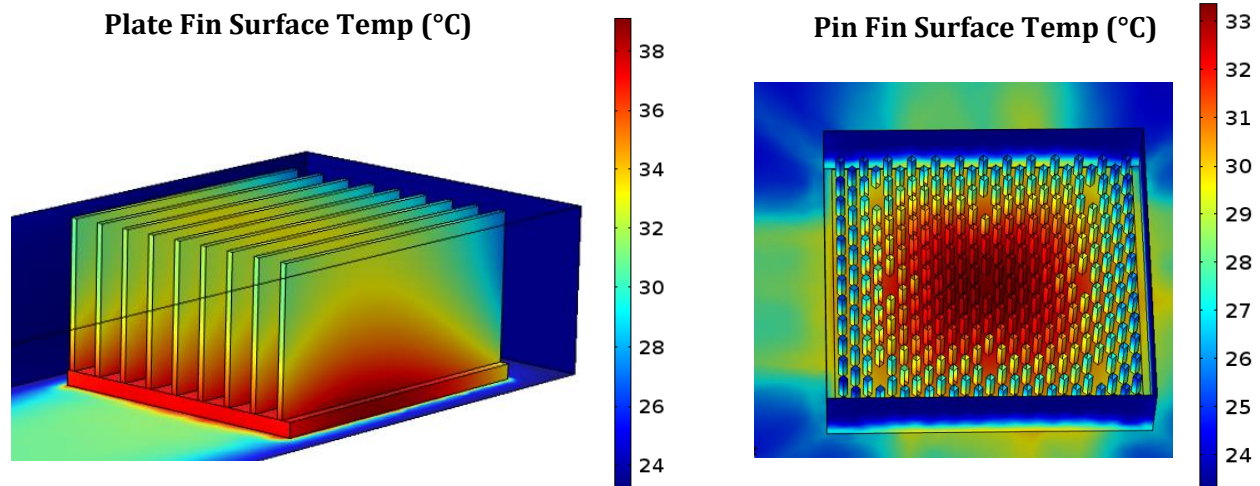


Figure 8: Plate and Pin Thermal Simulation Results

## 3.4 Theoretical Analysis

### 3.4.1 Plate Fin Analysis

Analysis of the bi-modular plate fin design was done using specific equations found in scientific literature [2] as the team was conducting research on heatsink optimization. These equations, which were solely suited for plate fin heatsinks, accounted for the pressure drop across the heatsink, which would indicate the velocity of the air moving through the heatsink, as well as the total thermal resistance of the bi-modular plate fin design. However due to the pressure drop being so low, the air speed through the heatsink was estimated to be the max rated value given from the cooling fan's spec sheet. This allowed for only the thermal resistance equation to be the primary focus. The total thermal resistance of the heatsink consisted of the conductive resistance from the baseplate, the conductive resistance from the fins, and the convective resistance from the cooling fan. The equation representing the total thermal resistance is represented in Equation 2 along with the thermal equivalent circuit shown in Figure 9.

$$R_{total} = R_{base} + \frac{1}{2 \cdot \text{Number of Fins}} (R_{fin} + R_{convective}) \quad \text{Equation 2}$$

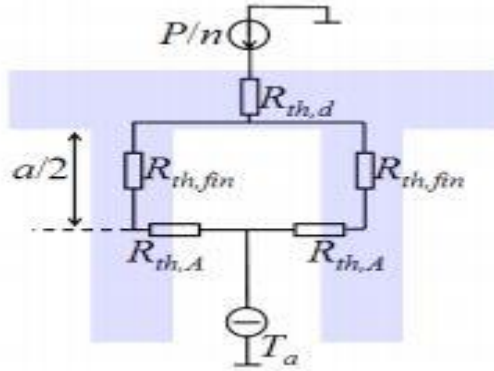


Figure 9: Thermal Equivalent Circuit

Additionally, an example of the thermal resistance computations for a plate fin heatsink is included in Appendix C. Mathcad was used for this analysis.

The weight of the Bi-Modular Plate Fin design was found by making a CAD drawing of the heatsink and using the inbuilt tools in SolidWorks to find the total mass, then combining that value with the mass of the cooling fans and screws that would be used in the design. The parameters for the bi-modular plate fin heatsink used along with the thermal resistance and weight is provided in the Table 4.

Table 4. Bi-Modular Plate Fin Parameters

<b>Bi- Modular Plate Fin Design</b>	
Length	127 mm
Width	126.5 mm
Base Thickness	6.4 mm
Height of Fin	62.8 mm
Thickness of Fin	2.5 mm
Number of Fins	9
Spacing of Fins	13 mm
Weight of Design	954 g
Calculated Thermal Resistance	0.335 K/W

### 3.4.2 Pin Fin Analysis

Though the thermal equivalent circuit of the pin fin is similar to that of the plate fin, the analyses of their thermal resistances is relatively different. This is mainly due to the different fin geometries as well as orientations of the fan(s). Team 13 was unable to find sufficient sources to benchmark the pin fin calculations while analyzing impingement air flow. The team was also unable to find Nusselt number equations for this direction of flow over the cylinder. This was a challenge since the Nusselt number heavily influences the convective heat transfer coefficient, which is a major property in determining the convective thermal resistance of the heatsink. Team 13 compromised, and proceeded to use Nusselt correlations for flow over a flat plate.

The team applied other assumptions to the analysis as well, including the following. For finding Reynold's and Nusselt Numbers, flow around only one cylinder would be analyzed. The air velocity was assumed to be uniform, and was found as the maximum volumetric flow rate of the fan divided by the channel area between the pins. For determining the Prandtl number, the temperature of air was assumed to 25°C (room temp) since new air was constantly moving through the array of pins. Additionally, once the convective thermal resistance was known, a correction factor was applied to account for the hindrance of air flow due to the surrounding array of fins. This was important because the original assumption looked only at flow over one fin. Dr. Kumar, a professor in fluids, advised Team 13 that this would account for between 20-30% of the total convective thermal resistance. A 25% correction factor was applied, and total thermal resistances were found. The calculations were extensively verified with omni-directional thermal resistances provided by manufacturers. Team 13 checked 60 different variations of heatsink geometries and fan flow rates. The error in the calculations was found to be between 10-60%. Example calculations can be found in Appendix D. These computations were applied to the pin fin heatsink that was ordered for testing. The properties of this heatsink are shown in Table 5.

Table 5. Bi-Modular Pin Fin Parameters

<b>Bi- Modular Pin Fin Design</b>	
Length	113.7mm
Width	113.7 mm
Base Thickness	4.7 mm
Height of Fin	13.1
Diameter of Fin	3.2 mm
Number of Fins	313
Spacing of Fins	9 mm inline 4.5 mm staggered
Weight of Design	553 g
Calculated Thermal Resistance	0.196 K/W

## 3.5 Comparison

### 3.5.1 Testing, Simulation, & Calculations

After calculating the thermal resistance through equations, COMSOL simulations, and experimentation all results were assembled into a table to see how each value varied. The main values chosen for comparison were the junction temperature (where the power modules interface with the heatsink) and the thermal resistance. The error in thermal resistance was found by calculating how much the COMSOL and analytical values varied from the experimental values. These comparisons, along with the error in thermal resistance, were done on both the bi-modular pin fin and the bi-modular plate fin designs. The pin fin results are shown in the following table.

Table 6: Bi-Modular Pin Fin Results

Power Output (W)	Results	Junction Temp. (°C)	Thermal Resistance (°C/W)	Thermal Resistance Error
120	Experiment	37.9	0.127	---
	COMSOL	33.4	0.084	34%
	Analytical	43.6	0.196	54%
	Total Weight	2.212 kg (65.7 % weight reduction)		

Looking the Table 6, the thermal resistance error for the analytical model is 54% and the error for the COMSOL model is 34% with respect to the test results. While these error percentages remain high, the junction temperatures of each model remained quite consistent deviating only to a maximum of about 6°C from the experimental model. This indicated that a high error in the thermal resistance did not necessarily correlate to a high change in the junction temperatures relative the experimental value. This relationship is also indicated in Table 7 for the bi-modular plate fin design shown below.

*Table 7. Bi-Modular Plate Fin Results*

Power Output (W)	Results	Junction Temp. (°C)	Thermal Resistance (°C/W)	Thermal Resistance Error
120	Experiment	37.2	0.111	---
	COMSOL	39.2	0.132	18.9%
	Analytical	65.5	0.335	242%
	Total Weight	3.816kg (40.8 % weight reduction)		

Once again, high thermal resistance error values are documented and the maximum difference in the junction temperature relative to the experimental temperature was about 28°C. This indicated that junction temperature should be looked at more closely when comparing the different models, and that the pin fin calculations produced more accurate results than the plate fin calculations.

Errors for each model need be addressed for the factors given rise to their numbers. In this case, errors in the analytical portions of each design could have more than likely been due to the forced convective equations used for each. Forced convection over and through geometrical structures can be quite complicated and certain assumptions were made in order to make the equations simpler for calculation purposes. If these assumptions were not made, it is possible that the thermal resistance could have been lowered for both designs, in conjunction decreasing the error. This can also be said for the COMSOL simulations. It is possible that certain parameters were not used or entered incorrectly thus increasing error. While some of these errors were significant, due to the low junction temperature differences and time constraints it was decided to proceed with design selection. Additionally, it should be noted that had the decision to pursue a pin fin design dictated the fact that the large error in the plate fin calculations was not addressed as thoroughly. This decision is discussed more thoroughly in the proceeding section.

### 3.5.2 Plate Fin vs. Pin Fin Heatsink

The following table has been provided to give insight on the specifications of the heatsinks tested.

*Table 8: Comparison Between Plate and Pin Fin Designs*

Heatsink Design	Plate Fin	Pin Fin
Size	127mm X 127mm X 69.2mm	113.7mm X 113.7mm X 17.8mm
Fin Height	62.8 mm	13.1 mm
Fin Thickness/Diameter	2.5 mm	3.2 mm
Fin Spacing	13.2 mm	Inline: 9 mm
		Staggered: 4.5 mm
# Fins	9	313
Fin Surface Area/Volume	0.848 mm <sup>-1</sup>	1.40mm <sup>-1</sup>
Weight w/ Fans	0.954 kg (X4)	0.553 kg (X4)
Fan Speed	1.73 m <sup>3</sup> /min (X2)	3.03 m <sup>3</sup> /min
Fan Orientation	Lateral	Axial
Temperature	41°C	36°C
Thermal Resistance	0.153 K/W	0.112 K/W

Looking at both designs, it was clear that they would both be able to keep the power modules well below their failing temperature of 120°C. However since the priority of this project is to reduce the overall weight of the thermal system, the bi-modular pin fin design was ultimately chosen due to its ability to dissipate the appropriate amount of heat along with having a smaller weight and needing less fans. The pin fin design was 65.7% lighter than the CAPS heatsink and 42% lighter than the plate fin design. The decision to pursue a pin fin heatsink was also derived from the fact that cylindrical fins are known to help maximize the fin surface area per volume. Having a high surface area is important because the thermal resistance is inversely proportional to it. The lower the thermal resistance, the quicker the heat is dissipated. Minimizing the volume is also desired for this project, since volume is directly proportional to weight. Team 13 computed the fin surface area per volume for fins from both plate and pin fin designs. For the plate fin design

the ratio of total surface area to fin volume was  $0.848\text{mm}^{-1}$ . For the pin fin, this value was  $1.40\text{mm}^{-1}$ . This ratio was heavily influential in the decision to pursue a pin fin design for optimization.

## 3.6 Optimization

### 3.6.1 Assumptions

To help further increase the power density of the thermal management system, the pin fin heatsink design was chosen for optimization. The main goal of the optimization process was to minimize the heatsink weight. A point of reference for optimization was the pin fin heatsink obtained from a manufacturer for experimental testing. Team 13 determined that the optimized design needed to weigh less than the manufacturer's pin fin, 0.254kg. For the optimized heatsink, the baseplate size was kept constant at 115mm x 115mm. This was determined as the minimum size possible while having adequate room to properly fit and space two power modules, which are each 108mm x 46mm. When threaded holes were drilled into the manufacturer's baseplate, the holes had just enough depth to allow the heat source emulators to be properly secured. Because there was not much extra clearance, the baseplate thickness was kept at 4.7mm.

The main cost of a decrease in heatsink weight was determined to be an increase in thermal resistance. A higher thermal resistance would indicate less heat was transferred through the heatsink, which could lead to the overheating and therefore failure of the power modules. In order to ensure safe operating temperatures for the power modules in the range of 30-60°C, a goal value for the thermal resistance was determined to be 0.3 K/W or less. Based upon research completed at CAPS, the combined power loss from the two power modules was known to be 105.2 W or less. The power loss was assumed to be equivalent to the amount of heat applied to the heatsink. The air flow from the fan was also assumed to uniformly cover the entire heatsink area.

### 3.6.2 Procedure

To carry out the optimization, the geometry of the heatsink and the flow rate of the fan were varied. The following geometric parameters were varied: length of the pin fins from 5-40mm, the pin diameter from 2-5mm, and the number of pins from 100-300, initially. The pin spacing directly depended on the pin diameter and the number of pins. If the pins are too close together,

the impingement of the flow will be increased, which is a detriment to convective heat transfer. The fan flow rate was also varied from 0.02-0.05 m<sup>3</sup>/s.

One input parameter was changed at a time and plotted against either the thermal resistance or weight. A second parameter was then iterated and plotted on the same graph. This proved to be the best way gain an understanding of how each alteration affected the system without having an overwhelming amount of information at one time. The general trend was that as a geometric parameter was decreased, therefore decreasing the weight, and there was an increase in thermal resistance. Increasing fan speed led to a decrease in thermal resistance. Certain parameters had more dominant effects on the outcome. A change in length had much less of an impact on the thermal resistance than did a change in diameter or number of pins, which both have a greater effect on pin spacing and, therefore, airflow through the heatsink.

### 3.6.3 Analysis of Results

As the optimization results were analyzed, several key decisions were made in order to narrow the possibilities for an optimized design. After observing the results for different numbers of pins, it was determined that the design must have more than 200 pins to obtain the necessary thermal resistance. Similarly, the minimum acceptable diameter size was determined to be 3.0 mm because the cost to the thermal resistance was too great to further reduce the diameter. To streamline the optimization process, Team 13 decided to use an inline pin arrangement in which all the pins were evenly spaced across the baseplate. To have uniformly separated pins on a square baseplate, the total number of pins needed to have an integer square root value. Looking in the range of 200 to 300 pins, the options were reduced to 225 pins (15x15 array), 256 pins (16x16 array), and 289 pins (17x17 array).

A representation of the optimization results is shown in Figure 10 in which the diameter is varied from 2 to 5mm and each line denotes a different number of pins. As expected, the thermal resistance decreased with increasing diameter and number of pins. At the minimum diameter of 3.0mm, each of the pin values was found to be in the acceptable thermal resistance range of 0.3K/W or less. Therefore, the number of pins was chosen to be 225 pins in order to most reduce the heatsink weight. After varying the pin length from 5 to 40mm, a length of 10mm was selected as best option to decrease the weight and maintain the correct thermal resistance. The fan flow rate



was also investigated to gain an idea of how low the speed could be while still properly cooling the heatsink. The fan needs to operate at no less than  $0.04\text{m}^3/\text{s}$  to obtain the goal thermal resistance.

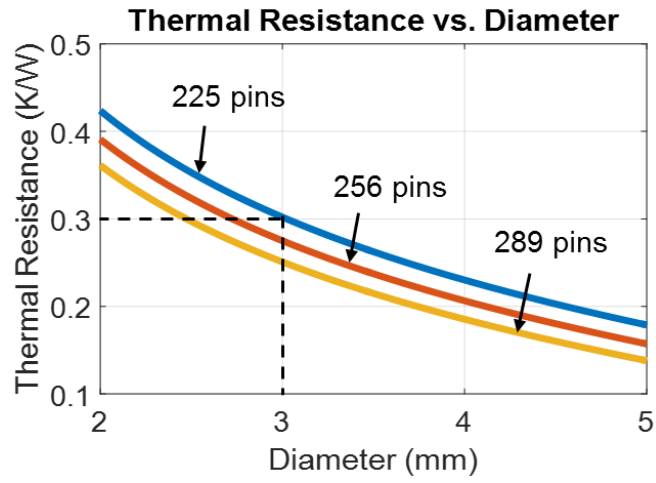


Figure 10: Pin Fin Optimization Plot

## 4. Final Design

### 4.1 Overview

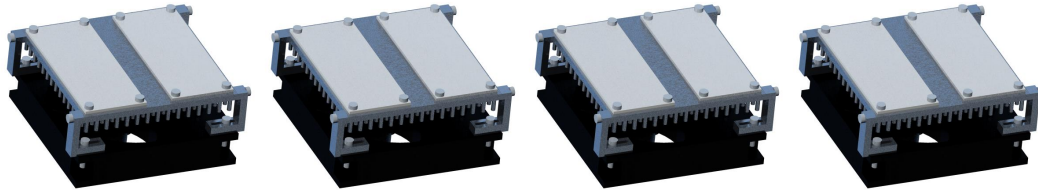


Figure 11: Final Optimized Thermal Cooling System (1.72 kg total)

#### 4.1.1 Optimized Design Specifications

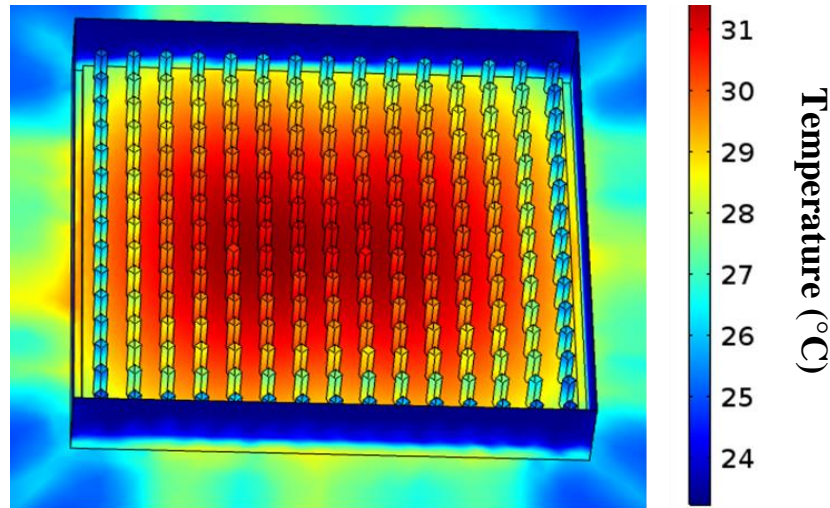
The optimized design had a total of 225 evenly spaced pin fins. The center to center distance between each of the adjacent pins was determined to be 4.71mm. The selected pin diameter was 3.0mm, and the length of the pins was 10.0mm. The heatsink material was chosen to be aluminum 6063-T5. The weight of the optimized design came to 211g. A new fan was also selected using the old fan available in the CAPS laboratory as a benchmark. The new fan was nearly half the weight of the old fan at 157g. The size was 120mm x 120mm x 25mm, which was the correct size for affixing the fan to the heatsink. The fan was found to achieve a flow rate of 0.051 m<sup>3</sup>/s, which exceeded the required flow rate determined through optimization. The voltage rating of the fan was 12V, and the power was 5.3W. The combination of the optimized heatsink and new fan delivered a weight reduction of about 34% when compared to that of the manufacturer's pin fin and the old fan. Taking the total system into account, the four optimized heatsink setups reduced the weight of the original CAPS heatsink by approximately 71%. The power density was increased from 2.5 kW/kg to 6.54 kW/kg.

#### 4.1.2 Simulation Results

The optimized heatsink design was also verified using COMSOL Multiphysics. The simulation, shown in Figure 12, was conducted assuming an ambient air temperature of 23.25°C, a flow rate of 0.051 m<sup>3</sup>/s, and a power loss of 105.2W. The result was a maximum surface temperature of 31.6°C, which was well within the desired temperature range. The thermal resistance was then calculated to be 0.0794 K/W. These results indicated the design performed

even better than expected due to errors in the pin fin calculations, which slightly overestimated the thermal resistance.

### Pin Fin Temperature Distribution



*Figure 12: Thermal Simulation for Optimized Design*

## 4.2 Design for Manufacturing

### 4.2.1 Heat Source Emulator

The heat source emulators were built using two different methods, one included copper plates and the other method did not. Both methods of building the heat source emulators included the use of two 50-watt, 5-ohm power resistors, three connection wires, and a solder iron. This is the simplest way that Team 13 could imitate the original power converters being used by the CAPS researchers.

To build the heat source emulators a connection wire was soldered to each end of the power resistors and one connection wire was soldered in between the two power resistors making the connection. This concludes the manufacturing of the heat source emulator using method two, the first method continues with attaching the power resistors on top of a 108 mm by 46 mm copper plate using a generous amount of thermal grease.

The assembly of the heat source emulators took less time than anticipated, the most time-consuming part was waiting for the thermal grease to set. The thermal grease does not dry but Team 13 let it sit overnight so that the power resistors were not sliding all over the copper plate.

To connect the heat source emulators to the heatsinks the same method was used with the thermal grease. For the emulators with copper plate, a generous coat of thermal grease was applied onto the copper plate and that plate was laid onto the heatsink. The emulators were then also secured with six screws, four on each corner of the copper plate and two in the middle of the copper plate. The emulators without the copper plate were attached to the heatsink the same way. Thermal grease was applied to the bottoms of the power resistors and then two screws on each power resistor further secured the heat source emulators to the heatsinks.



*Figure 13: Method 1 (with copper plate)*



*Figure 14: Method 2 (without copper plate)*

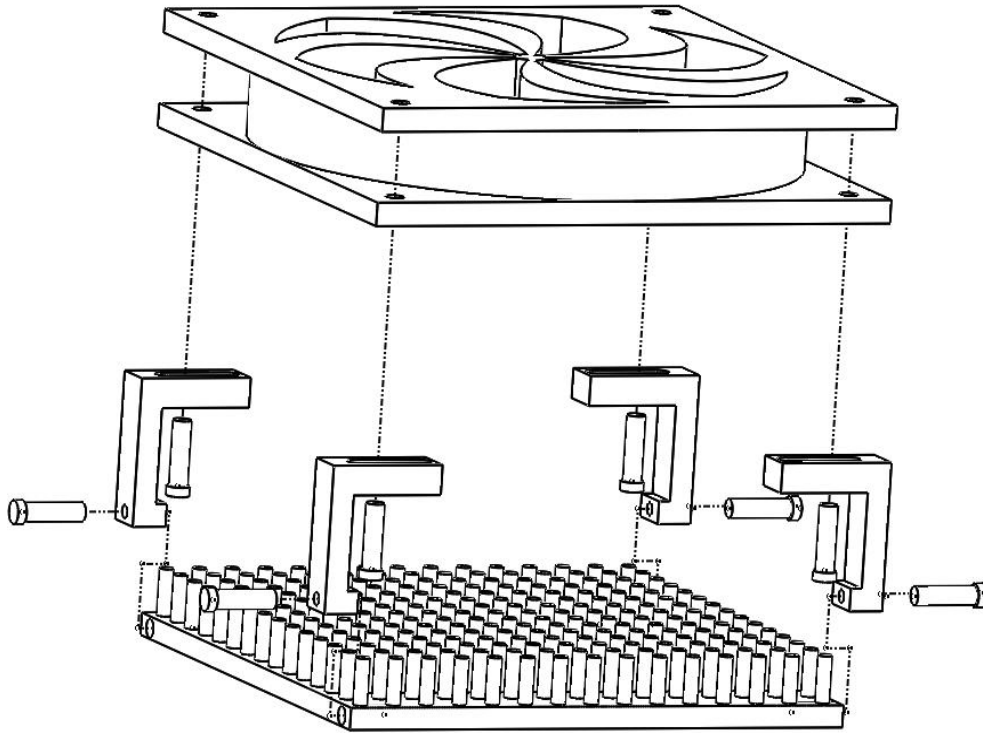
## 4.2.2 Heatsink Design

The final design and build for each individual heatsink system consisted of one pin fin heatsink, four connector pieces, eight screws and one cooling fan. 4 additional screw would be added per heatsink to attach the power modules. A total of four heatsink systems would need to be built to accommodate all eight of the power modules. The completed list of parts includes four heatsinks, 16 connector pieces, 48 4-40 screws, 16 4-40 nuts and four cooling fans.

Looking at the heatsink, to lessen complications during manufacturing it was decided that each heatsink would be made from Aluminum 6063 and ordered from a heatsink supplier where personal customization is possible. This allows for the optimized heatsink design to be fully manufactured with the correct geometrical sizing needed. The method that the supplier will use to manufacture each of the heatsinks would be cold forging. Cold forging is the preferred forging method when working with soft metals, such as aluminum, in order to deform the material into predetermined complex shapes. This method allows for better tolerance, small impurity content, improved surface finish, and lower cost as compared to other methods. Once ordered, a total of four holes will be machined into each of the heatsink's baseplates for screws to be inserted. The L-Bracket connector pieces will be used as mounts that will secure the cooling fan firmly to the heatsink while it is in operation. Their drawing, as well as the heatsink drawing is included in Appendix E. The connector pieces will be made from aluminum and will be water jetted. Both the screws and cooling fans will be bought from a supplier as well. The fan to be used is specified in Appendix F, and the screws and nuts can be ordered from Home Depot. A complete parts list is also included in Appendix H.

Assembly for the each of the four heatsink systems is simple and straight forward. The four connector pieces will first be screwed into the side of the baseplate of each heatsink to serve as four attachment points for the cooling fan. Once the connector pieces are screwed and tightened into the baseplate, the cooling fan is then mounted to the top side of each connector piece. This will be done by inserting four screws into the premade holes of the cooling fan and aligning them with the holes of the connector pieces. Once the screws are inserted in conjunction with the holes of the cooling fan and connector pieces, it will then be tightened to form a rigid and stiff connection. This assembly process will ensure that the fan is perfectly aligned with the center of the heatsink and that none of the parts will shift during operation. The time that it takes to assemble

each heatsink system is estimated to be at most 15 minutes. The exploded view for the assembly process can be seen in Figure 15.



*Figure 15: Exploded View of Heatsink System*

Additionally, when the thermal system is used with the power modules, the modules will be screwed into the top of the baseplate. They must have a separation of at least 15 mm between them, but for this design they can be up to 19 mm apart.

### 4.3 Design for Reliability

For this design to be feasible, each heatsink system needs to be able to be assembled multiple times along its lifetime to ensure proper maintenance and function throughout multiple uses. In practice these heatsink systems will rarely need to be disassembled, however making sure the heatsink can dissipate the required amount of heat for long periods of time is of prime importance to ensure reliability.

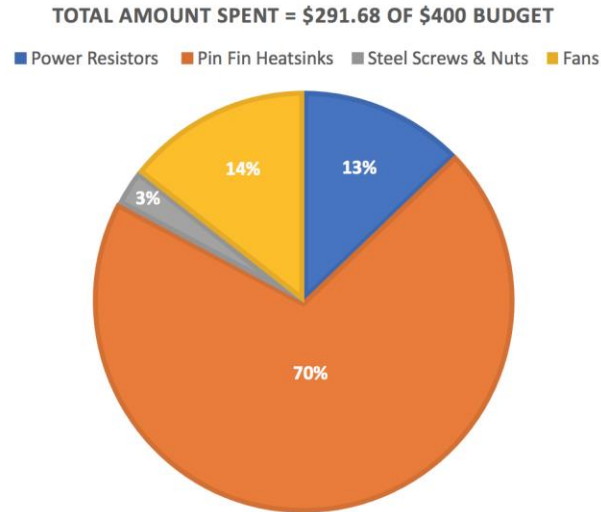
The main factor that would cause the system to fail was identified to primarily be from the cooling fan either not working properly or failing to work at all. Without the proper cooling of air

flowing through the heatsink, the power modules are bound to reach high temperatures within minutes, resulting in them overheating. To mitigate this potential failure, cooling fans were selected with values that ensured long lifetimes from continuous operation. The cooling fan selected was from the manufacturer Sunon-Fans with an estimated life expectancy of 70,000hrs at 40°C. Even though the fan is not directly touching the heatsink baseplate, it is still beneficial to consider the baseplate temperature to ensure that the surrounding space will fall within the 40°C range. From previous testing of the non-optimized heatsink, the baseplate had an average temperature of 37.9°C when dissipating 120 W. This temperature range ensure that the surrounding space of the heatsink system will remain under 40°C allowing the cooling fan to run its full lifetime of 70000hrs so long as the fans are placed in a position where they can draw in cooler air. It is recommended that each of the 4 heatsinks should be set at least a distance of 1 inch apart from one another to be sure that the ambient temperatures remain low.

## 4.4 Design for Economics

Team 13 was given a specified budget of \$400, which covered the price of designing an optimized heatsink for a PV converter, a selected manufactured heatsink to use for testing, and the materials needed to build multiple heat source emulators. This budget was used primarily for testing purposes, and does not indicate the total cost of the thermal management system.

The total cost of Designing and Testing a Thermal Management System for a SiC PV Converters thus far is \$291.68. This price includes multiple heatsinks, fans, power resistors, screws, and nuts. As can be observed in Figure 16, the heatsinks make up a majority of the cost at \$204. The connectors, copper, thermal grease, wiring, plate fin heatsinks, the three original fans used for testing, and the two power supplies were not a burden on the team's budgeting.



*Figure 16: Percent of Money Spent from Budget on Each Component for Project*

Team 13 purchased 3 pin fin heatsinks, 2 fans, 10 power resistors, 100 screws, and 100 nuts. Team 13's sponsor, Dr. Li, provided the team with the 2 plate fin heatsinks that were tested, the copper used under the emulators, the 3 original fans that were used for testing, the power supplies that were used for the tests, and the wiring and thermal grease that were needed to build the heat source emulators. The connectors that Team 13 used to connect the fans and the heatsinks were built by the FAMU-FSU College of Engineering Machine Shop.

The overall cost of the thermal management system for the converter, however, is not equivalent to the cost of this project. The thermal management system does not require any testing equipment or the same number of heatsinks that were purchased. Team 13 found a cylindrical pin fin heatsink with similar properties to the one that was optimized, and it will likely be the one that CAPS uses for their converter as its less expensive as getting one custom designed. This heatsink is manufactured by Cool Innovations and has measurements of 113.7 mm X 113.7 mm X 17.8 mm, with a thermal resistance of 2.2 °C/W. The heatsink costs \$50 plus \$27 for shipping and the fan cost is \$15 each plus \$10 for shipping. Assuming all 16 connector pieces take 30 minutes to manufacture in a shop, the connector brackets cost about \$30 to produce in total, or \$7.5 per assembly. The 100 screws and 100 nuts cost \$8.40 together as well. Taking all of this into consideration, all 4 heatsink and fan assemblies would cost CAPS about \$335.40 for parts, or \$83.85 per assembly. The cost distribution per assembly is included in Figure 17.



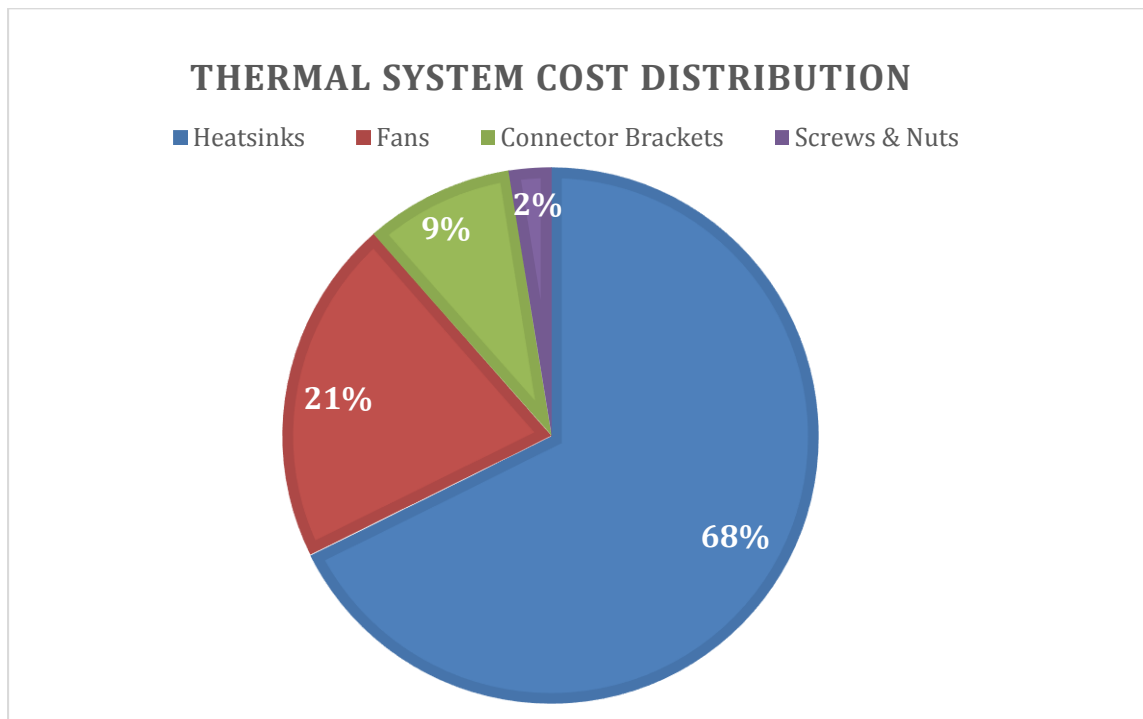


Figure 17: Pie Chart Showing Cost Distribution for PV Converter Thermal System

## 4.5 Operation Manual

### 4.5.1 Functional Analysis

The primary objective of this project was to design a lightweight heatsink for the power modules contained in a SiC PV converter. The original heatsink that was used for the converter was overdesigned. It was a plate fin heatsink with a total of 8 fans fixed on the sides and it housed 8 power modules on its baseplate. The thermal cooling system weighs a total of 6.45 kg, and was “cold to the touch” during operation. Team 13’s solution had to meet the following criteria.

- Prevent 8 power modules from exceeding  $120^{\circ}\text{C}$  (30 degrees below failure point) at steady state (while PV Converter produces 100 kW)
- Reduce the size and weight of the current design (6.45kg)
- Have a maximum thermal resistance of 0.792 K/W

Team 13 considered two different heatsink designs: a plate fin heatsink and a cylindrical pin fin heatsink. After conducting theoretical analyses, simulations using COMSOL, and experimental testing the group decided to pursue a thermal management system that consisted of 4 pin fin heatsinks with fans fixed to the bottom instead of on the sides. This configuration is known as impingement cooling, and provides more uniform cooling throughout. Each heatsink would house 2 power modules.

In addition to designing a thermal management system, Team 13 was tasked with the design, fabrication, and usage of a testing mechanism to verify and compare the capabilities of both plate and pin fin heatsink types. Power modules could not be used in order to protect against damage from an inadequate heatsink design. Heat source emulators were developed to mimic the SiC power modules, and were made from high power resistors connected in series. The resistors were chosen based on dimensional limitations, as well as based on the available power supply. The DC power supply had to provide the required amount of power needed to generate the amount of heat desired. Based on the available resources, Team 13 was able to come up with a solution, utilizing high power resistors with low resistances to simulate the emulators. A functional diagram of the thermal management system being tested with the heat source emulators is included in Figure 18. Figure 19 has the functional diagram for the heatsink being used inside the PV converter.

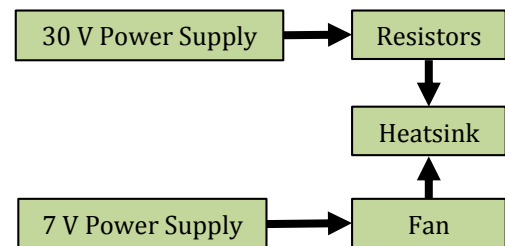


Figure 18: Functional Diagram of Heat Source Emulators with Active Heatsink

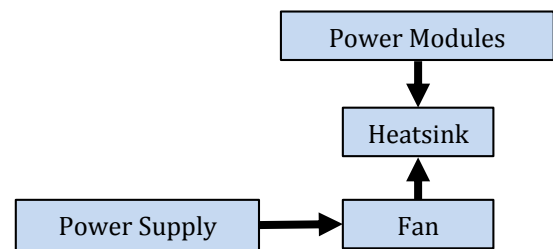


Figure 19: Functional Diagram of Power Modules with Active Heatsink

## 4.5.2 Product Specifications

Team 13 designed the heatsink shown in Figure 20. It's a simple pin fin heatsink made from Aluminum 6063 with a 15 x 15 array of 3mm diameter pins that are 10mm long. The base is 115mm x 115mm x 4.7 mm. This size allows for a 1.5 gap of separation between the two power modules (which is their minimum separation distance) and is just thick enough for 4-40 screws to

be used for connecting the fan, utilizing L shaped brackets that the team developed. The heatsink weighs 211 g.

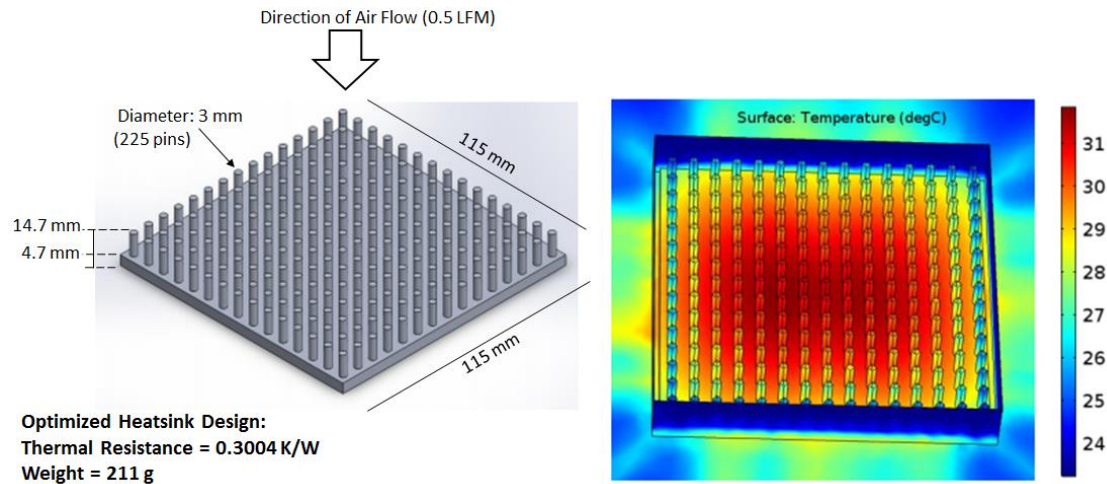


Figure 20: Optimized heatsink design (left) with COMSOL simulation (right)

A 120mm x 120mm x 25mm fan weighing 157 g is connected to the bottom of the heatsink, providing uniform cooling throughout the array. The fan has a volumetric flow rate of 0.5 LFM and is rated at 12V. A total of 16 screws, 4 L brackets, and 4 nuts are used in the assembly, weighing a total of 62 g.

At their maximum capacity, each power module has losses of 52.6W. This means that the max heat dissipation for each heatsink is 105.2 W. The max temperature that the baseplate will reach is 31.6° C. The weight of the entire optimized system is 1.72 kg (0.43 kg / heatsink & assembly). With this solution, the thermal management system weight was reduced by 77%, which results in a power density increase to 6.54 kW/kg.

Each SiC power module had dimensions known to be 108mm x 46mm x 2mm, with its max loss assumed to be 100 W. Due to this size constraint, only two resistors could be used to emulate a power module. For each emulator, two high power resistors were connected in series. Each was capable of handling 50W. The team had access to a 30V power supply, and using Ohm's law, they determined a total resistance of 10  $\Omega$  was needed per emulator. Thus, each resistor needed to be 5  $\Omega$ . Figure 21 shows the emulator configuration using Multisim.

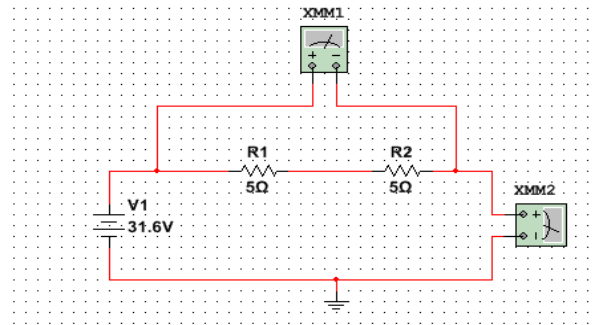


Figure 21: Heat Source Emulator Configuration

### 4.5.3 Product Assembly

The overall assembly of the heatsink and fan system is shown in Figure 22. The fan is connected to 4 L-brackets with screws and nuts. The top of the L-brackets are connected to the sides of the heatsink. When this assembly is used with the heat source emulators, the 2 sets of resistors are simply screwed into the top of the baseplate. As previously mentioned, the resistors are connected in series. It should be noted that during experimental testing, the team had screws that were sized for the fan attachments but not for the resistors which meant they were too long. These screws didn't supply the downward force needed to press the resistors firmly against the top of baseplate, which was important for heat transfer between the materials. In order to compensate for their extra length, the team used readily available nuts to fill in the gaps between the tops of the screws and resistors. This was an immediate solution that didn't require extra funds or time. It is not a permanent solution for the power modules, but was sufficient for testing purposes. Furthermore, the attachment of the power modules is similar to that attachment of the heat source emulators. The 2 power modules are screwed in to the top of the baseplate with a layer of thermal grease in between. A schematic of the setup of this assembly can be seen in Figure 23.

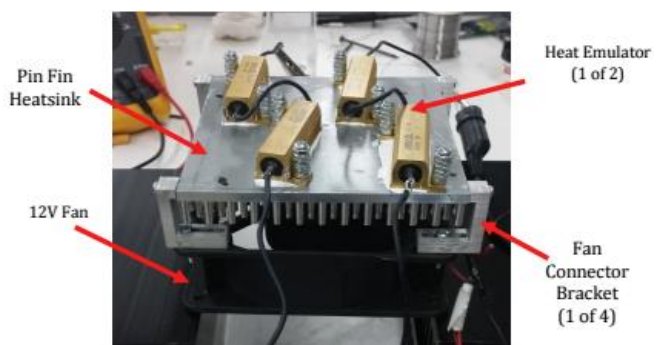
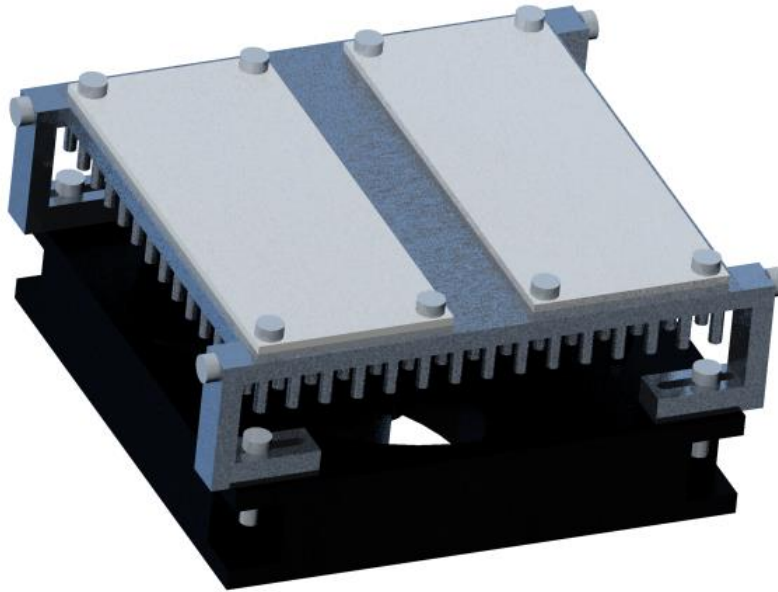


Figure 22: Test Setup with Heat Source Emulator



*Figure 23: CAD Model of Power Module Heatsink Assembly*

#### Heatsink & Fan Assembly:

1. Orient the fan on a flat surface facing upwards (fan specs label facing up).
2. Tightly screw all 4 L-Brackets into the holes in the sides of the heatsink, orienting them as shown Figure 5.
3. Place a screw facing downwards inside the slot at the bottom of each connector bracket.
4. Carefully place the heatsink and bracket assembly on the fan, aligning the loose screws with the holes on the outer edges of the fan.
5. Use the nuts provided to secure the screws, fastening the bottom of the connector bracket to the fan.

#### Emulator Construction:

The equipment, tools, and materials needed to construct the emulator is as follows:

- Four 50W, 5 $\Omega$  resistors → this makes two emulators
- Any wire gauge capable of handling up to 5A of current (about 5 ft)
- Wire cutters/strippers
- Soldering iron & solder

- Thermal compound
- 8 screws

Step-by-step instructions:

1. Cut 4 wire sections of about 3 – 4 inches in length.
2. Cut 2 wire sections of about 2 inches in length.
3. Solder the shorter wire to the end of one resistor and the end of another resistor. Do this again for the second emulator. (1x shorter wire per emulator)
4. Solder the longer wire to the other end of each resistor. Do this again for the second emulator. (2x longer wire per emulator)
5. Liberally apply thermal compound to the bottom of each resistor.
6. Use 4 screws to secure the emulator to the baseplate of the heatsink. Do this again for the second emulator. The finished emulator should look similar as shown in Figure 24.

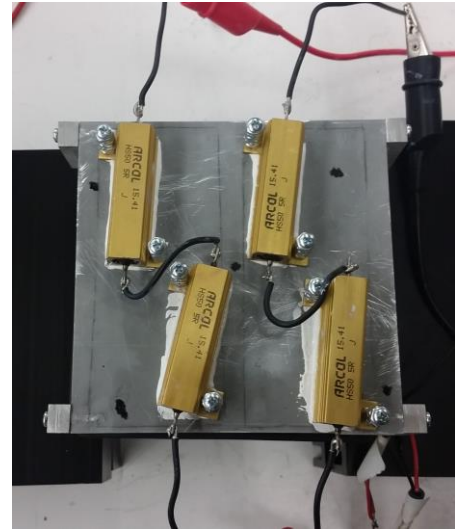


Figure 24: 2 Heat Source Emulators Side by Side

Once the emulators have been built, they must be tested to make sure they are in working order. Two methods should be used to verify this. The first method is to measure the resistance found across each resistor and the entire emulator to verify that they are in fact connected in series. The second test is to measure the connection of the emulator through a ‘diode’ or ‘connection’ test using a multimeter. Once both these methods have been verified, it can be safely assumed that the emulator is in full working condition.

#### 4.5.4 Operation Instructions

With the heatsink assembled and the emulators constructed and mounted to the baseplate, they can be tested with a heatsink. As mentioned before, an emulator refers to two resistors in series capable of handling up to 50W each. This gives a total power dissipation of no more than 100W. For using the emulators, at no time should the power exceed 100W.

For testing a heatsink, it is important to determine its thermal conductivity properties, namely the thermal resistance. Although this can be calculated, it can also be observed through

testing. Since the emulators are simply resistive components that will not exceed 100W, these resistors are capable of handling more heat than what will be generated during the testing. This makes them safer components to use and test on a heatsink without the risk of damaging equipment, if proper precautions are taken.

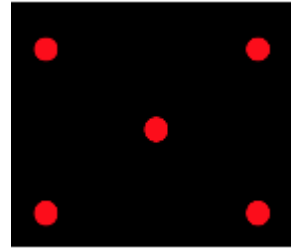
#### Emulator Operation:

Note: Due to the large amount of power required for each emulator (up to 100W) it may be necessary to use two power supplies or a single power supply with two independent channels.

Warning: Do not touch the power supply or heatsink with bare hands while powered up and in operation. Improper procedure can result in electric shock and/or burns.

1. Turn on the fan by connecting the positive and negative ends to the positive and negative ends of the 7V power supply, respectively.
2. Proceed to test the emulators.
3. Make sure the 30V power supply is off and turned down to 0 V.
4. Connect the power supply to the emulators by attaching the positive to one end of the emulator and the negative end to the other end. Do this for both emulators.
5. Turn on the power supply.
6. Starting with 0V, use an infrared temperature reader to measure the temperature of the baseplate in the 5 locations shown in Figure 8. Record the temperatures.
7. Increase the power output of the power supply by 15W. Be sure to keep the power supply the same for both emulators. Wait for steady state temperatures to be reached (about 5 minutes).
8. Using the infrared gun, measure the temperature of the baseplate in the 5 locations shown in Figure 25. Record temperatures. These will be averaged later.

9. Repeat steps 5-6, ranging the power from 0W-90W with increments of 15W. (The power supply team 13 used could not exceed 90W.)
10. After the final temperatures have been recorded, turn both power supplies down to 0V, and then turn it off.
11. Allow the heatsink to cool (2 hours).
12. Disconnect the power supply from the emulators and fans.



*Figure 25: Locations to Measure Temperatures*

### 4.5.5 Troubleshooting

Due to the nature of this project, only a few complications can arise during testing. The first complication would be from overheating if a fan is not used in conjunction with the heatsink while testing with the emulators. The emulators cannot endure temperatures exceeding 200C and if a fan is not used, the heatsink will not be able to dissipate the needed heat quickly enough to avoid overheating. If overheating does occur, the integrity of the emulators may be compromised. Using the test methods for operational use of the emulator as outlined in the product assembly section, one can verify the condition of the emulator. If the connection is broken, either the cabling or the resistors or both must be replaced, depending on the results shown from the multimeter connection test. One must also consider the situation where the fan fails during operation. This will likely be a product of a using the fan at max speed for prolonged periods of time which causes faster wear-and-tear. In an internal fault with the manufactured fan occurs, a new fan would need to be ordered. When the heatsinks are used in conjunction with the power modules, care must be taken to ensure that the fans have a reliable power source and do not exceed the rated voltage.

Another complication to consider for testing is if the emulators are in working condition but are not producing heat. This is most likely due to the power supply not supplying the required amount of power. It is important to know that if the power supply is at or near its maximum capacity, the power supply may have an overvoltage or overcurrent protection. If a test has triggered one of these protection schemes, the power supply voltage must be returned to zero, turned off, and unplugged from the power outlet for 5 minutes. This will essentially ‘reset’ the power supply and allow for use again. Of course, each power supply maker has different protection



schemes and thus, consulting the operation manual of the power supply will yield the best troubleshooting methods.

Prior to testing and use of the thermal management system and the heat source emulators, connection in wires should be checked. If a connection between the power supply and fans is severed via the connector falling off, the power should be turned off immediately to the entire system, including the emulators.

### 4.5.6 Regular Maintenance

There is not much regular maintenance that must take place for the thermal management system. The heatsink and brackets are not prone to rust because they are made from aluminum. The same goes for the screws, as they are made of zinc. Aluminum can be corrosive, so occasional visual check-ups can occur to ensure they're in good condition. Upon being used in conjunction with the power modules, care should be taken to ensure that the fans are working well and not being run at their peak capacity all the time. The fans should only be used up to their expected life cycle as well.

With regards to the heat source emulators, it is important to make sure there is plenty of thermal grease used between them and the heatsink. When they are being stored, they should be wiped clean and kept in a closed off environment away from dust.

### 4.5.7 Spare Parts

#### Heatsink:

To avoid operation interruption, it is recommended to have 1 extra fan in inventory. The following table is list of parts provided in the package.

*Table 9: Parts List for Thermal Cooling System*

Part Name	QTY
Pin Fin Heatsink Body	4
12 V DC Fan	4
L-Bracket Connector	16
4-40 screws	48
4-40 Nuts	16

Emulators:

1 testing unit is provided. To avoid operation disruption, it is recommended to have 1 extra resistor.

*Table 10: Parts List for Heat Source Emulator*

Part Name	QTY
5 $\Omega$ , 50W Resistor	4
5A Wire Gauge	1
Thermal Grease	1

# 5.Heatsink Selection Guide

## 5.1 Introduction

Selecting an affordable heatsink for power electronics, and for electronic packaging can be overwhelming with the vast array of options that are available. Several factors influence the performance of a heatsink, and it's important to understand how these parameters effect the thermal performance of the cooling mechanism. Heatsinks come in all different shapes and sizes, with a wide array of pin types and geometries. This guide's purpose is to delve into the basics of heatsinks, providing insight on several different types so that one may make an educated decision on selecting heatsink for their appropriate application. For more in depth explanations, the references of this guide are beneficial.

## 5.2 Terms and Properties

Conduction: This refers to heat transfer within an object or between adjacent objects due to a temperature gradient.

Convection: Convection occurs between a solid surface and a moving fluid (i.e. air) when they are at different temperatures.

Radiation: Radiation refers to heat transfer through electromagnetic waves between different object with finite temperature. However, in cooling applications with forced flow, and temperatures <100 degrees C, radiation only transfers a small portion of heat, and is typically neglected. In natural convection cases, radiation can play a significant role in the thermal properties, however, which is why surface color is sometimes considered.

Conductivity: The thermal conductivity of a material is a fixed parameter. It is defined as the rate at which heat passes through a specified material, expressed as the amount of heat that flows per unit time through a unit area with a temperature gradient of one degree per unit distance. It has units of Watts / (meter Kelvin).

Thermal Resistance: The thermal resistance (R) of a heatsink is arguably the most important parameter that one must look at when selecting a heatsink. It is a measurement of the temperature

difference by which an object or material resists heat flow. It has units of Kelvin / Watt. When the base and fluid temperatures are known as well as the rate of heat loss (Q) from the heat source, the thermal resistance is given by Equation 3.

$$R = \frac{\Delta T}{Q} \quad \text{Equation 3}$$

Fin efficiency: The fin efficiency is the ratio of the actual heat transfer through the fin base divided by the max possible heat transfer rate through the fin base (which can be obtained if the entire fin is at the base temperature). This quantity give insight into how long a fin should be [3]. This is typically used in heatsink design, but not in selection.

Fin effectiveness: This is the ratio of heat transfer from the fin to the heat transfer if the fin did not exist. It's a unitless quantity that tells one how much extra heat is being transferred by the fin. The desire is to have this ratio as large as possible while keeping the cost of the added fins as low as possible [3]. This value is also used more in heatsink design than in heatsink selection.

Surface Color: In heatsinks that use natural convection, the heatsink will typically have surface coloring to enhance the radiative heat transfer. Matte-Black surfaces radiate much better than shiny metal ones [7].

## 5.3 Selecting a Material: Copper vs. Aluminum

One important parameter for characterizing the performance of a heatsink is the material that it's made from. Heatsinks are traditionally made from either Aluminum or Copper alloys. These materials are typically used because they have high thermal conductivities. The thermal conductivity of a material has units of Watts per meter-Kelvin. The higher the thermal conductivity, the better the material is at dissipating heat. Copper has a higher thermal conductivity than aluminum, so in cases where it is important to dissipate the heat as quickly as possible, copper may be preferred. It also has higher spreading capability then Aluminum, so it can be preferred for cooling devices that are very small with a concentrated source of heat. However, copper is four to six times more expensive than Aluminum, and is roughly 3 times the weight of aluminum [5, 6]. So, in situations where expense or weight are significant factors,

aluminum is typically the preferred material so long as it can dissipate the heat at a sufficient rate [4].

## 5.4 Passive vs. Active Cooling

Typically, heatsinks can be considered either ‘active’ or ‘passive’. There are several advantages and to using both types. An active heatsink refers to a heatsink that is used in conjunction with a fan while a passive heatsink relies on natural convection which is the natural movement of air in the surrounding medium (with no fan). Active heatsinks cool much better than passive ones. They allow for minimal heatsink size and weight to be achieved. They also achieve significantly smaller thermal resistances. The thermal resistance of the cooling system typically reduces with increasing fan velocity. However, upon designing a heatsink, one must correlate the fan velocity with the fin density. For heatsinks with high approaching fan speed, that is 400 LFM (linear feet per minute) or greater, densely configured heatsinks are appropriate. Meanwhile for moderate speeds (200-400 LFM) heatsinks should have fin density that balances surface area and fin spacing. Meanwhile for low speed air flows (0-200 LFM) fins should be spaced sparsely [5].

Passive cooling, can also be a sufficient cooling system. This type of system is beneficial because it is not prone to failure. Active heatsinks can quickly reach critical temperatures if the fan fails, but if there’s no fan, this is not a worry. Passive heatsinks are larger than active ones, and are often less expensive. Sometimes large heatsinks are designed to be passive simply because the system won’t require so many fans for cooling. Fans also require voltage sources, which can sometimes be undesirable [5]. In systems where weight and size are not a significant constraint, passive cooling systems are often used. Most electronics use active systems, or combined systems that only utilize the fan when certain temperatures are reached.

## 5.5 Fin Arrangement

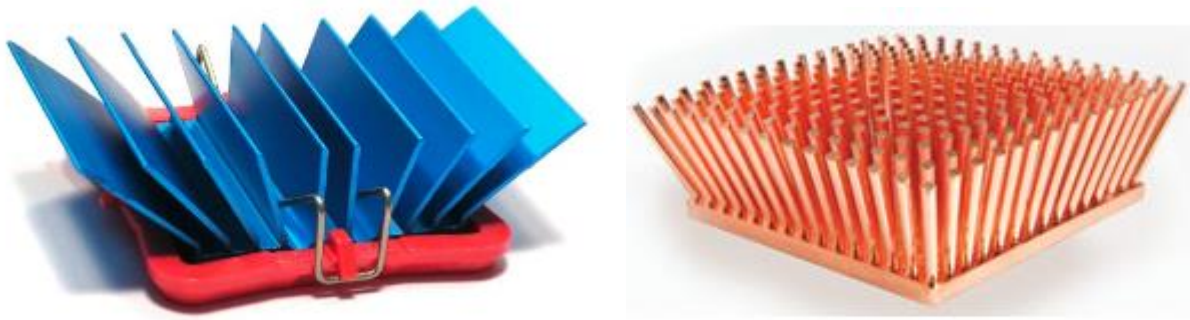
Commonly seen fin types include pin and straight fin heatsinks. Pin fin heatsinks have pins that extend from the base, and the pins are typically cylindrical, elliptical or square. Straight fin heatsinks have fins that extend from the base and run the entire length of the sink. Figure 26 below shows examples of a plate fin heatsink next to a pin fin heatsink [6].



*Figure 26: Example of Straight Fin heatsink (left) and Pin Fin Heatsink (Right) [5,6]*

The fins in straight fin heatsinks can vary in thickness and length, giving different thermal resistances. They are typically made by extrusion which is the most cost effective method for manufacturing a heatsink. For this reason, they are typically less expensive than pin fin heatsinks. However, one might choose a pin fin heatsink over a plate fin heatsink to achieve better thermal performance. Pin fin heatsinks are typically made by metal casting, forging or stamping [6]. Pin fin heatsinks pack as much surface area into a given volume as possible, which enhances the convection heat transfer. Cylindrical and elliptical pins have better surface area per unit volume compared to square and rectangular pins, however their manufacturing process can be more extensive. Overall, pin fin heatsinks have lower thermal resistances than straight fin heatsinks when the fluid flows axially across the pins rather than tangentially across them [7]. They are also typically lighter than straight fin heatsinks. In applications where weight is of serious concern, one might also consider a heatsink that has pins or fins which taper at the ends. Some other heatsinks commonly used are flared and splayed heatsinks.

Flaring the fins on a heatsink creates more convection surface area and decreased the flow resistance, allowing for better cooling. This type of design typically provides better thermal performance than straight fin heatsinks, but can have the same footprint. Splayed pin fin heatsinks do essentially the same thing for pin fin heatsinks. These configurations enable air to enter and exit the arrays of fins or pins more efficiently, providing a substantial improvement in cooling.



*Figure 27: Example of Flared Fin Heatsink (left) and Splayed Fin Heatsink (right) [5,6]*

## 5.6 Selecting from a Manufacturer

When selecting a heatsink from a manufacturer, one must primarily consider two things: the baseplate size and the thermal resistance that is needed for the application. The baseplate size can typically be benchmarked off the footprint size of whatever the heatsink will be used in conjunction with. The thermal resistance of the heatsink dictates whether the heatsink will be sufficient for the application. It is important to note that in forced cooling cases, the thermal resistance is heavily influenced by the air speed. So, when looking at heatsinks to purchase, each heatsink listed will typically have a series of thermal resistances listed with it, corresponding to different air speeds. To achieve that thermal resistance for the heatsink, a fan producing the same velocity must be used in conjunction with the heatsink. For some applications, one must keep in mind that fans do not provide completely uniform cooling, as the fan hub hinders the air flow.

## 5.7 Additional Information

For more detailed, application based explanations, Advanced Thermal Solutions has a series of guidelines which are included [6]:

For natural convection applications:

- 1) If the heat sink is mounted horizontally, it is recommended to use a pin fin heat sink.
- 2) If the heat sink is mounted vertically, both a pin fin heat sink and a straight fin can be used. For a straight fin heat sink, the fin-to-fin spacing has to be at least 6mm to enhance the natural convection flow.

For low velocity force convection ( $\leq 2$  m/s):

- 1) If the flow direction is unknown or unpredictable, choose a pin fin heat sink whose thermal resistance is less sensitive to flow orientation. A cross cut, flared fin heat sink is also a good choice for such applications.
- 2) If the heat sink is ducted, a straight fin heat sink is best choice.
- 3) If the heat sink is in free stream flow, a flared fin heat sink generally provides better performance than a straight fin or pin fin heat sink.

For forced convection ( $\geq 2$  m/s):

- 1) If the flow is un-ducted, make sure that the straight fins are parallel to the approaching flow, otherwise use a pin fin heat sink.
- 2) If the heat sink is ducted, both a pin fin and a straight fin heat sink are good choices. When calculating the thermal resistance of the heat sink theoretically or numerically, caution should be paid to choose right equations for flow mode. The flow may be in transition flow or turbulent flow regimes.

For air impingent applications:

If an air jet is impinged from above to a heat sink, a pin fin heat sink generally generates the lowest thermal resistance. But the fin pitch and fin height must be carefully chosen to maximize thermal performance. The same conclusion is applied to a heat sink with a fan on top. However, a straight fin heat sink with cross cuts can also be used for a fan heat sink if cost is the priority.



Here are other thumbs-up rules for choosing or designing a heat sink:

- 1) In natural convection applications, a heat sink with high aspect ratio fins benefits thermal performance. For high velocity force convection, high aspect ratio fins may not provide extra thermal benefits.
- 2) If the heat sink is required to have the ability to absorb a heat flux spike, it is better to use copper than aluminum.
- 3) For extreme performance and reduced weight, a heat pipe or vapor chamber can be embedded in a heat sink.

## 6. Project Management

### 6.1 Schedule

The schedule for Team 13 had the designing and testing of a thermal management system for a SiC PV converter completed just before April. The first semester that the team was working on this project was dedicated to determining a heatsink configuration, designing and building heat source emulators, as well as completing some of the testing and simulating for the plate fin and pin heatsinks. This all had to be completed before January because the team traveled to Raleigh, North Carolina to participate in Power America's Annual Conference. A more detailed schedule of the Fall 2016 semester is shown in the Gantt chart in Figure 29, Appendix A.

At the beginning of the spring semester, Team 13 attended the Power America Conference where they made a perfect pitch presentation and displayed a poster of the work done during the fall semester. After they returned, the team completed the testing and simulating of the plate fin and pin fin heatsinks. While the testing was happening, the optimization process was also occurring. The last step of this project was the creation of a heatsink selection guide that would walk Team 13's sponsor, Dr. Li, through choosing heatsinks for different applications. A more detailed schedule of the Spring 2017 semester is shown in the Gantt chart in Figure 30, Appendix A.

### 6.2 Resources

#### 6.2.1 CAPS

The Center for Advanced Power Systems allowed Team 13 to conduct all experiments in the CAPS laboratory. Members of Team 13 gained access to resources in this lab that made the entire project possible. CAPS provided the following: soldering equipment, power supplies for the fans and resistors, fans to use for testing, plate fin heatsinks to use for testing, a drill, screwdrivers, copper plates to be used for the emulators, wiring, thermal grease, computers that had more capabilities with COMSOL Software, as well as contacts with several graduate research assistants.

## 6.2.2 College of Engineering Machine Shop

Throughout the project, the college of engineering machine shop provided support and aid to Team 13. The machine shop made cuts on two plate fin heatsinks, and drilled holes in the appropriate locations. The machine shop also machined the connector pieces that were needed for mounting the fan to the pin fin heatsink, and drilled and threaded holes in the appropriate location on the team's pin fin heatsink.

## 6.2.3 College of Engineering Dean's Office

One other resource that was vital for the completion of this project was a scale, used to weigh the parts. Team 13 had trouble finding a scale with a low enough tolerance in various labs, but discovered that the front desk of the Dean's office had one that they could occasionally use. The team is grateful to have had this resource.

## 6.2.4 PowerAmerica

Additionally, PowerAmerica sponsored Team 13 to participate in the perfect pitch presentation at their annual PowerAmerica Meeting in Raleigh, North Carolina. Team 13 learned a great deal on the application of SiC devices at this annual meeting, and was happy to attend.

## 6.3 Procurement

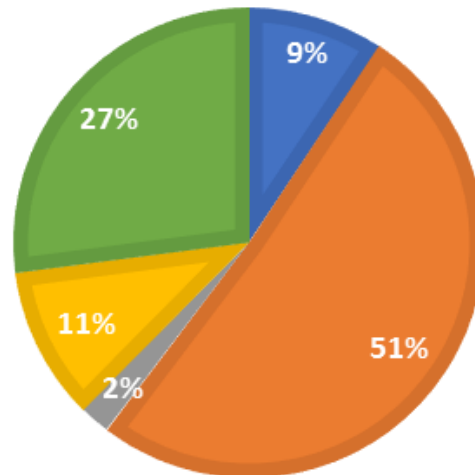
As stated, the team was given a specified budget of \$400, which covered the price of designing an optimized heatsink for a PV converter, a selected manufactured heatsink to use for testing, and the materials needed to build multiple heat source emulators.

The total cost of Designing and Testing Heatsinks for PV Converters was \$291.68 covering the cost of multiple heatsinks, fans, power resistors, screws, and nuts. The budget breakdown is included again in Figure 28.

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## COMPLETE \$400 BUDGET BREAKDOWN

■ Power Resistors    ■ Pin Fin Heatsinks    ■ Steel Screws & Nuts  
■ Fans    ■ Remaining Budget



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*Figure 28: Pie Chart Showing Complete Budget Distribution*

Overall, Team 13 purchased 3 pin fin heatsinks, 2 fans, 10 power resistors, 100 screws, and 100 nuts. Team 13's sponsor, Dr. Li, provided the team plenty of material to use for testing, and the team was also able to save money by using the machine shop at the College of Engineering. The team had \$108.32 remaining from the \$400 allocated.

Table 11 provides the detailed list of the materials purchased during the Fall 2016 semester and the Spring 2017 semester including part numbers, where the materials were ordered from, the quantity and price of the materials, and the amount of money remaining in the budget. Additionally, it should be noted that the budget for the project is not equivalent to the total cost of the thermal cooling system for the PV converter. That breakdown is included in the design for economics section of this report (section 4.4).

Table 11: Budget Information for Team 13

<b>Total Budget:</b>	<b>Order Cost:</b>	<b>Ordered From:</b>	<b>Part Number:</b>	<b>Item Description:</b>	<b>Quantity:</b>
400	37.29	Mouser Electronics	284-HS505RJ	50W 5Ω Power Resistors	10
	77	Cool Innovations	3-454507M	Pin Fin Heatsink	1
	8.4	McMaster-Carr	9027A113	Steel Screws 4-40 Thread, 3/4" Length	100
		McMaster-Carr	90480A005	Steel Hex Nut, 4-40 Thread Size	100
	127	Cool Innovations	3-454507M	Pin Fin Heatsink	2
	41.99	Digi-Key	259-1479-ND	Fans	2
<b>Budget Left:</b>	108.32				

## 6.4 Communication

### 6.4.1 Team Communication

For both Fall and Spring semesters communication efforts between group members of Team 13 were done through a group text message and Google Drive. The group text was effective in relaying concerns, questions, and ideas efficiently. The Google Drive was essential to sharing documents, reports, presentations, and results. This allowed the team members to simultaneously and easily access work without being in the same place.

During the Fall semester, Team 13 had a weekly meeting every Tuesday to discuss in person the progress of the project. The team members would update the rest of team about the portion of the project they were responsible for.

During the Spring semester, Team 13 held two critical meetings to coordinate the PowerAmerica conference, and project updates prior to spring semester vacation. However, members were in constant contact through the group text and the Google Drive.

### 6.4.2 Sponsor Communication

Along with the weekly meetings, Team 13 held biweekly meetings with their sponsor, Dr. Hui Li, and her graduate students, Thierry Kayiranga and Sandro Martin. The meetings provided a way to stay on track with the project and to make sure that any questions the team had would be answered in a timely matter. Dr. Li updated the team on any relevant information needed to complete the testing, simulations, or optimization.

The team also communicated with Dr. Li, Thierry, and Sandro through email. This allowed Team 13 to ask questions that didn't warrant a face to face meeting.

### 6.4.3 Faculty Communication

Team 13 had biweekly staff meetings with the senior design coordinator, Dr. Shih. This gave the team an opportunity to ask any questions regarding reports, presentations, or general project questions.

Along with Dr. Shih, the team communicated with Dr. Juan Ordonez, Dr. Guo, and Dr. Kumar. This correspondence was through email and face to face meetings which gave Team 13 the opportunity to ask more detailed project questions in relation to the faculty member's area of expertise.

## 7. Conclusion

After successful completion of the objective for designing a lightweight alternative to the original CAPS heatsink used in a SiC PV converter, Team 13 compiled the optimized results as well as a compact heatsink selection guide. This guide was given to sponsor and advisor, Dr. Hui Li, for future heatsink applications. It is important to be able to optimize the heatsink in applications where system size and weight are critical, and the heatsink selection guide will help to ease this process. Team 13 was able to successfully meet the requirements of the project and present a detailed design of the final product to sponsor, Dr. Li. If more time allotted, team 13 would have continued with manufacturing the finalized heatsink and would have obtained pricing information to do so.

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# Appendix A: Schedule

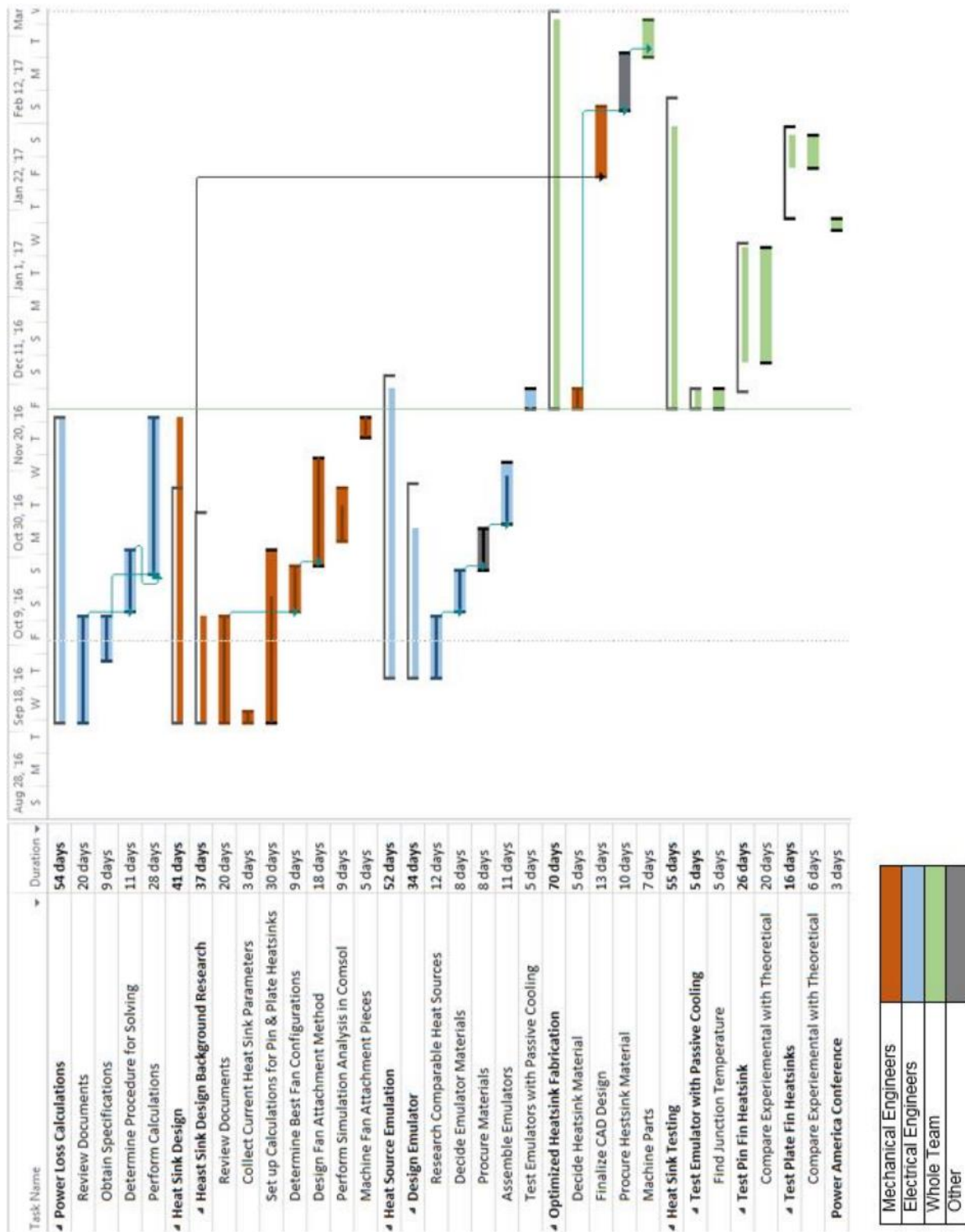


Figure 29: Fall 2016 Gantt Chart

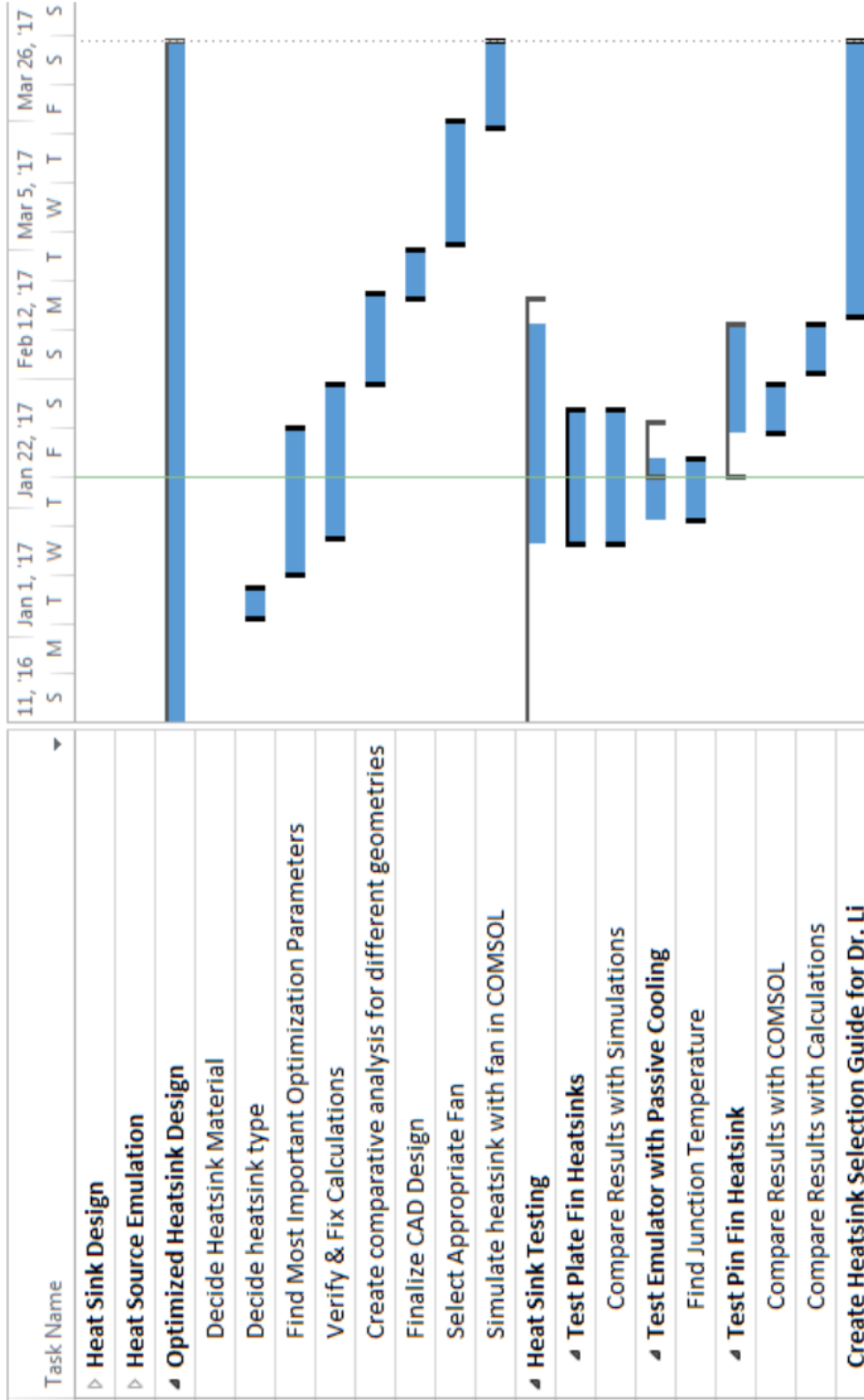


Figure 30: Spring 2017 Gantt Chart.

# Appendix B: House of Quality

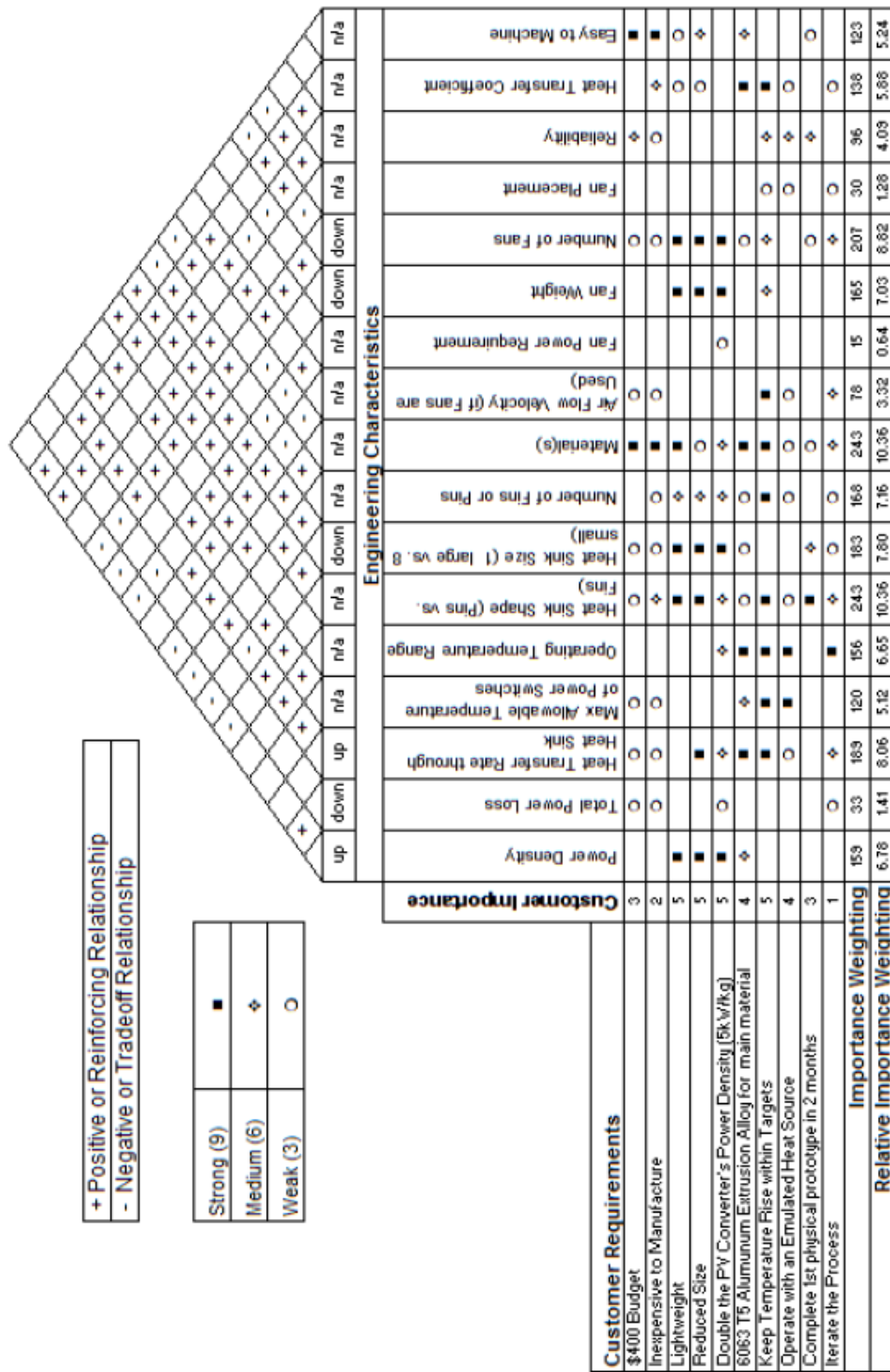


Figure 31: House of Quality

## Appendix C: Plate Fin Calculations

For a rectangular plate fin heat sink with the fan mounted so that the air flows parallel to the fins, a procedure has been developed for determining the thermal resistance of that heatsink. Because heat transfer is highly influenced by air flow, the pressure drop over the fins must first be determined. In order to do so, a variety of parameters must be calculated. These include the Reynolds number, the coefficients of sudden expansion and contraction, the aspect ratio, and the apparent friction factor, and the hydraulic diameter. From there, the pressure drop can be calculated and then plotted vs. the volumetric flow rate of air. The line from this plot is then compared to the air flow versus static pressure curve for a specific fan. At the point where those lines would intersect, the pressure drop is found.

Once the pressure drop is determined, the heat transfer coefficient can be calculated. It is a function of the Reynolds Number, Modified Reynolds Number, Nusselt Number and Prandtl Number. With the heat transfer coefficient known, the thermal resistance of the heatsink can be found. Sample calculations for finding the pressure drop and the thermal resistance are given as follows. It should be noted that these Mathcad files have been used several times with varied geometries and may not produce the exact results described in the report because of this. The procedure to find those results described has remained consistent though.

## PLATE FIN PRESSURE DROP

### Culham & Muzychka

A commercially bought fan with a stated flow rate will not actually be obtained due to the air pressure needed to force the air through the fin channels. The intersection between the commercially available fan curve and the analytical flow resistance curve indicates the operating point of the system. This point represents the actual air volume that will actually pass through the heat sink fin channels.

Len = Length of heatsink channels in flow direction

Wid = Width of Heatsink

Ht = Total Height

Baseplate thickness = 5.0mm

Height of fins = 60.0mm

tf = Thickness of fins

b = Spacing of fins/channel width

Nfin = Number of fins

Hydraulic diameter( $D_h$ ) =  $2 \times$  channel width

$K_e$  = Coefficient of Sudden Expansion

$K_c$  = Coefficient of Sudden Contraction

$\sigma$  = ratio of flow channels to that approaching heatsink

$\dot{V}$  = Volumetric flow rate of fan

$f_{app}$  = Apparent friction factor

$Re_n$  = Reynolds number

$L_{prime}$  = (Length of Heatsink/ Hydraulic Diameter)/Reynold's number

$\lambda$  = Aspect ratio (channel width/height of fin)

$\rho$  = Air density in kg/m<sup>3</sup>

$V_{av}$  = Air velocity in m/s

$\mu$  = dynamic viscosity in Pa\*s

$\Delta P$  = Pressure drop in pascals

$$\rho := 1.255 \frac{\text{kg}}{\text{m}^3} \quad \mu := 1.983 \cdot 10^{-5} \text{ Pa}\cdot\text{s}$$

Variable

$$\text{Base}_t := 4.7 \text{ mm}$$

$$\text{Nfin} := 15$$

$$\text{tf} := 1 \text{ mm}$$

$$\text{Hf} := 15 \text{ mm}$$

$$b := \frac{\text{Wid} - (\text{Nfin} \cdot \text{tf})}{\text{Nfin} - 1} = 7.05 \text{ mm}$$

$$\text{Dh} := 2 \cdot b = 0.014 \text{ m}$$

$$\text{Vav} := \frac{\text{Vdot}}{\text{Nfin} \cdot b \cdot \text{Hf}} = 20.529 \frac{\text{m}}{\text{s}}$$

$$\text{Ren} := \frac{\rho \cdot \text{Vav} \cdot \text{Dh}}{\mu} = 1.832 \times 10^4$$

$$\text{Lp} := \frac{\left(\frac{\text{Len}}{\text{Dh}}\right)}{\text{Ren}} = 4.402 \times 10^{-4}$$

$$f := \frac{\left(24 - 32.527 \cdot \lambda + 46.721 \cdot \lambda^2 - 40.829 \cdot \lambda^3 + 22.954 \cdot \lambda^4 - 6.089 \lambda^5\right)}{\text{Ren}} = 8.611 \times 10^{-4}$$

$$\text{fapp} := \frac{\left[\left(\frac{3.44}{\sqrt{\text{Lp}}}\right)^2 + (f \cdot \text{Ren})^2\right]^{0.5}}{\text{Ren}} = 8.992 \times 10^{-3}$$

$$\Delta P := \left(Kc + 4 \cdot \text{fapp} \cdot \frac{\text{Len}}{\text{Dh}} + Ke\right) \cdot \frac{\rho \cdot \text{Vav}^2}{2} = 120.135 \text{ Pa}$$

$$\text{mmh20} := \Delta P \cdot \frac{0.101971621298 \cdot \text{mm}}{\text{Pa}} = 0.482 \text{ in}$$

Fixed

$$\text{Vdot} := 69 \frac{\text{ft}^3}{\text{min}} = 0.033 \frac{\text{m}^3}{\text{s}}$$

$$\text{Len} := 113.7 \text{ mm} \quad \text{Wid} := 113.7 \text{ mm}$$

$$\text{Ht} := \text{Base}_t + \text{Hf}$$

$$\sigma := 1 - \frac{\text{Nfin} \cdot \text{tf}}{\text{Wid}} = 0.868$$

$$Kc := 0.42 \cdot (1 - \sigma^2) = 0.104$$

$$Ke := (1 - \sigma^2)^2 = 0.061$$

$$\lambda := \frac{b}{\text{Hf}} = 0.47$$

## Total Thermal Resistance of Plate Fin Heatsink

Hf = Height of Fin  
Hb = Height of Base  
Wid = Width of Fin  
Len = Length of Fin Channels  
Nfin = Number of Fins  
Vdot = Volumetric Flow Rate of Fan  
 $\mu$  = Kinematic Viscosity  
b = channel width  
tf = Thickness of Fin  
kf = Thermal conductivity of air  
k = Thermal conductivity of Heat sink material  
Pr = Prandtl Number

Vav = Channel Velocity  
Res = Reynolds Number  
modRen = Modified Reynolds Number  
Nu = Nusselt Number  
h = Convection Coefficient  
Ra = Convection Thermal Resistance  
Rfin = Conductive thermal resistance of fin  
Rd = Conductive Thermal Resistance of Base  
Rtotal = Total Thermal Resistance

$$\text{Len} := 118\text{mm}$$

$$\rho := 1.255 \frac{\text{kg}}{\text{m}^3}$$

$$\text{Pr} := 0.713$$

$$\text{Wid} := 125\text{mm}$$

$$\text{Hb} := 5\text{mm}$$

$$\nu := 15.11 \cdot 10^{-6} \frac{\text{m}^2}{\text{s}}$$

$$\text{Vdot} := 68.0 \frac{\text{ft}^3}{\text{min}}$$

$$\text{Hf} := 60\text{mm}$$

$$\text{Ht} := 69.85\text{mm} + 6.35\text{mm}$$

$$k_f := 0.0257 \frac{\text{W}}{\text{m}\cdot\text{K}}$$

$$\mu := 1.983 \cdot 10^{-5} \text{Pa}\cdot\text{s}$$

$$\text{tf} := 2.42\text{mm}$$

$$k := 209 \frac{\text{W}}{\text{m}\cdot\text{K}}$$

$$\text{Nfin} := 9$$

$$b := 13\text{mm}$$

$$\text{Vav} := \frac{\text{Vdot}}{\text{Nfin} \cdot b \cdot \text{Hf}} = 4.572 \frac{\text{m}}{\text{s}}$$

$$\text{Ren} := b \cdot \frac{\text{Vav}}{\nu} = 3.933 \times 10^3$$

$$\text{modRen} := \frac{\text{Ren} \cdot b}{\text{Len}} = 433.317$$

$$\text{Nu} := \left[ \left( \frac{\text{modRen} \cdot \text{Pr}}{2} \right)^{-3} + \left( 0.664 \sqrt{\text{modRen} \cdot \text{Pr}} \cdot \frac{1}{\sqrt{1 + \frac{3.65}{\sqrt{\text{modRen}}}}} \right)^{-3} \right]^{-\frac{1}{3}}$$

$$h := \text{Nu} \cdot \frac{k_f}{b} = 26.459 \frac{\text{kg}}{\text{K}\cdot\text{s}^3}$$

$$\text{Ra} := \frac{1}{h \cdot \text{Len} \cdot \text{Hf}} = 5.338 \frac{\text{K}\cdot\text{s}^3}{\text{m}^2 \cdot \text{kg}}$$

$$\text{Rfin} := \frac{\text{Hf}}{\text{tf} \cdot \text{Len} \cdot k} = 1.005 \frac{\text{K}\cdot\text{s}^3}{\text{m}^2 \cdot \text{kg}}$$

$$\text{Rd} := \frac{\text{Hb}}{\text{Wid} \cdot \text{Len} \cdot k} = 1.622 \times 10^{-3} \frac{\text{K}\cdot\text{s}^3}{\text{m}^2 \cdot \text{kg}}$$

$$\text{Rtotal} := \text{Rd} + \frac{(\text{Rfin} + \text{Ra})}{2\text{Nfin}} = 0.354 \frac{\text{K}\cdot\text{s}^3}{\text{m}^2 \cdot \text{kg}}$$



# Appendix D: Pin Fin Calculations

For finding the thermal resistance of the pin fin heatsink, the following schematics break down the convective and conductive components.

## Procedure for Calculation

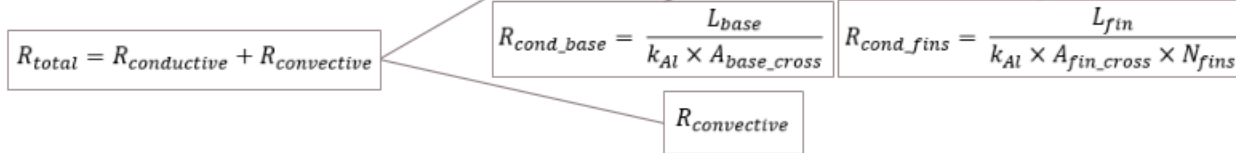
- Thermal Resistance found from COMSOL & Test results

$$\dot{Q} = \frac{\Delta T}{R_{total}} \rightarrow R_{total} = \frac{\Delta T}{\dot{Q}}$$

- Heatsink Mass (comparable to weight)

$$Mass_{heatsink} = \rho_{Al} \times Vol_{heatsink}$$

- Thermal Resistance Computed Analytically



## Convective Thermal Resistance for 1 Fin

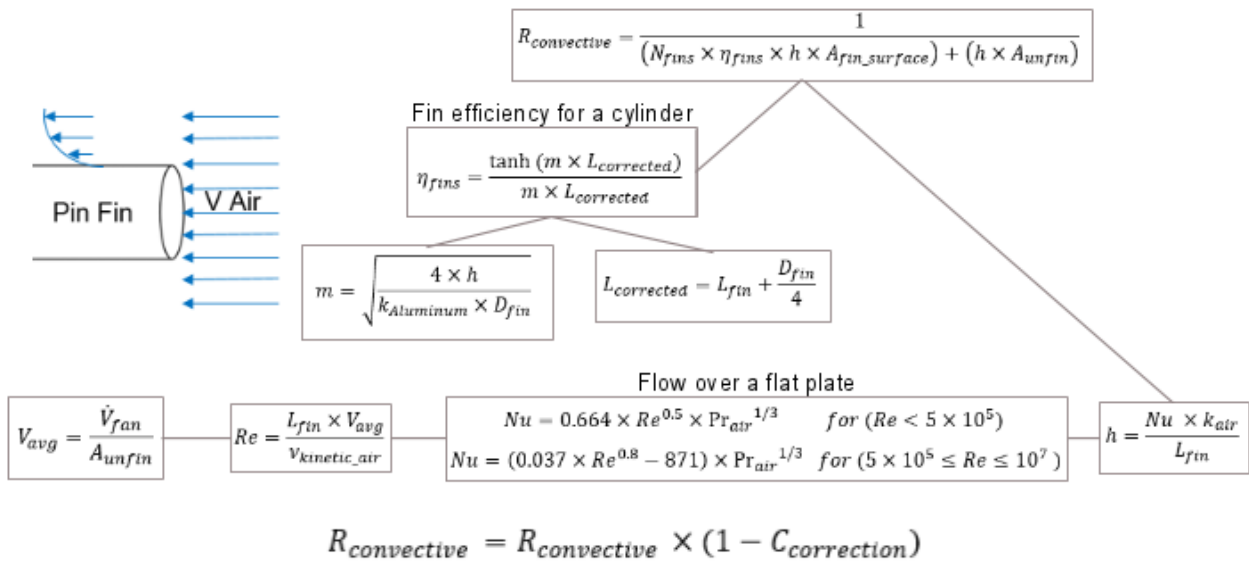


Figure 32: Pin Fin Calculation Schematic

The Matlab code that performed these calculations and used for optimization are included in the following pages.

```

%{
This program implements analysis of a cylindrical pin hin heatsink
analyzed with impingements flow. Nusselt Correlations for flow over a
flat plate are used
%}
clc, clf
clear all

%parameters to vary:
    %pin diameter
    %pin length
    %# of fins & fin spacing
    %fan speed

```

### diameter

```

%D_fin = linspace(0.002,.005,100);    %meters           Fin diameter
D_fin = .005;

```

### number of pins

```

num = 304;           %# of pins

```

### fin length

```

%fin length and corrected fin length for convection through fin tip
%l_fin = 0.0178 - .0047;
l_fin = linspace(.005,.04,100);

```

### flow rate

```

%V_dot = linspace(.01, .07, 100);
V_dot = .050498;    % m^3/s           volumetric flow rate of fan

%Constants
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
Power_Loss = 120;    %Watts
Q = Power_Loss;     %Watts

T_ambient = 22.6;   %deg C           temperature of air in room
T_chip_max = 120;   %deg C           max temperature that chip can operate at safely

```

```

%material constants
density_Al = 2700;    %kg/m^3
k_aluminum = 201;    %W/mK
k_air = 0.0257;      %W/mK    Air conductivity at 20 deg C
Pr = .713;           %Prandtl # for air at 20 deg C
v_k = 15.11 *10^-6;  %kinematic viscosity of air at 25 deg C

%Heatsink sizing properties
height = 0.0178;    %meters          total height of heatsink
l_x_base = .115;    %meters          heatsink base length
l_y_base = .115;    %meters          heatsink base width
t_base = 0.0047;    %meters          Thickness of base
l_fin_corr = l_fin + (D_fin./4); %meters Corrected fin length
r_fin = D_fin./2;   %meters          fin radius

%Area of Different parameters
A_pin = pi*D_fin.*l_fin;
A_pin_corr = pi*D_fin.*l_fin_corr;    %Area of 1 fin using corrected fin length
A_pin_top = pi.* r_fin.^2;            %Area of Fin Top
A_base = l_x_base * l_y_base;         %Base Area
A_unfin = A_base - (num.*A_pin_top);   %Exposed unfinned base area
A_fin_surface = (pi.*D_fin.*l_fin)+(pi.*r_fin.^2); %Surface area of 1 fin
A_surface_tot = num.*A_fin_surface + A_unfin;

Vol_tot = (A_base.*t_base) + (A_pin_top.*l_fin*num); %Heatsink total Volume
Vol_pin = A_pin_top.*l_fin*num;
Vol_current_pin = pi*.0016^2*(.0178-.0047)*304

mass = Vol_tot.*density_Al;

%calculations for Reynolds #
V_av = V_dot./A_unfin;    %average velocity through the pins

Re_critical = 5*10^5;     %critical reynolds number for laminar region
l_critical = Re_critical.*v_k./V_av; % length at which flow becomes turbulent
Re = V_av.*l_fin./v_k;   % actual reynolds number

fprintf('The critical fin length is %.2 meters',l_critical)
if l_fin < l_critical
    fprintf(' so the flow is assumed fully laminar along the fin.\n')
else
    fprintf(' so the flow is partially turbulent along the fin.\n')
end

% Reynolds number calculations
if Re < Re_critical
    Nu = 0.664.*Re.^(.5).*Pr.^(1/3);    %nusselt correlation for laminar flow along s
elseif (Re>=Re_critical) && (Re<=10^7) && (Pr>=0.6) && (Pr<=60)
    Nu = (0.037.*Re.^(.8)-871).*Pr.^(1/3); %nusselt correlation for laminar & turbulent

```

```

else
    error('There is no corresponding Nusselt # Equation')
end

fprintf('The Nusselt Number was determined to be %.3f\n', Nu)

h = Nu.*k_air./l_fin;      %W/K
fprintf('The heat transfer coefficient was determined to be %.3f\n W/K', h)

m = sqrt((4.*h)./(k_aluminum.*D_fin));
%Lm_rad = degtorad(m*l_fin_corr)
Lm = m.*l_fin_corr;
%n = tanh(Lm_rad)/Lm;
n = tanh(Lm)./Lm;
fprintf('The fin efficiency for 1 fin is %.8f\n', 100*n)

% conduction calculations
format long
R_cond_base = t_base / (k_aluminum.*A_base);
T_base_bottom = Q * R_cond_base;
R_cond_pin = l_fin ./ (k_aluminum.*A_pin_top.*num);

err = 0.25;

R_conv = 1./ ((num.*n.*h.*A_fin_surface)+(h.*A_unfin));
R_conv_new = (1-err).*R_conv;

R_contact = 0;
R_tot = R_contact+R_cond_base+R_cond_pin+R_conv_new;

T_chip = Q * R_tot + T_ambient;

effectiveness = Q./ (h.*A_base.*(T_chip-T_ambient));

```

Vol\_current\_pin =

3.2028e-05

# Appendix E: Engineering Drawings

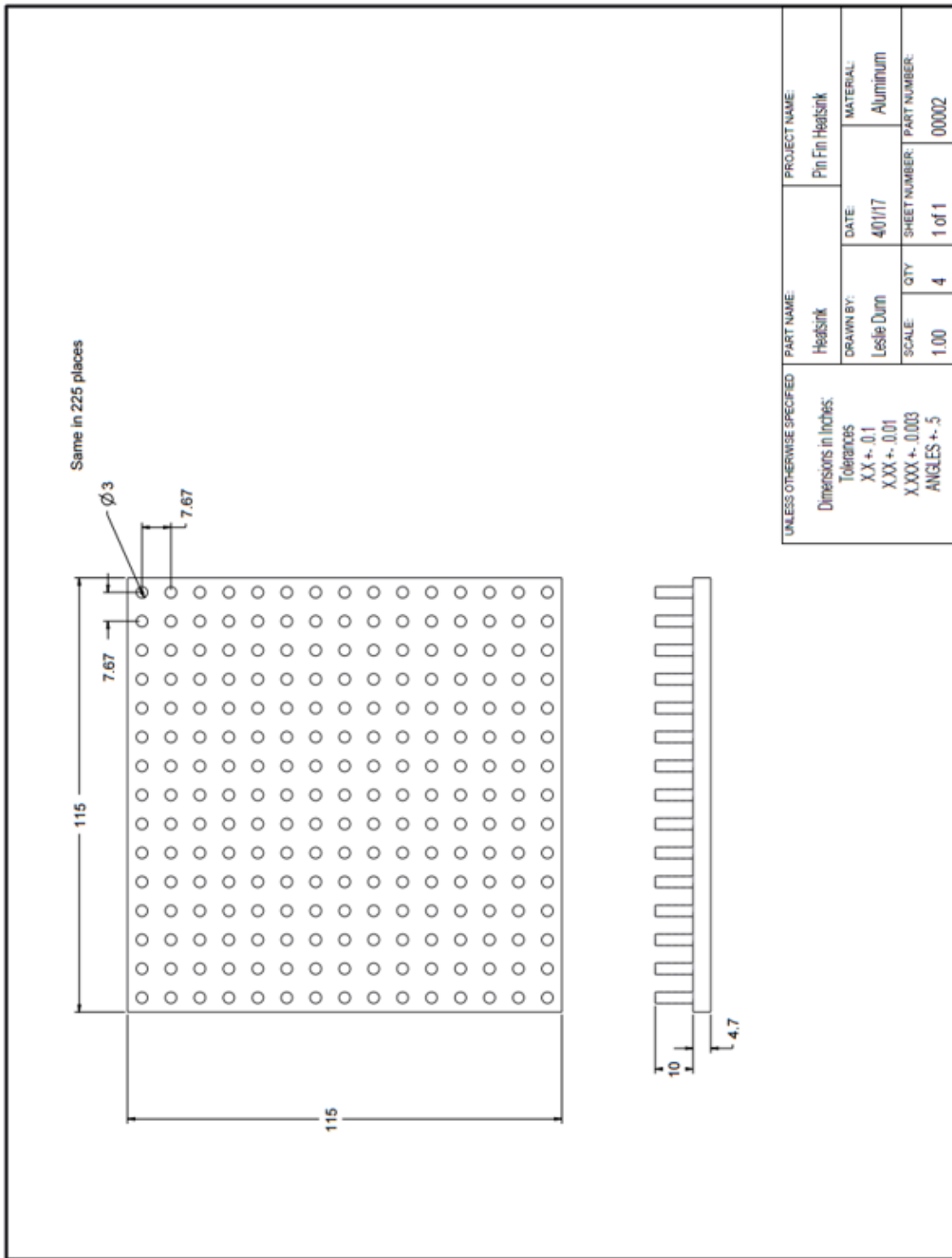


Figure 33: Optimized Heatsink CAD Drawing

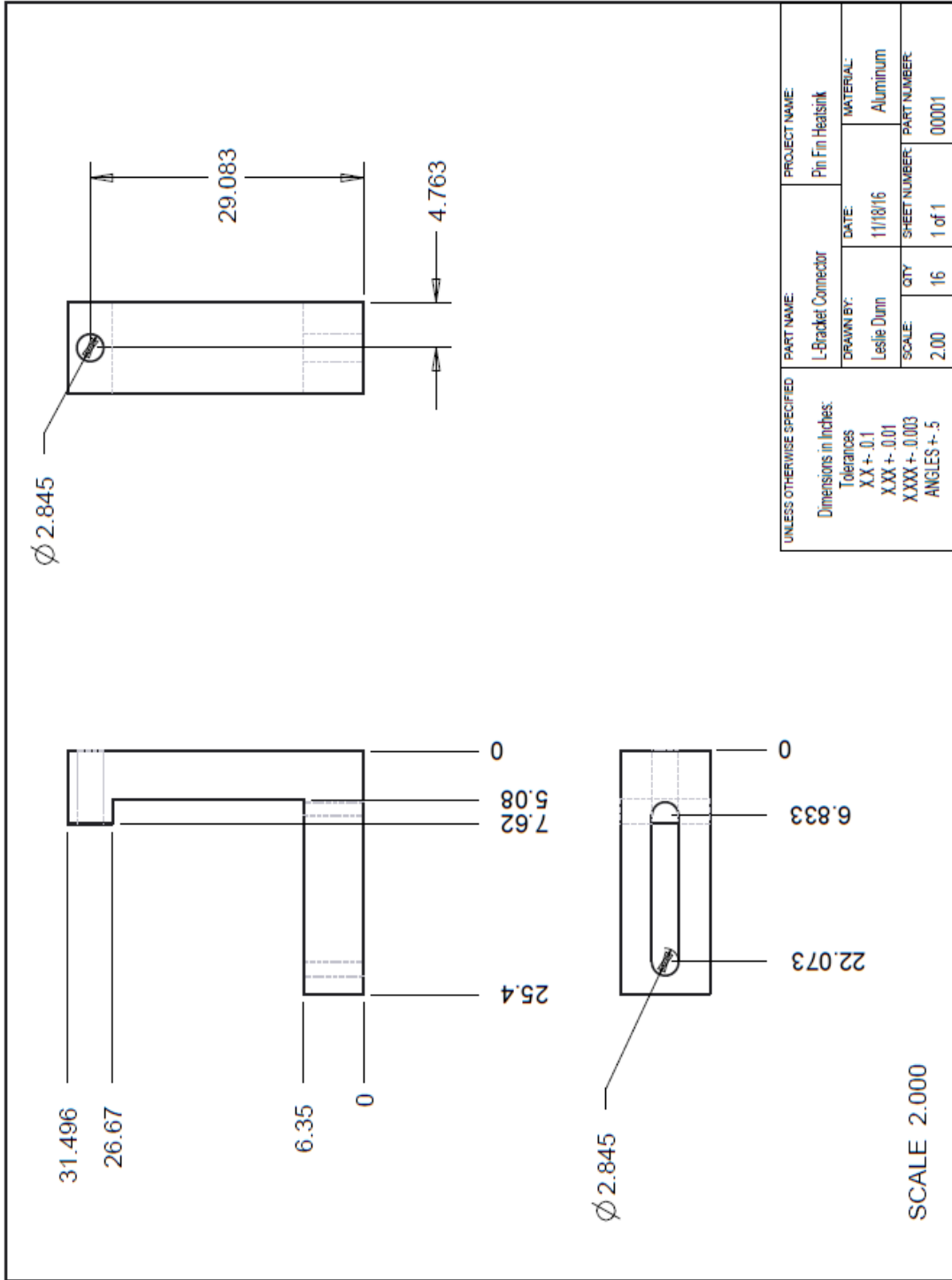



Figure 34: L-Bracket Connector CAD Drawing

## Appendix F: Fan



Product Overview	
<b>Digi-Key Part Number</b>	259-1479-ND
<b>Quantity Available</b>	632 Can ship immediately
<b>Manufacturer</b>	<a href="#">Sunon Fans</a>
<b>Manufacturer Part Number</b>	EEC0251B1-000U-A99
<b>Description</b>	FAN AXIAL 120X25MM 12VDC WIRE
<b>Lead Free Status / RoHS Status</b>	Lead free / RoHS Compliant
<b>Moisture Sensitivity Level (MSL)</b>	1 (Unlimited)
<b>Manufacturer Standard Lead Time</b>	14 Weeks

Documents & Media	
<b>Datasheets</b>	<a href="#">EEC0251B1-0000-A99 Spec Sheet</a>
<b>Product Training Modules</b>	<a href="#">DR MagLev Fan Series</a>
<b>Video File</b>	<a href="#">Sunon DR Maglev</a>
<b>RoHS Information</b>	<a href="#">RoHS Certificate</a>
<b>Featured Product</b>	<a href="#">E Series Fans</a>
<b>Online Catalog</b>	<a href="#">DR (E), DRMagLev® Series</a>

Figure 35: Selected Fan Specs

Product Attributes		Select All <input type="checkbox"/>
Categories	<a href="#">Fans, Thermal Management</a> <a href="#">DC Fans</a>	<input type="radio"/> <input checked="" type="radio"/>
Manufacturer	Sunon Fans	<input type="checkbox"/>
Series	<a href="#">DR MagLev®</a>	<input type="checkbox"/>
Part Status	Active	<input type="checkbox"/>
Voltage - Rated	12VDC	<input type="checkbox"/>
Size / Dimension	Square - 120mm L x 120mm H	<input type="checkbox"/>
Width	25.00mm	<input type="checkbox"/>
Air Flow	108.2 CFM (3.06m³/min)	<input type="checkbox"/>
Static Pressure	0.280 in H2O (69.7 Pa)	<input type="checkbox"/>
Bearing Type	Ball	<input type="checkbox"/>
Fan Type	Tubeaxial	<input type="checkbox"/>
Features	AutoRestart	<input type="checkbox"/>
Noise	44.5 dB(A)	<input type="checkbox"/>
Power (Watts)	5.30W	<input type="checkbox"/>
RPM	3100 RPM	<input type="checkbox"/>
Termination	2 Wire Leads	<input type="checkbox"/>
Ingress Protection	-	<input type="checkbox"/>
Operating Temperature	14 ~ 158°F (-10 ~ 70°C)	<input type="checkbox"/>
Approvals	CE, CUR, TUV, UL	<input type="checkbox"/>
Weight	0.346 lb (156.94g)	<input type="checkbox"/>
Current Rating	0.445A	<input type="checkbox"/>
Voltage Range	6 ~ 13.8VDC	<input type="checkbox"/>
Material - Frame	Polybutylene Terephthalate (PBT)	<input type="checkbox"/>
Material - Blade	Polybutylene Terephthalate (PBT)	<input type="checkbox"/>
Lifetime @ Temp.	100000 Hrs @ 40°C	<input type="checkbox"/>

Figure 36: Selected Fan Specs Cont.



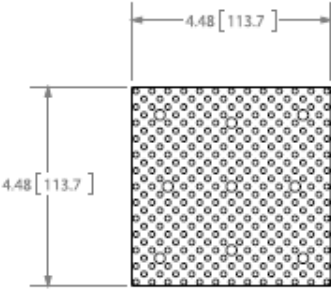
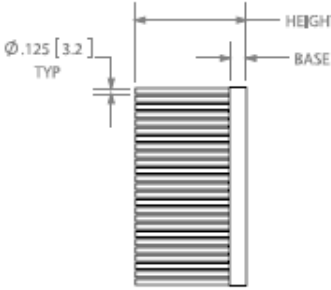
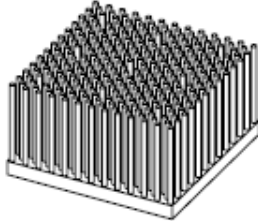
# Appendix G: Tested Pin Fin Heatsink

**COOLINNOVATIONS**  
ADVANCED HEAT SINKS
**FOOTPRINT 4.48" X 4.48"**

**SPARSE CONFIGURATION | ALUMINUM**

**SPECIFICATIONS**

<p><b>Overview</b></p> <ul style="list-style-type: none"> <li>• Provides outstanding cooling power</li> <li>• Omnidirectional</li> <li>• Recommended airspeed range: 0 to 800 LFM (0 to 4 m/s)</li> <li>• RoHS compliant</li> </ul>	<p><b>Technical</b></p> <ul style="list-style-type: none"> <li>• Material: Pure Aluminum</li> <li>• Mfg. process: Cold forging</li> <li>• Plating options: Black/clear anodize, chemical conversion coating</li> <li>• Base finish: Lapped Flatness: Better than 0.001 in/in Surface roughness: 16 RMS</li> </ul>	<p><b>Flexible Parameters</b></p> <ul style="list-style-type: none"> <li>• Footprint (length and width)</li> <li>• Height (pin length &amp; base thickness)</li> <li>• Single or multiple pins can be eliminated</li> <li>• Comprehensive machining (holes, threads, clearances, etc.)</li> </ul>
---	---	---

**THE PIN FIN APPROACH: Round pin formations produce outstanding cooling power**

P/N	Height in(mm)	Base in(mm)	Weight lb(g)	Thermal Resistance in °C/W			
				0(0)*	200(1)	400(2)	600(3)
3-454503M	0.30(7.6)	0.13(3.2)	0.30(137)	4.8	1.37	0.76	0.55
3-454505M	0.50(12.7)	0.19(4.7)	0.48(219)	3.1	0.89	0.49	0.35
3-454507M	0.70(17.8)	0.19(4.7)	0.56(253)	2.2	0.59	0.33	0.24
3-454510M	1.00(25.4)	0.25(6.4)	0.77(349)	1.71	0.43	0.24	0.181
3-454512M	1.20(30.5)	0.25(6.4)	0.84(383)	1.52	0.35	0.21	0.159
3-454515M	1.50(38.1)	0.38(9.5)	1.15(523)	1.40	0.30	0.185	0.144
3-454518M	1.80(45.7)	0.38(9.5)	1.27(574)	1.20	0.24	0.157	0.124
3-454520M	2.00(50.8)	0.38(9.5)	1.34(608)	1.08	0.21	0.142	0.113
3-454522M	2.20(55.9)	0.38(9.5)	1.41(642)	0.97	0.192	0.130	0.104
3-454525M	2.50(63.5)	0.38(9.5)	1.53(693)	0.81	0.167	0.116	0.093

Disclaimer: [www.coolinnovations.com](http://www.coolinnovations.com) \*Air Speed in LFM (m/s)

[www.coolinnovations.com](http://www.coolinnovations.com) • [sales@coolinnovations.com](mailto:sales@coolinnovations.com) • Tel: (905) 760-1992 • Fax: (905) 760-1994

Figure 37: Pin Fin Heatsink Selected for Testing

# Appendix H: Resistors

## HS Aluminium Housed Resistors **ARCOL**

Manufactured in line with the requirements of MIL 18546 and IEC 115, designed for direct heatsink mounting with thermal compound to achieve maximum performance.

- High Power to volume
- Wound to maximise High Pulse Capability
- Values from R005 to 100K
- Custom designs welcome
- RoHS Compliant

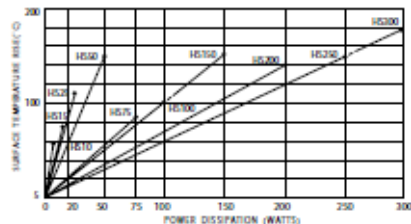


### Characteristics

Tolerance (Code):	Standard $\pm 5\%$ (J) and $\pm 10\%$ (K). Also available $\pm 1\%$ (F), $\pm 2\%$ (G) and $\pm 3\%$ (H)
Tolerance for low $\Omega$ values:	Typically $\geq R05 \pm 5\%$ < $R047 \pm 10\%$
Temperature coefficients:	Typical values < 1K 100ppm Std. > 1K 25ppm Std. For lower TCR's please contact Arcol
Insulation resistance (Dry):	10,000 M $\Omega$ minimum
Power dissipation:	At high ambient temperature dissipation derates linearly to zero at 200°C From R005 to 100K depending on wattage size
Ohmic values:	Specify by adding N before HS Series code, e.g. NHS50
Low inductive (NHS):	Divide standard HS maximum value by 4
NHS ohmic value:	Divide standard HS maximum working voltage by 1.414
NHS working volts:	

### Temp. Rise & Power Dissipation

Surface temperature of resistor related to power dissipation. The resistor is standard heatsink mounted using a proprietary heatsink compound.



### Heat Dissipation

Heat dissipation: Whilst the use of proprietary heat sinks with lower thermal resistances is acceptable, uprating is not recommended. For maximum heat transfer it is recommended that a heat sink compound be applied between the resistor base and heat sink chassis mounting surface. It is essential that the maximum hot spot temperature of 200°C is not exceeded, therefore, the resistor must be mounted on a heat sink of correct thermal resistance for the power being dissipated.

### Ordering Procedure

**Standard Resistor.** To specify standard: Series, Watts, Ohmic Value, Tolerance Code, e.g.: HS25 2R2 J  
**Non Inductive Resistor.** To specify add N, e.g.: NHS100 10R J

ARCOL UK Limited,  
 Threemilestone Ind. Estate,  
 Truro, Cornwall, TR4 9LG, UK.  
 T +44 (0) 1872 277431  
 F +44 (0) 1872 222002  
 E sales@arcolresistors.com

[www.arcolresistors.com](http://www.arcolresistors.com)

The information contained herein does not form part of a contract and is subject to change without notice. Arcol operates a policy of continual product development, therefore, specifications may change.

It is the responsibility of the customer to ensure that the component selected from our range is suitable for the intended application. If in doubt please ask Arcol.

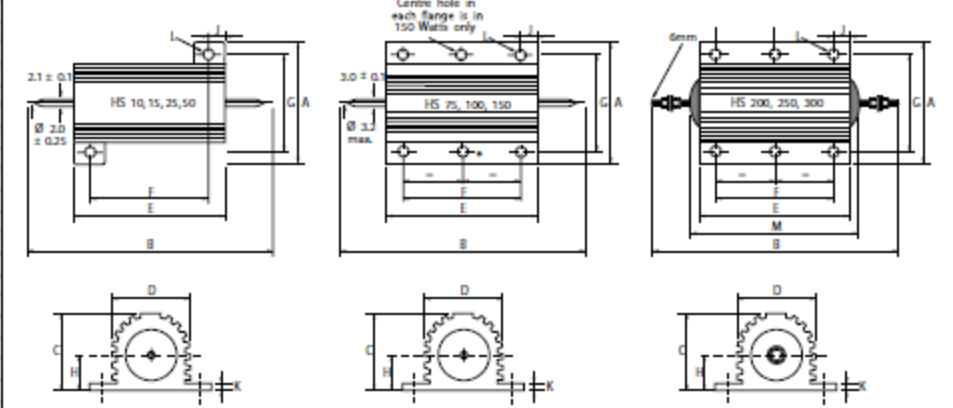
Figure 38: Resistors Used for Testing

# HS Aluminium Housed Resistors **ARCOL**

## Electrical Specifications

Size	Style MIL-R 18546	Power rating on std. heatsink @25°C	Watts with no heatsink @25°C	Resistance range	Limiting element voltage	Voltage proof AC Peak	Voltage proof AC rms.	Approx weight gms	Typical surface rise HS mounted	Standard heatsink	
										cm <sup>2</sup>	Thickness mm
HS10	RE 60	10	5	R005-10K	160	1400	1000	4	5.8	415	1
HS15	RE 65	15	7	R005-10K	265	1400	1000	7	5.1	415	1
HS25	RE 70	25	9	R005-36K	550	3500	2500	14	4.2	535	1
HS50	RE 75	50	14	R01-86K	1250	3500	2500	32	3.0	535	1
HS75		75	24	R01-50K	1400	6363	4500	85	1.1	995	3
HS100		100	30	R01-70K	1900	6363	4500	115	1.0	995	3
HS150		150	45	R01-100K	2500	6363	4500	175	1.0	995	3
HS200		200	50	R01-50K	1900	7070	5000	475	0.7	3750	3
HS250		250	55	R01-50K	2200	7070	5000	600	0.6	4765	3
HS300		300	60	R01-68K	2500	7070	5000	700	0.6	5790	3

## HS10-HS300 Standard



## Dimensions (mm)

Size	A Max	B Max	C Max	D Max	E Max	F±0.3	G±0.3	H Max	J Max	K Max	L ±0.25*	M Max
HS10	16.5	30.0	8.8	8.5	15.9	11.3	12.4	4.5	2.4	1.8	2.4	
HS15	21.0	36.5	11.0	11.2	19.9	14.3	15.9	5.5	2.8	1.8	2.4	
HS25	29.0	51.0	14.6	14.0	27.3	18.3	19.8	7.3	4.7	2.6	3.2	
HS50	29.7	72.5	14.8	14.2	49.1	39.7	21.4	8.5	5.2	2.6	3.2	
HS75	47.5	72.0	24.1	27.3	48.7	29.0	37.0	11.8	10.4	3.7	4.4	
HS100	47.5	88.0	24.1	27.3	65.2	35.0	37.0	11.8	15.4	3.7	4.4	
HS150	47.5	121.0	24.1	27.3	97.7	58.0	37.0	11.8	20.4	3.7	4.4	
HS200	72.5	145.7	41.8	45.5	89.7	70.0	57.2	20.5	10.4	5.5	5.1	103.4
HS250	72.5	167.0	41.8	45.5	109.7	89.0	57.2	20.5	10.4	5.5	5.1	122.4
HS300	72.5	184.4	41.8	45.5	127.7	104.0	59.0	20.5	12.4	5.5	6.6	141.4

\* HS200-HS300 Watts is ± 0.45

Figure 39: Resistors Used for Testing Cont.

## Appendix I: Parts List

*Table 12: Complete Parts List*

Part Name	QTY	Source
Pin Fin Heatsink Body	4	Custom Manufacturer
12 V DC Fan	4	digkey.com
L-Bracket Connector	16	Machined in shop
1/2" 4-40 screws	48	Home Depot
4-40 Nuts	16	Home Depot

## Biography

Melanie Gonzalez is a fifth-year senior engineering student studying electrical engineering. Her interests include control systems and embedded microprocessor design. She previously worked as an undergraduate research assistant at the Center for Advanced Power Systems. After graduation, she plans to obtain a Ph.D. in Electrical Engineering.

Tianna Lentino is a fourth-year senior at FSU working towards a B.S in electrical engineering. Her interests include systems engineering and power systems engineering. She has spent the past two summers working as an intern with Northrop Grumman. After she graduates, Tianna will start her career at Northrop Grumman.

James Hutchinson is a fifth-year senior attending Florida State University, studying mechanical engineering. He is a member of the National Society of Black Engineers, Golden Key Honor Society and the Foundations Chair of the Society of Engineering Entrepreneurs. His interests include CAD design, thermal fluids, and robotics. After graduation, he plans on working in Baltimore as a Systems Engineer for Northrop Grumman.

Colleen Kidder is pursuing a bachelor's degree in mechanical engineering and will graduate in May 2017. She is most interested in mechanics & materials and thermal fluids. Through internship experience at Crane Aerospace & Electronics, Colleen has developed engineering skills in a business environment. Upon graduation, she plans to take the Fundamentals of Engineering exam and begin her career at Northrop Grumman.

Leslie Dunn is a senior at Florida State University pursuing her B.S. in mechanical engineering. She worked part-time at the FSU Central Utilities Plant for 2 years where she conducted engineering work for a back-up fuel conversion. She's been an active member of the FAMU/FSU Society of Women Engineers for 3 years and is the current Vice President. After graduation, she will work as a project engineer for Pratt & Whitney.